FOURTH EDITION

Pressure Vessel Design Manual

Dennis Moss and Michael Basic

BH

Pressure Vessel Design Manual

This page intentionally left blank

PRESSURE VESSEL DESIGN MANUAL

Fourth Edition

Dennis R. Moss

Michael Basic

ELSEVIER

AMSTERDAM • BOSTON • HEIDELBERG • LONDON • NEW YORK • OXFORD
PARIS • SAN DIEGO • SAN FRANCISCO • SINGAPORE • SYDNEY • TOKYO

Butterworth-Heinemann is an imprint of Elsevier

B
H

Butterworth-Heinemann is an imprint of Elsevier
Linacre House, Jordan Hill, Oxford OX2 8DP, UK
The Boulevard, Langford Lane, Kidlington, Oxford OX5 1GB, UK
225 Wyman Street, Waltham, MA 02451, USA

Fourth edition 2013

British Library Cataloguing in Publication Data
A catalogue record for this book is available from the British Library

Library of Congress Cataloging-in-Publication Data
A catalog record for this book is availabe from the Library of Congress

ISBN: 978-0-12-387000-1

For information on all Elsevier publications
visit our web site at books.elsevier.com

Printed and bound in Great Britain

13 14 15 16 17   10 9 8 7 6 5 4 3 2 1

Contents

This page intentionally left blank

Preface to the 4th Edition

When I started the Pressure Vessel Design Manual 35 years ago, I had no idea where it would lead. The first edition alone took 10 years to publish. It began when I first started working for a small vessel shop in Los Angeles in 1972. I could not believe how little information was available to engineers and designers in our industry at that time. I began collecting and researching everything I could get my hands on. As I collected more and more, I began writing procedures around various topics. After a while I had a pretty substantial collection and someone suggested that it might make a good book. However I was constantly revising them and didn't think any of them were complete enough to publish. After a while I began trying to perfect them so that they could be published. This is the point at which the effort changed from a hobby to a vocation. My goal was to provide as complete a collection of equations, data and procedures for the design of pressure vessels that I could assemble. I never thought of myself as an author in this regard… but only the editor. I was not developing equations or methods, but only collecting and collating them. The presentation of the materials was then, and still is, the focus of my efforts. As stated all along "The author makes no claim to originality, other than that of format."

My target audience was always the person in the shop who was ultimately responsible for the designs they manufactured. I have seen all my goals for the PVDM exceeded in every way possible. Through my work with Fluor, I have had the opportunity to travel to 40 countries and have visited 60 vessel shops. In the past 10 years, I have not visited a shop that was not using the PVDM. This has been my reward. This book is now, and always has been, dedicated to the end user. Thank you.

The PVDM is a "designers" manual foremost, and not an engineering textbook. The procedures are streamlined to provide a weight, size or thickness. For the most part, wherever possible, it avoids the derivation of equations or the theoretical background. I have always sought out the simplest and most direct solutions.

Today, computers have changed the way we do our work. For the most part, designers and engineers rely on computers to perform their tasks. Computers are an integral part of the work process. I have been pleased that many of the procedures and techniques of the PVDM have been used by the software makers in the development of their software. So the question is raised…do we really need a book on how to perform manual calculations? After all, aren't computers capable of, and aren't they doing most of the work? I would offer the following points as a reply to this question;

1. As a method to develop an initial design.
2. As a basis for computer programs.
3. As a means to check and/or verify computer programs.
4. As a means to provide background and traceability.
5. As a historical basis. How was the equipment designed in the past?
6. For a comparison between designs.

This book does not always provide the most optimum solution. For more sophisticated and complex designs, the work process that has evolved in our company is to first develop a design and then hand it over to our FEA or stress specialists to evaluate and refine. Often, we have to be out for quote before a complete analysis can be done. This requires that we develop a design that will work, can be built, and do it quickly. However, it may not be the most economical design.

This edition is completely new in a number of ways. First, we have added a new chapter on design of high pressure vessels. Bits and pieces of this material have been scattered around for many years, but never comprehensively collected into a single source. I can state that this is the most complete collection of this material on this subject ever published.

Second, the sheer volume and scope of this edition makes this the most comprehensive manual on the design of pressure vessels ever published.

Third and most important, we have added a co-author to take over the PVDM. Please allow me to introduce Mr. Michael M. Basic. This is the most important addition to the PVDM. It is my sincere desire that this effort continue and my involvement is irrelevant to this goal. Therefore it is essential that the torch be passed to the next generation.

If I have an interest in seeing this book continuing, then it must be done under the direction of a new, younger and very talented person.

Finally, I would like to offer my warmest, heartfelt thanks to all of you that have made comments, contributions, sent me literature, or encouraged me over the past 35 years. It is immensely rewarding to have watched the book evolve over the years. This book would not have been possible without you!

Dennis R. Moss

1
General Topics

Design Philosophy

In general, pressure vessels designed in accordance with the ASME Code, Section VIII, Division 1, are designed by rules and do not require a detailed evaluation of all stresses. It is recognized that high localized and secondary bending stresses may exist but are allowed for by use of a higher design margin and design rules for details. It is required, however, that all loadings (the forces applied to a vessel or its structural attachments) must be considered. (See Reference 1, Para. UG-22).

While Section VIII, Division 1 provides formulas for thickness and stress of basic components, it is up to the designer to select appropriate analytical procedures for analyzing other components and to combine the calculated stresses in a manner appropriate with the intended operation of the equipment for an economical and safe design. For the supporting structures, the designer must also abide to the load combinations determined by the applicable building code.

Section VIII, Division 1 establishes allowable stresses by stating in Para. UG-23(c) that the maximum general primary membrane stress must be less than allowable stresses outlined in material sections. Further, it states that the maximum primary membrane stress *plus* primary bending stress may not exceed 1.5 times the allowable stress of the material sections. In other sections of the Code, including both design-by-rules and design-by-analysis methods in Section VIII, Division 2, higher stress levels are permitted if appropriate analyses are made. These higher allowable stresses clearly indicate that different stress levels for different stress categories are acceptable.

Section VIII, Division 2 also has a section outlining 'design-by-rules' requirements (Part 4). Though also classified as design-by-rules, the design margin is lower than in Division 1 and as a result will require more analysis. Part 4 clearly indicates that if rules are not provided for a specific detail, geometry, or loading, then an analysis in accordance with Part 5, or 'design-by-analysis', shall be performed. Most vessels designed to Part 4 will use both the rules in Part 4 as well as specific procedures in Part 5. Both Part 4 and Part 5 contain load combinations for the pressure envelope. The load combinations in Part 4 are based on the allowable stress design (ASD) load combinations from ASCE 7, with non-applicable loads removed.

It is general practice when doing a more detailed stress analysis to apply higher allowable stresses. In effect, the detailed evaluation of stresses permits substituting knowledge of localized stresses and the use of higher allowable stresses in place of the larger design margin used by the Code. This larger design margin really reflected lack of knowledge about actual stresses.

A calculated value of stress means little until it is associated with its location and distribution in the vessel and with the type of loading by which it was produced. Different types of stresses have different degrees of significance.

The designer must be familiar with the various types of loadings and their stresses in order to accurately understand the results of the analysis. The designer must also consider the stress categories to determine the allowable stress limits.

The following sections will provide the fundamental knowledge for determining and understanding the results of an analysis. The topics covered in Chapter 1 form the basis by which the rest of the book is to be used. A section on special problems and considerations is included to alert the designer to more complex problems that exist.

Stress Analysis

Stress analysis is the determination of the relationship between external forces applied to a vessel and the corresponding stresses. The emphasis of this book is not how to do stress analysis in particular, but rather how to analyze vessels and their component parts in an effort to arrive at an economical and safe design—the difference being that we analyze stresses where necessary to determine thickness of material and sizes of members. We are not so concerned with building mathematical models as with providing a step-by-step approach to the design of ASME Code vessels. It is not necessary to find every stress but rather to know the governing stresses and how they relate to the vessel or its respective parts, attachments, and supports.

The starting place for stress analysis is to determine all the design conditions for a given problem and then determine all the related external forces. We must then relate these external forces to the vessel parts which must resist them to find the corresponding stresses. By isolating the causes (loadings), the effects (stresses) can be more accurately determined.

The designer must also be keenly aware of the types of loads and how they relate to the vessel as a whole. Are the effects long or short term? Do they apply to a localized portion of the vessel or are they uniform throughout?

How these stresses are interpreted and combined, what significance they have to the overall safety of the vessel, and what allowable stresses are applied will be determined by three things:

1. The strength/failure theory utilized.
2. The types and categories of loadings.
3. The hazard the stress represents to the vessel.

Membrane Stress Analysis

Pressure vessels commonly have the form of spheres, cylinders, cones, ellipsoids, tori, or composites of these. When the thickness is small in comparison with other dimensions ($R_m/t \geq 10$), vessels are referred to as membranes and the associated stresses resulting from the contained pressure are called membrane stresses. These membrane stresses are average tension or compression stresses. They are assumed to be uniform across the vessel wall and act tangentially to its surface. The membrane or wall is assumed to offer no resistance to bending. When the wall offers resistance to bending, bending stresses occur in addition to membrane stresses.

In a vessel of complicated shape subjected to internal pressure, the simple membrane-stress concepts do not suffice to give an adequate idea of the true stress situation. The types of heads closing the vessel, effects of supports, variations in thickness and cross section, nozzles, external attachments, and overall bending due to weight, wind, and seismic activity all cause varying stress distributions in the vessel. Deviations from a true membrane shape set up bending in the vessel wall and cause the direct loading to vary from point to point. The direct loading is diverted from the more flexible to the more rigid portions of the vessel.

In any pressure vessel subjected to internal or external pressure, stresses are set up in the shell wall. The state of stress is triaxial and the three main defining stresses are:

σ_x = longitudinal/meridional stress

σ_ϕ = circumferential/latitudinal stress

σ_r = radial stress

In addition, there may be bending and shear stresses. The radial stress is a direct stress, which is a result of the pressure acting directly on the wall, and causes a compressive stress equal to the pressure on the surface on which it acts. In thin-walled vessels this stress is so small compared to the circumferential and longitudinal stresses that it is generally ignored. Thus we assume for purposes of analysis that the state of stress is biaxial. This greatly simplifies the method of combining stresses in comparison to triaxial stress states. For thick walled vessels ($R_m/t < 10$), the radial stress cannot be ignored and formulas are quite different from those used in finding membrane stresses in thin shells.

Since ASME Code, Section VIII, Division 1 is basically for designing by rules; a higher design margin and specific rules are used to allow for the high localized and secondary bending stresses at safe levels consistent with experience. This higher design margin can impose a penalty on design but requires much less analysis. The design techniques outlined in this book are a compromise between finding all stresses and utilizing minimum code formulas. This additional knowledge of stresses warrants the use of higher allowable stresses in some cases, while meeting the requirements that all loadings be considered.

In conclusion, "membrane stress analysis" is not completely accurate but allows certain simplifying assumptions to be made while maintaining a fair degree of accuracy. The main simplifying assumptions are that the stress is biaxial and that the stresses are uniform across the shell wall. For thin-walled vessels these assumptions have proven themselves to be reliable. No vessel meets the criteria of being a true membrane, but we can use this tool with a reasonable degree of accuracy.

Stress/Failure Theories

As stated previously, stresses are meaningless until compared to some stress/failure theory. The significance of a given stress must be related to its location in the vessel and the failure mode being evaluated. Historically, various "theories" have been derived to combine and measure stresses against the potential failure mode. A number of stress theories, also called "yield criteria", are available for describing the effects of multi-axial stresses. For purposes of this book, as these failure theories apply to pressure vessels, three theories will be discussed. They are the "maximum principal stress theory", the "maximum shear stress theory", and the "distortion energy theory".

Maximum Principal Stress Theory

This theory is the oldest, most widely used and simplest to apply. Both ASME Code, Section VIII, Division 1, and Section I use the maximum principal stress theory as a basis for design. This theory simply asserts that yielding occurs when the largest principal stress equals the yield strength. Stresses in the other directions are disregarded for this criteria. Only the maximum principal stress must be determined to apply this criterion. This theory is used for biaxial states of stress assumed in a thin-walled pressure vessel. As will be shown later it is unconservative in some instances of biaxial stress states. While the maximum principal stress theory more accurately predicts failure in brittle materials, it is not always accurate for ductile materials. Ductile materials often fail along lines 45° to the applied force by shearing, long before the tensile or compressive stresses are a maximum.

Where $\sigma 1 > \sigma 2$, this theory states that yielding will occur when

$$\sigma_1 = F_y$$

This theory is illustrated graphically for the four states of biaxial stress shown in Figure 1-1.

It can be seen that uniaxial tension or compression lies on the two axes. Inside the box (outer boundaries) is the elastic range of the material. Yielding is predicted for stress combinations by the outer line.

Maximum Shear Stress Theory

This theory asserts that yielding occurs when the largest difference of shear stress equals the shear yield strength. According to this theory, yielding will start at a point when the maximum shear stress at that point reaches one-half of the uniaxial yield strength, F_y. Thus for a biaxial state of stress where $\sigma_1 > \sigma_2$, the maximum shear stress will be $(\sigma_1 - \sigma_2)/2$.

This theory states that yielding will occur when

$$\frac{\sigma_1 - \sigma_2}{2} = \frac{F_y}{2}$$

This theory closely approximates experimental results in ductile materials and is also easy to use. This theory

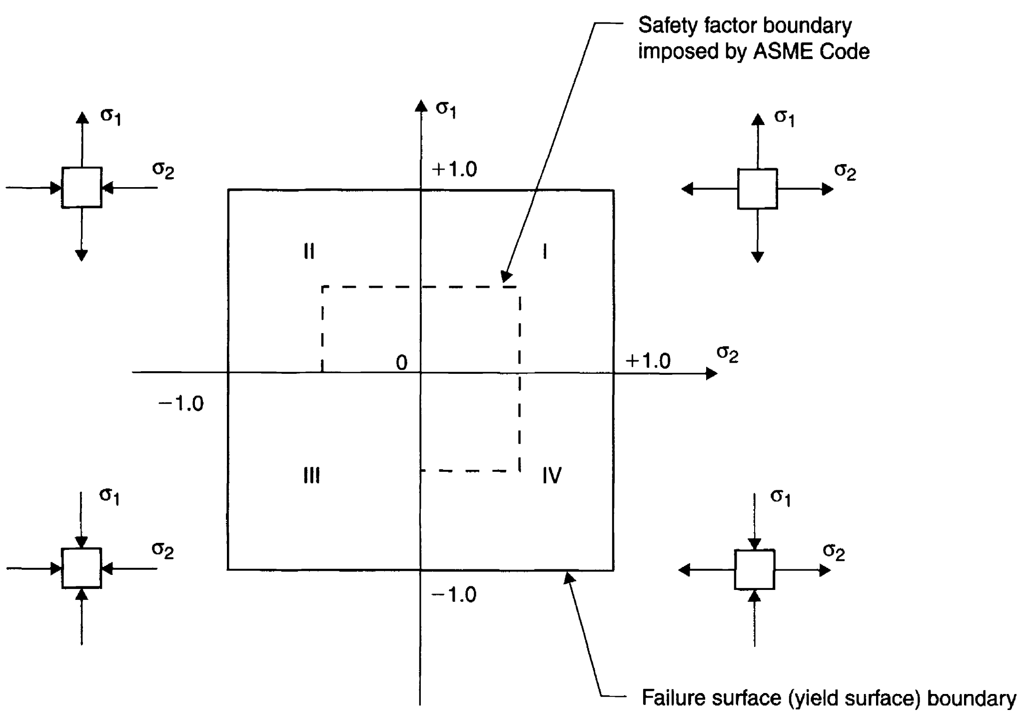


Figure 1-1. Graph of maximum stress theory. Quadrant I: biaxial tension; Quadrant II: tension; Quadrant III: biaxial compression; Quadrant IV: compression.

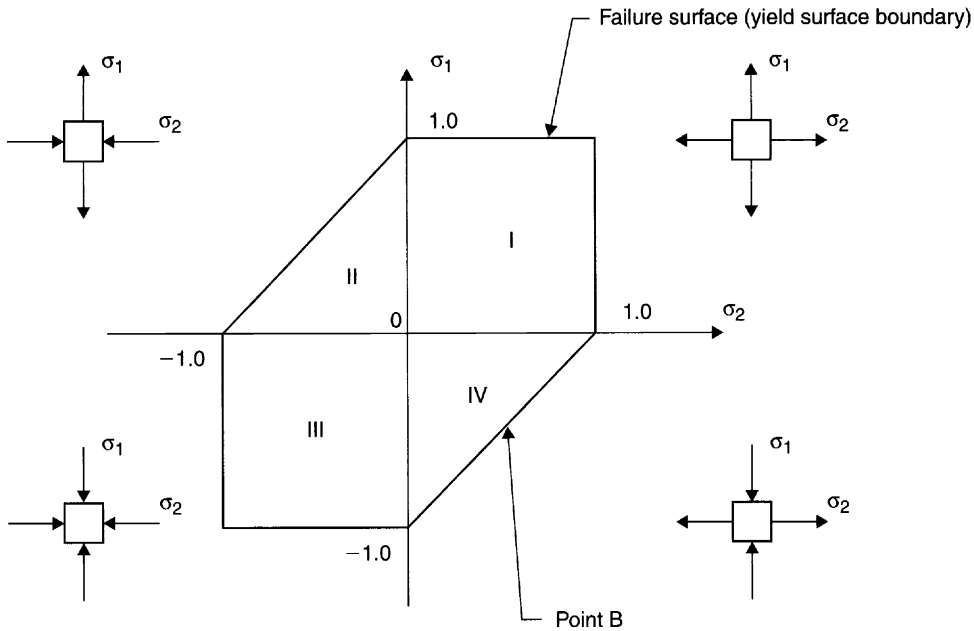


Figure 1-2. Graph of maximum shear stress theory.

also applies to triaxial states of stress. In a triaxial stress state, this theory predicts that yielding will occur whenever one-half the algebraic difference between the maximum and minimum stress is equal to one-half the yield stress. Where $\sigma_1 > \sigma_2 > \sigma_3$, the maximum shear stress is $(\sigma_1 - \sigma_3)/2$.

Yielding will begin when

$$\frac{\sigma_1 - \sigma_3}{2} = \frac{F_y}{2}$$

This theory is illustrated graphically for the four states of biaxial stress in Figure 1-2.

Both the pre-2007 ASME Section VIII, Division 2 and Part 4 of the new Section VIII, Division 2 utilize the maximum shear stress criterion for determining the primary thicknesses of a shell under internal pressure.

Distortion Energy Theory

This theory asserts that the total strain energy is composed of two parts; the strain energy required for hydrostatic strain and the strain energy required for distortion. In this theory, it is assumed that yielding will begin when the distortion component is equal to the uniaxial yield strength, F_y. Where $\sigma_1 > \sigma_2 > \sigma_3$, yielding will occur when

$$\frac{1}{2}\left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\right] = {F_y}^2$$

This theory is illustrated graphically for the four states of biaxial stress shown in Figure 1-3. This theory correlates even better with ductile test specimens than the maximum shear stress theory.

The new ASME Section VIII, Division 2, Part 5 utilizes the distortion energy theory to establish the equivalent stress in an elastic analysis where in the pre-2007 edition this was done with the maximum shear stress theory.

Comparison of the Three Theories

Figure 1-4 is an overlay of Figures 1-1, 1-2, and 1-3 and will illustrate the major differences between the three theories. For the case of biaxial stress state, all three theories are in agreement where their bounded areas graphically overlap. The bounded area by each theory indicates the elastic range of which there is no yielding, however, it is important to note that in quadrants II and IV that the maximum principal stress theory provides unconservative results. For example, consider point B at the midpoint of the line in Figure 1-2. It shows $\sigma_2 = (-)\sigma_1$; therefore the shear stress is equal to $(\sigma_2 - (-\sigma_1))/2$, which equals $(\sigma_2 + \sigma_1)/2$ or one-half the stress which would

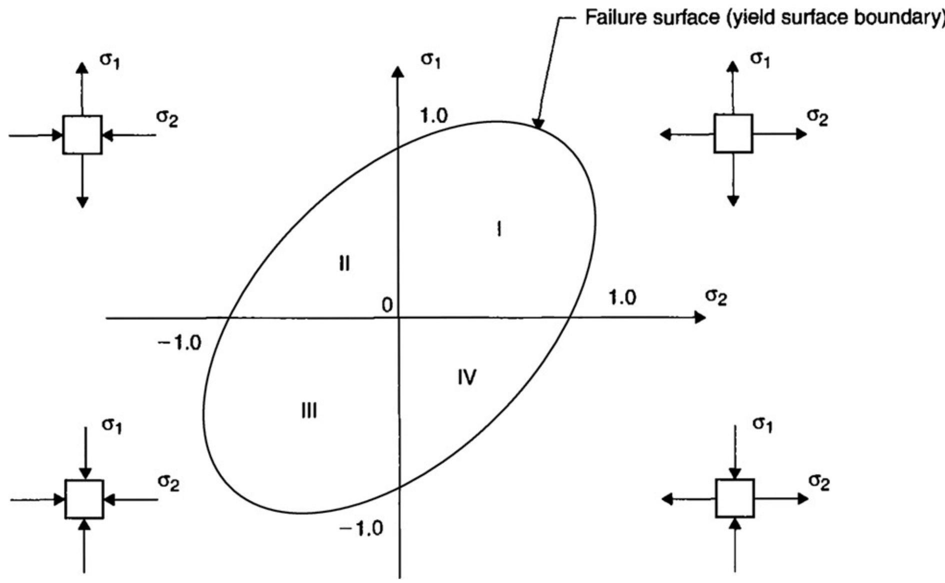


Figure 1-3. Graph of distortion energy theory.

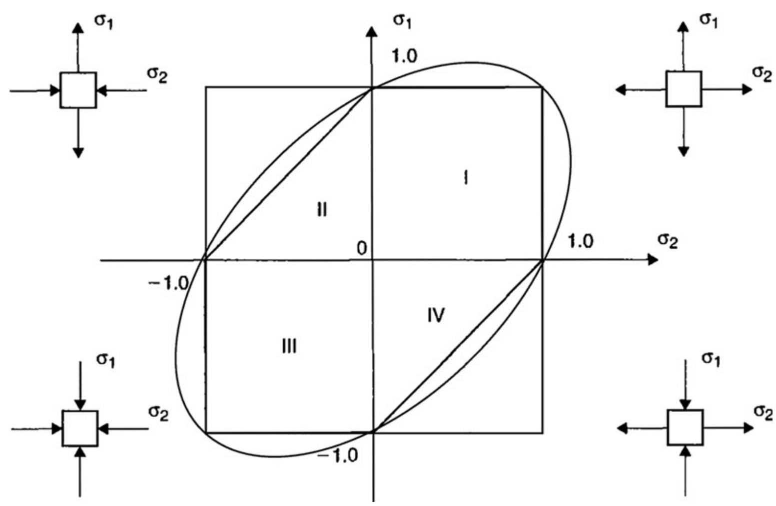


Figure 1-4. Comparison of the three theories.

cause yielding as predicted by the maximum principal stress theory.

For simple analysis upon which the thickness formulas for ASME Code, Section I or Section VIII, Division 1, are based, it makes little difference whether the maximum principal stress theory or maximum shear stress theory is used. For example, according to the maximum principal stress theory, for a cylinder only under internal pressure the controlling stress governing the thickness of a cylinder is σ_ϕ, the circumferential stress, since it is the largest of the three principal stresses. According to the maximum shear stress theory, the controlling stress would be one-half the algebraic difference between the maximum and minimum stress (this was defined as the stress intensity in the pre-2007 Section VIII, Division 2), and according to the distortion energy theory the equivalent stress is as shown:

- The maximum principal stress is the circumferential stress, σ_ϕ
  $\sigma_\phi = \sigma_1 = PR/t$
- The middle principal stress is the longitudinal stress, σ_r
  $\sigma_x = \sigma_2 = PR/2t$

- The minimum principal stress is the radial stress, σ_r
  $\sigma_r = \sigma_3 = (-)P$

Therefore, the stress used in the maximum principal stress theory is

$$\sigma_\phi = \frac{PR}{t},$$

the calculated stress used in the maximum shear stress theory (stress intensity) is

$$\sigma_\phi - \sigma_r = \frac{PR}{t} + P,$$

and for information, the equivalent stress used in the distortion energy theory is

$$\sigma_e = \frac{1}{\sqrt{2}}\left[(\sigma_\phi - \sigma_x)^2 + (\sigma_x - \sigma_r)^2 + (\sigma_r - \sigma_\phi)^2\right]^{0.5}$$

Note that in the derivation of the shell thickness calculation in the pre-2007 Section VIII, Division 2, $\sigma_r = (-)P/2$ was used as a representation of the average stress through the thickness. Also, for information, if σ_r is assumed to be zero for calculation purposes, $\sigma_e \approx 0.866$ PR/t for the distortion energy theory.

For a cylinder where P = 300 psi, R = 30 in., and t = .5 in., the three theories would compare as follows:

- *Maximum principal stress theory*
  $\sigma_\phi = \sigma_1 = PR/t = 300(30)/.5 = 18{,}000$ psi
- *Maximum shear stress theory*
  $\sigma_\phi - \sigma_r = PR/t + P = 300(30)/.5 + 300 = 18{,}300$ psi
- *Distortion energy theory*
  $\sigma_e = 15{,}850$ psi

Three points are obvious from a comparison of the maximum principal stress theory and the maximum shear stress theory:

1. For thin-walled pressure vessels, the two theories yield approximately the same results.
2. For thin-walled pressure vessels the radial stress is so small in comparison to the other principal stresses that it can be ignored and a state of biaxial stress is assumed to exist.
3. For thick-walled vessels ($R_m/t < 10$), the radial stress becomes significant in defining the ultimate failure of the vessel. The maximum principal stress theory is unconservative for designing these vessels. For this reason, this book has limited most of its application to thin-walled vessels where a biaxial state of stress is assumed to exist.

The results of the equivalent stress calculation from the distortion energy theory indicate that reduced shell thickness values may be obtained if a more rigorous analysis is performed. This is the basis for part of the design-by-analysis section in Section VIII, Division 2.

Failures in Pressure Vessels

Vessel failures can be grouped into four major categories, which describe *why* a vessel failure occurs. Failures can also be grouped into types of failures, which describe how the failure occurs. Each failure has a why and how to its history. It may have failed *through* corrosion fatigue *because* the wrong material was selected. The designer must be as familiar with categories and types of failure as with categories and types of stress and loadings. Ultimately they are all related.

Categories of Failures

1. *Material*—Improper selection of material; defects in material.
2. *Design*—Incorrect design data; inaccurate or incorrect design methods; inadequate shop testing.
3. *Fabrication*—Poor quality control; improper or insufficient fabrication procedures including welding; heat treatment or forming methods.
4. *Service*—Change of service condition by the user; inexperienced operations or maintenance personnel; upset conditions. Some types of service which require special attention both for selection of material, design details, and fabrication methods are as follows:
   a. Lethal
   b. Fatigue (cyclic)
   c. Brittle (low temperature)
   d. High temperature
   e. High shock or vibration
   f. Vessel contents
      - Hydrogen
      - Ammonia

- Compressed air
- Caustic
- Chlorides

Types of Failure Modes

1. *Elastic deformation*—Elastic instability or elastic buckling, must be evaluated by considering vessel geometry, stiffness as well as properties of materials.
2. *Excessive plastic deformation*—The primary stress limits as outlined in ASME Section VIII, Division 2, are intended to prevent excessive plastic deformation.
3. *Brittle fracture*—Can occur at low or intermediate temperatures. Brittle fractures have occurred in vessels made of low carbon steel in the 40°–50°F range during hydrotest where minor flaws exist. This is addressed greatly in material toughness.
4. *Stress rupture*—Italicized values in Section II, Part D indicate that allowable stress values are governed by time-dependent properties, e.g. stress rupture and creep rate.
5. *Plastic instability*—Incremental collapse; incremental collapse is cyclic strain accumulation or cumulative cyclic deformation. Cumulative damage leads to instability of vessel by plastic deformation. The primary plus secondary limits are intended to preclude any ratcheting and validate the use of elastic analysis.
6. *High strain*—Low cycle fatigue is strain-governed and occurs mainly in lower-strength/high-ductile materials. The peak stresses are used to evaluate this condition.
7. *Stress corrosion*—It is well known that chlorides cause stress corrosion cracking in stainless steels; likewise caustic service can cause stress corrosion cracking in carbon steels. Material selection is critical in these services.
8. *Corrosion fatigue*—Occurs when corrosive and fatigue effects occur simultaneously. Corrosion can reduce fatigue life by pitting the surface and propagating cracks. Material selection and fatigue properties are the major considerations.

Creep rupture from long term loading, creep-fatigue, and creep-buckling should also be addressed and procedures to do so within the Code have been around for some time now. In the past, at least some portions of the Code were limited to temperatures with the intent to avoid any time-dependent effects on the material, meaning that maximum permitted temperatures for materials were kept below the creep range.

In dealing with these various modes of failure, the designer must have an understanding and a picture of the state of stress in the various parts. It is against these failure modes that the designer must compare and interpret stress and strain values. Setting allowable stresses is not enough, as in the case of elastic instability one must consider geometry, stiffness, and the properties of the material. Material selection is a major consideration when related to the type of service. Design details and fabrication methods are as important as "allowable stress" in design of vessels for cyclic service. The designer and all those persons who ultimately affect the design must have a clear picture of the conditions under which the vessel will operate.

Loadings

Loadings or forces are the "causes" of stresses in pressure vessels. These forces and moments must be isolated both to determine *where* they apply to the vessel and *when* they apply to a vessel. Categories of loadings define where these forces are applied. Loadings may be applied over a large portion (general area) of the vessel or over a local area of the vessel. Remember both *general* and *local* loads can produce membrane and bending stresses. These stresses are additive and define the overall state of stress in the vessel or component. Stresses from local loads must be added to stresses from general loadings. These combined stresses are then compared to an allowable stress.

Consider a pressurized, vertical vessel bending due to wind, which has an inward radial force applied locally. The effects of the pressure loading are longitudinal and circumferential tension. The effects of the wind loading are longitudinal tension on the windward side and longitudinal compression on the leeward side. The effects of the local inward radial load are some local membrane stresses and local bending stresses. The local stresses would be both circumferential and longitudinal, tension on the inside surface of the vessel, and compressive on the outside. Of course the steel at any given point only sees a certain level of stress of the combined effect. It is the designer's job to combine the stresses from the various loadings to arrive at the worst probable combination of stresses utilizing both knowledge of the operation of the equipment and applicable load combinations, determine what section and part within the Code and corresponding failure theory to use, and compare the results to an acceptable stress level to obtain an economical and safe design.

This hypothetical problem serves to illustrate how categories and types of loadings are related to the stresses they produce. The stresses which are required for equilibrium of the vessel are primary stresses. The stresses due to pressure and wind are primary general membrane stresses since even if yielding occurred, redistribution of stresses would not be possible. These stresses should be limited to the Code allowable stress values, where increases for occasional loading may be allowed for certain sections of the Code.

On the other hand, the stresses from the inward radial load could be either a primary stress or secondary stress. It is a primary stress if it is produced from an unrelenting load or a secondary stress if produced by a relenting load. A general primary membrane stress will not redistribute upon yielding, whereas a primary local membrane stress will, and for a secondary stress the load will relax once slight deformation occurs.

This should make it obvious that the type and location of loading will determine the category of stress. This will be expanded upon later, but basically each combination of stresses (stress categories) will have different allowables, i.e.:

- Primary stress (P_m): $P_m < 1.0$ SE
- Primary local membrane (P_L): $P_L < 1.5$ SE
- Primary local membrane + primary bending ($P_L + P_b$): $P_L + P_b < 1.5$ SE
- Primary local membrane + primary bending + secondary ($P_L + P_b + Q$):
  $P_L + P_b + Q < 3$SE (or $2S_y$)

Whether a loading is steady, more or less continuous, or nonsteady, variable, or temporary will also have an effect on what level of stress will be acceptable. If in our hypothetical problem the loading had been pressure plus seismic plus local load, we would have a different case. Due to the relatively short duration of seismic loading, a higher "temporary" allowable stress would be acceptable. The vessel isn't expected to operate in an earthquake all the time though building codes are written such that vessels do not collapse in the event of an earthquake.

For *normal loads*, the vessel must support these loads more or less continuously during its useful life. As a result, the stresses produced from these loads must be maintained to an acceptable level.

For *occasional loads*, the vessel may experience some or all of these loadings at various times but not all at once and not more or less continuously. Therefore a temporarily higher stress is acceptable.

For *general loads* that apply more or less uniformly across an entire section, the corresponding stresses must be lower, since the entire portion of the vessel must support that loading.

For *local loads*, the corresponding stresses are confined to a small portion of the vessel and normally fall off rapidly in distance from the applied load. As discussed previously, pressurizing a vessel causes bending in certain components. But it doesn't cause the entire vessel to bend. The results are typically not as significant (except in cyclic service) as those caused by general loadings. Therefore a slightly higher allowable stress would be in order.

Loadings can be outlined as follows:

A. *Categories of loadings*
   1. *General loads*—Applied more or less continuously across a vessel section.
      a. Pressure loads—Internal or external pressure (design, operating, hydrotest, and hydrostatic head of liquid).
      b. Moment loads—Due to wind, seismic, erection, transportation.
      c. Compressive/tensile loads—Due to dead weight, installed equipment, ladders, platforms, piping, and vessel contents.
   2. Local loads—Due to reactions from supports, internals, attached piping, attached equipment, i.e., platforms, mixers, etc.
      a. Radial load—Inward or outward.
      b. Shear load—Longitudinal or circumferential.
      c. Torsional load.
      d. Moment load—Longitudinal or circumferential.
      e. Thermal load.

B. *Types of loadings*
   1. *Normal*—Long-term duration, continuous.
      a. Internal/external pressure.
      b. Dead weight.
      c. Vessel contents.
      d. Loadings due to attached piping and equipment.
      e. Loadings to and from vessel supports.
      f. Thermal loads.
      g. Wind loads.
   2. *Occasional loads*—Short-term duration; variable.
      a. Shop and field hydrotests.
      b. Earthquake.
      c. Erection.
      d. Transportation.
      e. Upset, emergency.
      f. Start up, shut down.

Table 1-1
Design load combinations

Load Combination		Thickness	Temperature	General Primary Membrane Allowable Stress	Description
1	P + Ps + D	Corroded	Design	SE	Pressure and weight
2	P + Ps + D + L	Corroded	Design	SE	Pressure, weight, and live loading
3	P + Ps + D + S	Corroded	Design	SE	Pressure, weight, and snow loading
4	P + Ps + D + 0.75L + 0.75S	Corroded	Design	SE	Pressure, weight, partial live load, and partial snow load
5	P + Ps + D + (W or 0.7E)	Corroded (4) (5)	Design	SE	Pressure, weight, and wind load or seismic load (see description of E)
6	P + Ps + D + 0.75(W or 0.7E) + 0.75L + 0.75S	Corroded (4) (5)	Design	SE	Pressure, weight, partial wind load or seismic load, partial live load, and partial snow load
7	0.6D + (W or 0.7E)	Corroded (4) (5)	Design	SE	Partial weight, and wind or seismic

P = design pressure
Ps = static head
D = dead load (e.g. weight of vessel, supports, internals, external appurtenances)
L = live load (e.g. appurtenance live loading)
E = earthquake load (strength level load)
W = wind load (e.g. wind load on vessel and appurtenances)
S = snow load (e.g. snow load on platforms)

1. For each load combination, the determination and evaluation of P = 0 (and Ps = 0) should be considered (e.g. hot shutdown).
2. For each load combination, the determination and evaluation of P = external pressure condition should be considered.
3. The effects of wind and seismic loading are not assumed to act concurrently.
4. For load combinations with wind loads, the possibility that a full wind load may occur while the vessel is empty should be considered.
5. For load combinations with seismic loads, it is typically conservative to use the uncorroded vessel weight in design load calculations and the corroded shell in stress calculations.

Stress

ASME Code, Section VIII, Division 1 vs. Division 2

ASME Code, Section VIII, Division 1 does not explicitly indicate the manner in which stresses should be combined but indicates that engineering judgment must be consistent with the philosophy of Division 1. ASME Code, Section VIII, Division 2, on the other hand, provides specific guidelines and stress categories, how they are combined, and the allowable stresses for each category and combination of categories. Part 5 of Division

2 is design-by-analysis whereas Division 1 and Part 4 of Division 2 are design-by-rules. Although stress analysis as utilized by Part 5 of Division 2 is beyond the scope of this book, the use of stress categories, definitions of stress, and allowable stresses is applicable.

Division 1 and the procedures outlined in this book consider a biaxial state of stress combined in accordance with the maximum principal stress theory. Division 2 considers triaxial stresses evaluated in accordance with the maximum shear stress theory and distortion energy theory. Just as you would not design a nuclear reactor to the rules of Division 1, you would not design an air receiver by the rules of Division 2. Each has its place and application. The following discussion on categories of stress and allowable stresses will utilize information from Division 2, which can be applied in general to all vessels.

Stress Categories

The shell thickness as computed by Code formulas for internal or external pressure alone is often not sufficient to withstand the combined effects of all other loadings. Detailed calculations consider the effects of each loading separately and then must be combined to give the total state of stress in that part.

Types of stress, stress categories, and allowable stresses are based on the type of loading that produced them *and* on the hazard they represent to the structure. Unrelenting loads produce primary stresses. Relenting loads (self-limiting) produce secondary stresses. Primary stresses must be kept lower than secondary stresses. Primary plus secondary stresses are allowed to be higher and so on. Before considering the combination of stress categories, we must first define the various *types* of stress and each *category*.

Types of Stress

There are many names to describe types of stresses. As these stresses apply to pressure vessels, we group all types of stress into three major classes of stresses, and subdivision of each of the groups is arranged according to their effect on the vessel. The following list of stresses describes types of stresses without regard to their effect on the vessel or component. They define a direction of stress or relate to the application of the load.

1. Tensile
2. Compressive
3. Normal
4. Shear
5. Membrane
6. Bending
7. Bearing
8. Axial
9. Discontinuity
10. Principal
11. Thermal
12. Tangential
13. Load controlled
14. Strain controlled
15. Circumferential
16. Longitudinal
17. Radial

Stress Categories

The foregoing list provides the categories and subcategories. It is, however, too general to provide a basis with which to combine stresses or apply allowable stresses. Stress categories are defined by the type of loading which produces them and the hazard they represent to the vessel.

1. *Primary stress*
   a. General membrane stress, P_m
   b. Local membrane stress, P_L
   c. Bending stress, P_b
2. *Secondary stress*
   a. Secondary membrane stress, Q_m
   b. Secondary bending stress, Q_b
3. *Peak stress,* F

Definitions and examples of these stresses follow.

Primary stresses: These stresses are normal or shear stresses which are required to satisfy equilibrium. They are produced by mechanical loads (load controlled) and when exceeding the yield strength can result in failure or gross distortion. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses are generally due to internal or external pressure or produced by sustained external forces and moments. Thermal stresses from thermal gradients and imposed displacements are never classified as primary stresses.

Primary general stresses are divided into membrane and bending stresses. The need for dividing primary general stress into membrane and bending is that the calculated value of a primary bending stress may be allowed to go to a higher value than that of a primary general membrane stress.

Primary general membrane stress, P_m. This stress is the average primary stress across a solid section, is produced by pressure or mechanical loads, and is remote from discontinuities such as head-shell intersections, cone-cylinder intersections, nozzles, and supports. Examples are:

a. Shells away from discontinuities due to internal pressure.
b. Compressive and tensile axial stresses due to wind.
c. Axial compression due to weight.
d. Nozzles within the limits of reinforcement due to internal pressure.

Primary local membrane stress, P_L. A primary local membrane stress is produced either by design pressure alone or by other mechanical loads. Primary local membrane stresses have some self-limiting characteristics like secondary stresses. Since they are localized, once the yield strength of the material is reached, the load is redistributed to stiffer portions of the vessel. However, since any deformation associated with yielding would be unacceptable, an allowable stress lower than a secondary stress is assigned. The ability of primary local membrane stresses to redistribute after the material yields allows for a higher allowable stress but only in a local area.

The bending stresses associated with a local loading are almost always classified as secondary stresses. Therefore, the membrane stresses from a WRC-107-type analysis must be broken out separately and combined with general primary stresses due to internal pressure, for example.

Examples of local primary membrane stresses exist:

a. Where internal pressure is the origin of stress and at a discontinuity:
   1. On the shell near a nozzle or other opening
   2. Head-shell juncture
   3. Cone-cylinder juncture
   4. Shell-flange juncture
   5. Head-skirt juncture
   6. Shell-stiffening ring juncture
b. Where non-pressure applied loads are the origin of stress and at a discontinuity:
   1. Support lugs
   2. Nozzle external loads
   3. Beam supports
   4. Major attachments

Primary bending stress, P_b. Primary bending stresses are due to pressure or mechanical loads and are through the thickness. There are relatively few areas where primary bending occurs:

a. Center of a flat head or crown of a dished head.
b. Shallow conical head.
c. In the ligaments of closely spaced openings.

Secondary stresses. These stresses are normal or shear stresses which are required to satisfy an imposed strain pattern. The basic characteristic of a secondary stress is that it is self-limiting. As defined earlier, this means that local yielding and minor distortions can satisfy the conditions which caused the stress to occur. Application of a secondary stress cannot cause structural failure of the vessel due to the restraints offered by the body to which the part is attached. Secondary stresses can develop at structural discontinuities but are also used to describe through thickness gradients away from structural discontinuities. Secondary stresses are also produced by sustained loads other than internal or external pressure.

Structural discontinuities that develop secondary stresses should be placed apart by at least $2.5\sqrt{R_m t}$. This restriction is to eliminate the additive effects of edge moments and forces.

Secondary stresses are divided into two additional groups, membrane and bending, though the Code makes no distinction in nomenclature for Q_m or Q_b. Examples of each are as follows:

Secondary membrane stress, Q_m.

a. Axial thermal gradients in shells, cones, or formed heads.
b. Thermal gradients between the shell and head.
c. Thermal stresses due to differential thermal expansion within a nozzle wall.
d. Pressure stress at an isolated ligament.

Secondary bending stress, Q_b.

a. Axial thermal gradients in shells, cones, or formed heads.
b. Thermal gradients between the shell and head.
c. Head-shell juncture
d. Nozzles outside the limits of reinforcement due to pressure and external loading.
e. Thermal stresses due to differential thermal expansion within a nozzle wall.

Peak stress, F. Peak stresses are the additional stresses due to stress intensification in highly localized areas. They apply to both sustained loads and self-limiting loads. There are no significant distortions associated with peak stresses.

Peak stresses are additive to primary and secondary stresses present at the point of the stress concentration. Peak stresses are only significant in fatigue conditions or brittle materials. Peak stresses are sources of fatigue cracks and apply to normal and shear stresses. Examples are:

a. Stress at the corner of a discontinuity (e.g. fillet weld or corner).
b. Thermal stresses due to differential thermal expansion within a nozzle wall.
c. Thermal stresses in cladding or weld overlay.
d. Stress due to notch effect (stress concentration).

Stress Limits

Once the various stresses of a component are calculated, they must be combined and this final result compared to an allowable stress (Table 1-1). Table 1-1 is derived basically from ASME Code, Section VIII, Division 2, and borrowed for application to Division 1 vessels and determining allowable stresses. It should be used as a guideline only because Division 1 recognizes only two categories of stress—primary membrane stress and primary bending stress. Since the calculations of most secondary and peak stresses are not included in this book, these categories can be considered for reference only. In addition, Division 2 utilizes load combinations, by which short-term loads (such as seismic) are reduced when combined with other loads. It also sets allowable limits of combined stresses for fatigue loading where secondary and peak stresses are major considerations.

Table 1-2
Allowable Stresses for Stress Classifications and Categories

Stress Classification or Category	Allowable Stress
General primary membrane, P_m	SE
Primary membrane stress plus primary bending stress across the thickness, $P_m + P_b$	1.5SE
Local primary membrane, P_L	1.5SE
Local primary membrane plus primary bending, $P_L + P_b$	1.5SE
Secondary membrane plus secondary bending, $Q_m + Q_b$	$3SE < 2F_y$
P + Q	$3SE < 2F_y$
$P_m + P_b + Q_m + Q_b$	$3SE < 2F_y$
$P_L + P_b + Q_m + Q_b$	$3SE < 2F_y$
Peak, F	S_a
P + Q + F	S_a
$P_m + P_b + Q_m + Q_b + F$	S_a
$P_L + P_b + Q_m + Q_b + F$	S_a

Notes:
F_y = minimum specified yield strength at design temperature
E = joint efficiency
S = allowable stress per ASME Code, Section VIII, Division 1, at design temperature
S_a = alternating stress for any given number of cycles from design fatigue curves
The term 3SE shall be used in lieu of $2F_y$ when the ratio of minimum specified yield strength to ultimate strength exceeds 0.7 or S is governed by time-dependent properties.

Thermal Stresses

Whenever the expansion or contraction that would occur normally as a result of heating or cooling an object is prevented, thermal stresses are developed. Thermal stresses are always caused by some form of mechanical restraint.

Thermal stresses are “secondary stresses” because they are self-limiting. That is, yielding or deformation of the part relaxes the stress (except thermal stress ratcheting). Thermal stresses will not cause failure by rupture in ductile materials except by fatigue over repeated applications. They can, however, cause failure due to excessive deformations.

Mechanical restraints are either internal or external. External restraint occurs when an object or component is supported or contained in a manner that restricts thermal movement. An example of external restraint occurs when piping expands into a vessel nozzle creating a radial load on the vessel shell. Internal restraint occurs when the temperature through an object is not uniform. Stresses from a “thermal gradient” are due to internal restraint. Stress is caused by a thermal gradient whenever the temperature distribution or variation within a member creates a differential expansion such that the natural growth of one fiber is influenced by the different growth requirements of adjacent fibers. The result is distortion or warpage.

A transient thermal gradient occurs during heat-up and cool-down cycles where the thermal gradient is changing with time. Thermal gradients may be logarithmic or linear across a vessel wall. Given a steady heat input inside or outside a tube the heat distribution will be logarithmic if there is a temperature difference between the inside and outside of the tube. This effect is significant for thick-walled vessels. A linear temperature distribution may be assumed if the wall is thin. Stress calculations are much simpler for linear distribution.

Thermal stress ratcheting is progressive incremental inelastic deformation or strain that occurs in a component that is subjected to variations of mechanical and thermal stress. Cyclic strain accumulation ultimately can lead to incremental collapse. Thermal stress ratcheting is the result of a sustained load and a cyclically applied temperature distribution.

The fundamental difference between mechanical stresses and thermal stresses lies in the nature of the loading. Thermal stresses as previously stated are a result of restraint or temperature distribution. The stress pattern must only satisfy the requirements for equilibrium of the internal forces. The result being that yielding will relax the thermal stress. If a part is loaded mechanically beyond its yield strength, the part will continue to yield until it breaks. The external load remains constant, thus the internal stresses cannot relax.

The equations and relationships for thermal stresses become increasingly complex when considering thermal gradients, transient thermal gradients, logarithmic gradients, and partial restraint. The basic equations follow. If the temperature of a unit cube is changed from T_1 to T_2 and the growth of the cube is fully restrained:

where T_1 = initial temperature, °F
T_2 = new temperature, °F
α = mean coefficient of thermal expansion in./in./°F
E = modulus of elasticity, psi
υ = Poisson's ratio = .3 for steel
ΔT = mean temperature difference, °F

Case 1: If the bar is restricted only in one direction but free to expand in the other direction, the resulting uniaxial stress, σ, would be;

$\sigma = -E\,\alpha\,(T_2 - T_1)$
If $T_2 > T_1$, σ is compressive (expansion).
If $T_1 > T_2$, σ is tensile (contraction).

Case 2: If restraint is in both directions, x and y, then;

$\sigma_x = \sigma_y = -\alpha\,E\,\Delta T/(1 - \upsilon)$

Case 3: If restraint is in all three directions, x, y, and z, then;

$\sigma_x = \sigma_y = \sigma_z = -\alpha\,E\,\Delta T/(1 - 2\upsilon)$

A linear thermal gradient through the thickness of a vessel wall, due to temperature difference between the outer and inner wall (thin wall) may be shown as follows;

$$\sigma = +/-\,E\,\alpha\,\Delta T/(2(1-\upsilon))$$
$$= .715\,E\,\alpha\,\Delta T\ (\text{if } \upsilon = .3)$$

This is a bending stress. If the hotter side is on the inside surface, the hot side is in compression since it wants to expand but is restricted, and if the cold side is on the outside surface it is in tension since it wants to contract but is restricted. Note that this is not a function of vessel diameter or thickness. The stress is due to internal restraint.

A sudden temperature change, ΔT, that penetrates a short distance (but not across the entire shell thickness) is as follows:

$$\sigma = +/-\,E\,\alpha\,\Delta T/((1-\upsilon))$$
$$= 1.43\,E\,\alpha\,\Delta T\ (\text{if } \upsilon = .3)$$

The average temperature between a nozzle attached to a rigid wall, ΔT, has an upper limit of discontinuity stress of the following:

$\sigma = 1.83\,E\,\alpha\,\Delta T\ (\text{if } \upsilon = .3)$

Discontinuity Stresses

Vessel sections of different thickness, material, diameter, and change in directions would all have different displacements if allowed to expand freely. However, since they are connected in a continuous structure, they must deflect and rotate together. The stresses in the respective parts at or near the juncture are called discontinuity stresses. Discontinuity stresses are necessary to satisfy compatibility of deformation in the region. They are local in extent but can be of very high magnitude. Discontinuity stresses are self-limiting but some stresses require to be classified as local primary membrane stresses to avoid distortion. That is, once the structure has yielded, the forces causing excessive stresses are reduced. In a typical application they will not lead to failure. Discontinuity stresses do become an important factor in fatigue design where cyclic loading is a consideration. Design of the juncture of the two parts is a major consideration in reducing discontinuity stresses.

It is necessary to superimpose the general membrane stresses with the discontinuity stresses. From superposition of these two states of stress, the total stresses are obtained. Generally when combined, a higher allowable stress is permitted. Due to the complexity of determining discontinuity stresses, solutions will not be covered in detail here. The designer should be aware that for designs of high pressure (>1,500 psi), brittle material or cyclic loading, discontinuity stresses may be a major consideration.

There are two major methods for determining discontinuity stresses:

1. *Displacement Method*—Conditions of equilibrium are expressed in terms of displacement.
2. *Force Method*—Conditions of compatibility of displacements are expressed in terms of forces.

Fatigue Analysis for Cyclic Service

Some vessels are subjected to periodic repetitions of mechanical and thermal loads and the resulting stresses during their service life. When a vessel is subject to repeated loading that could cause failure by the development of progressive fracture, the vessel is considered to be in cyclic service. The ASME Code, Section VIII, Division 1, does not specifically provide details for the design of vessels in cyclic service. However ASME Section VIII, Division 2 has detailed procedures for determining if a vessel in cyclic service requires a detailed fatigue analysis or not, and how to conduct the analysis.

Not every vessel in cyclic service is exposed to the number and magnitude of stress cycles that could shorten its design life. The Code recognizes this and has provided a "screening procedure" to determine whether a vessel is exempt from a rigorous fatigue analysis or not. A detailed fatigue analysis is not mandatory when the limits given by the Code are not exceeded.

Failure depends upon the number of repetitions at a given range of stress rather than the total time under load. When the stress level falls below a certain limit, the number of allowable cycles is said to be indefinite. This value of stress is known as the endurance or fatigue limit, and indicates that a very large number of cycles could occur.

Cracks are more likely to occur in areas of high stress. As such, pressure parts are the most susceptible. Since the longitudinal seams are stressed to twice that of the circumferential joints, they would be most suspect. In addition cracks typically begin on the inside of the vessel.

It is recognized that Code formulas for design of details, such as heads, can result in yielding in localized regions. Thus, localized stresses, exceeding the yield strength of the material, may be encountered even though low allowable stresses have been used in the design. These vessels, while safe for relatively static conditions, could develop "progressive fracture" after a large number of repeated loadings due to high localized and secondary bending stresses.

Fatigue failure can be a result of pressure or temperature fluctuations, as well as other loadings. Fatigue failures have occurred in boiler drums due to temperature variations in the shell at the feed water inlet.

Behavior of metal under fatigue conditions varies significantly from normal stress-strain relationships. Damage accumulates during each cycle of loading and develops localized regions of high stress until subsequent repetitions finally cause cracks to begin and propagate. Progressive fractures develop at stress risers and discontinuities even though the average stress in the vessel may be at design levels.

Factors Influencing Design

Vessels in cyclic service require special consideration for both design and fabrication. Design details play a major role in eliminating or reducing regions of stress risers and concentrations. It is not uncommon to have the design life of a vessel cut in half by poor design details. Although it is not possible to eliminate all stress risers and concentrations, some design details to be avoided are as follows;

1. Use integral construction
2. Avoid fillet welds for attachments to pressure boundary
3. Avoid reinforcing pads
4. Avoid threaded connections
5. Avoid partial penetration welds
6. Avoid stud bolt connections
7. Avoid nozzles in knuckle region of heads

In fatigue service the localized stresses at abrupt changes in thickness, abrupt changes in section, such as at a head junction or nozzle opening, misalignment, defects in construction and thermal gradients are the significant stresses. In general it is always beneficial to minimize peak stresses to the lowest level possible. Peak stresses often become the key stresses involving a fatigue analysis. Peak stresses occur at stress concentrations due to;

1. Fillet welds - high stress at corner of welds
2. Changes in thickness
3. Offset plates
4. Change in geometry
5. Welds attaching clips
6. Welds attaching nozzles

In welded regions, the influencing factors are unknown at the design stages, so they must be compensated for in overall safety factors applied to the procedure. The weld influencing factors are as follows;

1. Local surface notches such as weld bead roughness, weld ripples, undercut, local shrinkage grooves, local root concavity and welding start/stop craters.
2. Variation of the material properties in the various weld zones.
3. Residual stresses
4. Internal defects
5. Weld repairs

In addition Section VIII, Division 2 contains data for smooth bar design fatigue curves and welded joint fatigue curves. The curves represent testing conducted in air, and can be adjusted for the effect of corrosive environments. In general, corrosive environments have a strong, detrimental effect on fatigue behavior. Fatigue cracks in corrosive environments can occur at lower stress ranges; they can occur earlier and propagate faster. Corrosion can cause pitting, non-uniform wall thickness, crevices, gaps, and cracks… all of which reduce fatigue life.

Fabrication tolerances can be equally important details. Normal tolerances for vessels are not adequate. The ASME tolerance for out of roundness of 1% is not always acceptable. Peaking and banding tolerances may need to be much lower than the Code allows. Offsets between plates should be carefully controlled.

Additional factors for carbon steel vessels in cyclic service:

1. Vessels shall be PWHT
2. Material to be normalized
3. Material to be fine grain practice (7 or finer)
4. Plate material shall be 100% UT examined
5. All welds full penetration
6. All main seams ground flush
7. All attachment welds, internal or external, shall be ground smooth or contoured

In actuality, the surface finish, geometry, welded condition, temperature, environmental properties, and non-uniform material properties are drastically different than the test samples. As such, the ASME Code modified the data from the smooth bar fatigue curves by 2 on strain (or stress), or 20 on the number of cycles, whichever was more conservative.

Histograms

A histogram should be developed for each vessel to better assess the quantity of cycles the vessel or component will be subjected to. The purpose of the histogram(s) is to break down the loading history into individual cycles. The loading histogram should be determined based on the specified loadings provided in the UDS (User's Design Specification). The loading histogram should include all significant operating loads and events that are applied to a component. Examples of various histograms are given below.

Cycle Counting Methods

Section VIII, Division 2, Annex 5-B gives procedures for developing a histogram as well as two procedures for counting cycles. There are four major methods used for cycle counting as follows;

1. Rainfall Cycle Counting Procedure
2. Max-Min Cycle Counting Procedure
3. Reservoir Cycle Counting Procedure
4. Simplified Cycle Counting Procedure

Items 1 and 2 are described in Section VIII, Division 2. Procedures 3 and 4 are from other sources. Details for counting are not provided here.

Screening Procedure

Section VIII, Division 1 does not have either a screening method or a procedure for performing a fatigue analysis. However, Section VIII, Division 2 has both. However, it is still acceptable to build and stamp a vessel in cyclic service to Section VIII, Division 1 but to do the fatigue analysis per Section VIII, Division 2. There is no requirement for a vessel to be built to Section VIII, Division 2 simply because it is in cyclic service. On the other hand, Section VIII, Division 1 vessels are not exempt from fatigue analysis because they are not built to Section VIII, Division 2.

There are three Screening Criteria detailed in Section VIII, Division 2 for determining whether a fatigue analysis must be performed or if the vessel is exempt. The three criteria are as follows;

1. Screening Based on Experience with Comparable Equipment
2. Screening Method A
3. Screening Method B

Detailed descriptions of the screening methods are the following;

Screening Based on Experience with Comparable Equipment: Section VIII, Division 2, Paragraph 5.5.2.2 allows a new vessel to be exempted from fatigue analysis if comparable equipment with successful experience and similar loadings is obtained.

The criteria are;

1. Successful experience over a sufficient time frame
2. Similar histogram
3. Addressed in UDS (User's Design Specification)
4. Comparable equipment
5. Similar or same operating conditions

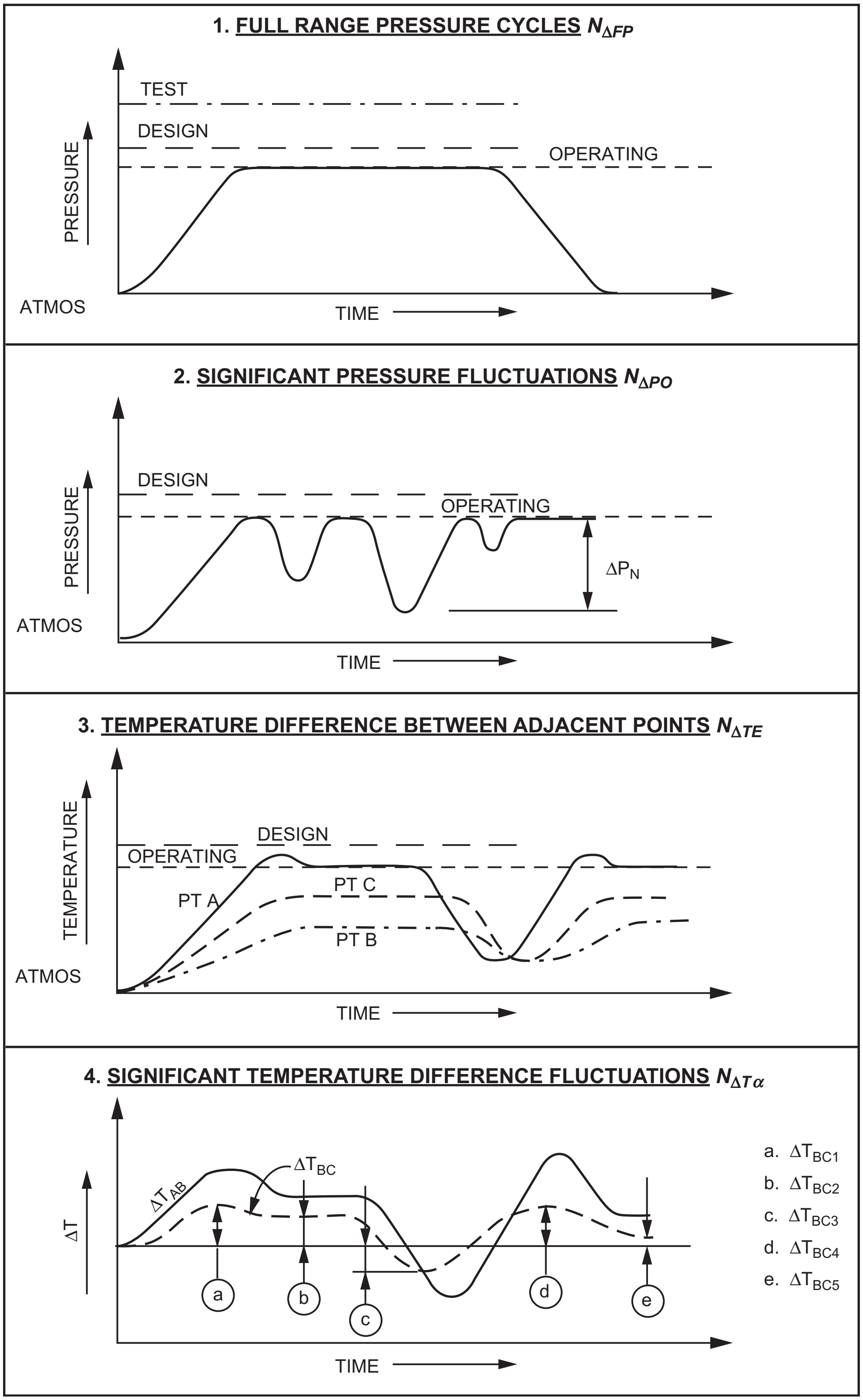
Screening Method A: Section VIII, Division 2, Paragraph 5.5.2.3 can only be used for vessels constructed of materials with an ultimate tensile strength less than or equal to 80,000 PSI (550 Mpa). It is a more simple method of A and B and is more conservative. The steps are as follows;

1. Determine load history… a histogram.
2. Determine the quantity of full range pressure cycles, $N_{\Delta FP}$, including start up and shut down. One full range pressure cycle would encompass starting up and shutting down.
3. Determine quantity of partial pressure fluctuations, $N_{\Delta PO}$. The cycles to be included in this category will vary depending on the type of construction; 20% of the design pressure for integral construction, and 15% for non-integral construction.
4. Determine the quantity of cycles for variation in temperature between adjacent points, $N_{\Delta TE}$. An adjacent point for a shell and dished head is $L = 2.5\ (Rt)^{1/2}$ or 3.5a for flat plates, where 'a' is equal to the radius of a hot spot or a heated area within a plate. This number is modified based on the range of temperatures by the procedure outlined in Section VIII, Division 2.
5. Determine the quantity of cycles for components having different coefficients of expansion, $N_{\Delta T\alpha}$. This does not include cladding.
6. Compare the total number of cycles, $N_T = N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha}$ with the criteria listed in Table 5.9 of ASME Section VIII, Division 2, shown in Table 1-3.

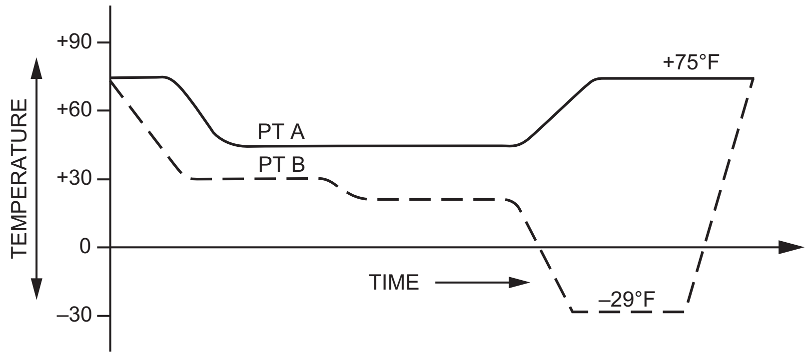
Table 1-3
Fatigue screening criteria for method A

Description		
Integral Construction	Attachments and nozzles in the knuckle region of formed heads	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 350$
	All other components	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 1000$
Non-Integral Construction	Attachments and nozzles in the knuckle region of formed heads	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 60$
	All other components	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 400$

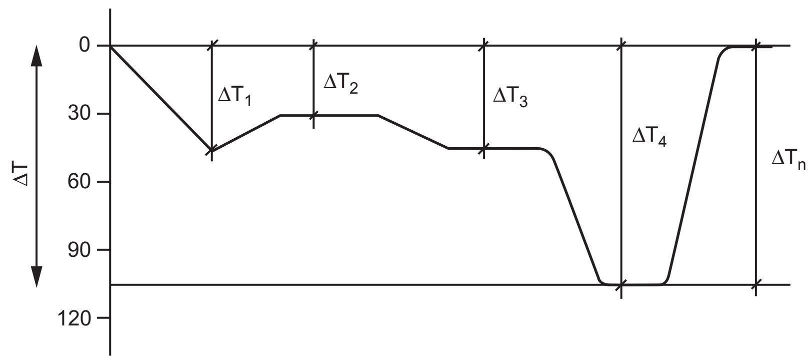
EXAMPLES OF HISTOGRAMS



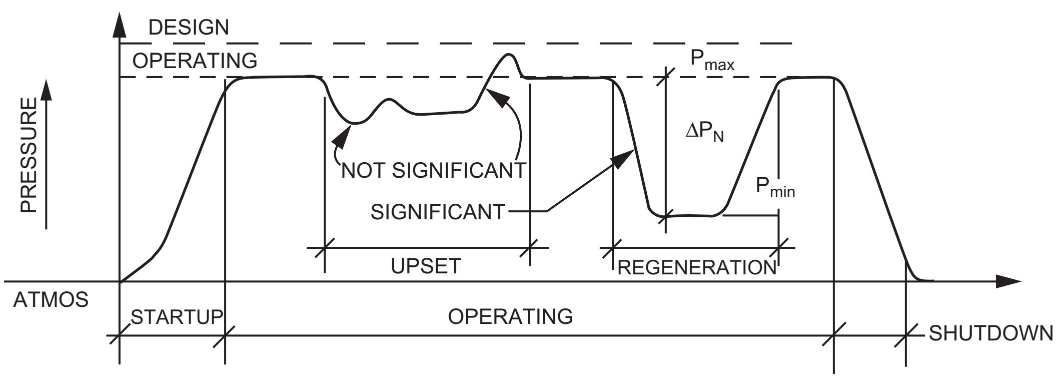
5. TEMPERATURE DIFFERENCE BETWEEN ADJACENT POINTS



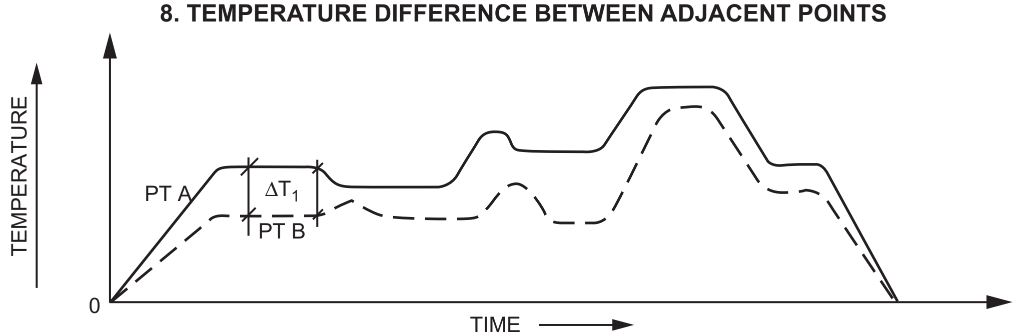
6. SIGNIFICANT TEMPERATURE DIFFERENCE FLUCTUATIONS

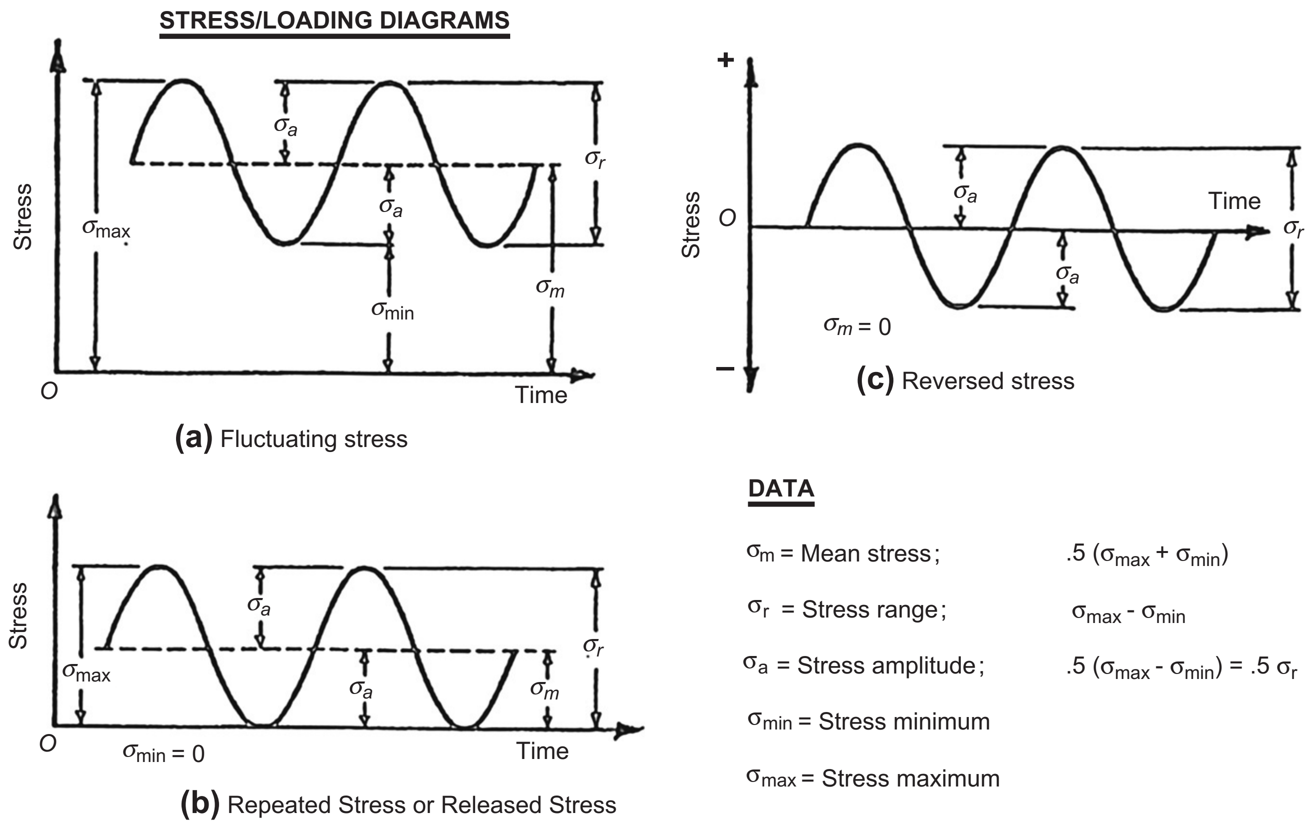


7. SIGNIFICANT PRESSURE FLUCTUATIONS



8. TEMPERATURE DIFFERENCE BETWEEN ADJACENT POINTS





Screening Method B: Section VIII, Division 2, Paragraph 5.5.2.4 indicates that the fatigue screening Method B may be used for all materials. It is not a simple, single procedure as is outlined in method A. Six of the steps require stress calculations, and not satisfying any of the six calculation steps results in a requirement to perform a detailed fatigue analysis.

1. Determine load history… a histogram.
2. Determine screening criteria factors, C_1 and C_2, where for integral construction of attachments and nozzles in the knuckle region of formed heads, $C_1 = 4$ and $C_2 = 2.7$, and for other components $C_1 = 3$ and $C_2 = 2$. For non-integral construction, $C_1 = 5.3$ and $C_2 = 3.6$, and for other components $C_1 = 4$ and $C_2 = 2.7$.
3. Determine the quantity of full range pressure cycles, $N_{\Delta FP}$, including start up and shut down. One full range pressure cycle would encompass starting up and shutting down. N(S) is the number of cycles from the fatigue curve at the stress amplitude.

$$N_{\Delta FP} \leq N(C_1 S)$$

4. Determine the maximum range of pressure fluctuation (excluding startups and shutdowns), ΔP_N, and the corresponding number of cycles, $N_{\Delta P}$. Note that P is the design pressure and S is the allowable stress at design temperature.

$$\Delta P_N \leq \frac{P}{C_1}\left(\frac{S_a(N_{\Delta P})}{S}\right)$$

5. Determine the maximum temperature difference between two adjacent points of the vessel during normal operation, and during startup and shutdown, ΔT_N, and the corresponding number of cycles, $N_{\Delta TN}$. The value α is the coefficient of thermal

expansion, and E_{ym} is the modulus of elasticity at the mean temperature of the cycle.

$$\Delta T_N \leq \left(\frac{S_a(N_{\Delta TN})}{C_2 E_{ym} \alpha}\right)$$

6. Determine the maximum temperature difference between two adjacent points of the vessel during normal operation, and during startup and shutdown, ΔT_R, and the corresponding number of cycles, $N_{\Delta TR}$.

$$\Delta T_R \leq \left(\frac{S_a(N_{\Delta TR})}{C_2 E_{ym} \alpha}\right)$$

7. Determine the range of temperature difference between two adjacent points for components made from different materials during normal operation, ΔT_M, and the corresponding number of cycles, $N_{\Delta TM}$. The values E_{y1} and E_{y2} are the moduli of elasticity for materials one and two, respectively, at the mean temperature of the cycle.

$$\Delta T_M \leq \left(\frac{S_a(N_{\Delta TM})}{C_2\left(E_{y1}\alpha_1 - E_{y2}\alpha_2\right)}\right)$$

8. Determine the equivalent stress range from the full mechanical loads, excluding pressure but including piping reactions, ΔS_{ML}, and the corresponding number of cycles, $N_{\Delta S}$.

$$\Delta S_{ML} \leq S_a(N_{\Delta S})$$

Fatigue Assessment: Section VIII, Division 2, Part 5 contains methods for performing an actual fatigue analysis. They are as follows:

1. Elastic Stress Analysis and Equivalent Stresses. This is based off of the pre-2007 Section VIII, Division 2 methodology. Stress ranges will be the output values using this analysis.
2. Elastic-Plastic Stress Analysis and Equivalent Strains. Both stress and strain ranges will be the output using this analysis.
3. Elastic Analysis and Structural Stress (for welds). This method was incorporated into Section VIII, Division 2, to allow for a treatment of welded joints. Stress ranges will be the output values using this analysis.

Ratcheting: Protection against ratcheting shall be performed even if the fatigue screening criteria are met. Ratcheting is progressive incremental inelastic strain that is a result of either mechanical or thermal stress (where thermal stress ratcheting is used to indicate that thermal stresses are mostly responsible for ratcheting action). As ratcheting causes cyclic straining of the material, it can lead to failure by fatigue or collapse. If the loading results in primary plus non-cyclic secondary stresses, ratcheting will be avoided. Shakedown would occur if only initial plastic deformations occurred but upon unloading and reloading only elastic primary and secondary stresses were developed, hence the term, 'shakedown to elastic action'. The methods outlined in Section VIII, Division 2, Part 5 are as follows:

1. Elastic Stress Analysis – Elastic Ratcheting Analysis Method. If the limits of $P_L + P_b + Q$ and including general thermal effects are met by limiting this value to S_{PS}, which is equal to the greater of 3S or $2S_y$, where S is the allowable tensile stress and S_y is the yield strength and where the average value between the specified highest and lowest temperatures are used.
2. Elastic Stress Analysis – Simplified Elastic-Plastic Method. This method may be used in the case where the method indicated above shows the $P_L + P_b + Q$ stress limits are not satisfied, but indicates that the $P_L + P_b + Q$ range and excluding thermal effects must be less than S_{PS}. Additionally, the effective alternating equivalent stress amplitude must include the fatigue penalty factor, $K_{e,k}$, which is based off of the simplified elastic-plastic criteria from the pre-2007 Section VIII, Division 2. Finally, a thermal stress ratcheting assessment must be made.
3. Elastic Stress Analysis – Thermal Stress Ratcheting Assessment. This section will evaluate the allowable limit on the secondary stress range from cyclic thermal loading.

General

The fatigue exemption is performed on a component or part basis since the stress level varies on a component by component basis. One component may be exempt, while another may not be. For any component that is not exempt, a fatigue analysis must be performed for that component.

It should be noted that not all vessels will have a 20 or 30 year life. Some vessels will have a significantly shorter life. The number of allowable cycles for the most highly stressed component will determine the life of the vessel. After the design life of the vessel is reached, the owner or user must then determine whether to retire the vessel, or apply some periodic inspections and NDE to determine if the vessel can continue in service.

If the screening method determines that a fatigue analysis is required, ASME Section VIII, Division 2, Part 5 gives detailed methods for performing this analysis. Basically the fatigue analysis process consists of a rigorous stress analysis of the whole vessel to find the points of highest stress. The highest stress is then used on a fatigue curve for that material to find the allowable number of cycles. If the allowable number of cycles exceeds the actual number of cycles, then that part and vessel are acceptable. If, on the other hand, the actual number of cycles exceeds the allowable number of cycles, then the design must be altered until an acceptable result is achieved, or a shortened design life is acceptable.

Fatigue Curves

Fatigue curves are used to determine the number of allowable cycles. The fatigue curve is also known as the S – N diagram, because one axis represents stress, S, and the other axis represent number of cycles, N. Each material group has their own fatigue curve based on test results and are shown in ASME Section VIII, Division 2, Annex 3-F.

The fatigue curves can be used in several ways as follows;

1. If the number of cycles is known, then you can determine the maximum allowable alternating stress that corresponds to that number of cycles. As long as the actual stress is less than this value, then the design is acceptable.
2. If the actual alternating stress is known, then you can determine the maximum number of allowable cycles based on that stress. If this quantity is greater than the actual number of cycles designed for, then the design is acceptable. As an alternative, the design life can be determined by this method, given the allowable number of cycles.

Permissible Number of Cycles

The permissible number of cycles, N, is based on the alternating equivalent stress amplitude taken from the applicable fatigue curve. Corrections for temperature should be made as follows;

$$N = 10^{\times}\left(E_T/E_{FC}\right)$$

Where;

X = Exponent used to compute the permissible number of cycles
E_T = Modulus of elasticity at temperature evaluated
E_{FC} = Modulus of elasticity used to establish the design fatigue curve.

Effective Alternating Equivalent Stress

The value of alternating stress taken from the fatigue curve is subject to other factors given by the ASME Code. The ultimate allowable stress for a given number of cycles should be adjusted for these factors as follows (for elastic stress analysis and equivalent stresses;

$$S_a = \frac{K_f \cdot K_{e,k} \cdot \Delta S_{P,k}}{2}$$

Where;
K_f = Fatigue strength reduction factor
$K_{e,k}$ = Fatigue penalty factor
$\Delta S_{P,k}$ = Effective equivalent stress range

Cumulative Usage (Damage), U

Often vessels are not subjected to the same range of stress throughout its entire life, but rather to a variety of stress levels for different periods. The results are additive to each other and known as total cumulative damage. S_E in the figure indicates the endurance limit. The check of total cumulative damage can be determined by the following procedure as follows;

Let n_1 = Number of cycles endured at S_1 where N_1 is the fatigue life at that stress

Let n_2 = Number of cycles endured at S_2 where N_2 is the fatigue life at that stress

Let n_3 = Number of cycles endured at S_3 where N_3 is the fatigue life at that stress

Then; $D_{f,k} = [\, n1 / N1 + n2 / N2 + n3 / N3 + \ldots + nk / Nk] =$ Cumulative Damage ≤ 1

Definitions

Adjacent Points: Any two points less than distance $2(R_m t)^{1/2}$ apart.

Cycle: A cycle is a relationship between stress and strain that is established by a specified loading at a specific location. More than one stress-strain cycle can be produced at a given location.

Endurance Limit: The value of stress below which a material can presumably endure an infinite number of cycles, S_E.

Fatigue Life: The number of stress cycles which can be sustained for a given condition, N_A.

Nominal Stress: The stress calculated by simple theory without taking into account variations in stress caused by holes, grooves, fillets and other structural discontinuities.

Peak: The point at which the stress histogram goes from positive to negative.

Stress Cycle: The smallest part of the stress-time function which is repeated periodically and identically. A stress cycle is a condition in which the alternating stress goes from an initial value to a maximum value, then to a minimum value and then returns to the initial value. A single operational cycle may result in one or more stress cycles.

Stress Cycles Endured: The number of cycles endured at any stage of life, N_T.

S-N Diagram: AKA fatigue curve. A plot of alternating stress, S_a, against the maximum number of allowable of cycles, N_A.

Valley: The point at which the stress histogram goes from negative to positive.

Notation

C_1, C_2 = Factors for fatigue screening B

D_f = Cumulative fatigue damage

K_f = Fatigue strength reduction factor

$K_{e,k}$ = Fatigue penalty factor

ΔP_n = Fluctuating pressure

$\Delta S_{P,k}$ = Effective equivalent stress range

ΔS_{ML} = Equivalent stress range from the full range of mechanical load cycles (excluding pressure)

ΔT = Operating temperature range

ΔT_M = Temperature difference between two adjacent points for components made from difference materials

ΔT_N = Temperature difference between two adjacent points for components including startup and shutdown

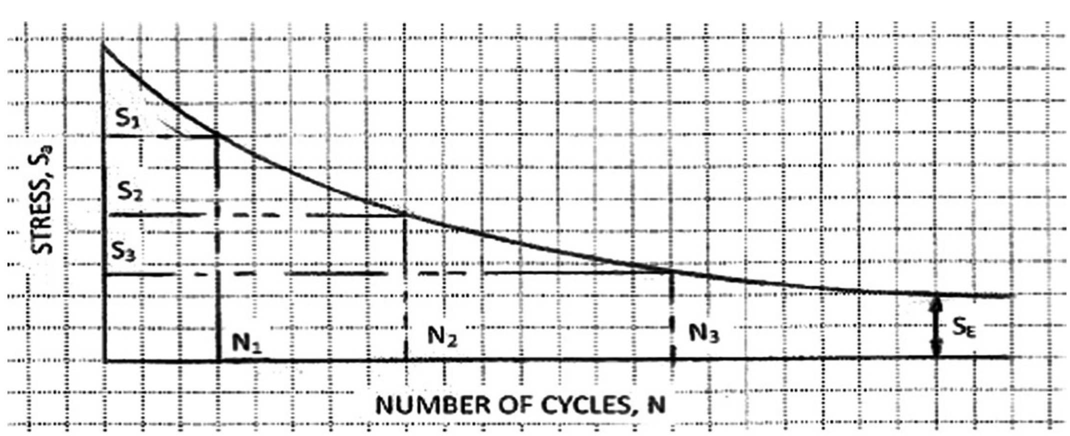
ΔT_R = Temperature difference between two adjacent points for components including startup and shutdown

$N_{\Delta FP}$ = Number of full range pressure cycles including startup and shutdown

$N_{\Delta P}$ = Number of significant partial pressure cycles

$N_{\Delta PO}$ = Number of significant pressure fluctuation cycles, not including startup or shutdown.

$N_{\Delta S}$ = Number of significant full range of mechanical load cycles (excluding pressure)



$N_{\Delta TE}$ = Number of cycles in metal temperature between adjacent points.
$N_{\Delta TM}$ = Allowable number of cycles for temperature fluctuations for components fabricated from different materials
$N_{\Delta TN}$ = Allowable number of cycles for maximum temperature difference between any two adjacent points
$N_{\Delta TR}$ = Allowable number of cycles for range of temperature fluctuations
$N_{\Delta T\alpha}$ = Number of cycles for components having different coefficients of expansion
S = Maximum allowable stress, tension, ASME VIII-1
S_a = Maximum alternating stress, PSI
S_m = Maximum allowable stress, tension, ASME VIII-2

Creep

Creep is a time dependent phenomena by which the material permanently deforms under stress, and occurs at elevated temperatures. It is a function of the material, stress, temperature, and time. There are several failure modes associated with creep, namely creep rupture, creep fatigue, and creep buckling. Higher stress levels or higher temperatures will result in faster creep rates.

Creep occurs in any metal or alloy at a temperature slightly above the recrystallization temperature. At this temperature the atoms become sufficiently mobile to allow time dependent rearrangement of the metallurgical structure.

Rupture has different mechanisms dependent on the temperature. At room temperature, failure in metals occurs through the grains. This is known as “transcrystalline”. At elevated temperatures however, failure occurs around the grains. This is called “intercrystalline”. Simply put, at room temperature there is greater strength in the grain boundaries then in the grains themselves. However as temperature is increased, a point is reached where the grains and grain boundaries have equal strength. This temperature is known as the “equicohesive temperature”. Below the “equicohesive temperature”, initial deformation is elastic. Above the equicohesive temperature, the deformation is plastic.

Creep damage begins with microscopic voids in the material. In time the voids link up into fissures, and finally cracks. The voids, fissures and cracks form at grain boundaries. Creep consists of three distinct stages, primary, secondary and tertiary. Descriptions are as follows;

1. Primary Creep or First Stage: After initial elastic strain, the rate of creep decreases since the effects of strain hardening of the material are greater than the effect of annealing.
2. Secondary Creep or Second Stage: Also known as “steady state creep”. In second stage, there is a constant creep rate. The effects of strain hardening and annealing counteract each other.
3. Tertiary Creep or Third Stage: The creep rate rapidly increases. There is a drastically increased strain rate associated with a rapid deformation, terminating in stress rupture. As the cross sectional area of the material reduces (necking) the stress level is increased.

For any given alloy, a coarse grain size has greater strength at elevated temperatures than its fine grain counterpart. Relative slight changes in alloy composition can also alter creep strength appreciably. An increase in alloy content will generally result in better creep resistance. The order of selection of materials is as follows;

1. Carbon Steel
2. C-1/2 Mo
3. 1 Cr
4. 1-1/4 Cr
5. 2-1/4 Cr

Creep begins at different temperatures for different materials and the following may be used as a very general guideline;

1. Aluminum
   200°F to 400°F
2. Titanium
   600°F
3. Carbon Steel
   700°F
4. Low Alloy
   700°F

5. Stainless Steels
   800°F
6. Nickel Alloys
   800°F
7. Refractory metals & alloys
   1800°F to 2800°F

Creep behavior is such that even at constant stress and temperature, strain will develop as shown on the creep curve. According to the Norton-Bailey power law, which models primary and secondary creep, creep strain, ε_c, is more affected by stress than other factors. The creep strain increases no more than linearly with time, but increases exponentially with stress. Creep strain depends on, and is influenced by each of the following four factors:

1. Metallurgy – Alloy and grain size
2. Temperature
3. Time
4. Stress

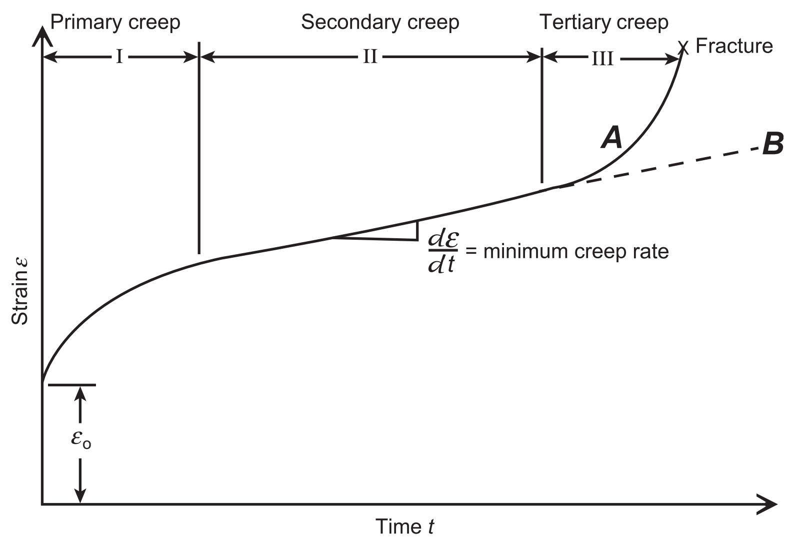


Figure 1-5. Typical creep curve showing the three steps of creep. Curve *A*, constant-load test; curve *B*, constant-stress test.

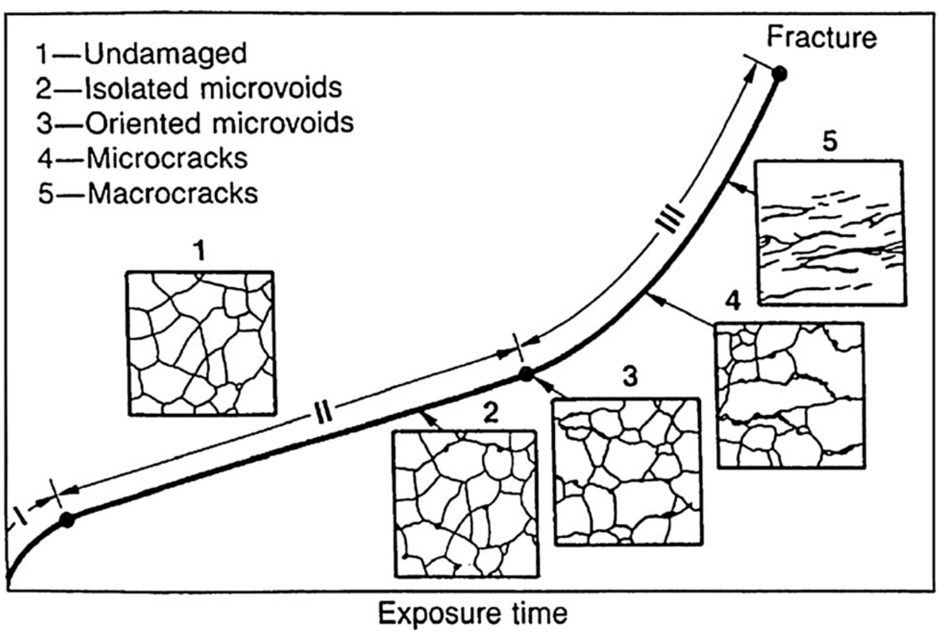


Figure 1-6. Idealized creep curve and corresponding microstructural damage.

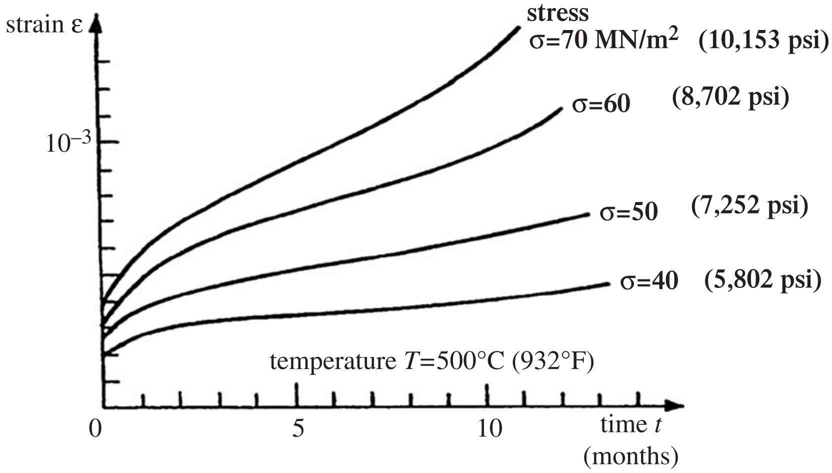
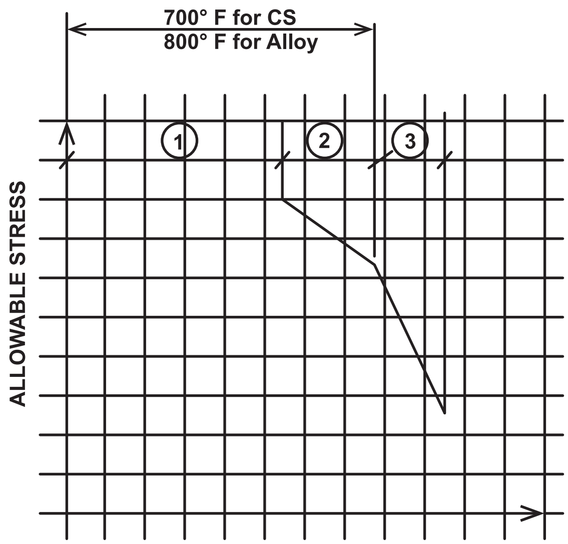


Figure 1-7. Creep curves for carbon steel (Hult, 1966).



① UTS / FS

② $2F_y/3$

③ 1% CREEP IN 100,000 HOURS

Figure 1-8. Idealized graph of allowable stress criteria.

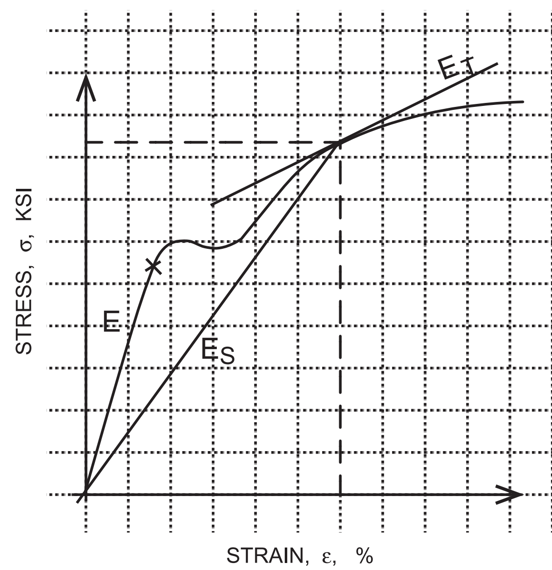


Figure 1-9. Stress-strain curve showing tangent and secant modulus.

There are several design methods utilized in the industry for analyzing creep problems. Three of the most prevalent are the Larson-Miller relation, the Norton-Bailey power law, and the MPC Omega method. The Larson-Miller relation attempts to extrapolate creep rupture data from experimental results. The Norton-Bailey power law is used to determine creep strain and creep strain rate. The MPC Omega method can be used to calculate accumulated and future strain, total damage and damage rate, creep rate, and remaining life. In addition there are several industry standards that provide guidelines, criteria, and design data for materials in the creep range. These are API-530, API-579, and WRC-443.

There are three distinct cases where creep is a factor in the design of equipment. These are as follows:

1. Continuous service in creep range
2. Creep Fatigue: Non-continuous, or intermittent service in the creep range
   a. Low-cycle fatigue
   b. High-cycle fatigue
   c. Thermal fatigue
3. Creep Buckling: Based on geometry of the component operating in the creep range.

Allowable Tensile Stress in the Creep Range

The ASME Code establishes allowable stresses at temperatures where creep and stress rupture govern based on the following:

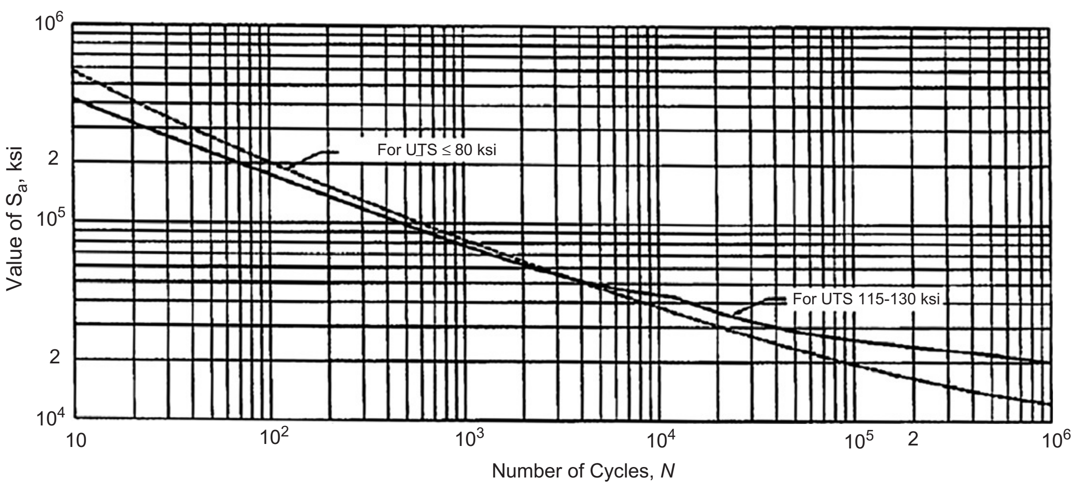
1. 100% of the average stress to yield a creep rate of 0.01%/1,000 hours,
2. 100% of the average stress to produce rupture in 100,000 hours (11.41 years),
3. 80% of the minimum stress to cause rupture in 100,000 hours.

In the ASME Code, Section II, Part D, Table 2A, the allowable tensile stress values for the materials that are governed where creep and stress rupture govern are indicated by *italics*. However this is not the sole criteria for determining if creep should be a consideration. Creep temperature limits are also given inadvertently in the fatigue curves of ASME Section VIII, Division 2. For the materials shown in Appendix 5, Fig. 5-110.1, the limit is 700°F. For materials listed in Fig. 5-110.4, the limit is 800°F. ASME Section III, Subsection NH, Appendix T also shows similar temperatures for the onset of creep.

Generally, misunderstanding arises when the design temperature is greater than these temperature limits, 700°F for carbon and low alloy materials and 800°F for alloy materials. The confusion arises because some of the materials above these temperature limits are not shown in *italics*. Many interpret this to mean that creep is not a consideration. However the magnitude of creep strain at any given temperature depends not just upon the stress level, but repeated applications and time at elevated temperature. The Code acknowledges this by limiting the temperatures when using the fatigue charts/tables.

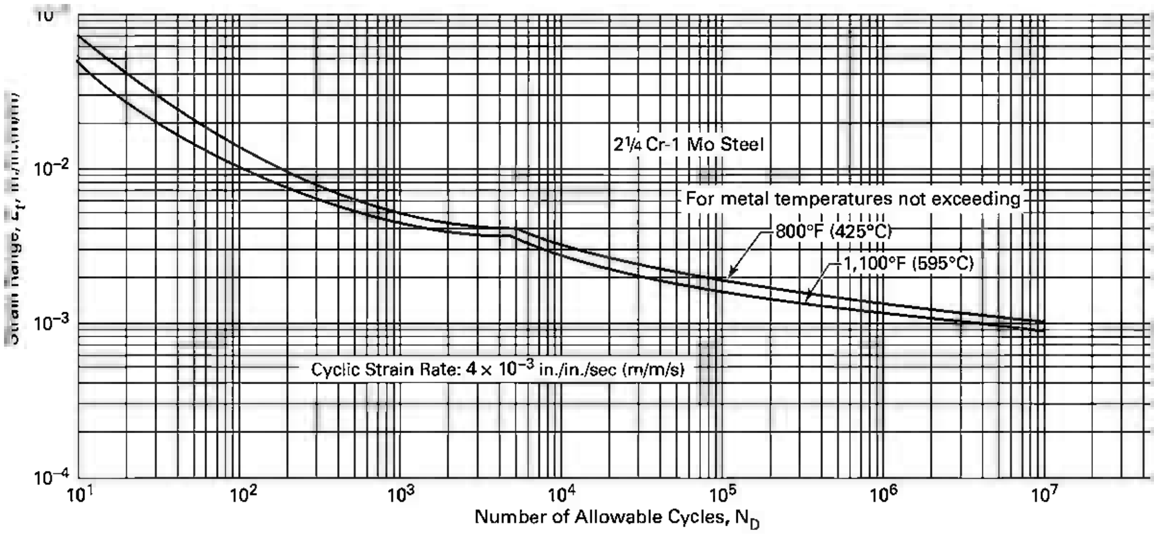
Allowable Compressive Stress in the Creep Range

The ASME Code gives temperature limitations for most groups of materials under external pressure and axial compression for cylinders, and for external pressure for spheres. The limitations are shown in graphical form in Section II, Part D, Mandatory Appendix 3, Figures



GENERAL NOTES:
(a) $E = 30 \times 10^6$ psi.
(b) Interpolate for UTS 80.0-115.0 ksi.
(c) Table 5-110.1 contains tabulated values and a formula for an accurate interpolation of these curves.

Figure 1-10. Design fatigue curves for carbon, low alloy, series 4XX, high alloy steels and high tensile steels for temperatures not exceeding 700°F and $N \leq 10^6$ (Use Fig. 5-110.1.1 for $N \geq 10^6$). (*Reprinted by permission, ASME*).



N_d, Number of Cycles [Note (1)]	ϵ_t, Strain Range (in./in.) (m/m) at Temperature	
	800°F (425°C)	900-1,000°F (480-595°C)
10^1	0.056	0.040
4×10^1	0.023	0.0163
10^2	0.013	0.0097
2×10^2	0.0094	0.0070
4×10^2	0.0070	0.0056
10^3	0.0052	0.0042
2×10^3	0.0044	0.0039
4×10^3	0.0040	0.0035
10^4	0.0032	0.00265
2×10^4	0.0026	0.00215
4×10^4	0.0023	0.00182
10^5	0.00195	0.00158
2×10^5	0.00173	0.00142
4×10^5	0.00155	0.00130
10^6	0.00137	0.00118

NOTE:
(1) Cycle strain rate: 4×10^{-3} in./in./sec (m/m/s)

Figure 1-11. Design Fatigue Strain Range, ε_t, for 2 ¼ Cr-1Mo Steel. (*Reprinted by permission, ASME.*)

3-500.1, 3-500.2 and 3-500.3. These temperatures are also a function of radius to thickness ratio. As the vacuum charts are based on short-term tensile tests, they are limited to certain temperatures since creep reduces the critical buckling stress. Since "time at temperature" is a critical factor for elevated temperature designs, it may be possible to design for short term loadings using this method.

For Code Case 2286 (now adopted into Section VIII, Division 2), the temperature limits are indicated within the said document.

At elevated temperatures, the effect of creep is to reduce the critical buckling stress. The critical buckling stress is dependent on the magnitude of the load and the time at load. Either load or time at elevated temperature will have a detrimental effect.

Additional information and design limits for elevated temperature buckling and instability are given in ASME III, Division 1, Subsection NH, Appendix T, Para T-1500.

Creep-Fatigue

For vessels in cyclic service, below the creep range, the fatigue-stress curves of Section VIII, Division 2 may be used to determine the number of acceptable cycles.

Above the creep range, a fatigue-strain curve is used. Within Section VIII, Division 2, a fatigue screening analysis is required. If the component fails the screening, a fatigue evaluation is required by the ASME Code. For vessels in the creep range, data for a fatigue evaluation is contained in Section III, Division 1, Subsection NH, Appendix T.

The methodology in Section III uses an interaction diagram, whereby a total fraction of creep damage is compared to a total fraction of fatigue damage using a material specific graph. For the creep damage, the fraction is calculated by determining the actual time at some temperature and dividing by an allowable time at some temperature and stress condition. For the fatigue damage, the fraction is calculated by determining the actual number of cycles at some at some temperature by an allowable number of cycles at that temperature. The summation of both total fractions must be less than or equal to a value as determined graphically from the interaction graph and is a function of material.

Often times, vessels are subjected to short term loadings that overstress some portion of the vessel for a very short duration. This short duration, overstressed condition, may in fact be acceptable providing the number of cycles at that overstressed condition is acceptable. A fatigue evaluation can validate this situation. The Code allowable stresses are based on continuous service up to 100,000 hours. If the vessel or component is going to be exposed to the overstressed condition for less than 100,000 hours, then the acceptability hinges on the number of allowable cycles rather than allowable stress.

The "design life" based on the allowable number of cycles must be monitored by the end user. At the end of life, the vessel must be retired, changed service or a systematic NDE program established for the continued safe operation of the equipment.

There is no general empirical or mathematical equation to relate creep strain to metallurgy or temperature since it is a combination of all these factors. One common equation for creep strain is the Norton-Bailey power law;

$$\varepsilon_c = A\,\sigma^m\,t^n$$

where,

ε_c = Creep strain, in/in
A = Material constant calculated from isochronus curves
σ = Applied stress, psi
t = Time, hours
m = Material constant calculated from isochronus curves. Always greater than 1
n = Time hardening coefficient, typically between 1/3 and 1/2.

Isochronus curves: These curves are available for many materials from ASME Section III, Division 1, Subsection NH, Class 1 for components in elevated temperature service. The word "isochronus" is derived from the words "iso", meaning equal, and "chronous" meaning time. It is a stress-strain curve for a given material, temperature, and time duration.

Creep-Buckling

One of the failure modes for cylindrical or conical under axial compression is buckling. In cylindrical components, buckling is a phenomenon that occurs when the cylinder fails in compression before the ultimate compressive strength is reached. It is a function of geometry, material properties, and is affected by imperfections in shape.

There are two kinds of failure due to buckling. The first is "general buckling" and involves bending of the axis of the cylinder resulting in instability. This is the type addressed by Euler and designed for by a "slenderness ratio" method.

The other type of buckling is a result of local instability that may or may not result in a change in the axis of the cylinder. This type is known as "local buckling" and the stability against local buckling is dependent on D/t ratios.

Creep-buckling is an issue when the cylindrical component operates in the creep range. Again, creep is a function of stress, temperature, and time.

Since the compressive loads in Section II, Part D do not account for time dependent loads, the Welding Research Council (WRC) published Bulletin 443 for calculating design limits for elevated temperature buckling based on theory as well as factors from Section III, Division 1- Subsection NH. Bulletin 443 presents equations for cylinders under axial compression as well as external pressure, and spheres under external pressure.

Creep-buckling is an issue when the cylindrical component operates in the creep range. The protection against buckling is determined by calculating a critical stress or failure point and applying a safety factor. This safety factor is a variable in the design of such components.

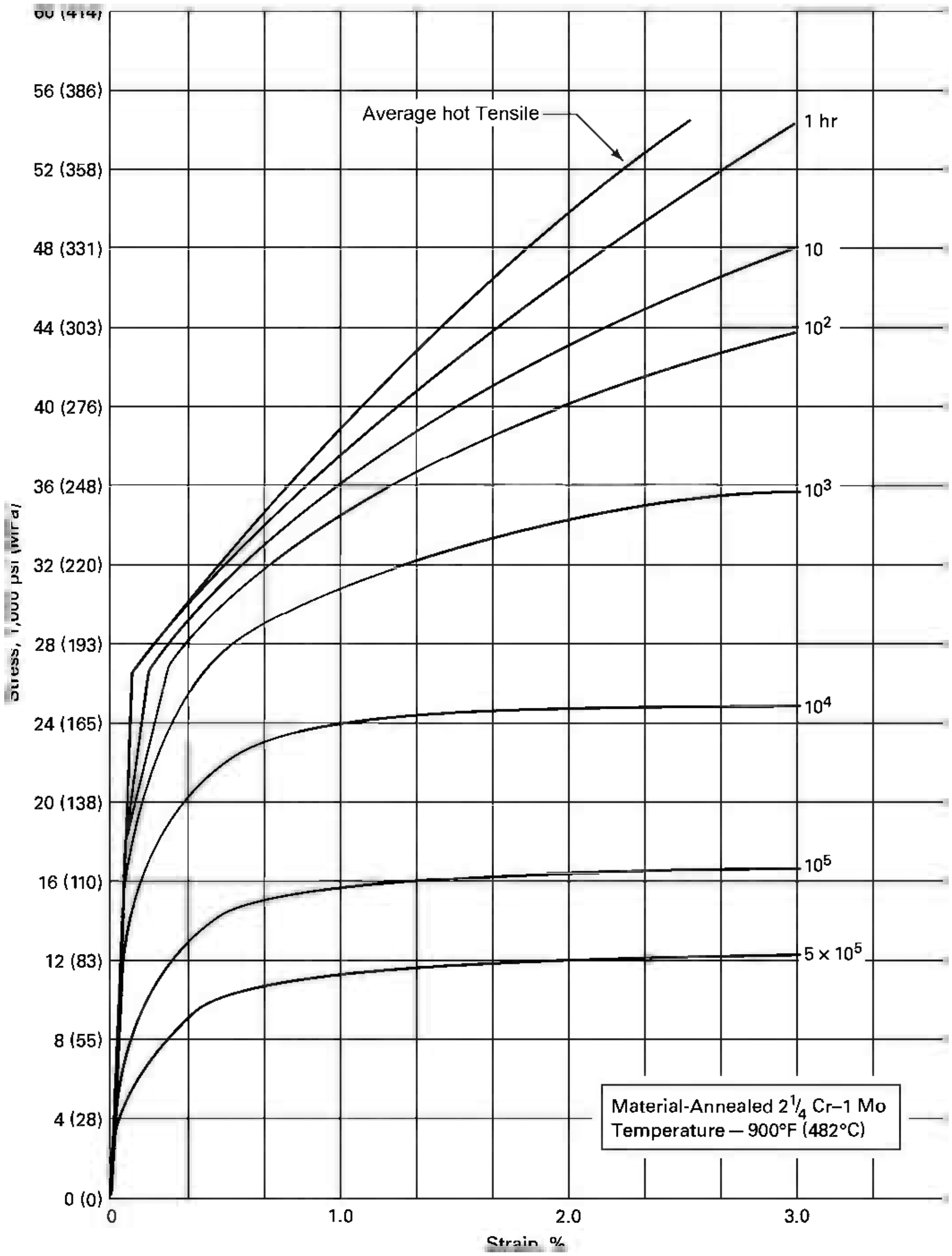


Figure 1-12. Average Isochronous Stress-Strain Curves. (*Reprinted by permission, ASME.*)

ASME Section II Part D, Mandatory Appendix 3 requires one to perform a creep-buckling analysis to obtain the allowable compressive stress at elevated temperature. Welding Research Council (WRC) Bulletin 443 has the following formula to calculate the critical buckling stress at elevated temperatures for a cylinder under axial compression:

$$\sigma_c = \left[\frac{E_t E_s}{3(1-\mu^2)}\right]^{1/2} \times \frac{t}{r_o}$$

Data

σ_c = critical buckling stress, axial, psi
t = wall thickness, in.
r_o = outside radius, in
ν = Poisson's ratio (0.3 for steel)
μ = 1/2-(1/2- ν)E_s/E
E = modulus of elasticity, psi
B = ASME Code B factor, psi
ε = strain, in/in
ε_{cr} = critical strain, in/in
α = factor for imperfection of cylinder

E_t, tangent modulus: When the critical stress exceeds the proportional limit of the material, the modulus of elasticity at that particular point on the curve is no longer valid. Instead the modulus of elasticity decreases to the local tangent value, E_t. The tangent modulus is used for buckling stresses above the proportional limit. Beyond this transition point, yield or creep will govern. The tangent modulus may be found from the isochronus stress-strain curve for that time or by using the MPC Project Omega data. The modulus is the slope of the line, tangent to the curve at the point of % strain.

E_s, secant modulus: The secant modulus is the slope of the line drawn from the origin to the point of % strain. The secant modulus may be found from the isochronus stress-strain curve for that time or by using the MPC Project Omega data.

E_e, equivalent modulus.

Per Mandatory Appendix 3-500, paragraph (c)(3), the tangent modulus, E_t, shall be used for σ_c above the proportional limit (transition point). The above formula can be used for both above and below the proportional limit.

Up to the proportional limit the σ_c formula reduces to:

$$B = \frac{AE}{2}$$

where A = 0.125 t/r_o

In the σ_c formula, let $E_tE_s = E_e^2$ where E_e is the equivalent modulus.

$$\sigma_c = \left[\frac{E_e^2}{3(1-\nu^2)}\right]^{1/2} \times \frac{t}{r_o}$$

Up to the proportional limit, $E_e = E$, $\mu = \nu = 0.3$ so that:

$$\sigma_c = \left[\frac{E^2}{3(1-0.3^2)}\right]^{1/2} \times \frac{t}{r_o}$$

$$\sigma_c = \frac{E}{[3(1-0.3^2)]^{1/2}} \times \frac{t}{r_o}$$

$$\sigma_c = (0.605)(E)\left(\frac{t}{r_o}\right) \text{ (Timoshenko)}$$

The Code formula for a cylinder under axial compression is based on an imperfection factor, $\alpha = 0.207$ and a design factor, D.F. = 2. The factor, α, accounts for theoretical (for a perfect shape) versus test results (imperfect shape):

$$B = (\alpha)\left(\frac{\sigma_c}{2}\right) = \frac{(0.207)(0.605)(E)\left(\frac{t}{r_o}\right)}{2}$$

$$B = (0.125)\left(\frac{t}{r_o}\right)\left(\frac{E}{2}\right)$$

$$B = \frac{AE}{2} \text{ (Code formula)}$$

Therefore, the σ_c formula is the continuation of the Code formula value up to the proportional limit. The steps for calculating σ_c are as follows.

1. Calculate the proportional limit, $\sigma_{prop,}$ by the isochronous curve.
2. Calculate the critical stress, σ_c, for $E = E_t = E_s$ using E from Section II, Part D. If the calculated value of σ_c is less or equal to σ_{prop} then the value of this is the value of σ_c.
3. If the calculated value of σ_c is greater than σ_{prop} then calculate the values of E_t and E_s to determine E_e and μ.
4. Use E_e and μ to calculate the value of σ_c and the corresponding value of ε_{cr}.
5. Calculate the allowable axial compressive stress using σ_c, a modified imperfection factor, α, and the appropriate design factor.

The following procedure should be used for determining the critical buckling stress for items operating in the creep range. ASME Code Vacuum charts stop at

800°F for carbon and low alloy steel. Above this temperature an alternative method must be used to determine the critical buckling stress.

This method shown below is referenced in ASME Code, Section II, Part D, Para 3-500(a) (3) which in turn refers to WRC Bulletin 443. The procedure is as follows;

Data

σ_c = critical buckling stress, psi
η = calculated per equation
E_s = secant modulus, psi
E_t = tangent modulus, psi
μ = calculated per equation
ν = Poisson's ratio, 0.3 for steel
r = radius of vessel, in.
t = thickness, in.

Sample Problem # 1, Creep Buckling (WRC – 443 Method)

Cone dimensions and properties:

Material = SA-240-304H
Design temp. = 1,150°F
Cone diameter at large end = 6.6284 in.
Cone diameter at small end = 2.6284 in.
Corroded thickness = 0.1036 in.
Length of cone = 12 in.
Equivalent length of cone = 12.166 in.
Half apex angle = 9.46°
Equivalent diameter of cone = 4.966 in.
Equivalent radius of cone = 2.483 in
Poisson's ratio = 0.3
Modulus of elasticity, E = 21,600 ksi
Tangent modulus, E_t = 8,431 ksi
Secant modulus, E_s = 17,700 ksi

$$\mu = 1/2 - (1/2 - \nu)E_s/E$$

$$\mu = 0.3361$$

$$\eta = 1 - \frac{(1 - E_t/E_s)}{1 + (1 - 4\mu^2)E_t/3E_s}$$

$$\eta = 0.5183$$

$$\sigma_c = \frac{\eta E_s t^2}{4(1 - \mu^2)r^2}$$

$$\sigma_c = 4,501 \cdot ksi$$

Cryogenic Applications

The word "cryogenics" comes from two Greek words, "kryos" meaning icy cold, and "genes" meaning to form. The term cryogenics was first used in 1875 and has come into general usage since 1955. Cryogenic temperatures are normally considered as below (−)150°F.

This fundamental characteristic of cryogenic technology has found its way into almost every major industry. Cryogenic applications are found in the steel, space, refining, welding, chemical, glass, cement, food, electronics and medical industries. In terms of volume of products, the steel and chemical industries represent the largest consumers of these products.

The principle gases include propane, ammonia, CO_2, argon, oxygen, helium, hydrogen, nitrogen, ethane, ethylene, methane and chlorine. Oxygen, nitrogen, argon and helium are mainly produced by the process of liquefaction.

There is a variety of equipment associated with separation and liquefaction of gases. These include heat exchangers, cold box equipment, distillation columns, storage vessels and compressors. In addition the storage, transportation, distribution and ultimate consumption of cryogenic fluids also require an array of mechanical equipment, piping, valves and instrumentation.

Due to the brittle nature of carbon steel at low temperature, this material is not suitable for cryogenic applications. Carbon steels can be utilized in low temperature service down to only about (−) 50°F, and then with the right testing and precautions. This is well above what would be considered a cryogenic application.

Most nonferrous metals are suitable for low temperature service. Essentially, all copper, aluminum, and high nickel based alloys remain tough and ductile in the cryogenic range. Low temperature applications also utilize low nickel alloys, such as 2-1/4 Ni and 3-1/2 Ni. Cryogenic applications utilize 9% Ni, stainless steel and aluminum. Austenitic stainless steels are capable of exposure to temperatures to absolute zero, (−) 459°F.

One of the major benefits of handling gases in liquefied form is the enormous reduction in volume resulting from the liquefaction of gas. Savings are achieved because one cubic foot of liquefied gas is equivalent to many hundreds of cubic feet of gas volume at normal pressure and temperature. Thus the handling of cryogenic fluids requires less container space.

Typical containers for cryogenic liquids are pressure vessels, spheres or tanks. In any case the normal storage device consists of a double wall, much like a thermos. Between the two walls is an insulation barrier to keep the cold within the inner vessel. Where the service requirements become critical in terms of minimizing heat loss, evacuation of the inner space becomes economically justifiable. The maintenance of this vacuum becomes one of the long term service problems. Vacuum requirements frequently begin for temperatures below (−) 200°F.

In double wall vessels and tanks the inner wall is subjected to the low temperature and the material required must correspond to that temperature. The outer vessel shell is used to support the insulation, contain the vacuum, if required, and support the total weight of the tank, contents and insulation. The outer vessel may be ordinary carbon steel, since it is a structural part only. The outer vessel does not need to be ASME Code stamped unless requested.

The inner vessel can be supported off the outer vessel in a number of ways but must minimize heat loss paths. The typical way to support the inner vessel is through a series of sway rods and support rods. These rods as well as the nozzles that penetrate both inner and outer shells, cause the most amount of heat loss. This heat loss results in boil-off of the cryogenic liquid and subsequent product loss.

If not properly designed, heat losses through the attachments to the inner shell, will account for more heat loss than the rest of the surfaces. Proper design will limit heat loss through these components to approximately 20% of the total calculated heat loss.

Many of the developments in the cryogenic field have only been possible because of the development of high efficiency insulation. Unlike most of our processes which are insulated to keep the heat in, cryogenic insulation is designed to keep the heat out.

Insulation becomes less important as the size of the vessel increases. The most important factor for heat loss is the surface area to volume ratio. As the size of the vessel increases, this ratio decreases and heat loss is less.

There are four general types of insulation used for cryogenic applications;

1. High Vacuum Insulation
2. Multiple Layer (super-insulation with vacuum)
3. Powder (perlite or silica aerogel used with no vacuum, partial vacuum, or full vacuum)
4. Rigid Foam (does not require a vacuum)

High Vacuum insulation is used for applications below (−) 200°F in double wall cryogenic vessels, as opposed to insulation between the inner and outer vessel walls. Surface reflectivity and emissivity are important factors for these applications and are a function of surface treatment, finish and cleanliness. High vacuum systems are designed to 10^{-9} torr and less. The better the surface finish and the higher emissivity, the better the insulating properties.

Specifications

A partial list of relevant Codes, Standards and Regulations is as follows;

US Std's & Codes:
Note CGA stands for "Compressed Gas Association"
CGA P-8.3
CGA P-12
CGA P-25
ASTM C740-97
European Std's & Codes:
EN 1252 CV
EN 12213
EN 12300
EN 13458 CV
EN 14197 CV
IGC DOC6/02/E
IGC DOC21/85/E/F/D
ISO/WD 21009-1
BS 5429:1976
BS 7777-4:1993

Service Considerations

The following is a list of services that require special consideration with regard to vessel design and/or material selection;

1. Cyclic Service: Fatigue
2. Lethal Service
3. Vibration Service
4. Shock Service; Thermal or Impulse loading
5. Low Temperature Service
6. Cryogenic
7. High Temperature Service
8. Creep
9. Creep –Fatigue
10. Corrosion Service
11. Contents: The contents of the vessel may require special design considerations and/or material selection. The following is a list of some of these special services;
    a. Hydrogen Service
    b. Wet H_2S Service (Sour Water)
    c. Caustic Service
    d. Ammonia Service
    e. Chloride Service
    f. Amine Service
    g. Sulfuric Acid
    h. Hydrochloric Acid
    i. Hydrofluoric Acid
    j. Polythionic Acid
12. Types of Hydrogen Service Failures;
    a. SSC: Sulfide Stress Cracking
    b. ASSC: Alkaline Sulfide Stress Cracking
    c. SZC: Soft Zone Cracking
    d. HIC: Hydrogen Induced Cracking
    e. SOHIC: Stress Oriented HIC
    f. SCC: Stress Corrosion Cracking
    g. GHSC: Galvanic Hydrogen Stress Cracking
    h. EC: Environmental Cracking
13. Types of Stress Corrosion Cracking (SCC);
    a. Chloride
    b. Polythionic
    c. Caustic
    d. MEA/DEA
    e. Sulfide
    f. Ammonia

Miscellaneous Design Considerations

TABLE 1-4
Guidelines for establishing design pressure & temperatures

Design Pressure		
Item	**Maximum Operating Pressure (PSIG)**	**Design Pressure (PSIG)**
1	Vacuum (Full or Partial)	50 PSIG Internal + Full Vacuum
2	0 to 5	50 PSIG Internal + Full Vacuum
3	6 to 35	50
4	36 to 100	Operating + 15
5	101 to 250	Operating + 25
6	251 to 500	Operating + 10 %
7	501 to 1,000	Operating + 50
8	over 1,000	Operating + 5%

Design Temperature		
Item	**Maximum Operating Temperature (°F)**	**Design Temperature (°F)**
1	Below (−) 10	Operating (−) 25
2	(−) 9 to 14	Operating (−) 20
3	15 and up	Operating + 50

TABLE 1-5
General size categories of vessels

Type	Small		Medium		Large	
HORIZONTAL / VERTICAL	DIA	<10 Ft	DIA	10 to 15 Ft	DIA	>15 Ft
	WEIGHT	<25 Tons	WEIGHT	25 to 50 Tons	WEIGHT	>50 tons
TRAYED COLUMNS	DIA	<10 Ft	DIA	10 to 15 Ft	DIA	15 to 25 Ft
	WEIGHT	<30 Tons	WEIGHT	30 to 100 Tons	WEIGHT	>100 Tons
	LENGTH	<50 Ft	LENGTH	50 to 100 Ft	LENGTH	>100 Ft
REACTORS & HIGH PRESSURE VESSELS	DIA	6 to 9 Ft	DIA	10 to 14 Ft	DIA	>15 Ft
	WEIGHT	<200 Tons	WEIGHT	200 to 500 Tons	WEIGHT	>500 Tons
	THK	2" to 4"	THK	4" to 6"	THK	>6"

Items to be Included in a User's Design Specification (UDS) for ASME VIII-2 Vessels

The following items should be included as part of the UDS.

1. Title to include Vessel name, Description and Item number
2. Project Name and Description
3. Scope
4. User
5. List of all Design Codes and Standards
6. List of all Specifications
7. List of all Design Drawings
8. List of all Standard Drawings

9. Design Basis
10. Design Data
11. Description of Loads, Load Cases, and Load Combinations (e.g. Wind, Seismic, Snow, Erection, etc.)
12. Location & Site Data
13. Fatigue Evaluation
14. Creep Evaluation
15. Design Life
16. Operating Data
17. Material Specifications
18. Inspections
19. Pressure Relief Device and Location
20. Special Service (Sour water, Hydrogen, Lethal, Cyclic, etc.)
21. Heat Treatment
22. Pressure Tests
23. Conflicts
24. Nozzle Loads
25. Design Certification

References

[1] ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, 2010 Edition, American Society of Mechanical Engineers.

[2] ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, 2010 Edition, American Society of Mechanical Engineers.

[3] Popov EP. Mechanics of Materials. Prentice Hall, Inc; 1952.

[4] Bednar HH. Pressure Vessel Design Handbook. Van Nostrand Reinhold Co; 1981.

[5] Harvey JF. Theory and Design of Modern Pressure Vessels. Van Nostrand Reinhold Co; 1974.

[6] Hicks EJ, editor. Pressure Vessels—A Workbook for Engineers, Pressure Vessel Workshop, Energy—Sources Technology Conference and Exhibition. Houston: American Society of Petroleum Engineers; January 19–21, 1981.

[7] Pressure Vessel and Piping Design, Collected Papers 1927–1959, American Society of Mechanical Engineers, 1960.

[8] Brownell LE, Young EH. Process Equipment Design. John Wiley and Sons; 1959.

[9] Roark RJ, Young WC. Formulas for Stress and Strain. 5th Edition. McGraw Hill Book Co; 1975.

[10] Burgreen D. Design Methods for Power Plant Structures. C. P. Press; 1975.

[11] Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2, American Society of Mechanical Engineers.

2
General Design

Procedure 2-1: General Vessel Formulas [1,2]

Notation

P = internal pressure, psi
D_i, D_o = inside/outside diameter, in.
S = allowable or calculated stress, psi
E = joint efficiency
L = crown radius, in.
R_i, R_o = inside/outside radius, in.
K, M = coefficients (See Note 3)
σ_x = longitudinal stress, psi
σ_ϕ = circumferential stress, psi
R_m = mean radius of shell, in.
t = thickness or thickness required of shell, head, or cone, in.
r = knuckle radius, in.

Notes

1. Formulas are valid for:
   a. Pressures <3,000 psi.
   b. Cylindrical shells where $t \leq 0.5\ R_i$ or $P \leq 0.385\ SE$. For thicker shells see Reference 1, Para. 1–2.
   c. Spherical shells and hemispherical heads where $t \leq 0.356\ R_i$ or $P \leq 0.665\ SE$. For thicker shells see Reference 1, Para. 1–3.
2. All ellipsoidal and torispherical heads having a minimum specified tensile strength greater than 80,000 psi shall be designed using S = 20,000 psi at ambient temperature and reduced by the ratio of the allowable stresses at design temperature and ambient temperature where required.

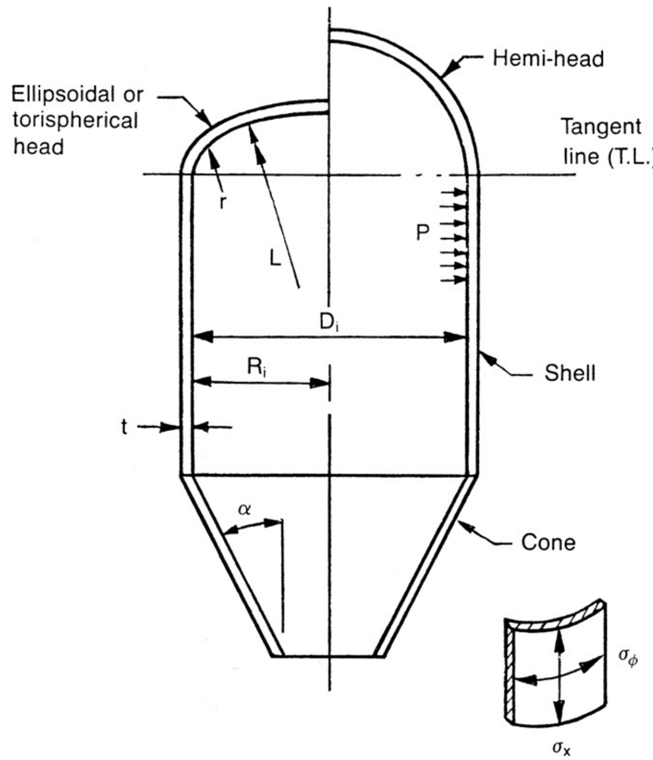


Figure 2-1. General configuration and dimensional data for vessel shells and heads.

3. Formulas for factors:

$$K = 0.167\left[2 + \left(\frac{D}{2h}\right)^2\right]$$

$$M = 0.25\left(3 + \sqrt{\frac{L}{r}}\right)$$

Table 2-1
General vessel formulas

Part	Stress Formula	Thickness, t		Pressure, P		Stress, S	
		I.D.	O.D.	I.D.	O.D.	I.D.	O.D.
Shell							
Longitudinal [1, Section UG-27(c)(2)]	$\sigma_x = \frac{PR_m}{0.2t}$	$\frac{PR_i}{2SE + 0.4P}$	$\frac{PR_o}{2SE + 1.4P}$	$\frac{2SEt}{R_i - 0.4t}$	$\frac{2SEt}{R_o - 1.4t}$	$\frac{P(R_i - 0.4t)}{2Et}$	$\frac{P(R_o - 1.4t)}{2Et}$
Circumferential [1, Section UG-27(c)(1); Section 1-1 (a)(1)]	$\sigma_\phi = \frac{PR_m}{t}$	$\frac{PR_i}{SE - 0.6P}$	$\frac{PR_o}{SE + 0.4P}$	$\frac{SEt}{R_i + 0.6t}$	$\frac{SEt}{R_o - 0.4t}$	$\frac{P(R_i + 0.6t)}{Et}$	$\frac{P(R_o - 0.4t)}{Et}$
Heads							
Hemisphere [1, Section 1-1 (a)(2); Section UG-27(d)]	$\sigma_x = \sigma_\phi = \frac{PR_m}{2t}$	$\frac{PR_i}{2SE - 0.2P}$	$\frac{PR_o}{2SE + 0.8P}$	$\frac{2SEt}{R_i + 0.2t}$	$\frac{2SEt}{R_o - 0.8t}$	$\frac{P(R_i + 0.2t)}{2Et}$	$\frac{P(R_o - 0.8t)}{2Et}$
Ellipsoidal [1, Section 1-4(c)]	See Procedure 2-7	$\frac{PD_iK}{2SE - 0.2P}$	$\frac{PD_oK}{2SE + 2P(K - 0.1)}$	$\frac{2SEt}{KD_i + 0.2t}$	$\frac{2SEt}{KD_o - 2t(K - 0.1)}$	See Procedure 2-7	
2:1 S.E. [1, Section UG-32(d)]	See Procedure 2-7	$\frac{PD_i}{2SE - 0.2P}$	$\frac{PD_o}{2SE + 1.8P}$	$\frac{2SEt}{D_i + 0.2t}$	$\frac{2SEt}{D_o - 1.8t}$	See Procedure 2-7	
100%–6% Torispherical [1, Section UG-32(e)]	See Procedure 2-7	$\frac{0.885PL_i}{SE - 0.1P}$	$\frac{0.885PL_o}{SE + 0.8P}$	$\frac{SEt}{0.885L_i + 0.1t}$	$\frac{SEt}{0.885L_o - 0.8t}$	See Procedure 2-7	
Torispherical L/r < 16.66 [1, Section 1-4(d)]	See Procedure 2-7	$\frac{PL_iM}{2SE - 0.2P}$	$\frac{PL_oM}{2SE + P(M - 0.2)}$	$\frac{2SEt}{L_iM + 0.2t}$	$\frac{2SEt}{L_oM - t(M - 0.2)}$	See Procedure 2-7	
Cone							
Longitudinal	$\sigma_x = \frac{PR_m}{2t\cos\propto}$	$\frac{PD_i}{4\cos\propto(SE + 0.4P)}$	$\frac{PD_o}{4\cos\propto(SE + 1.4P)}$	$\frac{4SEt\cos\propto}{D_i - 0.8t\cos\propto}$	$\frac{4SEt\cos\propto}{D_o - 2.8t\cos\propto}$	$\frac{P(D_i - 0.8t\cos\propto)}{4Et\cos\propto}$	$\frac{P(D_o - 2.8t\cos\propto)}{4Et\cos\propto}$
Circumferential [1, Section 1-4(e); Section UG-32(g)]	$\sigma_\phi = \frac{PR_m}{t\cos\propto}$	$\frac{PD_i}{2\cos\propto(SE - 0.6P)}$	$\frac{PD_o}{2\cos\propto(SE + 0.4P)}$	$\frac{2SEt\cos\propto}{D_i + 1.2t\cos\propto}$	$\frac{2SEt\cos\propto}{D_o - 0.8t\cos\propto}$	$\frac{P(D_i + 1.2t\cos\propto)}{2Et\cos\propto}$	$\frac{P(D_o - 0.8t\cos\propto)}{2Et\cos\propto}$

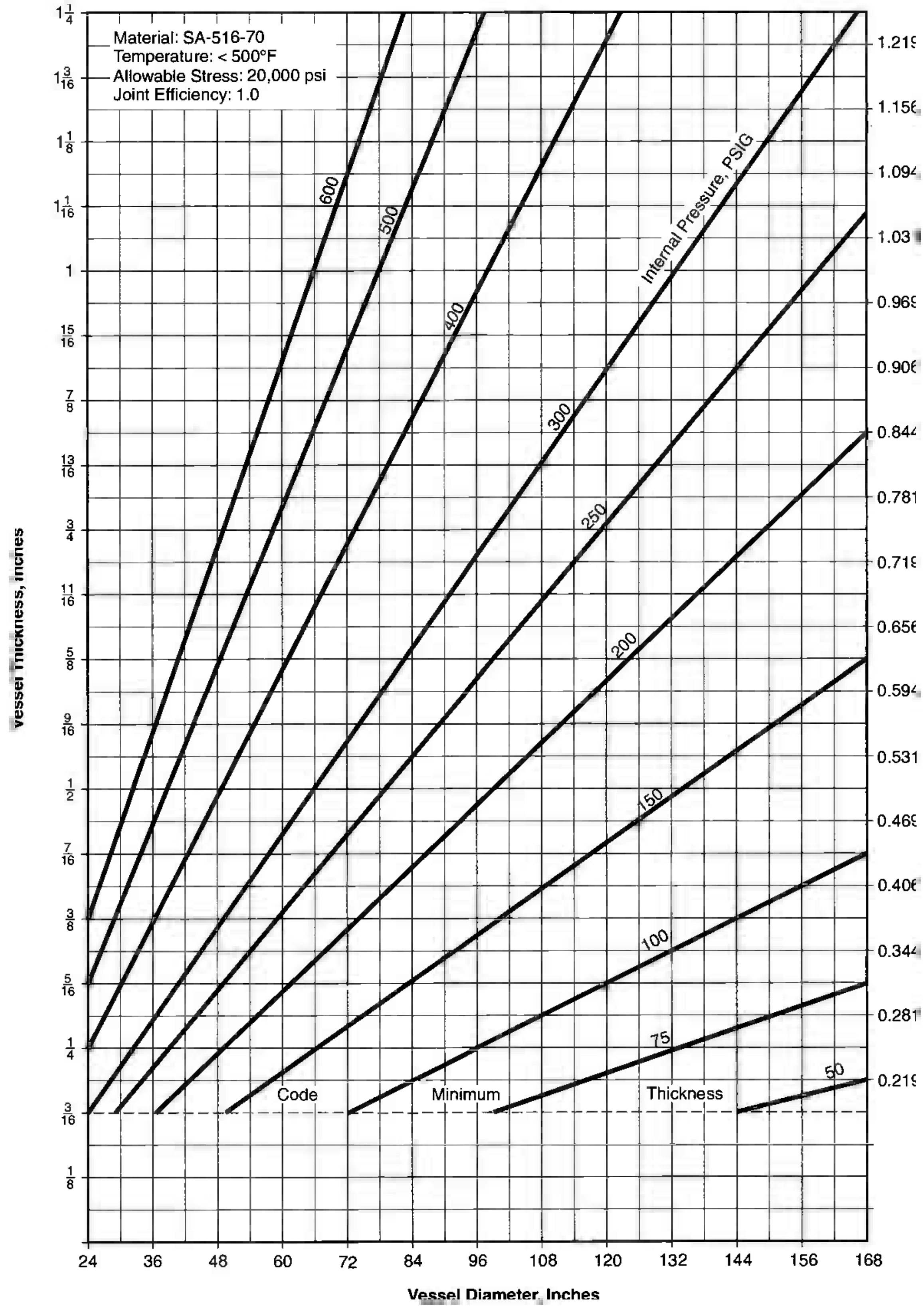


Figure 2-1a. Required shell thickness of cylindrical shell.

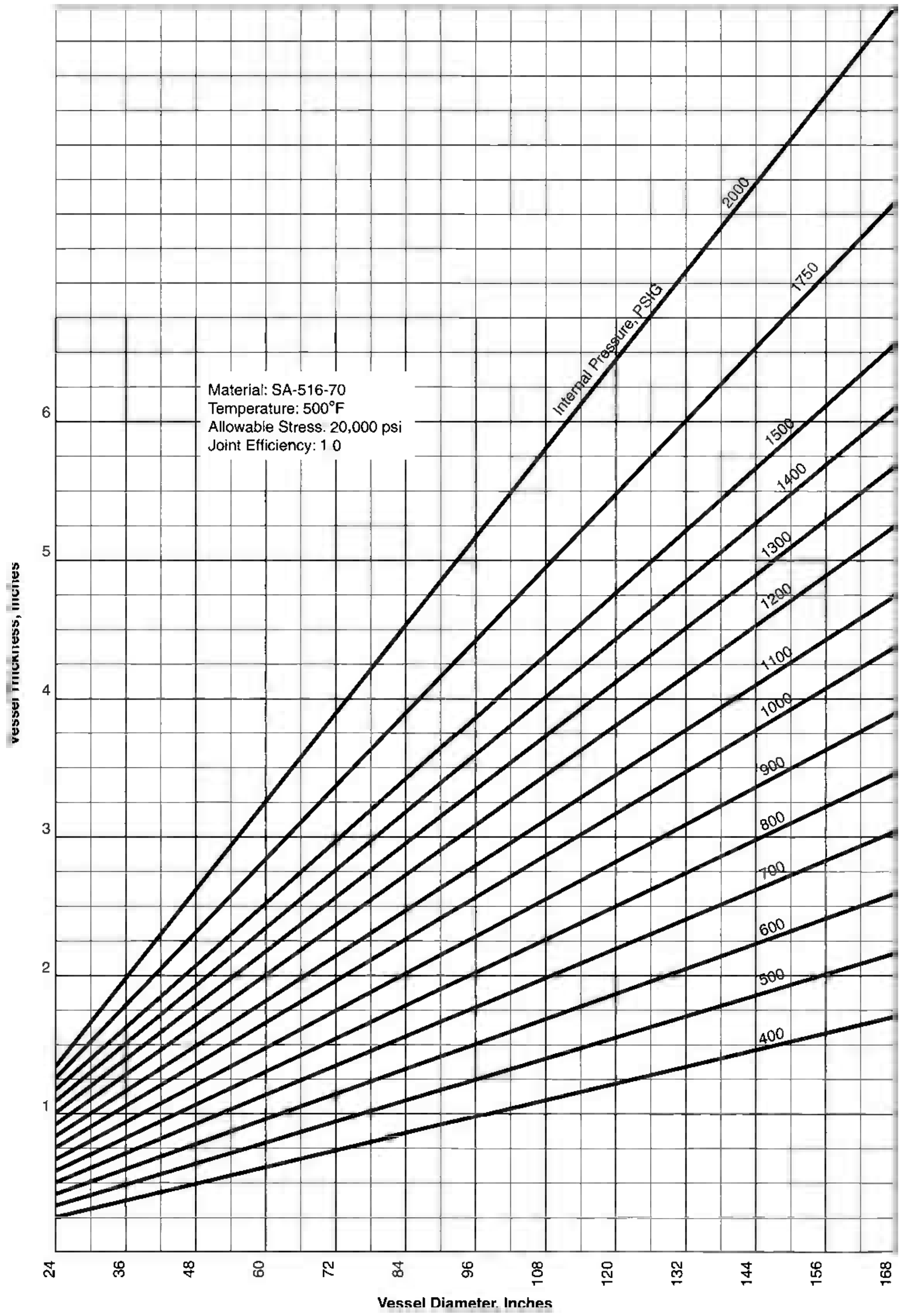


Figure 2-1a. (*continued*).

Procedure 2-2: External Pressure Design

Notation

A = factor "A," strain, from ASME Section II, Part D, Subpart 3, dimensionless
A_s = cross-sectional area of stiffener, in.^{2}
B = factor "B," allowable compressive stress, from ASME Section II, Part D, Subpart 3, psi
D = inside diameter of cylinder, in.
D_o = outside diameter of cylinder, in.
D_L = outside diameter of the large end of cone, in.
D_s = outside diameter of small end of cone, in.
E = modulus of elasticity, psi
I = actual moment of inertia of stiffener, in.^{4}
I_s = required moment of inertia of stiffener, in.^{4}
I'_s = required moment of inertia of combined shell-ring cross section, in.^{4}
L = for cylinders—the design length for external pressure, including 1/3 the depth of heads, in. For cones—the design length for external pressure (see Figures 2-1b and 2-1c), in.
L_e = equivalent length of conical section, in.
L_s = length between stiffeners, in.
L_{T-T} = length of straight portion of shell, tangent to tangent, in.
P = design internal pressure, psi
P_a = allowable external pressure, psi
P_x = design external pressure, psi
R_o = outside radius of spheres and hemispheres, crown radius of torispherical heads, in.
t = thickness of cylinder, head or conical section, in.
t_e = equivalent thickness of cone, in.
α = half apex angle of cone, degrees

Unlike vessels which are designed for internal pressure alone, there is no single formula, or unique design, which fits the external pressure condition. Instead, there is a range of options available to the designer which can satisfy the solution of the design. The thickness of the cylinder is only one part of the design. Other factors which affect the design are the length of cylinder and the use, size, and spacing of stiffening rings. Designing vessels for external pressure is an iterative procedure. First, a design is selected with all of the variables included, then the design is checked to determine if it is adequate. If inadequate, the procedure is repeated until an acceptable design is reached.

Vessels subject to external pressure may fail at well below the yield strength of the material. The geometry of the part is the critical factor rather than material strength. Failures can occur suddenly, by collapse of the component.

External pressure can be caused in pressure vessels by a variety of conditions and circumstances. The design pressure may be less than atmospheric due to condensing gas or steam. Often refineries and chemical plants design all of their vessels for some amount of external pressure, regardless of the intended service, to allow for steam cleaning and the effects of the condensing steam. Other vessels are in vacuum service by nature of venturi devices or connection to a vacuum pump. Vacuums can be pulled inadvertently by failure to vent a vessel during draining, or from improperly sized vents.

External pressure can also be created when vessels are jacketed or when components are within multichambered vessels. Often these conditions can be many times greater than atmospheric pressure.

When vessels are designed for both internal and external pressure, it is common practice to first determine the shell thickness required for the internal pressure condition, then check that thickness for the maximum allowable external pressure. If the design is not adequate then a decision is made to either bump up the shell thickness to the next thickness of plate available, or add stiffening rings to reduce the "L" dimension. If the option of adding stiffening rings is selected, then the spacing can be determined to suit the vessel configuration.

Neither increasing the shell thickness to remove stiffening rings nor using the thinnest shell with the maximum number of stiffeners is economical. The optimum solution lies somewhere between these two extremes. Typically, the utilization of rings with a spacing of 2D for vessel diameters up to about eight feet in diameter and a ring spacing of approximately "D" for diameters greater than eight feet, provides an economical solution.

The design of the stiffeners themselves is also a trial and error procedure. The first trial will be quite close if the old API-ASME formula is used. The formula is as follows:

$$I_s = \frac{0.16 D_o^3 P_x L_s}{E}$$

Stiffeners should never be located over circumferential weld seams. If properly spaced they may also double as insulation support rings. Vacuum stiffeners, if combined with other stiffening rings, such as cone reinforcement rings or saddle stiffeners on horizontal vessels, must be designed for the combined condition, not each independently. If at all possible, stiffeners should always clear shell nozzles. If unavoidable, special attention should be given to the design of a boxed stiffener or connection to the nozzle neck.

Design Procedure For Cylindrical Shells

Step 1: Assume a thickness if one is not already determined.

Step 2: Calculate dimensions "L" and "D." Dimension "L" should include one-third the depth of the heads. The overall length of cylinder would be as follows for the various head types:

W/(2) hemi-heads	$L = L_{T-T} + 0.333D$
W/(2) 2:1 S.E. heads	$L = L_{T-T} + 0.1666D$
W/(2) 100% − 6% heads	$L = L_{T-T} + 0.112D$

Step 3: Calculate L/D_o and D_o/t ratios

Step 4: Determine Factor "A" from ASME Code, Section II, Part D, Subpart 3, Fig G: Geometric Chart for Components Under External or Compressive Loadings (see Figure 2-1e).

Step 5: Using Factor "A" determined in step 4, enter the applicable material chart from ASME Code, Section II, Part D, Subpart 3 at the appropriate temperature and determine Factor "B."

Step 6: If Factor "A" falls to the left of the material line, then utilize the following equation to determine the allowable external pressure:

$$P_a = \frac{2AE}{3(D_o/t)}$$

Step 7: For values of "A" falling on the material line of the applicable material chart, the allowable external pressure should be computed as follows:

$$P_a = \frac{4B}{3(D_o/t)}$$

Step 8: If the computed allowable external pressure is less than the design external pressure, then a decision must be made on how to proceed. Either (a) select a new thickness and start the procedure from the beginning or (b) elect to use stiffening rings to reduce the "L" dimension. If stiffening rings are to be utilized, then proceed with the following steps.

Step 9: Select a stiffener spacing based on the maximum length of unstiffened shell (see Table 2-1a). The stiffener spacing can vary up to the maximum value allowable for the assumed thickness. Determine the number of stiffeners necessary and the corresponding "L" dimension.

Step 10: Assume an approximate ring size based on the following equation:

$$I = \frac{0.16 D_o^3 P_x L_s}{E}$$

Step 11: Compute Factor "B" from the following equation utilizing the area of the ring selected:

$$B = \frac{0.75 P D_o}{t + A_s/L_s}$$

Step 12: Utilizing Factor "B" computed in step 11, find the corresponding "A" Factor from the applicable material curve.

Step 13: Determine the required moment of inertia from the following equation. Note that Factor "A" is the one found in step 12.

$$I_s = \frac{\left[D_o^2 L_s \left(t + A_s/L_s\right) A\right]}{14}$$

Step 14: Compare the required moment of inertia, I, with the actual moment of inertia of the selected member. If the actual exceeds that which is required, the design is acceptable but may not be optimum. The optimization process is an iterative process in which a new member is selected, and steps 11 through 13 are repeated until the required size and actual size are approximately equal.

Notes

1. For conical sections where $\alpha < 22.5$ degrees, design the cone as a cylinder where $D_o = D_L$ and length is equal to L.
2. If a vessel is designed for less than 15 psi, and the external pressure condition is not going to be stamped on the nameplate, the vessel does not have to be designed for the external pressure condition.

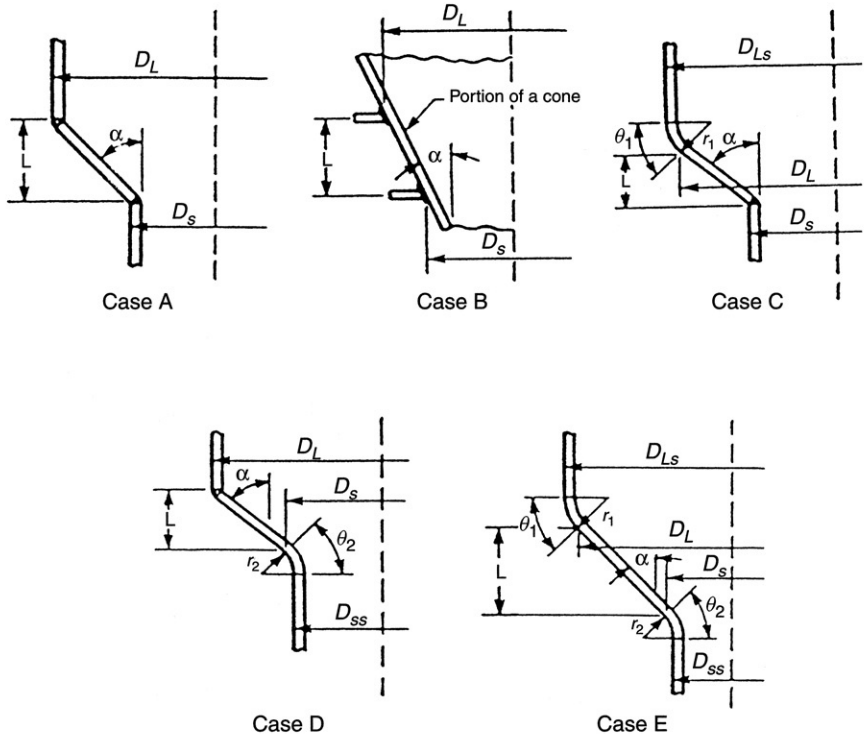


Figure 2-1b. External pressure cones 22 1/2° $< \alpha <$ 60°.

For Case B, $L_e = L$
For Cases A, C, D, E:

$$L_e = 0.5\left(1 + \frac{D_s}{D_L}\right)$$

$$t_e = t \cos \alpha$$

$$D_{L/t_e} =$$

$$L_{e/D_L} =$$

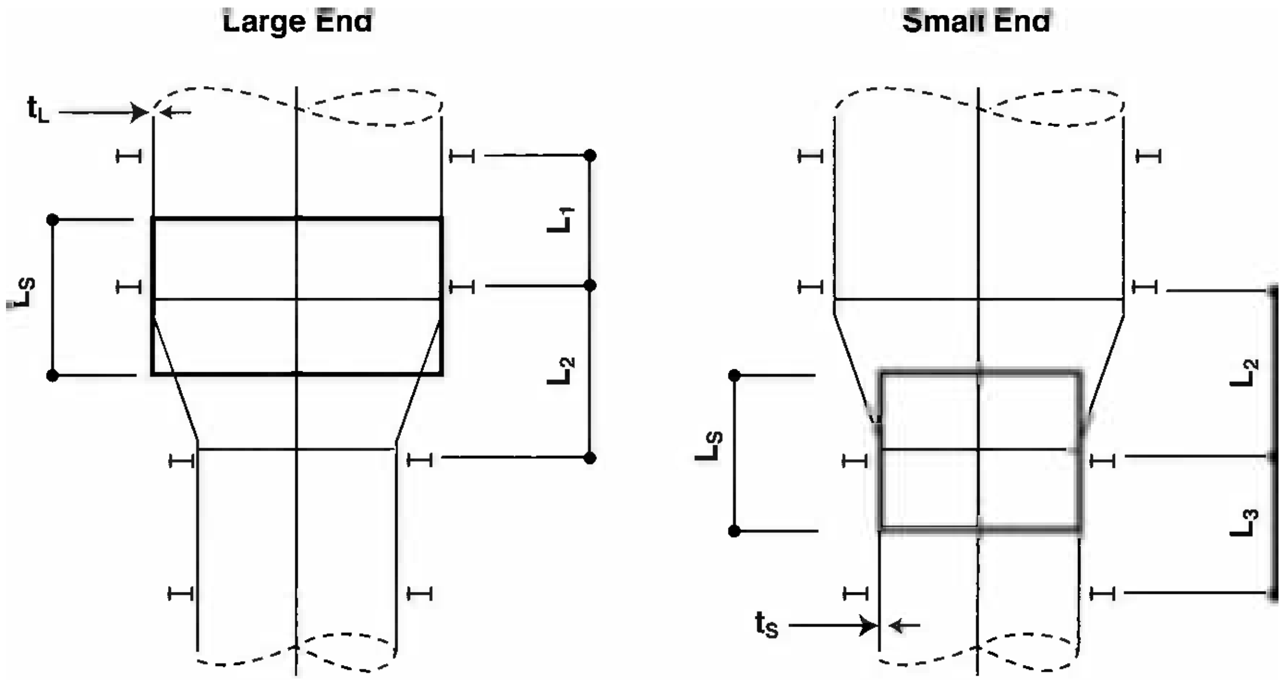


Figure 2-1c. Combined shell/cone design for stiffened shells.

Design stiffener for large end of cone as cylinder where:

$$D_o = D_L$$

$$t = t_L$$

$$L_s = \frac{L_1}{2} + \frac{L_2}{2}$$

Design stiffener for small end of cone as cylinder where:

$$D_o = D_s$$

$$t = t_s$$

$$L_s = \frac{L_3}{2} + \frac{L_2}{2}\left[1 + \frac{D_s}{D_L}\right]$$

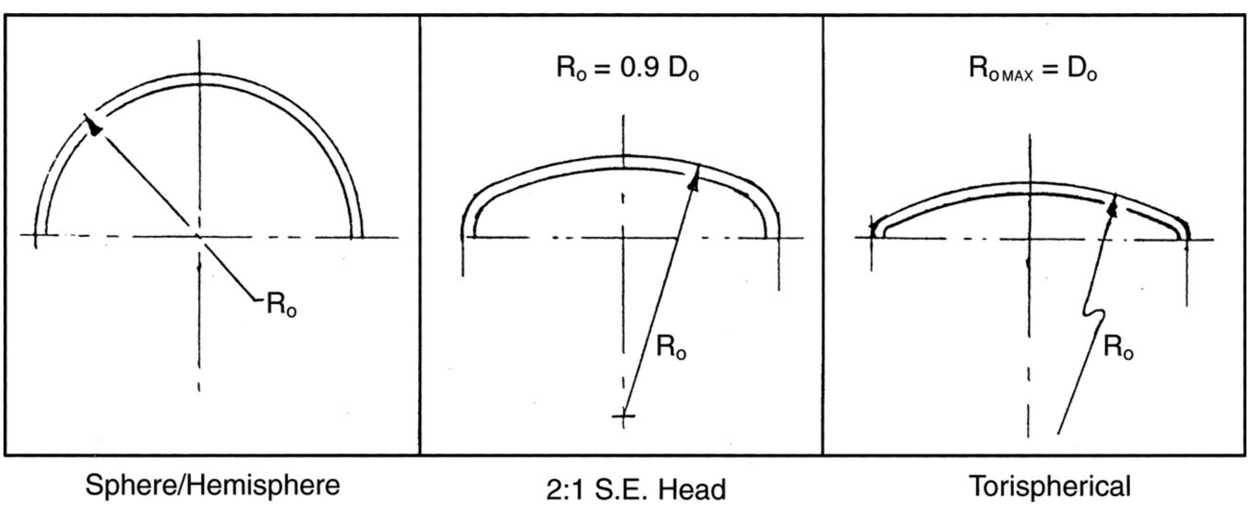


Figure 2-1d. External pressure ~ spheres and heads.

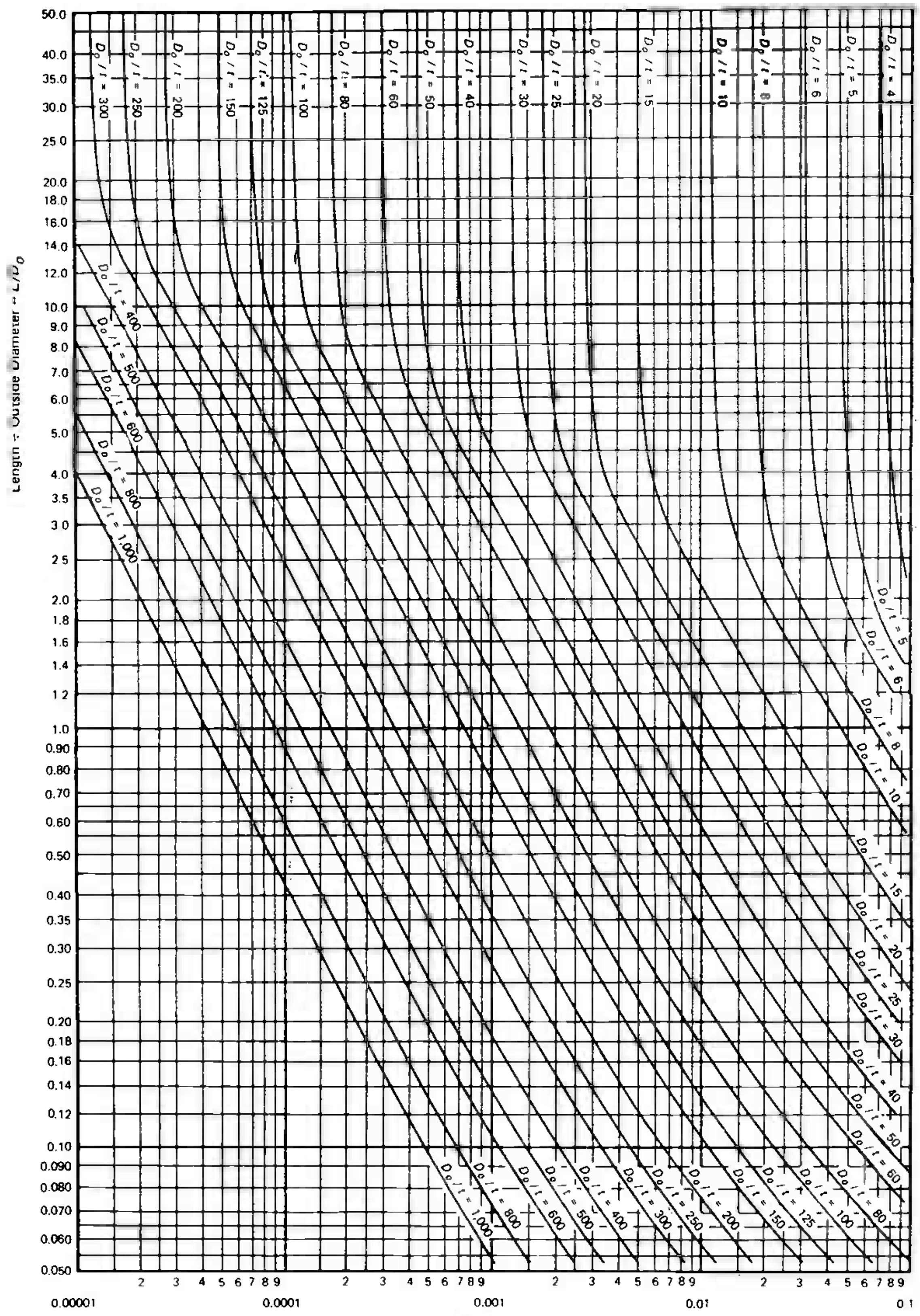


Figure 2-1e. Geometric chart for components under external or compressive loadings (for all materials). (*Reprinted by permission from the ASME Code, Section VIII, Div. 1.*)

Design Procedure For Spheres and Heads

Step 1: Assume a thickness and calculate Factor "A."

$$A = \frac{0.125t}{R_o}$$

Step 2: Find Factor "B" from applicable material chart.

$$B =$$

Step 3: Compute P_a.

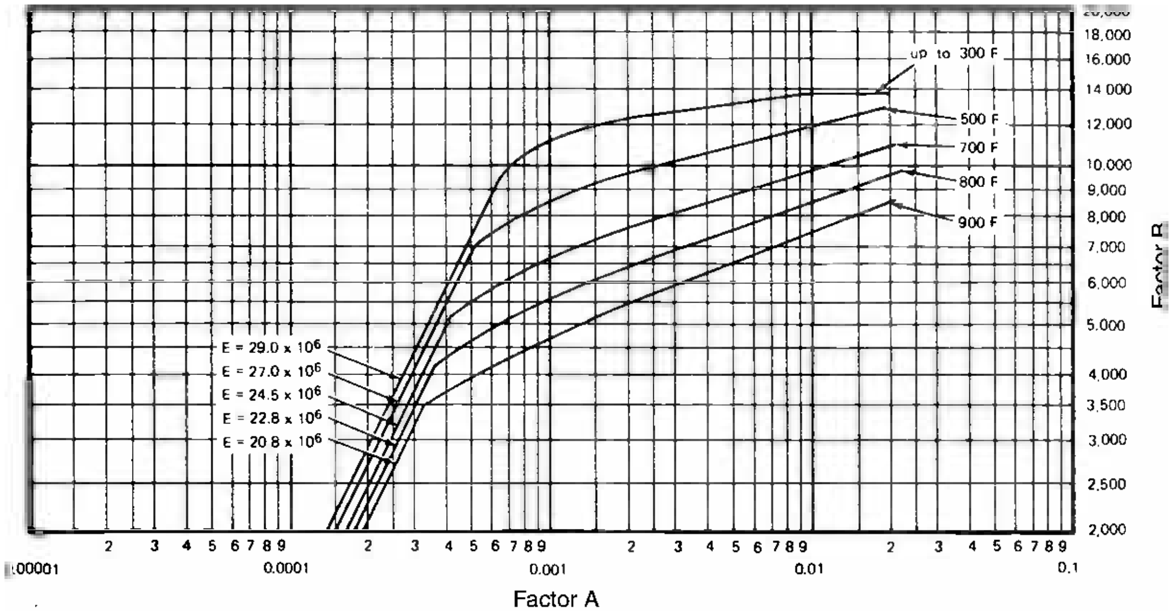


Figure 2-1f. Chart for determining shell thickness of components under external pressure when constructed of carbon or low-alloy steels (specified minimum yield strength 24,000psi to, but not including, 30,000psi). *(Reprinted by permission from the ASME Code, Section VIII, Div. 1.)*

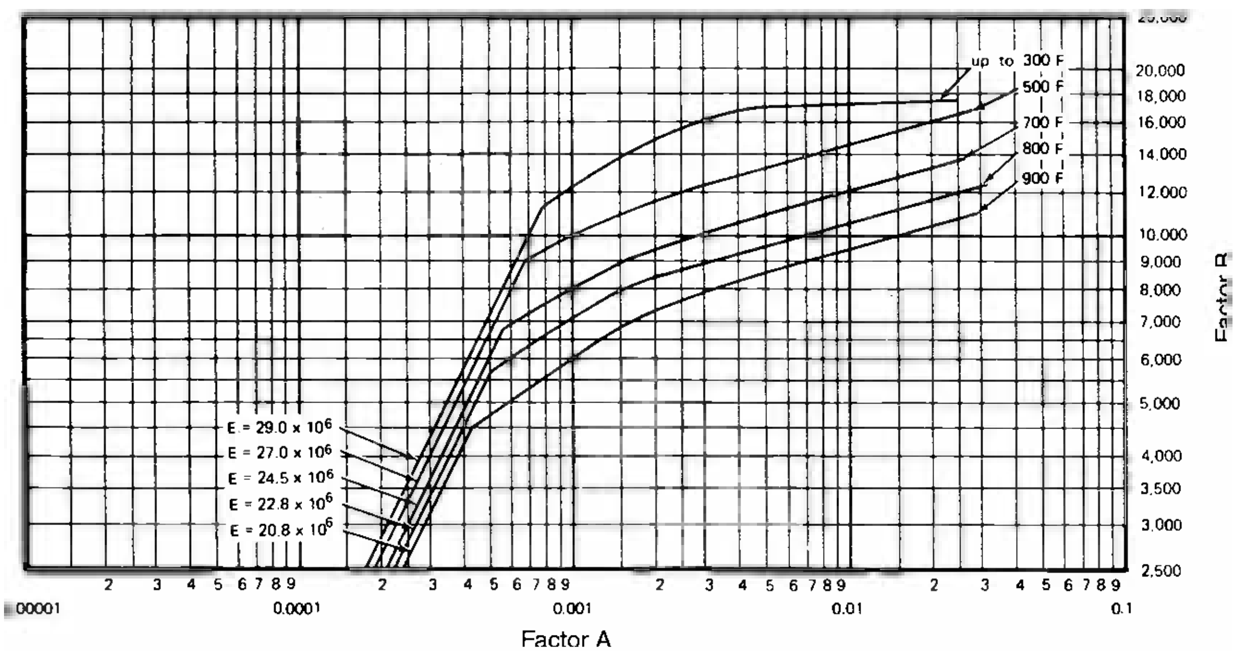


Figure 2-1g. Chart for determining shell thickness of components under external pressure when constructed of carbon or low-alloy steels (specified minimum yield strength 30,000 psi and over except materials within this range where other specific charts are referenced) and type 405 and type 410 stainless steels. (*Reprinted by permission from the ASME Code, Section VIII, Div. 1.*)

A to left of material line $P_a = \dfrac{0.0625E}{(R_o/t)^2}$

A to right of material line $P_a = \dfrac{Bt}{R_o}$

Notes

1. As an alternative, the thickness required for 2:1 S.E. heads for external pressure may be computed from the formula for internal pressure where $P = 1.67\ P_x$ and $E = 1.0$.

Table 2-1a
Maximum length of unstiffened shells

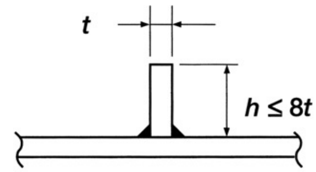
Diameter (in.)	Thickness (in.)															
	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1	1 1/16	1 1/8	1 3/16
36	204															
	∞															
42	168	280														
	313	∞														
48	142	235	358													
	264	437	∞													
54	122	203	306	437												
	228	377	∞													
60	104	178	268	381												
	200	330	499	∞												
66	91	157	238	336	458											
	174	293	442	626	∞											
72	79	138	213	302	408	537										
	152	263	396	56t	∞											
78	70	124	193	273	369	483	616									
	136	237	359	508	686	∞										
84	63	110	175	249	336	438	559									
	123	212	327	462	625	816	∞									
90	57	99	157	228	308	402	510	637								
	112	190	300	424	573	748	∞									
96	52	90	143	210	284	370	470	585	715							
	103	173	274	391	528	689	875	∞								
102	48	82	130	190	263	343	435	540	661	795						
	94	160	249	363	490	639	810	1,005	∞							
108	44	76	118	176	245	320	405	502	613	738	875					
	87	148	228	337	456	594	754	935	∞							
114	42	70	109	162	223	299	379	469	571	687	816					
	79	138	211	311	426	555	705	874	1,064	∞						
120	39	65	101	149	209	280	355	440	536	642	762	894				
	74	128	197	287	400	521	660	819	997	∞						
126	37	61	95	138	195	263	334	414	504	603	715	839	974			
	69	120	184	266	374	490	621	770	938	1,124	∞					
132	35	57	88	129	181	242	315	391	475	569	673	789	916	1,053		
	65	113	173	248	348	462	586	727	884	1,060	1,253	∞				
138	33	54	83	121	169	228	297	369	449	538	636	744	864	994		
	62	106	163	234	325	437	555	687	836	1,002	1,185	∞				
144	31	51	78	114	158	214	275	350	426	510	603	705	817	940	1,073	
	59	98	154	221	304	411	526	652	793	950	1,123	1,312	∞			
150		49	74	107	148	201	261	332	405	485	573	669	774	891	1,017	1,152
		92	146	209	286	385	499	619	753	902	1,066	1,246	1,442	∞		
156		46	70	101	140	189	248	309	385	462	546	637	737	846	966	1,095
		87	138	199	271	363	475	590	717	859	1,015	1,186	1,373	∞		
162		44	67	96	133	178	233	294	367	440	520	608	703	806	919	1,042
		83	131	189	258	342	448	562	684	819	968	1,131	1,309	1,509	∞	
	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1	1 1/16	1 1/8	1 3/16

Notes:

1. All values are in inches.
2. Values are for temperatures up to 500°F.
3. Top value is for full vacuum, lower value is half vacuum.
4. Values are for carbon or low-alloy steel ($F_y > 30{,}000$ psi) based on Figure 2-1g.

Table 2-1b
Moment of inertia of bar stiffeners

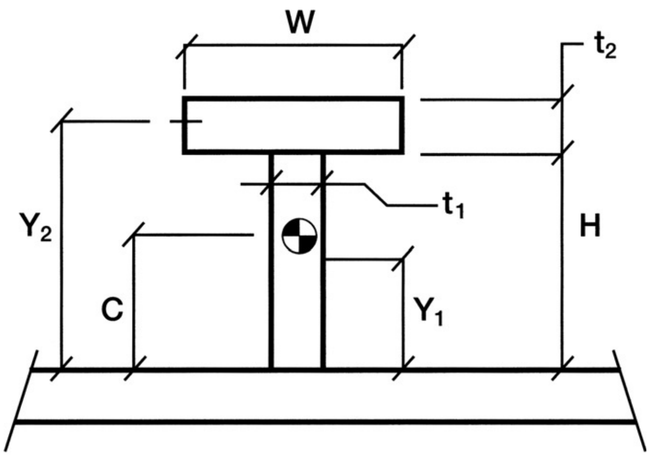
		Height, h, in.														
Thk t, in.	**Max, ht, in.**	**1**	**1½**	**2**	**2½**	**3**	**3½**	**4**	**4½**	**5**	**5½**	**6**	**6½**	**7**	**7½**	**8**
¼	2	0.020	0.070	0.167												
		0.250	0.375	0.5												
5/16	2.5	0.026	0.088	0.208	0.407											
		0.313	0.469	0.625	0.781											
⅜	3	0.031	0.105	0.25	0.488	0.844										
		0.375	0.563	0.75	0.938	1.125										
7/16	3.5		0.123	0.292	0.570	0.984	1.563									
			0.656	0.875	1.094	1.313	1.531									
½	4		0.141	0.333	0.651	1.125	1.786	2.667								
			0.75	1.00	1.25	1.50	1.75	2.00								
9/16	4.5			0.375	0.732	1.266	2.00	3.00	4.271							
				1.125	1.406	1.688	1.969	2.25	2.53							
⅝	5				0.814	1.41	2.23	3.33	4.75	6.510						
					1.563	1.875	2.188	2.50	2.813	3.125						
11/16	5.5					1.55	2.46	3.67	5.22	7.16	9.53					
						2.063	2.406	2.75	3.094	3.438	3.78					
¾	6					1.69	2.68	4.00	5.70	7.81	10.40	13.5				
						2.25	2.625	3.00	3.375	3.75	4.125	4.50				
13/16	6.5						2.90	4.33	6.17	8.46	11.26	14.63	18.59			
							2.844	3.25	3.656	4.063	4.469	4.875	5.281			
⅞	7							4.67	6.64	9.11	12.13	15.75	20.02	25.01		
								3.50	3.94	4.375	4.813	5.25	5.688	6.125		
1	8							5.33	7.59	10.42	13.86	18.00	22.89	28.58	35.16	42.67
								4.00	4.50	5.00	5.50	6.00	6.50	7.00	7.50	8.00



$$I = \frac{th^3}{12}$$

Note: Upper value in table is the moment of inertia. Lower value is the area.

Table 2-1c
Moment of inertia of composite stiffeners



$$I_1 = \frac{t_1 H^3}{12}$$

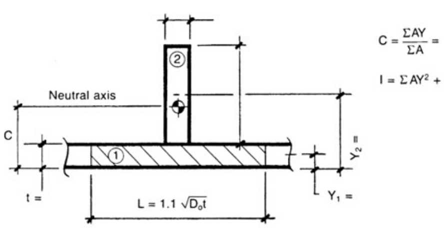
$$I_2 = \frac{w t_2^3}{12}$$

$$c = \frac{\sum A_n Y_n}{\sum A}$$

$$I = \sum A_n Y_n^2 + \sum I - C \sum A_n Y_n$$

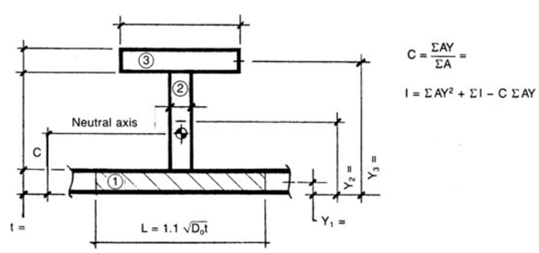
Type	H	W	t_1	t_2	$\sum A$	$\sum I$	C	I
1	3	3	0.375	0.5	2.63	0.87	2.50	2.84
2	3	4	0.5	0.5	3.50	1.17	2.50	3.80
3	4	4	0.375	0.5	3.50	2.04	3.28	6.45
4	4	5	0.5	0.625	5.13	2.77	3.41	9.28
5	4.5	5	0.5	0.5	4.75	3.85	3.57	11.25
6	5	4	0.5	0.625	5.00	5.29	3.91	15.12
7	5.5	4	0.5	0.5	4.75	6.97	4.01	17.39
8	6	5	0.5	0.625	6.13	9.10	4.69	25.92
9	6	6	0.625	0.625	7.50	11.37	4.66	31.82
10	5.5	6	0.875	0.875	10.01	12.47	4.42	37.98
11	6.5	6	0.75	0.75	9.38	17.37	4.99	48.14
12	7	6	0.625	0.75	8.88	18.07	5.46	51.60
13	8	6	0.75	1	12.00	32.50	6.25	93.25
14	8	6	1	1	14.00	43.16	5.93	112.47

Moment of Inertia of Stiffening Rings



Part	Area: A	Y	Y^2	AY	AY^2	I
1						
2						
Σ =						

Figure 2-1h. Case 1: Bar-type stiffening ring.



Part	Area: A	Y	Y^2	AY	AY^2	I
1						
2						
3						
Σ =						

Figure 2-1i. Case 2: T-type stiffening ring.

Stiffening ring check for external pressure

Ls		$B = 0.75 \dfrac{PD_o}{t + A_s/L_s}$		Moment of inertia w/o shell
t		If $B \leq 2{,}500$ psi,		$I_s = \dfrac{D_o^2 L_s(t + A_s/L_s)A}{14}$
P		$A = 2B/E$		Moment of inertia w/ shell
D_o				$I_s = \dfrac{D_o^2 L_s(t + A_s/L_s)A}{10.9}$
A_s		If $B > 2{,}500$ psi, determine A from applicable material charts		
E = modulus of elasticity				

From Ref. 1, Section UG-29.

Procedure 2-3: Properties of Stiffening Rings

Notation

A_C = Cross sectional area of composite area, in^2
A_r = Cross sectional area of ring, in^2
A_S = Cross sectional area of shell, in^2
D_m = Mean diameter of shell, in
E = Modulus of elasticity at design temperature, PSI
F_y = Minimum specified yield strength at design temperature, PSI
K_8, K_9 = Zick's coefficients
I_C = Moment of inertia of composite section, in^4
P = Internal pressure, PSIG
P_X = External pressure, PSIG
Q = Load at saddle, Lbs
R_m = Mean radius of shell, in
r = Inside radius of shell, in
R = Inside radius of shell in feet
S_{13} = Circumferential stress in shell due to load Q, PSI
S_{14} = Circumferential stress in ring due to load Q, PSI
t = Thickness, shell, in
ν = Poisson's ratio
σ_S = Stress in shell due to internal pressure, PSI
σ_T = Stress in shell due to external pressure, PSI

Derivation of Formula for Influence of Shell with Stiffening Ring

Length of shell acting with ring, L

$$.5\,L = (R_m\,t)^{1/2} / \left(3\left(1 - v^2\right)\right)^{1/4}$$

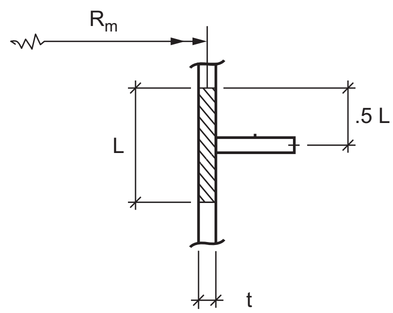


Figure 2-2. Stiffening Ring Dimensions.

For $v = .3$;

$$.5\,L = (R_m\,t)^{1/2} / 1.285$$
$$= .78(R_m\,t)^{1/2} = .55\,(D_m\,t)^{1/2}$$

$$L = 1.56\,(R_m\,t)^{1/2} = 1.1(D_m\,t)^{1/2}$$

Lateral Buckling of Stiffening Rings per CC2286

Lateral buckling is dependent on stiffener geometry. The requirements for stiffener geometry per CC2286 are as follows;

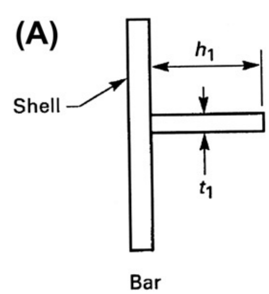


Figure 2-3. (A), (B), (C). Stiffener Geometry Per CC2286

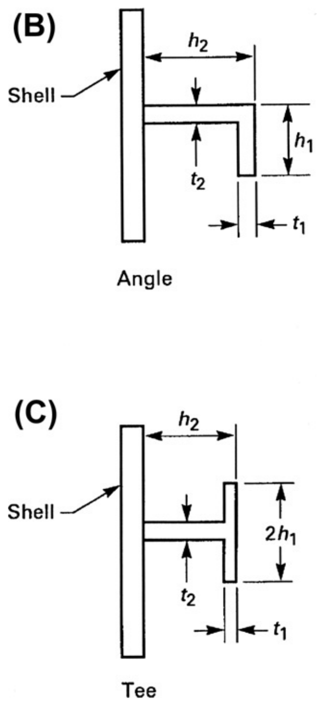


Figure 2-3. (*continued*).

Case 1: Flat bar stiffener, flange of a tee stiffener, or outstanding leg of an angle stiffener;

$$h_1/t_1 \le .375\ (E/F_y)^{1/2}$$

Case 2: Web of tee stiffener or leg of angle stiffener attached to the shell;

$$h_2/t_2 \le (E/F_y)^{1/2}$$

Shell Stresses Due to Internal or External Pressure on the Ring Section

- Stress in shell due to external pressure, σ_S

  $\sigma_S = (P_X\ L\ R_m)/A_C$
- Stress in shell due to internal pressure, σ_T

  $\sigma_T = [(P\ R_m)/t][A_S/A_C]$

Horizontal Vessel: Shell Stresses at Internal or External Ring Section Due to Load Q

Case 1: External Ring Stiffener

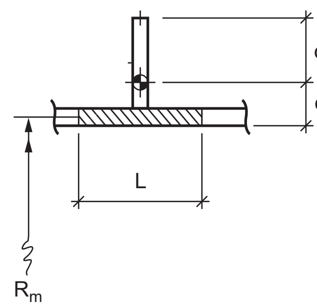


Figure 2-4. External stiffening ring dimensions.

- Stress in shell, S_{13}

  $S_{13} = (-)(K_8\ Q)/A_C + (K_9\ Q\ r\ C)/I_C$
- Stress in ring, S_{14}

  $S_{14} = (-)(K_8\ Q)/A_C(-)(K_9\ Q\ r\ d)/I_C$
- Combined stresses;

  If S_{13} is negative;

$(-)S_{13}(-)\sigma_S < .5\ F_y$

If S_{13} is positive;

$(+)S_{13}(+)\sigma_T < 1.5\ S$

Case 2: Internal Ring Stiffener

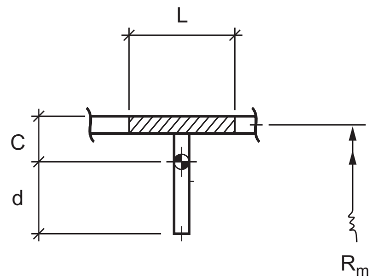


Figure 2-5. Internal stiffening ring dimensions.

- Stress in shell, S_{13}

  $S_{13} = (-)(K_8\,Q)/A_C(-)(K_9\,Q\,r\,C)/I_C$
- Stress in ring, S_{14}

  $S_{14} = (-)\ (K_8\,Q)/A_C(+)(K_9\,Q\,r\,d)/I_C$
- Combined stresses;

  $(-)\ S_{13}\ (-)\ \sigma_S < .5\ F_y$

Sample Problem Given

$Q = 267$ kips

$\theta = 150^\circ$

$A_r = 1 \times 8 = 8\ \text{in}^2$

$A_S = L\,t = 12.93(.875) = 11.31\ \text{in}^2$

$A_C = 19.31\ \text{in}^2$

$S = 20$ KSI

$F_y = 32.6$ KSI

$I_C = 135.88\ \text{in}^4$

$C = 2.27$ in

$d = 6.61$ in

$K_8 = .3$

$K_9 = .032$

$r = 78$ in

$P = 175$ PSIG

$P_X = (-)\ 15$ PSIG

$R_m = 78.5625$

$t = 0.875$ in (corr)

$$L = 1.56\ (R_m\,t)^{1/2}$$

$$= 1.56\ (78.5625\ (875))^{1/2}$$

$$= 12.93\ \text{in}$$

Case 1: External Ring

- Stress in shell, S_{13}

$$S_{13} = (-)\ (K_8\,Q)/A_C + (K_9\,Q\,r\,C)/I_C$$

$$= (-)\ .3\ (267)/19.31 + [.032\ (267)\ 78\ (2.27)]/135.88$$

$$= (-)\ 4.15 + 11.13 = (+)\ 6.98\ \text{KSI}$$

- Stress in ring, S_{14}

$$S_{14} = (-)\ (K_8\,Q)/A_C\ (-)\ (K_9\,Q\,r\,d)/I_C$$

$$= (-)\ .3\ (267)/19.31\ (-)\ [.032\ (267)\ 78\ (6.61)]/135.88$$

$$= (-)\ 4.15\ (-)\ 32.42 = (-)\ 36.57\ \text{KSI}$$

- Stress in shell due to external pressure, σ_S

$$\sigma_S = (P_X\,L\,R_m)/A_C$$

$$= [(-)\ 15\ (12.93)78.5625)]/19.31$$

$$= (-)7.89\ \text{KSI}$$

- Stress in shell due to internal pressure, σ_T

$$\sigma_T = [(P\,R_m)/t]\ [A_S/A_C]$$

$$= [175\ (78.5625)/.875]\ [11.31/19.31]$$

$$= (+)\ 9.2\ \text{KSI}$$

- Combined stresses;

  Since S_{13} is positive;

$$(+)\ S_{13}\ (+)\ \sigma_T < 1.5\ S$$

$$6.98 + 9.2 = 16.18\ \text{KSI}$$

$$< 1.5\ S = 1.5\ (20) = 30\ \text{KSI OK}$$

Case 2: Internal Ring (Same properties)

- Stress in shell, S_{13}

$$S_{13} = (-)\ (K_8\ Q)/A_C\ (-)\ (K_9\ Q\ r\ C)/I_C$$
$$= (-)\ 4.15\ (-)\ 11.13 = (-)\ 15.28\ \text{KSI}$$

- Stress in ring, S_{14}

$$S_{14} = (-)\ (K_8\ Q)/A_C\ (+)\ (K_9\ Q\ r\ d)/I_C$$
$$= (-)\ 4.15\ (+)\ 32.42 = (+)\ 28.27\ \text{KSI}$$

- Combined stresses;

$$(-)\ S_{13}\ (-)\ \sigma_S < .5\ F_y$$
$$(-)\ 15.28\ (-)\ 7.89 = (-)23.17\ \text{KSI}$$
$$<.5\ F_y = .5\ (32.6) = 16.3\ \text{KSI}$$

No Good!

Conclusion: Use an external ring or increase the properties of the internal ring.

Procedure 2-4: Code Case 2286 [1,8,21]

Nomenclature

A = cross-sectional area of cylinder, $\pi(D_o - t)t$, in^2
A_S = cross-sectional area of a ring stiffener, in^2
A_F = cross-sectional area of a large ring stiffener, in^2
D_i = inside diameter of cylinder, in.
D_o = inside diameter of cylinder, in.
D_e = outside diameter of assumed equivalent cylinder for design of cones or conical sections, in.
D_S = outside diameter at small end of cone, or conical section between lines of support, in.
D_L = outside diameter at large end of cone, or conical section between lines of support, in.
E = modulus of elasticity at design temperature, ksi
f_a = longitudinal compressive membrane stress from axial load, ksi
f_b = longitudinal compressive membrane stress from bending moment, ksi
f_h = circumferential compressive membrane stress from external pressure, ksi
f_q = longitudinal compressive membrane stress from pressure load on end of cylinder, ksi
f_v = shear stress from applied loads, ksi
f_x = $f_a + f_q$, ksi
F_{ba} = allowable longitudinal compressive membrane stress from bending moment, ksi
F_{ca} = allowable longitudinal compressive membrane stress from uniform axial compression with $\lambda_c > 0.15$, ksi
F_{aha} = allowable longitudinal compressive membrane stress from uniform axial compression with hoop compression with $0.15 < \lambda_c < 1.2$, ksi
F_{bha} = allowable longitudinal compressive membrane stress from bending moment with hoop compression, ksi
F_{cha} = allowable longitudinal compressive membrane stress from uniform axial compression with hoop compression for $0.15 < \lambda_c < 1.2$, ksi.
F_{cha} = F_{aha} when $f_q = 0$.
F_{hba} = allowable circumferential compressive membrane stress from bending moment with hoop compression, ksi
F_{he} = elastic circumferential compressive membrane failure stress from external pressure, ksi
F_{ha} = allowable circumferential compressive membrane stress from external pressure, ksi

F_{hva} = allowable circumferential compressive membrane stress from shear load with hoop compression, ksi
F_{hxa} = allowable circumferential compressive membrane stress from uniform axial compression with hoop compression, for $\lambda_c \leq 0.15$, ksi
F_{va} = allowable shear stress from applied shear load, ksi
F_{ve} = elastic shear failure stress from applied shear load, ksi
F_{vha} = allowable shear stress from shear load with hoop compression, ksi
F_{xa} = allowable compressive membrane stress from axial compression, for $\lambda_c \leq 0.15$, ksi
F_{xe} = elastic longitudinal compressive membrane failure (local buckling) stress from hoop compression, ksi
F_{xha} = allowable longitudinal compressive membrane stress from uniform axial compression with hoop compression for $\lambda_c \leq 0.15$, ksi.
FS = stress reduction factor or design factor (circumferential and longitudinal directions have separate values)
I = moment of inertia of cylinder cross section, $\pi(D_o - t)^3 t/8$, in^4
I_S' = moment of inertia of ring stiffener plus effective length of shell about its centroidal axis of combined section, in^4
I_S = moment of inertia of ring stiffener about its centroidal axis, in^4
K = coefficient to approximate end conditions for λ_c calculation (2.1 for free standing vessels supported at grade)
$L, L_1, L_2\ldots$ = distance of unstiffened vessel between lines of support, in.
$L_B, L_{B1}, L_{B2}\ldots$ = distance between bulkheads or large rings, in.
L_c = axial length of cone or conical section, in.
L_F = one-half of the sum of the distances, L_B, from one large ring to the next or line of support, in.
L_S = one-half of the sum of the distance, L, from one ring to the next of line of support, in.
L_t = overall length of vessel including 1/3 depth of each head, in.
L_u = laterally unsupported length for a free-standing vessel, for a skirt-supported vessel with no guide wires, the distance is from the top tangent line to the base of the skirt, in.
M = bending moment across cylinder cross section, in-kips
P = design external pressure, ksi
P_a = allowable external pressure
Q = uniform axial compression, kips
Q_p = axial compression as a result of external pressure, ksi
r = radius of gyration of a cylinder, $r = 0.25*(D_o^2 + D_i^2)^{1/2}$, in.
R = radius to centerline of shell, in.
R_c = radius of centroid of combined ring stiffener and effective length of shell, $R_c = R + Z_c$, in.
R_o = radius to outside of shell
s = section modulus of cylinder cross section, $\pi(D_o - t)^2 t/4$, in.
t = thickness of shell, less corrosion allowance, in.
t_c = thickness of cone, less corrosion allowance, in.
T/T = tangent to tangent length, in
V = shear force, kips
Z_c = radial distance from centerline of shell to centroid of combined section of ring and effective length of shell, in.

$$Z_c = \frac{A_S Z_S}{A_S + L_e t}$$

Z_S = radial distance from centerline of shell to centroid of ring stiffener (positive for outside rings), in.
λ_c = slenderness factor for column buckling

$$\lambda_c = \frac{K L_u}{\pi r}\sqrt{\frac{F_{xa} FS}{E}}$$

In 1998, Code Case 2286 *Alternative Rules for Determining Allowable External Pressure and Compressive*

Stresses for Cylinders, Cones, Spheres, and Formed Heads was approved for Section VIII, Divisions 1 and 2 of the ASME Code. Currently, Code Case 2286 may be applied only for Division 1, since the rules found in Code Case 2286 were absorbed into Part 4 of Division 2.

Code Case 2286 not only has rules for cylinders, cones, spheres, and formed heads, but also for the sizing of cone-cylinder junction rings, stiffening rings for external pressure, tolerances, and reinforcement for openings. When 2286 is used, it must be used for the entire vessel. Using 2286 will require a more rigorous analysis than the vacuum chart method in Division 1 and may provide a thinner shell, and fewer and/or smaller stiffening rings. In these cases, the overall cost of the vessel may be lowered as a result of the additional analysis and may be justified in many cases. An analogy may be used in comparing the use of Division 2 over Division 1 in that more design and analysis of the vessel is required in Division 2, however as a result, lower design margins are allowed. It should be noted that local jurisdictions may not always accept the use of 2286 for all equipment though this should be evaluated on a case by case basis.

Code Case 2286 assumes that shells are axisymmetric, that for unstiffened vessels the shells are the same thickness, and that for stiffened cylinders and cones the thickness between stiffeners is uniform. Additionally, capacity reduction factors (or knockdown factors) are provided for general use, but are incorporated in the allowable stress equations. Stress reduction factors (FS) must be found for each direction of loading so that the values of FS are determined independently for both the longitudinal and circumferential directions. The stress reduction factors cover elastic and inelastic buckling, and plastic collapse behavior for elements in compression.

There are several differences between Code Case 2286 and the older vacuum chart method. One difference is the upper limit of temperatures that each method has. Code Case 2286 has a limit of 800F for carbon steels found in UCS-23.1 and 800F for stainless steels in UHA-23, whereas the vacuum charts have higher limits for the more common carbon steels and stainless steels (except in a few cases). See Section II, Part D, Mandatory Appendix 3 for criteria and figures with temperature limits for various classes of materials under external pressure and axial compression. It should be noted that neither method has accounted for the effects of creep on buckling. A complete list of what 2286 covers that the vacuum chart method in Division 1 does not may be found in the 2286 document. A partial list of differences is as follows:

	Code Case 2286	Vacuum Chart Method
Carbon steel (UCS-23.1) temperature limits	800F	900F for most carbon steels
Stainless steel (UHA-23) temperature limits	800F	800F-1500F for most stainless steels
Do / t upper limit	2,000	1,000
Slenderness ratio (KL/r)	200	None
Combined loads	Yes	No
Stiffener geometry requirements	Yes	No

In discussing the differences between the two procedures, most vessels in the refining industry have D_o / t ratios far less than 1,000, so the lower limits of the vacuum charts are usually not a concern. The limitations on slenderness ratios of 200 are from AISC specifications and were based off of engineering judgment, economics in construction, and handling concerns. The evaluation of combined load cases (such as external pressure with uniform axial compression and bending) is also something that is not considered in the vacuum chart method. The use of Code Case 2286 requires the calculation of individual load cases first, and then determining if the combined load cases fulfill the acceptance criteria. This can become tedious since calculations are typically done for each weld plane.

Additionally, tolerances for cylindrical and conical shells are provided when subjected to external pressure, and uniform axial compression and bending. If the tolerances are exceeded, the allowable buckling stresses must be adjusted. In the case of vessels with large diameter over thickness ratios (approximately 300 or higher), it may be prudent to discuss shell tolerances with the fabricator to ensure that if tolerances cannot be met, the reduced allowable buckling stresses can be used for checking the design of the vessel.

The procedure for designing a simple cylindrical vessel to Code Case 2286 or Section VIII, Division 2 is to first establish geometry as well as determine shell thickness values to begin with. Typically, some thickness is established from, say, internal pressure and those thickness values are used for starting the procedure in Code Case 2286. External loads from wind and seismic design criteria must be established to complete the procedure.

The allowable primary membrane compressive stress shall be less than or equal to the maximum allowable tensile stress from Section II, Part D. The procedure is iterative. First, the allowable stresses are determined for each loading condition without being combined, and then each combined case is evaluated.

Step 1: Determine the allowable external pressure.

First, calculate M_x, C_h, and F_{he} using the equations supplied in the Code.

Next, calculate F_{ic}, which is the buckling stress that would be found if FS = 1 in the allowable stress equations. For external pressure, the equations are associated by the F_{he}/F_y ratios.

Then, calculate FS, where the equations are associated with the value of F_{ic} as compared to F_y.

Calculate F_{ha} with the calculated F_{ic} and the calculated FS value. This is the allowable stress value to use in the thickness calculations, assuming no other loadings are occurring.

Finally, determine the allowable pressure.

If the allowable pressure is not sufficient for design conditions, increase the thickness or add ring stiffeners.

Step 2: Determine the allowable longitudinal stress due to axial compression.

First, calculate M_x, c(bar), C_x, and F_{xe} using the equations supplied in the Code.

Next, calculate F_{ic}, which is the buckling stress that would be found if FS = 1 in the allowable stress equations. For uniform axial compression, the equations are associated by the D_o/t ratios.

Then, calculate FS, where the equations are associated with the value of F_{ic} as compared to F_y.

Calculate F_{xa} with the calculated F_{ic} and the calculated FS value.

Calculate λ_c to see if the vessel is subject to column buckling. If it is not then the allowable longitudinal stress due to axial compression is F_{xa}.

If it is, then F_{ca} must be calculated, which will be a reduced value of F_{xa}.

Finally, determine if the actual stress is less than or equal to the allowable stress due to uniform axial compression.

Step 3: Determine the allowable longitudinal stress due to bending moment.

First, using the D_o/t ratio and the $\gamma = F_y D_o/ET$ ratio, determine the appropriate of the four equations to use for F_{ba}.

Next, calculate F_{ic}, which is the buckling stress that would be found if FS = 1 in the allowable stress equations.

Then, calculate FS, where the equations are associated with the value of F_{ic} as compared to F_y.

Calculate F_{ba} with the calculated F_{ic} and the calculated FS value.

Finally, determine if the actual stress is less than or equal to the allowable stress due the bending moment.

Step 4: Determine the allowable in-plane shear stress due to a shear force:

First, calculate α_v, M_x, C_v, F_{ve}, and η_v using the equations supplied in the Code.

Next, calculate F_{ic}, which is the buckling stress that would be found if FS = 1 in the allowable stress equations.

Then, calculate FS, where the equations are associated with the value of F_{ic} as compared to F_y.

Calculate F_{va} with the calculated F_{ic} and the calculated FS value.

Finally, determine if the actual stress is less than or equal to the allowable stress due the shear force. Note that the shear stress may be calculated at various angles around the circumference.

Step 5: Determine the interaction of the stresses under combined loads.

Step 6: Determine the size of the ring stiffener if stiffeners are needed. Ring stiffeners may be sized as either small rings or large rings, as either ring is considered a line of support.

There are several examples using the incorporated Code Case 2286 in Division 2 that may be used as reference in the document ASME PTB-3 Section VIII Division 2 Example Problem Manual.

Procedure 2-5: Design of Cones

Notation

A = factor A from ASME
A_r = Excess area available in junction without a ring, in^2
A_S = Cross sectional area of ring, in^2
A_{eL} = Effective area of reinforcement, large end, in^2
A_{eS} = Effective area of reinforcement, small end, in^2
A_{rL} = Area of reinforcement required at large end, in^2
A_{rS} = Area of reinforcement required at small end, in^2
A_{TL} = Equivalent area of shell, cone, and ring at large end, in^2
A_{TS} = Equivalent area of shell, cone, and ring at small end, in^2
B = Factor B from ASME, PSI
Ca = Corrosion Allowance, in
D_e = OD of equivalent cylinder,in
E_1, E_2 = Joint efficiency in shell or cone
E_s, E_c, E_r = Modulus of elasticity of shell, cone or ring, psi
$f_{1\text{-}4}$ = Axial load per unit circumference excluding pressure, Lbs/in-circ
F_L = Longitudinal load at large end if this is a line of support, Lbs/in
F_S = Longitudinal load at small end if this is a line of support, Lbs/in
I = Available moment of inertia, ring only, in^4
I_S = Required moment of inertia, in^4
I' = Available moment of inertia of combined shell-ring section, in^4
I_S' = Required moment of inertia of combined shell-ring section, in^4
K = Ratio defined herein
L = Length of cone along axis of cone, in
L_a = Maximum distance along shell from junction that can be included as part of reinforcement, in
L_b = Maximum distance along shell from junction to centroid of ring, in
L_C = Length of cone along shell, in
L_{ce} = Length of equivalent cylinder for external pressure, in
L_L = Distance along shell at large end, from junction to first stiffening element, in
L_S = Distance along shell at small end, from junction to first stiffening element, in
M = Factor at large end if this is a LOS
M_1, M_2 = Moment at Elev 1 or 2, in-Lbs
N = Factor at small end if this is a LOS
P = Internal pressure, psig
P_x = External pressure, psig
$Q_{L1\text{-}4}$ = Axial load per unit circumference including pressure, Lbs/in-circ
t_{S1}, t_{S2} = Thickness required, shell, in
t_{rc} = Thickness required, cone, in
t_e = Thickness, equivalent cylinder for large end, in
$t_{1\text{-}2}$ = Thickness of shell, small/large end, in
t_c = thickness of cone, in
t_{rx} = Thickness required for cone external pressure, in
S_s, S_c, S_R = Allowable stress of shell, cone and ring respectively, psi
W_1, W_2 = Weights at Elev 1 or 2, Lbs
X, Y = Factors for cones where $\alpha > 30°$
X_1, Y_1 = ASME ratios defined herein
Δ = From Tables 2-2, 2-3 and 2-4, degrees
σ_X = Membrane longitudinal stress plus discontinuity longitudinal stress due to bending, PSI
σ_ϕ = Membrane hoop stress plus average discontinuity hoop stress, PSI
LOS = Line of Support

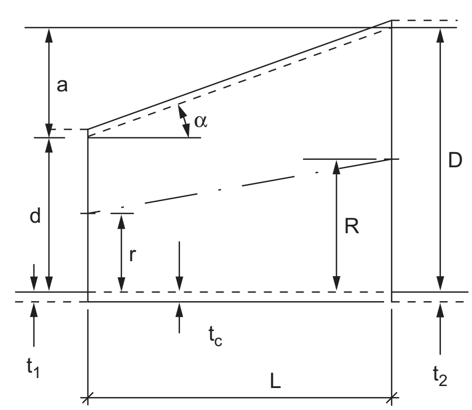


Figure 2-6. Eccentric cone.

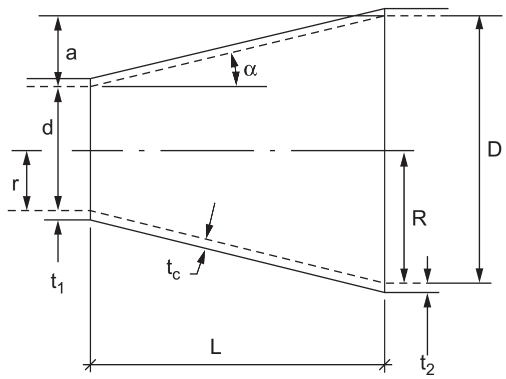


Figure 2-7. Concentric cone.

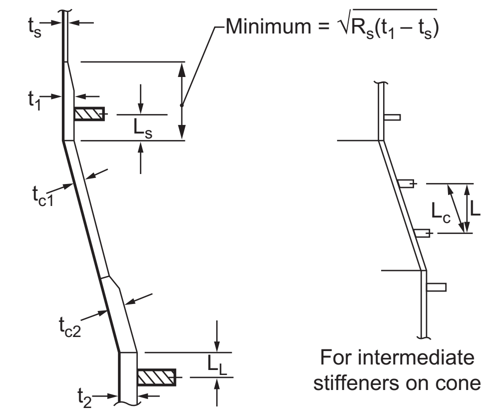


Figure 2-8. Dimensions of cone-cylinder intersections.

Geometry

FOR ECCENTRIC CONE…

$a = D - d$

$\tan\alpha = a/L$

$L = a\tan\alpha$

FOR CONCENTRIC CONE…

$a = .5\,(D - d)$

$\tan\alpha = a/L$

$L = a\tan\alpha$

$$L_C = \left[L^2 + (R - r)^2\right]^{1/2}$$

1.0. Design Cone for Internal Pressure

1.1. Thickness Required

- Required thickness of cone, Internal Pressure, $\alpha \le 30°$;
  Small End;

  $t_{C1} = (P\,r)/(\cos\alpha\,(S_C\,E_2 - .6P))$

Large End;

  $t_{C2} = (P\,R)/(\cos\alpha(S_C\,E_2 - .6P))$

- Required thickness of Shell;

Small End;

  $t_{S1} = (P\,r)/(S_S\,E_1 - .6P)$

Large End;

  $t_{S2} = (P\,R)/(S_S E_1 - .6P)$

1.2. Reinforcement Required

- Values of X_1 and Y_1
  X_1 = Smaller of $S_S\,E_1$ or $S_C\,E_2$
  Y_1 = Greater of $S_S\,E_S$ or $S_C\,E_C$

Table 2-2
Values of Δ, degrees - large end, α ≤ 30 degrees, internal pressure

P/X_1	Δ
.001	11
.002	15
.003	18
.004	21
.005	23
.006	25
.007	27
.008	28.5
.009	30

Large End

- $P/X_1 =$
  Δ = From Table 2-2
  If $\Delta < \alpha$ then reinforcement is required
  If $\Delta \ge \alpha$ then no reinforcement is required

If reinforcement is required follow the following steps:

- Calculate ratio, K

  $K = Y_1/(S_R\,E_R) > 1$

  $K = 1$ if additional area of reinforcement is not required

- Area of reinforcement required at large end of cone, A_{rL}

Case 1: External Loads are Not Included

$$A_{rL} = \left[\left(P\,R^2\,K\right)/2\,X_1\right]\left[1-\left(\Delta/\alpha\right)\right]\tan\alpha$$

Case 2: External Loads are Included

$$A_{rL} = [(K\,Q_L\,R)/X_1]\,[1-(\Delta/\alpha)]\tan\alpha$$

From the worksheet, select the largest value of Q_L for the large end in tension.

Reinforcement Available

- Effective area available at large end, A_{el}

  $$A_{el} = \left(t_2 - t_{S2}\right)(R\,t_2)^{1/2} + \left(t_C - t_r\right) \times (R\,t_C/\cos\alpha)^{1/2}$$

- Area required at large end, A_r

  $$A_r = A_{rL} - A_{eL}$$

If A_r is negative, the design is adequate as is. If A_r is positive, then either a ring must be added or the area of section increased. A_r is the net area required for the ring only!

- If a ring is required, the maximum distance to centroid of ring, L_b

  $$L_b = .25\,(R\,t_2)^{1/2}$$

- If the shell is made thicker to provide more area for reinforcement, then the maximum length of shell included in the reinforcement shall be;

  $$L_a = (R\,t_2)^{1/2}$$

Small End

Table 2-3
Values of Δ, degrees - small end, α ≤ 30 degrees, internal pressure

P/X_1	Δ
.002	4
.005	6
.010	9
.02	12.5
.04	17.5
.08	24
.1	27
.125	30

- $P/X_1 =$
  $\Delta =$ From Table 2-3
  If $\Delta < \alpha$ then reinforcement is required
  If $\Delta \geq \alpha$ then no reinforcement is required

If reinforcement is required follow the following steps:

Case 1: External Loads are Not Included:

$$A_{rS} = \left[\left(P\,r^2\,K\right)/2\,X_1\right]\left[1-\left(\Delta/\alpha\right)\right]\tan\alpha$$

Case 2: External Loads are Included

$$A_{rS} = [(K\,Q_L\,r)/X_1]\,[1-(\Delta/\alpha)]\tan\alpha$$

From the worksheet, select the largest value of Q_L for the small end in tension.

Reinforcement Available

- Effective area available at large end, A_{eS}

  $$A_{eS} = .78\left(t_1 - t_{S1}\right)(r\,t_1)^{1/2} + ((t_C - t_r)/\cos\alpha)^{1/2}$$

- Area required at large end, A_r

  $$A_r = A_{rS} - A_{eS}$$

If A_r is negative, the design is adequate as is. If A_r is positive, then either a ring must be added or the area of section increased. A_r is the net area required for the ring only!

- If a ring is required, the maximum distance to centroid of ring, L_b

  $$L_b = .25\,(r\,t_1)^{1/2}$$

- If the shell is made thicker to provide more area for reinforcement, then the maximum length of shell included in the reinforcement shall be;

  $$L_a = (r\,t_1)^{1/2}$$

2.0. Design Cone for External Pressure

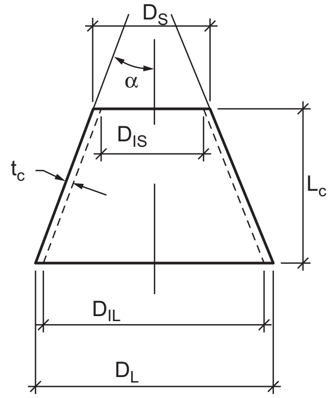


Figure 2-9. Dimensions of concentric cone for external pressure.

2.1. Thickness Required

For $\alpha \leq 22.5°$

- Design the cone as a cylinder where $D_o = D_L$ and the length is equal to L.
- t_{rx} = ________

For α > 22.5°

- Determine half apex angle, α

  $a = \arctan [.5(D_{iL} - D_{iS})/L]$
- Length of equivalent cylinder, L_{ce}

  $L_{ce} = L/\cos \alpha$
- Diameter of equivalent cylinder, D_e

  $D_e = .5 [(D_L + D_S)/\cos \alpha]$
- Thickness of equivalent cylinder, t_e

  $t_e = t_C \cos \alpha$
- Calculate ratios;

  $L_{ce}/D_e =$

  $D_e/t_e =$
- Design the cone as a cylinder with the properties calculated above
- t_{rx} = ________

2.2. Reinforcement Required

LARGE END

- Values of X_1 and Y_1

  X_1 = Smaller of $S_S E_1$ or $S_C E_2$

  Y_1 = Greater of $S_S E_S$ or $S_C E_C$

Table 2-4
Values of Δ, degrees - large end, α ≤ 60 degrees, external pressure

P_X/X_1	Δ
0	0
.002	5
.005	7
.010	10
.02	15
.04	21
.08	29
.1	33
.125	37
.15	40
.2	47
.25	52
.3	57
.35	60

Note: If $P_X / X_1 > .35$ Use $\Delta = 60$

- $P_X/X_1 =$

  Δ = From Table 2-4

  If $\Delta < \alpha$ then reinforcement is required

  If $\Delta \geq \alpha$ then no reinforcement is required

If reinforcement is required follow the following steps:

- Calculate ratio, K

  $K = Y_1/(S_R E_R) > 1$

  $K = 1$ if no additional area is required

Case 1: External Loads are Not Included

$$A_{rL} = \left[\left(P_X R^2 K\right)/2 X_1\right] \left[1 - .25\left(P_X R\right) \left(\Delta/\alpha\right)\right]$$

Case 2: External Loads are Included

$$A_{rL} = [(K Q_L R \tan \alpha)/X_1] [1 - .25$$
$$((P_X R - Q_L)/Q_L) (\Delta/\alpha)]$$

From the worksheet, select the largest value of Q_L for the large end in compression.

Reinforcement Available

- Effective area available at large end, A_{eL}

  $A_{eL} = .55 (D_L t_2)^{1/2} \left(t_2 + t_C/\cos \alpha\right)$

- Area required at large end, A_r

$$A_r = A_{rL} - A_{eL}$$

If A_r is negative, the design is adequate as is. If A_r is positive, then either a ring must be added or the area of section increased. A_r is the net area required for the ring only!

- If a ring is required, the maximum distance to centroid of ring, L_b

$$L_b = .25\,(R\,t_2)^{1/2}$$

- If the shell is made thicker to provide more area for reinforcement, then the maximum length of shell included in the reinforcement shall be;

$$L_a = (R\,t_2)^{1/2}$$

If the Large End is a Line of Support (LOS)

1. Assume size of ring;
   Size = _____
   A = ______
   I = _______
2. Determine length of cylinder acting with cone, L_L
   L_L = ______
3. Determine length of cone, L_C

$$L_C = \left[L^2 + (R - r)^2\right]^{1/2}$$

4. Calculate equivalent area of cylinder, cone and ring, A_{TL}

$$A_{TL} = .5\,(L_L\,t_2) + .5\,(L_C\,t_C) + A_S$$

5. Calculate Value M

$$M = .5\,(-R\tan\alpha) + .5\,L_L + \left[\left(R^2 - r_2\right)/ \times (3\,R\tan\alpha)\right]$$

6. Calculate Value F_L

$$F_L = P_X\,M + f_n\,\mathrm{Tan}\,\alpha$$

   Choose f_n as worst value, compression
7. Calculate Factor B

$$B = .75\,[(F_L\,D_L)/A_{TL}]$$

8. Find Factor A from applicable material curve or calculate as follows;

$$A = 2\,B\,/E_n$$

   Where E_n lesser of E_S, E_C or E_r
9. Required moment of inertia, I_S or I_S'

$$I_S = \left[\left(A\,D_L^2\,A_{TL}\right)/14\right]$$

$$I_S' = \left[\left(A\,D_L^2\,A_{TL}\right)/10.9\right]$$

10. Compare required values of I with actual;
   $I > I_S$
   $I' > I_S'$

SMALL END

Case 1: External Loads are Not Included

$$A_{rs} = \left(P_X\,r^2\,K\,\mathrm{Tan}\,\alpha\right)/2\,X_1$$

Case 2: External Loads are Included

$$A_{rs} = (K\,Q_L\,r\,\mathrm{Tan}\,\alpha)/X_1$$

From the worksheet, select the largest value of Q_L for the small end in compression.

Reinforcement Available

- Effective area available at small end, A_{eS}

$$A_{eS} = .55\,(t_1 - t_S)(D_S\,t_1)^{1/2} + (t_C - t_r/\cos\alpha)$$

- Area required at large end, A_r

$$A_r = A_{rS} - A_{eS}$$

If A_r is negative, the design is adequate as is. If A_r is positive, then either a ring must be added or the area of section increased. A_r is the net area required for the ring only!

- If a ring is required, the maximum distance to centroid of ring, L_b

$$L_b = .25\,(r\,t_1)^{1/2}$$

- If the shell is made thicker to provide more area for reinforcement, then the maximum length of shell included in the reinforcement shall be;

$$L_a = (r\,t_1)^{1/2}$$

If the Small End is a Line of Support (LOS)

1. Assume size of ring;
   Size = _____
   A = ______
   I = _______
2. Determine length of cylinder acting with cone, L_S
   L_S = ______

3. Determine length of cone, L_C

$$L_C = \left[L^2 + (R - r)^2\right]^{1/2}$$

4. Calculate equivalent area of cylinder, cone and ring, A_{TS}

$$A_{TS} = .5\,(L_S\,t_1) + .5\,(L_C\,t_C) + A_S$$

5. Calculate Value N

$$N = .5\,(r \tan\alpha) + .5\,L_S + \left[\left(R^2 - r^2\right)/ \times (6\,r \tan\alpha)\right]$$

6. Calculate Value F_S

$$F_S = P_X\,N + f_n \tan\alpha$$

Choose f_n as worst value, compression

7. Calculate Factor B

$$B = .75\,[(F_S\,D_S)/A_{TS}]$$

8. Find Factor A from applicable material curve or calculate as follows;

$$A = 2\,B/E_n$$

Where E_n lesser of E_S, E_C or E_r

9. Required moment of inertia, I_S or I_S'

$$I_S = \left[\left(A\,D_S^2\,A_{TS}\right)/14\right]$$

$$I_{S'} = \left[\left(A\,D_S^2\,A_{TS}\right)/10.9\right]$$

10. Compare required values of I with actual;

$I > I_S$

$I' > I_S'$

3.0. Design of Cones with Half Apex Angle, α, Between 30° and 60°, Internal Pressure Only!

Based on CC2150

LARGE END;

- Required thickness of shell;

$$t_{S2} = (P\,R)/(S_S\,E_1 - .6P)$$

- Required thickness of cone

$$t_{C2} = (P\,R)/(\cos\alpha\,(S_C\,E_2 - .6P))$$

CHECK STRESSES

- Determine ratio;

$R/t_2 =$

- Determine values of factors, X and Y, from the appropriate figure. Use Fig 2-10 if shell and cone are the same thickness. Use Fig 2-11 if the shell and cone are not the same thickness.

X =

Y =

- Calculate longitudinal stress, σ_X

$$\sigma_X = \left(P\,R/t_2\right)\left[.5 + X\,(R/t_2)^{1/2}\right]$$

- Calculate circumferential stress, σ_ϕ

$$\sigma_\phi = \left(P\,R/t_2\right)\left[1 - Y\,(R/t_2)^{1/2}\right]$$

- Allowable stresses;

$\sigma_X < 3\,S_S$

$\sigma_\phi < 1.5\,S_S$

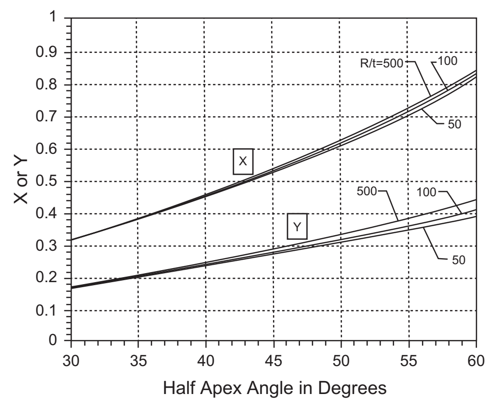


Figure 2-10. *X* and *Y* for cone thickness= t (Use when Cone and Shell are the same thickness)

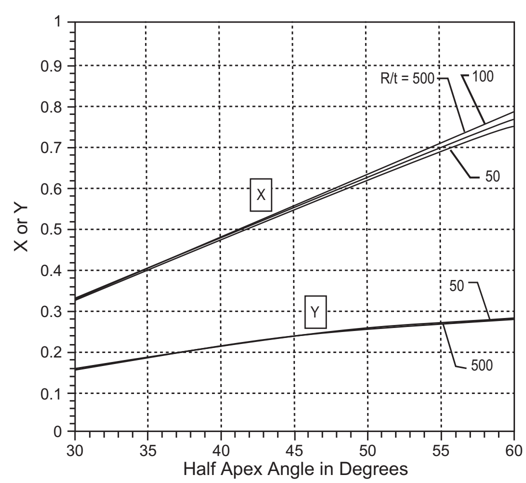


Figure 2-11. *X* and *Y* for cone thickness= t/cos α (Use when Cone and Shell are the not same thickness)

Sample Problem

$\alpha = 45°$
P = 500 psig
D = 120 in
R = 60.125 in Corr
DT = 600°F
Matl: SA-516-70
S_S = 19,400 PSI
E = 1
Ca = 0.125 in
Code: ASME VIII-1

- Required thickness of Shell;

$$t_{S2} = (P\,R)/(S_S\,E_1 - .6P)$$
$$= (500\,(60.125))/(19400 - 300)$$
$$= 1.57 + .125 = 1.699$$

Use t_2 = 1.75 in (1.625 in Corr)

- Required thickness of cone

$$t_{C2} = (P\,R)/[\cos\alpha\,(S_C\,E - .6P)]$$
$$= (500\,(60.125))/[\cos 45\,(19400 - 300)]$$
$$= 2.22 + .125 = 2.35$$

Use t_C = 2.375 in (2.25 in Corr)

CHECK STRESSES

- Determine ratio;

$$R/t_2 = 60.125/1.625 = 37$$

- Determine values of factors, X and Y, from Fig 2-11
  X = .53
  Y = .24
- Calculate longitudinal stress, σ_X

$$\sigma_X = \left(P\,R/t_2\right)\left[.5 + X\,(R/t_2)^{1/2}\right]$$
$$= \left[500\left(60.125\right)\Big/1.625\right]$$
$$\left[.5 + .53\,(60.125/1.625)^{1/2}\right]$$
$$= 68,691 \text{ PSI}$$

Allowable stress = 3 S_S = 3 (19,400) = 58,200 PSI
Therefore design is not acceptable!
Increase shell thickness by 0.125 in and recalculate stresses

- Calculate circumferential stress, σ_ϕ

$$\sigma_\phi = \left(P\,R/t_2\right)\left[1 - Y\,(R/t_2)^{1/2}\right]$$
$$= \left[500\left(60.125\right)\Big/1.625\right]$$
$$\left[1 - .24\,(60.125/1.625)^{1/2}\right]$$
$$= (-)\ 8,507 \text{ PSI (Acceptable)}$$

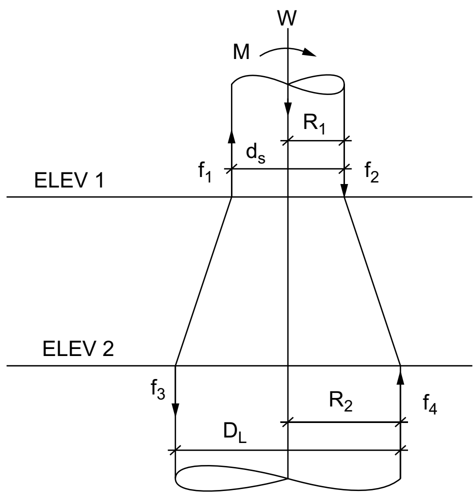
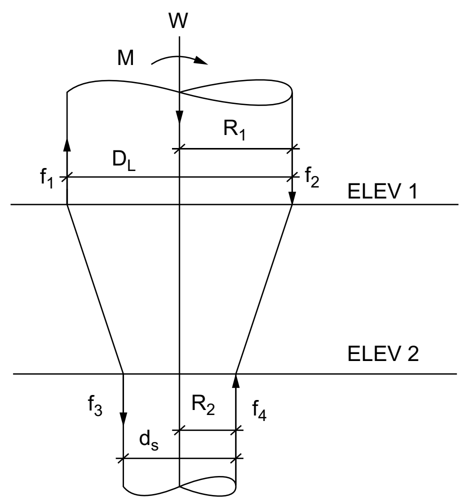
Notes

1. For small vessels or horizontal vessels with cones where there are not significant external loads, then the external loads may be neglected.
2. Line of Support: This indicates a formal work line or an elevation on the vessel used for external pressure calculations. It does not mean that the vessel is supported from that point.
3. The worksheet should be used to determine the loadings at the plane being investigated.
4. External Load Conditions;

INTERNAL PRESSURE:
a: Large End: .5PR + f_n is in tension, use this value in the procedure. If this value is (-), in compression, then a stress analysis is required.
b: Small End: .5PR + f_n is in tension, use this value in the procedure. If this value is (-), in compression, then a stress analysis is required.

EXTERNAL PRESSURE:
c: Large End: .5P_x R + f_n is in compression, use this value in the procedure. If this value is (+), in tension, then a stress analysis is required.
d: Large End: .5P_x R + f_n is in compression, use this value in the procedure. If this value is (+), in tension, then a stress analysis is required.

WORKSHEET TO DETERMINE LOADS AT CONE-CYLINDER INTERSECTIONS

CASE 1	CASE 2
	

Loads at Elevations due to External Forces, f_n (2)

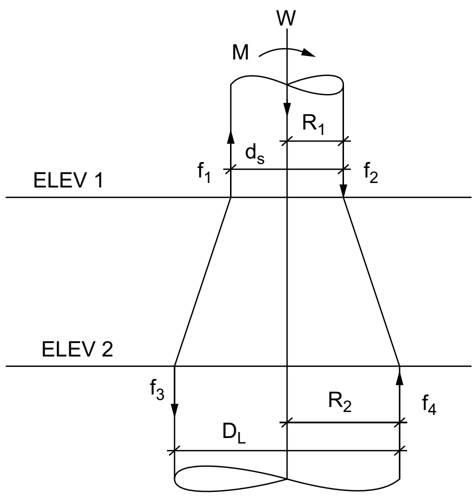
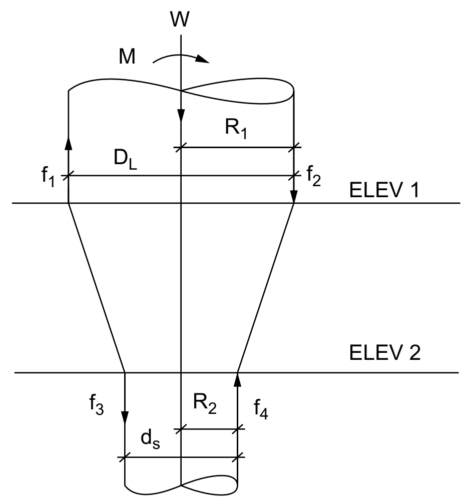
ELEV / SMALL END	$f_1 = \frac{-W_1}{\pi d_s} + \frac{4M_1}{\pi d_s^2}$		LARGE END	$f_1 = \frac{-W_1}{\pi D_L} + \frac{4M_1}{\pi D_L^2}$	
ELEV / SMALL END	$f_2 = \frac{-W_1}{\pi d_s} - \frac{4M_1}{\pi d_s^2}$		LARGE END	$f_2 = \frac{-W_1}{\pi D_L} - \frac{4M_1}{\pi D_L^2}$	
ELEV 2 / LARGE END	$f_3 = \frac{-W_2}{\pi D_L} + \frac{4M_2}{\pi D_L^2}$		SMALL END	$f_3 = \frac{-W_2}{\pi d_s} + \frac{4M_2}{\pi d_s^2}$	
ELEV 2 / LARGE END	$f_4 = \frac{-W_2}{\pi D_L} - \frac{4M_2}{\pi D_L^2}$		SMALL END	$f_4 = \frac{-W_2}{\pi d_s} - \frac{4M_2}{\pi d_s^2}$	

TOTAL LOAD, Q_L (1)			DATA			
	INT	EXT				
Q_{L1} = +/- .5 P_n R_1 +/- f_1			W_1		D_L	
Q_{L2} = +/- .5 P_n R_1 +/- f_2			W_2		d_s	
Q_{L3} = +/- .5 P_n R_2 +/- f_3			M_1		R_1	
Q_{L4} = +/- .5 P_n R_2 +/- f_4			M_2		R_2	
			P		P_X	

NOTES:

1. The expression (.5 P_n R_n) is (+) for internal pressure and (-) for external pressure. The sign of f_n is + or - based on the results of the calculation above
2. f_1 and f_3 are (-) if uplift due to moment is < weight. f1 and f3 are (+) if uplift due to moment is > weight.

WORKSHEET TO DETERMINE LOADS AT CONE-CYLINDER INTERSECTIONS (EXAMPLE)

CASE 1	CASE 2
	

Loads at Elevations due to External Forces, f_n (2)

Elev	Case 1 End	Case 1 Formula	Case 1 Value	Case 2 End	Case 2 Formula	Case 2 Value
ELEV	SMALL END	$f_1 = \frac{-W_1}{\pi d_s} + \frac{4M_1}{\pi d_s^2}$	(+) 735 Lbs/In	LARGE END	$f_1 = \frac{-W_1}{\pi D_L} + \frac{4M_1}{\pi D_L^2}$	
		$f_2 = \frac{-W_1}{\pi d_s} - \frac{4M_1}{\pi d_s^2}$	(-) 1121 Lbs/In		$f_2 = \frac{-W_1}{\pi D_L} - \frac{4M_1}{\pi D_L^2}$	
ELEV 2	LARGE END	$f_3 = \frac{-W_2}{\pi D_L} + \frac{4M_2}{\pi D_L^2}$	(+) 180 Lbs/In	SMALL END	$f_3 = \frac{-W_2}{\pi d_s} + \frac{4M_2}{\pi d_s^2}$	
		$f_4 = \frac{-W_2}{\pi D_L} - \frac{4M_2}{\pi D_L^2}$	(-) 396 Lbs/In		$f_4 = \frac{-W_2}{\pi d_s} - \frac{4M_2}{\pi d_s^2}$	

TOTAL LOAD, Q_L (1)			DATA			
	INT	EXT				
Q_{L1} = +/- .5 P_n R_1 +/- f_1	(+)1489 Lbs/ in	(+) 622 Lbs / in	W_1	36,500 Lbs	D_L	120.438"
Q_{L2} = +/- .5 P_n R_1 +/- f_2	(-)367 Lbs/in	(-) 1234 Lbs/in	W_2	41,100 Lbs	d_s	60.3125"
Q_{L3} = +/- .5 P_n R_2 +/- f_3	(+) 1685 Lbs/in	(-) 45 Lbs/in	M_1	2,652,000 in-Lbs	R_1	30.156"
Q_{L4} = +/- .5 P_n R_2 +/- f_4	(+) 1109 Lbs/in	(-) 621 Lbs/in	M_2	3,288,000 in-Lbs	R_2	60.219"
			P	50 PSIG	P_x	7.5 PSIG

NOTES:

1. The expression (.5 P_n R_n) is (+) for internal pressure and (-) for external pressure. The sign of f_n is + or - based on the results of the calculation above
2. f_1 and f_3 are (-) if uplift due to moment is < weight. f1 and f3 are (+) if uplift due to moment is > weight.

Procedure 2-6: Design of Toriconical Transitions [1,3]

Notation

P = internal pressure, psi
S = allowable stress, psi
E = joint efficiency
P_1, P_2 = equivalent internal pressure, psi
f_1, f_2 = longitudinal unit loads, lb/in.
σ_1, σ_2 = circumferential membrane stress, psi
α = half apex angle, deg
m = code correction factor for thickness of large knuckle
P_x = external pressure, psi
M_1, M_2 = longitudinal bending moment at elevation, in.-lb
W_1, W_2 = dead weight at elevation, lb

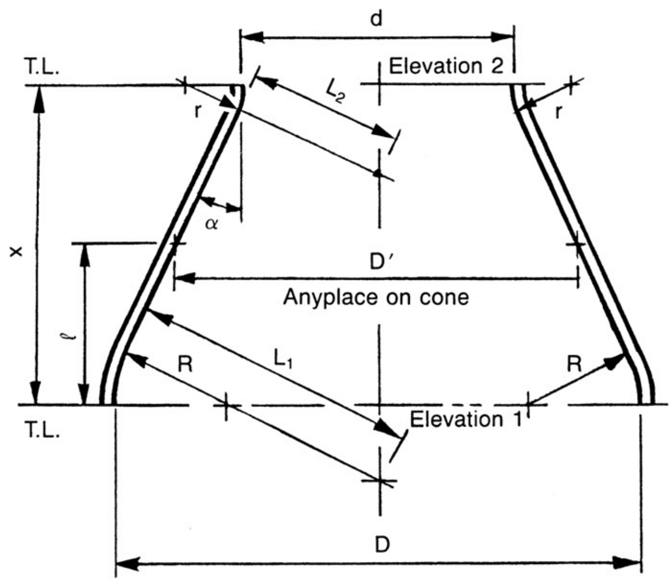
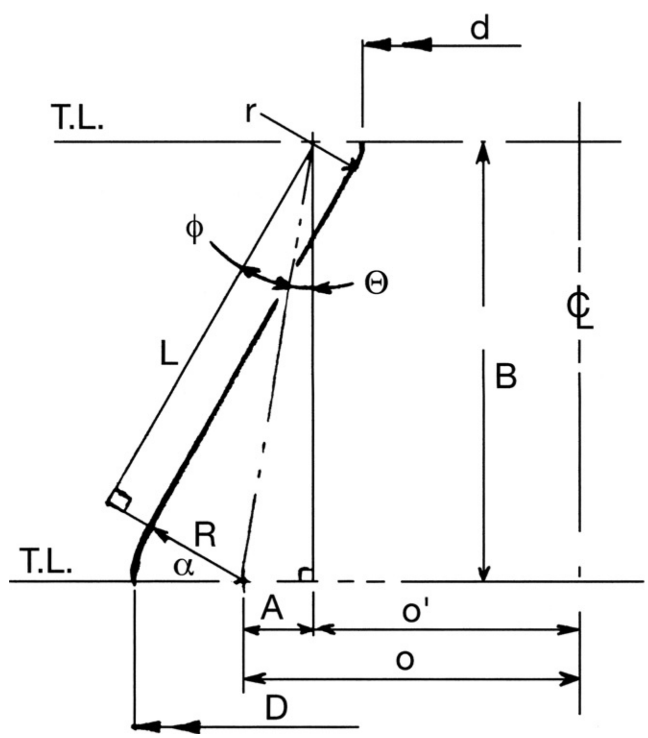


Figure 2-12. Dimensional data for a conical transition.

Calculating Angle, α

Case 1



$$o > o'$$

Step 1:

$$\sin\phi = \frac{R + r}{\sqrt{A^2 + B^2}} =$$

$$\phi = \_\_\_\_\_\_\_\_$$

Step 2:

$$\tan\Theta = \frac{A}{B}$$

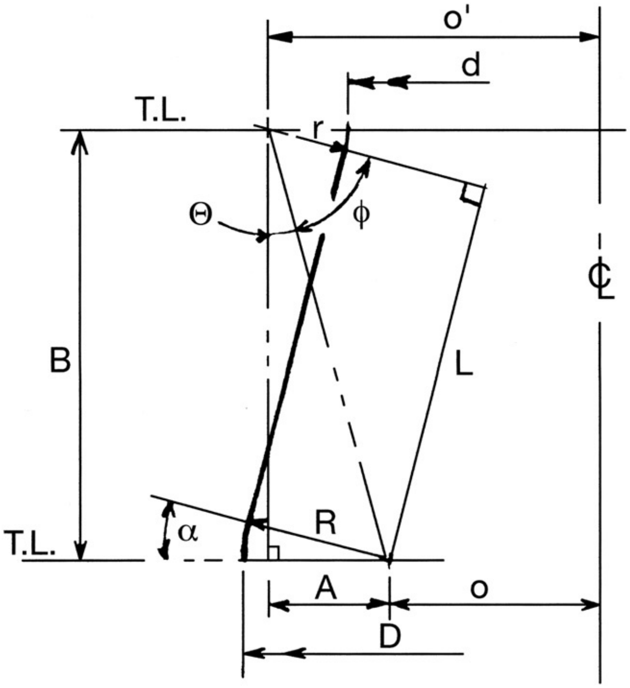
$$\Theta = \_\_\_\_\_\_\_\_$$

Step 3:

$$\alpha = \phi + \Theta$$

$$\alpha = \_\_\_\_\_\_\_\_$$

$$L = \cos\phi\sqrt{A^2 + B^2}$$



Case 2

$$o' > o$$

Step 1:

$$\cos\phi = \frac{R + r}{\sqrt{A^2 + B^2}} =$$

$\phi =$ ______________

Step 2:

$$\tan\Theta = \frac{A}{B}$$

$\Theta =$ ______________

Step 3:

$$\alpha = 90 - \Theta - \phi$$

$\alpha =$ ______________

$$L = \cos\phi\sqrt{A^2 + B^2}$$

Dimensional Formulas

$$D_1 = D - 2(R - R\cos\alpha)$$

$$D_2 = D + 2(R - R\cos\alpha)$$

$$D' = D - 2R\left(1 - \frac{1}{\cos\alpha}\right) - 2\ell\tan\alpha$$

$$L_1 = \frac{D_1}{2\cos\alpha}$$

$$L_2 = \frac{D_2}{2\cos\alpha}$$

$$m = 0.25\left(3 + \sqrt{\frac{L_1}{R}}\right)$$

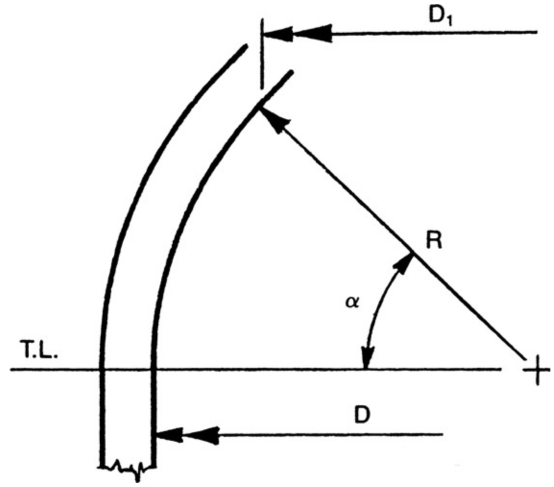


Figure 2-13. Dimensional data for the large end of a conical transition.

Large End (Figure 2-13)

- *Maximum longitudinal loads, f_1*
  (+) tension; (–) compression

$$f_1 = \frac{-W_1}{\pi D_1} \pm \frac{4M_1}{\pi D_1^2}$$

- *Determine equivalent pressure, P_1.*

$$P_1 = P + \frac{4f_1}{D_1}$$

- *Circumferential stress, D_1.*
  Compression:

$$\sigma_1 = \frac{PL_1}{t} - \frac{P_1L_1}{t}\left[\frac{L_1}{2R}\right]$$

- *Circumferential stress at D_1 without loads, σ_1.* Compression:

$$\sigma_1 = \frac{PL_1}{t}\left(1 - \frac{L_1}{2R}\right)$$

- *Thickness required knuckle, t_{rk}* [1, section 1-4(d)]. With loads:

$$t_{rk} = \frac{P_1L_1m}{2SE - 0.2P_1}$$

Without loads:

$$t_{rk} = \frac{PL_1m}{2SE - 0.2P}$$

- *Thickness required cone, t_{rc}* [1, section UG-32(g)]. With loads:

$$t_{rc} = \frac{P_1D_1}{2\cos\alpha(SE - 0.6P_1)}$$

Without loads:

$$t_{rc} = \frac{PD_1}{2\cos\alpha(SE - 0.6P)}$$

Small End (Figure 2-14)

- *Maximum longitudinal loads, f_2.*

  (+) tension; (–) compression

$$f_2 = \frac{-W_2}{\pi D_2} \pm \frac{4M_2}{\pi D_2^2}$$

- *Determine equivalent pressure, P_2.*

$$P_2 = P + \frac{4f_2}{D_2}$$

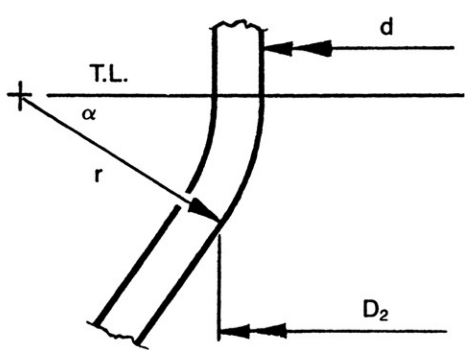


Figure 2-14. Dimensional data for the small end of a conical transition.

- *Circumferential stress at D_2.*

  Compression:

$$\sigma_2 = \frac{PL_2}{t} + \frac{P_2L_2}{t}\left[\frac{L_2}{2r}\right]$$

- *Circumferential stress at D_2 without loads, σ_2.*

  Compression:

$$\sigma_2 = \frac{PL_2}{t}\left(1 - \frac{L_2}{2r}\right)$$

- *Thickness required cone, at D_2, t_{rc}* [1, section UG-32(g)]. With loads:

$$t_{rc} = \frac{P_2D_2}{2\cos\alpha(SE - 0.6P_2)}$$

Without loads:

$$t_{rc} = \frac{PD_2}{2\cos\alpha(SE - 0.6P)}$$

- *Thickness required knuckle.* There is no requirement for thickness of the reverse knuckle at the small end of the cone. For convenience of fabrication it should be made the same thickness as the cone.

Additional Formulas (Figure 2-15)

- *Thickness required of cone at any diameter D', $t_{D'}$*

$$t_{D'} = \frac{PD'}{2\cos\alpha(SE - 0.6P)}$$

- *Thickness required for external pressure* [1, section UG-33(f)].

$$t_e = t\cos\alpha$$
$$D_L = D_2 + 2t_e$$

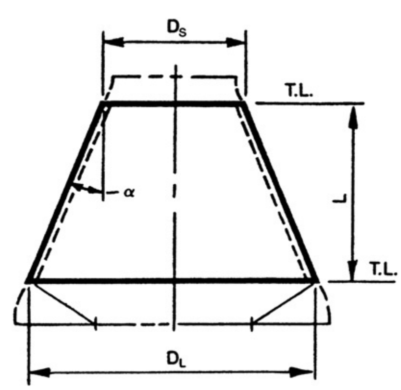


Figure 2-15. Dimensional data for cones due to external pressure.

$$D_s = D_1 + 2t_e$$
$$L = X - \sin\alpha(R+t) - \sin\alpha(r-t)$$
$$L_e = \frac{L}{2}\left(1+\frac{D_S}{D_L}\right)$$
$$\frac{L_e}{D_L} =$$
$$\frac{D_L}{t_e} =$$

Using these values, use Figure 2-1e to determine Factor A.

- *Allowable external pressure, P_a.*

$$P_a = \frac{2AEt_e}{D_L};\quad P_a > P_x$$

where E = modulus of elasticity at design temperature.

Notes

1. *Allowable stresses.* The maximum stress is the compressive stress at the tangency of the large knuckle and the cone. Failure would occur in local yielding rather than buckling; therefore the allowable stress should be the same as required for cylinders. Thus the allowable circumferential compressive stress should be the lesser of 2SE or F_y. Using a lower allowable stress would require the knuckle radius to be made very large—well above code requirements. See Reference 3.
2. Toriconical sections are mandatory if angle α exceeds 30° unless the design complies with Para. 1-5(e) of the ASME Code [1]. This paragraph requires a discontinuity analysis of the cone-shell juncture.
3. No reinforcing rings or added reinforcement is required at the intersections of cones and cylinders, providing a knuckle radius meeting ASME Code requirements is used. The minimum knuckle radius for the large end is not less than the greater of 3t or 0.12(R + t). The knuckle radius of the small end (flare) has no minimum. (See [Reference 1, Figure UG-36]).
4. Toriconical transitions are advisable to avoid the high discontinuity stresses at the junctures for the following conditions:
   a. High pressure—greater than 300 psig.
   b. High temperature—greater than 450 or 500°F.
   c. Low temperature—less than –20°F.
   d. Cyclic service (fatigue).

Procedure 2-7: Stresses in Heads Due to Internal Pressure [2,3]

Notation

L = crown radius, in.
r = knuckle radius, in.
h = depth of head, in.
R_L = latitudinal radius of curvature, in.
R_m = meridional radius of curvature, in.
σ_ϕ = latitudinal stress, psi
σ_x = meridional stress, psi
P = internal pressure, psi

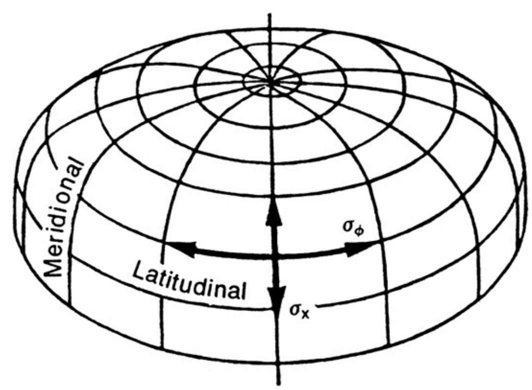


Figure 2-16. Direction of stresses in a vessel head.

Formulas

Lengths of R_L and R_m for ellipsoidal heads:

- At equator:

$$R_m = \frac{h^2}{R}$$

$$R_L = R$$

- At center of head:

$$R_m = R_L = \frac{R^2}{h}$$

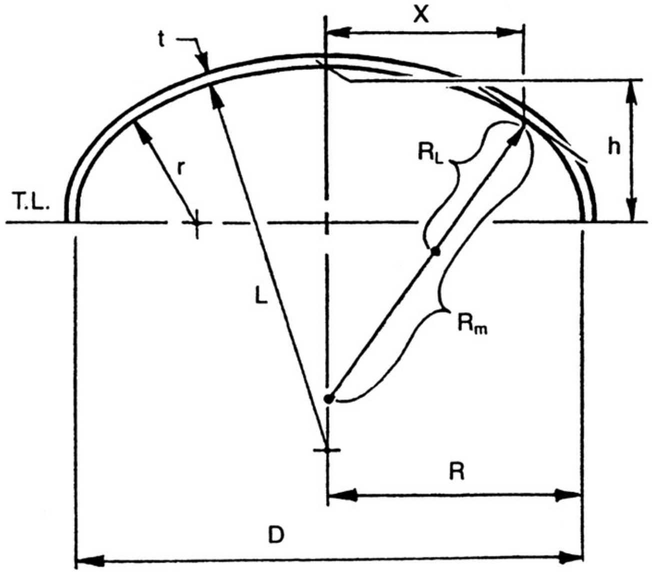


Figure 2-17. Dimensional data for a vessel head.

- At any point X:

$$R_L = \sqrt{\frac{R^4}{h^2} + X^2\left(1 - \frac{R^2}{h^2}\right)}$$

$$R_m = \frac{R_L^3 h^2}{R^4}$$

Notes

1. Latitudinal (hoop) stresses in the knuckle become compressive when the R/h ratio exceeds 1.42. These heads will fail by either elastic or plastic buckling, depending on the R/t ratio.
2. Head types fall into one of three general categories: hemispherical, torispherical, and ellipsoidal. Hemispherical heads are analyzed as spheres and were covered in the previous section. Torispherical (also known as flanged and dished heads) and ellipsoidal head formulas for stress are outlined in the following form.

TORISPHERICAL HEADS			
σ_x		σ_ϕ	
At Junction of Crown and Knuckle			
$\sigma_x = \frac{PL}{2t}$		$\sigma_\phi = \frac{PL}{4t}\left(3 - \frac{L}{R}\right)$	
In Crown			
$\sigma_x = \frac{PL}{2t}$		$\sigma_\phi = \sigma_x$	
In Knuckle			
See following section		See following section	
At Tangent Line			
$\sigma_x = \frac{PL}{2t}$		$\sigma_\phi = \frac{PR}{t}$	

ELLIPSOIDAL HEADS			
σ_x		σ_ϕ	
At Any Point X			
$\sigma_x = \frac{PR_L}{2t}$		$\sigma_\phi = \frac{PR_t}{t}\left(1 - \frac{R_L}{2R_m}\right)$	
At Center of Head			
$\sigma_x = \frac{PR^2}{2th}$		$\sigma_\phi = \sigma_x$	
At Tangent Line			
$\sigma_x = \frac{PR}{2t}$		$\sigma_\phi = \frac{PR}{t}\left(1 - \frac{R^2}{2h^2}\right)$	

CIRCUMFERENTIAL COMPRESSION STRESS IN KNUCKLE REGION OF TORISPHERICAL HEAD DUE TO INTERNAL PRESSURE

DATA
B = ASME Code allowable compressive stress, PSI
C.a. = Corrosion allowance, in
D = Inside diameter, in
DT = Design temperature, °F
F_C = Allowable compressive stress, PSI
K = ASME Code coefficient
L = Crown radius, use .9 D for 2:1 S.E. heads, in
M = ASME Code coefficient
P = Design pressure, PSIG
r = Knuckle radius, in
S = Allowable stress, tension, PSI
t = Head thickness, new, in
t_C = Head thickness, corroded, in
σ_φ = Circumferential stress, PSI

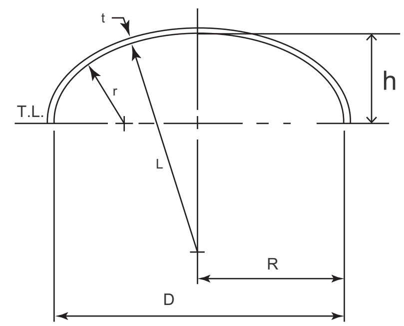
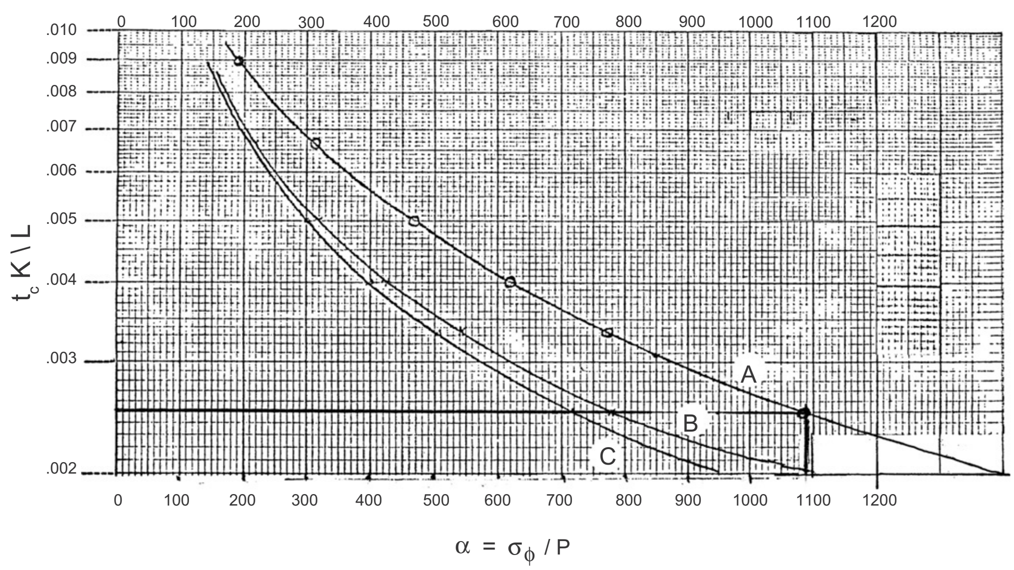


Figure 2.18 Dimensions of heads



Figures 2.19 Plot of head parameters

This procedure allows the designer to either check the compressive stress in an existing head, or to determine the thickness required for a new head. It should be noted that, the calculated thickness for internal pressure may not be adequate for the buckling stress in the knuckle. The example shown clearly illustrates this point.

Table 2-5
Head values for Figure 2.19

Ratio or Factor	Curve A	Curve B	Curve C
L / D	1.0	.9	.8
R / D	.06	.1	.1
M	1.77	1.5	1.46
K	1.13	1.33	1.37

Calculations

- Calculate coefficients K & M;

$$K = .167\left[2\,(D/2\,h)^2\right]$$

$$M = .25\left[3 + (L/r)^{1/2}\right]$$

- Determine ratios,
  L/D = ____
  r/D = _____
  L/r = ______
  L/K = ______
  t_C K / L = ______
- From Figure 2.19, using the appropriate curve, determine α
  α = _______
- Using value α, calculate σ_ϕ

$$\sigma_\phi = P\,\alpha$$

ALLOWABLE STRESS;

$$F_C = \text{Lesser of } 1.5\,S \text{ or } 1.5\,B$$

To find factor B…

$A = .125\, t_C/r$
B = From Chart
Use ______

EXAMPLE;

Material: SA-516-70
Head Proportions: 100% - 6%
D = 114 in
P = 25 PSIG
DT = 500°F
S = 20,000 PSI
E = 1.0
C.a. = .125 in
L = 114 in
R = 6.875 in
L/r = 16.58

- Thickness required for internal pressure, t_r

$$M = .25\left[3 + (L/r)^{1/2}\right] = 1.768$$

$$t_r = (PLM)/(2SE - .2P)$$
$$= (25\,(114)\,1.768)/(2\,(20{,}000)1.0\, - 5)$$
$$= .126 + .125 = .251 \text{ Use } .375 \text{ in}$$
$$t_C = .25 \text{ in}$$

- Determine circumferential stress, σ_ϕ

$$t_C\, K/L = .25\,(1.13)/114 = .00248$$

From curve A; $\alpha = \sigma_\phi$ / P = 1090

$$\sigma_\phi = 1090\,P = 1090\,(25) = 27{,}250 \text{ PSI}$$

- Allowable stress,
  Lesser of…

$$1.5\,S = 1.5 \times 20{,}000 = 30{,}000 \text{ PSI}$$
$$A = .125\, t_C/r = .125\,(.25)/6.875$$
$$= .00454$$
$$B = 13{,}000 \text{ PSI}$$
$$1.5\,B = 19{,}500 \text{ PSI}$$

Therefore design is not adequate. Increase thickness and recalculate;

TRIAL 2;

- Determine required σ_ϕ / P ratio;

$$\sigma_\phi/P = 19{,}500/25 = 780$$

- Find the t_C K / L ratio from graph for

$$\sigma_\phi/P = 780$$
$$t_C\, K/L = .0033$$

- Therefore t_r = .0033 (114) / 1.13 = .332
  The new thickness would be .332 + .125 = .458, use .5 in
- Check;

$$t_C = .5 - .125 = .375$$
$$t_C\, K/L = .00372$$
$$\sigma_\phi/P = 690$$
$$\sigma_\phi = 690\,P = 690\,(25) = 17{,}250 \text{ PSI}$$

Therefore design is acceptable.

Procedure 2-8: Design of Intermediate Heads [1,3]

Notation

A = factor A for external pressure
A_s = shear area, in.^{2}
B = allowable compressive stress, psi
F = load on weld(s), lb/in.
τ = shear stress, psi
E = joint efficiency
E_1 = modulus of elasticity at temperature, psi
S = code allowable stress, psi
H_D = hydrostatic end force, lb
P_i = maximum differential pressure on concave side of head, psi
P_e = maximum differential pressure on convex side of head, psi
K = spherical radius factor (see Table 2-6)
L = inside radius of hemi-head, in
= 0.9D for 2:1 S.E. heads, in
= KD for ellipsoidal heads, in
= crown radius of F & D heads, in

Table 2-6
Spherical radius factor, K

$\frac{D}{2h}$	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6	2.8	3.0
K	.5	.57	.65	.73	.81	.9	.99	1.08	1.18	1.27	1.36

Reprinted by permission from ASME Code Section VIII Div. 1, Table UG-33.1.

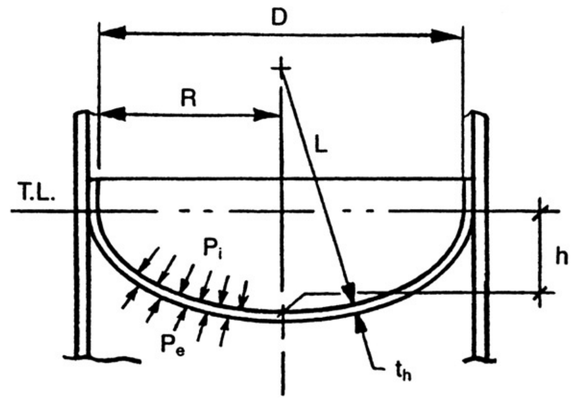


Figure 2-20. Dimensional data for an intermediate head.

Required Head Thickness, t_r

- *Internal pressure, P_i.* Select appropriate head formula based on head geometry. For dished only heads as in Figure 2-21, Case 3:

$$t_r = \frac{5P_iL}{6S}$$

- *External pressure, P_e.* Assume corroded head thickness, t_h

$$\text{Factor A} = \frac{0.125t_h}{L}$$

Factor B can be taken from applicable material charts in Section II, Part D, Subpart 3 of Reference 1.
Alternatively (or if Factor A lies to the left of the material/temperature line):

$$B = \frac{AE_1}{2}$$

$$t_r = \frac{P_eL}{B}$$

The required head thickness shall be the greater of that required for external pressure or that required for an internal pressure equal to $1.67 \times P_e$. See Reference 1, Para. UG-33(a).

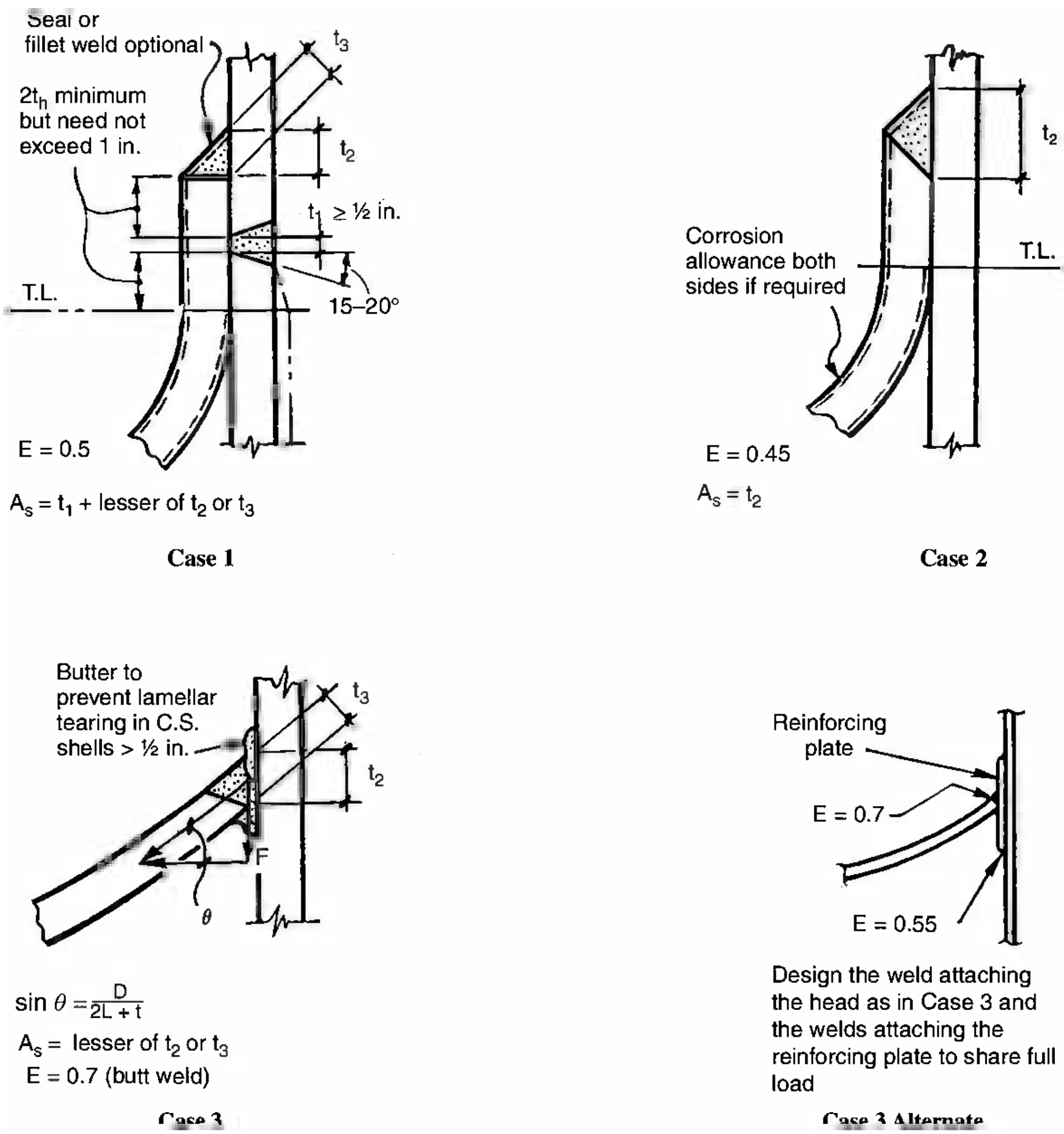


Figure 2-21. Methods of attachment of intermediate heads.

Shear Stress

- *Hydrostatic end force, H_D.*

$$H_D = \frac{P\pi D^2}{4}$$

where P = 1.5 × greater of P_i or P_e. (See Reference 1, Figure UW-13.1.)

- *Shear loads on welds, F.*

$$F = \frac{H_D}{\pi D \sin\theta}$$

Note: sin θ applies to Figure 2-21, Case 3 head attachments only!

- *Shear stress, τ.*

$$\tau = \frac{F}{A_s}$$

- *Allowable shear stress, SE.*

Procedure 2-9: Design of Flat Heads [1,2,4,5,6]

Notation

C = attachment factor
D = long span of noncircular heads, in.
d = diameter of circular heads or short span of non-circular heads, in.
E = joint efficiency (Cat. A seam only)
ℓ = length of straight flange measured from tangent line, in.
P = internal pressure psi
r = inside corner radius of head, in.
S = code allowable stress, tension, psi
t = minimum required thickness of head, in.
t_f = thickness of flange of forged head, in.
t_h = thickness of head, in.
t_r = minimum required thickness of seamless shell, in.
t_s = thickness of shell, in.
t_w = thickness of weld joint, in.
t_p = minimum distance from outside of head to edge of weld prep, in.
Z = factor, dependent on d/D ratio
Q_o = shear force per unit length, lb/in.
N_o = axial tensile force per unit length, lb/in.
M_o = radial bending moment, in.-lb/in.
ν = Poisson's ratio, 0.3 for steel

$\left.\begin{matrix} a_{1,2,3} \\ b_{1,2,3} \end{matrix}\right\}$ = Influence coefficients for head

$\left.\begin{matrix} a_{4,5,6} \\ b_{4,5,6} \end{matrix}\right\}$ = Influence coefficients for shell

Formulas

- *Circular heads.*

$$t = d\sqrt{\frac{CP}{SE}}$$

- *Noncircular heads.*

$$t = d\sqrt{\frac{ZCP}{SE}}$$

where $Z = 3.4 - \frac{2.4d}{D}; < 2.5$

- *Dimensionless factors.*

$$m = \frac{t_r}{t_s}$$

$$\beta = \sqrt[4]{\frac{12\left(1-\nu^2\right)}{d^2 t_s^2}}$$

$$a_1 = (-)3(1-\nu)\frac{d}{t_h}$$

$$a_2 = 2(1-\nu)$$

$$a_3 = \frac{3d(1-\nu)}{32t_h}$$

$$a_4 = (-)\frac{t_h}{t_s}\left[\frac{(\beta d)^2}{2}\right]$$

$$a_5 = (-)\frac{t_h}{t_s}\left(\frac{\beta d}{2}\right)$$

$$a_6 = (-)\frac{t_h}{t_s}\left(\frac{2-\nu}{8}\right)$$

$$b_1 = \frac{6(1-\nu)d^2}{(\beta d)^2 t_s t_h}$$

$$b_2 = (-)\frac{3(1-\nu)d}{(\beta d)^2 t_s}$$

$$b_3 = (-)\frac{3(1-\nu)d^2}{16(\beta d)^2 t_s t_h}$$

$$b_4 = (-)(\beta d)\left(\frac{t_h}{t_s}\right)^2$$

$$b_5 = (-)0.5\left(\frac{t_h}{t_s}\right)^2$$

$$b_6 = 0$$

Cases

Case 1 (Figure 2-22)

1. C = 0.17 for forged circular or noncircular heads.
2. $r \geq 3t_h$
3. C = 0.1 for circular heads if

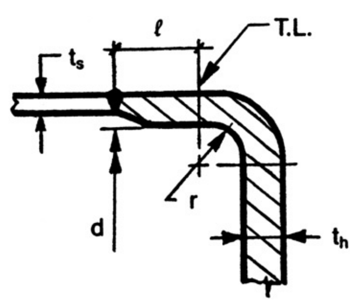


Figure 2-22. Case 1: Flanged head [1, Figure UG-34 (a)].

$$\ell \geq \left(1.1 - \frac{0.8t_s^2}{t_h^2}\right)\sqrt{dt_h}$$

or if the requirement of the flange length is not met and if

$$t_s \geq 1.12t_h\sqrt{1.1 - \frac{\ell}{\sqrt{dt_h}}}$$

for length $2\sqrt{dt_s}$ and taper is 4:1 minimum.

Case 2 (Figure 2-23)

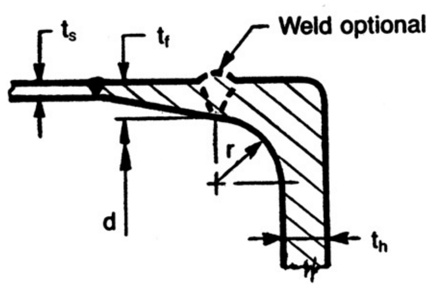


Figure 2-23. Case 2: Forged head [1, Figure UG-34 (b-1)].

1. $C = 0.17$
2. $t_f \geq 2t_s$
3. $r \geq 3t_f$
4. For forged circular or noncircular heads.

Case 3 (Figure 2-24)

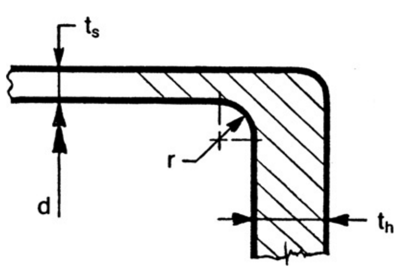


Figure 2-24. Case 3: Integrally forged head [1, Figure UG-34 (b-2)].

1. $C = 0.33$ m but ≥ 0.2
2. $r \geq 0.375$ in. if $t_s \leq 1.5$ in.
3. $r \geq 0.25t_s$ if t_s is greater than 1.5 in. but need not be greater than 0.75 in.

Case 4 (Figure 2-25)

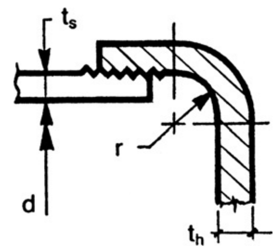


Figure 2-25. Case 4: Screwed flat head [1, Figure UG-34(c)].

1. $C = 0.3$
2. $r \geq 3t_h$
3. Design threads with 4:1 safety factor against failure by shear, tension, or compression due to hydrostatic end force.
4. Seal welding optional.
5. Threads must be as strong as standard pipe threads.

Case 5 (Figure 2-26)

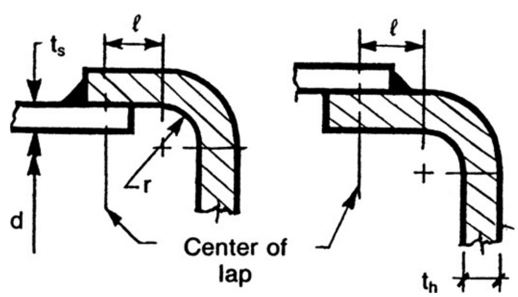


Figure 2-26. Case 5: Lap welded head [1, Figure UG-34(c)].

1. Circular heads: $C = 0.13$ if

$$\ell \geq \left(1.1 - \frac{0.8t_s^2}{t_h^2}\right)\sqrt{dt_h}$$

2. Noncircular heads and circular heads regardless of ℓ: $C = 0.2$.
3. $r \geq 3t_h$

Case 6 (Figure 2-27)

1. $C = 0.13$
2. $d \leq 24$ in.

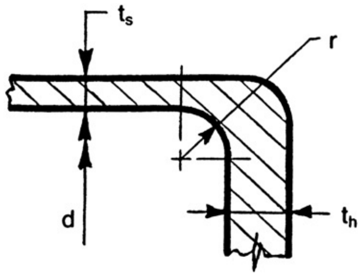


Figure 2-27. Case 6: Integrally forged head [1, Figure UG-34(d)].

3. $0.05 \leq t_h/d \leq 0.25$
4. $t_h \geq t_s$
5. $r \geq 0.25t_h$
6. Head integral with shell by upsetting, forging, or spinning.
7. Circular heads only.

Case 7 (Figure 2-28)

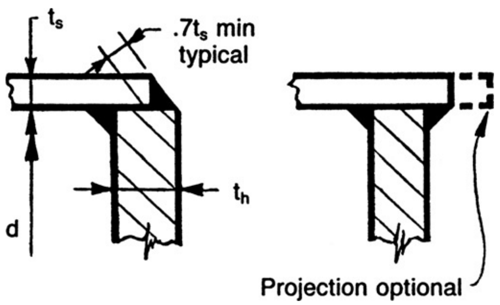


Figure 2-28. Case 7: Welded flat heads [1, Figure UG-34(e)(f)].

1. Circular heads: $C = 0.33$ m but ≥ 0.2. If $m < 1$, then shell cannot be tapered within $2\sqrt{dt_s}$ from inside of head.
2. Noncircular heads: $C = 0.33$
3. Liquid penetrant (L.P.) or magnetic particle test (M.T.) end of shell and O.D. of head if t_s or t_h is greater than 1/2 in. thick (before and after welding).

Case 8 (Figure 2-29)

1. Circular heads: $C = 0.33$ m but ≥ 0.2.

   $t_w \geq 2t_r$ and $\geq 1.25t_s$ but $\leq t_h$

If $m < 1$, then shell cannot be tapered within $2\sqrt{dt_s}$ from inside of head.

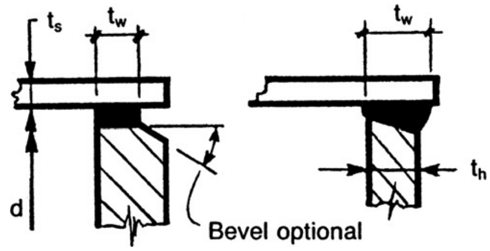


Figure 2-29. Case 8: Welded flat heads (Full penetration welds required) [1, Figure UG-34(g)].

2. Noncircular heads: $C = 0.33$
3. See Note 3 in Case 7.

Case 9 (Figure 2-30)

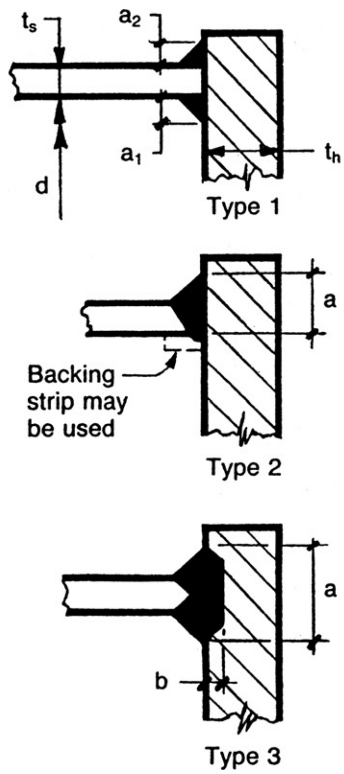


Figure 2-30. Case 9: Welded flat heads [1, Figure UG-34(h), Figure UW-13.2(g) (e-1)(f)].

1. Circular heads only.
2. $C = 0.33$
3. $t_s \geq 1.25t_r$
4. L.P./M.T. end of shell and O.D. of head if t_s or t_h is greater than ½ in. thick (before and after welding).
5. *Type 1*: $a_1 + a_2 \geq 2t_s$

   $0.5a_2 \leq a_1 \leq 2a_2$

Type 2: $a \geq 2t_s$
Type 3: $a + b \geq 2t_s$
$b = 0$ is permissible

Case 10 (Figure 2-31)

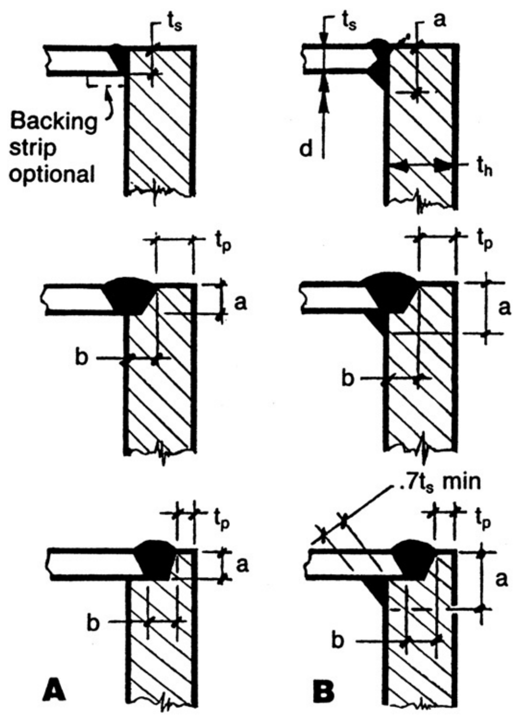


Figure 2-31. Case 10: Welded flat heads [1, Figure UG-34(h)(i), Figure UW-13.2(a)(b)(c)(d)].

1. For Figure 2-31A: $C = 0.33$ and $t_s \geq 1.25t_r$
2. For Figure 2-31B: $C = 0.33$ m but ≥ 0.2
3. $t_p \geq$ (t_s or 0.25 in.)
4. $t_w \geq t_s$
5. $a + b \geq 2t_s$
6. $a \geq t_s$
7. L.P./M.T end of shell and O.D. of head if t_s or t_h is greater than 1/2 in. thick (before and after welding).

Case 11 (Figure 2-32)

1. $C = 0.3$
2. All possible means of failure (by shear, tension, compression, or radial deformation, including flaring, resulting from pressure and differential thermal expansion) are resisted by factor of safety of 4:1.
3. Seal welding may be used.
4. Circular heads only.

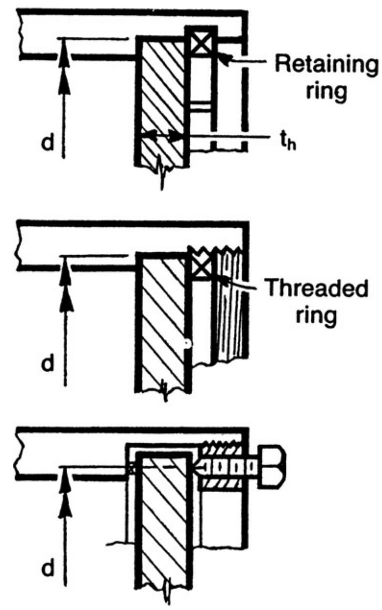


Figure 2-32. Case 11: Heads attached by mechanical lock devices [1, Figure UG-34(m)(n)(o)].

Case 12 (Figure 2-33)

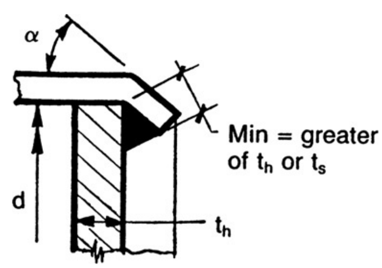


Figure 2-33. Case 12: Crimped head [1, Figure UG-34(r)].

1. $C = 0.33$
2. Circular plates only.
3. $d = 18$-in. maximum.
4. $\alpha = 30°$ minimum, $45°$ maximum.

Case 13 (Figure 2-34)

1. $C = 0.33$
2. Circular plates only.
3. $d = 18$-in. maximum.
4. $\alpha = 30°$ minimum, $45°$ maximum.
5. $t_s/d > P/S$ and $t_s/d > 0.05$

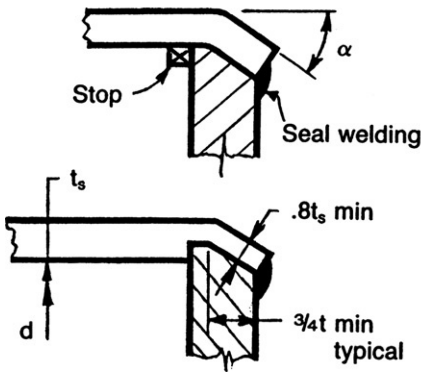


Figure 2-34. Case 13: Crimped heads [1, Section UG-34(s)].

6. Maximum allowable working pressure $\leq$ S/5d.
7. Crimping must be done at the proper forging temperature.

Stresses in Flat Heads

Maximum stress occurs at the junction, is axial in direction, and may be in either the head or the shell. When $t_h/t_s \leq 1$, the maximum stress is in the head at the junction. When $t_h/t_s > 1$, the maximum stress is in the shell at the junction. The bending moment M_o is a result of internal forces N_o and Q_o.

- *Internal force, Q_o.*

$$Q_o = Pd_m\left[\frac{(a_4 - a_1)b_3 - (a_3 - a_6)(b_4 - b_1)}{(a_4 - a_1)(b_5 - b_2) - (a_5 - a_2)(b_4 - b_1)}\right]$$

- *Bending moment, M_o.*

$$M_o = Pd_m^2\left[\frac{(a_3 - a_6)(b_5 - b_2) - (a_5 - a_2)b_3}{(a_4 - a_1)(b_5 - b_2) - (a_5 - a_2)(b_4 - b_1)}\right]$$

- *Axial stress in shell at junction, σ_s* [4, Equation 6.122].

$$\sigma_s = \frac{Pd_m}{4t_s} + \left|\frac{6M_o}{t_s^2}\right|$$

- *Axial stress in shell at junction, σ_h* [4, Equation 6.132].

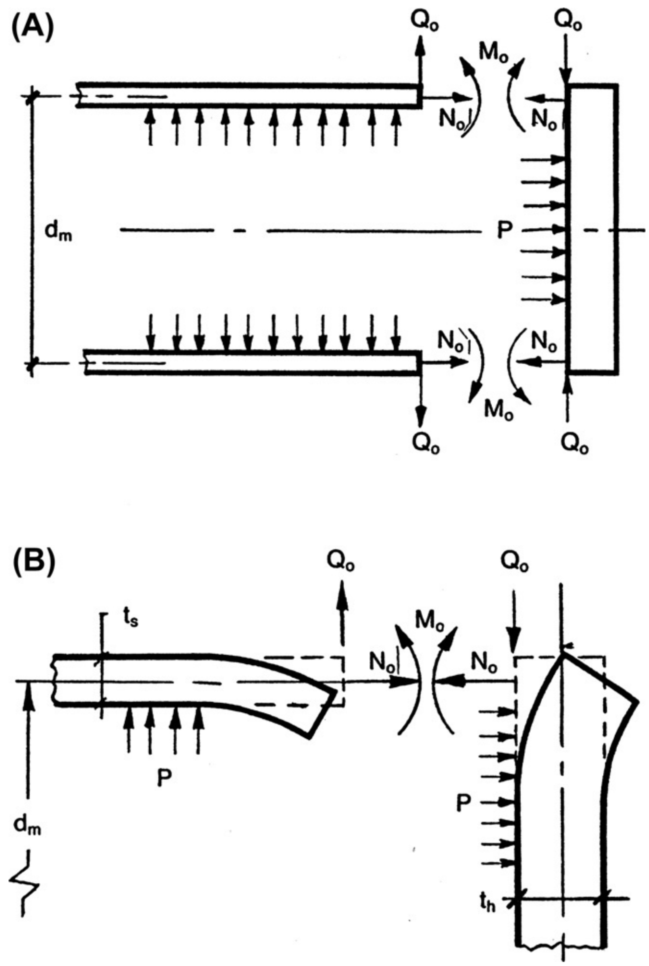


Figure 2-35. Discontinuity at flat head and cylindrical shell juncture.

$$\sigma_h = \left|\frac{Q_o}{t_h}\right| + \left|\frac{6M_o}{t_h^2} - \frac{3Q_o}{t_h}\right|$$

- *Primary bending stress in head, σ_b. Note:* Primary bending stress is maximum at the center of the head.

$$\sigma_b = (\pm)\frac{3(3+\nu)}{8}\left[\frac{Pd^2}{4t_h^2}\right]$$

(−) Inside head, compression
(+) Outside head, tension

Procedure 2-10: Design of Large Openings in Flat Heads [1]

Notation

P = internal pressure, psi
M_o = bending moment in head, in.-lb
M_h = moment acting on end of hub or shell at juncture, in.-lb
M_D = component of moment M_o due to H_D, in.-lb
M_T = component of moment M_o due to H_T, in.-lb
H = hydrostatic end force, lb
H_D = hydrostatic end force on area of central opening, lb
H_T = $H - H_D$, lb
S_H = longitudinal hub stress, psi
S_R = radial stress in head, psi
S_T = tangential stress in head, psi
S_{HS} = longitudinal hub stress, shell, psi
S_{RS} = radial stress, head, at O.D., psi
S_{TS} = tangential stress, head, at O.D., psi
S_{HO} = longitudinal hub stress at central opening, psi
S_{RO} = radial stress, head, at central opening, psi
S_{TO} = tangential stress, head, at central opening, psi
Z, Z_1, Y, T, U, F, V, f, e, d, L, X_1, and θ are all factors.

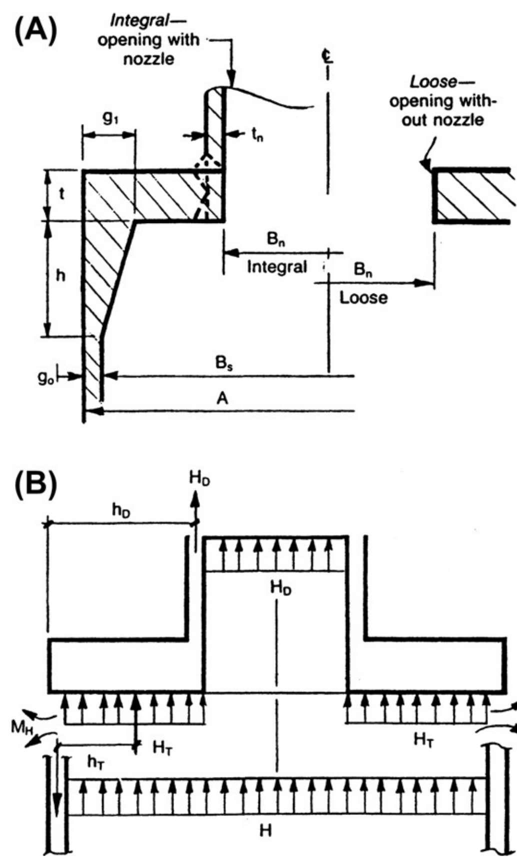


Figure 2-36. Dimensions (A) and loading diagram (B) for a flat integral head with opening.

Factor Formulas

1. Calculate geometry factors:

$$\frac{g_1}{g_o} =$$

$$K = \frac{A}{B_n} =$$

$$h_o = \sqrt{B_n g_o} =$$

$$\frac{h}{h_o} =$$

2. Using the factors calculated in Step 1, find the following factors in Procedure 3-1.

$Z =$
$Y =$
$T =$
$U =$
$F =$
$V =$
$f =$

3. Using the values found in the preceding steps, compute the following factors:

$$e = \frac{F}{h_o} =$$

$$d = \frac{U h_o g_o^2}{V} =$$

$$L = \frac{te + 1}{T} + \frac{t^3}{d} =$$

$$Z_1 = \frac{2K^2}{K^2 - 1} =$$

Stress and Moment Calculations

1. *Hydrostatic end forces, H, H_D, H_T.*

$$H = \frac{\pi B_s^2 P}{4}$$

$$H_D = \frac{\pi B_n^2 P}{4}$$

$$H_T = H - H_D$$

2. *Moment arms, h_D and h_T.*
   - Integral:

$$h_D = \frac{A - B_n - t_n}{2}$$

   - Loose:

$$h_D = \frac{A - B_n}{2}$$

   - Integral or loose:

$$h_T = \frac{B_s - B_n}{4} + \frac{g_o}{2}$$

3. *Moments.*

$$M_D = h_D H_D$$

$$M_T = h_T H_T$$

$$M_o = M_D + M_T$$

4. *Stresses in head and hub.*

$$S_H = \frac{fM_o}{Lg_1^2 B_n}$$

$$S_R = \frac{(1.33te + 1)M_o}{Lt^2 B_n}$$

   - Integral:

$$S_T = \frac{YM_o}{t^2 B_n} - ZS_R$$

   - Loose:

$$S_T = \frac{YM_o}{t^2 B_n}$$

5. *Factor, θ.*
   - Integral:

$$B_1 = B_n + g_o$$

If $f \geq 1$,

$$\theta = \frac{0.91(g_1/g_o)^2 B_1 V S_H}{fh_o}$$

   - Loose:

$$\theta = \frac{B_n S_T}{t}$$

6. *Moment at juncture of shell and head, M_H.*

$$M_H = \frac{\theta}{\dfrac{1.74h_o V}{g_o^3 B_1} + \dfrac{\theta}{M_o}\left(1 + \dfrac{Ft}{h_o}\right)}$$

where h_o, g_o, V, B_1, and F refer to shell.

7. *Factor X_1.*

$$X_1 = \frac{M_o - M_H\left(1 + \dfrac{Ft}{h_o}\right)}{M_o}$$

where F and h_o refer to shell.

8. *Stress at head-shell juncture.*

$$S_{HS} = \frac{1.1X_1 \theta h_o f}{(g_1/g_o)^2 B_s V}$$

$$S_{RS} = \frac{1.91M_H\left(1 + \dfrac{Ft}{h_o}\right)}{B_s t^2} + \frac{0.64FM_H}{B_s h_o t}$$

$$S_{TS} = \frac{X_1 \theta t}{B_s} - \frac{0.57M_H\left(1 + \dfrac{Ft}{h_o}\right)}{B_s t^2} + \frac{0.64FZM_H}{B_s h_o t}$$

where B_s, F, h_o, Z, f, g_o, g_1, and V refer to shell.

9. *Calculate stresses at head-nozzle juncture.*

$$S_{HO} = X_1 S_H$$

$$S_{RO} = X_1 S_R$$

$$S_{TO} = X_1 S_T + \frac{0.64FZ_1 M_H}{B_s h_o t}$$

where F, B_s, and h_o refer to shell.

Notes

1. This procedure is only applicable for integrally attached flat heads with centrally located openings which exceed one-half the head diameter. For applicable configurations see sketches in ASME Code, Figures UG-34(a), (b-1), (b-2), (d), or (g).
2. For details where inside corner of shell-head juncture is machined to a radius: $g_1 = g_o$ and $f = 1$.

3. The method employed in this procedure is to disregard the shell attached to the outside diameter of the flat head and then analyze the flat head with a central opening.
4. This procedure is based on Appendix 14 of ASME Section VIII, Division 1.

Procedure 2-11: Calculate MAP, MAWP, and Test Pressures

Notation

S_a = allowable stress at ambient temperature, psi
S_{DT} = allowable stress at design temperature, psi
S_{CA} = allowable stress of clad material at ambient temperature, psi
S_{CD} = allowable stress of clad material at design temperature, psi
S_{BA} = allowable stress of base material at ambient temperature, psi
S_{BD} = allowable stress of base material at design temperature, psi
C.a. = corrosion allowance, in.
t_{sc} = thickness of shell, corroded, in.
t_{sn} = thickness of shell, new, in.
t_{hc} = thickness of head, corroded, in.
t_{hn} = thickness of head, new, in.
t_b = thickness of base portion of clad material, in.
t_c = thickness of cladding, in.
R_n = inside radius, new, in.
R_c = inside radius, corroded, in.
R_o = outside radius, in.
D_n = inside diameter, new, in.
D_c = inside diameter, corroded, in.
D_o = outside diameter, in.
P_M = MAP, psi
P_W = MAWP, psi
P = design pressure, psi
P_S = shop hydro pressure (new and cold), psi
P_F = field hydro pressure (hot and corroded), psi
E = joint efficiency, see Procedure 2-1 and Appendix C

Definitions

Maximum Allowable Working Pressure (MAWP): The MAWP for a vessel is the maximum permissible pressure at the top of the vessel in its normal operating position at a specific temperature, usually the design temperature. When calculated, the MAWP should be stamped on the nameplate. The MAWP is the maximum pressure allowable in the “hot and corroded” condition. It is the least of the values calculated for the MAWP of any of the essential parts of the vessel, and adjusted for any difference in static head that may exist between the part considered and the top of the vessel. This pressure is based on calculations for every element of the vessel using nominal thicknesses exclusive of corrosion allowance. It is the basis for establishing the set pressures of any pressure-relieving devices protecting the vessel. The design pressure may be substituted if the MAWP is not calculated.

The MAWP for any vessel part is the maximum internal or external pressure, including any static head, together with the effect of any combination of loadings listed in UG-22 which are likely to occur, exclusive of corrosion allowance at the designated coincident operating temperature. The MAWP for the vessel will be governed by the MAWP of the weakest part.

The MAWP may be determined for more than one designated operating temperature. The applicable allowable stress value at each temperature would be used. When more than one set of conditions is specified for a given vessel, the vessel designer and user should decide which set of conditions will govern for the setting of the relief valve.

Maximum Allowable Pressure (MAP): The term MAP is no longer used by the Code but is used here as a matter of convenience. It refers to the maximum permissible pressure based on the weakest part in the new (uncorroded) and cold condition, and all other loadings are not taken into consideration.

Design Pressure: The pressure used in the design of a vessel component for the most severe condition of coincident pressure and temperature expected in normal operation. For this condition, and test condition, the maximum difference in pressure between the inside and outside of a vessel, or between any two chambers of a combination unit, shall be considered. Any thickness required for static head or other loadings shall be additional to that required for the design pressure.

Design Temperature: For most vessels, it is the temperature that corresponds to the design pressure. However, there is a maximum design temperature and a minimum design temperature for any given vessel. The minimum design temperature would be the MDMT (see Procedure 2-14). The MDMT shall be the lowest temperature expected in service or the lowest allowable temperature as calculated for the individual parts. Design temperature for vessels under external pressure shall not exceed the maximum temperatures given on the external pressure charts.

Operating Pressure: The pressure at the top of the vessel at which it normally operates. It shall be lower than the MAWP, design pressure, or the set pressure of any pressure relieving device.

Operating Temperature: The temperature that will be maintained in the metal of the part of the vessel being considered for the specified operation of the vessel.

Calculations

- *MAWP, corroded at Design Temperature P_ω. Shell*:

$$P_W = \frac{S_{DT}Et_{sc}}{R_c + 0.6t_{sc}} \text{ or } \frac{S_{DT}Et_{sc}}{R_o - 0.4t_{sc}}$$

2:1 S.E. Head:

$$P_W = \frac{2S_{DT}Et_{hc}}{D_c + 0.2t_{hc}} \text{ or } \frac{2S_{DT}Et_{hc}}{D_o - 1.8t_{hc}}$$

- *MAP, new and cold, P_M*

Shell:

$$P_M = \frac{S_aEt_{sn}}{R_n + 0.6t_{sn}} \text{ or } \frac{S_aEt_{sn}}{R_o - 0.4t_{sn}}$$

2:1 S.E. Head:

$$P_M = \frac{2S_aEt_{hn}}{D_n + 0.2t_{hn}} \text{ or } \frac{2S_aEt_{hn}}{D_o - 1.8t_{hn}}$$

- *Shop test pressure, P_s.*

$$P_s = 1.3P_M \text{ or } 1.3P_W\left[\frac{S_a}{S_{DT}}\right]$$

- *Field test pressure, P_F*

$$P_F = 1.3P$$

- *For clad vessels where credit is taken for the clad material, the following thicknesses may he substituted into the equations for MAP and MAWP:*

$$t_{sc}, t_{hc} = t_b + \left[\frac{S_{CD}}{S_{BD}}(t_c - C.a.)\right]$$

$$t_{sn}, t_{hn} = t_b + \left[\frac{S_{CA}t_c}{S_{BA}}\right]$$

Notes

1. Also check the pressure-temperature rating of the flanges for MAWP and MAP.
2. All nozzles should be reinforced for MAWP.
3. The MAP and MAWP for other components, i.e., cones, flat heads, hemi-heads, torispherical heads, etc., may be checked in the same manner by using the formula for pressure found in Procedure 2-1 and substituting the appropriate terms into the equations.
4. It is not necessary to take credit for the cladding thickness. If it is elected not to take credit for the cladding thickness, then base all calculations on the full base metal thickness. This assumes the cladding is equivalent to a corrosion allowance, and no credit is taken for the strength of the cladding.

Procedure 2-12: Nozzle Reinforcement

The following are only guidelines based on Section VIII, Division 1 of the ASME Code [1]. This is not an attempt to cover every possibility nor is it to become a substitute for using the Code.

1. *Limits.*
   a. No reinforcement other than that inherent in the construction is required for nozzles [1, Section UG-36(c)(3)]:
      - 3-in. pipe size and smaller in vessel walls 3/8–in. and less.
      - 2-in. pipe size and smaller in vessel walls greater than 3/8 in.

   b. Normal reinforcement methods apply to [1, Section UG-36(b)(1)]:
      - *Vessels 60-in. diameter and less*—1/2 the vessel diameter but not to exceed 20 in.
      - *Vessels greater than 60-in. diameter*—1/3 the vessel diameter but not to exceed 40 in.

   c. For nozzle openings greater than the limits of guideline 1.b., reinforcement shall be in accordance with Appendix 1-7 of the ASME Code.
2. *Strength.*
   It is advisable but not mandatory for reinforcing pad material to be the same as the vessel material [1, Section UG-41]:
   a. If a *higher strength material* is used, either in the pad or in the nozzle neck, no additional credit may be taken for the higher strength.
   b. If a *lower strength material* is used, either in the pad or in the nozzle, then the area taken as reinforcement must be decreased proportionately by the ratio of the allowable stress values of the two materials. Weld material taken as reinforcement must also be decreased as a proportion, assuming the weld material is the same strength as the weaker of the two materials joined.
3. *Thickness.*
   While minimum thicknesses are given in Reference 1, Section UG-16(b), it is recommended that pads be not less than 75% nor more than 150% of the part to which they are attached.
4. *Width.*
   While no minimum is stated, it is recommended that re-pads be at least 2 in. wide.
5. *Forming.*
   Reinforcing pads should be formed as closely to the contour of the vessel as possible. While normally put on the outside of the vessel, re-pads can also be put inside providing they do not interfere with the vessel's operation [1, Section UG-82].
6. *Tell-tale holes.*
   Reinforcing pads should be provided with a 1/4-in. tapped hole located at least 45° off the longitudinal center line and given an air-soap suds test [1, Section UW-15(d)].
7. *Elliptical or obround openings.*
   When reinforcement is required for elliptical or obround openings and the long dimension exceeds twice the short dimension, the reinforcement across the short dimension shall be increased to guard against excessive distortion due to twisting moment [1, Section UG-36(a)(1)].
8. *Openings in flat heads.*
   Reinforcement for openings in flat heads and blind flanges shall be as follows [1, Section UG-39]:
   a. *Openings < 1/2 head diameter*—area to be replaced equals 0.5d (t_r), or thickness of head or flange may be increased by:
      - Doubling C value.
      - Using C = 0.75.
      - Increasing head thickness by 1.414.

   b. *Openings > 1/2 head diameter*—shall be designed as a bolted flange connection. See Procedure 2-15.
9. *Openings in torispherical heads.*
   When a nozzle opening *and* all its reinforcement fall within the dished portion, the required thickness of head for reinforcement purposes shall be computed using M = 1 [1, Section UG-37(a)].
10. *Openings in elliptical heads.*
    When a nozzle opening *and* all its reinforcement fall within 0.8D of an elliptical head, the required thickness of the head for reinforcement purposes shall be equal to the thickness required for a seamless sphere of radius K(D) [1, Section UG-37(a)].
11. *General.*
    Reinforcement should be calculated in the corroded condition assuming maximum tolerance (minimum t). For non x-rayed vessels, t_r must be

computed using a stress value of 0.8S [1, Section UG-37(a)].

12. *Openings through seams.* [1, Section UW-14].
    a. Openings that have been reinforced may be located in a welded joint. E = joint efficiency of seam for reinforcement calculations. The ASME Code, Section Vlll, Division 1, does not allow a welded joint to have two different weld joint efficiencies. Credit may not be taken for a localized x-rayed portion of a spot or non x-rayed seam.
    b. Small nozzles that are not required to be checked per the Code can be located in circumferential joints providing the seam is x-rayed for a distance three times the diameter of the opening with the center of the hole at midlength.
13. *Re-pads over seams.*
    If at all possible, pads should not cover weld seams. When unavoidable, the seam should be ground flush before attaching the pad [1, Section UG-82].
14. *Openings near seams.*
    Small nozzles (for which the Code does not require the reinforcement to be checked) shall not be located closer than 1/2 in. to the edge of a main seam. When unavoidable, the seam shall be x-rayed, per ASME Code, Section UW-51, a distance of one and a half times the diameter of the opening either side of the closest point [1, Section UW-14].
15. *External pressure.*
    Reinforcement required for openings subject to external pressure only *or* where longitudinal compression governs shall only be 50% of that required for internal pressure and t_r is thickness required for external pressure [1, Section UG-37(d)].
16. *Ligaments.*
    When there is a series of closely spaced openings in a vessel shell and it is impractical to reinforce each opening, the construction is acceptable, provided the efficiency of the ligaments between the holes is acceptable [1, Section UG-53].
17. *Multiple openings.* [1, Section UG-42].
    a. For two openings closer than 2 times the average diameters and where limits of reinforcement overlap, the area between the openings shall meet the following:
       - *Must have a combined area equal to the sum of the two areas.*
       - *No portion of the cross-section shall apply to more than one opening.*
       - *Any overlap area shall be proportioned between the two openings by the ratio of the diameters.*
       - *If the area between the openings is less than 50% of that required for the two openings, the supplemental rules of Appendix 1-7(a) and (c) shall apply.*
    b. When more than two openings are to be provided with combined reinforcement:
       - *The minimum distance between the centers is* 1⅓ the average diameters.
       - *The area of reinforcement between the two nozzles shall be at least 50% of the area required for the two openings.*
    c. For openings less than 1⅓ times the average diameters:
       - *No credit may be taken for the area between the openings.*
       - *These openings shall be reinforced as in (d).*
    d. Multiple openings may be reinforced as an opening equal in diameter to that of a circle circumscribing the multiple openings.
18. *Plane of reinforcement.*
    A correction factor f may be used for "integrally reinforced" nozzles to compensate for differences in stress from longitudinal to circumferential axis of the vessel. Values of f vary from 1.0 for the longitudinal axis to 0.5 for circumferential axis [1, Section UG-37].

VESSEL DESCRIPTION:						ITEM NO:			SIZE:		
	1	2	3	4	5	Nozzle	1	2	3	4	5
Nozzle						t_n					
Location						t_{nc}					
Size and schedule						t_m					
P at elevation						Limit h					
I.D. New						A_2					
d (corroded)						A_3					
Shell/head t_{corr}						A_4					
Shell/head t_r						Pad size $t_e \times D_p$					
A						O.D. Nozzle					
Limit L						A_5					
A_1						A_T					

THICKNESS REQUIRED

Shell		Head		Nozzles					
$t_r = \dfrac{PR}{S - 0.6P}$		$t_{rh} = \dfrac{PD}{2S - 0.2P}$		$t_{rn} = \dfrac{PR_n}{S - 0.6P}$					
S, shell		*D		Nozzle					
		S, head		P					
*Note: D = R for heml-heads D = 0.9D if nozzle and reinforcement lie within 0.8D of 2:1 head D = L if nozzle and reinforcement lie within dished portion of a flanged and dished head.				R_n					
				S					
				t_{rn}					

FORMULAS

$A = dt_rF + 2t_nt_rF(1 - f_{r1})$		h = lesser of 2.5 t or $2.5\ t_{nc} + t_e$ h_1 = lesser of 2.5 t or $2.5\ (t_n - 2c.a.)$	
$A_1 = (2L - d)(t - Ft_r) - 2t_n(t - Ft_r)(1 - f_{r1})$			
$A_2 = 2h(t_n - t_m)f_{r1}$ $A_3 = 2h_1(t_n - 2\ c.a)f_{r1}$		$f_{r1} = \dfrac{S_{noz}}{S_{shell}} < 1$	< 1
A_4 = (WELDS) ts $(A_{41} + A_{43})f_{r1} + A_{42}f_{r4}$ $(A_5 = (D_P - d - 2t_n)t_e f_{r4}$		$f_{r4} = \dfrac{S_{pad}}{S_{shell}} < 1$	< 1
$A_T = A_1 + A_2 + A_3 + A_4 + A_5$			
L = greater of d or $R_n + t + t_{nc}$			

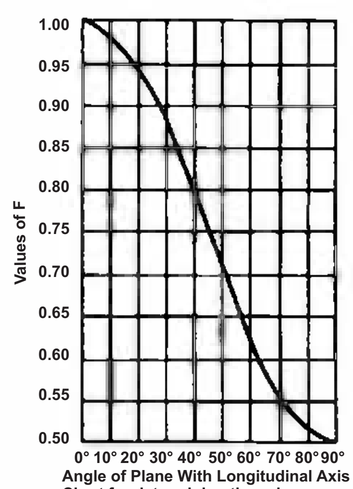


Chart for determining the value of F [1, Figure UG-37].

DESIGN DATA

Corrosion allowance. c.a.		Specific gravity	
Design liquid level		Thinning allowance	

Notes: Assumes E = 1 & f_{r1} = 1.0 for nozzle abutting vessel wall.

Figure 2-37. Worksheet for nozzle reinforcement calculations.

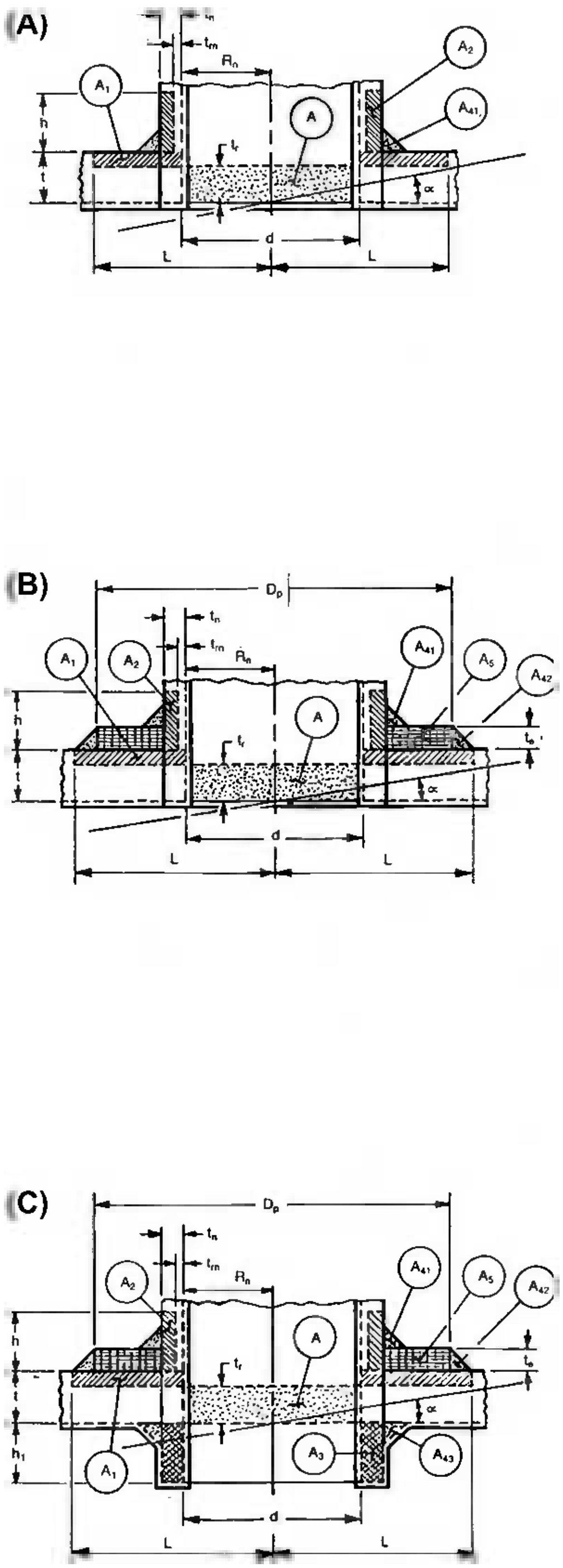


Figure 2-38. Typical nozzle connections.

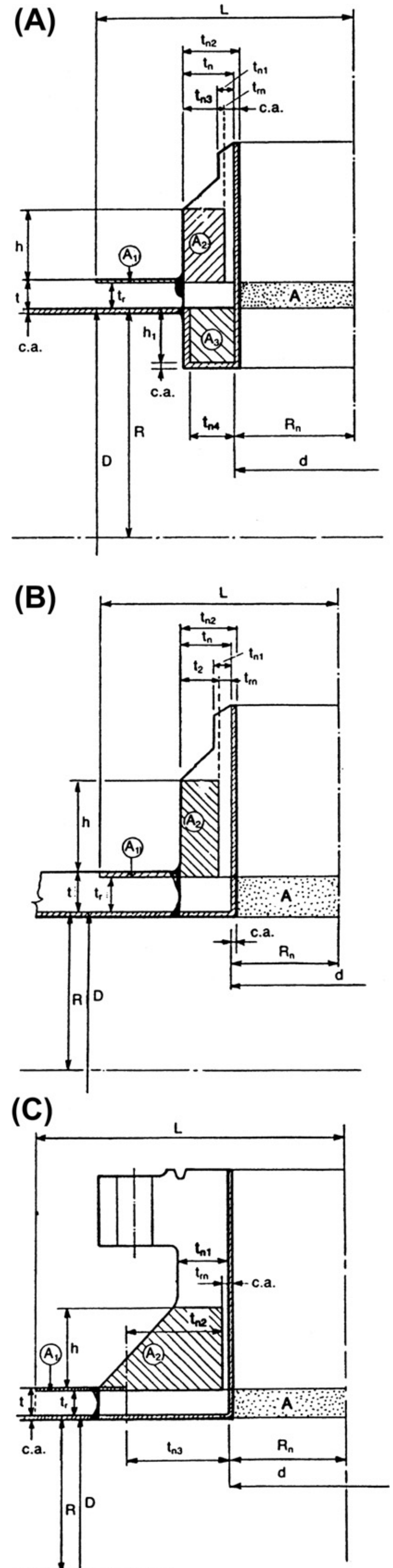
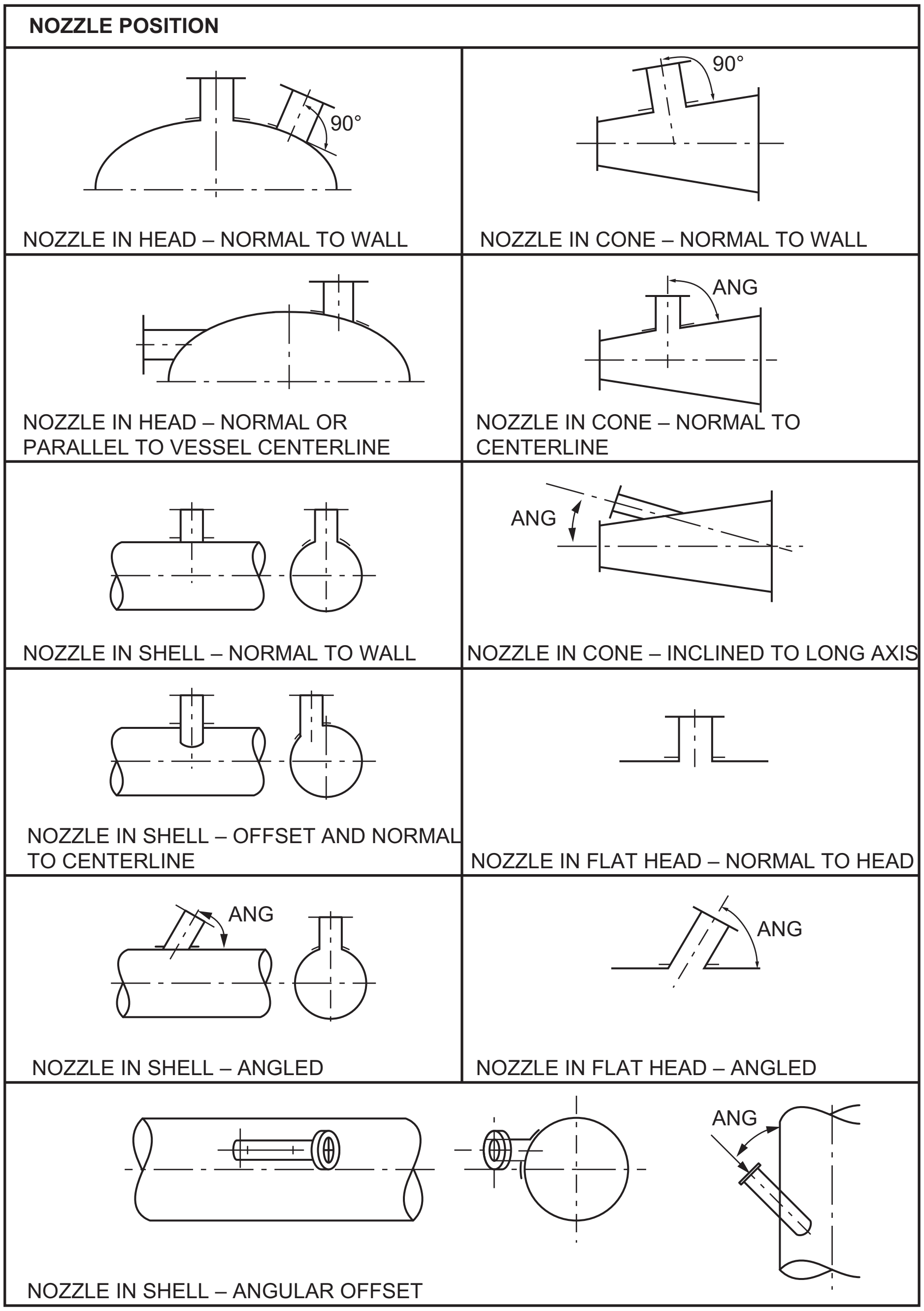


Figure 2-39. Typical self-reinforced nozzles.


NOZZLE POSITION
90°
90°
NOZZLE IN HEAD – NORMAL TO WALL
NOZZLE IN CONE – NORMAL TO WALL
ANG
NOZZLE IN HEAD – NORMAL OR PARALLEL TO VESSEL CENTERLINE
NOZZLE IN CONE – NORMAL TO CENTERLINE
ANG
NOZZLE IN SHELL – NORMAL TO WALL
NOZZLE IN CONE – INCLINED TO LONG AXIS
NOZZLE IN SHELL – OFFSET AND NORMAL TO CENTERLINE
NOZZLE IN FLAT HEAD – NORMAL TO HEAD
ANG
ANG
NOZZLE IN SHELL – ANGLED
NOZZLE IN FLAT HEAD – ANGLED
ANG
NOZZLE IN SHELL – ANGULAR OFFSET

Procedure 2-13 Find or Revise the Center of Gravity of a Vessel

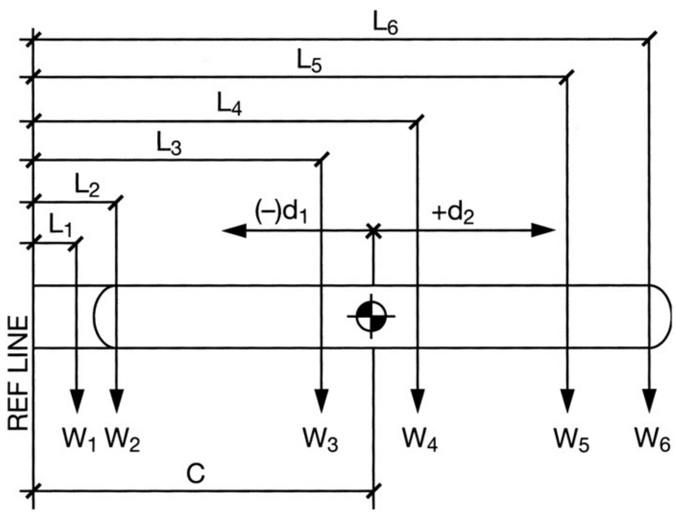


Figure 2-40. Load diagram for a typical vertical vessel.

Notation

C = distance to center of gravity, ft or in.
D′ = revised distance to C.G., ft or in.
d_n = distance from original C.G. to weights to add or remove, (+) or (−) as shown, ft or in.
L_n = distance from REF line to C.G. of a component weight, ft or in.
W_n = weight of vessel component, contents or attachments, lb
W′ = new overall weight, lb W + or – $\sum W_n$
W = overall weight, lb, $\sum W_n$
ω_n = revised unit weights, lb (+) to add weight (–) to remove weight

To find the C.G:

$$C = \frac{\sum L_n W_n}{W}$$

To revise C.G:

$$D' = C \pm \frac{\sum d_n \omega_n}{W'}$$

Procedure 2-14: Minimum Design Metal Temperature (MDMT)

Notation

R = use the lesser of R_1 or R_2
R_1 = ratio of thickness required at MDMT to the corroded thickness
R_2 = ratio of the actual stress to the allowable stress
t_{MT} = thickness required of the part at MDMT, in.
t_{DT} = thickness required of the part at design temperature, in.
t_n = thickness of the part, new, in (exclusive of thinning allowance for heads and undertolerance for pipe)
t_c = thickness of the part, corroded, in.
C.a. = corrosion allowance, in.
E = joint efficiency
S_{MT} = allowable stress at MDMT, psi
S_{DT} = allowable stress at design temperature, psi
S_a = actual tension stress in part due to pressure and all loadings, psi
T_1 = lowest allowable temperature for a given part based on the appropriate material curve of Figure 2-43, degrees F
T_2 = reduction in MDMT without impact testing per Figure 2-42, degrees F

This MDMT procedure is used to determine the lowest permissible temperature for which charpy impact testing is or is not required. The ASME Code requires this be determined for every pressure vessel and the MDMT be stamped on the nameplate. While every pressure vessel has its own unique MDMT, this may or may not be the MDMT that is stamped on the nameplate. Not only does every pressure vessel have its own unique MDMT, but every component of that pressure vessel has an MDMT. The vessel MDMT is the highest temperature of all the component MDMT's. On occasion, the MDMT is specified by the end user as an arbitrary value. The vessel fabricator is then responsible to verify that the actual MDMT of every component used in that pressure vessel is lower than the arbitrary value requested for the nameplate stamping. Considering this, there are various definitions for MDMT depending on how it is used. The definitions follow:

1. *Arbitrary MDMT:* A discretionary, arbitrary temperature, specified by a user or client, or determined in accordance with the provisions of UG-20. Some users have a standard value that has been chosen as the lowest mean temperature of the site conditions, such as 15°F.
2. *Exemption MDMT*: The lowest temperature at which the pressure vessel may be operated at full design pressure without impact testing of the component parts.
3. *Test MDMT:* The temperature at which the vessel is charpy impact tested.

The ASME Code rules for MDMT are built around a set of material exemption curves as shown in Figure 2-43. These curves account for the different toughness characteristics of carbon and low alloy steel and determine at what temperature and corresponding thickness impact testing will become mandatory.

There is an additional exemption curve (see Figure 2-42), which allows a decrease in the MDMT of every component, and thus the vessel, depending on one of several ratios specified. This curve would permit carbon steel, without impact testing, to be used at a temperature of (−)150°F, provided the combined stresses are less than 40% of the allowable stress for that material. Granted, the vessel would be more than twice as thick as it needed to be for the pressure condition alone, but if the goal was to exempt the vessel from impact testing, it could be accomplished.

Since impact testing is a major expense to the manufacturer of a pressure vessel, the designer should do everything to avoid it. Impact testing can always be avoided but may not be the most economical alternative. Following these steps will help eliminate the need for impact testing and, at the same time, will provide the lowest MDMT.

1. Upgrade the material to a higher group.
2. Increase the thickness of the component to reduce the stress in the part.
3. Decrease the pressure at MDMT. This is a process change and may or may not be possible. Sometimes a vessel does not operate at full design pressure at the low temperature condition but has alternate conditions, such as shutdown or depressurization. These alternate low temperature conditions can also be stamped on the nameplate.

Formulas

$$R_1 = \frac{t_r E}{t_c}$$

Table 2-7
Determination of MDMT (Example)

Part	Material	Material Group	S_{MT} ksi	S_{DT} ksi	t_n	t_{DT} (3)	t_{MT} (3)	t_c	S_a ksi	R_1	R_2	T_1	T_2	MDMT °F
Shell	SA-516-70	B	17.5	16.6	1.00	0.869	0.823	0.875	13.97	0.799	0.798	31°	20.1°	+11
Head(1)	SA-516-70	B	17.5	16.6	0.857	0.653	0.620	0.732	14.89	0.847	0.851	21.8	15°	+7
10″ Noz(2)	SA-53-B	B	12.8	12.2	0.519	0.174	0.166	0.394	5.26	0.421	0.410	−5.18	59°	
10″ 300# Flg. (4)	SA-105	B	17.5	16.6	0.519	0.128	0.121	0.394	5.26	0.307	0.300	−5.18	70°	
30″ Blind (5)	SA-266-2	B	17.5	16.6	6.06	—	1.48	5.94	—	—	—	51°	105°	−54
30″ Body Flg.	SA-266-2	B	17.5	16.6	Same as Shell									+11
Wear PL	SA-516-70	B	17.5	16.6	1.00	—	—	1.00	—	—	—	—	—	+11(6)
Bolting	SA-193-B7	—	—	—	—	—	—	—	—	—	—	—	—	−40

Notes:

1. The governing thickness for heads is based on that portion of the head which is in tension. For a 2:1 S.E. head this is the crown position where R = 0.90.
2. Includes pipe 12½% under tolerance.
3. Thickness exclusive of C.a.
4. Thickness at the hub (weld attachment) governs.
5. The governing thickness of flat heads and blind flanges is 1/4 of actual thickness.
6. Since the tension stress in the wear plate is less than the tension stress in the shell, the MDMT for the shell will govern.

$$R_2 = \frac{S_a}{S_{MT}}$$

$$t_c = t_n - \text{C.a.}$$

$$T_2 = (1 - R)100$$

$$\text{MDMT} = T_1 - T_2$$

Procedure

Step 1: Determine the lowest anticipated temperature to which the vessel will be subjected.
Step 2: Compare the lowest combined pressure-temperature case with the MDMT for each component.
Step 3: Determine if any components must be impact tested in their proposed material grade and thickness. This would establish the MDMT.
Step 4: Establish the overall MDMT as the highest value of MDMT for each of the component parts.

Notes

1. For flat heads, tubesheets, and blind flanges, the thickness used for each of the respective thickness' is that thickness divided by 4.
2. For corner, fillet, or lap-welded joints, the thickness used shall be the thinner of the two parts being joined.
3. For butt joints, the thickness used shall be the thickest joint.
4. For any Code construction, if the vessel is stress relieved and that stress relieving was not a Code requirement, the MDMT for that vessel may be reduced by 30° without impact testing.

Design Conditions (for example)

D.T. = 700°F
P = 400 PSIG
C.a. = 0.125
R_i = 30 in
E (Shell) = 0.85
E (Head) = 1.00
MDMT for vessel = + 11°F

General Notes on Assignment of Materials to Curves (Reprinted with permission from ASME Code, Section VIII, Div. 1.)

a. Curve A—all carbon and all low alloy steel plates, structural shapes, and bars not listed in Curves B, C, and D below.
b. Curve B

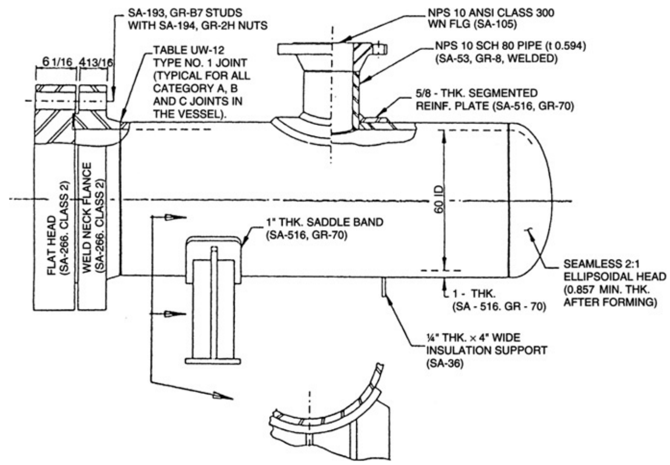
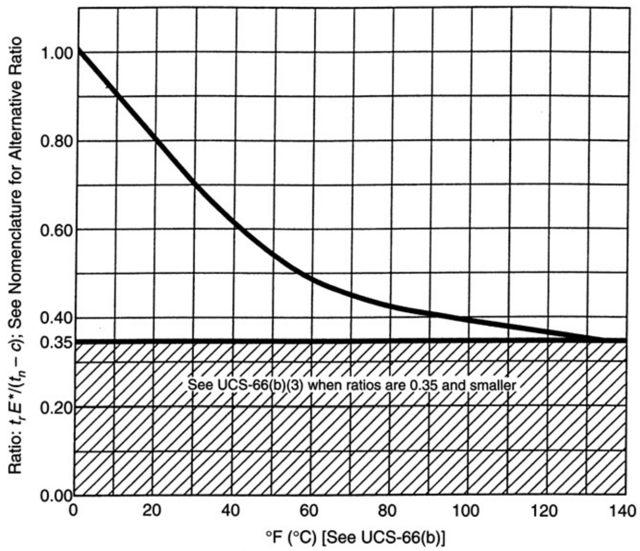


Figure 2-41. Dimensions of vessel used for MDMT example.

1. SA-285 Grades A and B
SA-414 Grade A
SA-515 Grade 60
SA-516 Grades 65 and 70 if not normalized
SA-612 if not normalized
SA-662 Grade B if not normalized
2. all materials of Curve A if produced to fine grain practice and normalized which are not listed for Curves C and D below.
3. except for bolting (see (e) below), plates, structural shapes, and bars, all other product forms (such as pipe, fittings, forgings, castings, and tubing) not listed for Curves C and D below.
4. parts permitted under UG-11 shall be included in Curve B even when fabricated from plate that otherwise would be assigned to a different curve.

c. Curve C
1. SA-182 Grades F21 and F22 if normalized and tempered
SA-302 Grades C and D
SA-336 Grades F21 and F22 if normalized and tempered
SA-387 Grades 21 and 22 if normalized and tempered
SA-516 Grades 55 and 60 if not normalized
SA-533 Grades B and C
SA-662 Grade A
2. all materials of Curve B if produced to fine grain practice and normalized and not listed for Curve D below.



Nomanclature (Note reference to General Notes of Fig. UCS-66-2.)

t_r = required thickness of the component under consideration in the corroded condition for all applicable loadings (General Note (2), based on the applicable joint efficiency E (General Note (3), in. (mm)

t_n = nominal thickness of the component under consideration before corrosion allowance is deducted, in. (mm)

c = corrosion allowance, in. (mm)

E^* = as defined in General Note (3)

Alternative Ratio = S^*E^* divided by the product of the maximum allowable stress value from Table UCS-23 times E, where S^* is the applied general primary membrane tensile stress and E and E^* are as defined in General Note (3)

Figure 2-42. Reduction in minimum design metal temperature without impact testing.

d. Curve D
   SA-203
   SA-508 Class 1
   SA-516 if normalized
   SA-524 Classes 1 and 2
   SA-537 Classes 1, 2, and 3
   SA-612 if normalized
   SA-622 if normalized
e. For bolting the following impact test exemption temperature shall apply:

Spec. No.	Grade	Impact Test Exemption Temperature, °F
SA-193	B5	−20
SA-193	B7	−40
SA-193	B7M	−55
SA-193	B16	−20
SA-307	B	−20
SA-320	B L7, L7A, L7M, L43	Impact tested
SA-325	1, 2	−20
SA-354	BC	0
SA-354	BD	+20
SA-449		−20
SA-540	B23/24	+10

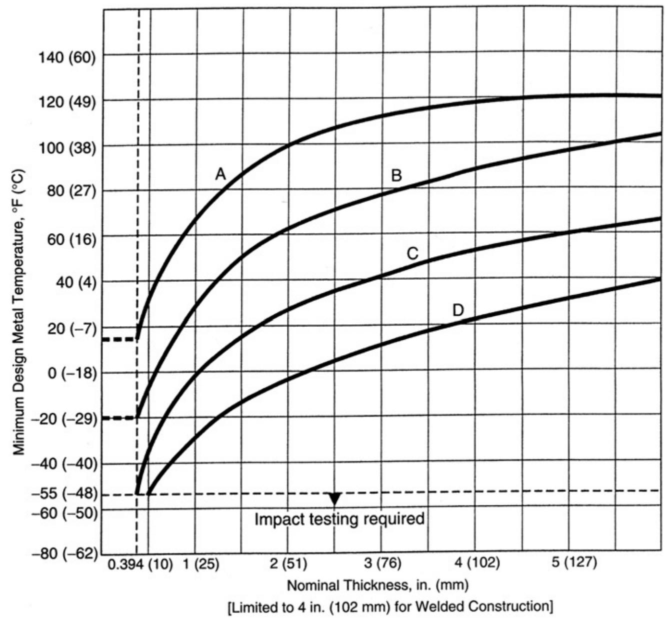


Figure 2-43. Impact test exemption curves.

f. When no class or grade is shown, all classes or grades are included.
g. The following shall apply to all material assignment notes:
   1. Cooling rates faster than those obtained by cooling in air, followed by tempering, as permitted by the material specification, are considered to be equivalent to normalizing or normalizing and tempering heat treatments.
   2. Fine grain practice is defined as the procedures necessary to obtain a fine austenitic grain size as described in SA-20.

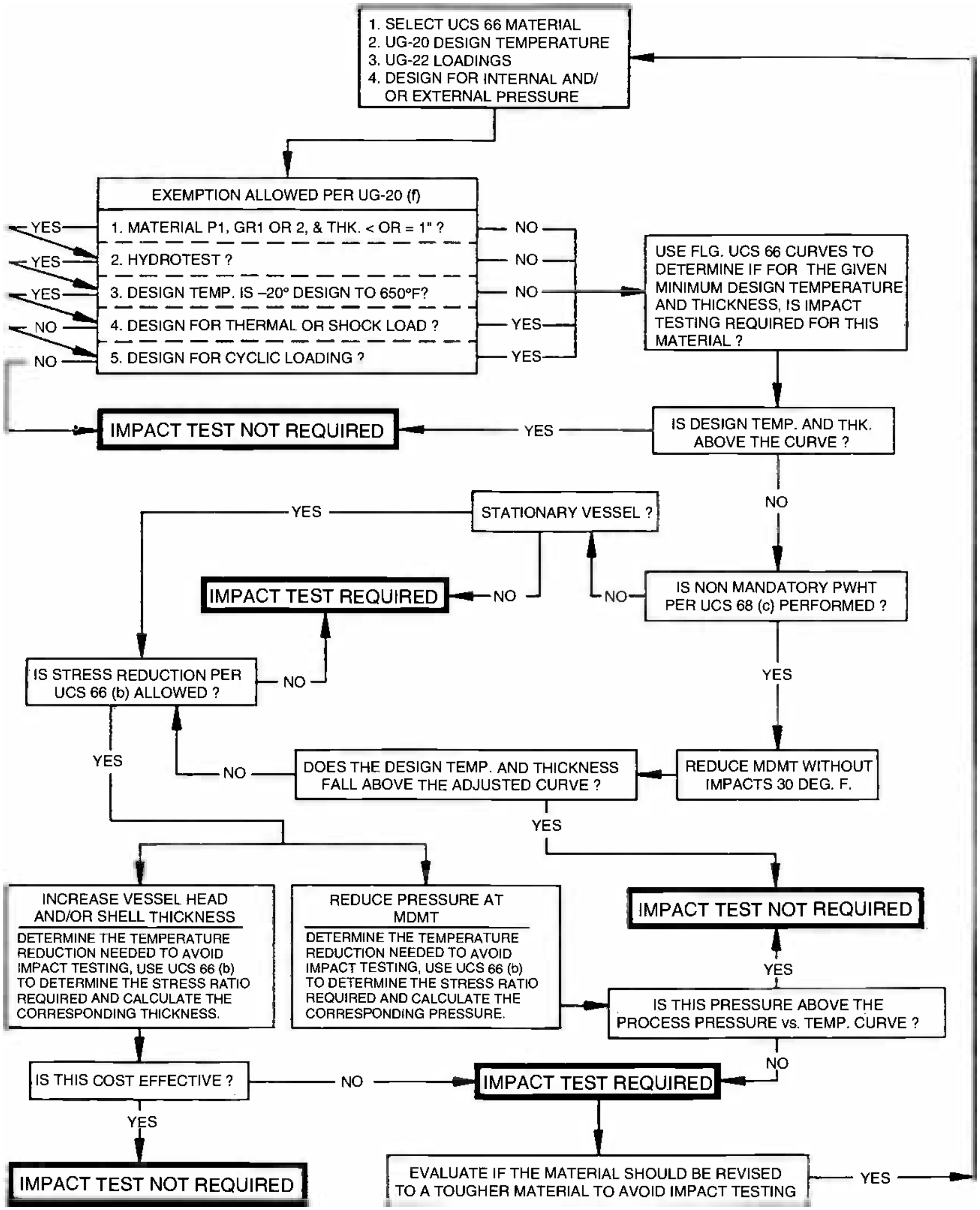


Figure 2-44. Flow chart showing decision-making process to determine MDMT and impact-testing requirements.

Procedure 2-15: Buckling of Thin Wall Cylindrical Shells [21]

This section provides commentary on the buckling of cylinders subject to external pressure, uniform axial compression, a bending moment across the cross-section, and in-plane shear stresses. Cylinders may also be subject to any combination of these loads. Unstiffened or large ring-stiffened cylinders may fail by local buckling or column buckling, and small ring-stiffened cylinders may fail by local buckling, column buckling, general instability, or local buckling of ring stiffeners. The rings described herein are assumed to be circumferential ring stiffeners. The terms 'large' and 'small' are consistent with Code Case 2286 (and the incorporated Code Case 2286 methodology in Section VIII, Division 2, Part 4), where the term 'large' refers to a bulkhead where the number of circumferential lobes is assumed to be two ($n = 2$) and the term 'small' refers to any larger number of circumferential lobes.

Local buckling describes failure by buckling of the cylinder in a radial direction. Column buckling for a cylinder describes failure by out-of-plane buckling while the shape of the cross section prior to buckling is circular. General instability describes failure where one or more ring stiffeners buckles along with the cylinder in a circumferential pattern with at least two waves. Local buckling of ring stiffeners refers to buckling of elements of the ring stiffener.

Note that an element in compression subject to 1) elastic buckling, will buckle before the element can develop the yield stress, 2) inelastic buckling, will buckle after some of the element develops the yield stress, and 3) plastic collapse, will collapse after all of the element develops the yield stress.

External Pressure

Buckling of the cylinder due to external pressure may occur in the elastic, inelastic, or plastic range. It is a function of L/r_g and D/t ratios as well as the physical properties of the material. Decreasing the effective length below the critical length has a positive effect the strength of the cylinder under external pressure. The critical length is the length at which the unstiffened cylinder

Decreasing the effective length may be done with ring stiffeners. Where cylinders with 'large' stiffening rings are used, the large rings will act as bulkheads whereby the shell portion between the rings acts similar to an unstiffened cylinder. In this case the large rings are assumed not to buckle with the shell. In general, where cylinders with 'small' stiffening rings are used the failure mode of general instability is introduced. Specifically within the Code (Code Case 2286 and Section VIII, Division 2, Part 4), the general instability failure mode is precluded by increasing the required moment of inertia of a small ring by a value of 20%. The difference between the large and small rings in Code Case 2286 is seen by using the more general equation representing the small ring, and using a value of $n = 2$ which yields the equation for the large ring. Figure 2-45 illustrates the meaning of circumferential buckling waves, where the circle represents the original shape.

Local buckling of ring stiffeners may be accomplished by using compact shapes for ring elements. Code Case 2286 and Section VIII, Division 2, Part 4 include geometry requirements (used from AISC) to ensure local buckling of ring stiffeners is avoided.

Axial Compression

Cylinders subject to uniform axial compression can fail by global or local buckling in the elastic, inelastic, and plastic range. Global buckling is usually determined by the length to radius of gyration ratio (L/r_g), and local buckling is determined by the diameter to thickness ratio (D/t). Each of these can be either elastic or inelastic. In some cases, circumferential ring stiffeners can be used on cylinders, but these are used mostly when external pressure is a concern. They may have a positive impact on axial compression if they are close enough to each other. Code Case 2286 and Section VIII, Division 2, Part 4 indicate that if the rings are less than $15(R_o t)^{1/2}$ from each other, then the circumferential rings are permitted to increase the calculated local buckling strength due to axial compression and bending. Longitudinal, or stringer, stiffeners may be used to increase the allowable axial capacity of the cylinder; however these are not typically used in process plants and are more common in offshore platform supports.

Cylinders under axial compression are more sensitive to geometric imperfections in the elastic range than in the

Figure 2-45. Circumferential buckling waves for $n = 2$ (left) and $n = 6$ (right)

inelastic range. For Code Case 2286, the reduction factor for shape imperfections is 0.207 for D_o/t values of greater than or equal to 1,247. The factor becomes higher for lower D_o/t ratios. The capacity reduction factors are already built into the allowable stress equations in Code Case 2286.

Bending Moment

Cylinders subject to a bending moment across the cross-section have similar characteristics as the case of a cylinder under uniform axial compression. Ovaling of the cross-section occurs during bending however as outlined in Code Case 2286 the allowable stress for a cylinder in bending is greater than a cylinder under uniform axial compression for the same geometry.

Shear

Cylinders subject to in-plane shear stresses can also fail in the elastic, inelastic, and plastic range. Though shear buckling is rarely a controlling factor in the design of process vessels, the interaction may have an unfavorable effect.

Interaction

The effect of combining internal/external pressure with axial tension/compression may be represented by an ellipse created using a yield criterion. In the four quadrants created by the yield criterion, four combinations of the internal/external pressure with axial tension/compression may be represented. In the case of external pressure, quadrants three and four (external pressure and axial compression, and external pressure and axial tension) are representative of what is discussed here. Furthermore, only quadrant three is evaluated in Code Case 2286 and Section VIII, Division 2, Part 4 since the topic of internal pressure is not addressed.

Procedure 2-16: Optimum Vessel Proportions [16-20]

This procedure specifically addresses drums but can be made applicable to any kind of vessel. The basic question is: What vessel proportions, usually expressed as L/D ratio, will give the lowest weight for a given volume? The maximum volume for the least surface area, and weight, is of course a sphere. Unfortunately, spheres are generally more expensive to build. Thus, spheres are not the most economical option until you get to very large volumes and for some process applications where that shape is required.

For vessels without pressure, atmospheric storage vessels, for example, the optimum L/D ratio is 1, again using the criteria for the maximum volume for the minimum surface area. This optimum L/D ratio varies with the following parameters:

Pressure.
Allowable stress.
Corrosion allowance.
Joint efficiency.

In *Process Equipment Design*, Brownell and Young suggest that for vessels less than 2 in. in thickness, the optimum L/D ratio is 6 and for greater thicknesses is 8. However, this does not account for the parameters just shown. Others have suggested a further breakdown by pressure categories:

Pressure (PSIG)	L/D Ratio
0–250	3
250–500	4
>500	5

Although this refinement is an improvement, it still does not factor in all of the variables. But before describing the actual procedure, a brief description of the sizing of drums in general is warranted. Here are some typical types of drums:

Knock-out drums.
Accumulator drums.
Suction drums.
Liquid–vapor separators.
Liquid–liquid separators.
Storage vessels.
Surge drums.

Typically the sizing of drums is related to a process consideration such as liquid holdup (surge), storage volume, or velocity considerations for separation. Surge volume in process units relates to the response time required for the alarms and operators to respond to upstream or downstream conditions.

For small liquid holdup, vessels tend to be vertical, while for large surge volumes they tend to be horizontal. For small volumes of liquid it may be necessary to increase the L/D ratio beyond the optimum proportions to allow for adequate surge control. Thus there may be an economic L/D ratio for determining the least amount of metal for the given process conditions as well as a practical operating L/D ratio.

For liquid–vapor separators the diameter of the vessel is determined by the velocity of the product and the time it takes for the separation to occur. Baffles and demister pads can speed up the process. In addition, liquid–vapor separators must provide for minimum vapor spaces. The sizing of vessels is of course beyond this discussion and is the subject of numerous articles.

An economic L/D ratio is between 1 and 10. L/D ratios greater than 10 may produce the lowest surface-area-to-volume ratio but should be considered impractical for most applications. Obviously plot space is also a consideration in ultimate cost. In general, the higher the pressure the larger the ratio, and the lower the pressure the lower the ratio. As previously stated, the optimum L/D ratio for an atmospheric drum is 1. Average pressure vessels will range between 3 and 5.

Two procedures are included here and are called Method 1 and Method 2. The two procedures, though similar in execution, yield different results. Both methods take into account pressure, corrosion, joint efficiency, and allowable stress. Even with this much detail, it is impossible to determine exactly what proportions will yield the lowest overall cost, since there are many more variables that enter into the ultimate cost of a vessel. However, determining the lowest weight is probably the best parameter in achieving the lowest cost.

The procedure for determining the optimum L/D ratios for the two methods is as follows:

Given

V, volume
P, pressure
C, corrosion allowance
S, allowable stress
E, joint efficiency

Method 1

1. Calculate F_1.
2. From Fig. 2-46, using F_1 and vessel volume, V, determine the vessel diameter, D.
3. Use D and V to calculate the required length, L.

Method 2

1. Calculate F_2.
2. From Fig. 2-47 determine L/D ratio.
3. From the L/D ratio, calculate the diameter, D.
4. Use D and V to calculate the required length, L.

Table 2-8
Optimum vessel proportions—comparison of two methods

V (cu. ft.)	P (PSIG)	Method[1]	D (ft)	L (ft)	t (in.)	W (lb)	L/D
1500	150	1	7.5	34	0.5625	20,365	4.5
		2	8.5	23.6	0.625	20,086	2.8
	300	1	6	53	0.8125	35,703	8.8
		2	7.5	31.5	0.8125	28,668	4.2
2000	150	1	7	52	0.5	25,507	7.4
		2	9	28.4	0.625	24,980	3.2
	300	1	6.5	61	0.875	51,179	9.4
		2	8.5	32.4	1.125	39,747	3.8
3000	150	1	8.5	53	0.625	40,106	6.3
		2	10.5	31.1	0.6875	35,537	3
	300	1	7.5	68	0.9375	65,975	9.1
		2	9.5	39.2	1.25	69,717	4.1
5000	150	1	10	64	0.6875	62,513	6.4
		2	11.5	44.3	1.125	86,781	3.9
	300	1	8.5	88	1.125	107,861	10.4
		2	11.5	44.3	1.375	106,067	3.9

[1]*Methods are as follows, based on graphs: Method 1: K. Abakians,* Hydrocarbon Processing*, June 1963. Method 2: S.P. Jawadekar,* Chemical Engineering*, Dec. 15,1980.*

Optimum Vessel Proportions for Vessels with (2) 2:1 S.E. Heads

Notation

V = vessel volume, cu ft
P = internal pressure, PSIG
L = length, T–T, ft
T = shell thickness, in.
W = vessel weight, lb
D = diameter, ft
C = corrosion allowance, in.
A = surface area, sq ft
F_n = vessel ratios
S = allowable stress, psi
E = joint efficiency
w = unit weight of plate, PSF
L_e = equivalent length of cylinder equal to the volume of a vessel with (2) 2:1 S.E. heads
h = height of cone, ft
R = radius, ft
C_1, K_1 = constant for ellipsoidal heads

Diameter for Different L/D Ratios

L/D	D
3	$\sqrt[3]{\frac{6V}{5\pi}}$
4	$\sqrt[3]{\frac{12V}{13\pi}}$
5	$\sqrt[3]{\frac{3V}{4\pi}}$
6	$\sqrt[3]{\frac{12V}{19\pi}}$
7	$\sqrt[3]{\frac{6V}{11\pi}}$
8	$\sqrt[3]{\frac{12V}{25\pi}}$

Equations

$$L_e = L + 0.332D$$

$$V = \frac{\pi D^3}{12} + \frac{\pi D^2 L}{4}$$

$$D = \sqrt[3]{\frac{4V}{\pi\left(0.3333 + \frac{L}{D}\right)}}$$

$$W = Aw$$

$$A = 2.18\,D^2 + \pi DL$$

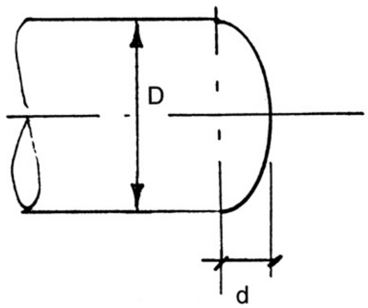
$$t = \frac{PR}{SE - 0.6P} + C$$

$$L = \frac{4V}{\pi D^2} - \frac{D}{3}$$

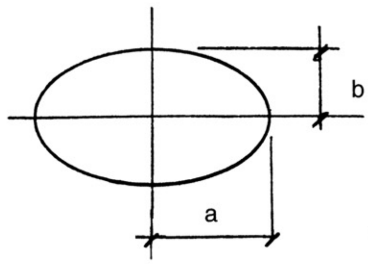
$$F_1 = \frac{P}{CSE}$$

$$F_2 = C\left(\frac{SE}{P} - 0.6\right)$$

Atmospheric Tank Proportions



Ellipsoidal Heads



Flat Elliptical Ends

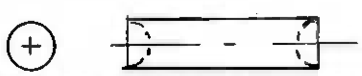
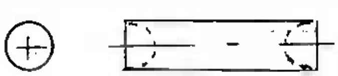
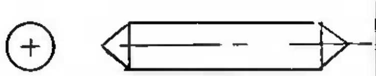
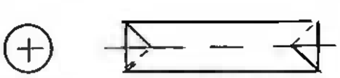
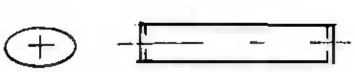
$$K_1 = \frac{2d}{R}$$

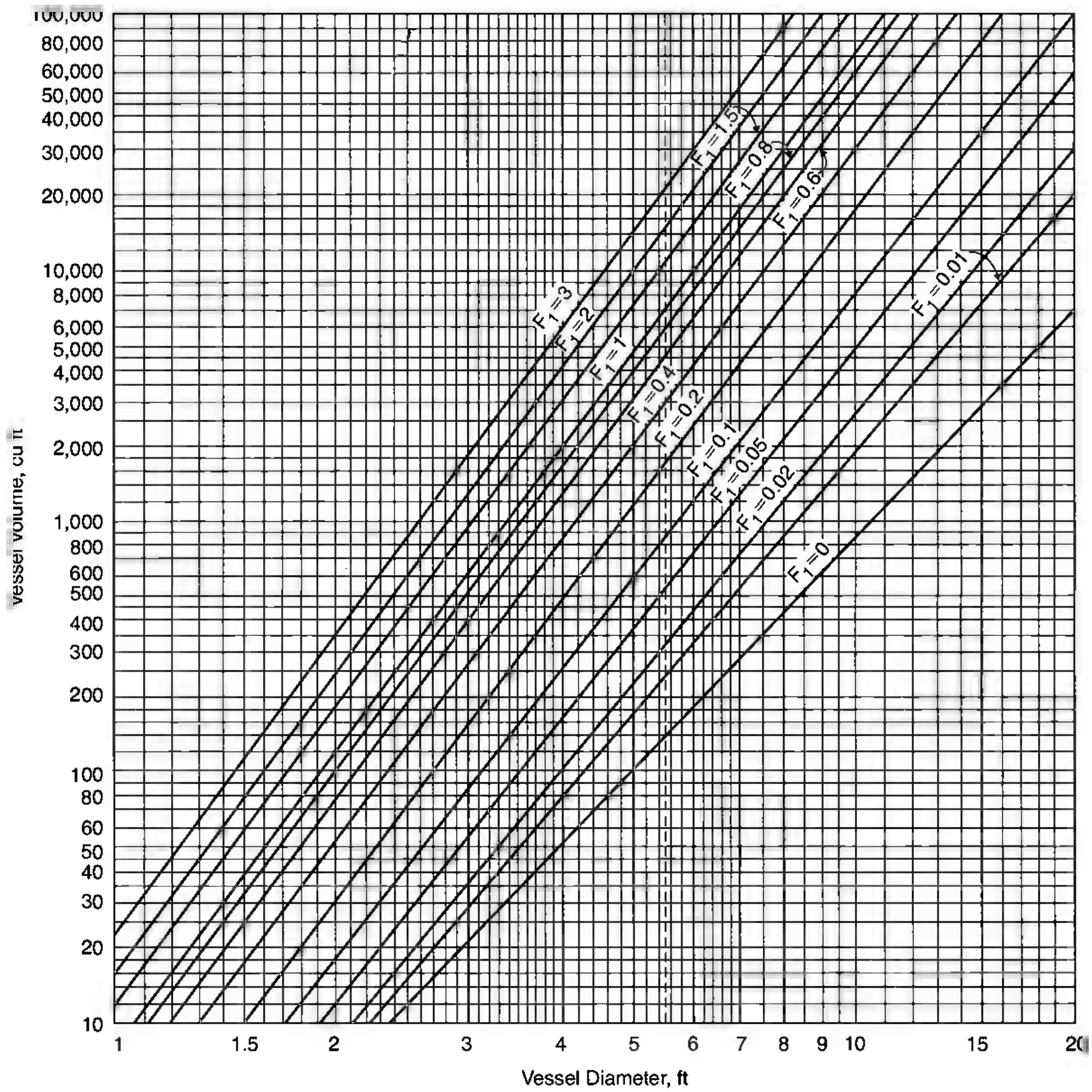
$$K_2 = \frac{b}{a}$$

$$C_1 = 2 + \frac{K_1^2}{\sqrt{1 - K_1^2}} \ln\left(\frac{1 + \sqrt{1 - K_1^2}}{1 - \sqrt{1 - K_1^2}}\right)$$

Note: For 2:1 S.E. Heads, $C_1 = 2.76$ and $K_1 = 0.5$.

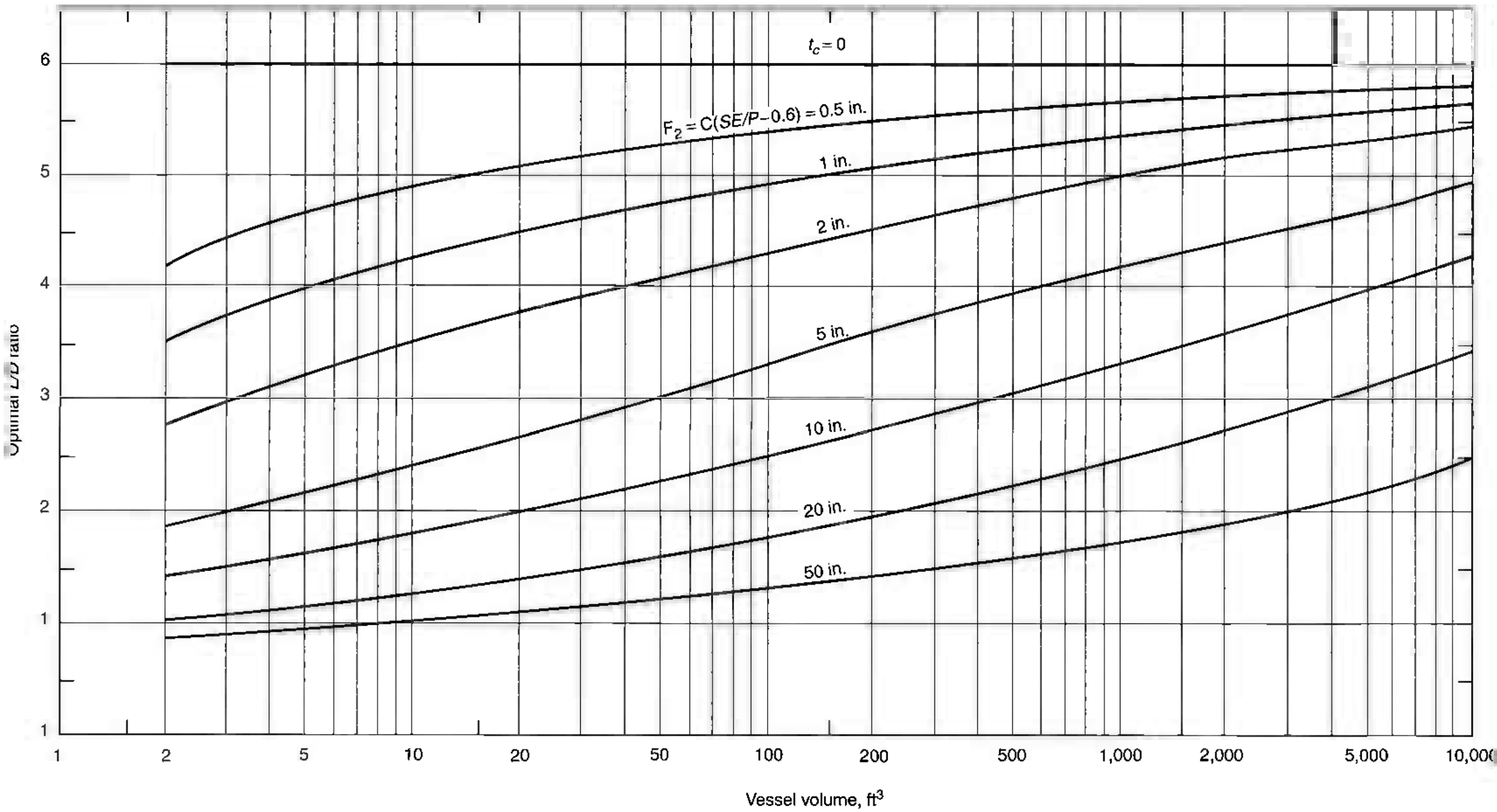
Table 2-9
Optimum tank proportions

Case	Optimum Proportions	Volume
Cylinder with flat ends 	$L = D$	$2\pi R^3$
Cylinder with ellipsoidal heads 	$L = R(C_1 + 4K_1)$	$\pi R^3\left(\frac{3C_1 - 8K_1}{3}\right)$
Cylinder with internal ellipsoidal heads 	$L = R(C_1 + 4K_1)$	$\pi R^3\left(\frac{3C_1 + 8K_1}{3}\right)$
Cylinder with internal hemi-heads 	$L = 8R$	$6.66\pi R^3$
Cylinder with conical ends 	$h = 0.9R$ $L = 0.9R$	$1.5\pi R^3$
Cylinder with internal conical ends 	$h = 0.9R$ $L = 3.28R$	$2.68\pi R^3$
Elliptical tank with flat ends 	$L = 2K_2 a\sqrt{\frac{2}{1 + K_2^2}}$	$2K_2^2\pi a^3\sqrt{\frac{2}{1 + K_2^2}}$



(From K. Abakians, *Hydrocarbon Processing and Petroleum Refiner*, June 1963.)

Figure 2-46. Method 1: Chart for determining optimum diameter.



(From S.P. Jawadekar, *Chemical Engineering*, Dec. 15, 1980.)

Figure 2-47. Method 2: Chart for determining the optimum L/D ratio.

Procedure 2-17: Estimating Weights of Vessels and Vessel Components

Estimating of weights of vessels is an important aspect of vessel engineering. In the conceptual phase of projects, weights are estimated in order to determine costs and budgets for equipment, foundations, erection, and transportation. Estimated weights also help to get more accurate bids from suppliers. Accurate weights are necessary for the design of the vessel itself to determine forces and moments.

There are a number of different types of weights that are calculated. Each weight is used for different purposes.

1. ***Fabricated weight***: Total weight as fabricated in the shop.
2. ***Shipping weight***: Fabrication weight plus any weight added for shipping purposes, such as shipping saddles.
3. ***Erection weight***: Fabrication weight plus any weight installed for the erection of the equipment, such as any insulation, fireproofing, piping, ladders, platforms.
4. ***Empty weight***: The overall weight of the vessel sitting on the foundation, fully dressed, waiting for operating liquid.
5. ***Operating weight***: Empty weight plus any operating liquid weight.
6. ***Test weight***: This weight can be either shop or field test weight, that is, the vessel full of water.

There are a number of ways to estimate the weights of vessels, depending on the accuracy required. Vessel weights can be estimated based on computer design programs. These programs typically calculate the volume of metal for the vessel shell and head and add weights for supports, nozzles, trays, and other components. Another fast and easy way to get the volume of metal in the shell and heads is to use the surface area in square feet and multiply this by the unit weight for the required thickness in pounds per square foot.

In addition to the base weight of metal in the shell and heads, the designer must include an allowance for plate overages per Table 2-11. The mill never rolls the plates the exact specified thickness since there would be the possibility of being below thickness. The safety margin added by the mill is referred to as *plate overage* or *overweight percentage*. The plate overage varies by the thickness of the material.

In addition to the plate overage, the fabricator (or head manufacturer) also adds a *thinning allowance* to the head to ensure that the head meets the minimum thickness in all areas. Depending on the type of head, the diameter, and the thickness required, a thinning allowance can be determined. This can be as much as 1.5 in. for large-diameter hemi-heads over 4 in. thick! The metal does not disappear during the forming process but may "flow" to the areas of most work.

On a typical spun 2:1 S.E. head, the straight flange will get thicker and the knuckle will get thinner due to forming. The crown of the head should remain about the same. Therefore the completed head has a thickness averaging the initial thickness of the material being formed.

After the weights of all the components are added for a total weight, an additional percentage is typically added to allow for other components and welding. The typical percentages are as follows:

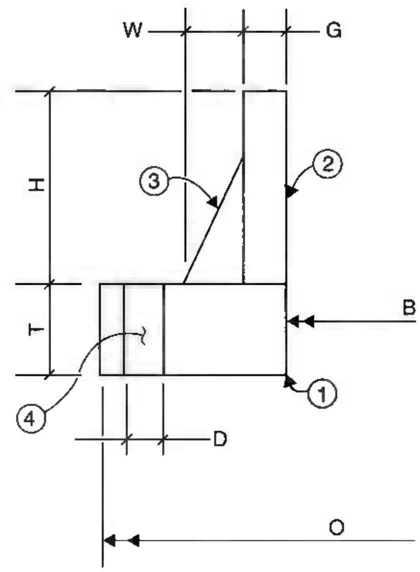
$<$50,000 lb	Add 10%
50,000-75,000 lb	Add 8%
75,000-100,000 lb	Add 6%
$>$100,000 lb	Add 5%

The weight of any individual component can easily be calculated based on the volume of the material times the unit density weight given in Table 2-11. Any shape can be determined by calculating the surface area times the thickness times the density. The designer need only remember the density of steel for most vessels of 0.2833 lb/in.^{3} to determine any weight. For vessels or components of other materials, either the density of that material or the factor for that material relative to carbon steel can be used. These values are also listed in the following tables.

Table 2.10
General formulas for computing weights of vessel components

ITEM		WEIGHT FORMULAS	THICKNESS FORMULAS	DATA
SHELL	Per Ft	$10.68\ D_m\ t$	ASME Section VIII, Div 1	D = ID of shell or cone, in
	C.S.	$.89\ D_m\ L\ t$	SHELL;	d = ID, small end of cone, in
	Other	$\pi\ D_m\ L\ t\ \delta$	$t = (P\ R_i) / (S\ E - .6\ P)$	D_m = Mean diameter of shell, in
SPHERE	C.S.	$.89\ D_m^2\ t$		t = Thickness, in
	Other	$\pi\ D_m^2\ t\ \delta$	2:1 S.E. HEAD;	A_c = Area of cone, in^2
HEMI	C.S.	$.445\ D_m^2\ t$	$t = (P\ R_i) / (S\ E - .2\ P)$	L = Length of shell or cone, in
	Other	$1.57\ D_m^2\ t\ \delta$		δ = Density of material, PCI
2:1 SE HEAD	C.S.	$.307\ D_m^2\ t$	HEMI HEAD;	R = Estimated radius of hemi head, in
	Other	$1.084\ D_m^2\ t\ \delta$	$t = (P\ R_i) / (2\ S\ E - .2\ P)$	R_i = Inside radius, in
CONE	C.S.	$.2833\ A_c\ t$		S = Allowable stress, PSI
	Other	$A_c\ t\ \delta$	CONE;	P = Internal pressure, PSIG
			$t = (P\ R_i) / [\cos\alpha\ (S\ E - .6\ P)]$	E = Joint efficiency
			ASME Section VIII, Div 2	α = Half apex angle of cone, degrees
			SHELL;	**FORMULAS**
			$t = R_i\ (e^{P/S} - 1)$	$D_m = D + t$ Use shell thickness in formula
NOTES			HEMI HEAD;	$R = .5\ D_m - .25\ t$
1. Add thinning allowance to all head thickness			$t = R_i\ (e^{.5P/S} - 1)$	For 30° cone, $A_c = 1.57\ (D^2 - d^2)$
				For all other cones; $A_c = \pi\ [.5\ (D + d)]\ [L^2 + (.5\ (D - d)^2)]^{1/2}$

Calculation of Weight of Weld Neck Flange



Data

T = thickness of flange
O = flange OD
D = bolt hole diameter
H = hub height
G = hub thickness at small end
W = width of hub
B = ID of flange
V = volume, in.^{3}
d = density of material, lb/in.^{3}
N = number of bolts/holes

Formulas

1. $\left[\frac{O^2\pi}{4} - \frac{B^2\pi}{4}\right]T = (+)$
2. $\{B + G\}\pi GH = (+)$
3. $0.5\{(B + 2G + W)\pi WH\} = (+)$
4. $\left[\frac{D^2\pi}{4}\right]T\,N = (-)$
5. $V = 1 + 2 + 3 - 4 =$
6. weight $= V \times d =$

Table 2-11
Weights of carbon steel plate and stainless steel sheet, PSF

Thickness (in.)	Raw Weight	Weight Including % Overweight	% Overweight	Thickness (in.)	Raw Weight	Weight Including % Overweight	% Overweight
0.125	5.1	5.65	10.75	0.875	35.7	36.91	3.38
0.1875	7.66	8.34	9	0.9375	38.28	39.54	3.38
0.25	10.2	10.97	7.5	1	40.8	42.02	3
0.3125	12.76	13.61	6.75	1.0625	43.38	44.65	3
0.375	15.3	16.22	6	1.125	45.94	47.28	3
0.4375	17.86	18.79	5.25	1.25	51	52.53	3
0.5	20.4	21.32	4.5	1.375	56.15	57.78	3
0.5625	22.97	23.98	4.5	1.5	61.2	63.04	3
0.625	25.6	26.46	3.75	1.625	66.35	68.29	3
0.6875	28.07	29.1	3.75	1.75	71.4	73.54	3
0.75	30.6	31.63	3.38	1.875	76.56	78.8	3
0.8125	33.17	34.27	3.38	2	81.6	84.05	3
Stainless Steel Sheet							
Thickness Gauge	Weight	Thickness Gauge	Weight				
10 GA	5.91	20 GA	1.58				
11 GA	5.25	24 GA	1.05				
12 GA	4.59	26 GA	0.788				
14 GA	3.28	28 GA	0.656				
16 GA	2.63	30 GA	0.525				
18 GA	2.1						

Note*: % Overweight is based on standard mill tolerance added to the thickness of plate to guarantee minimum thickness.*

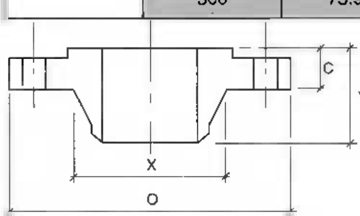
Table 2-12
Weights of flanges, 2 in. to 24 in. (lb)

Size (in.)	Rating					
	150	300	600	900	1500	2500
2	9	10	12	25	25	42
	4	*8*	*10*	*25*	*25*	*39*
3	14	16	20	32	48	94
	14	*16*	*20*	*32*	*48*	*86*
4	16	26	41	51	73	145
	19	*27*	*41*	*54*	*73*	*135*
6	25	45	77	110	164	380
	28	*50*	*86*	*113*	*159*	*345*
8	40	70	111	187	273	580
	48	*80*	*140*	*197*	*301*	*530*
10	56	94	180	268	454	1075
	70	*120*	*230*	*290*	*507*	*1025*
12	86	140	226	372	670	1525
	105	*184*	*295*	*413*	*775*	*1300*
14	111	190	334	562	940	
	135	*249*	*378*	*494*	*975*	
16	141	250	462	685	1250	
	176	*324*	*527*	*619*	*1300*	
18	153	305	531	924	1625	
	214	*416*	*665*	*880*	*1750*	
20	188	380	678	1164	2050	
	284	*516*	*855*	*1107*	*2225*	
24	270	540	959	2107	3325	
	398	*763*	*1175*	*2099*	*3625*	

Notes:
1. Top value in block is the weight of a weld neck flange.
2. Bottom value in block is the weight of a blind flange.

Table 2-13
Dimensions and weights of large-diameter flanges, 26 in. to 60 in., ASME B16.47, Series B

Size (in.)		Dimensions						Weight	
		O	C	Y	X	N	d	RFWN	Blind
26	150	30.94	1.62	3.5	29.64	36	0.75	120	340
	300	34.12	3.5	5.69	27.62	32	1.25	400	860
28	150	32.94	1.75	3.75	28.94	40	0.75	140	415
	300	36.25	3.5	5.88	29.75	36	1.25	450	970
30	150	34.94	1.75	3.94	31	44	0.75	150	470
	300	39	3.69	6.22	32	36	1.375	550	1250
32	150	37.06	1.81	4.25	33.06	48	0.75	170	550
	300	41.5	4.06	6.62	34	32	1.5	685	1550
34	150	39.56	1.94	4.34	35.12	40	0.875	210	660
	300	43.62	4.06	6.81	36.12	36	1.5	750	1635
36	150	41.62	2.06	4.62	37.19	44	0.875	240	780
	300	46.12	4.06	7.12	38	32	1.625	840	1835
38	150	44.25	2.12	4.88	39.12	40	1	290	905
	300	48.12	4.38	7.56	40	36	1.625	915	2150
40	150	46.25	2.19	5.06	41.31	44	1	310	1025
	300	50.12	4.56	7.81	42	40	1.625	990	2425
42	150	48.25	2.31	5.25	43.38	48	1	345	1175
	300	52.5	4.69	8.06	44	36	1.75	1135	2745
44	150	50.25	2.38	5.38	45.38	52	1	370	1310
	300	54.5	5	8.44	46.19	40	1.75	12635	3150
46	150	52.81	2.44	5.69	47.44	40	1.125	435	1490
	300	57.5	5.06	8.75	43.38	36	1.875	1470	3560
48	150	54.81	2.56	5.88	49.5	44	1.125	480	1680
	300	59.5	5.06	8.81	50.31	40	2	1575	3850
50	150	56.81	2.69	6.06	51.5	48	1.125	520	1900
	300	61.5	5.44	9.25	52.38	44	2	1710	4365
52	150	58.81	2.75	6.19	53.56	52	1.125	550	2080
	300	63.5	5.62	9.56	54.44	48	2	1840	4800
54	150	61	2.81	6.38	55.62	56	1.125	620	2290
	300	65.88	[illegible]38	9.44	56.5	48	2	1980	4965
56	150	63	2.88	6.56	57.69	60	1.125	680	2500
	300	69.5	6.06	10.56	58.81	36	2.375	2600	6240
58	150	65.94	2.94	6.88	59.69	48	1.25	830	2800
	300	71.94	6.06	10.81	60.94	40	2.375	2770	6675
60	150	67 94	3	7.06	61.81	52	1.25	1075	3030
	300	73.94	5.94	10.69	62.94	40	2.375	2870	6930



d = bolt hole diameter, in.
N = number of bolt holes

Table 2-14
Weights of nozzles and manways, 1 in. to 60 in.

Rating Size	150	300	600	900	1500	2500
1″	4	5	6	11	13	18
1.5″	6	9	11	16	21	34
2″	8	10	13	27	42	47
3″	15	21	22	34	78	110
4″	21	31	42	60	110	160
6″	37	55	81	127	215	360
8″	54	81	132	207	335	520
10″	72	116	215	310	650	1000
12″	107	158	261	418	940	1350
14″	132	232	407	613	950	
16″	163 **400**	289 **705**	549 **1260**	751 **1610**	1175 **3250**	
18″	200 **479**	340 **875**	639 **1530**	1042 **2270**	1475 **5200**	
20″	235 **593**	421 **1065**	783 **1925**	1283 **2800**	1725 **5430**	
24″	310 **825**	587 **1600**	1100 **2685**	2287 **5455**	2650 **9000**	

Rating Size	150	300
26″	230	490
28″	260	550
30″	280 **1126**	665 **2880**
32″	305	805
34″	345	880
36″	385 **1726**	980 **3685**
38″	440	1045
40″	465	1125
42″	510 **2387**	1275 **4600**
44″	540	1365
46″	610	1620
48″	660 **2970**	1775 **5515**
50″	705	1915
52″	740	2025
54″	800	2170
56″	835	2790
58″	970	2970
60″	1050 **5760**	3080 **8675**

Notes:

1. Weights include pipe and WN flg.
2. Lower weight in box is weight of manway and includes nozzle, blind, and bolts.
3. Class 1500 manways are based on LWN.

Table 2-15
Weights of valve trays, PSF

Dia.	One Pass		Two Pass		Four Pass	
	C.S.	Alloy	C.S.	Alloy	C.S.	Alloy
<84″	13	11	14	12		
84″ to 180″	12	10	13	11	15	13
>180″	11.5	9.5	12.5	10.5	14.5	12.5

Notes:

1. Compute area on total cross-sectional area of vessel. The downcomer areas compensate for the weight of downcomers themselves.
2. Tray weights include weights of trays and downcomers.

Table 2-16
Weights of tray supports and downcomer bars (lb)

ID (in.)	C.S.	Alloy	ID (in.)	C.S.	Alloy	ID (in.)	C.S.	Alloy
30	25	17	102	113	72	174	287	174
36	28	19	108	119	75	180	294	178
42	34	23	114	123	77	186	344	207
48	37	25	120	176	108	192	354	212
54	44	35	126	183	112	198	362	218
60	47	38	132	188	116	204	374	226
66	50	40	138	195	119	210	385	231
72	53	44	144	202	122	216	396	239
78	55	46	150	244	149	222	407	245
84	99	62	156	251	152	228	418	252
90	103	65	162	271	162	234	428	259
96	109	68	168	278	167	240	440	265

Notes:

1. Tray support weights include downcomer bolting bars as well.
2. Tray support ring sizes are as follows:

   C.S.: 1/2″ × 2 1/2″
   Alloy: 5/16″ × 2 1/2″

Table 2-17
Thinning allowance for heads

Thickness	Diameter	
	<150″	>150″
0.125″ to 1″	0.0625	None
1″ to 2″	0.125	0.25
2″ to 3″	0.25	0.25
3″ to 3.75″	0.375	0.375
3.75″ to 4″	0.5	0.5
over 4.25″	0.75	0.75

Table 2-18
Weights of pipe (PLF)

Size (in.)	Schedule												
	10	20	30	STD	40	60	XS	80	100	120	140	160	XXS
0.75	0.8572			1.131	1.131		1.474	1.474				1.937	2.441
1	1.404			1.679	1.679		2.172	2.172				2.844	3.659
1.25	1.806			2.273	2.273		2.997	2.997				3.765	5.214
1.5	2.085			2.718	2.718		3.631	3.631				4.859	6.408
2	2.638			3.653	3.653		5.022	5.022				7.444	9.029
2.5	3.531			5.793	5.793		7.661	7.661				10.01	13.69
3	4.332			7.576	7.576		10.25	10.25				14.32	18.58
3.5	4.973			9.109	9.109		12.5	12.5				17.69	22.85
4	5.613			10.79	10.79	12.66	14.98	14.98		19		22.51	27.54
5	7.77			14.62	14.62		20.78	20.78		27.04		32.96	38.55
6	9.289		17.02	18.97	18.97		28.57	28.57		36.39		45.3	53.16
8	13.4	22.36	24.7	28.55	28.55	35.64	43.39	43.39	50.87	60.63	67.76	74.69	72.42
10	18.2	28.04	34.24	40.48	40.48	54.74	54.74	64.33	76.93	89.2	104.1	115.6	
12	24.2	33.38	43.77	49.56	53.52	73.16	65.42	88.51	107.2	125.5	139.7	160.3	
14	36.71	45.68	54.57	54.57	63.37	84.91	72.09	106.1	130.7	150.7	170.2	189.1	
16	42.05	52.36	62.58	62.58	82.77	107.5	82.77	136.5	164.8	192.3	223.5	245.1	
18	47.39	59.03	82.06	70.59	104.6	138.2	93.45	170.8	208	244.1	274.2	308.5	
20	52.73	78.6	104.1	78.6	122.9	166.4	104.1	208.9	256.1	296.4	341.1	379	
22	58.1			86.6			114.8						
24	63.41	94.62	140.8	94.62	171.2	238.1	125.5	296.4	367.4	429.4	483.1	541.9	
26				102.6			136.2						
28				110.7			146.8						
30	98.9	157.6	196.1	118.7			157.6						
32				126.7			168.2						
34				134.7			178.9						
36				142.7			189.6						
42				166.7			221.6						

Table 2-19
Weights of alloy stud bolts + (2) nuts per 100 pieces

Length (in.)	Stud Diameter, in.												
	0.5	**0.625**	**0.75**	**0.875**	**1**	**1.125**	**1.25**	**1.375**	**1.5**	**1.625**	**1.75**	**1.875**	**2**
3	29	49	76										
3.25	30	51	79										
3.5	31	53	82	120									
3.75	32	55	85	124									
4	34	57	88	128	188								
4.25	35	59	91	132	194								
4.5	36	61	94	136	199	246							
4.75	37	63	97	140	205	253							
5	39	65	100	144	210	259	330						
5.25	40	67	103	148	216	266	338						
5.5	41	69	106	152	221	272	347						
5.75		71	109	156	227	279	355						
6		73	112	160	232	285	363	460	568	700			
6.25			115	164	238	292	371	470	580	714			
6.5			118	168	243	298	380	480	592	728			
6.75				172	249	305	388	490	604	742			
7				176	254	311	396	500	616	756	900	1062	1227
7.25					260	318	404	510	628	770	916	1080	1248
7.5					265	324	413	520	640	784	932	1098	1270
7.75					271	331	421	530	652	798	948	1116	1291
8					276	337	429	540	664	812	964	1134	1312
8.25						344	437	550	676	826	980	1152	1334
8.5						350	446	560	688	840	996	1170	1355
8.75						357	454	570	700	854	1012	1188	1376
9						363	462	580	712	868	1028	1206	1398
9.25						370	470	590	724	882	1044	1224	1419
9.5						376	479	600	736	896	1060	1242	1440
9.75						383	487	610	748	910	1076	1260	1462
10						389	495	620	760	924	1092	1278	1483
10.25								630	772	938	1108	1296	1508
10.5								640	784	952	1124	1314	1526
10.75								650	796	966	1140	1332	1547
11								660	808	980	1156	1350	1569
11.25								670	820	994	1172	1368	1590
11.5								680	832	1008	1188	1385	1611
11.75								690	844	1022	1204	1404	1633
12								700	856	1036	1220	1422	1654
Add per additional 1/4″ length	1.5	2	3	4	5.5	6.5	8.5	10	12	14	16	18	21.5

Table 2-20
Weights of saddles and baseplates (lb)

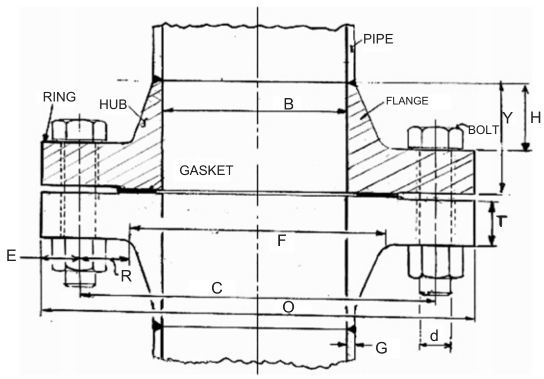
ID (in.)	Two CS Saddles	1/2 × 6 Baseplate	3/4 × 8 Baseplate	ID (in.)	Two CS Saddles	1/2 × 6 Baseplate	3/4 × 8 Baseplate
24	100	70	150	138	3060	390	790
30	150	90	190	144	3400	410	820
36	260	105	215	150	3700	430	855
42	330	125	250	156	4000	450	885
48	380	140	285	162	4250	460	920
54	440	160	320	168	4500	480	950
60	510	170	350	174	4750	490	985
66	590	190	385	180	5000	510	1020
72	680	200	420	186	5250	530	1050
78	910	220	450	192	5500	540	1080
84	1050	240	485	198	5750	560	1120
90	1160	260	520	204	6000	580	1150
96	1230	280	550	210	6250	590	1190
102	1730	290	585	216	6500	610	1220
108	1870	310	615	222	6750	630	1250
114	2330	330	650	228	7000	650	1290
120	2440	340	690	238	7250	660	1320
126	2700	360	720	240	7500	680	1360
132	2880	380	755				

Table 2-21
Density of various materials

Material	d (lb/in.^{3})	PCF	Weight Relative to C.S.
Steel	0.2833	490	1.00
300 SST	0.286	494	1.02
400 SST	0.283	489	0.99
Nickel 200	0.321	555	1.13
Permanickel 300	0.316	546	1.12
Monel 400	0.319	551	1.13
Monel 500	0.306	529	1.08
Inconel 600	0.304	525	1.07
Inconel 625	0.305	527	1.08
Incoloy 800	0.287	496	1.01
Incoloy 825	0.294	508	1.04
Hastelloy C4	0.312	539	1.10
Hastelloy G30	0.297	513	1.05
Aluminum	0.098	165	0.35
Brass	0.297	513	1.05
Cast iron	0.258	446	0.91
Ductile iron	0.278	480	0.98
Copper	0.322	556	1.14
Bronze	0.319	552	1.13

SPECIAL DESIGN RFWN FLANGES FOR CLASS 2500#

Dimensions for Weld Neck Flange

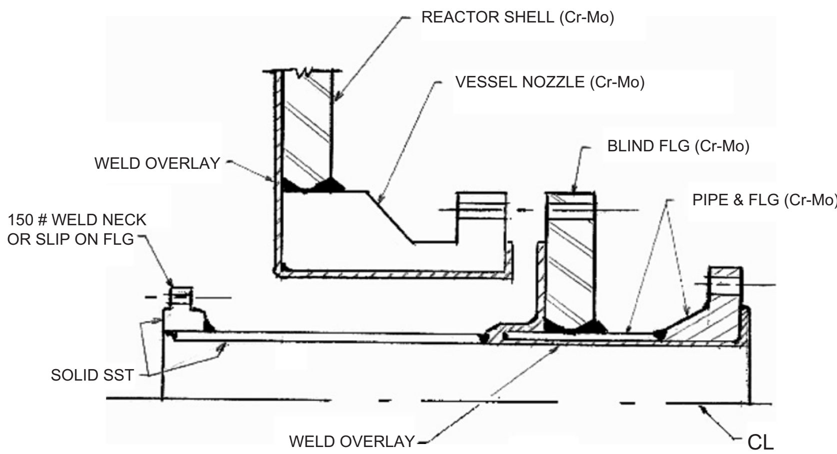


Notes:

1. These dimensions and weights should be used for estimating purposes only.
2. ASME B16.5 dimensions for 2500# flanges stop at 12" NPS. These dimensions should be considered an extension of the B16.5 size range.
3. These dimensions can be used as a starting place for special designed flanges.

SIZE B	O	T	Y	C	N	d	G	E	R	F	H	WT
14	34	7.5	18.75	27	16	2.75	1.406	2.63	3.38	20.25	11	1750
16	38	7.75	19.5	30	16	3	1.593	2.88	3.63	22.75	11.5	2250
18	41	8.25	20.5	34	20	3	1.781	2.88	3.63	26.75	12	2850
20	45	8.5	22	39.25	24	3	1.93	2.88	3.63	32	13.25	3800
24	52	9.5	24	45	20	3.5	2.34	3.5	4	37	14.25	5500

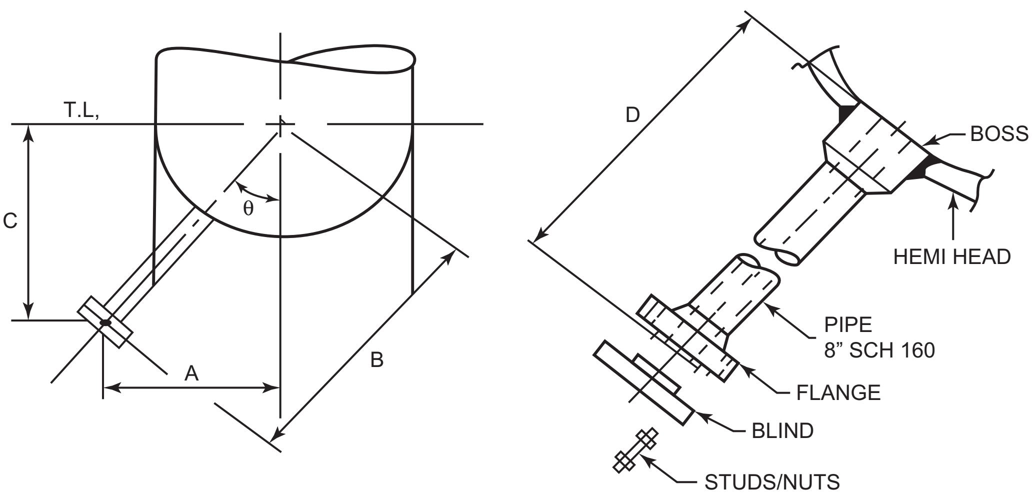
WEIGHTS OF QUENCH NOZZLES



SIZE	RATING	B	O	W_N	W_b	W_S	W_f	W_p	W_T
10" X 4"	600	10	9	433	230	70	41	39	815
	900			558	290	93	51		1030
	1500			888	507	181	73		1690
	2500			1625	1025	236	145		3070
12" X 6"	600	12	11	533	295	89	77	70	1065
	900			715	413	122	110		1425
	1500			1683	775	302	164		3000
	2500			1995	1300	548	380		4300
14" X 8"	600	14	13.5	631	378	116	111	115	1350
	900			817	494	157	187		1770
	1500			1929	975	383	273		3675
	2500			2400	1600	780	580		7475

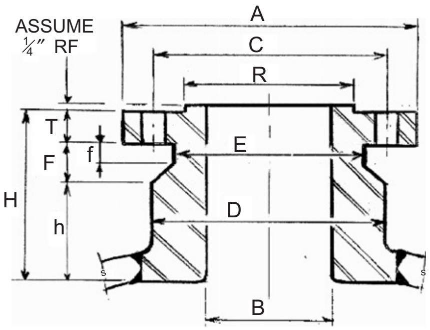
DATA	NOTES
W_N = Weight, Nozzle, Lbs	1) 600# & 900# assumes Type 'HB' Conn
W_b = Weight, Blind, Lbs	2) 1500# & 2500# assumes Type 'F' Conn
W_s = Weight, Studs & Nuts for Blind, Lbs	3) W_p assumes weight of 12" of Sch 160 Pipe and the 150# internal flange
W_f = Weight, Flange, Lbs	
W_p = Weight, Internal Pipe and Flange, Lbs	
W_T = Weight, Total, Lbs	
B = Bore of Vessel Nozzle, In	
O = OD of Internal Flange, In	

DIMENSIONS/WEIGHTS OF BOTTOM CATALYST DUMP NOZZLES



DIA	RATING	A	B	C	D	θ	WEIGHT	DIA	RATING	A	B	C	D	θ	WEIGHT
8	600	69	139	120	91	30	875	13	600	96	169	138	91	35	875
	900						1030		900						1030
	1500						1260		1500						1260
	2500						1825		2500						1825
9	600	76	152	132	98	30	925	14	600	101	176	144	92	35	885
	900						1080		900						1040
	1500						1310		1500						1270
	2500						1875		2500						1830
10	600	84	146	120	87	35	850	15	600	111	172	132	82	40	825
	900						1000		900						980
	1500						1235		1500						1210
	2500						1800		2500						1775
11	600	88	139	126	73	35	770	16	600	121	188	144	92	40	840
	900						920		900						1040
	1500						1155		1500						1270
	2500						1715		2500						1835
12	600	92	161	132	89	35	870	NOTES:							
	900						1075	1. Weights & dimensions are for estimating purposes only							
	1500						1250	2. Dimensions are based on 8" nozzle							
	2500						1815	3. Dimensions are in Inches, weights in Lbs							

Table of Dimensions and Weights for Large Diameter, Self-Reinforced Manways

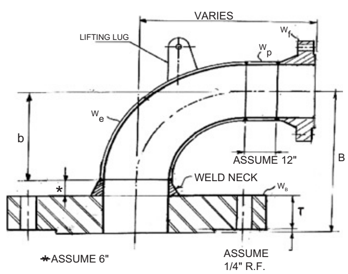


Formulas	Data
f = d + .25	N = Qty Studs d = Dia, Studs, in
C = A - 2 d ; Min = (B_S N) / π	L = Length of Studs, in C = Bolt Circle, in
E = C - X	W_N = Weight, Nozzle, Lbs W_S = Weight, Studs & Nuts, Lbs
R = B + 2a	X = Width across Flats of Nuts, in a = Width of Raised Face, in = Gasket Width + .5" Min
F = f + (D - E)/2	B_S = Min Bolt Spacing = 2.1 d
h = H - T - F -.25	
L = 2 T + 2 d + 1.5	
A = C + 2 d	

Nom Size	A	B	C	D	E	F	H	T	Studs N	Studs d	Studs L	R	W_N	W_S
24" 600#	37	24	33	30	28.25	2.13	18	4	24	1.875	13.25	27.25	1500	370
24" 900#	41	24	35.5	31.62	29.5	3.188	20	5.5	20	2.5	17.5	27.25	2160	715
24" 1500#	46	24	39	33.62	30	5.563	22	8	16	3.5	24.5	27.25	3400	1550
24" 2500#	52	24	45	45	39.5	6.5	24	9.5	20	3.5	27.5	27.25	6000	2110
30" 600#	48	30	40.25	43	33.94	9	24	4.63	28	2	14.75	36.13	1600	535
30" 900#	50	30	43	44.5	35.75	10.44	26	6.56	20	2.5	19.62	36.5	3400	770
30" 1500#	52	30	44.75	46	37.5	12.63	28	8	24	2.5	22.5	39.25	5500	1025
30" 2500#	58	30	52	50	45.88	5.31	30	9.5	24	3	26.5	39.75	12000	1730
36" 600#	52	36	47.5	47	44	4	29	4.88	28	2.25	15.75	40.25	2500	730
36" 900#	54	36	51	51	45.63	6.44	31	6.75	20	3.5	22	40.25	3800	1820
36" 1500#	56	36	50	53	46	6.25	33	8.25	28	2.5	23	44.25	10,660	1115
36" 2500#	62	36	56	54.38	51.38	4.75	36	9.5	28	3	26.5	44.25	15,000	2020

Notes:
1) These dimensions and weights should be used for estimating purposes only. Detail designs have not been calculated.
2) All flange designs shown are non-standard except for the 24" 600# thru 1500# only.
3) All flange designs based on 2-1/4 Cr - 1 Mo material at 850 deg F.

REACTOR INLET ESTIMATED WEIGHTS



DATA

W_B = Weight, Blind, Lbs

W_f = Weight, Outlet Flange, Lbs

W_e = Weight, Elbow, Lbs

W_p = Weight, Pipe, Lbs

W_T = Weight, Total

$W_T = W_B + W_f + W_e + W_p$

BLIND

SIZE	RATING	T	W_B
24	600	4	1175
	900	5.5	2080
	1500	7.5	3500
	2500	9.5	5580
30	600	4.63	2080
	900	6.56	3725
	1500	8	5000
	2500	10.5	7140
36	600	4.88	3300
	900	6.75	5700
	1500	8.25	5400
	2500	12.25	9700

NOZZLE FLANGE

SIZE	RATING	W_f
10	600	180
	900	268
	1500	454
	2500	1075

NOZZLE FLANGE

SIZE	RATING	W_f
12	600	226
	900	372
	1500	670
	2500	800
14	600	334
	900	562
	1500	940
	2500	1750
16	600	462
	900	685
	1500	940
	2500	2260
18	600	531
	900	924
	1500	1625
	2500	2850
20	600	678
	900	1164
	1500	2050
	2500	3800

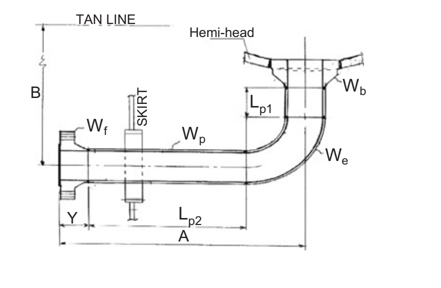
PIPE & ELBOW

SIZE	b	W_p	W_e
8	12	75	117
10	15	160	226
12	18	200	300
14	21	250	410
16	24	300	530
18	27	350	675
20	30	400	840

NOTES

1. Dim "B" = T + b + 6"
2. Pipe & elbow assumed as Sch 160
3. Weld neck is included in blind flg weight
4. Studs and nuts are not included

REACTOR BOTTOM OUTLET ESTIMATED WEIGHTS



DATA

d = Nominal pipe Size, in

W_b = Weight, Boss, Lbs

W_e = Weight, Elbow, Lbs

W_p = Weight, Pipe, Lbs

W_f = Weight, Flange, Lbs

W_T = Weight, Total

$W_T = W_b + W_e + W_p + W_f$

EQUATIONS

$B_{min} = .5\,D + 1.5\,d + L_{p1} + 6"$

A_{min} = 8' to 11.5' Dia : .5D + 14"
12' to 16' Dia : .5D + 16"

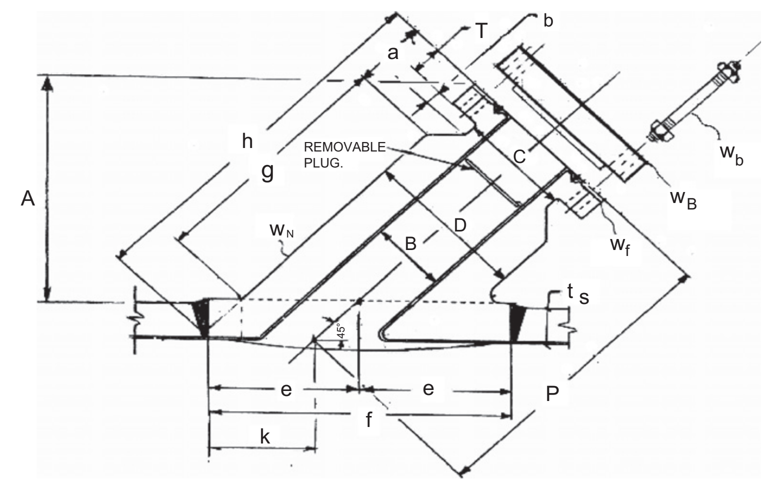
$L_{p2} = A - 1.5\,d - Y$

NOTES

1. Pipe & Elbow are assumed as Sch 160
2. Flange is assumed as 1500#

DIA Ft	d in	A in	B in	Y in	L_{p1} in	L_{p2} in	WEIGHTS (Lbs)				
							W_e	W_f	W_b	W_p	W_T
8	6	62	69	6.75	6	46.25	53	164	90	198	505
	8		72	8.38		41.63	117	273	150	300	840
	10		75	10		37	226	454	232	416	1330
9	6	68	75	6.75	6	52.25	53	164	90	220	527
	8		78	8.38		47.63	117	273	150	335	875
	10		81	10		43	226	454	232	475	1390
10	10	74	83	8.38	8	49	226	454	232	550	1470
	12		86	10		44.88	375	670	320	705	2070
	14		89	11.13		41.25	570	940	380	780	2670
11	10	80	92	8.38	8	55	226	454	232	610	1520
	12		95	10		50.88	375	670	320	785	2150
	14		98	11.13		47.25	570	940	380	875	2765
12	12	88	100	8.38	10	58.88	375	670	320	920	2285
	14		103	10		55.25	570	940	380	1035	2925
	16		106	11.13		51.75	800	1250	490	1260	3800
13	12	94	109	10	10	64.88	375	670	320	1000	2365
	14		112	11.13		61.25	570	940	380	1130	3020
	16		115	11.75		57.75	800	1250	490	1385	3925
14	14	100	120	11.13	12	67.25	570	940	380	1255	3145
	16		123	11.75		63.75	800	1250	490	1547	4087
	18		126	12.25		60.12	1200	1625	620	1860	5300
15	14	106	129	11.75	12	73.25	570	940	380	1350	3240
	16		132	12.25		69.75	800	1250	490	1670	4210
	18		135	12.88		66.13	1200	1625	620	2010	5455
16	16	112	140	12.25	14	75.75	800	1250	490	1830	4370
	18		143	12.88		72.13	1200	1625	620	2220	5665
	20		146	14		68	1650	2050	760	2590	7050

DIMENSIONS & WEIGHTS OF SIDE CATALYST DUMP NOZZLES



NOTES:

1. Weights & dimensions are for estimating only
2. Dimensions are in inches
3. Weights are in Lbs
4. Shell thickness assumed as follows; 600# = 5", 900# = 6", 1500# = 7", 2500# = 8"
5. There is no weight allowance for the shell portion of the nozzle

FORMULAS:

$A = .707\,P + .354\,D$

$a = T + b + .5\,(D - C)$

$h = g + a + 1.414\,t_s$

$f = 2\,e$

$e = .707\,D + t_s$

SIZE	RATING	A	B	C	D	e	f	g	h	b	a	P	T	WEIGHT
6"	600	22.6	6	9.88	14	14.9	29.8	20.5	32.68	1.25	5.18	25	2.125	850
	900	25.1		10	15	16.6	33.2	22.5	37.13	1.44	6.13	28	2.438	1045
	1500	28.2		10.31	15.5	18	36	25	42.38	1.62	7.48	32	3.5	1300
	2500	32.2		11.38	19	21.4	42.9	27	48.62	2.25	10.31	36	4.5	2320
8"	600	35.5	8	11.94	16.5	16.7	33.4	36	48.84	1.38	5.84	42	2.438	1800
	900	38.4		12.44	18.5	19	38	38	53.66	1.63	7.16	45	2.75	2530
	1500	40.6		12.94	19	20.4	40.8	40	58.43	1.88	8.53	48	3.88	2960
	2500	44.5		14.12	21.75	23.4	46.8	42	64.38	2.25	11.07	52	5.25	4425
10"	600	45.6	10	15	20	19	38	48	61.5	1.5	6.5	55	2.75	3400
	900	48.6		15.5	21.5	21.2	42.4	50	66	1.8	7.56	58	3	4300
	1500	52		16.06	23	23.4	46.8	52	71.5	2.2	9.83	62	4.5	5470
	2500	57.5		17.38	26.5	26.7	53.4	54	79	2.75	13.81	68	6.75	8030

Table 2-22
Weight of (1) hemispherical head, based on inside diameter, LBS

Specified Min Thk, t_s (in)	Thk w/ thinning allowance, T_a	Diameter, in															
		24	36	48	60	72	84	96	108	120	132	144	156	168	180	192	204
0.380	**0.560**	151	334	590	918	1318	1790	2334	2950	3638							
0.500	**0.690**	186	412	725	1126	1616	2194	2860	3614	4456							
0.630	**0.810**	223	490	861	1337	1917	2600	3388	4281	5277	6377	7582	8890				
0.750	**0.940**	260	570	1000	1550	2220	3010	3920	4950	6100	7373	8763	10275				
0.880	**1.060**	297	650	1138	1763	2524	3421	4454	5624	6930	8371	9950	11664	13514			
1.000	**1.190**	335	730	1279	1978	2830	3835	4991	6300	7760	9374	11140	13056	15126			
1.130	**1.500**	434	938	1635	2524	3606	4880	6345	8000	9854	11900	14131	16558	19177	21989	24992	28189
1.250	**1.630**	475	1024	1780	2746	3920	5300	6892	8690	10697	12912	15335	17966	20806	23855	27110	30575
1.380	**1.750**	516	1110	1927	2970	4236	5726	7441	9380	11543	13931	16543	19380	22440	25725	29233	32967
1.500	**1.880**	559	1197	2075	3194	4558	6153	7992	10073	12393	14954	17755	20796	24078	27600	31362	35365
1.630	**2.000**	602	1285	2225	3421	4874	6582	8547	10769	13247	15980	18971	22218	25721	29480	33500	37770
1.750	**2.130**	645	1375	2375	3650	5195	7014	9105	11468	14104	17011	20191	23644	27370	31365	35635	40177
1.880	**2.250**	690	1465	2528	3880	5520	7450	9665	12170	14964	18045	21415	25075	29021	33256	37780	42592
2.000	**2.380**	735	1556	2682	4112	5846	7885	10228	12875	15827	19083	22645	26510	30679	35152	39930	45013
2.130	**2.750**			3150	4818	6838	9209	11933	15010	18440	22220	26354	30840	35679	40870	46413	52310
2.250	**2.880**			3311	5057	7172	9655	12508	15728	19316	23273	27600	32293	37355	42786	48586	53242
2.380	**3.000**				5300	7510	10105	13084	16450	20650	24330	28850	33750	39914	44708	50763	57203
2.500	**3.130**				5540	7850	10555	13664	17173	21082	25391	30100	35210	40722	46634	52946	59660
2.630	**3.250**					8190	11010	14246	17900	21969	26455	31358	36678	42414	48566	55135	62120
2.750	**3.380**					8532	11465	14831	18630	22860	27524	32620	38148	44109	50500	57330	64587
2.880	**3.500**						11925	15240	19365	23755	28600	33885	39625	45810	52445	59530	67060
3.000	**3.750**						12850	16605	20840	25555	30750	36430	42590	49225	56345	63945	72025
3.250	**4.000**							17800	22330	27370	32925	38990	45570	52660	60265	68380	77010
3.500	**4.250**							19000	23830	29200	35110	41570	48570	56115	64200	72840	82020
3.750	**4.500**							20225	25345	31040	37310	44160	51585	59590	68165	77320	87050
4.000	**5.000**							22700	28400	34765	41760	49400	57675	66600	76150	86350	97190
4.250	**5.250**								29960	36650	44000	52040	60750	70125	80175	90900	102300
4.500	**6.000**								34700	42400	50850	60075	70000	80840	92370	104675	117750
5.000	**6.500**									46286	55485	65516	76380	88077	100607	113971	128167
5.250	**7.000**									50242	60185	71025	82762	95400	108928	123357	138683
5.500	**7.250**									52241	62558	73805	85982	99086	113121	128084	143976
5.750	**7.750**										67355	79418	92475	106525	121570	137605	154635
6.000	**8.000**										69775	82250	95750	110275	125825	142400	160000

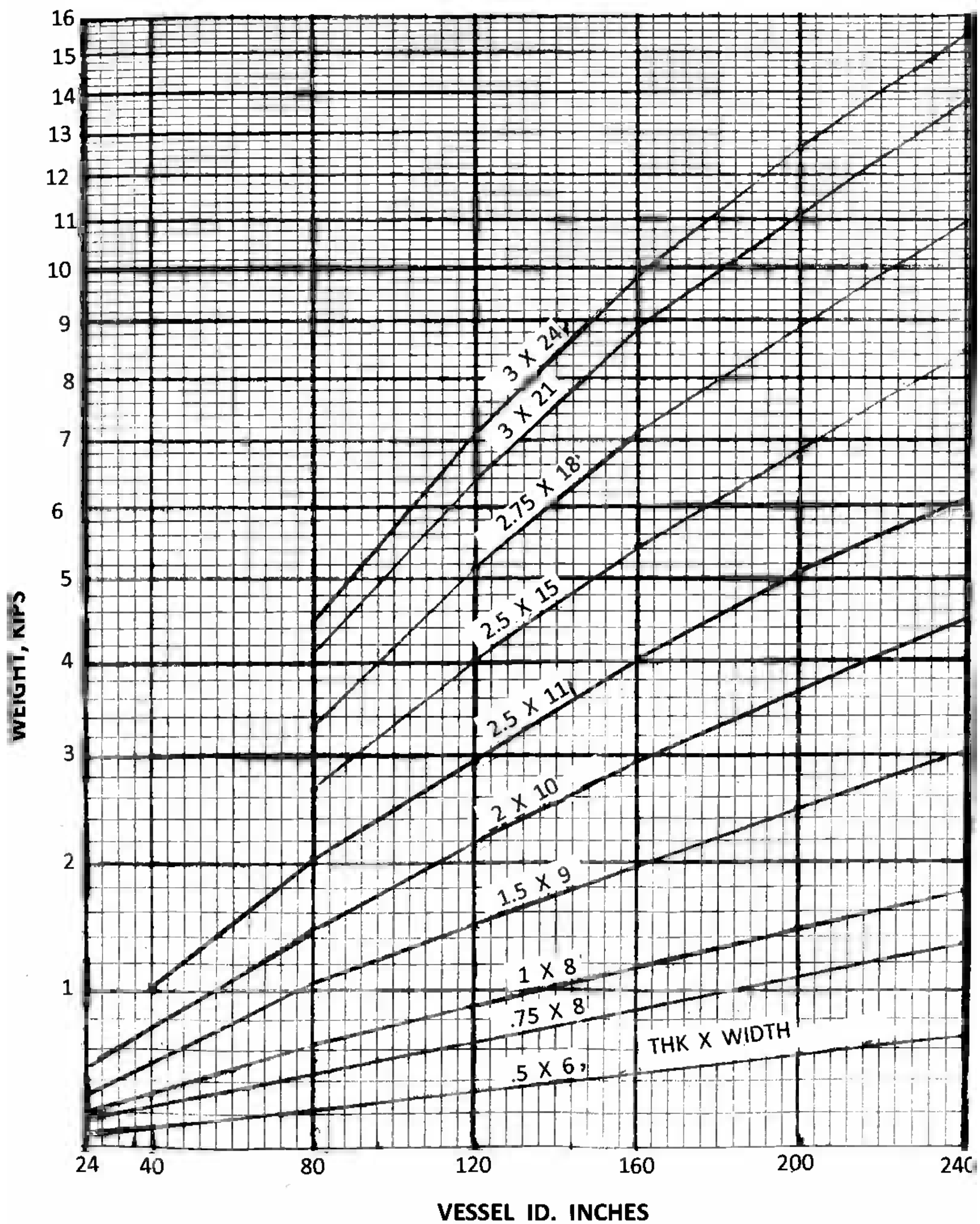
Notes:

1. Assumes carbon steel material (unit wt = .2833 PCI)
2. Thinning allowance, T_a, per Table
3. Formulas are as follows: $W = .445\ D_m^2\ t$; $D_m = D + t$ and $t = t_s + T_a$

Table 2-23
Thinning allowance for hemi-heads

Thickness (in)	Thinning Allowance, T_a
0.125 to 1	0.19
1 to 2	0.38
2 to 3	0.63
3 to 3.75	0.75
4 to 4.5	1
4.5 to 5	1.5
5 to 5.5	1.75
5.75 to 6	2

Weights of base plates



Ladder and Platform (L&P) Estimating

The following is a listing of average breakdowns, both cost and weight, for ladders and platforms (L&Ps) for refinery-type projects. Note that L&Ps include pipe supports, guides, and davits as well as ladders and platforms. Because this data is "average," it is meant to be averaged over an entire project and not to find the cost or weight of any individual item or vessel.

1. **Estimated Price Breakdown**:

• Platforms		30 PSF @ \$2.50/lb	= \$75/sq ft
• Ladders:	Caged	24 lb/ft @ \$3.00/lb	= \$72/ft
	Uncaged	10 lb/ft @ \$2.35/lb	= \$23/ft
• Misc.		\$2.50/lb	
• Handrail:	Straight	\$32/ft	
	Circular	\$42/ft	

2. **Estimated Weight Breakdown** (as a breakdown of the total quantity):

Item		Percentage (%)	Cost (\$/lb)
Platforms:	Circular	30–35%	\$2.50
	Rectangular	50–55%	\$2.00
Ladders:	Caged	7–9%	\$3.00
	Uncaged	2–3%	\$2.25
Misc.		5–10%	\$2.50
Total		100%	

3. **Average Cost of L&Ps** (assuming 100 tons):

Item	Weight (tons)	Cost (\$1000)	% (cost)
Platf Circ	31	155	34
Platf Rect	51	204	45
Ladder Caged	8	48	11
Ladder (uncaged)	2.5	11.25	2
Misc.	7.5	37.5	8
	100 T	\$455.75	100%

Average \$/lb $= 455.75/100 \times 2 =$ \$2.28/lb

4. **Average % Detailed Weight Breakdown for Trayed Columns:**

Item	Large	Medium
Ladders	13.1%	9.3%
Framing	33.3	44.2
Grating	25.3	23.5
Handrailing	18.2	9.7
Pipe supports	3.0	1.6
Bolting	2.5	2.5
Davits	4.1	7.4
Misc.	0.5	1.8
	100%	100%

5. If no estimate of L&Ps is available, an ROM weight estimate can be determined by taking 5% of the overall vessel weights for the project as a total L&P weight. A percentage breakdown may be made of this overall value as noted.

Notes:

1. Miscellaneous weights:

a. Concrete	144 PCF
b. Water	62.4 PCF
c. Gunnite	125 PCF
d. Refractory	65–135 PCF
e. Calcium silicate insulation	13.8 PCF

2. Estimate weight of liquid holdup in random packed columns as 13% of volume.

3. Weights of demister pads and support grids is as follows:

Type	Density (PCF)
931	5
326	7.2
431	9
421	10.8 (multipiece)
	12 (single piece)
Grid	3 PSF

4. Estimate weights of platforming as follows:

Type	Weight
Circular platform	30 PSF
Rectangular platform	20 PSF
Ladder with cage	24 PLF
Ladder without cage	10 PLF

5. Weight of anchor chairs per anchor bolt (wt each, lb):

Anchor Bolt Dia (in.)	Weight (lb)
1	11
1.25	12
1.50	15
1.75	20
2.0	38
2.25	48
2.5	63

Procedure 2-18: Design of Jacketed Vessels

External jackets are used to heat or cool the contents of a vessel. In effect, this turns the pressure vessel into a heat exchanger. Jacketing is an optimum means to accomplish this in terms of control, efficiency and product quality. The advantages are as follows;

1. All liquids can be used. Steam is ideal for heating. Water or Glycol are ideal for cooling.
2. Circulation, temperature and velocity of the heating/cooling media can be carefully controlled.
3. Jackets may be fabricated from a much less expensive metal than the shell.
4. Cleaning and maintenance can be minimized providing a "clean" media is utilized.

Jackets are frequently used in combination with internal heating coils or agitators. Jacketed vessels are much more common in the food, beverage and chemical industry then in the refining industry. The applications are limited in the refining industry because the services in refining application tend to be fouling and not clean. Most refinery applications for heat exchange are dirty services and therefore must be capable of being taken apart for cleaning. Since a jacketed vessel can only be cleaned with steam or chemicals, it is not considered practical for most refinery services.

The types of jackets used on vessels are as follows;

1. Conventional (AKA "Plain")
2. Jacket with a spiral baffle
3. Spiral pipe coil welded to the shell
4. Spiral half pipe coil welded to the shell
5. Dimpled jacket

Conventional Jacket

Used with steam or cooling. Liquid flow velocities are low and the flow is poorly distributed. Natural convection equations are suitable and cooling coefficients have low values. Conventional jackets are best applied to small vessels or high pressure applications, where the vessel internal pressure is twice the jacket pressure as a minimum. The conventional jacket is the most common type. An often used variation of this configuration is made by dividing the straight side into two or more separate jackets.

Jacket with a Spiral baffle

This design is a variation of the conventional jacket. The internal, spiral baffle allows for high flow velocities to be reached. Some clearance must be allowed for between the baffles and the inside of the jacket, to allow for assembly, fabrication and tolerances. Although this clearance may be small, the total leakage area per baffle turn may be substantial when compared with the cross sectional flow area of the baffle passage. Thus, the actual velocity may only be a fraction of the calculated value. To compensate for this leakage around the baffles it is recommended that a 10% allowance be applied to either the total area required or total heat required.

Spiral Pipe Coil Welded to the Shell

This design is not used frequently due to the minimal contact area between the shell and coil. It does however alleviate one of the problems of constructing a half-pipe coil. That is the cutting of the coils in half and the subsequent wastage. Some techniques have been developed to form flat strips into half pipe coils to save wastage and labor. There is just as much welding required for attaching a full pipe coil as there is with the half pipe.

Spiral Half Pipe Coil welded to the shell

This design provides high velocity and turbulence within the jacket. This in turn will result in an unusually high film coefficient. The half pipe coil is recommended for high temperature and all liquid applications. It is better than conventional jackets because the pressure drop can be carefully controlled and calculated. It is not however practical for small vessels, less than 500 gallons. Because there are no limitations to the number of inlet and outlet connections, this type of jacketing can be divided into multiple zones for maximum flexibility and efficiency.

For maximum heat transfer, the coils should be spaced about ¾" (19mm) apart, but other spacing is aceptable. Standard sizes are 2", 3" or 4" NPS.

Dimpled Jackets

There are two main types of dimpled jackets. These are;

1. Integral construction
2. Bolted-on or clamp-on type.

The bolted-on types are less expensive but do not have completely reliable heat transfer characteristics due to the "fit" of the clamp-on sections. Sometimes a heat transfer mastic is used between the two surfaces. Advantages are as follows;

1. They are cheap.
2. They are completely replaceable.
3. The service can be easily altered.
4. The shape and configuration can be easily modified.
5. They do not exert any external pressure on the vessel shell.

The integral type utilizes the inner surface of the dimple jacket for the vessel shell. Advantages of the integral type are as follows;

1. They are cheaper than a conventional jacket.
2. The savings increase with higher pressures and larger vessels.
3. They are cheaper than the half pipe jacket for low internal pressures.

General

External Pressure

The inner vessel shall be designed to resist the maximum differential pressure. Typically this would be the design pressure of the jacket. The differential pressure should include accidental vacuum in the inner vessel. Particular attention should be given to the effects of local loads and differential expansion.

The spiral baffles may not be considered as contributing stiffness to the shell for the case of external pressure since they are not closed circumferential rings. If stiffening rings are required, they must be placed inside the vessel. This may be a problem from a process standpoint. The alternatives are to increase the shell thickness, or shorten the jacket into smaller sections to reduce the L dimension.

Per ASME Code, if the internal pressure is greater than 1.67 times the external pressure, the external pressure will not govern.

The half pipe coil causes an external pressure on the vessel shell but only inside the coil area, and then not in a complete 360° circumferential zone. Therefore the effects on the vessel shell are negligible.

Design

There are two distinct aspects of the design of jackets for heat transfer. These are the thermal design and the physical design. The thermal design falls into three parts:

1. Determine the proper design basis:
   a. Vessel proportions
   b. Maximum depth of liquid
   c. Time required to heat/cool
   d. Agitated or non-agitated
   e. Type of operation; batch or continuous
2. Calculating the required heat load
3. Computing the required surface area

Physical design includes the following:

1. Selecting the type of jacket
2. Determine the areas of the vessel shell to be heated or cooled
3. Determine whether external connections are:
   a. Separate
   b. Series
   c. Parallel
4. Determine pressure drop

The type of operation is characterized in the following cases:

1. Batch operation: Heating
2. Batch operation: Cooling
3. Continuous operation: Heating
4. Continuous operation: Cooling

Pressure Drop

It is important that pressure drop be considered in the design of an external jacket. This will establish the practical limits on the length of passageway inside the jacket.

Pressure drop in conventional jackets without spiral baffles and dimple jackets are deliberately excluded from this procedure. Other sources should be consulted for these applications.

For design purposes, the half pipe coil and conventional jackets with spiral baffles are treated as coils. This procedure converts the shape of the passageway into an equivalent diameter pipe coil.

Large pressure drops may mean the passageway is not capable of transmitting the required quantity of liquid at the available pressure. In addition the fluid velocities inside the passageway should be kept as high as possible to reduce film buildup.

There are no set rules or parameters for maximum allowable pressure drop. Rather, an acceptable pressure drop is related to the velocity required to effect the heat transfer. For liquids a minimum velocity of 1 to 3 feet per second should be considered as minimum. For gases "rho V squared" should be maintained around 4000.

Pressure drop in spiral passageways is dependent on whether the flow is laminar or turbulent. Typically flows are laminar at low fluid velocities and turbulent at high fluid velocities. In spiral passageways a secondary circulation takes place called the "Double Eddy" or "Dean Effect". While this circulation increases the friction loss, it also tends to stabilize laminar flow, thus increasing the "critical" Reynolds number.

In general flows are laminar at Reynolds numbers less than 2000 and turbulent when greater than 4000. At Reynolds numbers between 2000 and 4000 intermittent conditions exist that are called the "critical zone".

For steam flow the pressure drop will be high near the inlet and decrease approximately as the square of the velocity. From this relationship, combined with the effects of increased specific volume of the steam due to pressure drop, it can be shown that the average velocity of the steam in the coil is ¾ of the maximum inlet velocity. For the purposes of calculating pressure drop this ratio may be used to determine the average quantity of steam flowing within the spiral passageway.

In cases where a high velocity is required in order to utilize a high heat transfer coefficient, the pressure drop becomes important. If the pressure drop exceeds the pressure output of a positive displacement pump, then there is a real possibility that the pressure of the jacket will exceed the design conditions.

In the case of a centrifugal pump, the velocity of the fluid will adjust to the pump pressure available. This may result in a lower heat transfer coefficient then required.

For either a dimple jacket design or a half pipe jacket, the pressure drop will be higher than that of an equivalent conventional jacket, due to the increased turbulence.

The preferred method to reduce pressure drop is to shorten the path of the fluid and manifold the inlets and outlets by "parallel" routing. The "series" type external piping connections will increase the overall pressure drop. See Figure 2-50.

Heat Transfer Coefficient, U

The Heat Transfer Coefficient, U, is dependent on the following variables;

1. Thermal conductivity of metal, medium and product
2. Thickness of metal in vessel shell
3. Fluid velocity
4. Specific heat
5. Density and viscosity
6. Fouling factor (oxidation, scaling)
7. Temperature differences (driving force)
8. Trapped gasses in liquid flow
9. Type of flow regime (laminar versus turbulent, turbulent being better)

Agitation

In order to effect the best heat transfer possible, it is preferable to have the contents circulated with an agitator or mixer. This is desirable but not mandatory.

In many cases agitators are provided as part of the process for blending or mixing, not purely to enhance heat transfer.

However, there is a "natural circulation" set up by the heating of a product inside a vessel that will occur. This circulation is driven by the warmer product rising in the tank.

Agitation will also help to prevent buildup or fouling of the vessel wall. This provides for the use of a higher heat transfer coefficient.

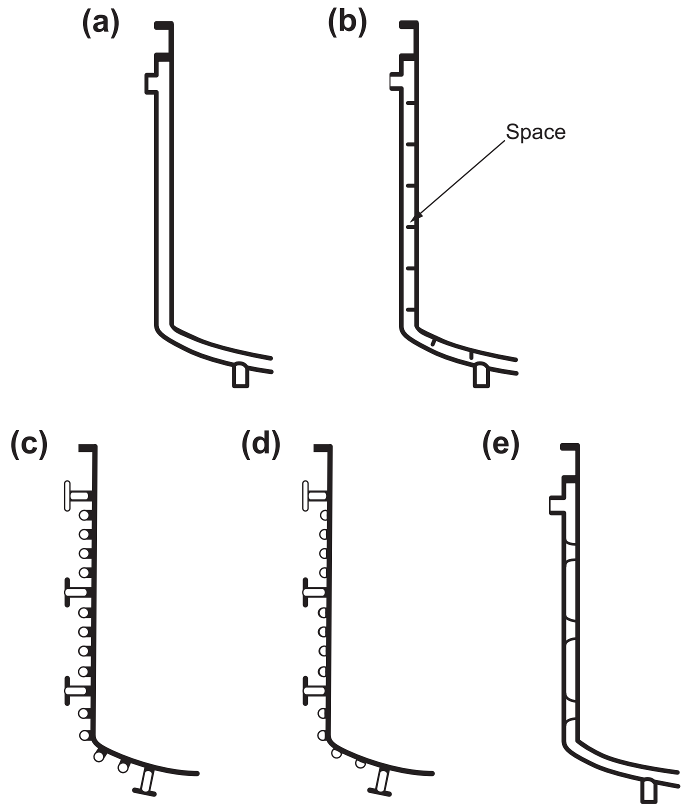


Figure 2-48. Types of Jacket Construction (a) Plain Jacket (b) Jacket with Spiral Baffle (c) Pipe Coil Welded to Shell (d) Half-Pipe Welded to Shell (e) Dimpled Jacket.

Notation

A_C = Cross sectional area of jacket flow area, in^2
A_r = Cross sectional area of jacket flow area, in^2
D = Vessel OD, Ft
D_e, d_e = Equivalent inside diameter of passageway, D_e is in Ft, d_e is inches
f = Friction factor
F_{LF} = Laminar flow factor
g = Acceleration due to gravity$=4.17 \times 10^8$ Ft/Hr^2
j = Jacket width, in
L = Design length of jacket section, in
L_P = Length of path of travel, Ft
L_{BP} = Baffle pitch, in
M = Mass flow rate, Lbs/Hr
N = Number of turns of spiral baffle or half pipe
P = Internal design pressure in jacket, PSI
p = Pitch of half pipe coil, in
Q = Total heat required, BTU/Hr
Q_L = Heat loss from the exterior of the vessel shell and jacket, BTU/Hr
r = Radius of toroidal section of closure, in
R_e = Reynolds number
R_{ec} = Critical Reynolds number
R_i = Inside radius of vessel, in
R_j = Inside radius of jacket, in
R_S = Outside radius of vessel, in
S = ASME Code allowable stress, tension, PSI
t_C = Thickness of closure bar, in
t_j = Thickness of jacket, in
t_{rJ} = Thickness required, jacket, in
t_{rC} = Thickness required, closure bar, in
t_S = Thickness of vessel shell, in
U = Heat transfer coefficient, $BTU/Hr/Ft^2/°F$
V = Velocity of media, FPS or FPH
W = Rate of flow in jacket, Lbs/Hr
Y, Z = Weld sizes, in
ρ = Density of fluid, PCF
ΔP = Pressure drop, PSI
ΔP_L = Straight line pressure drop, PSI
μ = Dynamic viscosity, cP

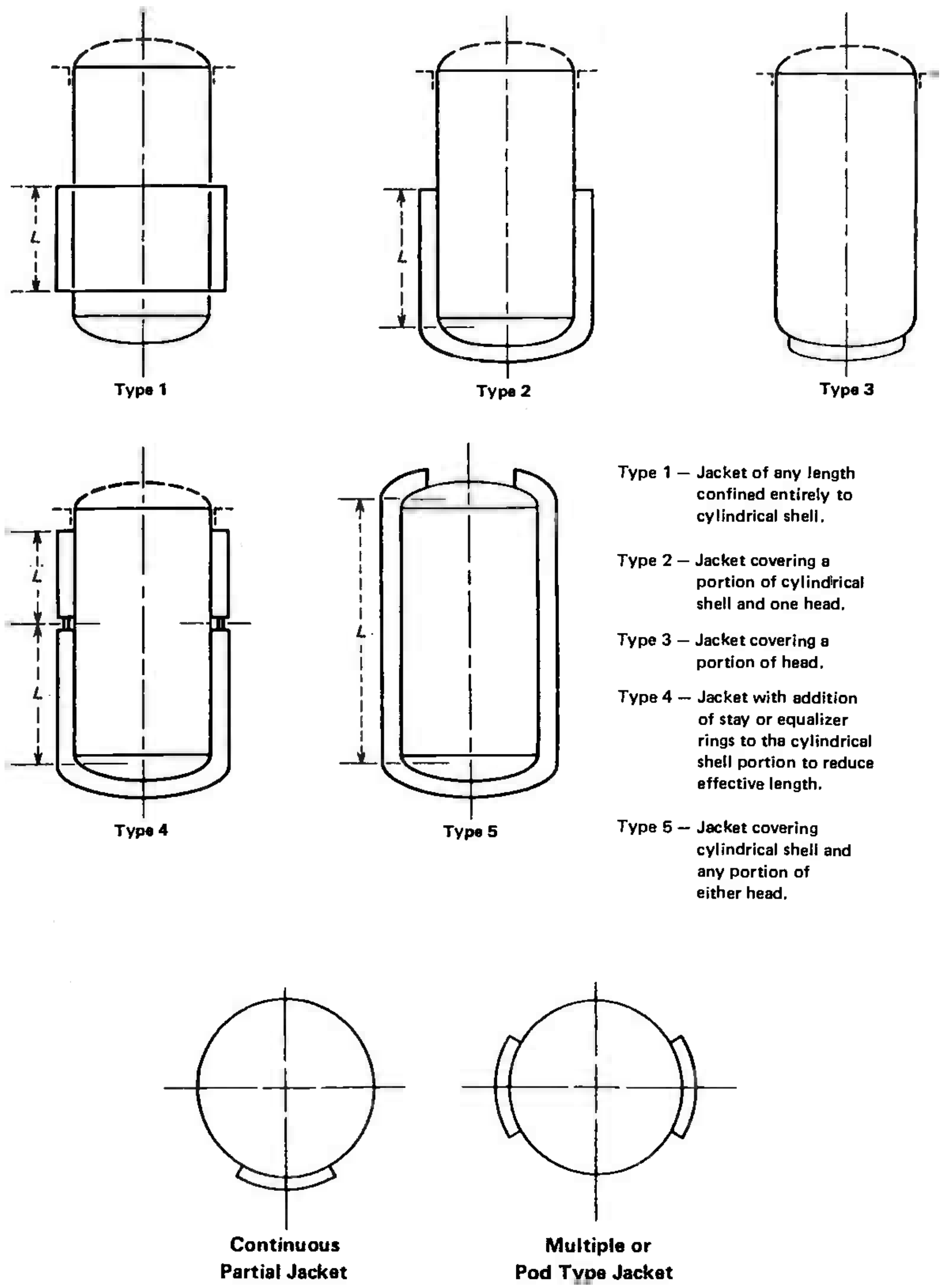
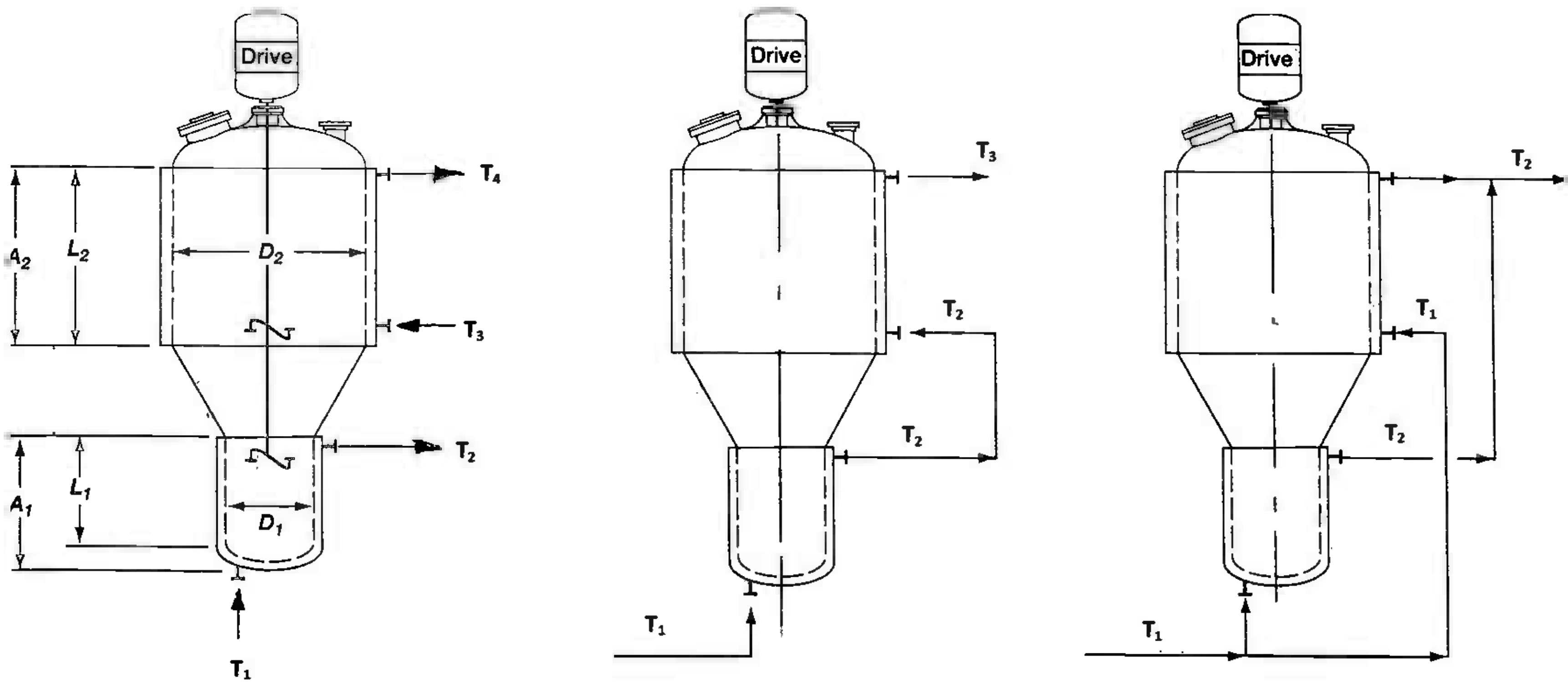


Figure 2-49. Some acceptable types of jacketed vessels.



TYPE 1
JACKETS SEPARATE

TYPE 2
JACKETS IN SERIES

TYPE 3
JACKETS IN PARALLEL

Figure 2-50. External piping configurations for multi-jacketed vessles.

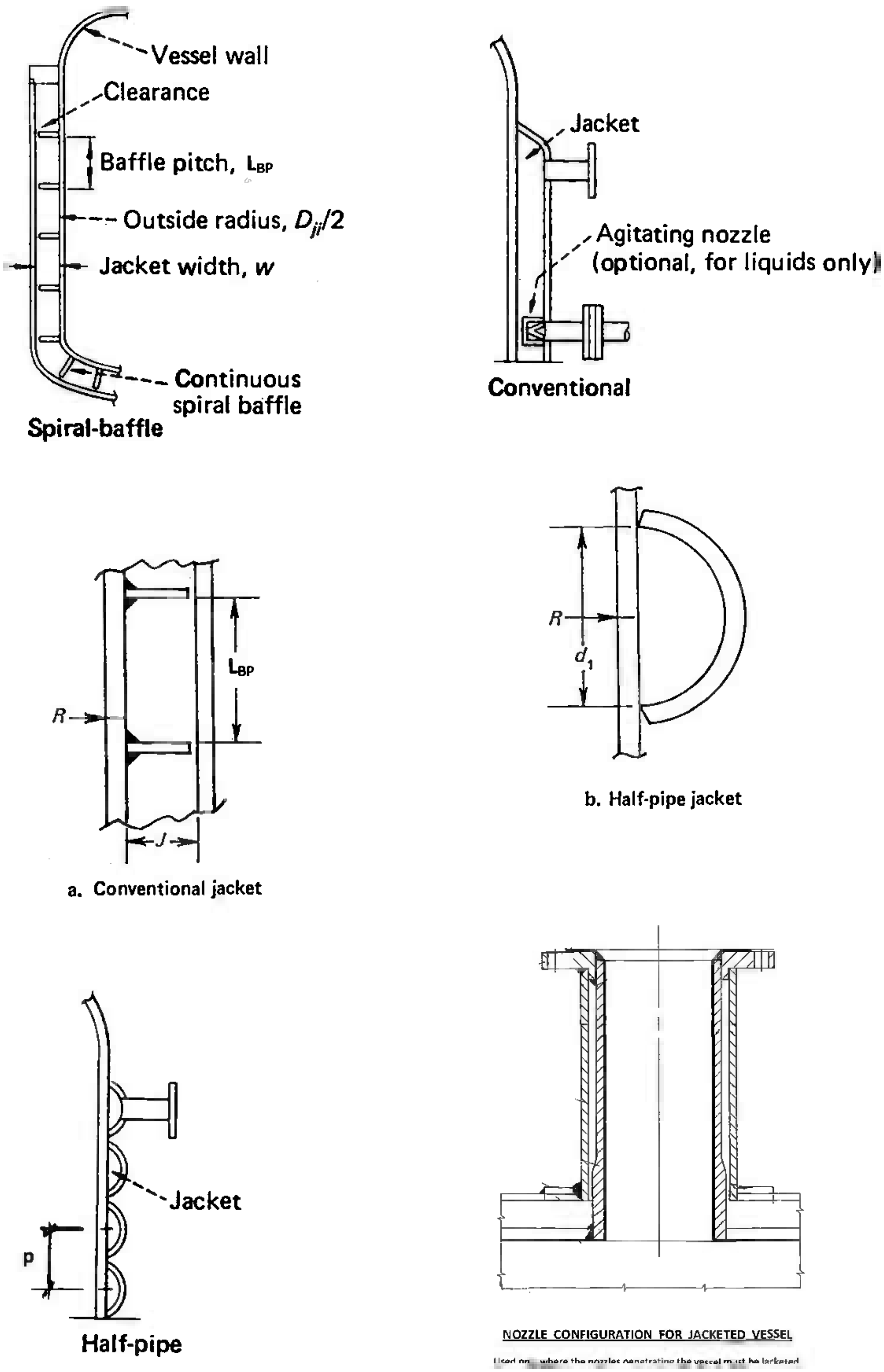


Figure 2-51. Miscellaneous jacket details.

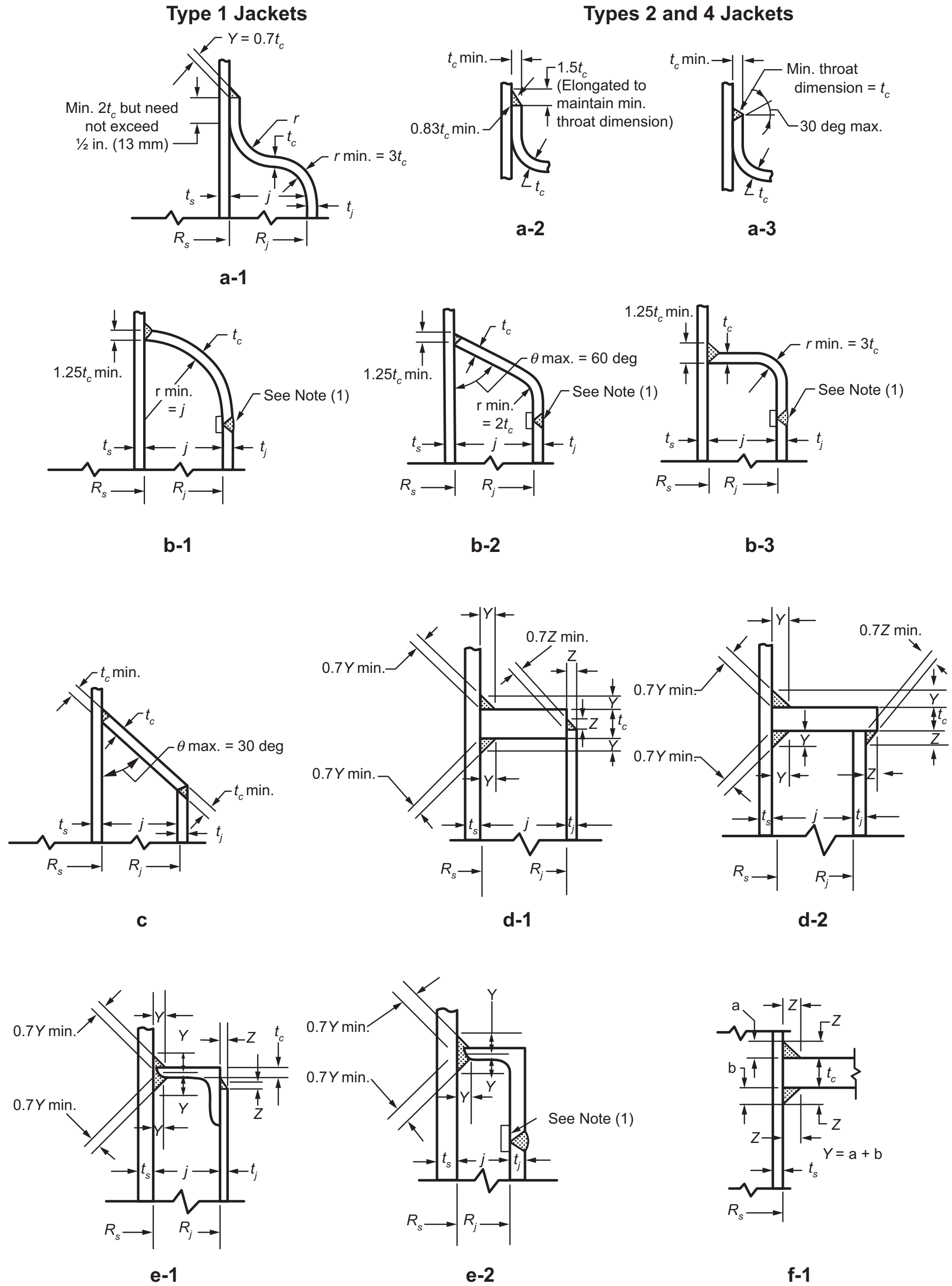


Figure 2-52a. Some acceptable types of jacket closures.

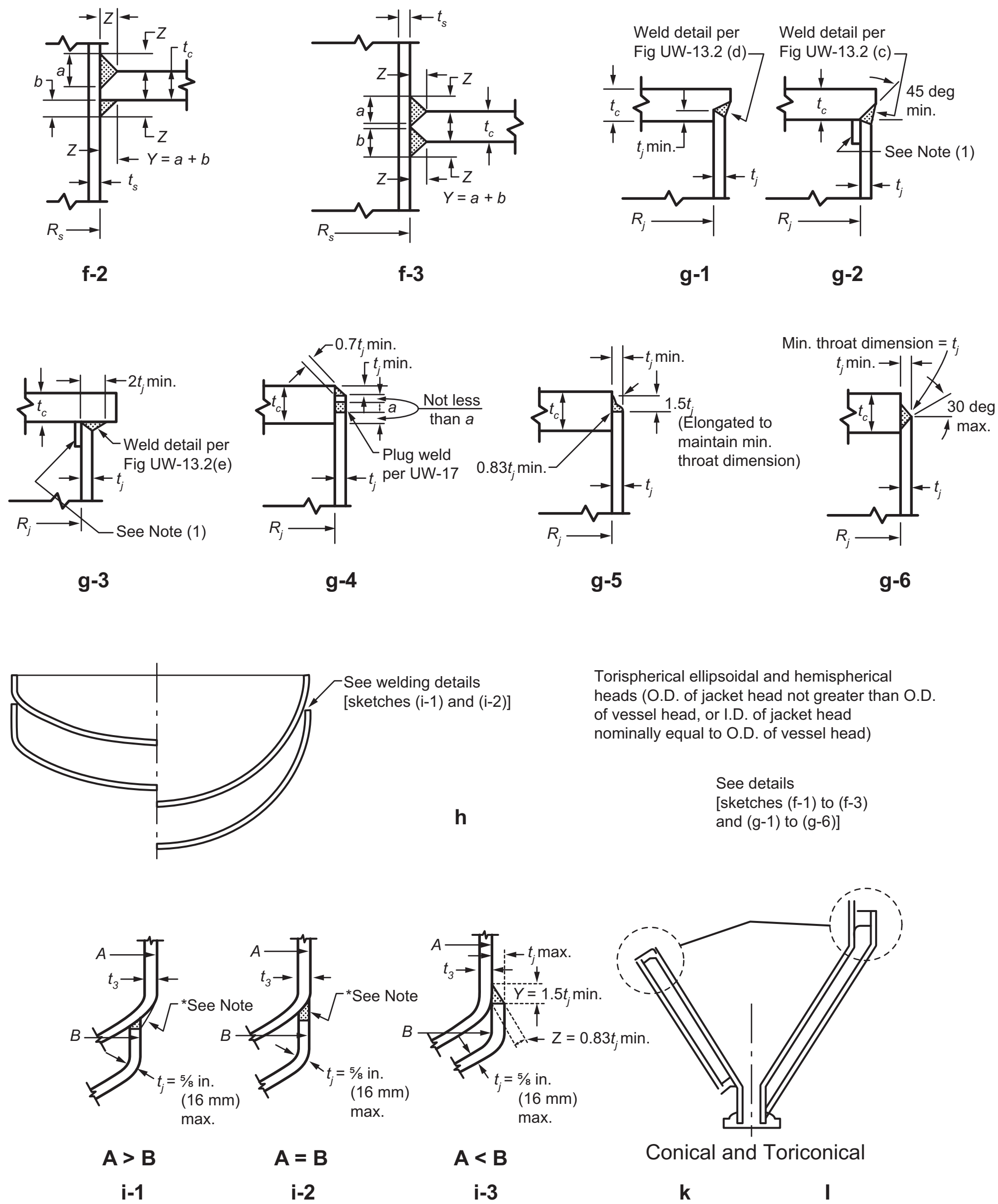


Figure 2-52b. Some acceptable types of jacket closures (Continued)

Table 2-24
Notes To Figure 2-52

Fig	Notes/Remarks	Fig	Notes/Remarks
a-1	a. Type 1 jacket only!	f-1, f-2 & f-3	a. May be used for any type jacket
	b. $t_{rc} \geq t_{rj}$		b. For Type 1 jacket, the closure bar thk shall be the greater of the following;
	c. $r \geq 3\ t_c$		1. $t_{rc} = 2\ t_{rj}$
	d. $t_{rc} = .625$" (16 mm) max		2. $t_{rc} = .707\ J\ (P / S)^{1/2}$
	e. $Y \geq .7\ t_c$		c. For all other jacket types;
a-2 & a-3	a. Type 2 or 4 jackets only!		$t_{rc} = 1.414\ [(P\ R_s J) / S)^{1/2}]$
	b. $t_{rc} \geq t_{rj}$		$J = [(2\ S\ t_s^2)/(P\ Rj)] - .5\ (t_s + tj)$
	c. $r \geq 3t_c$		d. Weld sizes;
	d. $t_{rc} = .625$" (16 mm) max e. $Y \geq .83\ t_c$		$Y \geq$ Smaller of $1.5\ t_c$ or $1.5\ t_s$
b-1 & b-2	a $t_{rc} \geq t_{rj}$		where $Y =$ Sum of $a + b$
b-3	a. $t_{rc} \geq t_{rj}$		$Z =$ Min fillet weld size used to maintain the min Y dimension
	b. $t_{rc} \geq .707\ J\ (P / S)^{1/2}$	g-1, g-2 & g-3	a. May be used with any type jacket
c	a. Type 1 jacket only!		
	b. $\theta = 30°$ max	g-4, g-5 & g-6	a. May be used with any type jacket
	c. $t_{rc} = (P\ r) / [\cos\theta\ (SE - .6\ P)]$		b. $t_{rc} = .625$" (16 mm) max
	d. $t_{rc} \geq t_{rj}$	h	a. Type 3 jacket only!
d-1, d-2, e-1, & e-2	a. Type 1 jacket only! b. $t_{rj} = .625$" (16 mm) max		b. Attachment welds shall be in accordance with Figures i-1, i-2 or i-3
	c. Min thk of closure bar shall be the greater of the following;		c. $t_{rj} = .625$" (16 mm) max
	1. $t_{rc} = 2\ t_{rj}$ 2. $t_{rc} = .707\ J\ (P / S)^{1/2}$	i-1, i-2 & i-3	a. Weld details for jackets attahed to heads
	d. $Y \geq$ Smaller of $.75\ t_c$ or $.75\ t_s$	k or l	a. Closures for conical or toriconical closures only
	e. $Z \geq tj$		b. Type 2 jacket only!

Note 1:
For sketches b-1, b-2, b-3, e-2, g-2 and g-3, a backing strip may be used with a full penetration weld.

Procedure 2-19: Forming Strains/Fiber Elongation

Nomenclature

d = Original ID of pipe or tube, in
d_f = Final OD of pipe or tube, in
d_o = OD of pipe or tube, in
D = Finished ID of pipe or tube, in
D_b = Diameter of blank plate or intermediate product, in
D_f = Final OD of pipe or tube, in
D_o = Original OD of head, in
FE = Fiber elongation, %
L_1 = Developed length of two knuckles of head or cone, in
L_2 = Developed length of crown of head or hemisphere, in
L = Initial length of swaged or flared section of pipe or tube, in
L_f = Final length of swaged or flared section of pipe or tube, in
r_o = Outside radius of pipe or tube, in
R_c = Centerline bend radius of pipe or tube, in
R_f = Final mean radius, in
R_o = Original mean radius of plate for head. Assume ∞ if forming begins with a flat plate
t = Thickness of plate, in
t_A = Wall thickness of pipe or tube before bending or forming, in
t_B = Wall thickness of pipe or tube after forming, in
ε_f = Forming strain or fiber elongation, %

Expected Wall Thinning of Pipe Bends

$$t_B = [R_c/(R_c + .5\ d_o)]\ (t_A)$$

$$R_f = R_c + r_o - .5\ t_B$$

Blank Diameter for Formed Heads (Developed Length)

Dimensions…

$$a = .5(D_m - 2\ r)$$

$$\text{Sin}\ \alpha = a/(L - r)$$

$$\alpha =$$

$$\beta = 90 - \alpha$$

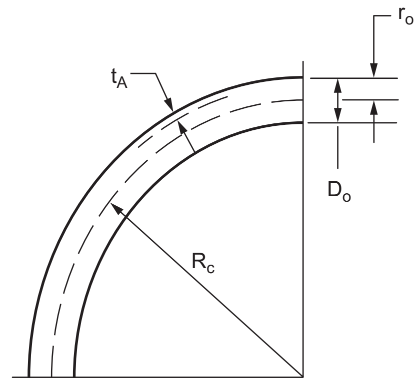


Figure 2-53. Dimensions of pipe bend.

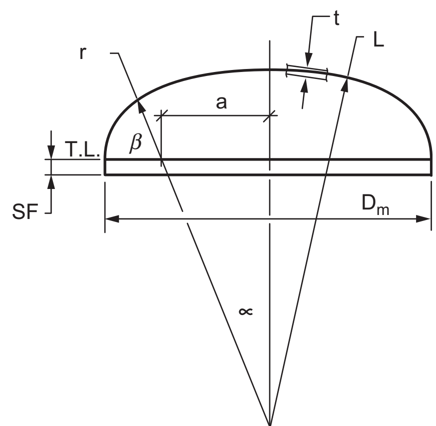


Figure 2-54. Dimensions of head.

- Length of knuckles, L_1
  $L_1 = 2r\beta\ (\pi/180)$
- Length of crown, L_2
  $L_2 = 2L\alpha\ (\pi/180)$
- Diameter of blank necessary, D_b (developed length)
  $D_b = L_1 + L_2 + 2\ SF + 1$ in

Assumes 1 in for machining.

Example # 1: Formed Head

Given: 100 in ID, 2:1 S.E. Head, 1 in thick

$D_m = 101$ in

$L = .9\ D_m = 90.9$ in

$r = .1727\ D_m = 17.44$ in

$a = 0.5(D_m - 2r) = .5(101 - 34.88) = 33.06$ in

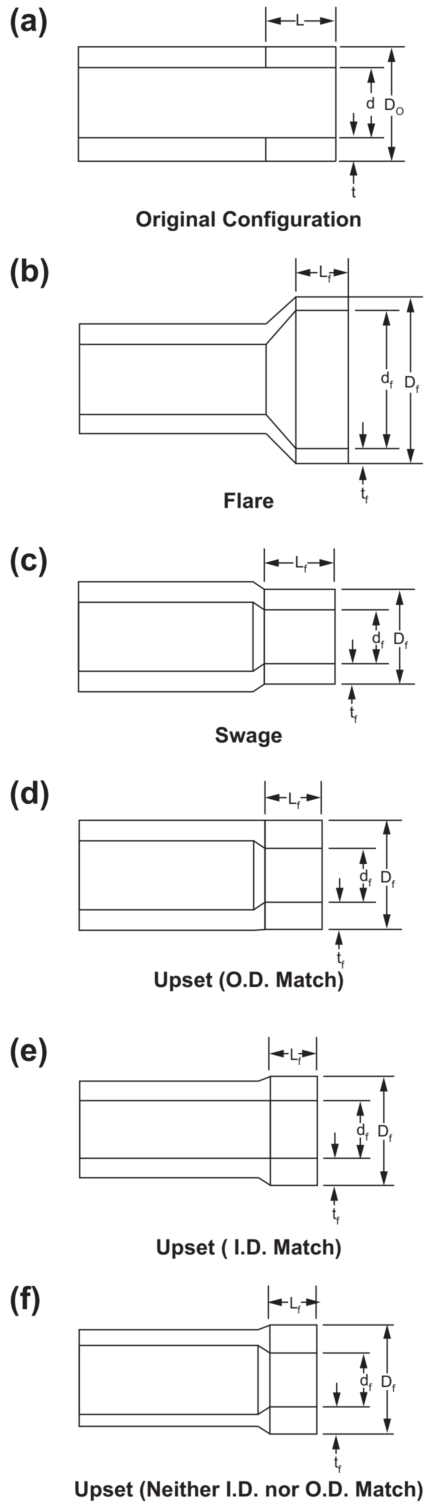


Figure 2-55. Cold forming operations for flaring, swaping, and upsetting of tubing.

$\sin \alpha = a/(L\text{-}r) = 33.06/(90.9 - 17.44) = .452$

$\alpha = 26.75°$

$\beta = 90 - \alpha = 63.25$

- Length of knuckles, L_1

  $L_1 = 2r\beta(\pi/180)$

  $2(17.44)(63.25)(\pi/180) = 38.50$ in
- Length of crown, L_2

  $L_2 = 2L\alpha(\pi/180) =$

  $2(90.9)(26.75)(\pi/180) = 84.88$ in
- Diameter of blank necessary, D_b (Developed length)

  $D_b = L_1 + L_2 + 2\ SF + 1$ in

  $= 38.50 + 84.88 + 3 + 1 = 127.38$
- Per ASME Section VIII-1

  $FE = 75\ t/R_f(1 - R_f/R_o)$

  $= (75)(1)/(17.44) = 4.30\%$

  Since $R_o = \infty$, i.e. starting with flat plate, the term $(1 - R_f / R_o)$ drops out.
  Since $FE \leq 5\%$, no heat treatment required!
- Per ASME Section VIII-2

  $\varepsilon_f = 100\ L_n(D_b/D_i)$

  $= 100\ L_n(127/100) = 23.9\%$

  Since $FE > 5\%$, heat treatment is required

Example #2: Pipe Bend

Pipe Size: 4" Sch 40

$d_o = 4.5$ in

$t_A = 0.154$ in

$R_c = 6$ in

- Wall thinning after forming, t_B

  $t_B = [R_c/(R_c + .5\ d_o)](t_A)$

  $= [6/(6 + .5(4.5)](.154) = 0.112$ in
- Forming strain, ε_f (per ASME VIII-2)

  $\varepsilon_f = \max\left[(r/R_f), ((t_A\text{-}t_B)/t_A)\right].\ 100$

  Where

$$R_f = R_c + r_o - .5\ t_B$$
$$= 6 + 2.25 - .056 = 8.194\ \text{in}$$
$$\varepsilon_f = \max\left[(2.25/8.194), ((0.154 - 0.112)/0.154)\right] \cdot 100 = 27.5\%$$

Notes

1. This procedure is for carbon steel and low alloy material only.
2. As far as ASME Code is concerned, there is no difference between fiber elongation, FE, and forming strain, ε_f. In ASME VIII-1, it is called fiber elongation. In ASME VIII-2, it is called forming strain. In reality, the only way to determine actual fiber elongation is to remove a sample section of material, macroetch the sample, and view under a microscope.
3. For formed vessel heads, the fiber elongation in the knuckle will always govern. However the FE should be calculated for the crown radius as well.
4. For pipe bends, the inside of the pipe bend gets thicker, while the outside of the bend tends to stretch the material and thinning occurs. The maximum FE is at the far outside of the bend, also known as the extrados.
5. Requirements for heat treatment;

For P No. 1, Groups 1 & 2;

a. All material with a FE > 5% should be heat treated except following…
b. FE can be as high as 40% providing none of the following conditions exist;
   - Vessel is in lethal service
   - Impact testing is not required
   - t > 0.625 in
   - Forming temperature is between 250°F to 900°F

For all other material groups;

6. For all other material, see applicable material sections of ASME Code.
7. When heat treatment is required, it shall be stress relieved.
8. Normalizing is only required when called out by customer or material specifications.
9. If the section is hot formed above the normalizing temperature, and allowed to air cool, no further heat treatment is required.
10. Requirements per ASME Section VIII, Division 1 shall be per UCS-79.
11. Requirements per ASME Section VIII, Division 2 shall be per Table 6.1.

Table 2-25
Formulas for calculating forming strains

Type of Part Being Formed	Forming Strain
Heads and conical sections – formed by spinning or dishing	$\varepsilon_f = 100\ln\left(\frac{D_h}{D_o - 2t}\right)$
Shell – cylinders formed from plate	$\varepsilon_f = \frac{50t}{R_f}\left(1 - \frac{R_f}{R_o}\right)$
Heads and conical sections – formed by pressing	$\varepsilon_f = \frac{75t}{R_f}\left(1 - \frac{R_f}{R_o}\right)$
Tubing and pipe-bends	$\varepsilon_f = \max\left[\left(\frac{r_o}{R_f}\right), \left(\frac{t_A - t_B}{t_A}\right)\right] \cdot 100$
Tubing and pipe-flares, swages and upsets	*outside diameter hoop strain* $\varepsilon_f = \left(\frac{D - D_f}{D}\right) \cdot 100$ *inside diameter hoop strain* $\varepsilon_f = \left(\frac{d - d_f}{d}\right) \cdot 100$ *axial strain* $\varepsilon_f = \left(\frac{L - L_f}{L}\right) \cdot 100$ *radial strain* $\varepsilon_f = \left(\frac{t - t_f}{t}\right) \cdot 100$

References

[1] ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, 2010 Edition, American Society of Mechanical Engineers.

[2] Harvey JF. Theory and Design of Modern Pressure Vessels. 2nd Edition. Van Nostrand Reinhold Co; 1974.

[3] Bednar HH. Pressure Vessel Design Handbook. Van Nostrand Reinhold Co; 1981.

[4] Brownell LE, Young EH. Process equipment design. John Wiley and Sons, Inc.; 1959. Section 6.2, pp. 157–159.

[5] Watts GW, Lang HA. The Stresses in a Pressure Vessel with a Flat Head Closure, ASME Paper No. 51-A–146, 1951.

[6] Burgreen D. Design methods for power plant structures. C. P. Press; 1975.

[7] Manual of Steel Construction. 8th Edition. American Institute of Steel Construction, Inc; 1980.

[8] ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, 2010 Edition, American Society of Mechanical Engineers.

[9] Levinson IS. Statics and Strength of Materials. Prentice Hall, Inc 1971. Chapter 16.

[10] Troitsky MS. Tubular Steel Structures, Theory and Design. 2nd Edition. James F. Lincoln Arc Welding Foundation; 1990. Chapter 2.

[11] Dambski JW. Interaction Analysis of Cylindrical Vessels for External Pressure and Imposed Loads. ASME Technical Paper 80-C2/PVP-2. American Society of Mechanical Engineers; 1990.

[12] Harvey JF. Theory and Design of Pressure Vessels. Van Nostrand Reinhold Co; 1985. Chapter 8.

[13] AWWA D-100-96, Welded Steel Tanks for Water Storage, American Water Works Association, 6666 W. Quincy Ave, Denver, CO 80235, Section 4.0.

[14] Jawad MH, Farr JR. Structural analysis and Design of Process Equipment. 2nd Edition. John Wiley and Sons; 1989. Chapter 5.3.

[15] Bednar HH. Pressure Vessel Handbook. 2nd Edition. Van Nostrand Reinhold Co.; 1986. 48–55.

[16] Kerns GD. New charts speed drum sizing. Petroleum Refiner; July 1960.

[17] Jawadekar SP. Consider corrosion in L/D calculation. Chemical Engineering December 15, 1980.

[18] Abakians K. Nomographs give optimum vessel size. Hydrocarbon Processing and Petroleum Refiner June 1963.

[19] Gerunda A. How to size liquid-vapor separators. Chemical Engineering May 4, 1981.

[20] Brownell LE, Young EH. Process Equipment Design. John Wiley & Sons, Inc.; 1959. Chapter 5.

[21] Galambos, Theodore V., *Guide to Stability Design Criteria for Metal Structures*. John Wiley & Sons. INC., 1998, Chapter 14.

3
Flange Design

Introduction

Standard flanges should be used wherever possible. The cost of designing and fabricating a custom flange is expensive and should be a last resort. If a custom flange is required, and there is no alternative, then the procedures included in this chapter can be utilized for designing a custom flange. The ASME Code accepts the standard pressure-temperature ratings of ASME B16.5 for flanges, classes 150 to 2500, ½” to 24”. For larger diameter flanges use ASME B16.47.

Flange Standards

1. ASME/ANSI B16.5 (1/2” to 24” Class 150 to 2500)
2. ASME/ANSI B16.47
   • Series A: Replaced the Old API 605 (26” to 60”, Class 75 to Class 900)
   • Series B: Based on MSS SP 44 Steel Pipeline flanges designed to ASME Section VIII Large flanges (26” to 60”, Class 150 to Class 900)
3. TEMA Flanges, Class R
4. ASME/ANSI B16.1 Cast Iron Pipe Flanges
5. ASME/ANSI B16.24 Bronze Flanges and Flanged Fittings
6. AWWA C207 Steel Pipe Flanges for Water Works Services
7. BS 3293 British Standards – Specification for Carbon Steel Pipe Flanges (over 24” nominal size)
8. API Specification 6A (2,000 to 20,000 psi for wellhead equipment)
9. MSS SP-51 Corrosion Resistant Cast Flanges and Flanged Fittings
10. MSS SP-65 High Pressure Chemical Industry Flanges

Flange Design

In general, special designs as outlined in this procedure are done for large or high-pressure designs. Flanges in this category will be governed by one of two conditions:

1. Gasket seating force, W_{m2}
2. Hydrostatic end force, H

For high-pressure flanges, typically the hydrostatic end force, H, will govern. For low-pressure flanges, the gasket seating force will govern. Therefore the strategy for approaching the design of these flanges will vary. The strategy is as follows:

• *For low-pressure flanges*
  a. Minimize the gasket width to reduce the force necessary to seat the gasket.
  b. Use a larger number of smaller diameter bolts to minimize the bolt circle diameter and thus reduce the moment arm which governs the flange thickness.
  c. Utilize hubless flanges (either lap joint or plate flanges) to minimize the cost of forgings.
• *For high-pressure flanges*
  High-pressure flanges require a large bolt area to counteract the large hydrostatic end force. Large bolts, in turn, increase the bolt circle with a corresponding increase in the moment arm. Thicker flanges and large hubs are necessary to distribute the bolt loads. Seek a balance between the quantity and size of bolts, bolt spacing, and bolt circle diameter.

Design Strategy

Step 1: Determine the number and size of bolts required. As a rule of thumb, start with a number of bolts equal to the nominal size of the bore in inches, rounded to the nearest multiple of four. First, calculate W_{m1} or W_{m2}. A_m is equal to the larger of W_{m1} or W_{m2} divided by S_a. The quantity of bolts required is:

$$n = A_m/R_a$$

To find the size of bolt for a given quantity:

$$R_a = A_m/n$$

With these two equations a variety of combinations can be determined.

Step 2: Determine the bolt circle diameter for the selected bolt size.

$$C = B + 2g_1 + 2R$$

The flange O.D. may now be established.

$$A = C + 2E$$

Step 3: Check the minimum bolt spacing (not an ASME requirement). Compare with the value of B_s in Table 3-3.

$$B_s = C/n$$

Note: Dimensions R_a, R, E, and B_s are from Table 3-3.

Step 4: After all of the preliminary dimensions and details are selected, proceed with the detailed analysis of the flange by calculating the balance of forces, moments, and stresses in the appropriate design form.

Bolt Loads in Bolted Flange Connections Bolted flange connections are designed to balance the gasket reaction with the bolt loads. Since these two loads are applied at different locations on the flange, a moment is developed in the flange. The flange geometry is proportioned to accommodate this internal moment. Flange design is the process of determining the load applied at the gasket due to seating or operating conditions and then finding the corresponding bolt load to counteract that load.

The ASME Code defines three separate distinct bolt loads used in the design of flanges. The three separate bolt loads are as follows;

1. W Flange design bolt load for operating or gasket seating as may apply
2. W_{m1} Required bolt load for operating condition
3. W_{m2} Required bolt load for gasket seating

The values of W_{m1} and W_{m2} are used to determine the bolt area required for the operating and gasket seating conditions respectively. Once the bolt area required is determined, and the actual bolt size and quantity selected, then the actual bolt load W, can be determined. Thus W is defined as the "Design Bolt Load" since it is not based on the theoretical required loads, but on the actual quantity and sizes of bolts used.

Ordinarily, the bolt area is selected to correspond closely with the minimum required. If excess bolting is provided, as is common practice with low pressure flanges, some recognition of this excess bolting should be made in the flange design to guard against excessive flange stress being developed when the bolts are tightened and to provide reasonable protection against abuse due to over tightening.

Factor "m" Gasket factor m is a dimensionless constant, also referred to as the maintenance factor. It has been found that the pressure on gasket faces required to avoid leakage must bear a minimum ratio, "m", to the hydrostatic pressure expected to be confined. The value m depends on the type of gasket material and the initial pressure to which the gasket is installed. It also depends on the type of flange facing used, but design methods usually account for this by choosing an effective gasket width. The term 'm' is actually a "pressure ratio". The ASME Code has used various pressure ratios, defined as the follows;

m = effective ratio @ mean gasket diameter
r = contact pressure ratio @ outside gasket diameter
m_a = absolute pressure ratio @ inside gasket diameter

Actual pressure ratios can be defined by the following equation;

$$m = \left(W - P^* \pi/4^* G^2\right) / \left(P^* \pi/4^* \left(OD^2 - ID^2\right)\right)$$

Equations for r or m_a can be accomplished by substituting OD or ID for G in the above equation. Relationships between the terms can be defined as follows;

$$m_a < m + 1/2$$

$$m_a = r + 1$$

The terms m, r and m_a are based on the same fundamental assumption; that some multiple of the confined *unit pressure* must be kept as a *unit pressure* on the gasket surfaces. This assumption is probably valid for small pressure ratios and soft gaskets with fluid-like characteristics. In this manner the pressure ratio is purported to be an indicator of leakage prediction. However it is a poor indicator of leakage prediction for hard or metal gaskets where the gasket material does not flow.

When the gasket width is increased, the m value is smaller than required for a narrow gasket from the equation. This would seem to indicate a dimensional constant involving load per linear inch of gasket, ratioed to the pressure.

The equation for m above was the basis for the development of the values given in the ASME Code. We do not calculate the value of m but utilize the values from the table.

Gasket Facing and Selection

The gasket facing and type correspond to the service conditions, fluid or gas handled, pressure, temperature, thermal shock, cyclic operation, and the gasket selection. The greater the hazard, the more care that should be invested in the decisions regarding gasket selection and facing details.

Facings which confine the gasket, such as male and female, tongue and groove, and ring joint offer greater security against blowouts. Male and female and tongue and groove have the disadvantage that mating flanges are not alike. These facings, which confine the gasket, are known as enclosed gaskets and are required for certain services, such as TEMA Class "R."

For tongue and groove flanges, the tongue is more likely to be damaged than the groove; therefore, from a maintenance standpoint, there is an advantage in placing the tongue on the part which can be transported for servicing, i.e., blind flanges, manway heads, etc. If the assembly of these joints is horizontal then there will be less difficulty if the groove is placed in the lower side of the joint. The gasket width should be made equal to the width of the tongue. Gaskets for these joints are typically metal or metal jacketed.

Table 3-1
Types of gaskets and surface finish

Type of Gasket		Surface Finish	Notes
1	Ring Type (Flat)	Concentric Serrated or stock finish	Thickness of gasket should be at least 3 times the depth of the grooves. Previously most commonly known as "compressed asbestos". Typical thickness' are 1/16" and 1/8".
2	Solid Metal a. Flat b. Profile c. Profile with Filler	 Concentric Serrated Very Smooth Smooth	Brute force seals. Gaskets made of flat metal. Relatively thin compared to width. Metal selection dependent on corrosion and temperature.
3	Spiral Wound (Note 1) a. Inner Ring Only (Style RIR) b. Outer Ring Only (Style CG) c. Inner and Outer Ring (Style CGI) d. No Inner or Outer Ring (Style R)	Smooth or Serrated; 125 to 250 AARH	Consists of preformed "V" shaped strip of metal which is wound into a spiral form with a compressible filler.
4	Metal Jacketed a. Flat Metal jacketed b. Corrugated Metal Jacketed	Very Smooth	Many variations. Consists of soft, compressible filler, either partially or wholly enclosed in a metal jacket.
5	Corrugated Metal a. With Filler b. Without Filler	Smooth	Line contact seals consisting of thin metal, corrugated or embossed. Can be coated with filler.
6	Elastomers	Concentric Serrated	Rubber, Fiber, Teflon, etc.
7	Ring Type Joint (RTJ) a. Hex b. Oval	Very Smooth	Metal gaskets sealed by line of contact and wedging action
8	Special a. Delta Ring b. Lens Ring c. Double Cone d. Bridgman e. O-Ring (Metal)	Very Smooth	Metal gasket sealed by line of contact and/or wedging action.

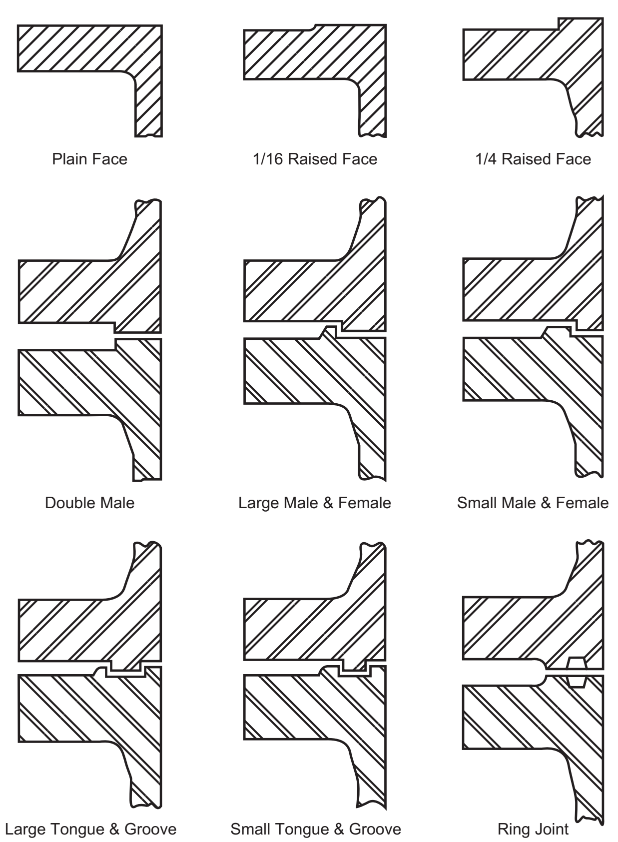
Notes:

1. ASME B16.20 requires solid metal inner rings for following;
   a. 900#: 24" & larger
   b. 1500#: 12" & larger
   c. 2500#: 4" & larger
   d. Vacuum service
   e. Surface finish $>$ 125 AARH

Table 3-2
Recommendations for gasket/facing selection

	Design Conditions			
Service	**Pressure Rating**	**Temperature Range, F**	**Facing**	**Gasket Type**
Oil or Hydrogen	150 to 600	1000 & Below	RF	Corrugated, Double Jacketed or Spiral Wound
		Above 1000	RTJ	Oval Ring
	900 and Higher	Any	RTJ	Oval Ring
Steam and Boiler Feedwater	900 or Lower	1000 & Below	RF	Corrugated, Double Jacketed or Spiral Wound
	1500	Any	RTJ	Oval Ring
Air	300 or Lower	750 & Below	RF	1/16" Flexible Graphite
		>750–1000	RF	Corrugated, Double Jacketed or Spiral Wound
		>1000–1200	RF	Spiral Wound
Water	150	250 & Below	RF	1/16" Flexible Graphite
		>250	RF	Corrugated, Double Jacketed or Spiral Wound
	300-900	Any	RF	Corrugated, Double Jacketed or Spiral Wound
	1500	Any	RTJ	Oval Ring
Fluid Catalyst	300 or Lower	1000 & Below	RF	Corrugated, Double Jacketed or Spiral Wound
		>1000–1200	RF	Spiral Wound
Toxic Fluids including Acids & Caustics	300 or Lower	750 & Lower	RF	Corrugated, Double Jacketed or Spiral Wound
Refrigerants and Refrigerated Hydrocarbons	Any	Below (–) 50	RF	Spiral Wound, Teflon Filled
		(–) 50 to 400	RF	Corrugated, Double Jacketed or Spiral Wound

Types of Contact Faces for Flanges



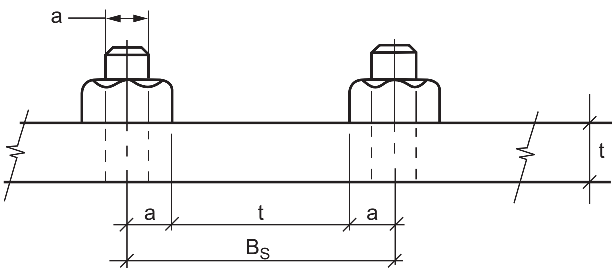
Gasket Contact Surface Finishes

The following is a list of typical surface finishes for flange faces;

1. Stock Finish: 250 to 500 AARH: This finish is produced with a continuous spiral groove generated with a round nosed tool. This finish is suitable for all ordinary service conditions and is the most widely used gasket surface finish
2. Spiral Serrated: 125 to 250 AARH: This finish is produced with a continuous spiral groove but utilizes a 90° included angle "V" tool. The groove is 1/64" deep and the feed is 1/32" for all sizes.
3. Concentric Serrated: 125 to 250 AARH: This finish is produced with concentric grooves using the same tools and parameters as the "spiral serrated" finish.
4. Smooth Finish: 63 to 125 AARH: This finish can be generated with a variety of tooling, but shows no tool marks apparent to the naked eye.
5. Cold Water Finish: 32 to 63 AARH: This finish is very smooth and has the appearance of a ground finish. It is mirrorlike in appearance.
6. Flat Face: Stock or Serrated
7. 1/16" Raised Face: Stock or Serrated
8. ¼" Raised Face: Stock or Serrated

9. Male & Female: Smooth
10. Tongue & Groove: Smooth
11. Side Wall of Ring Joint: 63 AARH

Bolt Spacing – Maximum and Minimum



Maximum Bolt Spacing: B_s Max $= 2a + t$
Minimum Bolt Spacing: B_s Min $= 2$ to $2.5 \times a$ (See Note 2)
Actual Bolt Spacing: $B_s = (\pi C)/n$
$C =$ Bolt Circle Diameter, in
$n =$ Quantity of Bolts (Multiple of 4)
$a =$ Bolt/Stud Diameter, in
$t =$ Flange Thickness, in

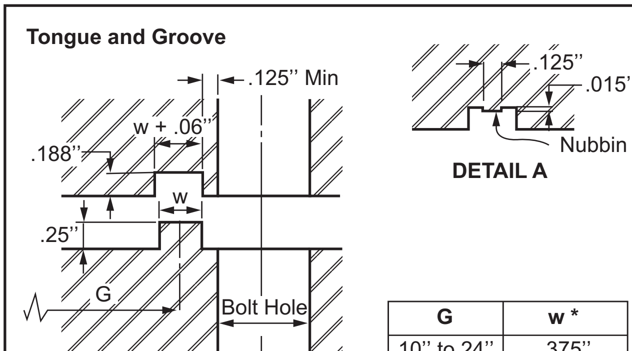
If bolt spacing exceeds B_s Max, multiply m_o and m_G by;

$$\sqrt{B_s/(2a + t)}$$

Notes

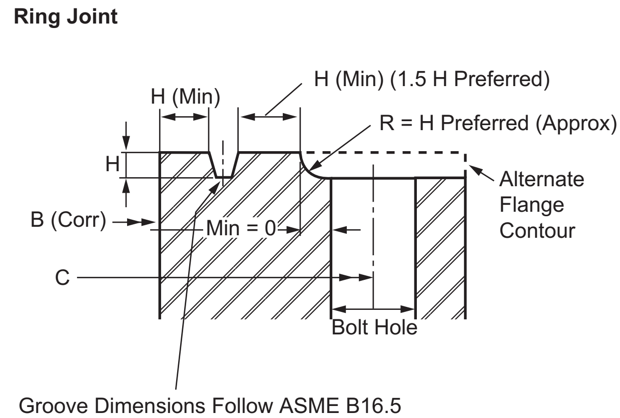
1. The requirements for bolt spacing is no longer an ASME Code requirement. However it is still good design practice.
2. B_s Min is based on wrench clearances. See Table 3-3.

Dimensions of Flange Faces



G	w *
10" to 24"	.375"
26" to 48"	.5"
48" to 72"	.625"

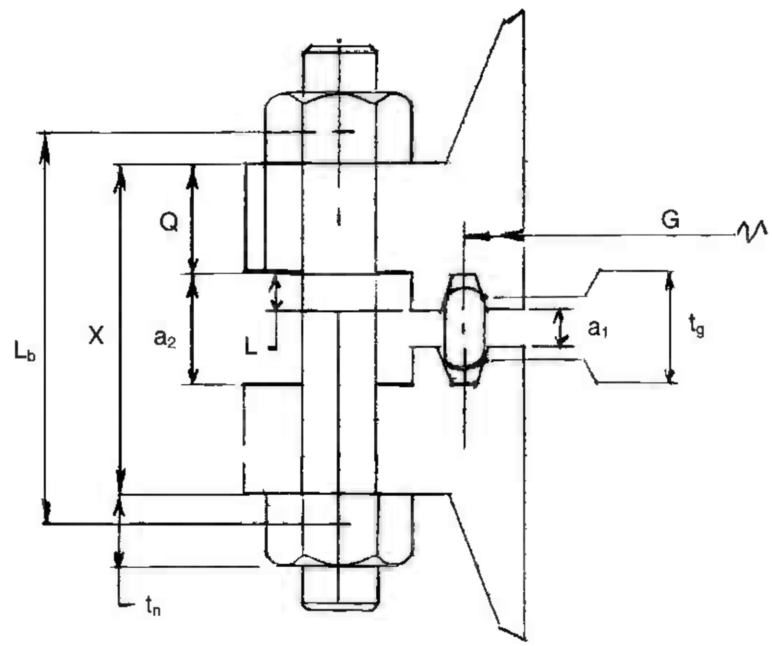
*Suggested Minimum



Notes

1. The procedures as outlined herein are based on Taylor Forge Bulletin No. 502, 7th Edition, entitled "Modern Flange Design." The forms and tables have been duplicated here for the user's convenience. The design forms are fast and accurate and are accepted throughout the industry. For additional information regarding flange design, please consult this excellent bulletin.
2. Flange calculations are done either as "integral" or "loose." A third classification, "optional," refers to flanges which do not fall into either of the foregoing categories and thus can be designed as either integral or loose. Definitions and examples of these categories are:
   - *Integral*—Hub and flange are one continuous structure either by manufacture or by full penetration welding. Some examples are:
     a. Welding neck flanges.
     b. Long weld neck flanges.
     c. Ring flanges attached with full penetration welds. Use design form "Type 1: Weld Neck Flange Design (Integral)," or "Type 3: Ring Flange Design."
   - *Loose*—Neither flange nor pipe has any attachment or is non-integral. It is assumed for purposes of analysis, that the hubs (if used) act independent of the pipe. Examples are:
     a. Slip-on flanges.
     b. Socket weld flanges.
     c. Lap joint flanges.
     d. Screwed flanges.
     e. Ring flanges attached without full penetration welds.

FLANGE DIMENSIONS FOR RTJ GASKET



Flange Size & Rating: ______________________

L = ______________________

Q = ______________________

Bolting:

Qty, n = ______________________

Size: ______________________

Length: ______________________

t_n = ______________________

d = ______________________

d_m = ______________________

$A_b = .25\,(\pi\, d^2\, n)$

Gasket:

w = ______________________

h = ______________________

Dimensions:

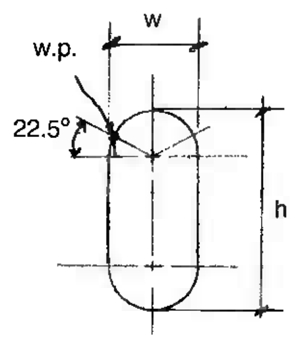
a_1 = ______________________

$a_2 = a_1 + 2L$

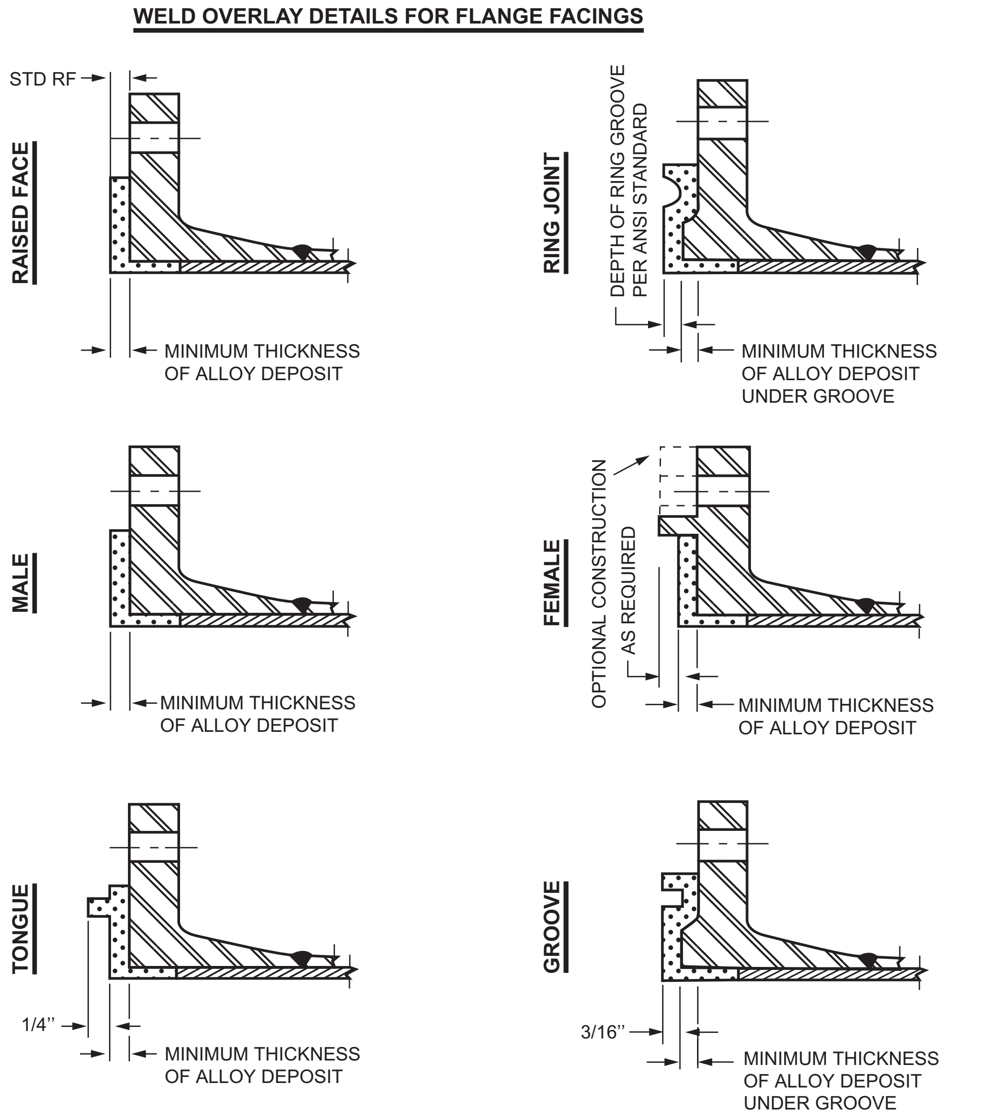
$X = 2Q + a_2$

$L_b = X + t_n$

$t_g = (h - w) + 2\,(\text{Sin } 22.5)\,(.5\,w)$



RTJ Gasket



Notes:

1. Raised face is not included in minimum flange thickness for 400#, 600#, 900#, 1500# and 2500#. Therefore the raised face can be removed and replaced with weld overlay. ASME B16.5 - 1996, Para 6.3.13 states; "*with ring joint groove or female face, the minimum flange thickness shall be first provided and then sufficient metal added so that the bottom of the groove is in the same plane as the flange edge of a full thickness flange*". Therefore the weld overlay must be assumed as not contributing to the flange or, as an option, the flange must be checked as a solid alloy flange.

2. The raised face thickness is included in the minimum flange thickness for 150# and 300#. Therefore the raised face may not be removed without rerating the flange. Weld overlay must be added above the elevation of the raised face.

Notes *(Cont)*

Use design form "Type 2: Slip-On Flange Design (Loose)," or "Type 3: Ring Flange Design."

3. Hubs have no minimum limit for h and g_o, but values of $g_o < 1.5\,t_n$ and $h < g_o$ are not recommended. For slip-on flanges as a first trial, use $g_1 = 2$ times pipe wall thickness.
4. The values of T, Z, Y, and U in Table 3-6 have been computed based on Poisson's ratio of 0.3.

Table 3-3
Dimensional data for bolts and flanges

	Standard Thread		8-Thread Series		Bolt Spacing						
Bolt Size	No. of Threads	Root Area, R_a, In^2	No. of Threads	Root Area, R_a, In^2	Minimum, B_S	Preferred	Minimum Radial Distance, R	Edge Distance, E	Nut Dimension, width across flats. W_{AF}	Max Fillet Radius at base of hub, r	Nut Stop, R_1
1/2"	13	0.126	N.A.		1.25	3	0.813	0.625	0.875	0.25	0.5
5/8"	11	0.202			1.5	3	0.938	0.75	1.063	0.313	0.594
3/4"	10	0.302			1.75	3	1.125	0.813	1.25	0.375	0.688
7/8"	9	0.419			2.0625	3	1.25	0.938	1.438	0.375	0.781
1"	8	0.551	8	0.551	2.25	3	1.375	1.063	1.63	0.438	0.875
1-1/8"	7	0.693	8	0.728	2.5	3	1.5	1.125	1.813	0.438	0.969
1-1/4"	7	0.89	8	0.929	2.8125	3	1.75	1.25	2	0.563	1.063
1-3/8"	6	1.054	8	1.155	3.0625	3.25	1.875	1.375	2.188	0.563	1.156
1-1/2"	6	1.294	8	1.405	3.25	3.25	2	1.5	2.375	0.625	1.25
1-5/8"	5.5	1.515	8	1.68	3.5	3.5	2.125	1.625	2.563	0.625	1.344
1-3/4"	5	1.744	8	1.98	3.75	3.75	2.25	1.75	2.75	0.625	1.438
1-7/8"	5	2.049	8	2.304	4	4	2.375	1.875	2.938	0.625	1.531
2"	4.5	2.3	8	2.652	4	4.25	2.5	2	3.125	0.688	1.625
2-1/4"	4.5	3.02	8	3.423	4.5	4.75	2.75	2.25	3.5	0.688	1.813
2-1/2"	4	3.715	8	4.292	5	5.25	3.063	2.375	3.875	0.813	2
2-3/4"	4	4.618	8	5.259	5.5	5.75	3.375	2.625	4.25	0.875	2.188
3"	4	5.621	8	6.324	6	6.25	3.625	2.875	4.625	0.938	2.375
3-1/4"	N.A.		8	7.49	6.5	8.125	3.88	3.25	5	1	2.563
3-1/2"			8	8.75	7	8.75	4.125	3.5	5.375	1.063	2.75
3-3/4"			8	10.11	7.5	9.375	4.375	3.75	5.75	1.125	2.938
4"			8	11.57	8	10	4.625	4	6.125	1.25	3.125

Notes:

1. All dimensions are in inches except as shown.
2. Nut Stop dimension, R_1, refers to a modified dimension, R, such that the hub prevents the nut from spinning without use of a wrench.

Notes *(Cont)*

5. B is the I.D. of the flange and not the pipe I.D. For small-diameter flanges when B is less than 20 g_1, it is optional for the designer to substitute B_1 for B in Code formula for longitudinal hub stress, S_H. (See [1, Para. 2-3 of Section VIII, Division 1].)

6. In general, bolts should always be used in multiples of 4. For large-diameter flanges, use many smaller bolts on a tight bolt circle to reduce the flange thickness. Larger bolts require a large bolt circle, which greatly increases flange thickness.
7. If the bolt holes are slotted to allow for swing-away bolting, substitute the diameter of the circle tangent to the inner edges of the slots for dimension A and follow the appropriate design procedures.
8. Square and oval flanges with circular bores should be treated as "inscribed" circular flanges. Use a bolt circle passing through the center of the outermost bolt holes. The same applies for noncircular openings; however, the bolt spacing becomes more critical. The spacing factor can be less than required for circular flanges since the metal available in the corners tends to spread the bolt load and even out the moment.
9. Design flanges to withstand both pressure and external loads, use "equivalent" pressure P_e as follows:

$$P_e = \frac{16M}{\pi G^3} + \frac{4F}{\pi G^2} + P$$

where M = bending moment, in.-lb
F = radial load, lb

10. Hubs: Minimum 3:1 taper. Ideal taper is 4:1. This allows the hub stresses to dissipate over a longer distance.
11. Maximum bolt spacing is 2a + t. For minimum bolt spacing see Table 3–3
12. Gasket width check:

$$N_{min} = (A_b\, S_a)/(2\pi yG)$$

13. Unit stress on a gasket: This value should not exceed twice the gasket yield point when the bolts are stressed to their nominal value (20,000 psi for alloy bolts)

$$S_g = (A_b\, S_a)/.785\left[(d_o - .125)^2 - d_i^2\right]$$

14. Only hubless flanges can be machined from plate. See UW-13(f) and Appendix 2-2.
15. The design of flanges includes the following;
    a. Flange type, integral or loose
    b. Flange configuration, lap joint, weld neck, slip on, etc.
    c. Gasket selection (type, material, dimensions)
    d. Flange facing type (RF, FF, M&F, T&G, ring joint)
    e. Flange face finish (smooth, rough, serrated)
    f. Bolting
    g. Hub proportions
    h. Flange width
16. In welded construction the nozzle neck comprised of the vessel or pipe wall to which it is attached, is considered to act as the hub.
17. The yield point of gasket material is related to the "m" value. It is not a "yield stress" in the conventional sense.
18. The flange and bolt design temperature may be reduced by 10% if the flange is not insulated.
19. Unless otherwise ordered, the manufacturer can supply either the spiral or concentric grooves if a "serrated" finish is specified.

Procedure 3-1: Design of Flanges [1–4]

Notation

A = flange O.D., in.
A_b = cross-sectional area of bolts, in.^{2}
A_m = total required cross-sectional area of bolts, in.^{2}
a = nominal bolt diameter, in.
B = flange I.D., in.
B_1 = flange I.D., in.
B_s = bolt spacing, in.
b = effective gasket width, in.
b_o = gasket seating width, in.
C = bolt circle diameter, in.
d = hub shape factor
d_1 = bolt hole diameter, in.
E, h_D, h_G, h_T, R = radial distances, in.
e = hub shape factor
F = hub shape factor for integral-type flanges
F_L = hub shape factor for loose-type flanges

f = hub stress correction factor for integral flanges
G = diameter at gasket load reaction, in.
g_o = thickness of hub at small end, in.
g_1 = thickness of hub at back of flange, in.
H = hydrostatic end force, lb
H_D = hydrostatic end force on area inside of flange, lb
H_G = gasket load, operating, lb
H_p = total joint-contact surface compression load, lb
H_T = pressure force on flange face, lb
h = hub length, in.
h_o = hub factor
M_D = moment due to H_D, in.-lb
M_G = moment due to H_G, in.-lb
M_o = total moment on flange, operating, in.-lb
M'_o = total moment on flange, seating
M_T = moment due to H_T, in.-lb
m = gasket factor
m_o = unit load, operating, lb
m_g = unit load, gasket seating, lb
N = width of gasket, in.
n = number of bolts
v = Poisson's ratio, 0.3 for steel
P = design pressure, psi
S_a = allowable stress, bolt, at ambient temperature, psi
S_b = allowable stress, bolt, at design temperature, psi
S_{fa} = allowable stress, flange, at ambient temperature, psi
S_{fo} = allowable stress, flange, at design temperature, psi
S_H = longitudinal hub stress, psi
S_R = radial stress in flange, psi
S_T = tangential stress in flange, psi
T, U, Y, Z = K-factors (see Table 3-6)
T_r, U_r, Y_r = K-factors for reverse flanges
t = flange thickness, in.
t_n = pipe wall thickness, in.
V = hub shape factor for integral flanges
V_L = hub shape factor for loose flanges
W = flange design bolt load, lb
W_{m1} = required bolt load, operating, lb
W_{m2} = required bolt load, gasket seating, lb
w = width of raised face or gasket contact width, in. (See Table 3-5)
y = gasket design seating stress, psi

Formulas

$$h_D = \frac{C - \text{dia. } H_D}{2}$$

$$h_T = \frac{C - \text{dia. } H_T}{2}$$

$$h_G = \frac{C - G}{2}$$

$$h_o = \sqrt{B\ g_o}$$

$$H_D = \frac{\pi B^2 P}{4}$$

$$H_T = H - H_D$$

$$H_G = \text{operating} = W_{ml} - H$$
$$\text{gasket seating} = W$$

$$H = \frac{\pi G^2 P}{4}$$

$$m_o = \frac{M_o}{B}$$

$$m_G = \frac{M_G}{B}$$

$$M_D = H_D h_D$$

$$M_T = H_T h_T$$

$$M_G = W h_G$$

$$E = \frac{A - C}{2}$$

$$K = \frac{A}{B}$$

$$T = \frac{\left(1 - v^2\right)\left(K^2 - 1\right) U}{(1 - v) + (1 + v) K^2}$$

$$Z = \frac{K^2 + 1}{K^2 - 1}$$

$$Y = \left(1 - v^2\right) U$$

$$U = \frac{K^2(1 + 4.6052\ (1 + v/1 - v)\ \log_{10} K) - 1}{1.0472\ (K^2 - 1)\ (K - 1)(1 + v)}$$

B_1 = loose flanges = $B + g_1$

= integral flanges, $f < 1 = B + g_1$

= integral flanges, $f \geq 1 = B + g_o$

d = loose flanges = $\dfrac{U h_o g_o^2}{V_L}$

= integral flanges = $\dfrac{U h_o g_o^2}{V}$

= reverse flanges = $\dfrac{U_r h_o g_o^2}{V}$

e = loose flanges = $\dfrac{F_L}{h_o}$

= integral flanges = $\dfrac{F}{h_o}$

G = (if $b_o \leq 0.25$ in.) mean diameter of gasket face

= (if $b_o > 0.25$ in.) O.D. of gasket contact face − 2b

Stress Formula Factors

$\alpha = t\,e + 1$

$\beta = 1.333\,t\,e + 1$

$\delta = \dfrac{t^3}{d}$

$\gamma = \dfrac{\alpha}{T}$ or $\dfrac{\alpha}{T_r}$ for reverse flanges

$\lambda = \gamma + \delta$

$$\alpha_R = \frac{1}{K^2}\left[1 + \frac{3(K+1)(1-v)}{\pi Y}\right]$$

For factors, F, V, F_L, and V_L, see Table 2-7.1 of the ASME Code [1].

TYPE 1: WELD NECK FLANGE DESIGN (INTEGRAL)

1	DESIGN CONDITIONS				
Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2	GASKET AND FACING DETAILS		
Gasket		Facing	

3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = 0.5(A_m + A_b)S_a$	
m					

5	MOMENT CALCULATIONS				
Load		× Lever Arm		= Moment	
Operating					
$H_D = \pi B^2P/4$		$h_D = R + 0.5g_1$		$M_D = H_Dh_D$	
$H_G = W_{m1} - H$		$h_G = 0.5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = 0.5(R + g_1 + h_G)$		$M_T = H_Th_T$	
				M_o	
Seating					
$H_G = W$		$h_G = 0.5(C - G)$		M_o'	

6	K AND HUB FACTORS		
$K = A/B$		h/h_o	
T		F	
Z		V	
Y		f	
U		$e = F/h_o$	
g_1/g_o		$d = \frac{U}{V} h_o g_o^2$	
$h_o = \sqrt{Bg_o}$			

7	STRESS FORMULA FACTORS		
t			
$\alpha = te + 1$			
$\beta = 4/3\ te + 1$			
$\gamma = \alpha/T$			
$\delta = t^3/d$			
$\lambda = \gamma + \delta$			
$m_o = M_o/B$			
$m_G = M_o'/B$			

If bolt spacing exceeds $2a + t$, multiply m_o and m_G in above equation by: $\sqrt{\dfrac{\text{Bolt spacing}}{2a + t}}$

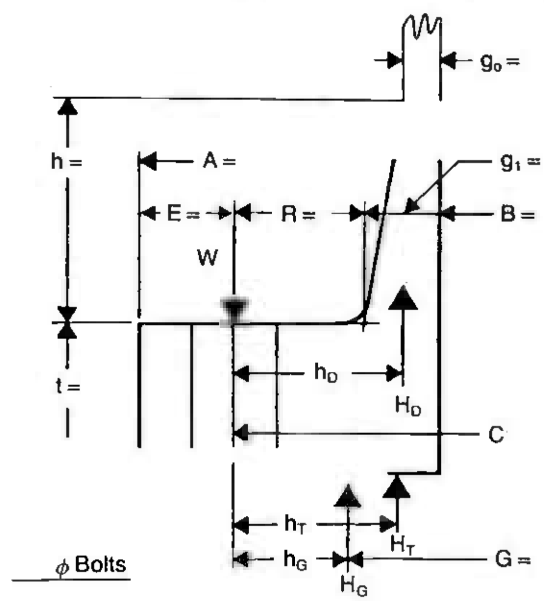


Figure 3-1. Dimensional data and forces for a weld neck flange (integral).

8	STRESS CALCULATIONS				
Allowable Stress	Operating		Allowable Stress	Seating	
$1.5\ S_{fo}$	Longitudinal hub, $S_H = fm_o/\lambda g_1^2$		$1.5\ S_{fa}$	Longitudinal hub, $S_H = fm_G/\lambda g_1^2$	
S_{fo}	Radial flange, $S_R = \beta m_o/\lambda t^2$		S_{fa}	Radial flange, $S_R = \beta m_G/\lambda t^2$	
S_{fo}	Tangential flange, $S_T = m_oY/t^2 - ZS_R$		S_{fa}	Tangential flange, $S_T = m_GY/t^2 - ZS_R$	
S_{fo}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$		S_{fa}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$	

Adapted from Taylor Forge International, Inc., by permission.

TYPE 2: SLIP-ON FLANGE DESIGN (LOOSE)

1	DESIGN CONDITIONS				
Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2	GASKET AND FACING DETAILS		
Gasket		Facing	

3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = 0.5(A_m + A_b)S_a$	
m					

5	MOMENT CALCULATIONS				
Load		× Lever Arm		= Moment	
Operating					
$H_D = \pi B^2P/4$		$h_D = R + g_1$		$M_D = H_Dh_D$	
$H_G = W_{m1} - H$		$h_G = 0.5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = 0.5(R + g_1 + h_G)$		$M_T = H_Th_T$	
				M_o	
Seating					
$H_G = W$		$h_G = 0.5(C - G)$		M_o'	

6	K AND HUB FACTORS		
$K = A/B$		h/h_o	
T		F_L	
Z		V_L	
Y			
U		$e = \frac{F_L}{h_o}$	
g_1/g_o		$d = \frac{U}{V}h_og_o^2$	
$h_o = \sqrt{Bg_o}$			

7	STRESS FORMULA FACTORS		
t			
$\alpha = te + 1$			
$\beta = 4/3\ te + 1$			
$\gamma = \alpha/T$			
$\delta = t^3/d$			
$\lambda = \gamma + \delta$			
$m_o = M_o/B$			
$m_G = M_o'/B$			

If bolt spacing exceeds 2a + t, multiply m_o and m_g in above equation by: $\sqrt{\frac{\text{Bolt spacing}}{2a + t}}$

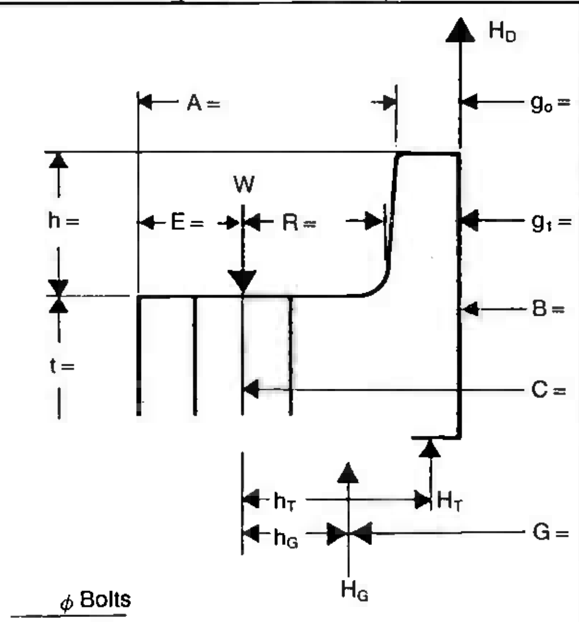


Figure 3-2. Dimensional data and forces for a slip-on flange (loose).

8	STRESS CALCULATIONS				
Allowable Stress	Operating		Allowable Stress	Seating	
1.5 S_{fo}	Longitudinal hub, $S_H = m_o/\lambda g_1^2$		1.5 S_{fa}	Longitudinal hub, $S_H = m_G/\lambda g_1^2$	
S_{fo}	Radial flange, $S_R = \beta m_o/\lambda t^2$		S_{fa}	Radial flange, $S_R = \beta m_G/\lambda t^2$	
S_{fo}	Tangential flange, $S_T = m_oY/t^2 - ZS_R$		S_{fa}	Tangential flange, $S_T = m_GY/t^2 - ZS_R$	
S_{fo}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$		S_{fa}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$	

Adapted from Taylor Forge International, Inc., by permission.

TYPE 3: RING FLANGE DESIGN

1		DESIGN CONDITIONS			
Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2	GASKET AND FACING DETAILS		
Gasket		Facing	

3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = 0.5(A_m + A_b)S_a$	
m					

5		MOMENT CALCULATIONS			
Load		× Lever Arm		= Moment	
		Operating			
$H_D = \pi B^2P/4$		$h_D = 0.5(C - B)$		$M_D = H_Dh_D$	
$H_G = W_{m1} - H$		$h_G = 0.5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = 0.5(h_D + h_G)$		$M_T = H_Th_T$	
				M_o	
		Seating			
$H_G = W$		$h_G = 0.5(C - G)$		M_o'	

6	SHAPE CONSTANTS		
K = A/B		Y	

If bolt spacing exceeds 2a + t, multiply M_o and M_o' in above equation by: $\sqrt{\dfrac{\text{Bolt spacing}}{2a + t}}$

7	FLANGE THICKNESS REQUIRED
t = greater of	
Operating	Seating
$t = \sqrt{\dfrac{M_oY}{S_{fo}B}}$	$t = \sqrt{\dfrac{M_o'Y}{S_{fa}B}}$

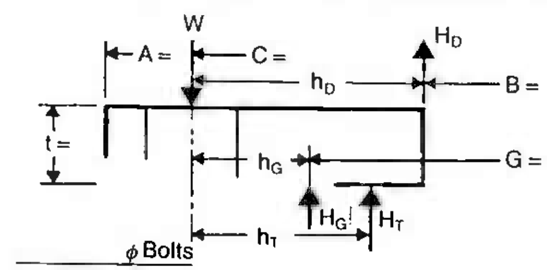


Figure 3-3. Dimensional data and forces for a ring flange.

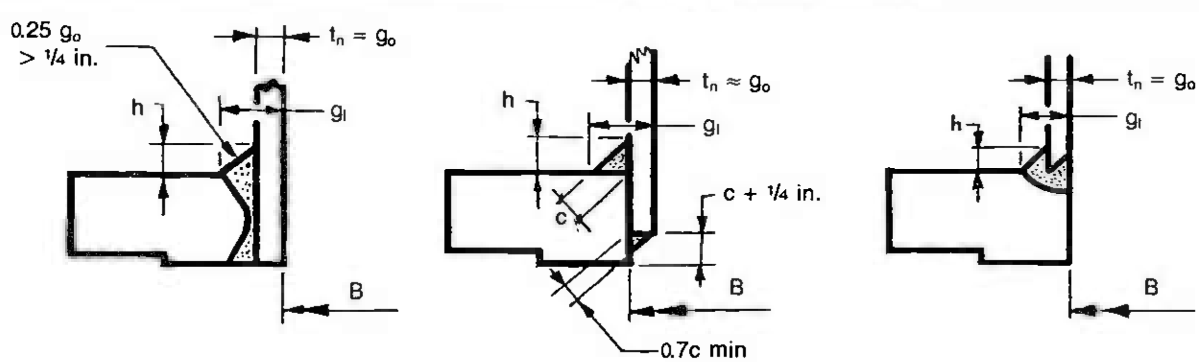


Figure 3-4. Various attachments of ring flanges. (All other dimensions and loadings per Figure 3-2.)

8 NOTES

If $g_o < 1.5t_n$ and $h < g_o$, design as integral. If $g_o > 1.5t_n$ and $h > g_o$, design as loose.
If $g_o \le 5/8$ in., $B/g_o \le 300$, $P \le 300$ psi and design temp. < 700°, design as integral or loose.

c = lesser of t_n or $\left\{\begin{array}{l}\text{loose: } 2t_m \\ \text{integral: } 2g_o\end{array}\right\}$ but not less than 1/4 in.

Adapted from Taylor Forge International, Inc., by permission.

TYPE 4: REVERSE FLANGE DESIGN

1	DESIGN CONDITIONS				
Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2	GASKET AND FACING DETAILS		
Gasket		Facing	

3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = 0.5(A_m + A_b)S_a$	
m					

5	MOMENT CALCULATIONS				
Load		× Lever Arm		=	Moment
		Operating			
$H_D = \pi B^2P/4$		$h_D = 0.5(C + g_1 - 2g_o - B)$		$M_D = H_D h_D$	
$H_G = W_{m1} - H$		$h_G = 0.5(C - G)$		$M_G = H_G h_G$	
$H_T = H - H_D$		$h_T = 0.5(C - (B + G)/2)$		$M_T = H_T h_T$	
Add moments algebraically, then use the absolute value $\lvert M_o \rvert$ in all subsequent calculations.				$\lvert M_o \rvert$	
		Seating			
$H_G = W$		$h_G = 0.5(C - G)$		M_o'	

6	K AND HUB FACTORS		
$K = A/B'$		h/h_o	
T		F	
Z		V	
Y		f	
U		$e = F/h_o$	
g_1/g_o		$d = \frac{U_R}{V} h_o g_o^2$	
$h_o = \sqrt{Ag_o}$			
$Y_R = \alpha_R Y$		$U_R = \alpha_R U$	
$\alpha_R = \frac{1}{K^2}\left[1 + \frac{3(K + 1)(1 - \nu)}{\pi Y}\right]$			
$T_R = \frac{(Z + \nu)}{(Z - \nu)}\alpha_R T$			

7	STRESS FORMULA FACTORS		
t		$\delta = t^3/d$	
$\alpha = te + 1$		$\lambda = \gamma + \delta$	
$\beta = 4/3\ te + 1$		$m_o = M_o/B'$	
$\gamma = \alpha/T_R$		$m_G = M_o'/B'$	

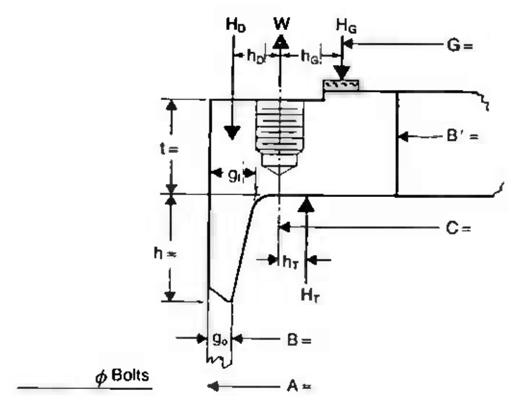


Figure 3-5. Dimensional data and forces for a reverse flange.

8	STRESS CALCULATIONS				
Allowable Stress	Operating		Allowable Stress	Seating	
1.5 S_{fo}	Longitudinal hub, $S_H = fm_o/\lambda g_1^2$		1.5 S_{fa}	Longitudinal hub, $S_H = fm_G/\lambda g_1^2$	
S_{fo}	Radial flange, $S_R = \beta m_o/\lambda t^2$		S_{fa}	Radial flange, $S_R = \beta m_G/\lambda t^2$	
S_{fo}	Tangential flange, $S_T = m_o Y_R/t^2 - ZS_R \frac{(0.67te + 1)}{\beta}$		S_{fa}	Tangential flange, $S_T = m_G Y_R/t^2 - ZS_R \frac{(0.67te + 1)}{\beta}$	
S_{fo}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$		S_{fa}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$	
S_{fo}	Tangential flange $S_T(A_T B')$ $= \frac{m_o}{t^2}\left[Y - \frac{2k^2\left(1 + \frac{2}{3}te\right)}{(k^2 - 1)\lambda}\right]$		S_{fa}	Tangential flange $S_T(A_T B') = \frac{m_G}{t^2}\left[Y - \frac{2k^2\left(1 + \frac{2}{3}te\right)}{(k^2 - 1)\lambda}\right]$	

Adapted from Taylor Forge International, Inc., by permission.

TYPE 5: SLIP-ON FLANGE, FLAT FACE, FULL GASKET

1 — DESIGN CONDITIONS

Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2 — GASKET AND FACING DETAILS

Gasket		Facing	

3 | **4 — LOAD AND BOLT CALCULATIONS**

$G = C - 2h_G$		$W_{m2} = b\pi Gy + H'_{GY}$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
$b = (C - B)/4$		$H_P = 2b\pi GmP$			
y		$H'_P = (h_G/h'_G)H_P$		A_b	
m		$H = G^2\pi P/4$		$W = 0.5(A_m + A_b)S_a$	
		$W_{m1} = H + H_P + H'_P$		$H'_{GY} = (h_G/h'_G)b\pi Gy$	

5 — MOMENT CALCULATIONS

Load		× Lever Arm		= Moment	
Operating					
$H_D = \pi B^2P/4$		$h_D = R + g_1$		$M_D = H_Dh_D$	
$H_T = H - H_D$		$h_T = 0.5(R + g_1 + h_G)$		$M_T = H_Th_T$	
				M_o	
Lever Arms					
$h_G = \dfrac{(C - B)(2B + C)}{6(B + C)}$			$h'_G = \dfrac{(A - C)(2A + C)}{6(C + A)}$		
Reverse Moment					
$H_G = W - H$		$h''_G = \dfrac{h_Gh'_G}{h_G + h'_G}$		$M_G = H_Gh''_G$	

6 — K AND HUB FACTORS

$K = A/B$		h/h_o	
T		F_L	
Z		V_L	
Y		$e = \dfrac{F_L}{h_o}$	
U			
g_1/g_o		$d = \dfrac{U}{V_L}h_og_o^2$	
$h_o = \sqrt{Bg_o}$			

7 — STRESS FORMULA FACTORS

t		$\delta = t^3/d$	
$\alpha = te + 1$		$\lambda = \gamma + \delta$	
$\beta = 4/3\ te + 1$		$m_o = M_o/B$	
$\gamma = \alpha/T$			

If bolt spacing exceeds 2a + t, multiply m_o in above equation by: $\sqrt{\dfrac{\text{Bolt spacing}}{2a + t}}$

8 — STRESS CALCULATIONS

Allowable Stress	Operating	
1.5 S_{fo}	Longitudinal hub, $S_H = m_o/\lambda g_1^2$	
S_{fo}	Radial flange, $S_R = \beta m_o/\lambda t^2$	
S_{fo}	Tangential flange, $S_T = m_oY/t^2 - ZS_R$	
S_{fo}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$	
S_{fo}	Radial stress at bolt circle $S_{RAD} = \dfrac{6M_G}{t^2(\pi C - nd_1)}$	

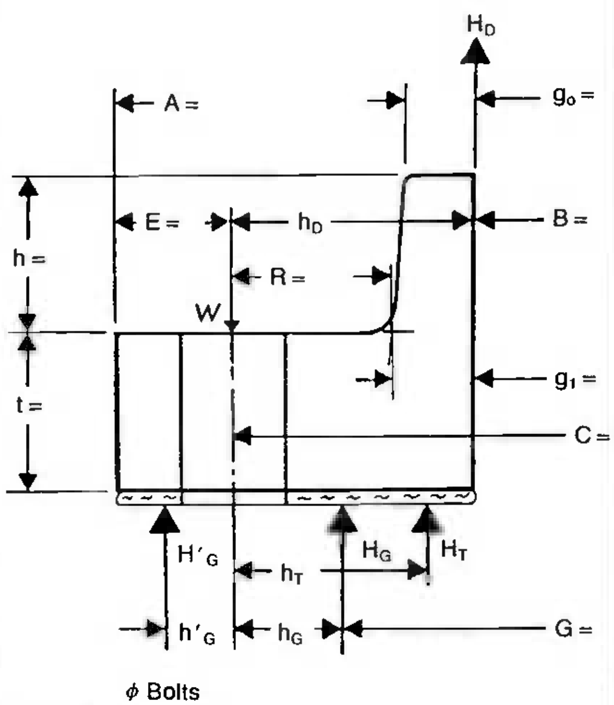


Figure 3-6. Dimensional data and forces for a slip-on flange, flat face, full gasket.

Adapted from Taylor Forge International, Inc., by permission.

Table 3-4
Gasket materials and contact facings[1]
Gasket factors (m) for operating conditions and minimum design stress (y)

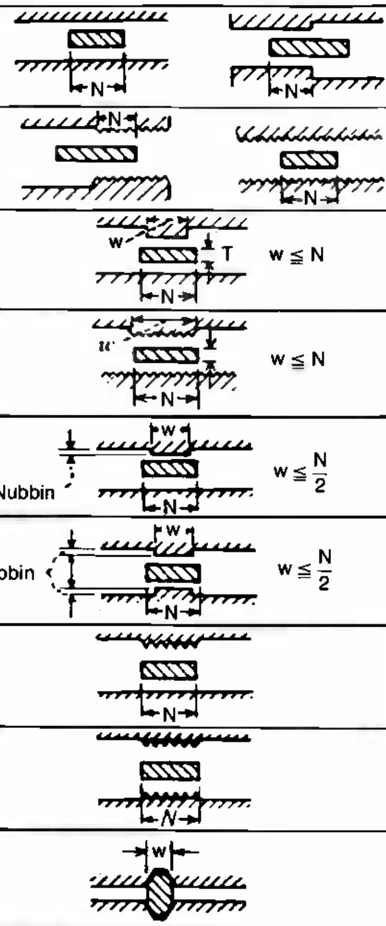
Gasket Material		Gasket Factor m	Min. Design Seating Stress y	Sketches and Notes	Use Facing Sketch (Refer to Table 3-5)	Use Column III (Refer to Table 3-5)
Self-energizing types: O rings, metallic, elastomer or other gasket types considered as self-sealing		0	0			
Elastomers without fabric or a high percentage of asbestos fiber: Below 75A Shore Durometer		0.50	0		(1a), (1b), (1e), (1d), (4), (5)	II
75A or higher Shore Durometer		1.00	200		(1a), (1b), (1e), (1d), (4), (5)	II
Asbestos with a suitable binder for the operating conditions	1/8 thick	2.00	1,600		(1a), (1b), (1e), (1d), (4), (5)	II
	1/16 thick	2.75	3,700			
	1/32 thick	3.50	6,500			
Elastomers with cotton fabric insertion		1.25	400		(1a), (1b), (1e), (1d), (4), (5)	II
Elastomers with asbestos fabric insertion, with or without wire reinforcement	3-ply	2.25	2,200		(1a), (1b), (1e), (1d), (4), (5)	II
	2-ply	2.50	2,900			
	1-ply	2.75	3,700			
Vegetable fiber		1.75	1,100		(1a), (1b), (1e), (1d), (4), (5)	II
Spiral-wound metal, asbestos filled	Carbon	2.50	10,000			II
	Stainless or Monel	3.00	10,000			
Corrugated metal, asbestos inserted or corrugated metal, jacketed asbestos filled	Soft aluminum	2.50	2,900			II
	Soft copper or brass	2.75	3,700			
	Iron or soft steel	3.00	4,500			
	Monel or 4%–6% chrome	3.25	5,500			
	Stainless steels	3.50	6,500			
Corrugated metal	Soft aluminum	2.75	3,700		(1a), (1b), (1c), (1d)	II
	Soft copper or brass	3.00	4,500			
	Iron or soft steel	3.25	5,500			
	Monel or 4%–6% chrome	3.50	6,500			
	Stainless steels	3.75	7,600			
Flat metal jacketed asbestos filled	Soft aluminum	3.25	5,500		(1a), (1b), (1c), (1d), (2)[2]	II
	Soft copper or brass	3.50	6,500			
	Iron or soft steel	3 75	7,600			
	Monel or	3.50	8,000			
	4%–6% chrome	3.75	9,000			
	Stainless steels	3.75	9,000			
Grooved metal	Soft aluminum	3.25	5,500		(1a), (1b), (1c), (1d), (2), (3)	II
	Soft copper or brass	3.50	6,500			
	Iron or soft steel	3.75	7,600			
	Monel or 4%–6% chrome	3.75	9,000			
	Stainless steels	4.25	10,100			
Solid flat metal	Soft aluminum	4.00	8,800		(1a), (1b), (1c), (1d), (2), (3), (4), (5)	I
	Soft copper or brass	4.75	13,000			
	Iron or soft steel	5.50	18,000			
	Monel or 4%–6% chrome	6.00	21,800			
	Stainless steels	6.50	26,000			
Ring joint	Iron or soft steel	5.50	18,000		(6)	I
	Monel or 4%–6% chrome	6.00	21,800			
	Stainless steels	6.50	26,000			

Notes:

1. This table gives a list of many commonly used gasket materials and contact facings with suggested design values of m and y that have generally proved satisfactory in actual service when using effective gasket seating width b, given in Table 3-5. The design values and other details given in this table are suggested only and are not mandatory.
2. The surface of a gasket having a lap should not be against the nubbin.

Reprinted by permission from ASME Code Section VIII Div. 1, Table 2-5.1

Table 3-5
Effective gasket width



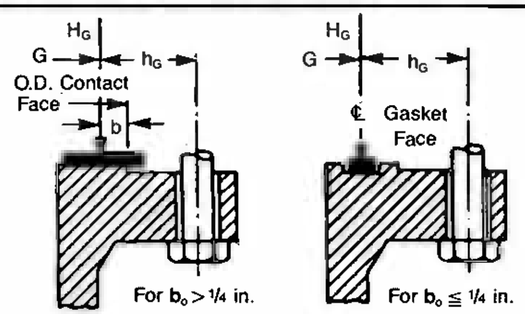
Facing Sketch (Exaggerated)	Basic Gasket Seating Width, b_o Column I	Column II
(1a)	$\frac{N}{2}$	$\frac{N}{2}$
(1b)[1]		
(1c) $w \leq N$	$\frac{w+T}{2}; \left(\frac{w+N}{4}\max\right)$	$\frac{w-T}{2}; \left(\frac{w-N}{4}\max\right)$
(1d) $w \leq N$		
(2) 1/64 in. Nubbin $w \leq \frac{N}{2}$	$\frac{w+N}{4}$	$\frac{w+3N}{8}$
(3) 1/64 in. Nubbin $w \leq \frac{N}{2}$	$\frac{N}{4}$	$\frac{3N}{8}$
(4)*	$\frac{3N}{8}$	$\frac{7N}{16}$
(5)*	$\frac{N}{4}$	$\frac{3N}{8}$
(6)	$\frac{w}{8}$	

Effective Gasket Seating Width, b

$b = b_o$, when $b_o \leq 1/4$ in.

$b = \frac{\sqrt{b_o}}{2}$, when $b_o > 1/4$ in.

Location of Gasket Load Reaction



Note: The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes

* Where serrations do not exceed 1/64-in. depth and 1/32-in. width spacing, sketches (1b) and (1d) shall be used.

Reprinted by permission from ASME Code Section VIII Div. 1, Table 2-5.2

Table 3-6
Table of Coefficients

K	T	Z	Y	U	K	T	Z	Y	U	K	T	Z	Y	U	K	T	Z	Y	U
1.001	1.91	1000.50	1911.16	2100.18	1.046	1.90	22.05	42.75	46.99	1.091	1.88	11.52	22.22	24.41	1.136	1.86	7.88	15.26	16.77
1.002	1.91	500.50	956.16	1050.72	1.047	1.90	21.79	41.87	46.03	1.092	1.88	11.40	21.99	24.16	1.137	1.86	7.83	15.15	16.65
1.003	1.91	333.83	637.85	700.93	1.048	1.90	21.35	41.02	45.09	1.093	1.88	11.28	21.76	23.91	1.138	1.86	7.78	15.05	16.54
1.004	1.91	250.50	478.71	526.05	1.049	1.90	20.92	40.21	44.21	1.094	1.88	11.16	21.54	23.67	1.139	1.86	7.73	14.95	16.43
1.005	1.91	200.50	383.22	421.12	1.050	1.89	20.51	39.43	43.34	1.095	1.88	11.05	21.32	23.44	1.140	1.86	7.68	14.86	16.35
1.006	1.91	167.17	319.56	351.16	1.051	1.89	20.12	38.68	42.51	1.096	1.88	10.94	21.11	23.20	1.141	1.86	7.62	14.76	16.22
1.007	1.91	143.36	274.09	301.20	1.052	1.89	19.74	37.96	41.73	1.097	1.88	10.83	20.91	22.97	1.142	1.86	7.57	14.66	16.11
1.008	1.91	125.50	239.95	263.75	1.053	1.89	19.38	37.27	40.96	1.098	1.88	10.73	20.71	22.75	1.143	1.86	7.53	14.57	16.01
1.009	1.91	111.61	213.40	234.42	1.054	1.89	19.03	36.60	40.23	1.099	1.88	10.62	20.51	22.39	1.144	1.86	7.48	14.48	15.91
1.010	1.91	100.50	192.19	211.19	1.055	1.89	18.69	35.96	39.64	1.100	1.88	10.52	20.31	22.18	1.145	1.86	7.43	14.39	15.83
1.011	1.91	91.41	174.83	192.13	1.056	1.89	18.38	35.34	38.84	1.101	1.88	10.43	20.15	22.12	1.146	1.86	7.38	14.29	15.71
1.012	1.91	83.84	160.38	176.25	1.057	1.89	18.06	34.74	38.19	1.102	1.88	10.33	19.94	21.92	1.147	1.86	7.34	14.20	15.61
1.013	1.91	77.43	148.06	162.81	1.058	1.89	17.76	34.17	37.56	1.103	1.88	10.23	19.76	21.72	1.148	1.86	7.29	14.12	15.51
1.014	1.91	71.93	137.69	151.30	1.059	1.89	17.47	33.62	36.95	1.104	1.88	10.14	19.58	21.52	1.149	1.86	7.25	14.03	15.42
1.015	1.91	67.17	128.61	141.33	1.060	1.89	17.18	33.04	36.34	1.105	1.88	10.05	19.38	21.30	1.150	1.86	7.20	13.95	15.34
1.016	1.90	63.00	120.56	132.49	1.061	1.89	16.91	32.55	35.78	1.106	1.88	9.96	19.33	21.14	1.151	1.86	7.16	13.86	15.23
1.017	1.90	59.33	111.98	124.81	1.062	1.89	16.64	32.04	35.21	1.107	1.87	9.87	19.07	20.96	1.152	1.86	7.11	13.77	15.14
1.018	1.90	56.06	107.36	118.00	1.063	1.89	16.40	31.55	34.68	1.108	1.87	9.78	18.90	20.77	1.153	1.86	7.07	13.69	15.05
1.019	1.90	53.14	101.72	111.78	1.064	1.89	16.15	31.08	34.17	1.109	1.87	9.70	18.74	20.59	1.154	1.86	7.03	13.61	14.96
1.020	1.90	50.51	96.73	106.30	1.065	1.89	15.90	30.61	33.65	1.110	1.87	9.62	18.55	20.38	1.155	1.86	6.99	13.54	14.87
1.021	1.90	48.12	92.21	101.33	1.066	1.89	15.67	30.17	33.17	1.111	1.87	9.54	18.42	20.25	1.156	1.86	6.95	13.45	14.78
1.022	1.90	45.96	88.04	96.75	1.067	1.89	15.45	29.74	32.69	1.112	1.87	9.46	18.27	20.08	1.157	1.86	6.91	13.37	14.70
1.023	1.90	43.98	84.30	92.64	1.068	1.89	15.22	29.32	32.22	1.113	1.87	9.38	18.13	19.91	1.158	1.86	6.87	13.30	14.61
1.024	1.90	42.17	80.81	88.81	1.069	1.89	15.02	28.91	31.79	1.114	1.87	9.30	17.97	19.75	1.159	1.86	6.83	13.22	14.53
1.025	1.90	40.51	77.61	85.29	1.070	1.89	14.80	28.51	31.34	1.115	1.87	9.22	17.81	19.55	1.160	1.86	6.79	13.15	14.45
1.026	1.90	38.97	74.70	82.09	1.071	1.89	14.61	28.13	30.92	1.116	1.87	9.15	17.68	19.43	1.161	1.85	6.75	13.07	14.36
1.027	1.90	37.54	71.97	79.08	1.072	1.89	14.41	27.76	30.51	1.117	1.87	9.07	17.54	19.27	1.162	1.85	6.71	13.00	14.28
1.028	1.90	36.22	69.43	76.30	1.073	1.89	14.22	27.39	30.11	1.118	1.87	9.00	17.40	19.12	1.163	1.85	6.67	12.92	14.20
1.029	1.90	34.99	67.11	73.75	1.074	1.88	14.04	27.04	29.72	1.119	1.87	8.94	17.27	18.98	1.164	1.85	6.64	12.85	14.12
1.030	1.90	33.84	64.91	71.33	1.075	1.88	13.85	26.69	29.34	1.120	1.87	8.86	17.13	18.80	1.165	1.85	6.60	12.78	14.04
1.031	1.90	32.76	62.85	69.06	1.076	1.88	13.68	26.36	28.98	1.121	1.87	8.79	17.00	18.68	1.166	1.85	6.56	12.71	13.97
1.032	1.90	31.76	60.92	66.94	1.077	1.88	13.56	26.03	28.69	1.122	1.87	8.72	16.87	18.54	1.167	1.85	6.53	12.64	13.89
1.033	1.90	30.81	59.11	64.95	1.078	1.88	13.35	25.72	28.27	1.123	1.87	8.66	16.74	18.40	1.168	1.85	6.49	12.58	13.82

1.034	1.90	29.92	57.41	63.08	1.079	1.88	13.18	25.40	27.92	1.124	1.87	8.59	16.62	18.26	1.169	1.85	6.46	12.51	13.74
1.035	1.90	29.08	55.80	61.32	1.080	1.88	13.02	25.10	27.59	1.125	1.87	8.53	16.49	18.11	1.170	1.85	6.42	12.43	13.66
1.036	1.90	28.29	54.29	59.66	1.081	1.88	12.87	24.81	27.27	1.126	1.87	8.47	16.37	17.99	1.171	1.85	6.39	12.38	13.60
1.037	1.90	27.54	52.85	58.08	1.082	1.88	12.72	24.52	26.95	1.127	1.87	8.40	16.25	17.86	1.172	1.85	6.35	12.31	13.53
1.038	1.90	26.83	51.50	56.59	1.083	1.88	12.57	24.24	26.65	1.128	1.87	8.34	16.14	17.73	1.173	1.85	6.32	12.25	13.46
1.039	1.90	26.15	50.21	55.17	1.084	1.88	12.43	24.00	26.34	1.129	1.87	8.28	16.02	17.60	1.174	1.85	6.29	12.18	13.39
1.040	1.90	25.51	48.97	53.82	1.085	1.88	12.29	23.69	26.05	1.130	1.87	8.22	15.91	17.48	1.175	1.85	6.25	12.10	13.30
1.041	1.90	24.90	47.81	53.10	1.086	1.88	12.15	23.44	25.77	1.131	1.87	8.16	15.79	17.35	1.176	1.85	6.22	12.06	13.25
1.042	1.90	24.32	46.71	51.33	1.087	1.88	12.02	23.18	25.48	1.132	1.87	8.11	15.68	17.24	1.177	1.85	6.19	12.00	13.18
1.043	1.90	23.77	45.64	50.15	1.088	1.88	11.89	22.93	25.20	1.133	1.86	8.05	15.57	17.11	1.178	1.85	6.16	11.93	13.11
1.044	1.90	23.23	44.64	49.05	1.089	1.88	11.76	22.68	24.93	1.134	1.86	7.99	15.46	16.99	1.179	1.85	6.13	11.87	13.05
1.045	1.90	22.74	43.69	48.02	1.090	1.88	11.63	22.44	24.66	1.135	1.86	7.94	15.36	16.90	1.180	1.85	6.10	11.79	12.96
1.182	1.85	6.04	11.70	12.86	1.278	1.81	4.16	8.05	8.85	1.434	1.74	2.89	5.56	6.10	1.75	1.60	1.97	3.64	4.00
1.184	1.85	5.98	11.58	12.73	1.281	1.81	4.12	7.98	8.77	1.438	1.74	2.87	5.52	6.05	1.76	1.60	1.95	3.61	3.96
1.186	1.85	5.92	11.47	12.61	1.284	1.80	4.08	7.91	8.69	1.442	1.74	2.85	5.48	6.01	1.77	1.60	1.94	3.57	3.93
1.188	1.85	5.86	11.36	12.49	1.287	1.80	4.05	7.84	8.61	1.446	1.74	2.83	5.44	5.97	1.78	1.59	1.92	3.54	3.89
1.190	1.84	5.81	11.26	12.37	1.290	1.80	4.01	7.77	8.53	1.450	1.73	2.81	5.40	5.93	1.79	1.59	1.91	3.51	3.85
1.192	1.84	5.75	11.15	12.25	1.293	1.80	3.98	7.70	8.46	1.454	1.73	2.80	5.36	5.89	1.80	1.58	1.89	3.47	3.82
1.194	1.84	5.70	11.05	12.14	1.296	1.80	3.94	7.63	8.39	1.458	1.73	2.78	5.32	5.85	1.81	1.58	1.88	3.44	3.78
1.196	1.84	5.65	10.95	12.03	1.299	1.80	3.91	7.57	8.31	1.462	1.73	2.76	5.28	5.80	1.82	1.58	1.86	3.41	3.75
1.198	1.84	5.60	10.85	11.92	1.302	1.80	3.88	7.50	8.24	1.466	1.73	2.74	5.24	5.76	1.83	1.57	1.85	3.38	3.72
1.200	1.84	5.55	10.75	11.81	1.305	1.80	3.84	7.44	8.18	1.470	1.72	2.72	5.20	5.71	1.84	1.57	1.84	3.35	3.69
1.202	1.84	5.50	10.65	11.71	1.308	1.79	3.81	7.38	8.11	1.475	1.72	2.70	5.16	5.66	1.85	1.56	1.83	3.33	3.65
1.204	1.84	5.45	10.56	11.61	1.311	1.79	3.78	7.32	8.05	1.480	1.72	2.68	5.12	5.61	1.86	1.56	1.81	3.30	3.62
1.206	1.84	5.40	10.47	11.51	1.314	1.79	3.75	7.26	7.98	1.485	1.72	2.66	5.08	5.57	1.87	1.56	1.80	3.27	3.59
1.208	1.84	5.35	10.38	11.41	1.317	1.79	3.72	7.20	7.92	1.490	1.72	2.64	5.04	5.53	1.88	1.55	1.79	3.24	3.56
1.210	1.84	5.31	10.30	11.32	1.320	1.79	3.69	7.14	7.85	1.495	1.71	2.62	5.00	5.49	1.89	1.55	1.78	3.22	3.54
1.212	1.83	5.27	10.21	11.22	1.323	1.79	3.67	7.09	7.79	1.500	1.71	2.60	4.96	5.45	1.90	1.54	1.77	3.19	3.51
1.214	1.83	5.22	10.12	11.12	1.326	1.79	3.64	7.03	7.73	1.505	1.71	2.58	4.92	5.41	1.91	1.54	1.75	3.17	3.48
1.216	1.83	5.18	10.04	11.03	1.329	1.78	3.61	6.98	7.67	1.510	1.71	2.56	4.88	5.37	1.92	1.54	1.74	3.14	3.45
1.218	1.83	5.14	9.96	10.94	1.332	1.78	3.58	6.92	7.61	1.515	1.71	2.54	4.84	5.33	1.93	1.53	1.73	3.12	3.43
1.220	1.83	5.10	9.89	10.87	1.335	1.78	3.56	6.87	7.55	1.520	1.70	2.53	4.80	5.29	1.94	1.53	1.72	3.09	3.40
1.222	1.83	5.05	9.80	10.77	1.338	1.78	3.53	6.82	7.50	1.525	1.70	2.51	4.77	5.25	1.95	1.53	1.71	3.07	3.38
1.224	1.83	5.01	9.72	10.68	1.341	1.78	3.51	6.77	7.44	1.530	1.70	2.49	4.74	5.21	1.96	1.52	1.70	3.05	3.35
1.226	1.83	4.98	9.65	10.60	1.344	1.78	3.48	6.72	7.39	1.535	1.70	2.47	4.70	5.17	1.97	1.52	1.69	3.03	3.33
1.228	1.83	4.94	9.57	10.52	1.347	1.78	3.46	6.68	7.33	1.540	1.69	2.46	4.66	5.13	1.98	1.51	1.68	3.01	3.30
1.230	1.83	4.90	9.50	10.44	1.350	1.78	3.43	6.63	7.28	1.545	1.69	2.44	4.63	5.09	1.99	1.51	1.68	2.98	3.28
1.232	1.83	4.86	9.43	10.36	1.354	1.77	3.40	6.57	7.21	1.55	1.69	2.43	4.60	5.05	2.00	1.51	1.67	2.96	3.26
1.234	1.83	4.83	9.36	10.28	1.358	1.77	3.37	6.50	7.14	1.56	1.69	2.40	4.54	4.99	2.01	1.50	1.66	2.94	3.23

(*Continued*)

Table 3-6
Table of Coefficients—cont'd

K	T	Z	Y	U	K	T	Z	Y	U	K	T	Z	Y	U	K	T	Z	Y	U
1.236	1.82	4.79	9.29	10.20	1.362	1.77	3.34	6.44	7.08	1.57	1.68	2.37	4.48	4.92	2.02	1.50	1.65	2.92	3.21
1.238	1.82	4.76	9.22	10.13	1.366	1.77	3.31	6.38	7.01	1.58	1.68	2.34	4.42	4.86	2.04	1.49	1.63	2.88	3.17
1.240	1.82	4.72	9.15	10.05	1.370	1.77	3.28	6.32	6.95	1.59	1.67	2.31	4.36	4.79	2.06	1.48	1.62	2.85	3.13
1.242	1.82	4.69	9.08	9.98	1.374	1.77	3.25	6.27	6.89	1.60	1.67	2.28	4.31	4.73	2.08	1.48	1.60	2.81	3.09
1.244	1.82	4.65	9.02	9.91	1.378	1.76	3.22	6.21	6.82	1.61	1.66	2.26	4.25	4.67	2.10	1.47	1.59	2.78	3.05
1.246	1.82	4.62	8.95	9.84	1.382	1.76	3.20	6.16	6.77	1.62	1.65	2.23	4.20	4.61	2.12	1.46	1.57	2.74	3.01
1.248	1.82	4.59	8.89	9.77	1.386	1.76	3.17	6.11	6.72	1.63	1.65	2.21	4.15	4.56	2.14	1.46	1.56	2.71	2.97
1.250	1.82	4.56	8.83	9.70	1.390	1.76	3.15	6.06	6.66	1.64	1.65	2.18	4.10	4.50	2.16	1.45	1.55	2.67	2.94
1.252	1.82	4.52	8.77	9.64	1.394	1.76	3.12	6.01	6.60	1.65	1.65	2.16	4.05	4.45	2.18	1.44	1.53	2.64	2.90
1.254	1.82	4.49	8.71	9.57	1.398	1.75	3.10	5.96	6.55	1.66	1.64	2.14	4.01	4.40	2.20	1.44	1.52	2.61	2.87
1.256	1.82	4.46	8.65	9.51	1.402	1.75	3.07	5.92	6.49	1.67	1.64	2.12	3.96	4.35	2.22	1.43	1.51	2.58	2.84
1.258	1.81	4.43	8.59	9.44	1.406	1.75	3.05	5.87	6.44	1.68	1.63	2.10	3.92	4.30	2.24	1.42	1.50	2.56	2.81
1.260	1.81	4.40	8.53	9.38	1.410	1.75	3.02	5.82	6.39	1.69	1.63	2.08	3.87	4.26	2.26	1.41	1.49	2.53	2.78
1.263	1.81	4.36	8.45	9.28	1.414	1.75	3.00	5.77	6.34	1.70	1.63	2.06	3.83	4.21	2.28	1.41	1.48	2.50	2.75
1.266	1.81	4.32	8.37	9.19	1.418	1.75	2.98	5.72	6.29	1.71	1.62	2.04	3.79	4.17	2.30	1.40	1.47	2.48	2.72
1.269	1.81	4.28	8.29	9.11	1.422	1.75	2.96	5.68	6.25	1.72	1.62	2.02	3.75	4.12	2.32	1.40	1.46	2.45	2.69
1.272	1.81	4.24	8.21	9.02	1.426	1.74	2.94	5.64	6.20	1.73	1.61	2.00	3.72	4.08	2.34	1.39	1.45	2.43	2.67
1.275	1.81	4.20	8.13	8.93	1.430	1.74	2.91	5.60	6.15	1.74	1.61	1.99	3.68	4.04	2.36	1.38	1.44	2.40	2.64
2.38	1.38	1.43	2.38	2.61	2.83	1.25	1.28	1.98	2.17	3.46	1.11	1.18	1.64	1.80	4.15	0.989	1.12	1.40	1.54
2.40	1.37	1.42	2.36	2.59	2.86	1.24	1.28	1.96	2.15	3.50	1.10	1.18	1.62	1.78	4.20	0.982	1.12	1.39	1.53
2.42	1.36	1.41	2.33	2.56	2.89	1.23	1.27	1.94	2.13	3.54	1.09	1.17	1.61	1.76	4.25	0.975	1.12	1.38	1.51
2.44	1.36	1.40	2.31	2.54	2.92	1.22	1.27	1.92	2.11	3.58	1.08	1.17	1.59	1.75	4.30	0.968	1.11	1.36	1.50
2.46	1.35	1.40	2.29	2.52						3.62	1.07	1.16	1.57	1.73					
					2.95	1.22	1.26	1.90	2.09						4.35	0.962	1.11	1.35	1.48
2.48	1.35	1.39	2.27	2.50	2.98	1.21	1.25	1.88	2.07	3.66	1.07	1.16	1.56	1.71	4.40	0.955	1.11	1.34	1.47
2.50	1.34	1.38	2.25	2.47	3.02	1.20	1.25	1.86	2.04	3.70	1.06	1.16	1.55	1.70	4.45	0.948	1.11	1.33	1.46
2.53	1.33	1.37	2.22	2.44	3.06	1.19	1.24	1.83	2.01	3.74	1.05	1.15	1.53	1.68	4.50	0.941	1.10	1.31	1.44
2.56	1.32	1.36	2.19	2.41						3.78	1.05	1.15	1.52	1.67	4.55	0.934	1.10	1.30	1.43
					3.10	1.18	1.23	1.81	1.99						4.60	0.928	1.10	1.29	1.42
2.59	1.31	1.35	2.17	2.38	3.14	1.17	1.23	1.79	1.97	3.82	1.04	1.15	1.50	1.65	4.65	0.921	1.10	1.28	1.41
2.62	1.30	1.34	2.14	2.35	3.18	1.16	1.22	1.77	1.94	3.86	1.03	1.14	1.49	1.64	4.70	0.914	1.09	1.27	1.39
2.65	1.30	1.33	2.12	2.32	3.22	1.16	1.21	1.75	1.92	3.90	1.03	1.14	1.48	1.62	4.75	0.908	1.09	1.26	1.38
2.68	1.29	1.32	2.09	2.30	3.26	1.15	1.21	1.73	1.90	3.94	1.02	1.14	1.46	1.61	4.80	0.900	1.09	1.25	1.37
2.71	1.28	1.31	2.07	2.27	3.30	1.14	1.20	1.71	1.88	3.98	1.01	1.13	1.45	1.60	4.85	0.893	1.09	1.24	1.36
2.74	1.27	1.31	2.04	2.25	3.34	1.13	1.20	1.69	1.86	4.00	1.009	1.13	1.45	1.59	4.90	0.887	1.09	1.23	1.35
2.77	1.26	1.30	2.02	2.22	3.38	1.12	1.19	1.67	1.84	4.05	1.002	1.13	1.43	1.57	4.95	0.880	1.08	1.22	1.34
2.80	1.26	1.29	2.00	2.20	3.42	1.11	1.19	1.66	1.82	4.10	0.996	1.13	1.42	1.56	5.00	0.873	1.08	1.21	1.33

Reprinted by permission of Taylor Forge International, Inc.

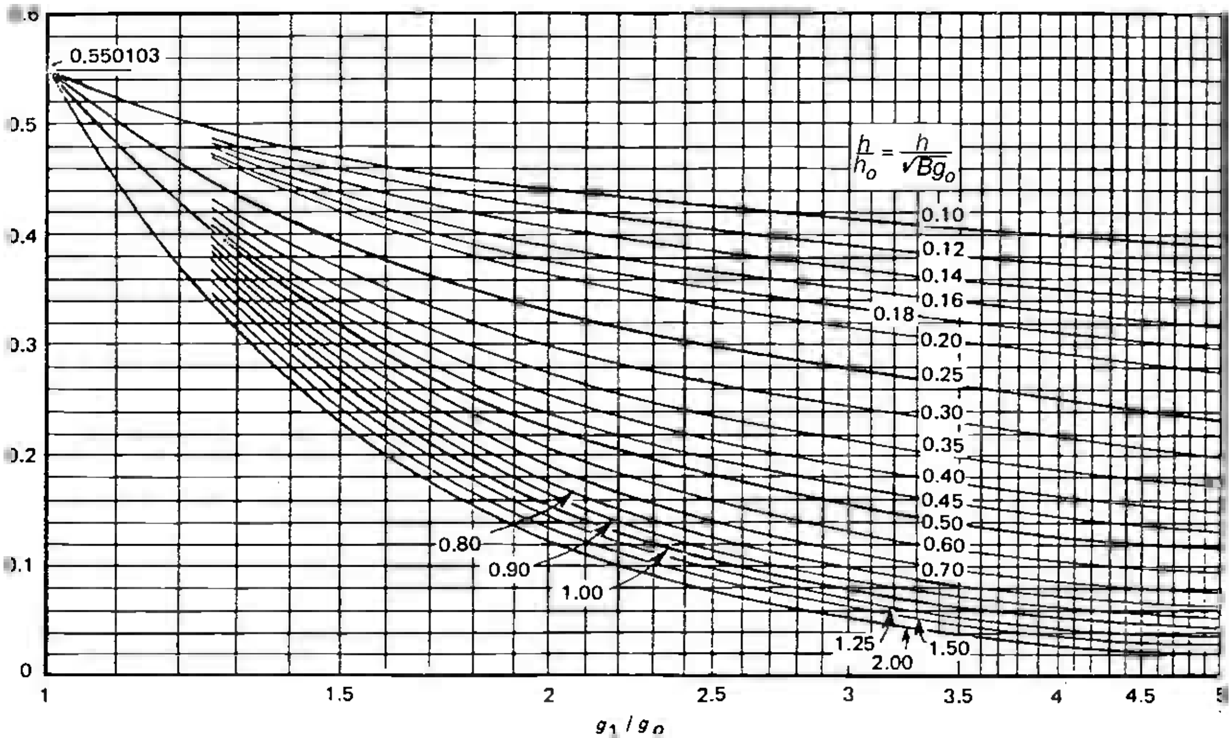


Figure 3-7. Values of V (integral flange factors). *(Reprinted by permission from the ASME Code, Section VIII, Div. 1, Figure 2-7.3.)*

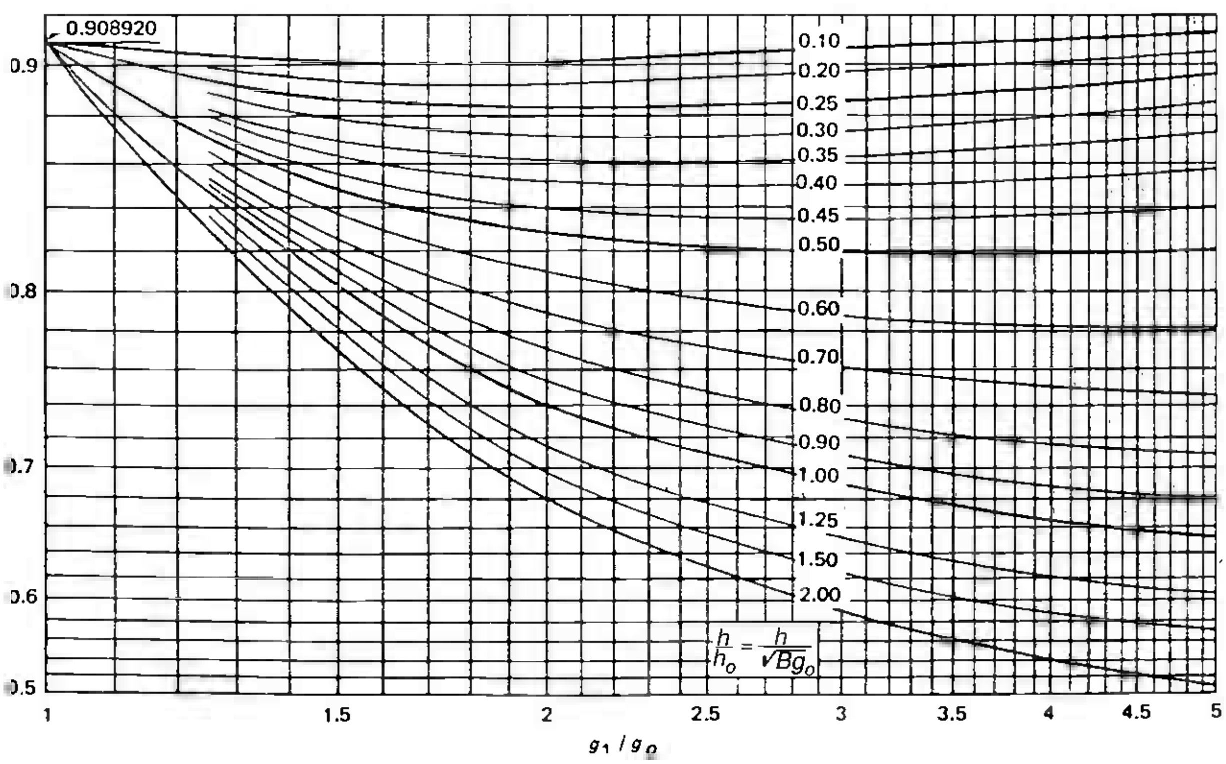


Figure 3-8. Values of F (integral flange factors). *(Reprinted by permission from the ASME Code, Section VIII, Div. 1, Figure 2-7.2.)*

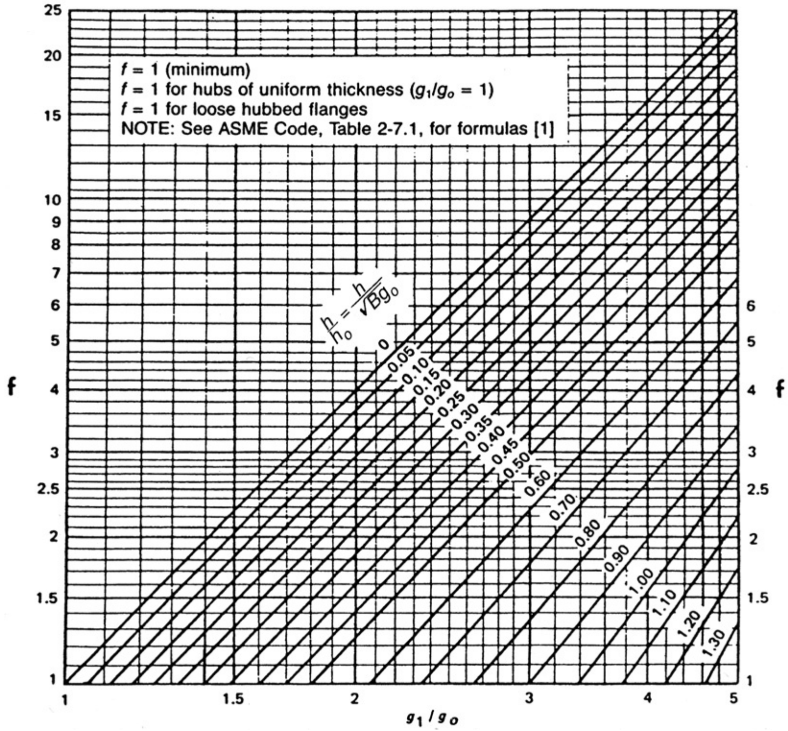


Figure 3-9. Values of f (hub stress correction factor). *(Reprinted by permission from the ASME Code, Section VIII, Div. 1, Figure 2-7.6.)*

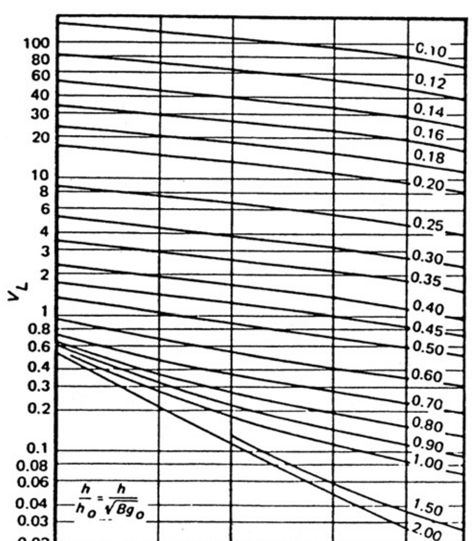


Figure 3-10. Values of V_L (loose hub flange factors). *(Reprinted by permission from the ASME Code, Section VIII, Div. 1, Figure 2-7.5.)*

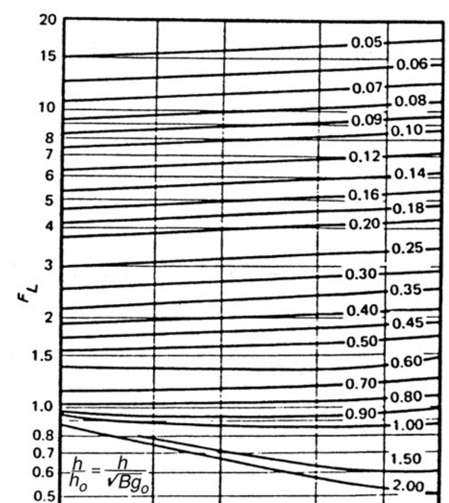


Figure 3-11. Values of F_L (loose hub flange factors). *(Reprinted by permission from the ASME Code, Section VIII, Div. 1, Figure 2-7.4.)*

Table 3-7
Number and Size of Bolts for Flanged Joints

Primary Service Pressure Rating	Bolting	Flange Facing	Nominal Pipe Size ½	¾	1	1¼	1½	2	2½	3	3½	4	5	6	8	10	12	14	16	18	20	24
	Number		4	4	4	4	4	4	4	4	8	8	8	8	8	12	12	12	16	16	20	20
	Diameter		½	½	½	½	½	⅝	⅝	⅝	⅝	⅝	¾	¾	¾	⅞	⅞	1	1	1⅛	1⅛	1¼
150 Pound	Length of Stud Bolts	1/16″ RF	2¼	2¼	2½	2½	2¾	3	3¼	3½	3½	3½	3¾	3¾	4	4½	4½	5	5¼	5¾	6	6¾
		RTJ			3	3	3¼	3½	3¾	4	4	4	4¼	4¼	4½	5	5	5½	5¾	6¼	6½	7¼
	Length of Mach, Bolts	1/16″ RF	1¾	2	2	2¼	2¼	2¾	3	3	3	3	3¼	3¼	3½	3¾	4	4¼	4½	4¾	5¼	5¾
	Number		4	4	4	4	4	8	8	8	8	8	8	12	12	16	16	20	20	24	24	24
	Diameter		½	⅝	⅝	⅝	¾	⅝	¾	¾	¾	¾	¾	¾	⅞	1	1⅛	1⅛	1¼	1¼	1¼	1½
300 Pound	Length of Stud Bolts	1/16″ RF	2½	2¾	3	3	3½	3¼	3¾	4	4¼	4¼	4½	4¾	5¼	6	6½	6¾	7¼	7½	8	9
		RTJ	3	3¼	3½	3½	4	4	4½	4¾	5	5	5¼	5½	6	6¾	7¼	7½	8	8¼	8¾	10
	Length of Mach, Bolts	1/16″ RF	2	2½	2½	2¾	3	3	3¼	3½	3¾	3¾	4	4¼	4¾	5¼	5¾	6	6½	6¾	7	7¾
	Number		4	4	4	4	4	8	8	8	8	8	8	12	12	16	16	20	20	24	24	24
	Diameter		½	⅝	⅝	⅝	¾	⅝	¾	¾	⅞	⅞	⅞	⅞	1	1⅛	1¼	1¼	1⅜	1⅜	1½	1¾
400 Pound	Length of Stud Bolts	¼″ RF	3	3¼	3½	3¾	4	4	4½	4¾	5¼	5¼	5½	5¾	6½	7¼	7¾	8	8½	8¾	9½	10½
		RTJ	3	3¼	3½	3¾	4	4¼	4¾	5	5½	5½	5¾	6	6¾	7½	8	8¼	8¾	9	9¾	11
		M&F, T&G	2¾	3	3¼	3½	3¾	3¾	4¼	4½	5	5	5¼	5½	6¼	7	7½	7¾	8¼	8½	9¼	10¼
	Number		4	4	4	4	4	8	8	8	8	8	8	12	12	16	20	20	20	20	24	24
	Diameter		½	⅝	⅝	⅝	¾	⅝	¾	¾	⅞	⅞	1	1	1⅛	1¼	1¼	1⅜	1½	1⅝	1⅝	1⅞
600 Pound	Length of Stud Bolts	¼″ RF	3	3¼	3½	3¾	4	4	4½	4¾	5¼	5½	6¼	6½	7½	8¼	8½	9	9¾	10½	11¼	12¾
		RTJ	3	3¼	3½	3¾	4	4¼	4¾	5	5½	5¾	6½	6¾	7¾	8½	8¾	9¼	10	10¾	11½	13¼
		M&F, T&G	2¾	3	3¼	3½	3¾	3¾	4¼	4½	5	5¼	6	6¼	7¼	8	8¼	8¾	9½	10¼	11	12½
	Number		4	4	4	4	4	8	8	8		8	8	12	12	16	20	20	20	20	20	20
	Diameter		¾	¾	⅞	⅞	1	⅞	1	⅞		1⅛	1¼	1⅛	1⅜	1⅜	1⅜	1½	1⅝	1⅞	2	2½
900 Pound	Length of Stud Bolts	¼″ RF	4	4¼	4¾	4¾	5¼	5½	6	5½		6½	7¼	7½	8½	9	9¾	10½	11	12¾	13½	17
		RTJ	4	4¼	4¾	4¾	5¼	5¾	6¼	5¾		6¾	7½	7½	8¾	9¼	10	11	11½	13¼	14	17¾
		M&F, T&G	3¾	4	4½	4½	5	5¼	5¾	5¼		6¼	7	7¼	8¼	8¾	9½	10¼	10¾	12¼	13¼	16¾
	Number		4	4	4	4	4	8	8	8		8	8	12	12	12	16	16	16	16	16	16
	Diameter		¾	¾	⅞	⅞	1	⅞	1	1⅛		1¼	1½	1⅜	1⅝	1⅞	2	2¼	2½	2¾	3	3½
1500 Pound	Length of Stud Bolts	¼″ RF	4	4¼	4¾	4¾	5¼	5½	6	6¾		7½	9½	10	11¼	13¼	14¾	16	17½	19¼	21	24
		RTJ	4	4¼	4¾	4¾	5¼	5¾	6¼	7		7¾	9¾	10¼	11¾	13½	15¼	16¾	18½	20¼	22¼	25½
		M&F, T&G	3¾	4	4½	4½	5	5¼	5¾	6½		7¼	9¼	9¾	11	13	14½	15¾	17¼	19	20¾	23¾
	Number		4	4	4	4	4	8	8	8		8	8	8	12	12	12					
	Diameter		¾	¾	⅞	1	1⅛	1	1⅛	1¼		1½	1¾	2	2	2½	2¾					
2500 Pound	Length of Stud Bolts	¼″ RF	4¾	4¾	5¼	5¾	6½	6¾	7½	8½		9¾	11½	13½	15	19	21					
		RTJ	4¾	4¾	5¼	6	6¾	7	7¾	8¾		10¼	12¼	14	15½	20	22					
		M&F, T&G	4½	4½	5	5½	6¼	6½	7¼	8¼		9½	11¼	13¼	14¾	18¾	20¾					

DERIVATION OF FLANGE MAXIMUM ALLOWABLE PRESSURE

		@ Design Temperature	@ Ambient Temperature
INTEGRAL TYPE	1. Calculate Moments M_1 through M_5 as follows:	@ Design Temperature	@ Ambient Temperature
	$M_1 = (\text{Lesser of } 1.5\,S_{fo} \text{ or } 2.5\,S_{fa})\dfrac{\lambda g_1^2 B}{f}$		
	$M_2 = \dfrac{S_{fo}\lambda Bt^2}{1.33te + 1}$		
	$M_3 = \dfrac{S_{fo}\lambda Bt^2}{Y\lambda - Z(1.33te + 1)}$		
	$M_4 = \dfrac{2S_{fo}\lambda Bt^2 g_1^2}{ft^2 + (1.33te + 1)g_1^2}$		
	$M_5 = \dfrac{2S_{fo}\lambda Bt^2 g_1^2}{ft^2 + Y\lambda g_1^2 - Z(1.33te + 1)g_1^2}$		
	M_{MAX} = Lesser of M_1 thru M_5		
LOOSE TYPE	1. Calculate the Maximum Allowable Moment	@ Design Temperature	@ Ambient Temperature
	$M_{MAX} = \dfrac{S_{fo}t^2 B}{Y}$		

2.

$$A_{m(MAX)} = \frac{2M_{MAX}(\text{@ Ambient Temperature})}{h_G S_a} - A_b$$

Note : If $A_{m2} > A_{m(MAX)}$, then the gasket width, seating stress, or bolting is insufficient.

3. Determine the Maximum Allowable Pressure set by the Maximum Allowable Moment: (*Operating Condition*)

$$\frac{M_{MAX}(\text{@ Design Temperature})}{0.785B^2 h_D + 6.28bGmh_G + 0.785(G^2 - B^2)h_T}$$

4. Determine the Maximum Allowable Pressure set by $A_{m(MAX)}$: (*Gasket Seating*)

$$\frac{S_b A_{m(MAX)}}{6.28bGm + 0.785G^2}$$

5. The Maximum Allowable Pressure = the lesser of 3. or 4.

Note that this pressure includes any static head applicable for the case under consideration.

Maximum Allowable Pressure = ______________________________

MAWP is based on corroded condition at design temperature.
When M_{MAX} is governed by M_2. Check integral type flange for new & uncorroded condition.
MAP (cold & corroded) is based on corroded condition @ ambient temperature.
MAP (new & cold) is based on new condition @ ambient temperature.

Procedure 3-2: Design of Spherically Dished Covers

1		DESIGN CONDITIONS			
Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Amb. temp., S_{fa}		Amb. temp., S_a	

2	GASKET AND FACING DETAILS		
Gasket		Facing	

3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = 0.5(A_m + A_b)S_a$	
m					

5	GASKET WIDTH CHECK
$N_{min} = A_bS_a/2y\pi G$	

6		MOMENT CALCULATIONS			
Load	×	Lever Arm		=	Moment
Operating					
$H_D = \pi B^2P/4$		$h_D = 0.5(C - B)$		$M_D = H_Dh_D$	
$H_G = H_P$		$h_G = 0.5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = 0.5(h_D + h_G)$		$M_T = H_Th_T$	
$H_r = H_D/\tan \beta_1$		h_r		$M_r = H_rh_r$	
β Calculation					
$\beta_1 = \arc\sin \dfrac{B}{2L + t}$		$M_o = M_D + M_G + M_T \pm M_r$ *Note:* M_r is (+) if ℄ of head is below the center of gravity; (−) if above.			
Seating					
$H_G = W$		h_G		$M_o' = Wh_G$	

7 FLANGE AND HEAD THICKNESS CALCULATION	
Head Thickness Required	
$t = \dfrac{5PL}{6S}$	
Flange Thickness Required	
$F = \dfrac{PB\sqrt{4L^2 - B^2}}{8S_{fo}(A - B)}$	
$J = \dfrac{M}{S_{fo}B}\left(\dfrac{A + B}{A - B}\right)$ where $M = M_o$ or M_o', whichever is greater	
$T = F + \sqrt{F^2 + J}$	

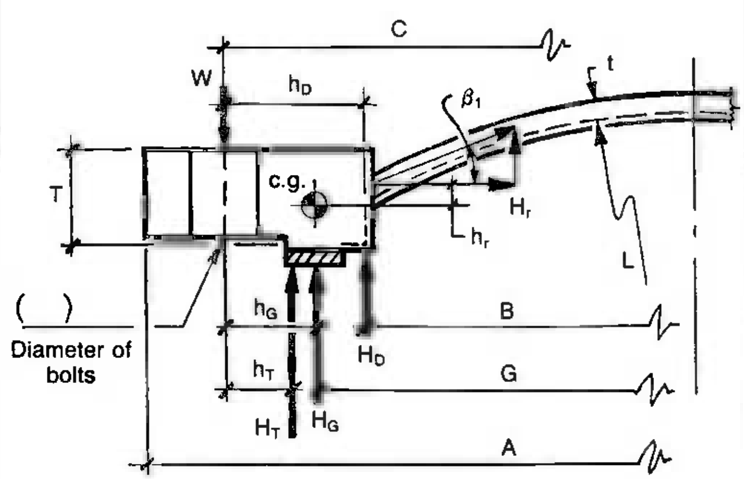


Figure 3-12. Dimensional data and forces for a spherically dished cover.

SPHERICALLY DISHED COVER

1	DESIGN CONDITIONS				
		Allowable Stresses (PSI)			
		Flange		Bolting	
Design pressure, P	150 PSIG				
Design temperature	400 deg F				
Flange material	SA-285-C	Design temp., S_{fo}	13,750	Design temp., S_b	25,000
Bolting material	SA-193-B7	Amb.temp., S_{fa}	13,750	Amb.temp., S_a	25,000

2	GASKET AND FACING DETAILS		
Gasket	.125 Thk DJAF	Facing	RF

3		4	LOAD AND BOLT CALCULATIONS		
N	1.00	$W_{m2} = b\pi Gy$	265,939	A_m = greater of W_{m2}/S_a or W_{m1}/S_b	15.50
b	.353	$H_P = 2b\pi GmP$	53,187		
G	53.29	$H = G^2\pi P/4$	334,558	A_b	20.38
y	4600 PSI	$W_{m1} = H_P + H$	387,745	$W = 0.5(A_m + A_b)S_a$	448,500
m	3.00				

5	GASKET WIDTH CHECK
$N_{min} = A_b S_a / 2y\pi G$	.33

6	MOMENT CALCULATIONS				
Load		x Lever Arm		= Moment	
Operating					
$H_D = \pi B^2 P/4$	321,628	$h_D = 0.5(C-B)$	2.50	$M_D = H_D h_D$	804,070
$H_G = H_P$	53,187	$h_G = 0.5(C-G)$	1.98	$M_G = H_G h_G$	105,310
$H_T = H - H_D$	12,930	$h_T = 0.5(h_D + h_G)$	2.24	$M_T = H_T h_T$	28,963
$H_r = H_D/\tan\beta_1$	547,325	h_r	1.375	$M_r = H_r h_r$	(−)752,572
β Calculation					
$\beta_1 = \arc\sin \dfrac{B}{2L+t}$	30.44 deg	$M_o = M_D + M_G + M_T \pm M_r$ *Note:* M_r is (+) if ε of head is below the center of gravity; (–) if above.		185,771	
Seating					
$H_G = W$	448,500	h_G	1.98	$M_o' = W h_G$	888,030

7 FLANGE AND HEAD THICKNESS CALCULATION	
Head Thickness Required	
$t = \dfrac{5PL}{6S}$	464+.125=.589 Use .625
Flange Thickness Required	
$F = \dfrac{PB\sqrt{4L^2-B^2}}{8S_{fo}(A-B)}$	.64
$J = \dfrac{M}{S_{fo}B}\left(\dfrac{A+B}{A-B}\right)$ where $M = M_o$ or M_o', whichever is greater	14.475
$T = F + \sqrt{F^2+J}$	4.498 Use 4.50

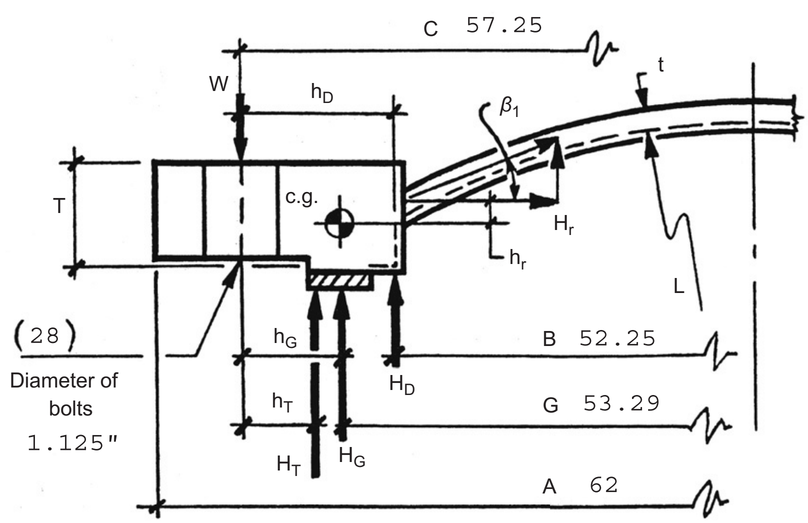


Figure 3-13. Dimensional and forces for a spherically dished cover.

Procedure 3-3: Design of Blind Flanges with Openings [1,4]

1	DESIGN CONDITIONS				
Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2	GASKET AND FACING DETAILS		
Gasket		Facing	

3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G (see below)		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = 0.5(A_m + A_b)S_a$	
m				$h_G = 0.5(C - G)$	

5	THICKNESS AND REINFORCEMENT CALCULATIONS
Dimension, G	
If $b_o \le 0.25$ in., G = mean gasket diameter	
If $b_o > 0.25$ in., G = lesser of raised face diameter or gasket O.D. − 2b	

Thickness Required			
Operating, t_o [1, UG-34(c)(2)] (See Note 1)		Seating, t_G	
$t_o = G\sqrt{\dfrac{0.3P}{S_{fo}} + \dfrac{1.9W_{m1}h_G}{S_{fo}G^3}}$		$t_G = G\sqrt{\dfrac{1.9Wh_G}{S_{fa}G^3}}$	

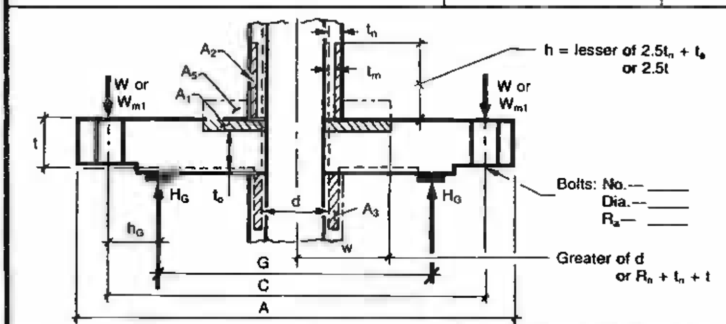


Figure 3-14. Dimensional data and forces for a blind flange.

Reinforcement			
$t_m = \dfrac{PR_n}{SE - 0.6P}$		$A_3 = 2t_n h$	
		A_4 = area of welds	
$A_r = 0.5dt_o$		$A_5 = t_e$(O.D. pad − O.D. nozzle)	
$A_1 = (t - t_o)(2w - d)$		$\Sigma A = A_1$ through A_5	
$A_2 = 2h(t_n - t_m)$		$\Sigma A > A_r$	

Notes

1. Reinforcement is only required for operating conditions not bolt up.
2. Options in lieu of calculating reinforcement:
   Option 1—No additional reinforcement is required if flange thickness is greater than 1.414 t_o.
   Option 2—If opening exceeds one-half the nominal flange diameter, the flange may be computed as an optional-type reducing flange.
   Option 3—No additional reinforcement is required if t_o is calculated substituting 0.6 for 0.3 in the equation for t_o (doubling of c value).

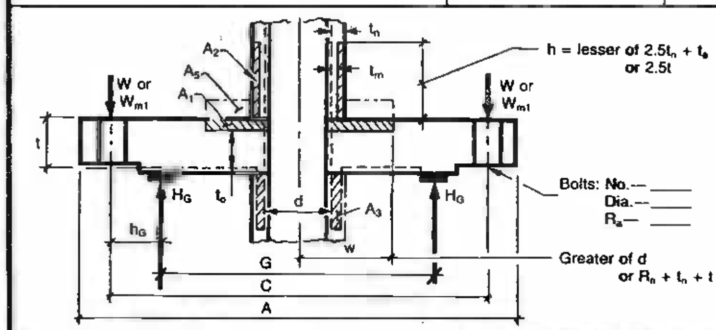
BLIND FLANGE WITH OPENING

1	DESIGN CONDITIONS				
Design pressure, P	293 PSIG	Allowable Stresses			
Design temperature	500 deg F	Flange		Bolting	
Flange material	SA-105	Design temp., S_{fo}	17,500	Design temp., S_b	25,000
Bolting material	SA-193-B7	Atm. temp., S_{fa}	17,500	Atm. temp., S_a	25,000
Corrosion allowance	.118				

2	GASKET AND FACING DETAILS		
Gasket		Facing	

3		4	LOAD AND BOLT CALCULATIONS		
N	1.315	$W_{m2} = b\pi Gy$	185,670	A_m = greater of W_{m2}/S_a or W_{m1}/S_b	17.55
b	.405	$H_P = 2b\pi GmP$	80,867		
G (see below)	39.44	$H = G^2\pi P/4$	357,957	A_b	84.26
y	3700	$W_{m1} = H_P + H$	438,824	$W = 0.5(A_m + A_b)S_a$	1,280,162
m	2.75			$h_G = 0.5(C - G)$	3.28

5	THICKNESS AND REINFORCEMENT CALCULATIONS		
Dimension, G			
If $b_o \leq 0.25$ in., G = mean gasket diameter If $b_o > 0.25$ in., G = lesser of raised face diameter or gasket O.D. − 2b		40.25-2(.405)=39.44	
Thickness Required			
Operating, t_o [1, UG-34(c)(2)] (See Note 1)		Seating, t_G	
$t_o = G\sqrt{\dfrac{0.3P}{S_{fo}} + \dfrac{1.9W_{m1}h_G}{S_{fo}G^3}}$	3.426	$t_G = G\sqrt{\dfrac{1.9Wh_G}{S_{fa}G^3}}$	3.40



A = 50 t = 4.12
C = 46
G = 39.44

Figure 3-15. Dimensional data and forces for a blind flange.

(32)2"dia studs
Ra = 2.625

Reinforcement			
$t_m = \dfrac{PR_n}{SE - 0.6P}$	.083	$A_3 = 2t_n h$	0
		A_4 = area of welds	0
$A_r = 0.5dt_o$	12.71	$A_5 = t_e$(O.D. pad − O.D. nozzle)	5.99
$A_1 = (t - t_o)(2w - d)$	5.31	$\Sigma A = A_1$ through A_5	13.01
$A_2 = 2h(t_n - t_m)$	1.71	$\Sigma A > A_r$	OK

Alternate Reinforcement check
1. t > 1.414 X 3.426 = 4.84 No Good!
2. Calculate t using C = .6 = 4.42" No Good!
Area of pad required = Ar-A1-A2 = 5.62 sq in
Use 16" OD X .813" Thk
A5 = (16-8.625) .813 = 5.99 Sq in

Procedure 3-4: Bolt Torque Required for Sealing Flanges [5–7]

Notation

A_b = cross-sectional area of bolts, $in.^2$
A_g = actual joint-contact area of gasket, $in.^2$
b = effective gasket seating width, in.
d = root diameter of threads, in.
d_m = pitch diameter of threads, in.
G = diameter at location of gasket load reaction, in.
M = external bending moment, in.-lb
m = gasket factor
N = gasket width, in.
n = number of bolts
E_b = modulus of elasticity of bolting material at temperature, psi
E_g = modulus of elasticity of gasket material at temperature, psi
P = internal pressure, psi
P_e = equivalent pressure including external loads, psi
P_r = radial load, lb
P_T = test pressure, psi
F = restoring force of gasket (decreasing compression force) from initial bolting strain, lb
F_{bo} = initial tightening force, lb
L_b = effective length of bolt, mid nut to mid nut, in.
W = total tightening force, lb
W_{ml} = $H + H_p$ = required bolt load, operating, lb
W_{m2} = required bolt load, gasket seating, lb
y = gasket unit seating load, psi
H = total hydrostatic end force, lb
H_P = total joint-contact surface compression load, lb
T = initial tightening torque required, ft-lb
t_g = thickness of gasket, in.
t_n = thickness of nut, in.
K = total friction factor between bolt/nut and nut/flange face
w = width of ring joint gasket, in.

Note: See Procedure 3-1 for values of G, N, m, b, and y.

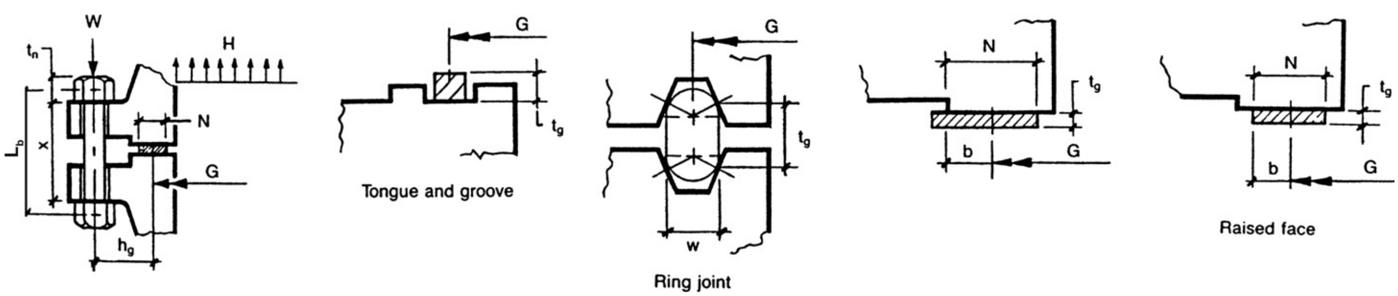


Figure 3-16. Flange and joint details.

Table 3-8
Bolting Dimensional Data

Size	¾ in.	⅞ in.	1 in.	1⅛ in.	1¼ in.	1⅜ in.	1½ in.	1⅝ in.
d	0.6273	0.7387	0.8466	0.9716	1.0966	1.2216	1.3466	1.4716
d_m	0.6850	0.8028	0.9188	1.0438	1.1688	1.2938	1.4188	1.5438
t_n	0.7344	0.8594	0.9844	1.1094	1.2188	1.3438	1.4688	1.5938
Size	**1¾ in.**	**1⅞ in.**	**2 in.**	**2¼ in.**	**2½ in.**	**2¾ in.**	**3 in.**	**3¼ in.**
d	1.5966	1.7216	1.8466	2.0966	2.3466	2.5966	2.8466	3.0966
d_m	1.6688	1.7938	1.9188	2.1688	2.4188	2.6688	2.9188	3.1688
t_n	1.7188	1.8438	1.9688	2.2031	2.4531	2.7031	2.9531	3.1875

Note: 3/4 and 7/8 in. bolts are UNC series threads. All others are 8 series threads. All dimensions are from ANSI B18.2.

Table 3-9
Modulus of Elasticity, E_b, 10^6 psi

Material	Temperature, °F								
	70°	200°	300°	400°	500°	600°	700°	800°	900°
Carbon steel A-307-B	27.9	27.7	27.4	27.0	26.4	25.7	24.8	23.4	18.5
Low alloy A-193-B7, B16, B7M	29.9	29.5	29.0	28.6	28.0	27.4	26.6	25.7	24.5
Straight chrome A-193-B6, B6X	29.2	28.7	28.3	27.7	27.0	26.0	24.8	23.1	22.1
Stainless A-193-B8 series	28.3	27.7	27.1	26.6	26.1	25.4	24.8	24.1	23.4

Note: Values per ASME Code, Section II.

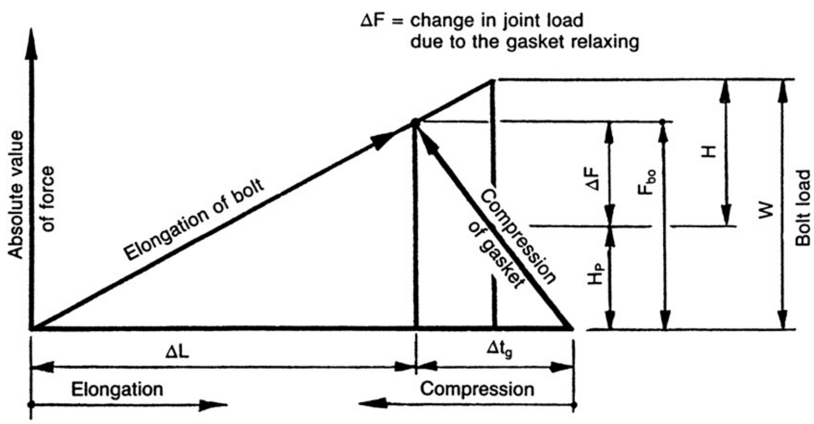


Figure 3-17. Typical joint diagram.

Design Data		Gasket Data		Bolting Data	
Flange size		Type		Nominal size	
Design pressure, P		Diameter of raised face		Quantity n	
Test pressure, P_T		O.D., I.D.		d	
		N or w		d_m	
Moment, M		y		E_b	
Radial load, Pr		m		$Ab = \frac{\pi d^2 n}{4}$	
		Eg			
Friction factor, K		tg		$\ell_b = x + t_n$	
Design temperature		b			
		G			

Modulus of Elasticity of Gasket Material, E_g

- Ring joint and flat metal: Select values from ASME Section II, or Appendix K of this book.
- Comp asb = 70 ksi
- Rubber = 10 ksi
- Grafoil = 35 ksi
- Teflon = 24 ksi
- Spiral wound = 569 ksi

Friction Factor, K

- Lubricated = 0.075–0.15
- Nonlubricated = 0.15–0.25

Calculations

- *Equivalent pressure, P_e, psi.*

$$P_e = \frac{16M}{\pi G^3} + \frac{4P_r}{\pi G^2} + P$$

- *Hydrostatic end force, H, lb.*

$$H = \frac{\pi G^2 P_e}{4}$$

- *Total joint-contact-surface compression load, H_p, lb.*

$$H_P = 2b\pi G m P_e$$

- *Minimum required bolt load for gasket seating, W_{m2}, lb.*

$$W_{m2} = \pi b G y$$

- *Actual joint area contact for gasket, A_g, in.^{2}*

$$A_g = 2\pi b G$$

- *Decreasing compression force in gasket, $\triangle F$, lb.*

$$\Delta F = \frac{H}{1 + \frac{A_b E_b t_g}{A_g E_g L_b}}$$

- *Initial required tightening force (tension), F_{bo}, lb.*

$$F_{bo} = H_p + \Delta F$$

- *Total tightening force required to seal joint, W, lb.*

$$W = \text{greater of } F_{bo} \text{ or } W_{m2}$$

- *Required torque, T, ft-lb.*

$$T = \frac{KWd_m}{12n}$$

Notes

1. Bolted joints in high-pressure systems require an initial preload to prevent the joint from leaking. The loads which tend to open the joint are:
   a. Internal pressure.
   b. Thermal bending moment.
   c. Dead load bending moment.
2. Either stud tensioners or torque wrenches are used for prestressing bolts to the required stress for gasket seating. Stud tensioners are by far the most accurate. Stud tension achieved by torquing the nut is affected by many variables and may vary from 10% to 100% of calculated values. The following are the major variables affecting tension achieved by torquing:
   a. Class of fit of nut and stud.
   b. Burrs.
   c. Lubrication.
   d. Grit, chips, and dirt in threads of bolts or nuts.
   e. Nicks.
   f. The relative condition of the seating surface on the flange against which the nut is rotated.
3. Adequate lubrication should be used. Nonlubricated bolting has an efficiency of about 50% of a well-lubricated bolt. For standard applications, a heavy graphite and oil mixture works well. For high temperature service (500°F to 1000°F), a high temperature thread compound may be used.
4. The stiffness of the bolt is only 1/3 to 1/5 that of the joint. Thus, for an equal change in deformation, the change of the load in the bolt must be only 1/3 to 1/5 of the change in the load of the joint.
5. Joints almost always relax after they have first been tightened. Relaxation of 10% to 20% of the initial preload is not uncommon. Thus an additional preload of quantity F is required to compensate for this "relaxing" of the joint.

Table 3-10
Bolt torques
Torque required in ft-lb to produce the following bolt stress

Bolt Size	15 ksi	30 ksi	45 ksi	60 ksi
½–13	15	30	45	60
⅝–11	30	60	90	120
¾–10	50	100	150	200
⅞–9	80	160	240	320
1–8	123	245	368	490
1⅛–8	195	390	533	710
1¼–8	273	500	750	1000
1⅜–8	365	680	1020	1360
1½–8	437	800	1200	1600
1⅝–8	600	1100	1650	2200
1¾–8	775	1500	2250	3000
1⅞–8	1050	2000	3000	4000
2–8	1125	2200	3300	4400
2¼–8	–	3180	4770	6360
2½–8	–	4400	6600	8800
2¾–8	–	5920	8880	11840
3–8	–	7720	11580	15440

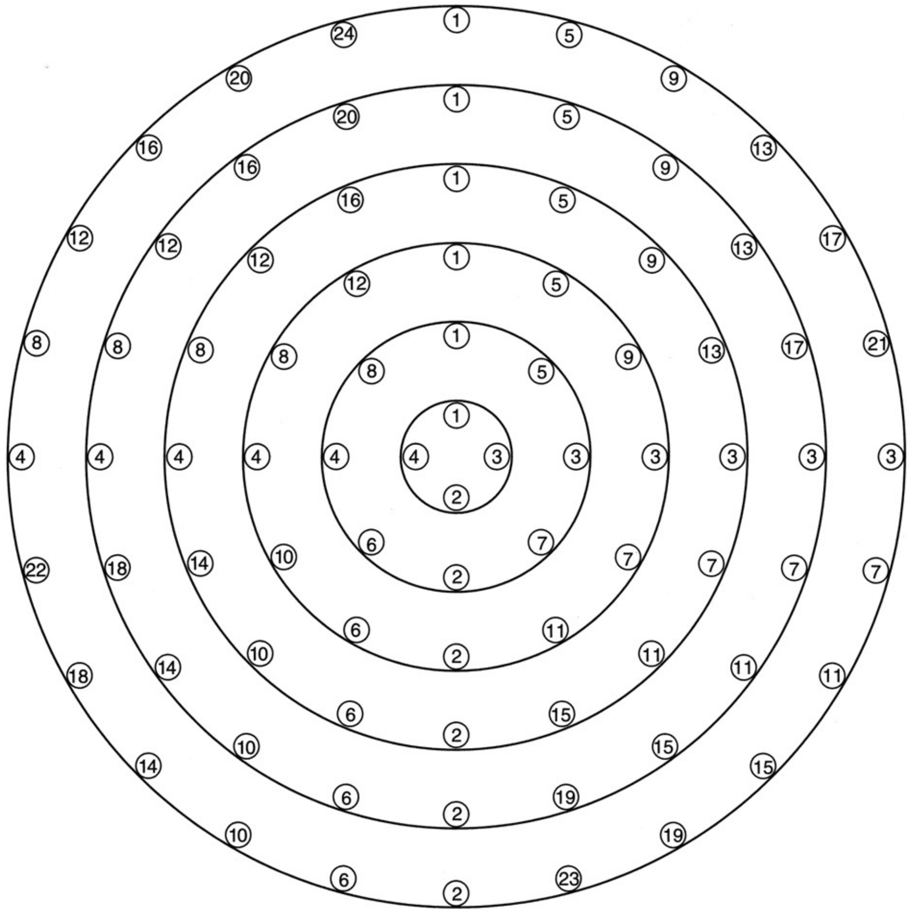


Figure 3-18. Sequence for tightening of flange bolts. *Note:* Bolts should be tightened to 1/3 of the final torque value at a time in the sequence illustrated in the figure. Only on the final pass is the total specified torque realized.

Procedure 3-5: Design of Studding Outlets

The calculations for a studding outlet will vary from standard flange calculations because these flanges are not hubbed or hubless. However many of the loads and moments calculated in a standard flange still apply. The bolt load calculations and gasket calculations are identical.

A studding outlet has other loads that combine with those due to pressure, gasket and bolting loads. The attachment of the shell directly to the studding outlet will cause a moment in the flange ring. This moment acts about the center of gravity of the flange ring and is additive to the other moments.

Another difference is that in a hubbed flange, the hydrostatic end force, H_D would be applied at the centroid of the hub. Since studding outlets are hubless, the load is transferred to the ID of the flange, just as in a spherically dished cover. This load is applied from the centroid of the flange area, and not from the bolt circle.

For studding outlets installed in cylindrical shells, there are two cases to be considered, the circumferential axis case and the longitudinal axis case. Normally, the worst case is the circumferential axis case because the circumferential load from the pressure stress is twice that of the longitudinal case. A check may be made of the longitudinal axis by reducing force T_2 in half and recalculating h_r from the center of shell to centroid of the flange on the longitudinal axis.

For a studding outlet in a sphere or the spherical portion of a head, the loading would be the same in either axis. Therefore the procedure would be identical except that the pressure loading, T_2 would be reduced to .5 PR.

STUDDING OUTLET

1	DESIGN CONDITIONS				
Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Amb. temp., S_{fa}		Amb. temp., S_a	

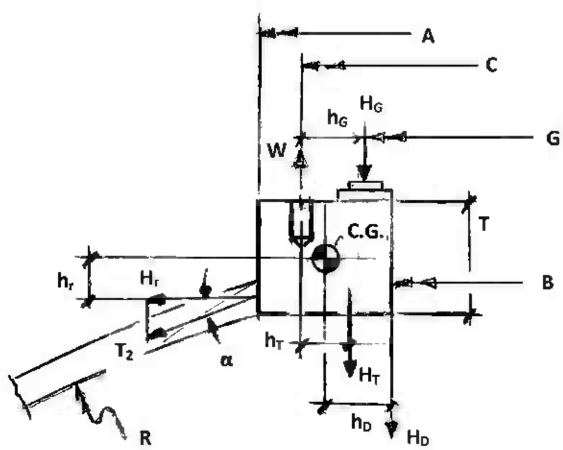
2	GASKET AND FACING DETAILS		
Gasket		Facing	

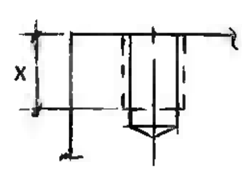
3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = .5(A_m + A_b)S_a$	
m					

5	GASKET WIDTH CHECK
$N_{min} = A_bS_a/2y\pi G$	

6	MOMENT CALCULATIONS				
Load	×	Lever Arm		=	Moment
		Operating			
$H_D = \pi B^2P/4$		$h_D = .5(A - B)$		$M_D = H_Dh_D$	
$H_g = W_{m1} - H$		$h_G = .5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = .5(h_D + h_G)$		$M_T = H_Th_T$	
$H_r = T_2 \cos \alpha$		h_r		$M_r = H_rh_r$	
$\alpha = \arcsin .5\,A/R$		$M_o = M_D + M_G + M_T +/- M_r$ Note: M_r is (+) if C.L. of shell/head is below C.G., (-) if above			
$T_2 = P\,R$					
		Seating			
$H_G = W$		h_G		$M_o' = Wh_G$	

7 FLANGE THICKNESS CALCULATION	
Thickness required- Gasket seating	
$T_G = \sqrt{\dfrac{M_o'}{S_{fa}B}\left[\dfrac{A+B}{A-B}\right]}$	
Thickness required- Operating	
$F = \dfrac{PB\sqrt{4L^2 - B^2}}{8S_{fo}(A - B)}$	
$J = \dfrac{M}{S_{fo}B}\left(\dfrac{A+B}{A-B}\right)$	
$T_o = F + \sqrt{F^2 + J}$	





X = Minimum depth of threads (UG-44(b))

$X = (.75\,d\,S_b) / S_{fo} > 1.5\,d$

TYPE 4: REVERS FLANGE DESIGN

1 DESIGN CONDITIONS

		Allowable Stresses			
Design pressure, P	400 PSIG	Flange		Bolting	
Design temperature	1,000 deg F				
Flange material	SB-564-800H	Design temp., S_{fo}	14,000	Design temp., S_b	32,300
Bolting material	SB-637	Amb.temp., S_{fa}	16,700	Amb.temp., S_a	37,000

2 GASKET AND FACING DETAILS

Gasket	Spiral Wound	Facing	RF

3 / **4 LOAD AND BOLT CALCULATIONS**

N	1.00	$W_{m2} = b\pi Gy$	222,110	A_m = greater of W_{m2}/S_a or W_{m1}/S_b	6.00
b	.3535	$H_P = 2b\pi GmP$	53,306		
G	20″	$H = G^2\pi P/4$	125,663	A_b	6.612
y	10,000PSI	$W_{m1} = H_P+H$	178,970	$W = .5(A_m + A_b)S_a$	233,322
m	3.0				

5 GASKET WIDTH CHECK

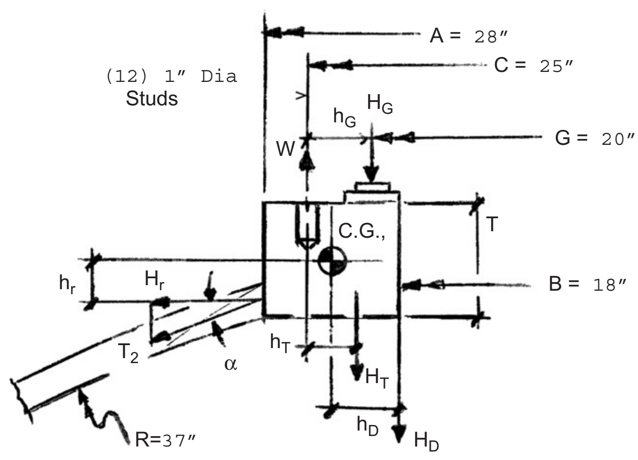
$N_{min}=A_bS_a/2y\pi G$	.195

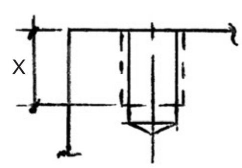
6 MOMENT CALCULATIONS

Load		x Lever Arm		= Moment	
Operating					
$H_D = \pi B^2P/4$	101,787	$h_D = .5(A-B)$	2.50	$M_D = H_Dh_D$	254,468
$H_g = W_{m1}-H$	53,306	$h_G = .5(C-G)$	2.50	$M_G = H_Gh_G$	133,265
$H_T = H - H_D$	23,876	$h_T = .5(h_D-h_G)$	2.313	$M_T = H_Th_T$	55,213
$H_r = T_2\cos\alpha$	13,700	h_r	4.20	$M_r = H_rh_r$	57,540
α = arc sin.5A/R	22.23 deg	$M_o = M_D + M_G + M_T +/- M_r$ Note: M_r is(+) if C.L.of shell/head is		500,486	
$T_2 = PR$	14,800	below C.G.,(-) if above			
Seating					
$H_G = W$	233,322	h_G	2.50	$M_o' = Wh_G$	583,305

7 FLANGE THICKNESS CALCULATION

Thickness required-Gasket seating	
$T_{\cdot G} = \sqrt{\dfrac{M_o'}{S_{fa}B}\left[\dfrac{A+B}{A-B}\right]}$	2.98″
Thickness required-Operating	
$F = \dfrac{PB\sqrt{4L^2-B^2}}{8S_{fo}(A-B)}$	.4614
$J = \dfrac{M}{S_{fo}B}\left(\dfrac{A+B}{A-B}\right)$	9.14
$T_o = F+\sqrt{F^2+J}$	3.52″





X = Minimum depth of threads (UG-44(b))

$X = (.75\ d\ s_b)/S_{fo} > 1.5d$

1.73 > 1.5″

OK

Procedure 3-6: Reinforcement for Studding Outlets

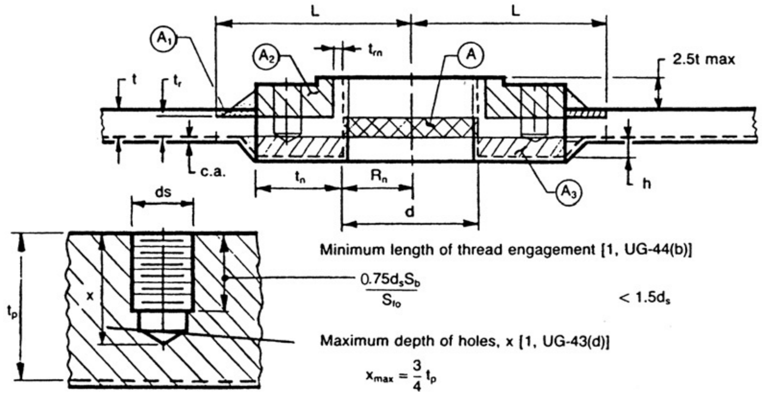


Figure 3-19. Typical studding outlet.

Table 3-11
Tapped Hole Area Loss, S, in.²*

d_s	⅝ in.	¾ in.	⅞ in.	**1 in.**	1⅛ in.	**1¼ in.**	1⅜ in.	**1½ in.**	1⅝ in.	1¾ in.	1⅞ in.	**2 in.**	**2 ¼ in.**
X	1.11	1.33	1.55	1.77	2.00	2.44	2.66	2.88	3.10	3.32	3.56	3.98	4.44
S	1.28	1.84	2.50	3.28	4.15	5.12	6.20	7.38	8.66	10.05	11.55	13.1	16.6

** Values of x and s are based on the shoulder of the hole at a depth of 1.5d_s and fully threaded.*

Calculation of Area of Reinforcement (Figure 3-19)

$$A = (dt_rF) + S$$

$$L = \text{Greater of } d \text{ or } R_n + t_n + t$$

$$A_1 = 2(L - R_n - t_n)(t - t_r)$$

$$A_2 = 2(t_p - h - t_r)(t_n - t_m)$$

$$A_3 = 2(ht_n)$$

$$A_T = A_1 + A_2 + A_3$$

Notes

1. Check plane which is nearest the longitudinal axis of the vessel and passes through a pair of studded holes.
2. S_b = allowable stress of stud material at design temperature.
   S_{fo} = allowable stress of flange material at design temperature.
3. A_2 as computed ignores raised face.

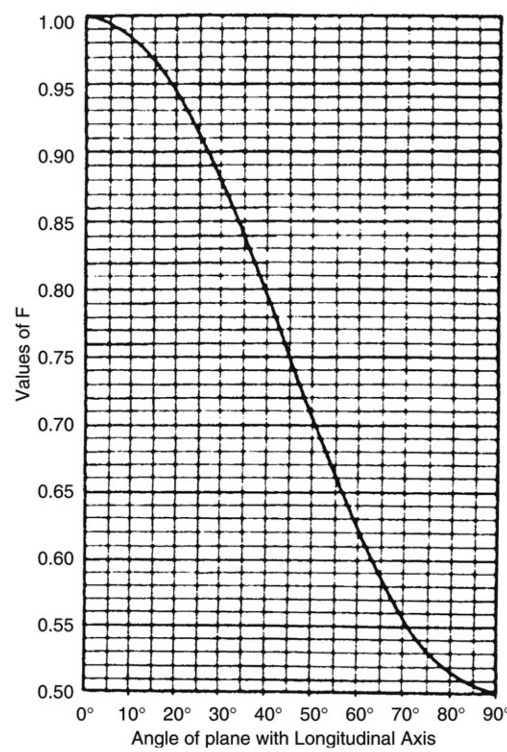


Figure 3-20. Chart for determining the value of F. *(Reprinted by permission from ASME Code, Section VIII, Div. 1, Figure UG-37.)*

Procedure 3-7: Studding Flanges

TYPE 6 : STUDDING FLANGE

1		DESIGN CONDITIONS			
Design pressure, P		Allowable Stresses			
Design temperature		Flange		Bolting	
Flange material		Design temp., S_{fo}		Design temp., S_b	
Bolting material		Atm. temp., S_{fa}		Atm. temp., S_a	
Corrosion allowance					

2	GASKET AND DETAILS		
Gasket		Facing	

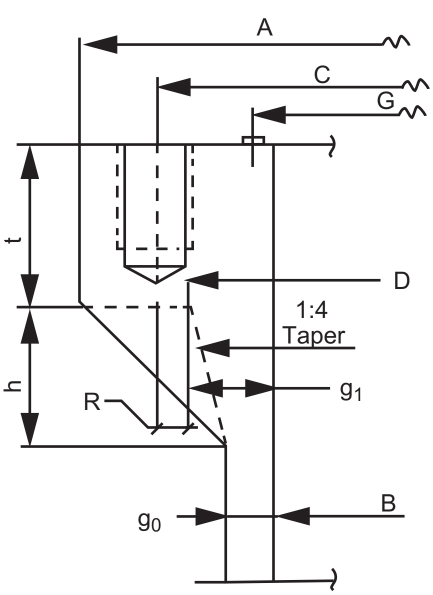
3		4	LOAD AND BOLT CALCULATIONS		
N		$W_{m2} = b\pi Gy$		A_m = greater of W_{m2}/S_a or W_{m1}/S_b	
b		$H_P = 2b\pi GmP$			
G		$H = G^2\pi P/4$		A_b	
y		$W_{m1} = H_P + H$		$W = .5(A_m + A_b)S_a$	
m					

5		MOMENT CALCULATIONS			
Load		x Lever Arm		=	Moment
		Operating			
$H_D = \pi B^2P/4$		$h_D = R + .5g_1$		$M_D = H_Dh_D$	
$H_G = W_{m1} - H$		$h_G = .5(C - G)$		$M_G = H_Gh_G$	
$H_T = H - H_D$		$h_T = .5(R + g_1 + h_G)$		$M_T = H_Th_T$	
				M_o	
		Seating			
$H_G = W$		$h_G = .5(C - G)$		M_o'	

6	K AND HUB FACTORS		
$K = A/B$		h/h_o	
T		F	
Z		V	
Y		f	
U		$e = F/h_o$	
g_1/g_o		$d = \frac{U}{V} h_o g_o^2$	
$h_o = \sqrt{Bgo}$			

7	STRESS FORMULA FACTORS		
t			
$\alpha = te + 1$			
$\beta = 4/3\ te + 1$			
$\gamma = \alpha/T$			
$\delta = t^3/d$			
$\lambda = \gamma + \delta$			
$m_o = M_o/B$			
$m_G = M_o'/B$			

if bolt spacing exceeds 2a + t, multiply m_o and m_G in above equation by: $\sqrt{\frac{\text{Bolt spacing}}{2a + t}}$



8		STRESS CALCULATIONS			
Allowable Stress	Operating		Allowable Stress	Seating	
$1.5\ S_{fo}$	Longitudinal hub, $S_H = fm_o/\lambda gt^2 = g_1^2$		$1.5\ S_{fa}$	Longitudinal hub, $S_H = fm_o = m_G/\lambda gt^2 = g_1^2$	
S_{fo}	Radial flange, $S_R = \beta m_o/\lambda gt^2 = t^2$		S_{fa}	Radial flange, $S_R = \beta m_o = m_G/\lambda gt^2 = t^2$	
S_{fo}	Tangential flange, $S_T = m_oY/t^2 - ZS_R$		S_{fa}	Tangential flange, $S_T = m_GY/t^2 - ZS_R$	
S_{fo}	Greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$		S_{fa}	Greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$	

Nomenclature

A_b = Total area of studs, in^2
A_m = Area of studs required, in^2
A_r = Area required for one stud, in^2
A_g = Actual gasket area, in^2
B_S = Stud spacing, in
d_S = Diameter of stud, in
f_{a1} = Load between b and r_i, Lbs/in
f_{a2} = Load between r_i and r_o, Lbs/in
f_{a3} = Load between r_o and a, Lbs/in
f_{an} = net load, Lbs/in
f_{aT} = Total load, Lbs/in
G_o, G_i = Gasket OD or ID
L = Minimum length of thread engagement, in
n = Number of studs
R_a = Root area of one stud, in^2
R_f = diameter of raised face, in
S_b = Allowable stress of stud at design temperature, PSI
S_{fo} = Allowable stress of flange material at design temperature, PSI
S_{if} = Hoop stress in flange at flange ID, PSI
S_{ri} = Hoop stress in flange at inner surface of main stud hole, PSI
S_{ro} = Hoop stress in flange at outer surface of main stud hole, PSI
S_{of} = Hoop stress at flange OD, PSI
S'_{if} = Corrected hoop stress at flange ID, PSI
Y_1 = Correction ratio

Establish Flange Dimensions

- Area of studs required, A_m

  $A_m = W_{m1}/S_b$
- Area required for one stud, A_r

  $A_r = A_m/n$
- Stud selection

  Qty, n = ________
  Dia, d_S = ________
  Root Area, R_a = ________
- Total area of studs, A_b

  $A_b = nR_a$
- Minimum length of thread engagement, L
  Greater of ….

$L = (.75\ d_s\ S_b)/S_{fo}$ or $1.5d_S$

- Depth of hole, F

  $F = L + .25''$Min
- Depth of drill tip, C_O

  $C_O = .288d_S$
- Determine bolt circle diameter, C
  Larger of …

1. $C = B + d_S + 2''$
2. $C = R_f + d_S + 2''$
3. $C = (B_S\ n)/\pi$

   Use, C = ________

- Hub proportions;
  Assuming a theoretical 1:4 taper for the hub, find corresponding dimensions of h and g_1

$g_1 = .25h + g_o$
$h = 4(g_1 - g_o)$

Use, h = ________
And g_1 = ________

$D = B + 2\ g_1$
$R = .5(C - D)$

- Flange OD, A
  Larger of …

1. $C + 2(d_S - .125'')$
2. $B + 2\ g_o + 2d_S$

   Use A = _______

- Minimum flange thickness, t_{min}

  $t_{min} = F + .5(a - r_o)$
- Gasket dimensions,

  G_o = ______
  G_i = ______
  N = ______
  b_o = N / 2 = ______
  b = $b_o^{1/2}$ / 2 = ______
  G = G_o − 2 b = ________
  R_f = ________
  Use OD of groove for M&F or T&G flange facing
- Gasket area, A_g

  $A_g = (\pi\ G_o^2)/4 - (\pi\ G_i^2)/4$
- Gasket width check, N_{min}

  $N_{min} = (A_h\ S_a)/(2\ \pi\ y\ G)$

WORKSHEET FOR STUDDING FLANGES

GIVEN		LOADS & STRESSES			
FLANGE OD, A OR A_1		$S_{if} = P[(a^2 + b^2) / (a^2 - b^2)]$		$f_{an} = f_{at} - [(F + .5C_O) f_{a2} / t]$	
FLANGE ID, B					
BOLT CIRCLE, C		$S_{ri} = [Pb^2/(a^2 - b^2)][1 + a^2/r_i^2]$		$Y_1 = f_{at} / f_{an}$	
HYDROSTATIC END FORCE, H					
REQUIRED BOLT LOAD, OPERATING, W_{m1}		$S_{ro} = [Pb^2/(a^2 - b^2)][1 + a^2/r_o^2]$		$S'_{if} = Y_1 S_{if}$	
AREA OF GASKET, A_G					
DIAMETER OF STUDS, d_S		$S_{of} = (2Pb^2)/(a^2 - b^2)$		**NOTES:**	
PRESSURE, P				1. If S'_{if} </= Allowable Stress, than the design is OK	
THICKNESS, t		$f_{a1} = .5(r_i - b)(S_{if} + S_{ri})$		2. If S'_{if} > Allowable Stress, then implement a or b below	
DIMENSIONS					
$a = .5(A \text{ or } A_1)$		$f_{a2} = .5(r_o - r_i)(S_{ri} + S_{ro})$		a. Increase flange proportions	
$b = .5B$				b. Add a shrink ring	
$r_i = .5(C - d_S)$		$f_{a3} = .5(a - r_o)(S_{ro} + S_{of})$			
$r_O = .5(C + d_S)$					
$F = L + .25''$		$f_{aT} = f_{a1} + f_{a2} + f_{a3}$			
$C_O = .288d_S$					

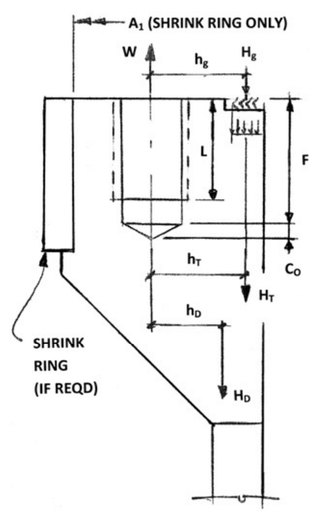


Figure 3.21. Dimensions & Forces

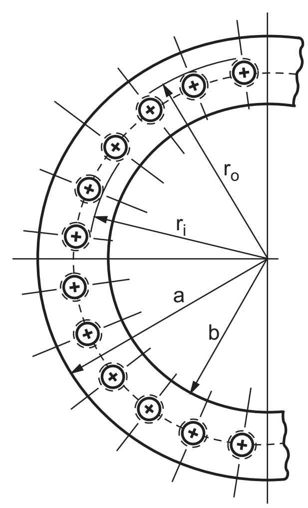


Figure 3.22. Plan View - Dimensions

TYPE 6 : STUDDING FLANGE

1	DESIGN CONDITIONS				
Design pressure, P	1700 PSIG	Allowable Stresses			
Design temperature	810 Deg F	Flange		Bolting	
Flange material	SA-182-F22	Design temp., S_{to}	22,360	Design temp., S_b	19,260
Bolting material	SA-193-B7	Atm. temp., S_{ta}	25,000	Atm. temp., S_a	23,000
Corrosion allowance	0				

2	GASKET AND FACING DETAILS
Gasket	Facing

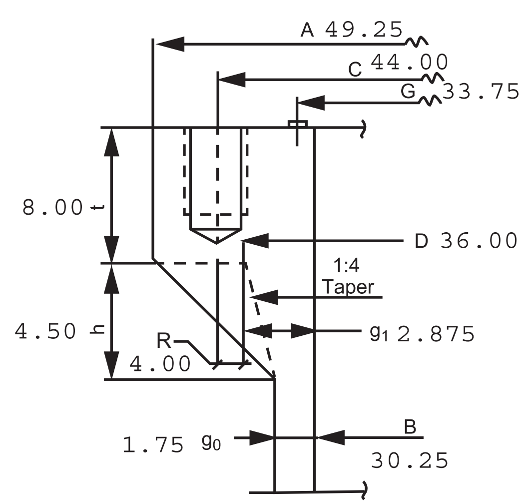
3		4	LOAD AND BOLT CALCULATIONS		
N	2	$W_{m2} = b\pi Gy$	530,143	A_m = greater of W_{m2}/S_a or W_{m1}/S_b	107.04
b	0.5	$H_p = 2b\pi GmP$	540,747		
G	33.75	$H = G^2\pi P/4$	1,520,850	A_b	126.2
y	10,000	$W_{m1} = H_p + H$	2,061,597	$W = .5(A_m + A_b)S_a$	2,682,260
m	3				

5	MOMENT CALCULATIONS				
Load		x Lever Arm		= Moment	
Operating					
$H_D = \pi B^2P/4$	1,221,770	$h_D = R + .5g_1$	5.438	$M_D = H_Dh_D$	6,643,375
$H_G = W_{m1} - H$	540,747	$h_G = 5(C - G)$	5.125	$M_G = H_Gh_G$	2,771,328
$H_T = H - H_D$	299,080	$h_T = .5(R + g_1 + h_G)$	6.00	$M_T = H_Th_T$	1,794,480
				M_O	11,209,183
Seating					
$H_G = W$	2,682,260	$h_G = .5(C - G)$	5.125	M'_O	13,746,583

6	K AND HUB FACTORS		
K = A/B	1.63	h/h_0	.618
T	1.65	F	.822
Z	2.21	V	.270
Y	4.15	f	1.00
U	4.56	$e = F/h_0$	.113
g_1/g_0	1.64	$d = \frac{U}{V}h_0g_0^2$	376
$h_0 = \sqrt{Bg_0}$	7.27		

7	STRESS FORMULA FACTORS
t	8.00
$\alpha = te + 1$	1.90
$\beta = 4/3\ te + 1$	2.21
$\gamma = \alpha/T$	1.15
$\delta = t^3/d$	1.36
$\lambda = \gamma + \delta$	2.51
$m_0 = M_0/B$	370,551
$m_G = M'_0/B$	454,432

if bolt spacing exceeds 2a + t, multiply m_o and m_G in above equation by: $\sqrt{\frac{\text{Bolt spacing}}{2a + t}}$



8	STRESS CALCULATION				
Allowable Stress	Operating		Allowable Stress	Seating	
1.5 S_{fo} 33,540	Longitudinal hub, $S_H = fm_o/\lambda g_1^2$	17,860	1.5 S_{fa} 37,500	Longitudinal hub, $S_H = fm_G/\lambda g_1^2$	21,903
S_{fo} 22,360	Radial flange, $S_R = \beta m_o/\lambda t^2$	5,097	S_{fa} 25,000	Radial flange, $S_R = \beta m_G/\lambda t^2$	6,251
S_{fo} 22,360	Tangential flange, $S_T = m_oY/t^2 - ZS_A$	12,763	S_{fa} 25,000	Tangential flange, $S_T = m_GY/t^2 - ZS_R$	15,650
S_{fo} 22,360	Greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$	15,311	S_{fa} 25,000	Greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$	18,776

WORKSHEET FOR STUDDING FLANGES

GIVEN		LOADS & STRESSES			
FLANGE OD, A OR A_1	49.25	$S_{if} = P [(a^2 - b^2) / (a^2 - b^2)]$	3760	$f_{an} = f_{at} - [(F + .5\ C_o)\ f_{a2} / t]$	22,346
FLANGE ID, B	30.25				
BOLT CIRCLE, C	44	$S_{ri} = [P\ b^2 / (a^2 - b^2)]\ [1 + a^2 / r_i^2]$	2498	$Y_1 = f_{at} / f_{an}$	1.18
HYDROSTATIC END FORCE, H	1,520,850				
REQUIRED BOLT LOAD, OPERATING, W_{m1}	2,061,597	$S_{ro} = [P\ b^2 / (a^2 - b^2)]\ [1 + a^2 / r_o^2]$	2172	$S'_{if} = Y_1\ S_{if}$	4421
AREA OF GASKET, A_G	205.77				
DIAMETER OF STUDS, d_S	2.75	$S_{of} = (2\ P\ b^2) / (a^2 - b^2)$	2060	**NOTES:**	
PRESSURE, P	1700			1. If S'_{if} </= Allowable Stress, than the design is OK	
THICKNESS, t	8	$f_{a1} = .5\ (r_i - b)\ (S_{if} + S_{ri})$	17,210	2. If S'_{if} > Allowable Stress, then implement a or b below	
DIMENSIONS					
$a = .5$ (A or A_1)	24.625	$f_{a2} = .5\ (r_o - r_i)\ (S_{ri} + S_{ro})$	6421	a. Increase flange proportions	
$b = .5$ B	15.125			b. Add a shrink ring	
$r_i = .5\ (C - d_S)$	20.625	$f_{a3} = .5\ (a - r_o)\ (S_{ro} + S_{of})$	2645		
$r_O = .5\ (C + d_S)$	23.375				
$F = L + .25''$	4.5	$f_{aT} = f_{a1} + f_{a2} + f_{a3}$	26,276		
$C_O = .288\ d_S$	0.792				

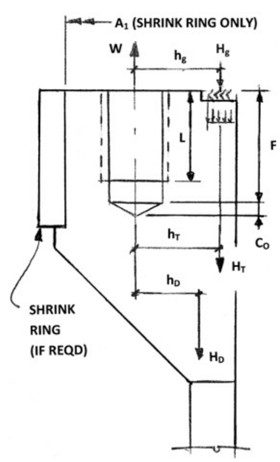


Figure 3.23. Dimensions & Forces

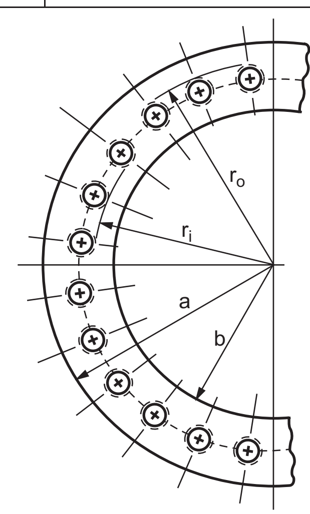
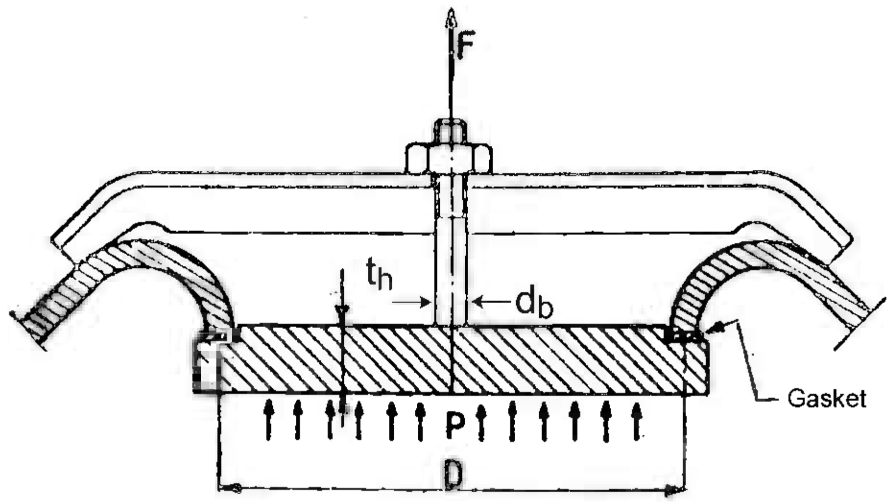


Figure 3.24. Plan View - Dimensions

Procedure 3-8: Design of Elliptical, Internal Manways

DESIGN OF ELLIPTICAL, INTERNAL MANWAYS



NOTATION

C = Attachment Factor

E = Joint Efficiency, Cat A seam only

F = Total force exerted by stud(s), Lbs

P = Design Pressure, PSIG

S = Allowable Stress, head, PSI

S_b = Allowable Stress, bolt, PSI

Z = Factor dependent on D/d ratio

CALCULATIONS

- Maximum force applied from stud(s), F

$$F = .785\, d_b^2\, S_b$$

- Determine attachment factor, C

$$C = \sqrt{0.31 + 0.95 \frac{F}{PD^2}}$$

- Since head is non-circular, determine factor, Z

$$Z = 3.4 - (2.4d) / D; < 2.5$$

- Thickness required for head, t_h

$$t_h = d\sqrt{\frac{Z\,C\,P}{S\,E}}$$

EXAMPLE

D = 20"

d = 14"

P = 750 PSIG

S = 20,000 PSI

S_b = 25,000 PSI

d_b = 1.25"

E = 1.0

$$F = .785\,(1.25^2)\,25{,}000 = 30{,}664 \text{ Lbs}$$

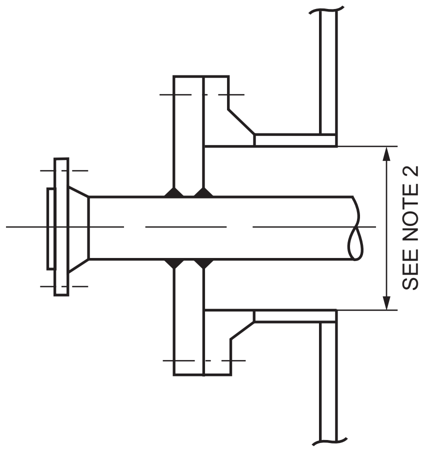
$$C = \sqrt{.31 + .95\,[\,30{,}664 / (750\,(20^2)\,]} = .638$$

$$Z = 3.4 - [2.4(14)] / 20 = 1.72$$

$$t_h = 14\sqrt{\frac{1.72(.638)\,750}{20{,}000\,(1.0)}} = 2.84"$$

Procedure 3-9: Through Nozzles

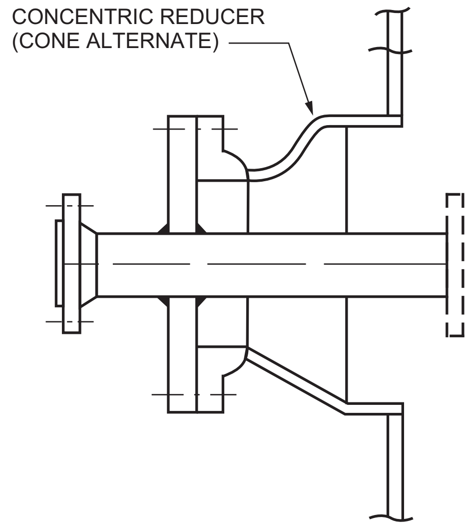
THROUGH NOZZLES



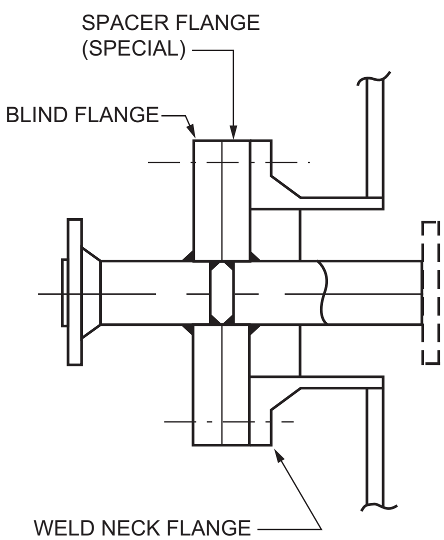
Notes:

1. Through flanges provide the ability to pull the internal pipe from outside
2. Ensure the ID of flange is large enough to clear the internal flange
3. Use where extra flexibility is required

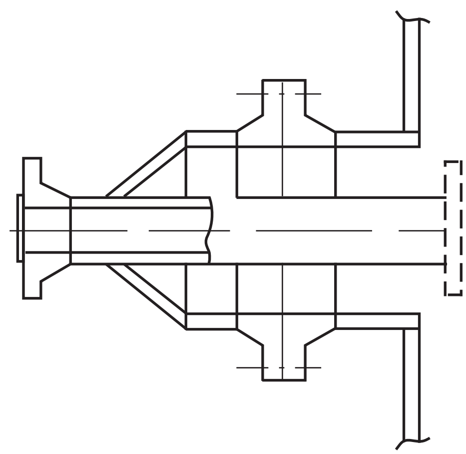
TYPE 1



TYPE 2



TYPE 3



TYPE 4

References

[1] ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, 2010 Edition, American Society of Mechanical Engineers.

[2] Modern Flange Design, 7th Edition, Bulletin 502, Taylor Forge International, Inc.

[3] Brownell LE, Young EH. Process Equipment Design. John Wiley and Sons, Inc; 1959. Chapter 12.

[4] ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, 2010 Edition, American Society of Mechanical Engineers.

[5] Radzinovsky EI. Bolt Design for Repeated Loading. Machine Design November 1952. pp. 135ff.

[6] Meyer G, Strelow D. Simple Diagrams Aid in Analyzing Forces in Bolted Joints. Assembly Engineering January 1972:28–33.

[7] Horsch R. Solve Complicated Force Problems with Simple Diagrams. Assembly Engineering December 1972:22–4.

This page intentionally left blank

4
Design of Vessel Supports

Introduction: Support Structures

There are various methods that are used in the support structures of pressure vessels, as outlined below.

- Skirt Supports
  1. Cylindrical
  2. Conical
  3. Pedestal
  4. Shear ring
- Leg Supports
  1. Braced
     a. Cross braced (pinned and unpinned)
     b. Sway braced
  2. Unbraced
  3. Stub columns
- Saddle Supports
- Lug Supports
- Ring Supports
- Combination Supports
  1. Lugs and legs
  2. Rings and legs
  3. Skirt and legs
  4. Skirt and ring girder

Skirt Supports

One of the most common methods of supporting vertical pressure vessels is by means of a rolled cylindrical or conical shell called a skirt. The skirt can be either lap-, fillet-, or butt-welded directly to the vessel. This method of support is attractive from the designer's standpoint because it minimizes the local stresses at the point of attachment, and the direct load is uniformly distributed over the entire circumference. The use of conical skirts is more expensive from a fabrication standpoint, and unnecessary for most design situations.

The critical line in the skirt support is the weld attaching the vessel to the skirt. This weld, in addition to transmitting the overall weight and overturning moments, must also resist the thermal and bending stresses due to the temperature drop in the skirt. The thinner the skirt, the better it is able to adjust to temperature variations. A "hot box" design is used for elevated temperatures to minimize discontinuity stresses at the juncture by maintaining a uniform temperature in the region. In addition, skirts for elevated temperature design will normally be insulated inside and outside for several feet below the point of attachment.

There are various methods of making the attachment weld of the skirt to the shell. The preferred method is the one in which the center line of the shell and skirt coincide. This method will minimize stresses at the juncture. Probably the most common method, however, is to make the OD of the skirt match the OD of the shell. Other methods of attachment include lap-welding, pedestal type, or a shear ring arrangement. The joint efficiency of the attachment weld also varies by the method of attachment and is usually the governing factor in determining the skirt thickness. This weld may be subject to cracking in severe cyclic service.

Because the skirt is an attachment to the pressure vessel, the selection of material is not governed by the ASME Code. Any material selected, however, should be compatible with the vessel material in terms of weldability. Strength for design is also not specified for support material by the ASME Code. Usually, in the absence of any other standard, the rules of the AISC Steel Construction Manual will be utilized. Nonmandatory Appendix G in the ASME Code, Section VIII, Division 1 contains general guidelines on skirt supports (and other types of supports). Additionally, Part 4 in the ASME Code, Section VIII, Division 2 contains rules regarding applied forces, localized stresses, and thermal gradients for skirt supports for vessels designed to Division 2, but may be used for good practice of skirt supports for vessels designed to Division 1. For elevated temperature design of a vessel with a support skirt made of different materials, the upper portion of the skirt should be the same material of the shell, however, the upper portion should also extend below the hotbox. A thermal analysis should be performed to determine the temperature gradient along the length of the skirt and the location where another material may be used for the skirt support.

The most common governing conditions for determining the thickness of the skirt are as follows:

1. Weight + overturning moment
2. Imposed loads from anchor chairs
3. Vessel erection

Leg Supports

A wide variety of vessels, bins, tanks, and hoppers may be supported on legs. The designs can vary from small vessels supported on 3 or 4 legs, to very large vessels and spheres up to 80 feet in diameter, supported on 16 or 20 legs. Sometimes the legs are also called columns or posts.

Almost any number of legs can be used, but the most common variations are 3, 4, 6, 8, 12, 16, or 20. Legs should be equally spaced around the circumference.

Leg supports may be braced or unbraced. Braced legs are those which are reinforced with either cross-bracing or sway-bracing. Sway braces are the diagonal members which transfer the horizontal loads, but unlike cross braces, they operate in tension only. The diagonal members in a sway-braced system are called tie rods, which transfer the load to each adjacent panel. Turnbuckles may be used for adjustments of the tie rods.

Cross braces, on the other hand, are tension and compression members. Cross braces can be pinned at the center or unpinned, and transfer their loads to the legs via wing plates or can be welded directly to the legs.

Bracing is used to reduce the number or size of legs required by eliminating bending in the legs. The bracing will take the horizontal loads, thus reducing the size of the legs to those determined by compression or buckling. The additional fabrication costs of bracing may not warrant the savings in the size of the legs, however. Bracing may also cause some additional difficulties with the routing of any piping connected to nozzles on the bottom of the vessel.

Legs may be made out of pipe, channels, angles, rectangular tubing, or structural sections. Legs may be welded directly to the vessel shell or head or may be bolted or welded to clips which are directly attached to the shell. It is preferable if the centroid of the leg coincides with the center line of the vessel shell to minimize the eccentric action. However, this may be more expensive from a welding and fit up viewpoint due to the coping and contouring necessary to accomplish this.

Very large vessels and tanks may require a circumferential box girder, compression ring, or ring girder at or near the attachment point of the legs to distribute the large localized loads induced by the columns and bracing. These localized stresses at the attachment point should be analyzed for the eccentric action of the legs, overturning moments, torsion of the ring, as well as the loads from any bracing.

Whereas skirt-supported vessels are more common in refinery service, leg-supported vessels are more common in the chemical industry. This may be due in part to the ventilation benefits and the toxicity of the stored or processed chemicals. Legs should not be used to support vessels in high-vibration, shock, or cyclic service due to the high localized stresses at the attachments.

Legs are anchored to the foundations by base plates, which are held in place by anchor bolts embedded in the concrete. For large vessels in high seismic areas, a shear bar may be welded to the underside of the base plate which, in turn, fits into a corresponding recessed groove in the concrete.

Saddle Supports

Usually, horizontal pressure vessels and tanks are supported on two vertical cradles called saddles. The use of more than two saddles is unnecessary and should be avoided. Using more than two saddles is normally a stress-related issue, which can be solved in a more conventional manner. The reason for not using more than two saddles is that it creates an indeterminate structure, both theoretically and practically. With two saddles, there is a high tolerance for soil settlement with no change in shell stresses or loading. Even where soil settlement is not an issue, it is difficult to ensure that the load is uniformly distributed. Obviously there are ways to accomplish this, but the additional expense is often unwarranted. Vessels 40-50 ft in diameter and 150 ft long have been supported on two saddles.

A methodology for the determination of the stresses in the shell and heads of a horizontal vessel supported on saddles was first published in 1951 by L. P. Zick. This effort was a continuation of others' work, started as early as the 1930s. This procedure has been used, with certain refinements since that time, and is often called Zick's analysis, or the stresses are referred to as Zick's stresses.

Zick's analysis is based on the assumption that the supports are rigid and are not connected to the vessel shell. In reality, most vessels have flexible supports which are attached to the vessel, usually by welding. Whatever the reason, and there are a myriad of them, Zick's assumptions may yield an analysis that is not 100% accurate. These results should, however, be viewed more in terms of the performance they have demonstrated in the past 45 years, than in the exact analytical numbers they produce. As a strategy, the procedure is successful when utilized properly. There are other issues that also would have an effect on the outcome of the numerical answers such as the relative rigidity of the saddle—from infinitely rigid to flexible. The answers should be viewed in light of the assumptions as well as the necessity for 5-digit accuracy.

The ASME Code, Section VIII, Division 2 contains rules for determining the actual and allowable stresses for a vessel being supported by two saddles, with or without reinforcing plates, and with or without stiffening rings. These rules are based largely on Zick's analysis. However, as with all other types of supports, the ASME Code does not have specific design procedures for the design of saddles. Typically, the allowable stresses utilized are those as outlined in the *AISC Steel Construction Manual*.

The saddle itself has various parts: the web, base plate, ribs, and wear plate. The web can be on the center line of the saddle or offset. The design may have outer ribs only or inner ribs only, but usually it has both. For designs in seismic areas, the ribs perform the function of absorbing the longitudinal, horizontal loads. The saddle itself is normally bolted to a foundation via anchor bolts. The ASME Code does specify the minimum included arc angle (contact angle) of 120°. The maximum efficient saddle angle is 180°, since the weight and saddle splitting force go to zero above the belt line. In effect, taking into account the 6° allowed for reduction of stresses at the horn for wear plates, the maximum angle becomes 168°.

Saddles may be steel or concrete. They may be bolted, welded, or loose. For the loose type, some form of liner should be used between the vessel and the saddle. The typical loose saddle is the concrete type. Usually one end of the vessel is anchored and the other end sliding. The sliding end may have bronze, oiled, or Teflon slide plates to reduce the friction caused by the thermal expansion or contraction of the vessel.

Longitudinal location of the saddles also has a large effect on the magnitude of the stresses in the vessel shell as well as a bearing on the design of the saddle parts themselves. For large diameter, thin-walled vessels, the saddles are best placed within 0.5R of the tangent line to take advantage of the stiffening effect of the heads. Other vessels are best supported where the longitudinal bending at the midspan is approximately equal to the longitudinal bending at the saddles. However, the maximum distance is 0.2 L.

Lugs and Ring Supports

Lugs. Lugs offer one of the least expensive and most direct ways of supporting pressure vessels. They can readily absorb diametral expansion by sliding over greased or bronzed plates, are easily attached to the vessel by minimum amounts of welding, and are easily leveled in the field.

Since lugs are eccentric supports they induce compressive, tensile, and shear forces in the shell wall. The forces from the eccentric moments may cause high localized stresses that are combined with stresses from internal or external pressure. In thin-walled vessels, these high local loads have been known to physically deform the vessel wall considerably. Such deformations can cause angular rotation of the lugs, which in turn can cause angular rotations of the supporting steel.

Two or four lug systems are normally used; however, more may be used if the situation warrants it. There is a wide variety of types of lugs, and each one will cause different stress distributions in the shell. Either one or two gussets can be used, with or without a compression plate. If a compression plate is used, it should be designed to be stiff enough to transmit the load uniformly along the shell. The base plate of the lug can be attached to the shell wall or unattached. Reinforcing pads can be used to reduce the shell stresses. In some cases, the shell course to which the lugs are attached can be made thicker to reduce the local stress.

The method shown utilizes the local load analysis developed by Bijlaard in the 1950s, which was further refined and described in the WRC Bulletin 107. This procedure uses the principles of flexible load surfaces.

When making decisions regarding the design of lugs, a certain sequence of options should be followed. The following represents a ranking of these options based on the cost to fabricate the equipment:

1. 2 lugs, single gusset
2. 2 lugs, double gussets
3. 2 lugs with compression plate
4. Add reinforcing pads under (2) lugs
5. Increase size of (2) lugs
6. 4 lugs, single gusset
7. 4 lugs, double gussets
8. 4 lugs with compression plates
9. Add reinforcing pads under (4) lugs
10. Increase size of (4) lugs
11. Add ring supports

Ring Supports. In reality, ring supports are used when the local stresses at the lugs become excessively high. As can be seen from the previous list, the option to go to complete, 360-degree stiffening rings would, in most cases, be the most expensive option. Typically, vessels supported by rings or lugs are contained within a structure rather than supported at grade and as such would be subject to the seismic movement of which they are a part.

Vessels supported on rings should only be considered for lower or intermediate temperatures, say below 400 or 500 degrees. Using ring supports at higher temperatures could cause extremely large discontinuity stresses in the shell immediately adjacent to the ring due to the differences in expansion between the ring and the shell. For elevated temperature design, rings may still be used, but should not be directly attached to the shell wall. A totally loose ring system can be fabricated and held in place with shear bars. With this system there is no interaction between the shell and the support rings.

The analysis for the design of the rings and the stresses induced in the shell employs the same principles as Lug Method 1, ring analysis. The eccentric load points are translated into radial loads in the rings by the gussets. The composite ring section comprised of the shell and ring is then analyzed for the various loads.

Procedure 4-1: Wind Design Per ASCE [1]

Notation

A_f = projected area, ft^2 (m^2)
$\bar{b}$ = mean hourly wind speed factor
C_f = force coefficient, shape factor 0.7, 0.8, and 0.9 for h/D_e of 1, 7, and 25, respectively (linear interpolation is permitted). See ASCE/SEI 7-10.
c = turbulence intensity factor
D_e = vessel effective diameter, from Table 4-4
F = design wind force

$q_z G C_f A_f$ (lb)(N)

F_i = design wind force of section under consideration, i = 1 to n, lb (N)
g_Q = peak factor for background response, use 3.4
g_R = peak factor for resonant response
g_v = peak factor for wind response, use 3.4
G = gust effect factor
G_f = gust response factor for flexible vessels
H_i = height from base of vessel to center of section under consideration, i = 1 to n, ft (m)
h = height of vessel, ft (m)
h_i = length of section under consideration, i = 1 to n, ft (m)
$I_{\bar{z}}$ = the intensity of turbulence at height z
K_d = wind directionality factor, use 0.95 for vessels when using ASCE/SEI 7-10 load combinations
K_z = velocity pressure exposure coefficient from Table 4-3a
K_{zt} = topographic factor, use 1.0 unless vessel is located near or on isolated hills. See ASCE/SEI 7-10 for specific requirements
L_z = integral length scale of turbulence, ft (m)
ℓ = integral length scale factor, ft (m)
M = overturning moment at base, ft-lb (N-m)
M_i = moment at base of section under consideration, i = 1 to n, ft-lb (N-m)
N_1 = reduced frequency
n_1 = fundamental natural frequency, Hz
Q = background response factor
q_z = velocity pressure at height z above the ground

$0.00256\ K_z K_{zt} K_d V^2$ (lb/ft^2) or
$0.613\ K_z K_{zt} K_d V^2$ (N/m^2)

R = resonant response factor

R_B, R_h, R_L, R_n = calculation factors
T = period of vibration, sec
V = basic wind speed from map, Figures 4-1a, 4-1b, and 4-1c, mph (m/s)
V_i = shear force at base of section under consideration, i = 1 to n, lb (N)
$\overline{V}_{\bar{z}}$ = mean hourly wind speed at height $\bar{z}$, ft/sec (m/s)
W = weight of vessel, lb (N)
z = height above ground level, ft (m)
$\bar{z}$ = equivalent height of vessel, ft (m)
z_{min} = minimum design height, ft (m), from Table 4-3
$\bar{\alpha}$ = mean hourly wind-speed power law exponent
β = damping ratio (structural), percent of critical from Table 4-3
bedrock, endbearing piles, or other rigid bases 0.2%
friction piles or mat foundations on soil 0.4%
$\bar{\in}$ = integral length scale power law exponent
η_B, η_h, η_L = calculation factors

The ASME Code does not give specific procedures for designing vessels for wind. However, Para. UG-22, "Loadings," does list wind as one of the loadings that must be considered. In addition, local, state, or other governmental jurisdictions will require some form of analysis to account for wind loadings. Client specifications and standards also frequently require consideration of wind. There is one nationally recognized standard that is most frequently used for wind design.

ASCE/SEI 7-10

This section outlines the wind design procedures for this standard. Wind design is used to determine the forces and moments at each elevation to check if the calculated shell thicknesses are adequate. The overturning moment at the base is used to determine all of the anchorage and support details. These details include the number and size of anchor bolts, thickness of skirt, size of legs, and thickness of base plates.

As a loading, wind differs from seismic in that it is more or less constant; whereas, seismic is of relatively short duration. In addition, the wind pressure varies with the height of the vessel. A vessel must be designed for the worst case of wind or seismic, but need not be designed for both simultaneously. While typically the worst case for seismic design is with the vessel full (maximum weight), the worst design case for wind is with the vessel empty. This will produce the maximum uplift due to the minimum restraining weight.

The wind forces are obtained by multiplying the projected area of each element, within each height zone by the basic wind pressure for that height zone and by the shape factor for that element. The total force on the vessel is the sum of the forces on all of the elements. The forces are applied at the centroid of the projected area.

Tall towers or columns should be checked for dynamic response. If the vessel is above the critical line in Figure 4-7, R_m/t ratio is above 200 or the h/D ratio is above 15, then dynamic stability should be investigated. This section does not consider aerodynamic damping effects, however it is possible that the aerodynamic damping contribution is negative under certain conditions. If this is the case, the overall effect of the structural damping would be reduced. See Procedure 6-5, "Vibration of Tall Towers and Stacks," for additional information.

Design Procedure

Risk category	=	______
Basic wind speed, V	=	______
Exposure category	=	______
Effective diameter, D_e	=	______
Height of vessel, h	=	______
Shape factor, C_f	=	______
Fundamental frequency, f n_t	=	______
Damping ratio, structural, β	=	______

Step 1: Give or determine the following:

Step 2: Determine if vessel is rigid or flexible.

a. If $n_1 \geq 1$ Hz, then vessel is considered rigid and:

$$F = q_z G_f C_f A_f$$

b. If $n_1 < 1$ Hz, then vessel is considered flexible and:

$$F = q_z G_f C_f A_f$$

Step 3: Calculate shear and moments at each elevation by multiplying force, F, and elevation, H_n, the distance to the center of the projected area.

Step 4: Sum the forces and moments at each elevation down to the base.

Determination of Gust Factor, G, for Vessels Where $n_1 \geq 1$ Hz

For rigid structures, the gust factor may be taken as 0.85 or as calculated below:

Given: D_e	=	______	(effective diameter)
h	=	______	(overall height)
g_Q, g_v	=	3.4	

Determine the following values from Table 4-3:
Calculate:

$\bar{\alpha}$	=	______	ℓ	=	______
$\bar{b}$	=	______	$\bar{\in}$	=	______
c	=	______	z_{min}	=	______

$$\bar{z} = \max(0.6h, z_{min})$$

$$I_{\bar{z}} = c\left(\frac{33}{\bar{z}}\right)^{1/6}, \ I_{\bar{z}} = c\left(\frac{10}{\bar{z}}\right)^{1/6} \text{(SI)}$$

$$L_{\bar{z}} = \ell\left(\frac{\bar{z}}{33}\right)^{\bar{\in}}, \ L_{\bar{z}} = \ell\left(\frac{\bar{z}}{10}\right)^{\bar{\in}} \text{(SI)}$$

$$Q = \sqrt{\frac{1}{1 + 0.63\left(\frac{D_e + h}{L_{\bar{z}}}\right)^{0.63}}}$$

$$G = 0.925\left(\frac{1 + 1.7 g_Q I_{\bar{z}} Q}{1 + 1.7 g_v I_{\bar{z}}}\right)$$

Determination of Gust Factor, G_f, for Vessels Where $n_1 < 1$ Hz

In addition to the tabular data above, the following must be given to determine the gust factor, G_f:

Given: n_1	=	______	(fundamental natural frequency)
V	=	______	(basic wind speed)
β	=	______	(damping ratio, structural)

Calculate ($\bar{z}, I_{\bar{z}}, L_{\bar{z}}, Q$ are as determined above):

$$g_R = \sqrt{2\ln(3,600 n_1)} + \frac{0.577}{\sqrt{2\ln(3,600 n_1)}}$$

$$\bar{V}_{\bar{z}} = \bar{b}\left(\frac{\bar{z}}{33}\right)^{\bar{\alpha}}\left(\frac{88}{60}\right)V, \ \bar{V}_{\bar{z}} = \bar{b}\left(\frac{\bar{z}}{10}\right)^{\bar{\alpha}} V \text{(SI)}$$

$$N_1 = \frac{n_1 L_{\bar{z}}}{\bar{V}_{\bar{z}}}$$

$$\eta_h = 4.6 n_1 h / \bar{V}_{\bar{z}}$$

$$\eta_B = 4.6 n_1 D_e / \bar{V}_{\bar{z}}$$

$$\eta_L = 15.4 n_1 D_e / \bar{V}_{\bar{z}}$$

$$R_h = \frac{1}{\eta_h} - \frac{1}{2\eta_h^2}\left(1 - e^{-2\eta_h}\right) \text{ for } \eta > 0,$$

$$R_h = 1 \text{ for } \eta = 0$$

$$R_B = \frac{1}{\eta_B} - \frac{1}{2\eta_B^2}\left(1 - e^{-2\eta_B}\right) \text{ for } \eta > 0,$$

$$R_B = 1 \text{ for } \eta = 0$$

$$R_L = \frac{1}{\eta_L} - \frac{1}{2\eta_L^2}\left(1 - e^{-2\eta_L}\right) \text{ for } \eta > 0,$$

$$R_L = 1 \text{ for } \eta = 0$$

$$R_n = \frac{7.47 N_1}{(1 + 10.3 N_1)^{5/3}}$$

$$R = \sqrt{\frac{1}{\beta} R_n R_h R_B (0.53 + 0.47 R_L)}$$

$$G_f = 0.925\left(\frac{1 + 1.7 I_{\bar{z}} \sqrt{g_Q^2 Q^2 + g_R^2 R^2}}{1 + 1.7 g_v I_{\bar{z}}}\right)$$

Sample Problem

Vertical vessel on skirt:

Risk category	=	III
Basic wind speed, V	=	115 mph
Exposure category	=	C
Effective diameter, D_e	=	8 ft
Height of vessel, h	=	200 ft
Shape factor, C_f	=	0.9
Fundamental frequency	=	0.57 Hz
Damping ratio (structural)	=	0.01
Empty weight, W	=	100 kips

Values from Table 4-3:

$\overline{\alpha}$	=	1/6.5	ℓ	=	500
$\overline{b}$	=	0.65	$\overline{\in}$	=	1/5.0
c	=	0.20	z_{min}	=	15

Calculate:

$$\overline{z} = \max(0.6h, z_{min}) = \max(0.6\ (200), 15) = 120 \text{ ft}$$

$$I_{\overline{z}} = c\left(\frac{33}{\overline{z}}\right)^{1/6} = 0.20\left(\frac{33}{120}\right)^{1/6} = 0.161$$

$$L_{\overline{z}} = \ell\left(\frac{\overline{z}}{33}\right)^{\overline{\in}} = 500\left(\frac{120}{33}\right)^{1/5.0} = 647 \text{ ft}$$

$$Q = \sqrt{\frac{1}{1+0.63\left(\frac{D_e+h}{L_{\overline{z}}}\right)^{0.63}}}$$

$$= \sqrt{\frac{1}{1+0.63\left(\frac{8+200}{647}\right)^{0.63}}} = 0.874$$

$$g_Q = g_v = 3.4$$

$$g_R = \sqrt{2\ \ln(3,600n_1)} + \frac{0.577}{\sqrt{2\ \ln(3,600n_1)}}$$

$$= \sqrt{2\ \ln(3,600(0.57))} + \frac{0.577}{\sqrt{2\ \ln(3,600(0.57))}}$$

$$= 4.05$$

$$\overline{V}_{\overline{z}} = \overline{b}\left(\frac{\overline{z}}{33}\right)^{\overline{\alpha}}\left(\frac{88}{60}\right)V = (0.65)\left(\frac{120}{33}\right)^{1/6.5}\left(\frac{88}{60}\right)(115)$$

$$= 134 \text{ ft/sec}$$

$$N_1 = \frac{n_1 L_{\overline{z}}}{\overline{V}_{\overline{z}}} = \frac{(0.57)(647)}{(134)} = 2.76$$

$$\eta_h = 4.6n_1h/\overline{V}_{\overline{z}} = 4.6(0.57)(200)/(134) = 3.92$$

$$\eta_B = 4.6n_1D_e/\overline{V}_{\overline{z}} = 4.6(0.57)(8)/(134) = 0.157$$

$$\eta_L = 15.4n_1D_e/\overline{V}_{\overline{z}} = 15.4(0.57)(8)/(134) = 0.525$$

$$R_h = \frac{1}{\eta_h} - \frac{1}{2\eta_h^2}\left(1-e^{-2\eta_h}\right)$$

$$= \frac{1}{(3.92)} - \frac{1}{2(3.92)^2}\left(1-e^{-2\ (3.92)}\right) = 0.222$$

$$R_B = \frac{1}{\eta_B} - \frac{1}{2\eta_B^2}\left(1-e^{-2\eta_B}\right)$$

$$= \frac{1}{(0.157)} - \frac{1}{2(0.157)^2}\left(1-e^{-2\ (0.157)}\right) = 0.903$$

$$R_L = \frac{1}{\eta_L} - \frac{1}{2\eta_L^2}\left(1-e^{-2\eta_L}\right)$$

$$= \frac{1}{(0.525)} - \frac{1}{2(0.525)^2}\left(1-e^{-2\ (0.525)}\right) = 0.725$$

$$R_n = \frac{7.47\ N_1}{(1+10.3N_1)^{5/3}} = \frac{7.47\ (2.76)}{(1+10.3(2.76))^{5/3}} = 0.074$$

$$R = \sqrt{\frac{1}{\beta}R_nR_hR_B(0.53+0.47R_L)}$$

$$= \sqrt{\frac{1}{(0.01)}(0.074)(0.222)(0.903)(0.53+0.47(0.725))}$$

$$= 1.13$$

$$G_f = 0.925\left(\frac{1+1.7I_{\overline{z}}\sqrt{g_Q^2Q^2+g_R^2R^2}}{1+1.7g_vI_{\overline{z}}}\right)$$

$$= 0.925\left(\frac{1+1.7(0.161)\sqrt{(3.4)^2(0.874)^2+(4.05)^2(1.134)^2}}{1+1.7(3.4)(0.161)}\right)$$

$$= 1.20$$

$$F = q_zG_fC_fA_f = 32.163K_z(0.9)A_f = 28.947A_fK_z$$

where $q_z = 0.00256K_zK_{zt}K_dV^2 = 32.163K_z$, A_f is calculated using the section length and the effective diameter, and K_z is determined using the elevations at the top of each section.

Determine Wind Force on Vessel

Elevation	q_z	G_f	C_f	h_i	A_f	Force on A_f, F_i	ΣShear, V_i	ΣMoment, M_i
190–200 ft	47.1 psf	1.20	0.90	10 ft	80 ft^2	4,061 lb	4,061 lb	20,303 ft-lb
170–190 ft	46.6 psf	1.20	0.90	20 ft	160 ft^2	8,034 lb	12,095 lb	181,855 ft-lb
150–170 ft	45.5 psf	1.20	0.90	20 ft	160 ft^2	7,848 lb	19,943 lb	502,228 ft-lb
130–150 ft	44.3 psf	1.20	0.90	20 ft	160 ft^2	7,644 lb	27,587 lb	977,520 ft-lb
110–130 ft	43.0 psf	1.20	0.90	20 ft	160 ft^2	7,417 lb	35,004 lb	1,603,423 ft-lb
95–110 ft	41.5 psf	1.20	0.90	15 ft	120 ft^2	5,371 lb	40,374 lb	2,168,759 ft-lb
85–95 ft	40.3 psf	1.20	0.90	10 ft	80 ft^2	3,472 lb	43,846 lb	2,589,860 ft-lb
75–85 ft	39.3 psf	1.20	0.90	10 ft	80 ft^2	3,391 lb	47,237 lb	3,045,274 ft-lb
65–75 ft	38.3 psf	1.20	0.90	10 ft	80 ft^2	3,303 lb	50,540 lb	3,534,161 ft-lb
55–65 ft	37.2 psf	1.20	0.90	10 ft	80 ft^2	3,205 lb	53,745 lb	4,055,587 ft-lb
45–55 ft	35.9 psf	1.20	0.90	10 ft	80 ft^2	3,094 lb	56,839 lb	4,608,510 ft-lb
35–45 ft	34.4 psf	1.20	0.90	10 ft	80 ft^2	2,966 lb	59,806 lb	5,191,735 ft-lb
27.5–35 ft	32.6 psf	1.20	0.90	7.5 ft	60 ft^2	2,110 lb	61,916 lb	5,648,191 ft-lb
22.5–27.5 ft	31.0 psf	1.20	0.90	5 ft	40 ft^2	1,337 lb	63,253 lb	5,961,112 ft-lb
17.5–22.5 ft	29.7 psf	1.20	0.90	5 ft	40 ft^2	1,282 lb	64,535 lb	6,280,580 ft-lb
0–17.5 ft	28.2 psf	1.20	0.90	17.5 ft	140 ft^2	4,255 lb	68,789 lb	7,447,165 ft-lb

Exposure Categories

The following ground roughness exposure categories are considered and defined in ASCE/SEI 7-10 Section 26.7.3:

- *Exposure B:* For buildings with a mean roof height of less than or equal to 30 ft (9.1 m). Urban and suburban areas, towns, city out skirts, wooded areas, or other terrain with numerous closely spaced obstructions having the size of single family dwellings or larger.
- *Exposure C:* For cases where Exposures B and D do not apply. Open terrain with scattered obstructions having heights generally less than 30 ft (9.1 m).
- *Exposure D:* Flat, unobstructed coastal areas directly exposed to wind blowing over open water.

Notes

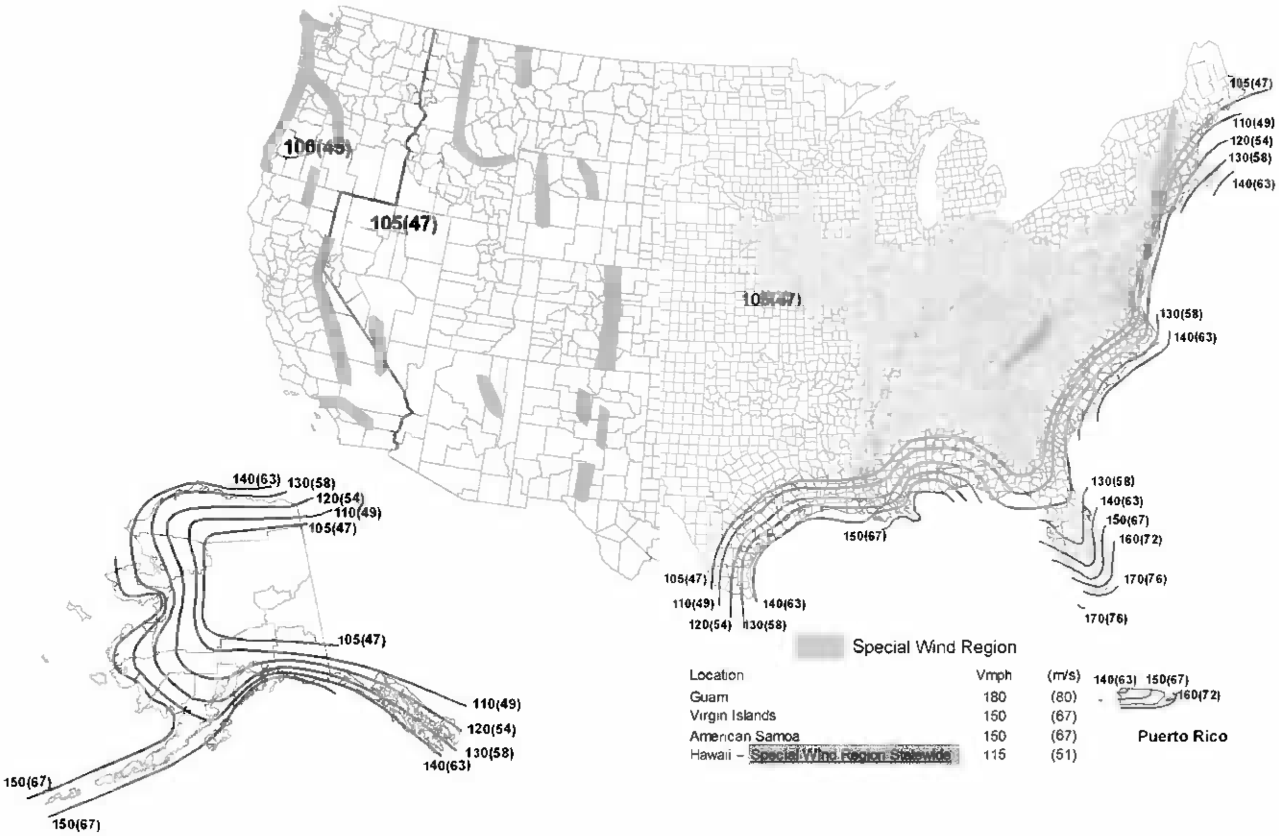
1. Most vessels will be classified as Category III.
2. The basic wind speeds on the map, Figures 4-1a, 4-1b, and 4-1c, correspond to a 3-sec. gust speed at 33 ft above the ground, in Exposure Category C with a 7% / 3% / 15% probability of exceedance in 50 years.
3. The constant, 0.00256 (0.613), reflects the mass density of air for the standard atmosphere (59°F (15 °C) at sea level pressure, 29.92 in. of mercury (101.325 kPa)). The constant is calculated by ½ ρ_{air}/g, where ρ_{air} is the density of air and g is the acceleration due to gravity. The mass density of the air will vary as function of altitude, latitude, temperature, weather, or season. This constant may be varied to suit the actual conditions if they are known with certainty. See ASCE/SEI 7-10.
4. Short, vertical vessels, vessels in structures, or horizontal vessels where the height is divided between two pressure zones may be more conveniently designed by applying the higher pressure uniformly over the entire vessel.
5. Deflection due to wind should be limited to 6 in. per 100 ft of elevation.

For 15 ft. $\leq z \leq z_g$

$$K_z = 2.01\left(z/z_g\right)^{2/\alpha}$$

For z < 15 ft.

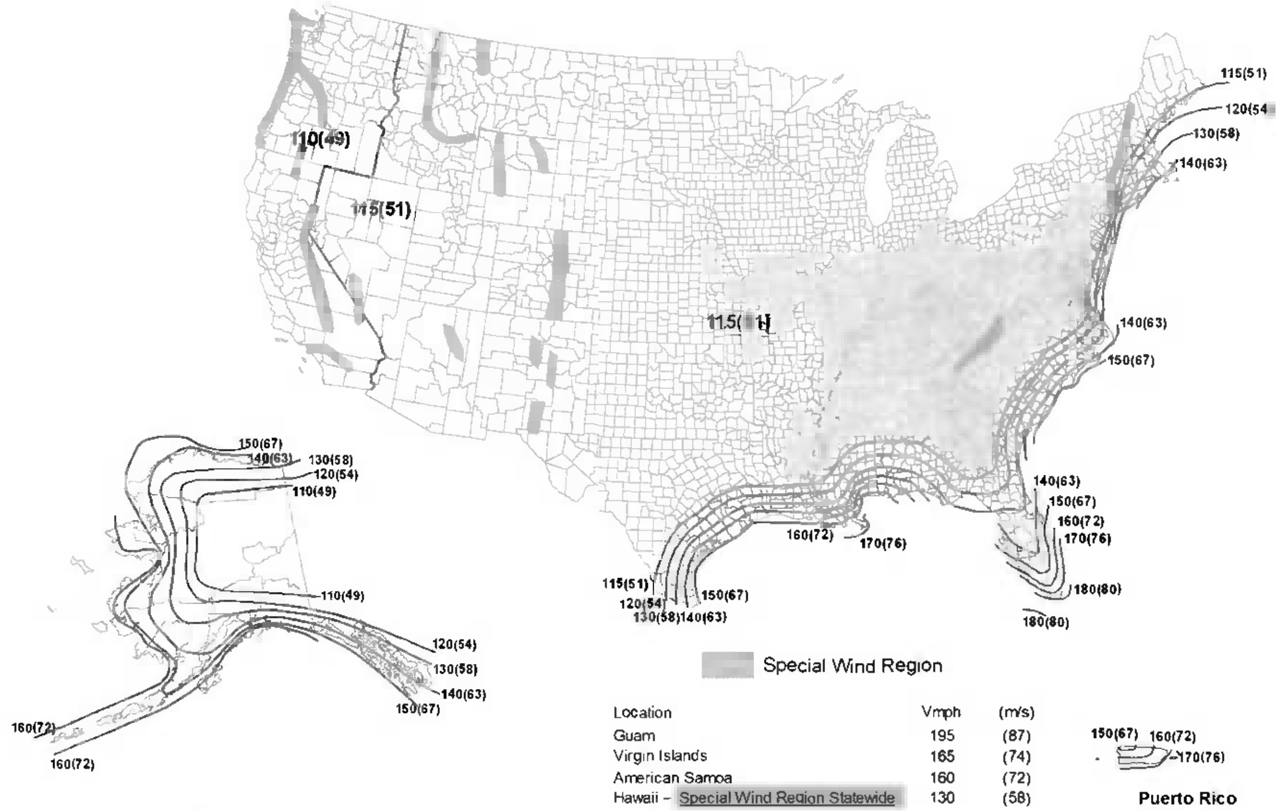
$$K_z = 2.01\left(15/Z_g\right)^{2/\alpha}$$



Notes:

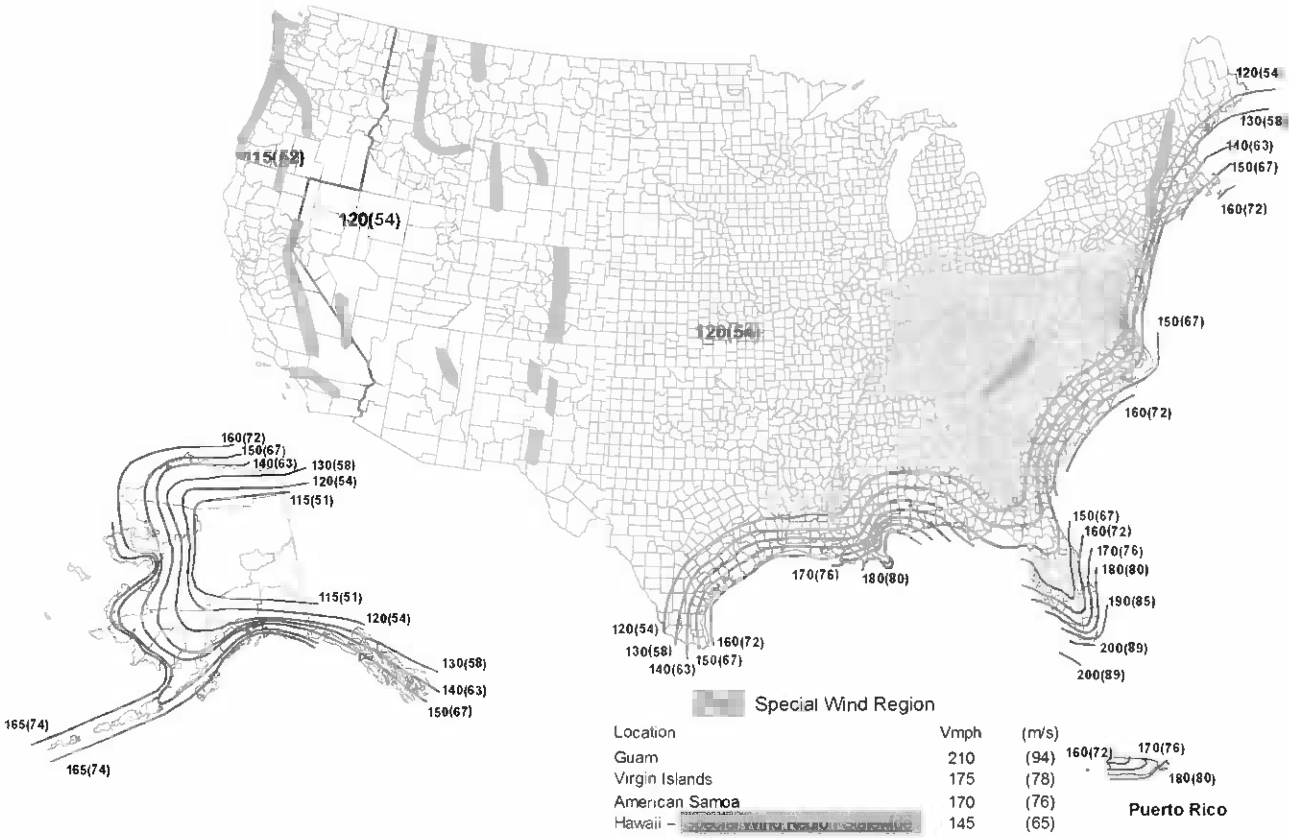
1. Values are nominal design 3-second gust wind speeds in miles per hour (m/s) at 33 ft (10m) above ground for Exposure C category.
2. Linear interpolation between contours is permitted.
3. Islands and coastal areas outside the last contour shall use the last wind speed contour of the coastal area.
4. Mountainous terrain, gorges, ocean promontories, and special wind regions shall be examined for unusual wind conditions
5. Wind speeds correspond to approximately a 15% probability of exceedance in 50 years (Annual Exceedance Probability = 0.00333, MRI = 300 Years)

Figure 4-1a. Basic Wind Speeds for Occupancy Category I Buildings and Other Structures. *With permission from ASCE.*



Notes:
1. Values are nominal design 3-second gust wind speeds in miles per hour (m/s) at 33 ft (10m) above ground for Exposure C category.
2. Linear interpolation between contours is permitted.
3. Islands and coastal areas outside the last contour shall use the last wind speed contour of the coastal area.
4. Mountainous terrain, gorges, ocean promontories, and special wind regions shall be examined for unusual wind conditions.
5. Wind speeds correspond to approximately a 7% probability of exceedance in 50 years (Annual Exceedance Probability = 0.00143, MRI = 700 Years).

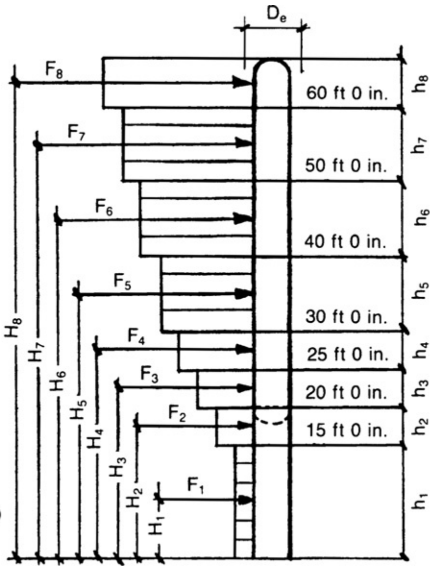
Figure 4-1b. Basic Wind Speeds for Occupancy Category II Buildings and Other Structures. *With permission from ASCE.*



Notes:

1. Values are nominal design 3-second gust wind speeds in miles per hour (m/s) at 33 ft (10m) above ground for Exposure C category.
2. Linear interpolation between contours is permitted.
3. Islands and coastal areas outside the last contour shall use the last wind speed contour of the coastal area.
4. Mountainous terrain, gorges, ocean promontories, and special wind regions shall be examined for unusual wind conditions.
5. Wind speeds correspond to approximately a 3% probability of exceedance in 50 years (Annual Exceedance Probability = 0.000588, MRI = 1700 Years)

Figure 4-1c. Basic Wind Speeds for Occupancy Category III and IV Buildings and Other Structures. *With permission from ASCE.*



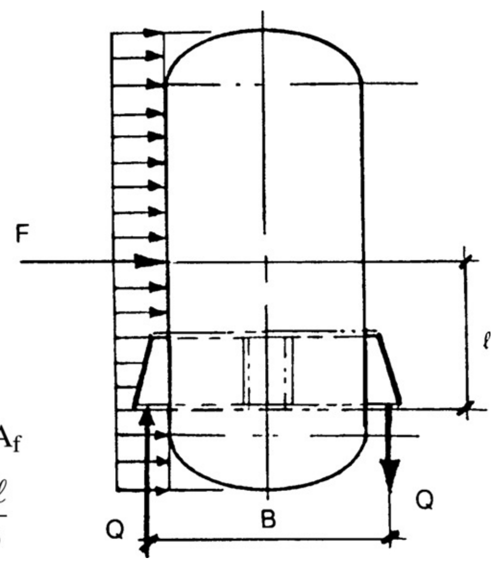
$A_f = h_n D_e$

$F_i = q_z G C_f A_f$

$M_i = F_i(h_i/2) + V_{i+1}(h_i) + M_{i+1}$

$M = \Sigma F_i H_i$

Figure 4-2. Vertical vessels.

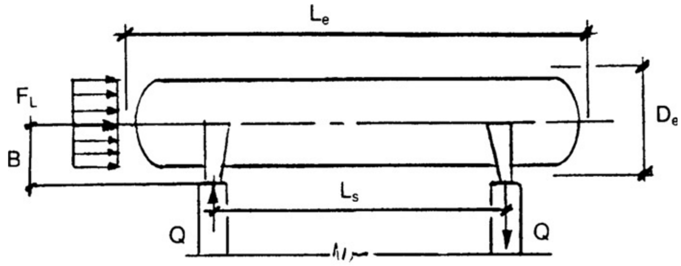


$A_f = L_e D_e$

$F = q_z G C_f A_f$

$Q = \dfrac{W}{N} \pm \dfrac{F\ell}{B}$

Figure 4-4. Vessels on lugs or rings.



Longitudinal

$A_L = \dfrac{\pi D_e^2}{4}$

$F_L = q_z G C_f A_L$

$Q = \dfrac{W}{2} \pm \dfrac{F_L B}{L_s}$

Transverse

$A_t = L_e D_e$

$F_t = 0.5(q_z G C_f A_t)$ (one support)

$M_t = F_t B$ (one support)

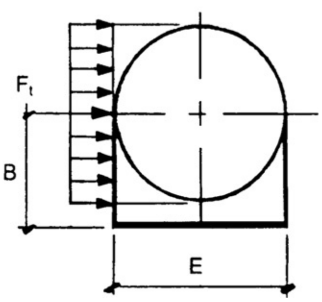
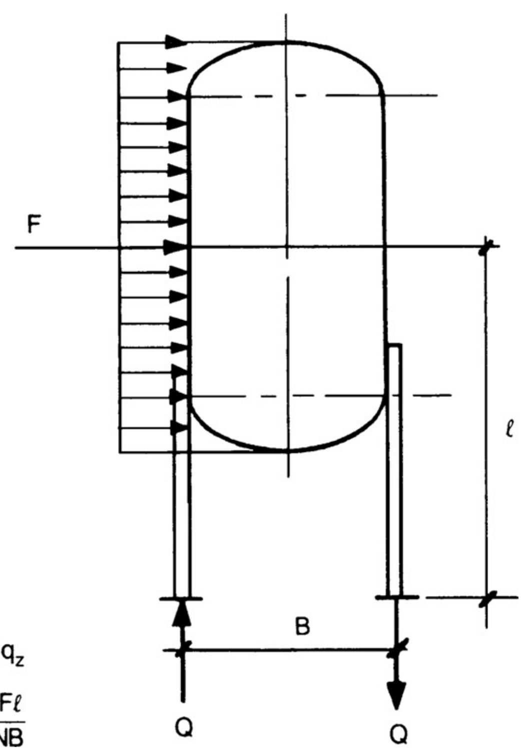


Figure 4-3. Horizontal vessels.



$A_f = L_e D_e$

$F = A_f C_f G q_z$

$Q = \dfrac{W}{N} \pm \dfrac{4F\ell}{NB}$

Figure 4-5. Vessels on legs.

Table 4-1
Importance Factor (Wind Loads)

Structure Category	I
I	0.87
II	1.00
III	1.15
IV	1.15

Table 4-2
Risk Category of Buildings and Other Structures for Flood, Wind, Snow, Earthquake, and Ice Loads

Buildings and structures that represent a low risk to human life in the event of failure.	Category I
All buildings and other structures not covered by Risk Categories I, III, and IV.	Category II
Buildings and other structures containing sufficient quantities of toxic or explosive substances to be dangerous to the public if released. Typically, the equipment inside of a refinery falls under this category.	Category III
Schools, non-emergency health care facilities, jails, non-essential power stations	Category III
Essential facilities	Category IV
Buildings and other structures containing sufficient quantities of toxic or explosive substances to be dangerous to the public if released. (Buildings and other structures containing these substances may be eligible to be classified in a lower category if it can be demonstrated to the jurisdictional authority through a special assessment that the lower risk category is acceptable).	Category IV

With permission from ASCE

Table 4-3
Miscellaneous coefficients

Expos.	α	z_g (ft/m)	$\bar{\alpha}$	$\bar{b}$	$\bar{c}$	ℓ (ft/m)	$\bar{\epsilon}$	$*z_{min}$ (ft/m)
B	7.0	1200/365.76	1/4.0	0.45	0.30	320/97.54	1/3.0	30/9.14
C	9.5	900/274.32	1/6.5	0.65	0.20	500/152.4	1/5.0	15/4.57
D	11.5	700/213.36	1/9.0	0.80	0.15	650/198.12	1/8.0	7/2.13

$*z_{min}$ = minimum height used to ensure that the equivalent height $\bar{z}$ is the greater of 0.6 h or z_{min}.

Table 4-3a*
Velocity pressure exposure coefficients, K_z

Height above ground level, z		Exposure Categories		
ft	(m)	B	C	D
0-15	(0-4.6)	0.57	0.85	1.03
20	(6.1)	0.62	0.90	1.08
25	(7.6)	0.66	0.94	1.12
30	(9.1)	0.70	0.98	1.16
40	(12.2)	0.76	1.04	1.22
50	(15.2)	0.81	1.09	1.27
60	(18.0)	0.85	1.13	1.31
70	(21.3)	0.89	1.17	1.34
80	(24.4)	0.93	1.21	1.38
90	(27.4)	0.96	1.24	1.40
100	(30.5)	0.99	1.26	1.43
120	(36.6)	1.04	1.31	1.48
140	(42.7)	1.09	1.36	1.52
160	(48.8)	1.13	1.39	1.55
180	(54.9)	1.17	1.43	1.58
200	(61.0)	1.20	1.46	1.61
250	(76.2)	1.28	1.53	1.68
300	(91.4)	1.35	1.59	1.73
350	(106.7)	1.41	1.64	1.78
400	(121.9)	1.47	1.69	1.82
450	(137.2)	1.52	1.73	1.86
500	(152.4)	1.56	1.77	1.89

Note: Linear interpolation for intermediate values of height z is acceptable.
K_z may be determined from the following formula:

Table 4-4
Effective diameter, D_e*

D (Vessel Diameter + 2 x Insulation Thickness)	Piping with or Without Ladders	Attached Piping, Ladders, and Platforms
$\leq$ 4ft – 0 in.	$D_e = 1.6D$	$D_e = 2.0D$
4ft - 0 in. – 8 ft – 0 in.	$D_e = 1.4D$	$D_e = 1.6D$
$>$ 8 ft – 0 in.	$D_e = 1.2D$	$D_e = 1.4D$

**Suggested only; not from ASCE.*

Procedure 4-2: Seismic Design – General

Pressure vessels and their supports must be designed to resist the forces and loadings anticipated during a seismic event … an earthquake. The seismic design is not defined by the ASME Code but by building codes (previously NBC, SBC, and UBC, but now IBC) that reference technical standards such as ASCE/SEI 7, ACI 318, and AISC 360. Many countries have their own seismic standards and there are international standards as well. The ASME Code states in UG-22 that the vessel and support structure must be designed to withstand the forces from a seismic event.

A seismic event causes the vessel to sway as a result of the ground motion. How much loading the vessel experiences is dependent on the type of foundation and supports, the size and proportions of the vessel, the geographic location of the vessel, and the type of soil. A tall, thin, slender cylindrical tower mounted at grade, is relatively flexible and will therefore have a long period and low frequency. By contrast a short, squat vessel will have a short period and higher frequency. Vessels mounted in or on structures will be influenced by the relative stiffness of the structure.

Seismic standards are all based on the geographical and statistical data for a given region. The standards use various criteria to estimate the loads on the vessel or structure and probability of occurrence. Some regions have high probability for very strong earthquakes to occur. Other regions are almost negligible in terms of seismic events. Seismic codes and standards date back to the 1920's. Modern, industrial societies have rigorous building codes that account for earthquakes. The building codes may not define all types of procedures but they allow for the various design procedures.

Additionally, the site class has an effect on the design loadings. In general, site classes comprised of hard rock will have less intense shaking than site classes composed of soft soils.

Seismic design procedures can be accomplished for most vessels by one of the two methods as follows;

a. Equivalent Lateral Force (ELF) – Static Analysis
b. Modal Response Spectrum Analysis – Dynamic Analysis

Seismic design criteria may provide linear and non-linear seismic response history procedures, but these are not as commonly used for vessels.

Equivalent Lateral Force

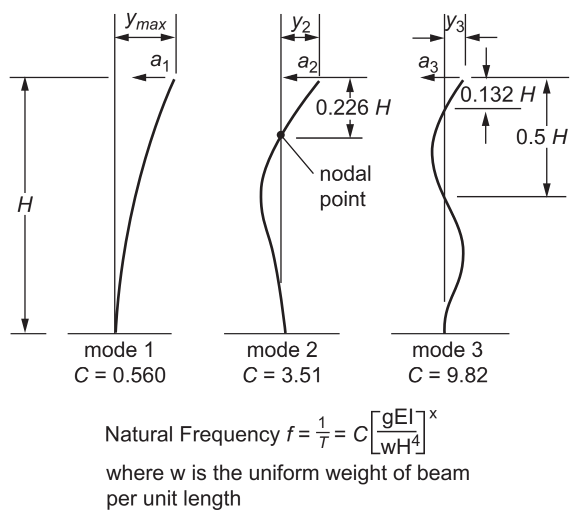
The ELF approximates the effect that the ground displacements would have on the structure by applying an equivalent force to the structure itself. A seismic event is a time-dependent phenomena whereby the loading is not applied instantaneously, but over a period of time. However, the ELF assumes that the entire earthquake force is applied instantaneously. The ELF is conservative and has served industry and society well for many years.

The procedure takes the total base shear and distributes it along the length of the column. Once the vertical distribution of the lateral seismic force is determined, the shear force and bending moment at each plane and the sum at the base of the column can be determined. The vertical component of the seismic design loads can be either added or subtracted to create the most stringent condition. These loads used with the corresponding load combinations are used to design all support components.

Modal Response Spectrum Analysis

The Modal Response Spectrum Analysis (also known as a dynamic analysis) more accurately depicts the response of the structure to the earthquake. This is done by considering the response of multiple modes instead of just the first one (as is done in the ELF). Whereas the use of a static analysis assumes that a load is applied relatively slowly, a dynamic analysis should be used if the application of the load is faster than the response of the structure. For this reason a dynamic analysis is mainly used for vertical vessels which are basically a cantilevered cylinder. A dynamic analysis frequently results in lower overturning moments than the ELF. Lower moments in turn translate into reduced thickness for skirt and base plate and fewer anchor bolts. For this reason the question is asked whether a dynamic analysis is less conservative than a static analysis, however the dynamic analysis is a more accurate representation of the way the structure responds to the earthquake-induced ground motion.

For rigid vessels, the first few modes may represent the majority of the modal mass participation, whereas for flexible vessels, the number of modes may be 20. It is for this reason that dynamic analyses lend themselves to computerized models. Many seismic design standards indicate that the number of modes to be included must have a combined mass participation of at least 90%.



Allowable Stresses and Load Combinations

Unlike wind, seismic events are short term loading conditions. As a result, the ASME Code Section VIII, Division 1 allows for an increase in the allowable stress of 1.2. Section VIII, Division 2, building codes, and design standards (such as ASCE/SEI 7) use load combinations and typically do not allow for an increase in allowable stress, however the seismic load is usually reduced when combined with other types of loads and so the effect is similar. The vessel may only experience an earthquake several times during the life of the equipment, though the vessel must be designed to withstand any seismic event.

Designing a vessel to be invulnerable to any earthquake would be both impractical and uneconomical. Building codes and design standards use the ability of the structure to yield and absorb energy in a ductile manner during a seismic event for design. This is part of the basis of the 'R' factor, which is used to reduce the design strength for a structure. As a result of the designed structure undergoing permanent deformation during an earthquake, some of the structure may be lightly or severely damaged. In the case of vessel support design, it should be understood that the anchor bolts provide a benefit to the vessel by yielding and absorbing energy that could otherwise have a greater impact on the support members.

Period of Vibration

Vessels will vibrate based on an exciting force such as wind or earthquake. There are two distinct types of loadings as a result of wind. The first is the static force from wind loading pressure against the vessel shell. The second is a dynamic effect from vortex shedding due to wind flow around the vessel. Tall, slender, vertical vessels are more susceptible to the effects of vortex shedding.

Vessels subject to an external force or ground motion will deflect to a specific shape and then return to its original position once the applied force is dissipated or removed. The extent of deflection is proportional to the applied force. The vessel, or its support, will act as a spring. In the passage to equilibrium, the vessel will vibrate freely, through its various modes. The period of vibration (POV) is the time it takes the vessel to deflect through one mode and return to its original position and is measured in seconds. The frequency, which is the inverse of POV, is the number of cycles per second.

The POV of a pressure vessel is a function of the vessel weight, diameter, height, weight distribution, temperature, flexibility, type of support, damping mechanisms and location if supported in a structure. Typically when we are discussing the period of vibration for a vessel we are talking about the "first" period of vibration, or the first "natural" or "fundamental" period of vibration.

All vibrating systems, of which vessels are included, have multiple modes of vibration, known as the first mode, the second mode, etc. Each individual mode will have its own unique characteristics for that particular system. The deflected shape of a vessel for any single mode of vibration is always the same for that vessel, regardless of the magnitude. In other words, though the amplitude of displacement changes with time, the relation between displacements throughout the height remains constant.

The mode with the lowest frequency (longest period) is called the first, or fundamental mode. The mode with the higher frequencies (shorter periods) are called the higher modes. Each mode would have a different POV and frequency.

The period of vibration is the inverse of the frequency of vibration. Typically the symbol for POV is T and is given in seconds. The symbol for frequency is f, and is given in hertz, which is cycles per second. $T = 1/f$ and $f = 1/T$.

Generally, vessels with a POV less than 0.30 seconds ($f \geq 3.33$ Hz) are considered rigid. Vessels with a POV between 0.30 and 0.75 seconds (1.33 Hz $< f <$ 3.33 Hz) are semirigid. Between 0.75 and 1.25 seconds (0.8 Hz $< f <$ 1.33 Hz) are semiflexible and vessels with a POV greater than 1.25 seconds (0.80 Hz) are flexible.

A vessel will have a different POV in the empty and full condition. It will have a different POV for the new and corroded condition. It will have a different POV for hot and cold conditions due to the modulus of elasticity of the steel at temperature. Vertical vessels on legs and skirts are the most flexible. Vessels on lugs and rings are normally supported in structures and therefore would be subject to the harmonics of the structure itself. Horizontal vessels vibrate with their supports as well and are dependent on pier deflection.

A vertical vessel is modeled as a cantilever beam whereas a horizontal vessel is modeled as a simply supported beam. A cantilever is a much more prone to vibration and deflection then a simply supported beam, therefore the POV is typically much higher. Guiding a vessel supported in a structure will greatly alter its POV because it changes the mode of vibration.

Wind and seismic design standards such as ASCE have base shear factors that are a function of the POV. This makes sense because the response of the vessel is

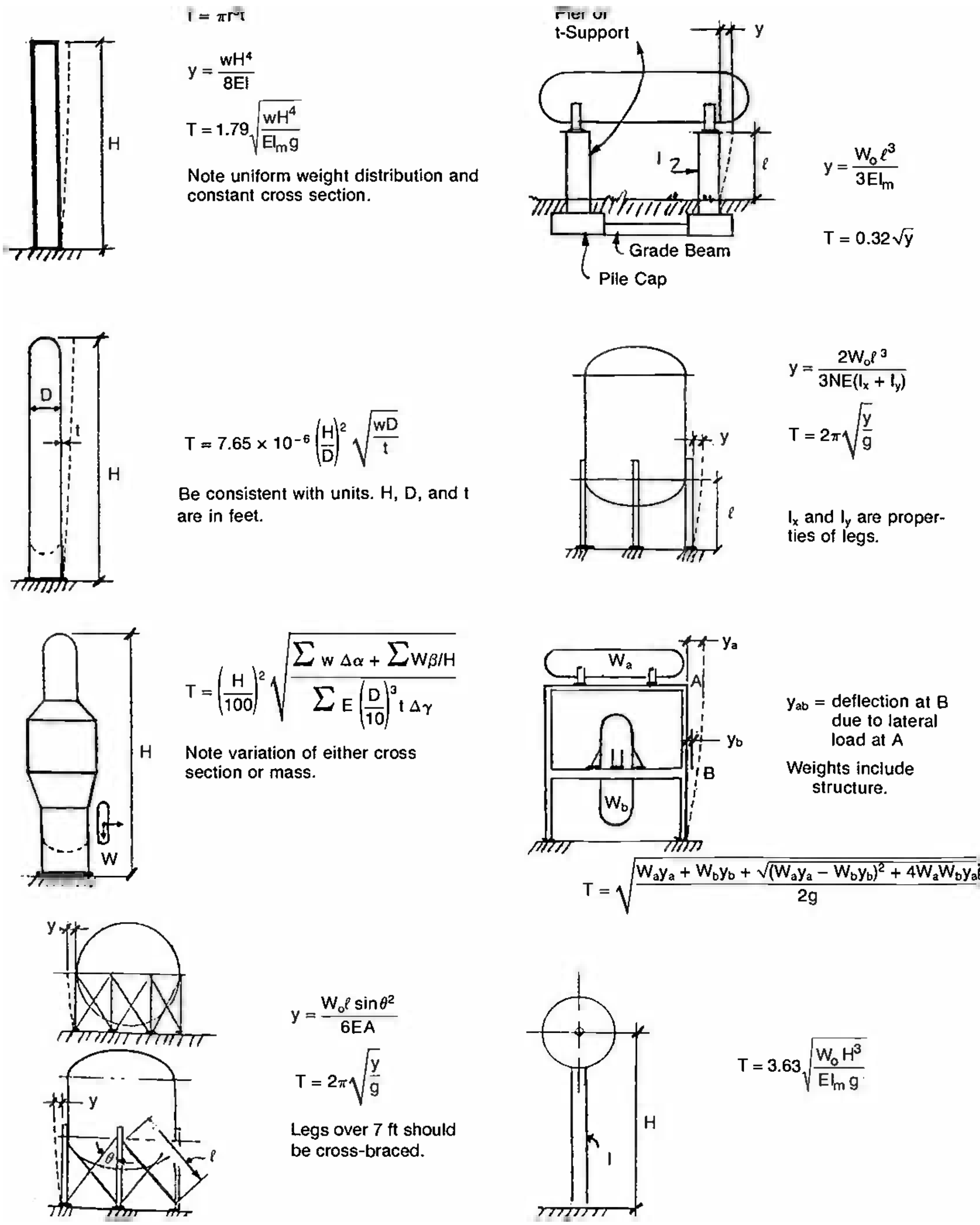
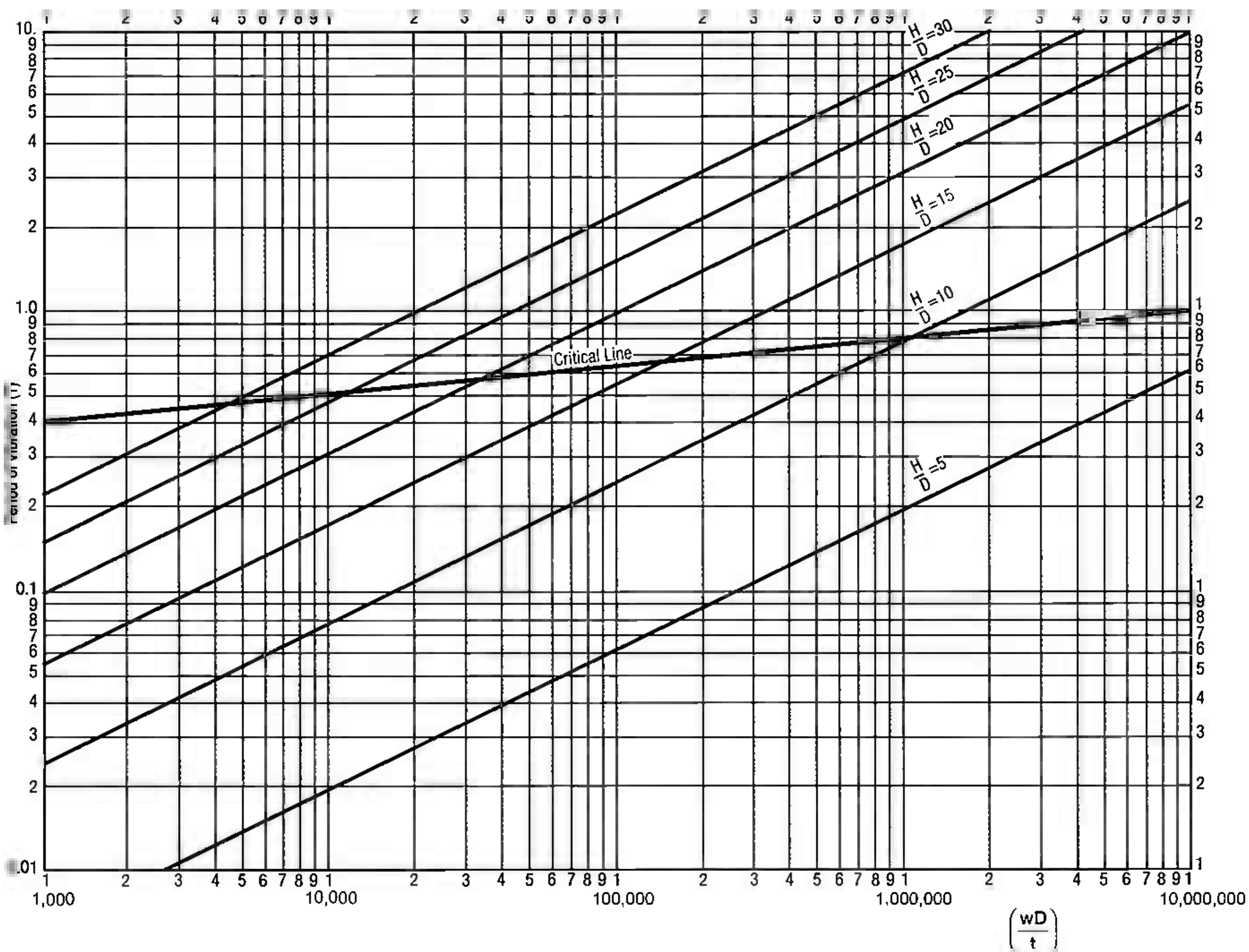


Figure 4-6. Formulas for period of vibration, T, and deflection, y.



General formula for cantilever

$$T = K\sqrt{\frac{wH^4}{EIg}}$$

which for steel cylidrical shell reduces to

$$T = 0.00000765\left(\frac{H}{D}\right)^2\sqrt{\frac{wD}{t}}$$

where T = period, sec
w = weight, lb per ft
H = height, ft
D = diameter of shell, ft
t = thickness of shell, ft

Constant 0.00000765 is based upon
E = modulus of elasticity of steel 30,000,000 lb per sq in
I = moment of inertia of shell area

$$= 3.142\left(\frac{D}{2}\right)^3 t$$

K = 1.79 for fundamental period of vibration

$g = 32.2$ ft per sec^2

Figure 4-7. Period of vibration for cylindrical steel shells. *Reprinted by permission of Fluor Daniel. Inc., Irvine CA.*

dependent on the relative rigidity of the vessel. The more rigid the vessel (lower POV, high frequency) the higher the base shear will be. The more flexible (higher POV, lower frequency) vessels would have a lower base shear.

Notes

1. Vessels mounted in structures at some elevation other than grade generally will experience amplified base motion near and above the natural frequencies of the support structure.
   - *Light vessels* (less than 1% of structure weight):
     a. If vessel frequency > structure frequency, then vessel is subjected to maximum acceleration of the structure.
     b. If vessel frequency < structure frequency, then vessel will not be affected by structure. It will respond as if it were mounted at grade.
   - *Medium vessels* (less than 20% of structure weight): Approximate methods may be used to develop the in-structure response spectra. The method used should account for interaction between vessel and structure (energy feedback). Consideration should be given to account for ductility of the vessel.
   - *Heavy vessels* (single large vessel or multiple large vessels): The vessel(s) is the principal vibrating element. It requires a combined seismic model, which simulates the mass and stiffness properties of vessel and structure.
2. For tall slender vessels, the main concern is bending. For short, squat vessels the main concern is base shear.
3. The procedures outlined in this chapter are static-force procedures, which assume that the entire seismic force due to ground motion is applied instantaneously. This assumption is conservative but greatly simplifies the calculation procedure. In reality earthquakes are time-dependent events and the full force is not realized instantaneously. ASCE/SEI 7 allows, and in some cases requires, that a dynamic analysis be performed in lieu of the static force method. Although much more sophisticated, often the seismic loadings are reduced significantly.

Procedure 4-3: Seismic Design for Vessels [2]

Notation

C_s = seismic response coefficient
E_h = effect of horizontal earthquake-induced forces
E_v = effect of vertical earthquake-induced forces
F_a = short-period site coefficient (at 0.2 second period)
F_v = long-period site coefficient (at 1.0 second period)
g = acceleration due to gravity, ft/sec^2
I_e = the importance factor
R = response modification coefficient
S_a = design spectral acceleration
S_S = mapped maximum considered earthquake (risk-targeted), 5 percent damped, spectral response acceleration parameter at short periods
S_1 = mapped maximum considered earthquake (risk-targeted), 5 percent damped, spectral response acceleration parameter at a period of 1 second
S_{DS} = design, 5 percent damped, spectral response acceleration parameter at short periods
S_{D1} = design, 5 percent damped, spectral response acceleration parameter at a period of 1 second
S_{MS} = the maximum considered earthquake (risk-targeted), 5 percent damped, spectral response acceleration parameter at short periods adjusted for site class effects
S_{M1} = the maximum considered earthquake (risk-targeted), 5 percent damped, spectral response acceleration parameter at a period of 1 second adjusted for site class effects
T = the fundamental period of the structure, seconds
T_L = long-period transition period, seconds (see Figure 4-8)
V = total design lateral force or shear at the base, lbs
W = effective seismic weight of the structure, lbs

Design Procedure

Step 1: Determine the following.
Risk Category from Table 4-5
Importance factor as follows:
1.00 for Risk Category I and II
1.25 for Risk Category III
1.50 for Risk Category IV
Site Class as determined by local soil conditions, see Table 4-6
Site Class D may be used unless a governing authority indicates E or F shall be used.
S_S and S_1 parameters
See http://earthquake.usgs.gov/designmaps
F_a and F_v parameters
See Tables 4-7 and 4-8

Step 2: Calculate S_{MS} and S_{M1}

$$S_{MS} = F_a S_s$$
$$S_{M1} = F_v S_1$$

Step 3: Calculate S_{DS} and S_{D1}

$$S_{DS} = (2/3)^* S_{MS}$$
$$S_{D1} = (2/3)^* S_{M1}$$

Step 4: Calculate S_a to develop a response spectrum

If $T < T_o$, $(T_o = 0.2\, S_{D1}/S_{DS})$,

$$S_a = S_{DS}\left(0.4 + 0.6\frac{T}{T_o}\right),$$

If $T_o \le T \le T_S$, $(T_S = S_{D1}/S_{DS})$, $S_a = S_{DS}$,

If $T_S < T \le T_L$, $S_a = \dfrac{S_{D1}}{T}$,

If $T > T_L$, $S_a = \dfrac{S_{D1}T_L}{T^2}$

Step 5: Determine Seismic Design Category (SDC) from S_{DS} and S_{D1} and use most severe*
SDC A-D are determined from Tables 4-9 and 4-10
SDC E is used where S_1 is greater than or equal to 0.75 for Risk Categories I, II, and III
SDC F is used where S_1 is greater than or equal to 0.75 for Risk Category IV

Step 6: Calculate the vertical seismic load, E_V
$E_v = 0.2 S_{DS} D$, where D is the effect of the dead load

Step 7: Determine the response modification factor*

R = 3	for elevated tanks, vessels, bins or hoppers on symmetrically braced legs (height limits may apply)
R = 2	for elevated tanks, vessels, bins or hoppers on unbraced or asymmetrically braced legs (height limits may apply)
R = 3	for horizontal vessels on welded steel saddle supports
R = 2 (3**)	steel stacks, chimneys, silos, skirt-supported vertical vessels

Step 8: Calculate the seismic response coefficient, C_s

$$C_s = \frac{S_{DS}}{(R/I_e)}, \text{ but need not exceed the following:}$$

$$C_s = \frac{S_{D1}}{T(R/I_e)} \text{ for } T \le T_L,$$

$$C_s = \frac{S_{D1}T_L}{T^2(R/I_e)} \text{ for } T > T_L,$$

and shall not be less than

$$C_s = 0.044 S_{DS} I_e \le 0.01$$

and additionally where $S_1 \ge 0.6$ g

$$C_s = 0.5 S_1/(R/I_e)$$

Step 9: Calculate the seismic base shear, V
$V = C_s W$ and $E_h = V$

* Additional detailing should be addressed for the use of these R factors per ASCE/SEI 7 which include the effects of the solids or fluids stored and their interaction with the structure, p-delta effects, and other requirements within other design standards.

** For these structures, R may be taken as a value of 3, however, additional instructions apply in these cases. If buckling of the support is determined to be the governing mode of failure, or if the structure is in Risk Category IV, then the seismic response coefficient must be determined using a value of $I_e/R = 1.0$ and checked against the critical buckling resistance (safety factor equal to 1.0).

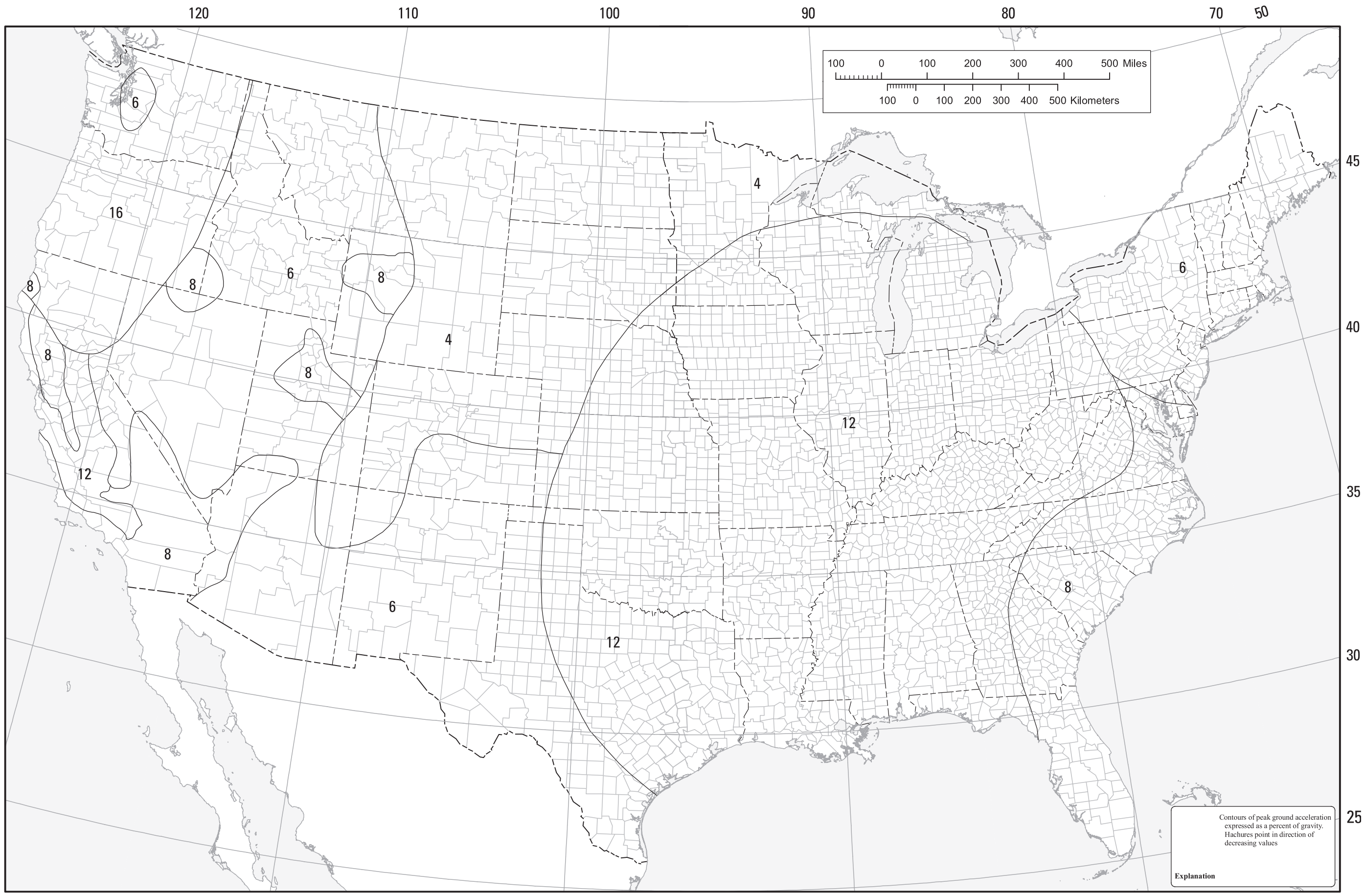


Figure 4-8. Mapped long-period transition period, T_L(s),for the conterminous United States. *With permission from ASCE.*

Table 4-5
Risk category of buildings and other structures for wind and earthquake loads

Use or Occupancy of Buildings and Structures	Risk Category
Buildings and other structures that represent a low risk to human life in the event of failure	I
All buildings and other structures except those listed in Risk Categories I, III, and IV	II
Buildings and other structures, the failure of which could pose a substantial risk to human life.	III
Buildings and other structures, not included in Risk Category IV, with potential to cause a substantial economic impact and/or mass disruption of day-to-day civilian life in the event of failure.	
Buildings and other structures not included in Risk Category IV (including, but not limited to, facilities that manufacture, process, handle, store, use, or dispose of such substances as hazardous fuels, hazardous chemicals, hazardous waste, or explosives) containing toxic or explosive substances where their quantity exceeds a threshold quantity established by the authority having jurisdiction and is sufficient to pose a threat to the public if released.	
Buildings and other structures designated as essential facilities.	IV
Buildings and other structures, the failure of which could pose a substantial hazard to the community.	
Buildings and other structures (including, but not limited to, facilities that manufacture, process, handle, store, use, or dispose of such substances as hazardous fuels, hazardous chemicals, or hazardous waste) containing sufficient quantities of highly toxic substances where the quantity exceeds a threshold quantity established by the authority having jurisdiction to be dangerous to the public if released and is suffi cient to pose a threat to the public if released.[a]	
Buildings and other structures required to maintain the functionality of other Risk Category IV structures.	

[a] Buildings and other structures containing toxic, highly toxic, or explosive substances shall be eligible for classification to a lower Risk Category if it can be demonstrated to the satisfaction of the authority having jurisdiction by a hazard assessment as described in Section 1.5.2 that a release of the substances is commensurate with the risk associated with that Risk Category.
With permission from ASCE

Table 4-6
Site classification

Site Class	$\overline{v}_s$	$\overline{N}$ or $\overline{N}_{cA}$	$\overline{S}_u$
A. Hard rock	>5,000 ft/s	NA	NA
B. Rock	2,500 to 5,000 ft/s	NA	NA
C. Very dense soil and soft rock	1,200 to 2,500 ft/s	>50	>2,000 psf
D. Stiff soil	600 to 1,200 ft/s	15 to 50	1,000 to 2,000 psf
E. Soft clay soil	<600 ft/s	<15	<1,000 psf
	Any profile with more than 10 ft of soil having the following characteristics: — Plasticity index $PI > 20$, — Moisture content $w \geq 40\%$, — Undrained shear strength $\overline{S}_u < 500$ psf		
F. Soils requiring site response analysis in accordance with Section 21.1	See Section 20.3.1 of ASCE/SEI 7.		

For SI: 1 ft/s = 0.3048 m/s; 1 lb/ft^2 = 0.0479 kN/m^2.
With permission from ASCE

Table 4-7
Site coefficient, F_a

	Mapped Risk-Targeted Maximum		Considered Earthquake (MCER) Parameter at Short Period	Spectral Response Acceleration	
Site Class	$S_S \leq 0.25$	$S_S =$ 0.5	$S_S = 0.75$	$S_S =$ 1.0	$S_S \geq$ 1.25
A	0.8	0.8	0.8	0.8	0.8
B	1.0	1.0	1.0	1.0	1.0
C	1.2	1.2	1.1	1.0	1.0
D	1.6	1.4	1.2	1.1	1.0
E	2.5	1.7	1.2	0.9	0.9
F	See Section 11.4.7				

Note: Use straight-line interpolation for intermediate values of S_S.
With permission from ASCE

Table 4-8
Site coefficient, F_v

	Mapped Risk-Targeted Maximum	Considered Earthquake (MCER) Parameter at 1-s Period		Spectral Response Acceleration	
Site Class	$S_1 \leq 0.1$	$S_1 =$ 0.2	$S_1 = 0.3$	$S_1 =$ 0.4	$S_1 \geq$ 0.5
A	0.8	0.8	0.8	0.8	0.8
B	1.0	1.0	1.0	1.0	1.0
C	1.7	1.6	1.5	1.4	1.3
D	2.4	2.0	1.8	1.6	1.5
E	3.5	3.2	2.8	2.4	2.4
F	See Section 11.4.7				

Note: Use straight-line interpolation for intermediate values of S_1.
With permission from ASCE

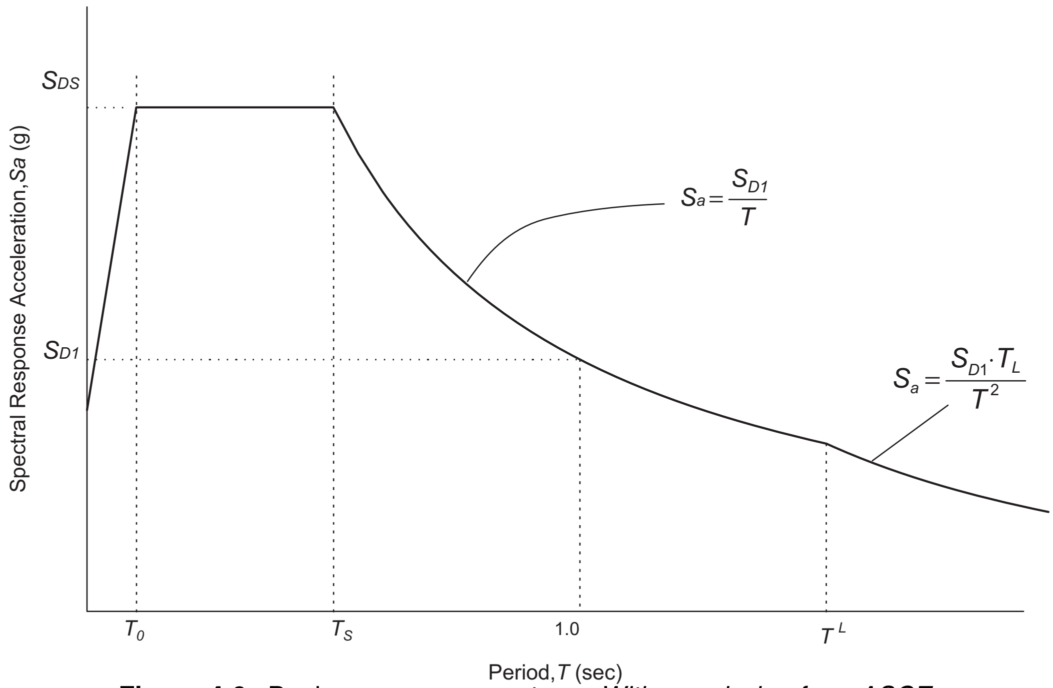


Figure 4-9. Design response spectrum. *With permission from ASCE*.

Table 4-9
SDC Based on S_{DS}

Value of S_{DS}	Risk Category: I or II or III	IV
$S_{DS} < 0.167$	A	A
$0.167 \le S_{DS} < 0.33$	B	C
$0.33 \le S_{DS} < 0.50$	C	D
$0.50 \le S_{DS}$	D	D

With permission from ASCE

Table 4-10
SDC Based on S_{D1}

Value of S_{D1}	Risk Category: I or II or III	IV
$S_{D1} < 0.067$	A	A
$0.067 \le S_{D1} < 0.133$	B	C
$0.133 \le S_{D1} < 0.20$	C	D
$0.20 \le S_{D1}$	D	D

With permission from ASCE

Procedure 4-4: Seismic Design – Vessel on Unbraced Legs [4–7]

Notation

A = cross-sectional area, leg, in.^{2}
V = base shear, lb
W = operating weight, lb
n = number of legs
C_v = vertical seismic factor
C_h = horizontal seismic factor
y = static deflection, in.
F_v = vertical seismic force, lb
F_h = horizontal seismic force, lb
F_a = allowable axial stress, psi
F_D = axial load due to dead weight, lb
F_L = axial load due to seismic or wind, lb
F_b = allowable bending stress, psi
F'_e = Euler stress divided by safety factor, psi
f_1 = maximum eccentric load, lb
V_n = horizontal load on leg, lb
F_n = maximum axial load, lb
f_a = axial stress, psi
f_b = bending stress, psi
E = modulus of elasticity, psi
g = acceleration due to gravity, 386 in./sec^2
e = eccentricity of legs, in.

M_b = overturning moment at base, in.-lb
M_t = overturning moment at tangent line, in.-lb
M = bending moment in leg, in.-lb
ΣI_1 = summation of moments of inertias of all legs perpendicular to F_h, in.^{4}
ΣI_2 = summation of moments of inertia of one leg perpendicular to F_h, in.^{4}
I = moment of inertia of one leg perpendicular to F_h, in.^{4}
C_1 = distance from centroid to extreme fiber, in.
C_m = coefficient, 0.85 for compact members
K_1 = end connection coefficient, 1.5-2.0
T = period of vibration, sec
r = least radius of gyration, in.

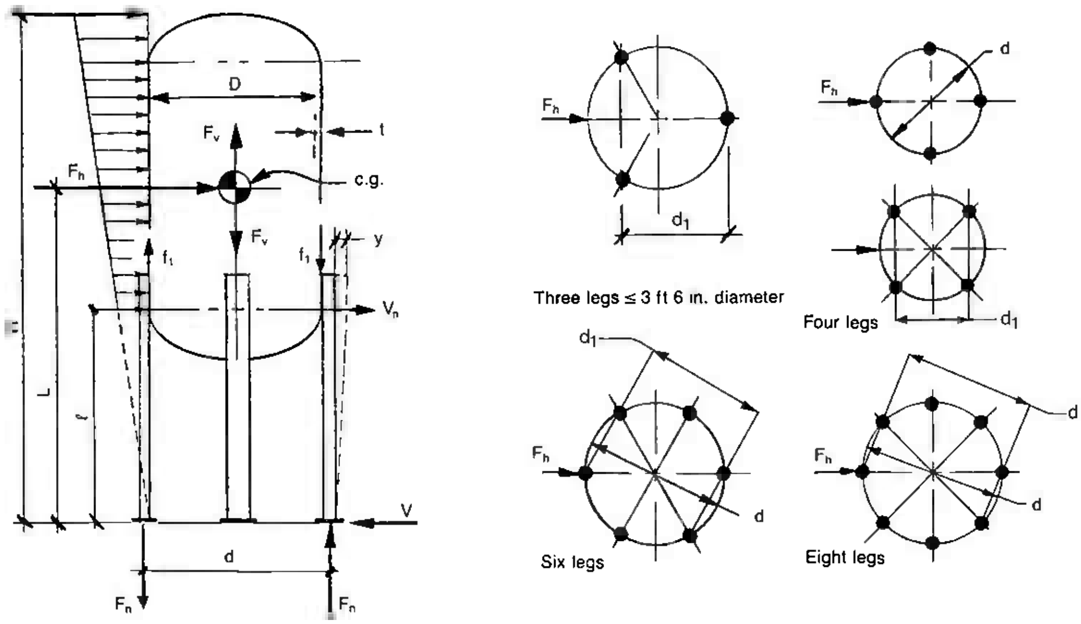


Figure 4-10. Typical dimensional data and forces for a vessel supported on unbraced legs.

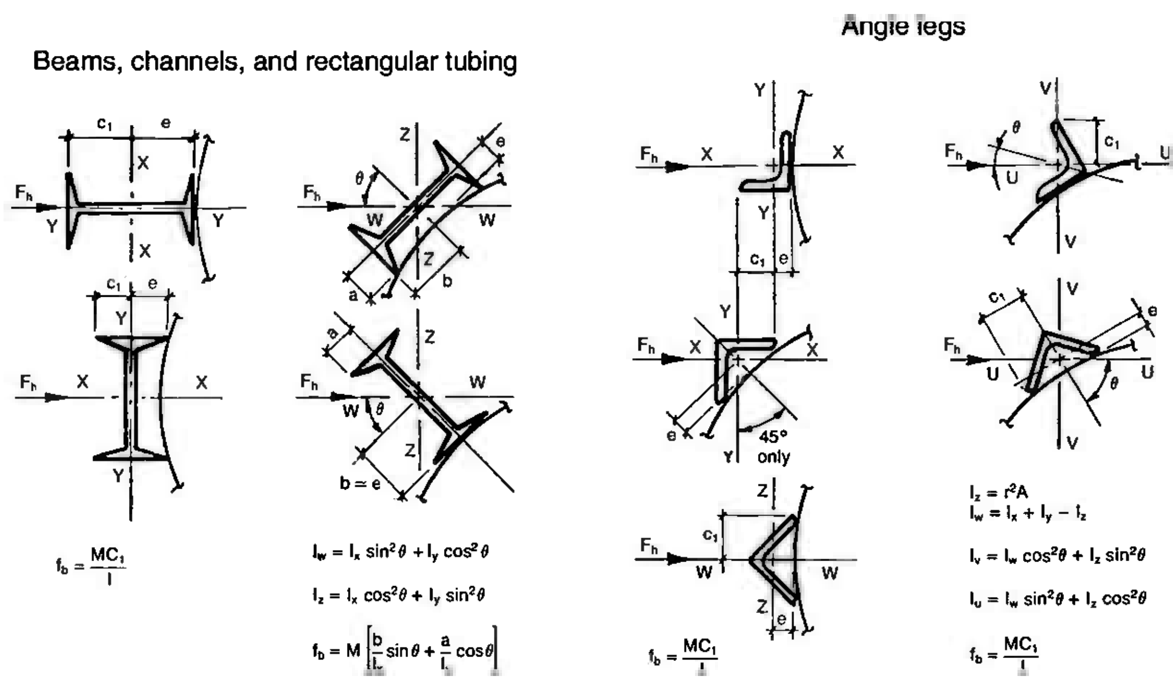


Figure 4-11. Various leg configurations.

Calculations

The following information is needed to complete the leg calculations:

No. ________	I_u = ________
Size ________	I_v = ________
A = ________	ΣI_1 = ________
r = ________	ΣI_2 = ________
I_x = ________	$K_1\ell/r$ = ________
I_y = ________	F_a = ________
I_z = ________	
I_w = ________	

- *Deflection, y, in.*

$$y = \frac{2\,W\,\ell^3}{3nE\sum I_2}$$

Note: Limit deflection to 6 in. per 100 ft or equivalent proportion. This calculation is based on the assumption that the support legs are pinned at the base and fixed at the vessel, and that the vessel is significantly more rigid than the supported legs.

- *Period of vibration, T, sec.*

$$T = 2\pi\sqrt{\frac{y}{g}}$$

- *Base shear, V, lb.*
  See Procedure 4-3.
- *Horizontal force at c.g. of vessel, F_h, lb.*

$$F_h = C_h W$$

- *Vertical force at c.g. of vessel, F_v, lb.*

$$\text{Downward: } (-)F_v = W$$

$$\text{or } (1 + C_v)W$$

$$\text{Upward: } (+)F_v = (C_v - 1)W$$

if vertical seismic is greater than 1.0

- *Overturning moment at base, in.-lb.*

$$M_b = LF_h$$

Note: Include piping moments if applicable.

- *Overturning moment at bottom tangent line, in.-lb.*

$$M_t = (L - \ell)F_h$$

- *Maximum eccentric load, lb.*

$$f_1 = \frac{-F_v}{n} - \frac{4M_t}{nD}$$

Note: f_1 is not considered in leg bending stress if legs are not eccentrically loaded.

- *Horizontal load distribution, V_n* (See Figure 4-12).
  The horizontal load on any one given leg, V_n, is proportional to the stiffness of that one leg perpendicular to the applied force relative to the stiffness of the other legs. The greater loads will go to the stiffer legs. Thus, the general equation:

$$V_n = \frac{VI}{\sum I_1} \text{ and } \sum V_n = V$$

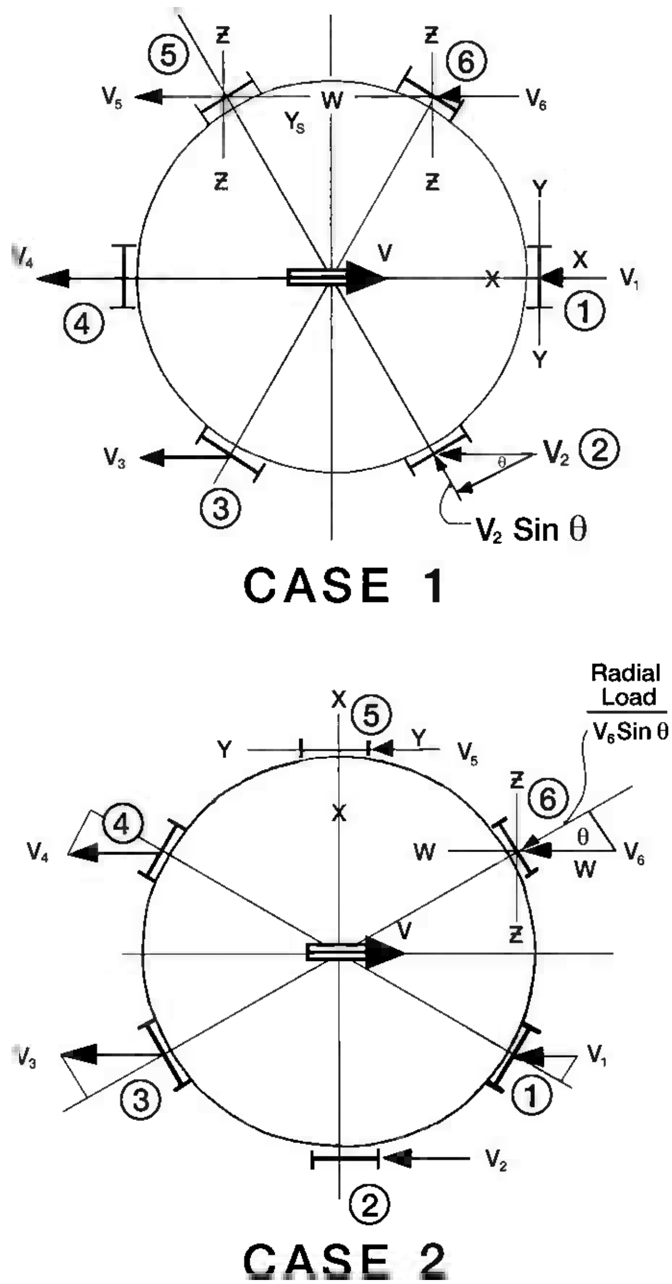


Figure 4-12. Load diagrams for horizontal load distribution.

- *Vertical load distribution, F_n* (See Figure 4-13).

  The vertical load distribution on braced and unbraced legs is identical. The force on any one leg is equal to the dead load plus the greater of seismic or wind and the angle of that leg to the direction of force, V.
- *Bending moment in leg, M, in.-lb.*

  $M = f_1 e \pm V_n \ell$
- *Axial stress in leg, f_a, psi.*

  $$f_a = \frac{F_n}{A}$$
- *Bending stress in leg, f_b, psi.*

  $f_b =$

  Select appropriate formula from Figure 4-11.

For Case 1	For Case 2
$F_D = \frac{F_v}{n}$	$F_D = \frac{F_v}{n}$
$F_L = \frac{4M}{nd}$	$F_L = \frac{4Md_1}{nd^2}$
$F_n = F_D \pm F_L \cos\phi_n$	
$F_n = F_D \pm F_L \cos\phi_n$	

- *Combined stress.*

  If $\frac{f_a}{F_a} \le 0.15$, then $\frac{f_a}{F_a} + \frac{f_b}{F_b} < 1$

  If $\frac{f_a}{F_a} > 0.15$, then $\frac{f_a}{F_a} + \frac{C_m f_b}{\left[1 - \frac{f_a}{F'_e}\right] F_b} < 1$

  where $C_m = 0.85$

  $$F_e{}' = \frac{12\pi^2 E}{23\left(\frac{k_1 \ell}{r}\right)^2}$$
- *Maximum compressive stress in shell, f_c, psi (See Figure 4-16).*

  $L_1 = W + 2\sqrt{(Rt)}$

  Above leg:

  $$f_c = \frac{f_1}{L_1 t}$$

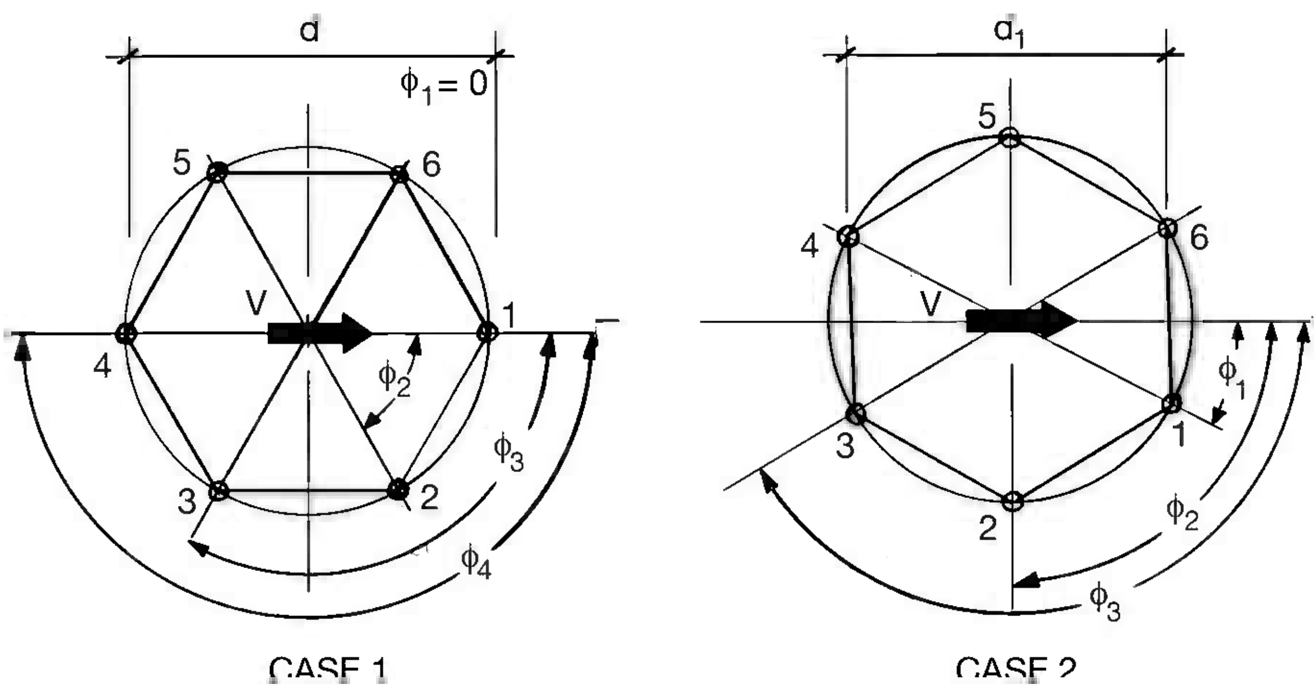


Figure 4-13. Load diagrams for vertical load distribution.

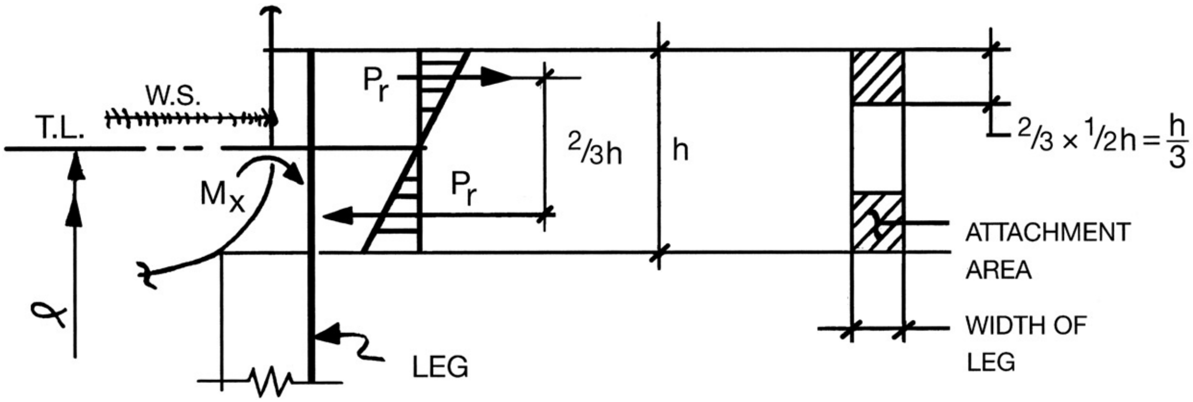


Figure 4-14. Application of local loads in head and shell.

General:

$$f_c = (-)\frac{F_v}{\pi Dt} - \frac{4M_t}{\pi D^2 t}$$

F_c = allowable compressive stress is factor "B" from ASME Code.

$$\text{Factor "A"} = \frac{0.125t}{R}$$

"B" = from applicable material chart of ASME Code, Section II, Part D, Subpart 3.

- *Shear load in welds attaching legs.*

$$\frac{f_1}{2h} = \frac{\text{lb}}{\text{in. of weld}}$$

See Table 4-14 for allowable loads on fillet welds in shear.

- *Local load in shell (See Figure 4-14).*

For unbraced designs, the shell or shell/head section to which the leg is attached shall be analyzed for local loading due to bending moment on leg.

$$M_x = V_n \ell \sin\theta$$

- *Anchor bolts.* If the weight $W > 4\ M_b/d$, then no uplift occurs and anchor bolts should be made a minimum of ¾ in. in diameter. If uplift occurs, then the cross-sectional area of the bolt required for tension alone would be:

$$A_b = \frac{f_2}{S_t}\ \text{in.}^2$$

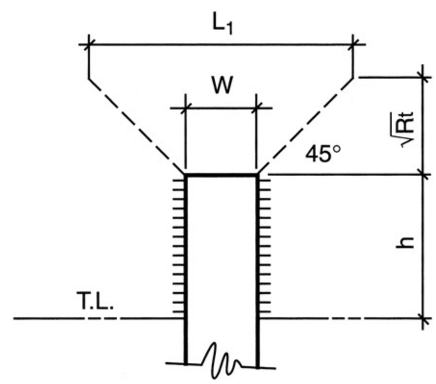


Figure 4-15. Dimensions of leg attachment.

where A_b = area of bolt required
f_2 = axial tension load
S_t = allowable stress in tension

Notes

1. Legs longer than 7 ft should be cross-braced.
2. Do not use legs to support vessels where high vibration, shock, or cyclic service is anticipated.
3. Select legs that give maximum strength for minimum weight for most efficient design. These sections will also distribute local loads over a larger portion of the shell.
4. Legs may be made of pipe, channel, angle, rectangular tubing, or beam sections.
5. This procedure assumes a one-mass bending structure which is not technically correct for tall vessels. Tall towers would have distributed masses and should be designed independently of support structure, i.e., legs.

Table 4-11
Vertical loads on legs, F_n

Quantity of Legs	Leg No.	Case 1	Case 2
6	1	$F_D + 1.000\ F_L$	$F_D + 0.866\ F_L$
	2	$F_D + 0.500\ F_L$	F_D
	3	$F_D - 0.500\ F_L$	$F_D - 0.866\ F_L$
	4	$F_D - 1.000\ F_L$	$F_D - 0.866\ F_L$
	5	$F_D - 0.500\ F_L$	F_D
	6	$F_D + 0.500\ F_L$	$F_D + 0.866\ F_L$
8	1	$F_D + 1.000\ F_L$	$F_D + 0.923\ F_L$
	2	$F_D + 0.707\ F_L$	$F_D + 0.382\ F_L$
	3	F_D	$F_D - 0.382$
	4	$F_D - 0.707\ F_L$	$F_D - 0.923\ F_L$
	5	$F_D - 1.000\ F_L$	$F_D - 0.923\ F_L$
	6	$F_D - 0.707\ F_L$	$F_D - 0.382\ F_L$
	7	F_D	$F_D + 0.382$
	8	$F_D + 0.707\ F_L$	$F_D + 0.923\ F_L$
10	1	$F_D + 1.000\ F_L$	$F_D + 0.951\ F_L$
	2	$F_D + 0.809\ F_L$	$F_D + 0.587\ F_L$
	3	$F_D + 0.309\ F_L$	F_D
	4	$F_D - 0.309\ F_L$	$F_D - 0.809\ F_L$
	5	$F_D - 0.809\ F_L$	$F_D - 0.951\ F_L$
	6	$F_D - 1.000\ F_L$	$F_D - 0.951\ F_L$
	7	$F_D - 0.809\ F_L$	$F_D - 0.587\ F_L$
	8	$F_D - 0.309\ F_L$	F_D
	9	$F_D + 0.309\ F_L$	$F_D + 0.587\ F_L$
	10	$F_D + 0.809\ F_L$	$F_D + 0.951\ F_L$
12	1	$F_D + 1.000\ F_L$	$F_D + 0.965\ F_L$
	2	$F_D + 0.866\ F_L$	$F_D + 0.707\ F_L$
	3	$F_D + 0.500\ F_L$	$F_D + 0.258\ F_L$
	4	F_D	$F_D - 0.258\ F_L$
	5	$F_D - 0.500\ F_L$	$F_D - 0.707\ F_L$
	6	$F_D - 0.866\ F_L$	$F_D - 0.965\ F_L$
	7	$F_D - 1.000\ F_L$	$F_D - 0.965\ F_L$
	8	$F_D - 0.866\ F_L$	$F_D - 0.707\ F_L$
	9	$F_D - 0.500\ F_L$	$F_D - 0.258\ F_L$
	10	F_D	$F_D + 0.258\ F_L$
	11	$F_D + 0.500\ F_L$	$F_D + 0.707\ F_L$
	12	$F_D + 0.866\ F_L$	$F_D + 0.965\ F_L$
16	1	$F_D + 1.000\ F_L$	$F_D + 0.980\ F_L$
	2	$F_D + 0.923\ F_L$	$F_D + 0.831\ F_L$
	3	$F_D + 0.707\ F_L$	$F_D + 0.555\ F_L$
	4	$F_D + 0.382\ F_L$	$F_D + 0.195\ F_L$
	5	F_D	$F_D - 0.195\ F_L$
	6	$F_D - 0.382\ F_L$	$F_D - 0.555\ F_L$
	7	$F_D - 0.707\ F_L$	$F_D - 0.831\ F_L$
	8	$F_D - 0.923\ F_L$	$F_D - 0.980\ F_L$
	9	$F_D - 1.000\ F_L$	$F_D - 0.980\ F_L$
	10	$F_D - 0.923\ F_L$	$F_D - 0.831\ F_L$
	11	$F_D - 0.707\ F_L$	$F_D - 0.555\ F_L$
	12	$F_D - 0.382\ F_L$	$F_D - 0.195\ F_L$
	13	F_D	$F_D + 0.195\ F_L$
	14	$F_D + 0.382\ F_L$	$F_D + 0.555\ F_L$
	15	$F_D + 0.707\ F_L$	$F_D + 0.831\ F_L$
	16	$F_D + 0.923\ F_L$	$F_D + 0.980\ F_L$

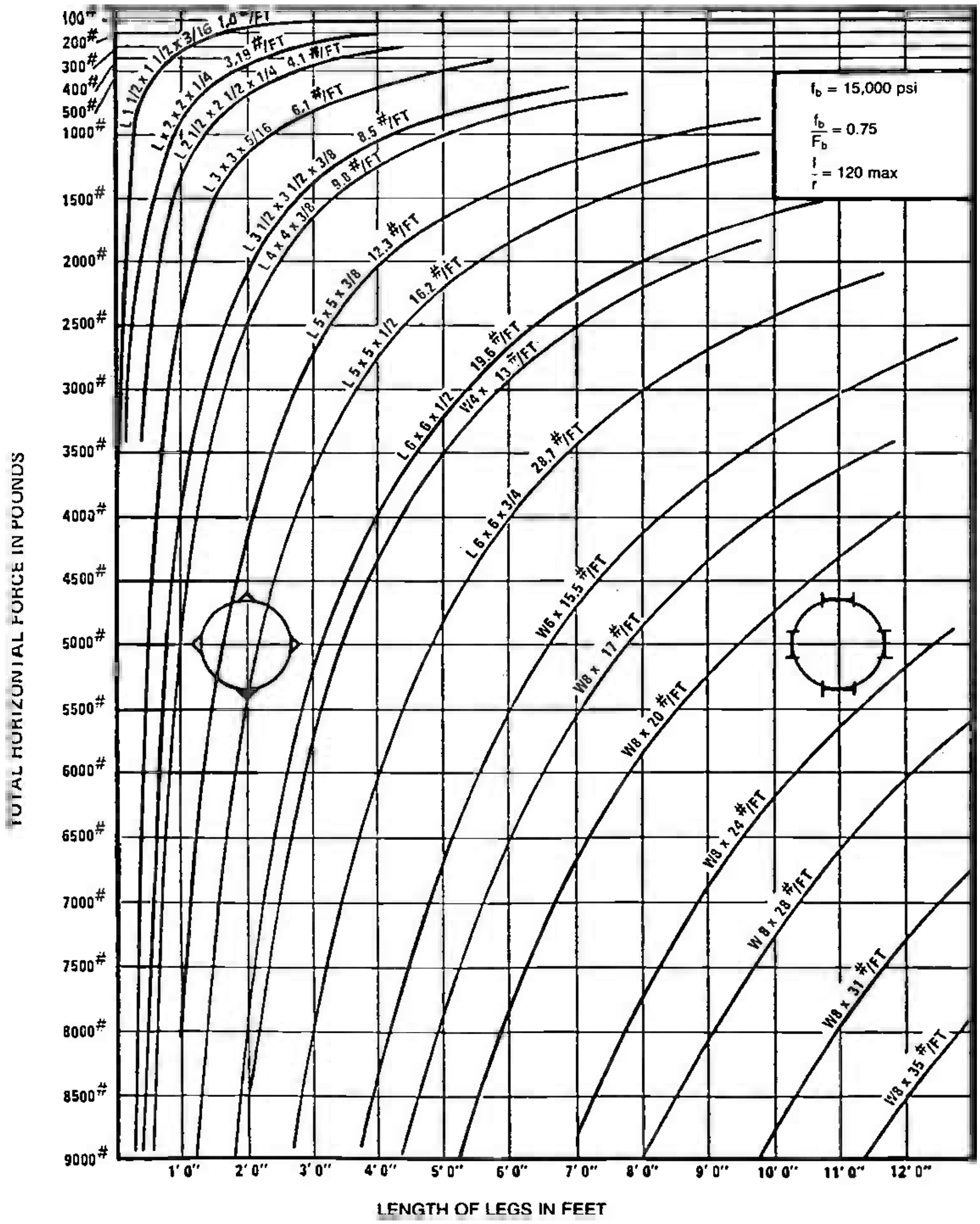
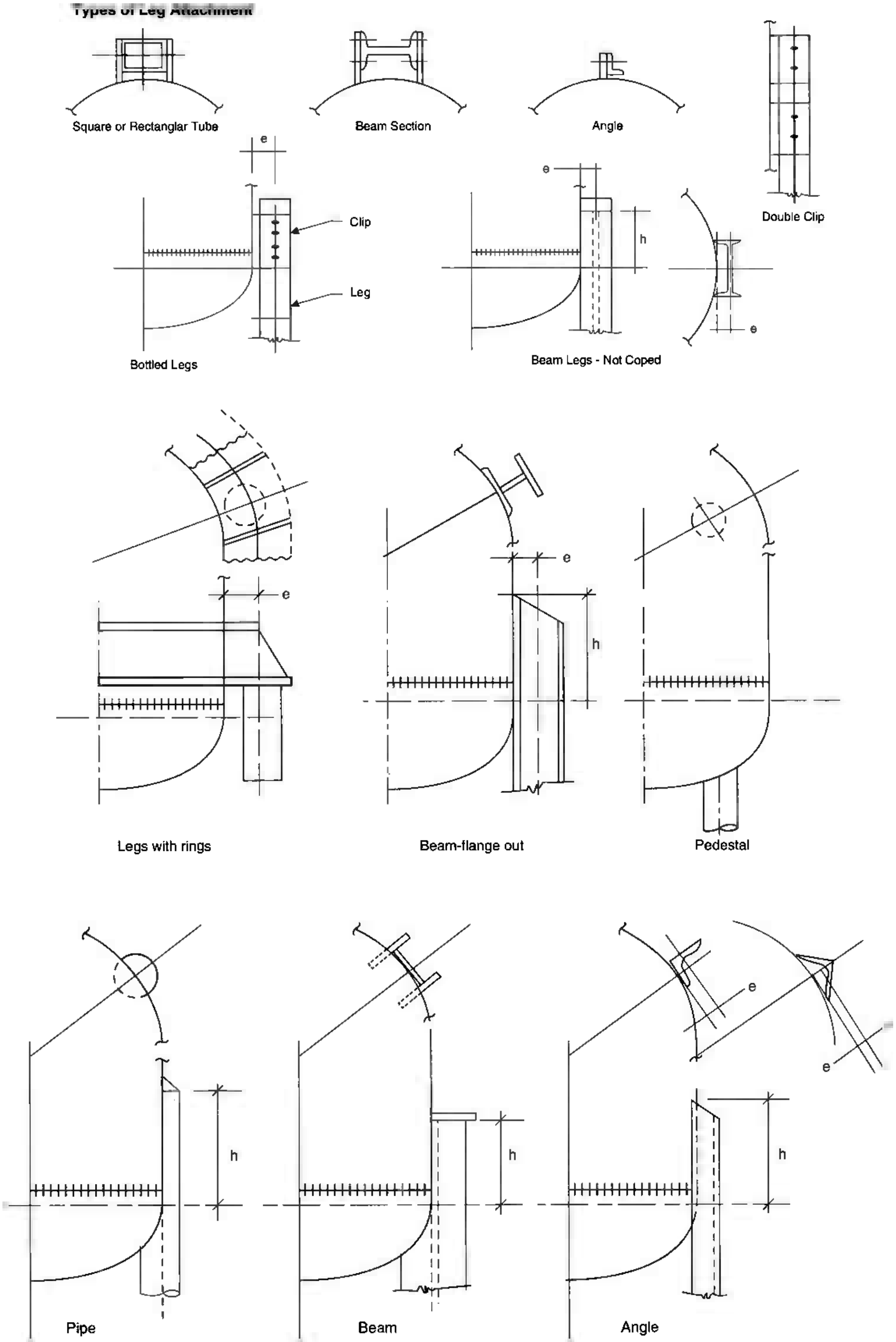


Figure 4-16. Leg sizing chart for vessel supported on four legs.

Types of Leg Attachment



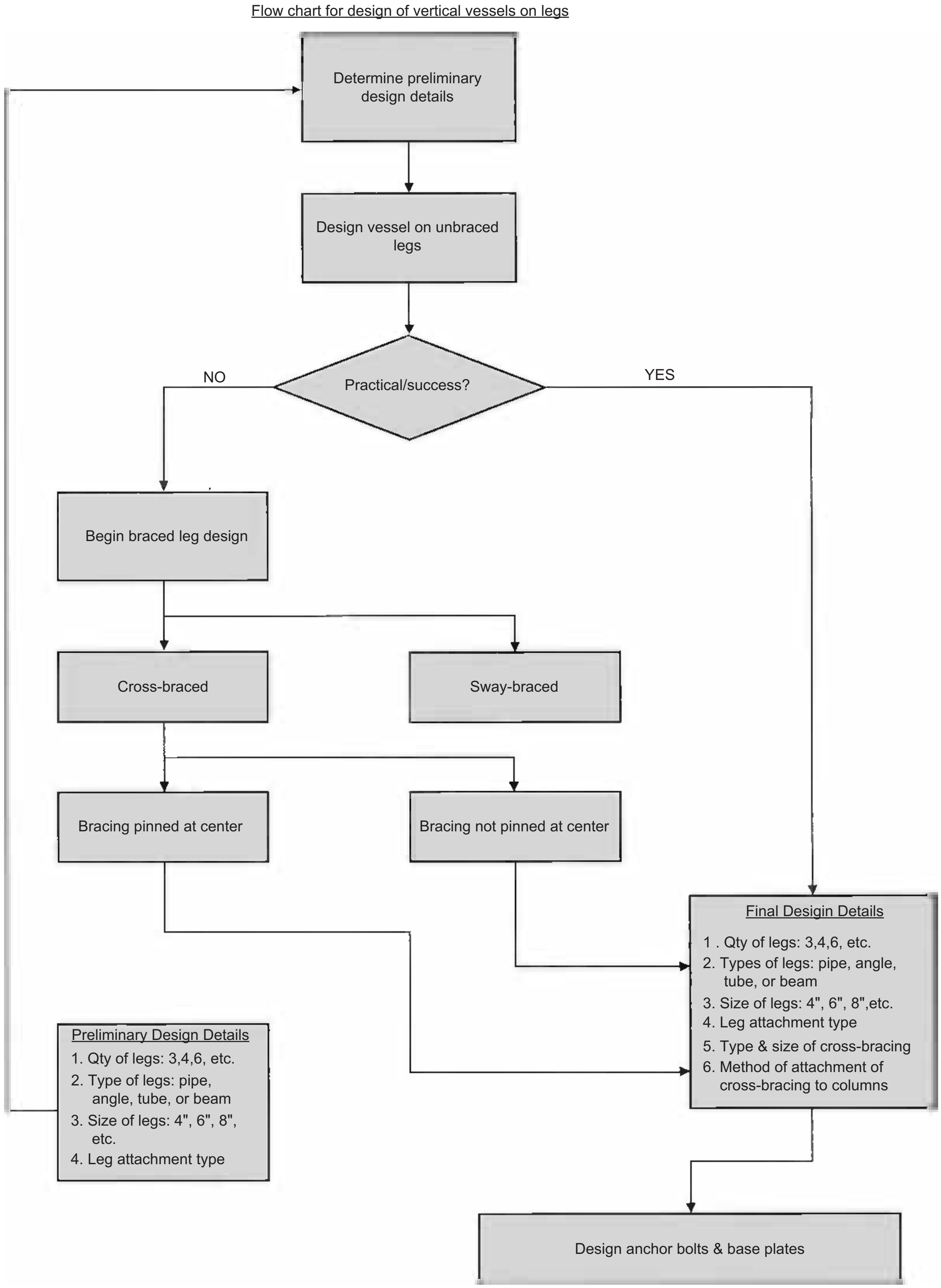


Figure 4-17. Flow chart for design of vertical vessels on legs.

Procedure 4-5: Seismic Design – Vessel on Braced Legs

Notation

A_b = Area, brace, in^2
A_c = Area, column, in^2
A_{br} = Area, brace, required, in^2
C_a = Corrosion allowance, in
D_c = Centerline diameter of columns, in
E = Modulus of elasticity, psi
f = Maximum force in brace, Lbs
f_a = Axial stress, compression, psi
f_t = Tension stress, psi
F_a = Allowable axial stress, psi
F_b = Allowable stress, bending, psi
F_c = Allowable stress, compression, psi
F_D = Axial load on column due to dead weight, lbs
F_h = Horizontal seismic force, Lbs
F_L = Axial load on column due to seismic or wind, lbs
F_t = Allowable stress, tension, psi
F_V = Vertical seismic force, Lbs
F_y = Yield strength of material at temperature, psi
g = Acceleration due to gravity, 386 in/sec^2
I_b = Moment of inertia, bracing, in^4
I_r = Required moment of inertia, in^4
I_c = Moment of inertia, column, in^4
k = End connection coefficient, columns
M_o = Overturning moment, in-Lbs
N = Number of columns
n = Number of active rods per panel use 1 for sway bracing; 2 for cross bracing
n′ = Factor for cross bracing, use 1 for unpinned and 2 for pinned at center
Q = Maximum axial force in column, Lbs
r_b = Radius of gyration, brace, in
r_c = Radius of gyration, column, in
S_r = Slenderness ratio
T = Period of vibration, seconds
V = Base shear, Lbs
V_n = Horizontal force per column, Lbs
W_o = Weight, operating, Lbs
w = Unit weight of liquid, pcf
ΔL = Change in length of brace, in
δ = Lateral deflection of vessel, in

Horizontal Load Distribution, V_n

The horizontal load on any one leg is dependent on the direction of the leg bracing. The horizontal force, V, is transmitted to the legs through the bracing. Thus, the general equation:

Table 4-12
Dimensions for d_1

No. of Legs	d_1
3	.750 D_C
4	.707 D_C
6	.866 D_C
8	.924 D_C
10	.951 D_C
12	.966 D_C
16	.981 D_C

$$V_n = \frac{V\left(\sin^2 \alpha_{n-1} + \sin^2 \alpha_n\right)}{N} \quad \text{and} \quad \Sigma V_n = V$$

Vertical Load Distribution, F_n

The vertical load distribution on braced and unbraced legs is identical. The force on any one leg is equal to the dead load plus the greater of seismic or wind and the angle of that leg to the direction of force, V. The general equation for each case is as follows:

For Case 1:

$$F_D = \frac{F_v}{N}$$

$$F_L = \frac{4M}{Nd}$$

$$F_n = F_D \pm F_L \cos \phi_n$$

For Case 2:

$$F_D = \frac{F_v}{N}$$

$$F_L = \frac{4Md_1}{Nd^2}$$

$$F_n = F_D \pm F_L \cos \phi_n$$

Design of Columns

- Base Shear,V
  Use worst case of wind or seismic
  V = ________
- Overturning Moment, M_o

  $M_o = L\,V$
- Maximum Dead load, F_D

  $F_D = (-)W_o/N$
- Maximum Earthquake/Wind Load, $F_{E/W}$

  $F_L = +/-\,4\,M_o/N\,D_c$
- Maximum Column Load, Q
  Select worst case from Table or use;

$Q = F_D +/-\ F_L$

Q max compression $= Q_C =$

Q max tension $= Q_T =$

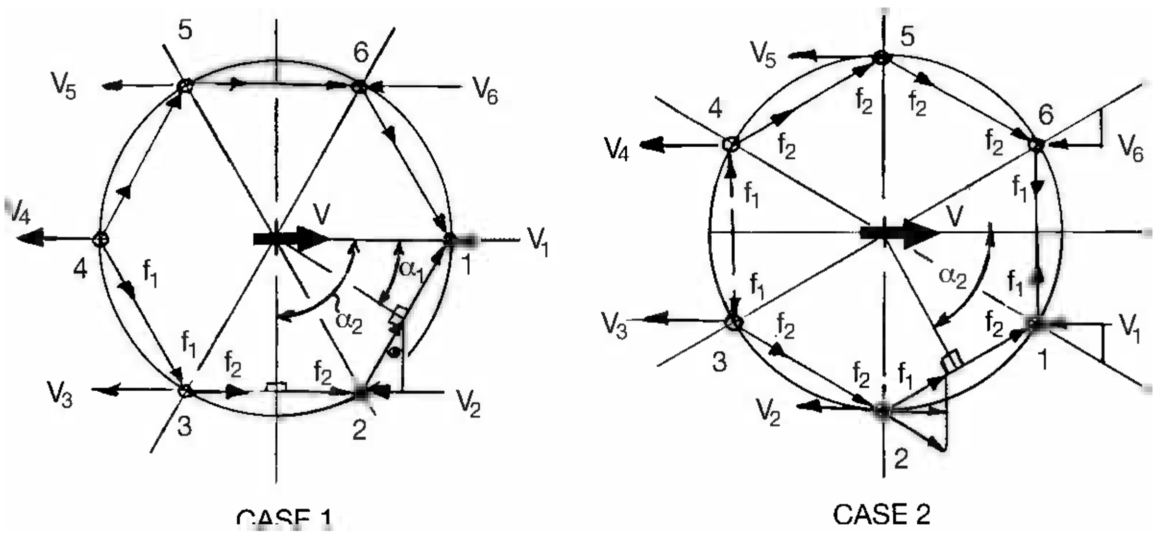
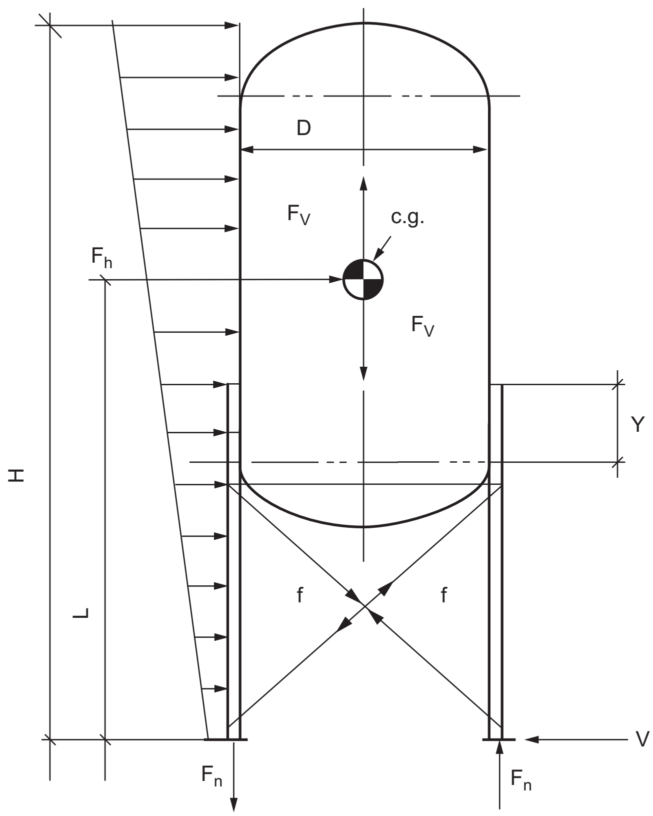


Figure 4-18. Load diagrams for horizontal load distribution.



Four legs (for illustration only)

Note: If there is no uplift then there is no tension force.

- Leg selection;

  Use: = __________

  A_C = __________

Compression Case

- Compressive stress, f_a

  $f_a = Q_C/A_C \leq F_a$

- Slenderness ratio, $S_r = k_h/r_c$
  $F_a =$

Tension Case

- Tension stress, f_t

  $f_t = Q_T/A_C \leq F_t$

- Allowable tension stress, F_t,

  $F_T = 1.2(0.6)F_y$

Cross Bracing

Note: Loads in cross bracing are tension and compression.

Compression Case

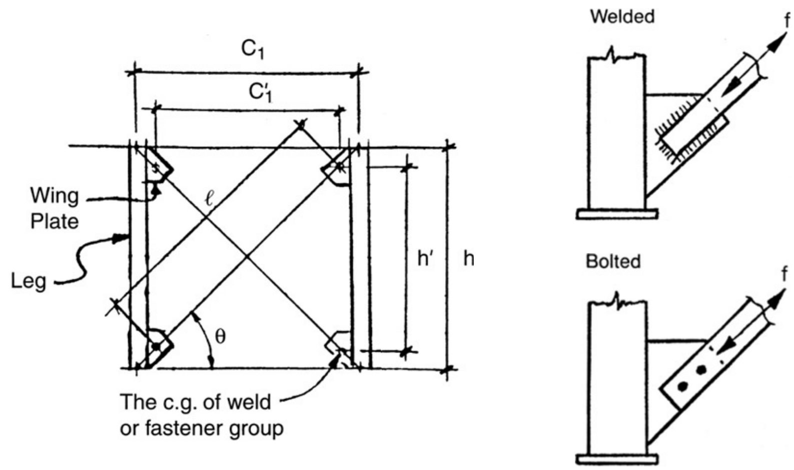
- Case 1: Pinned at center

  $I_r = FL_1{}^2/4\pi^2E$

  Case 2: Not pinned at center

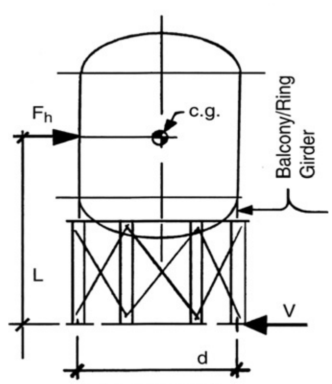
  $I_r = FL_1{}^2/\pi^2E$

ESTABLISH LEG & BRACE DIMENSIONS

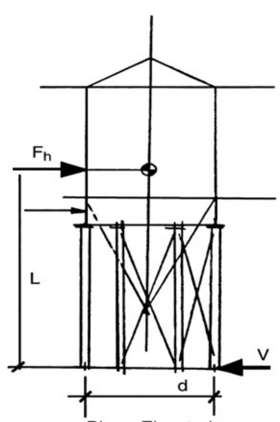


DIMENSIONS:

$\phi = 360 / N =$	$h =$
$R_c =$	$h' = h - 2 a_2 =$
$E = 2 [\sin .5 \phi (R_c)] =$	Find Angle θ ;
$E =$	$\tan \theta = E' / h' =$
$a_1 =$	$\theta =$
$a_2 =$	Find Length of Brace, L_1
$E' = E - 2 a_1 =$	$L_1 = E' / \sin \theta =$



Large Pressure Vessel



Bin or Elevated Tank

VESSEL ON BRACED LEGS - SEISMIC DESIGN			
	METHOD 1	**METHOD 2**	**METHOD 3**
V_n = Horiz shear per lug	Worst case from Table dependent on number of legs and direction of seismic force (between legs or through legs).	NA	$V_n = V/N$
f = Max force in brace	$f = V_n / n \sin\theta$	$f = 2 W_o / 2 N \sin\theta$	$f = V_n / \sin\theta$
ΔL = Change in length of brace	$\Delta L = (f L_1) / (E A_b)$	$\Delta L = (f L_1) / (E A_b)$	$\Delta L = (2 W_o L_1) / (2 N E A_b \sin\theta)$
δ = Lateral deflection of Vessel	$\delta = \Delta L / \sin\theta$	$\delta = \Delta L / \sin\theta$	$\delta = \Delta L / \sin\theta$
T = Period of vibration	$T = 2\pi(\delta / g)^{1/2}$	$T = 2\pi(\delta / g)^{1/2}$	$T = 2\pi(\delta / g)^{1/2}$

NOTES:

1. Approx POV per ASCE 7-05 ; $T_a = C_t h_n^x$

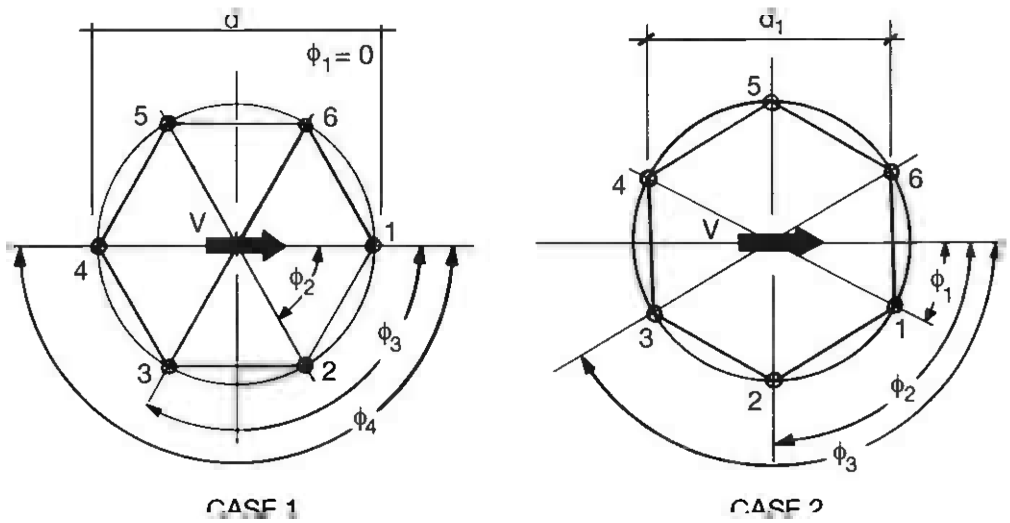


Figure 4-19. Load diagrams for vertical load distribution.

Table 4-13
Summary of loads, forces & moments at support locations

Qty of Columns	Leg No.	Case 1: At Columns		Case 2: Between Columns	
		Horiz (Vn)	Vertical (Q)	Horiz (Vn)	Vertical (Q)
6	1	+ 0.083 V	$F_D + 1.000\ F_L$	+ 0.125 V	$F_D + 0.866\ F_L$
	2	+ 0.208 V	$F_D + 0.500\ F_L$	+ 0.250 V	F_D
	3	+ 0.208 V	$F_D - 0.500\ F_L$	+ 0.125 V	$F_D - 0.866\ F_L$
	4	+ 0.083 V	$F_D - 1.000\ F_L$	+ 0.125 V	$F_D - 0.866\ F_L$
	5	+ 0.208 V	$F_D - 0.500\ F_L$	+ 0.250 V	F_D
	6	+ 0.208 V	$F_D + 0.500\ F_L$	+ 0.125 V	$F_D + 0.866\ F_L$
8	1	+ 0.036 V	$F_D + 1.000\ F_L$	+ 0.062 V	$F_D + 0.923\ F_L$
	2	+ 0.125 V	$F_D + 0.707\ F_L$	+ 0.187 V	$F_D + 0.382\ F_L$
	3	+ 0.213 V	F_D	+ 0.187 V	$F_D - 0.382\ F_L$
	4	+ 0.125 V	$F_D - 0.707\ F_L$	+ 0.062 V	$F_D - 0.923\ F_L$
	5	+ 0.036 V	$F_D - 1.000\ F_L$	+ 0.062 V	$F_D - 0.923\ F_L$
	6	+ 0.125 V	$F_D - 0.707\ F_L$	+ 0.187 V	$F_D - 0.382\ F_L$
	7	+ 0.213 V	F_D	+ 0.187 V	$F_D + 0.382\ F_L$
	8	+ 0.125 V	$F_D + 0.707\ F_L$	+ 0.062 V	$F_D + 0.923\ F_L$
10	1	+ 0.019 V	$F_D + 1.000\ F_L$	+ 0.034 V	$F_D + 0.951\ F_L$
	2	+ 0.075 V	$F_D + 0.809\ F_L$	+ 0.125 V	$F_D + 0.587\ F_L$
	3	+ 0.165 V	$F_D + 0.309\ F_L$	+ 0.180 V	F_D
	4	+ 0.165 V	$F_D - 0.309\ F_L$	+ 0.125 V	$F_D - 0.587\ F_L$
	5	+ 0.075 V	$F_D - 0.809\ F_L$	+ 0.034 V	$F_D - 0.951\ F_L$
	6	+ 0.019 V	$F_D - 1.000\ F_L$	+ 0.034 V	$F_D - 0.951\ F_L$
	7	+ 0.075 V	$F_D - 0.809\ F_L$	+ 0.125 V	$F_D - 0.587\ F_L$
	8	+ 0.165 V	$F_D - 0.309\ F_L$	+ 0.180 V	F_D
	9	+ 0.165 V	$F_D + 0.309\ F_L$	+ 0.125 V	$F_D + 0.587\ F_L$
	10	+ 0.075 V	$F_D + 0.809\ F_L$	+ 0.034 V	$F_D + 0.951\ F_L$
12	1	+ 0.011 V	$F_D + 1.000\ F_L$	+ 0.020 V	$F_D + 0.965\ F_L$
	2	+ 0.047 V	$F_D + 0.866\ F_L$	+ 0.083 V	$F_D + 0.707\ F_L$
	3	+ 0.119 V	$F_D + 0.500\ F_L$	+ 0.145 V	$F_D + 0.258\ F_L$
	4	+ 0.155 V	F_D	+ 0.145 V	$F_D - 0.258\ F_L$
	5	+ 0.119 V	$F_D - 0.500\ F_L$	+ 0.083 V	$F_D - 0.707\ F_L$
	6	+ 0.047 V	$F_D - 0.866\ F_L$	+ 0.020 V	$F_D - 0.965\ F_L$
	7	+ 0.011 V	$F_D - 1.000\ F_L$	+ 0.020 V	$F_D - 0.965\ F_L$
	8	+ 0.047 V	$F_D - 0.866\ F_L$	+ 0.083 V	$F_D - 0.707\ F_L$

(*Continued*)

Table 4-13
Summary of loads, forces & moments at support locations—cont'd

Qty of Columns	Leg No.	Case 1: At Columns		Case 2: Between Columns	
		Horiz (Vn)	Vertical (Q)	Horiz (Vn)	Vertical (Q)
	9	+ 0.119 V	$F_D - 0.500\ F_L$	+ 0.145 V	$F_D - 0.258\ F_L$
	10	+ 0.155 V	F_D	+ 0.145 V	$F_D + 0.258\ F_L$
	11	+ 0.119 V	$F_D + 0.500\ F_L$	+ 0.083 V	$F_D + 0.707\ F_L$
	12	+ 0.047 V	$F_D + 0.866\ F_L$	+ 0.020 V	$F_D + 0.965\ F_L$
16	1	+ 0.004 V	$F_D + 1.000\ F_L$	+ 0.009 V	$F_D + 0.980\ F_L$
	2	+ 0.021 V	$F_D + 0.923\ F_L$	+ 0.040 V	$F_D + 0.831\ F_L$
	3	+ 0.062 V	$F_D + 0.707\ F_L$	+ 0.084 V	$F_D + 0.555\ F_L$
	4	+ 0.103 V	$F_D + 0.382\ F_L$	+ 0.115 V	$F_D + 0.195\ F_L$
	5	+ 0.120 V	F_D	+ 0.115 V	$F_D - 0.195\ F_L$
	6	+ 0.103 V	$F_D - 0.382\ F_L$	+ 0.084 V	$F_D - 0.555\ F_L$
	7	+ 0.062 V	$F_D - 0.707\ F_L$	+ 0.040 V	$F_D - 0.831\ F_L$
	8	+ 0.021 V	$F_D - 0.923\ F_L$	+ 0.009 V	$F_D - 0.980\ F_L$
	9	+ 0.004 V	$F_D - 1.000\ F_L$	+ 0.009 V	$F_D - 0.980\ F_L$
	10	+ 0.021 V	$F_D - 0.923\ F_L$	+ 0.040 V	$F_D - 0.831\ F_L$
	11	+ 0.062 V	$F_D - 0.707\ F_L$	+ 0.084 V	$F_D - 0.555\ F_L$
	12	+ 0.103 V	$F_D - 0.382\ F_L$	+ 0.115 V	$F_D - 0.195\ F_L$
	13	+ 0.120 V	F_D	+ 0.115 V	$F_D + 0.195\ F_L$
	14	+ 0.103 V	$F_D + 0.382\ F_L$	+ 0.084 V	$F_D + 0.555\ F_L$
	15	+ 0.062 V	$F_D + 0.707\ F_L$	+ 0.040 V	$F_D + 0.831\ F_L$
	16	+ 0.021 V	$F_D + 0.923\ F_L$	+ 0.009 V	$F_D + 0.980\ F_L$

Notes:

1. Radius, R_n in equations will be R_1 if a girder is used, R_c if no girder is used.

Use: ____________

I_b = ____________

r_b = ____________

A_b = ____________

- Compressive Stress, f_a

$$f_a = Q_c/A_c \le F_a$$

- Slenderness ratio, $S_r = KL_1/n'r_b$ $n' = 1$ for not pinned, 2 for pinned
  $F_a =$

Tension Case

- Tension stress, f_t

$$f_t = f/A_b \le F_t$$

- Allowable tension stress, F_t,

$$F_T = 1.2(0.6)F_y$$

Sway Bracing

Note: Loads in sway bracing are tension only.

- Area of bracing required, A_{br}

$$A_{br} = f/F_t$$

- Allowable tensile stress, F_t,

$$F_T = 1.2(0.6)F_y$$

End Connections

- Shear per bolt = .5 f / number of bolts
- Shear per inch of weld = .5 f / inch of weld

Table 4-14
Allowable shear load in kips (bolts and welds per AISC steel construction manual, ASD method)

Bolt Size	A-307	A-325
.625”	3.68	7.36
.75”	5.30	10.6
.875”	7.21	14.4
1”	9.42	18.8
1.125”	11.9	23.8
WELD SIZE	E60XX	E70XX
.1875	2.39	2.78
.25	3.18	3.71
.3125	3.98	4.64
.375	4.77	5.57
.4375	5.56	6.50

Table 4-15
Suggested sizes of legs and cross-bracing

Vessel O.D. (in.)	Tan to Tan Length (in.)	Support Leg Angle Sizes (in.)	Base Plate Size (in.)	Bracing Angle Size (in.)	Bolt Size (in.)	Y (in.)
Up to 30	Up to 240	(3) 3 × 3 × ¼	6 × 6 × ⅜	2 × 2 × ¼	¾	12
	Up to 120	(4) 3 × 3 × ¼	6 × 6 × ⅜		¾	8
30 to 42	121 to 169	(4) 3 × 3 × ¼	6 × 6 × ⅜	2 × 2 × ¼	¾	10
	170 to 240	(4) 3 × 3 × ⅜	6 × 6 × ½		¾	12
	Up to 120	(4) 3 × 3 × ⅜	6 × 6 × ½	2½ × 2½ × ¼	¾	8
43 to 54	121 to 169	(4) 3 × 3 × ⅜	6 × 6 × ½		¾	10
	170 to 240	(4) 4 × 4 × ⅜	8 × 8 × ⅜		¾	12
	Up to 120	(4) 4 × 4 × ⅜	8 × 8 × ⅜	2½ × 2½ × ¼	1	8
55 to 56	121 to 169	(4) 4 × 4 × ½	8 × 8 × ½		1	10
	170 to 240	(4) 4 × 4 × ½	8 × 8 × ½		1	12
	Up to 120	(4) 5 × 5 × ⅜	9 × 9 × ½	3 × 3 × ¼	1⅛	8
67 to 78	121 to 169	(4) 5 × 5 × ⅜	9 × 9 × ½		1⅛	10
	170 to 240	(4) 6 × 6 × ½	10 × 10 × ½		1⅛	12
	Up to 120	(4) 6 × 6 × ½	10 × 10 × ½	3 × 3 × ¼	1⅛	10
79 to 80	121 to 169	(4) 6 × 6 × ½	10 × 10 × ½		1⅛	12
	170 to 240	(4) 6 × 6 × ½	10 × 10 × ½		1⅜	12
	Up to 120	(4) 6 × 6 × ½	10 × 10 × ½	3 × 3 × ⅜	1⅜	12
91 to 102	121 to 169	(6) 6 × 6 × ½	10 × 10 × ½		1⅜	12
	170 to 240	(6) 6 × 6 × ⅝	10 × 10 × ¾		1⅜	12

Post Connection Plate

See "Design of Ring Girders"

Notes

1. Cross-bracing the legs will conveniently reduce bending in legs due to overturning moments ("wind and earthquake") normally associated with unbraced legs. The lateral bracing of the legs must be sized to take lateral loads induced in the frame that would otherwise cause the legs to bend.
2. Legs may be made from angles, pipes, channels, beam sections, or rectangular tubing.
3. Legs longer than about 7 ft should be cross-braced.
4. Check to see if the cross-bracing interferes with piping from bottom head.
5. Shell stresses at the leg attachment should be investigated for local loads. For thin shells, extend "Y." Legs should be avoided as a support method for vessels with high shock loads or vibration service.

Procedure 4-6: Seismic Design – Vessel on Rings [4,5,8]

Notation

C_v, C_h = vertical/horizontal seismic factors
A_b = bearing area, in.^{2}
F_v, F_h, = vertical/horizontal seismic force, lb
N = number of support points
n = number of gussets at supports
P, P_e = internal/external pressure, psi
W = vessel weight under consideration, lb
σ_b = bending stress, psi
σ_ϕ = circumferential stress, psi
K_r = internal moment coefficient
C_r = internal tension/compression coefficient
Z = required section modulus, ring, in.^{3}
I_{1-2} = moment of inertia of rings, in.^{4}
S = code allowable stress, tension, psi
A_{1-2} = cross-sectional area, ring, in.^{2}
T_C, T_T = compression/tension loads in rings, lb
M = internal moment in rings, in.-lb

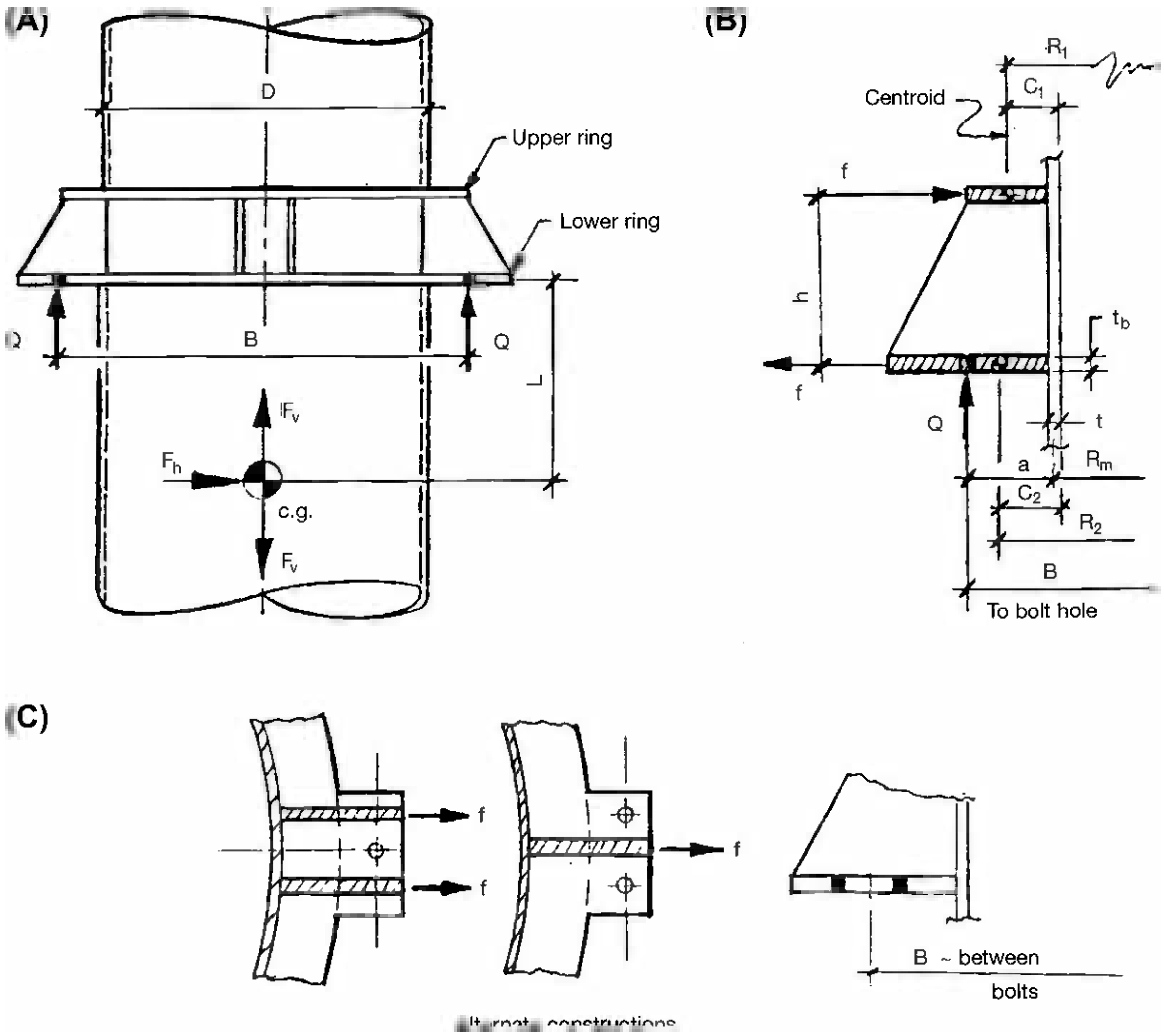


Figure 4-20. Typical dimensional data and forces for a vessel supported on rings.

M_b = bending moment in base ring, in.-lb, greater of M_x or M_y
B_p = bearing pressure, psi
Q = maximum vertical load at supports, lb
f = radial loads on rings, lb

- *Internal moment in rings, M_1 and M_2.*

Upper ring:

$$M_1 = k_r f R_1 \cos\theta$$

Lower ring:

$$M_2 = k_r f R_2 \cos\theta$$

Note: $\cos\theta$ is to be used for nonradial loads. Disregard if load f is radial.

- *Required section modulus of upper ring, Z.*

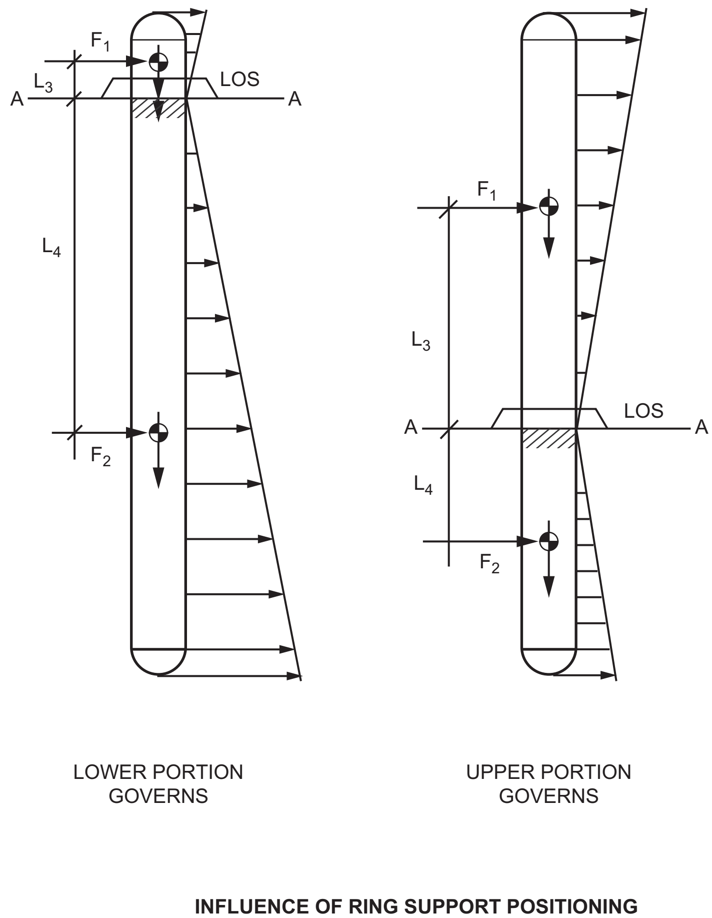
$$Z = \frac{M_1}{S}$$

Note: It is assumed the lower ring is always larger or of equal size to the upper ring.

- *Tension/compression loads in rings. Note*: In general the upper ring is in compression at the application of the loads and in tension between the loads. The lower ring is in tension at the loads and in compression between the loads. Since the governing stress is normally at the loads, the governing stresses would be:

Upper ring:

$$T_c = C_r f \cos\theta$$



M_{max} = GREATER OF ...

$M_{AA} = F_1 L_3$

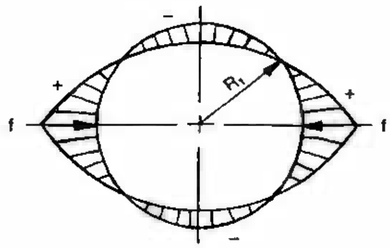
Or $F_2 L_4$

V_{max} = Greater of F_1 or F_2

Figure 4-21. Vessel supported on rings (Influence of support positioning).

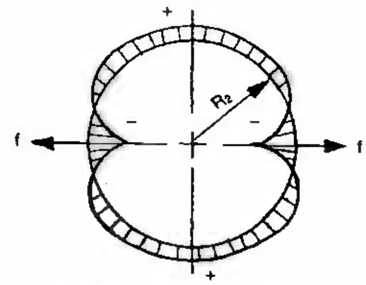
Upper Ring

Moment diagrams shown (typical)



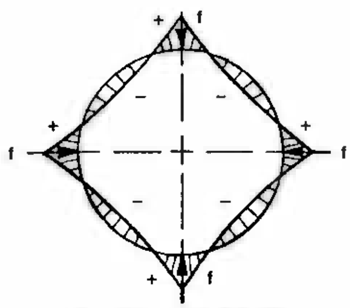
	K_r	C_r
At loads	+0.3183	0
Between loads	−0.1817	−0.5

Lower Ring

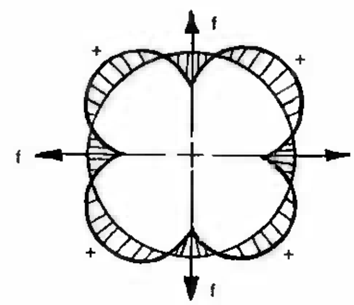


	K_r	C_r
At loads	−.3183	0
Between loads	+.1817	+.5

Two loads

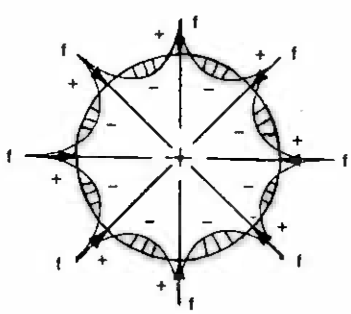


	K_r	C_r
At loads	+0.1366	−0.5
Between loads	−0.0705	−0.7071

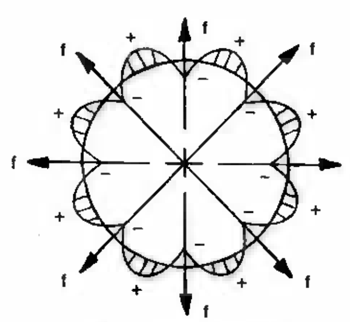


	K_r	C_r
At loads	−0.1366	+0.5
Between loads	+0.0705	+0.7071

Four loads



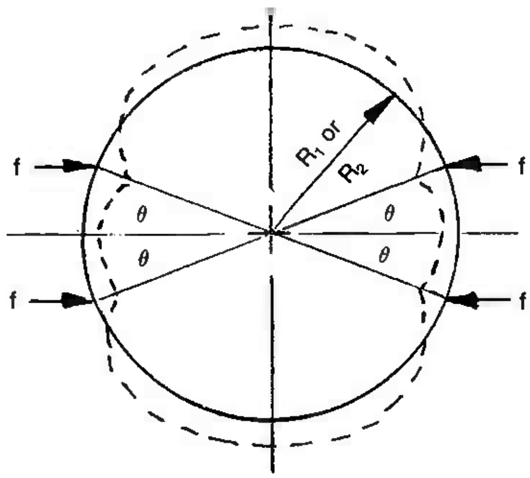
	K_r	C_r
At loads	+0.0661	−1.2071
Between loads	−0.034	−1.306



	K_r	C_r
At loads	−0.0661	+1.2071
Between loads	+0.034	+1.306

Eight Loads

Figure 4-22. Coefficients for rings.



	At Loads		Between Loads	
	K_r	C_r	K_r	C_r
1°	+0.619	−0.017	−0.365	−1.00
2°	+0.601	−0.041	−0.366	−0.999
3°	+0.584	−0.052	−0.363	−0.998
4°	+0.566	−0.071	−0.362	−0.997
5°	+0.550	−0.087	−0.360	−0.996
6°	+0.532	−0.105	−0.359	−0.995
7°	+0.515	−0.122	−0.357	−0.992
8°	+0.498	−0.138	−0.355	−0.990
9°	+0.481	−0.155	−0.352	−0.986
10°	+0.466	−0.171	−0.348	−0.985
15°	+0.387	−0.250	−0.329	−0.966
20°	+0.315	−0.321	−0.303	−0.940
25°	+0.254	−0.383	−0.270	−0.906
30°	+0.204	−0.433	−0.229	−0.866
35°	+0.167	−0.469	−0.183	−0.819
40°	+0.144	−0.492	−0.129	−0.766
45°	+0.137	−0.500	−0.070	−0.707

Figure 4-23. Coefficients for rings. (Signs in the table are for loads as shown. Reverse signs for loads are in the opposite direction.).

Lower ring:

$$T_T = C_r f \cos\theta$$

where C_r is the maximum positive value for T_T and the maximum negative value for T_c.

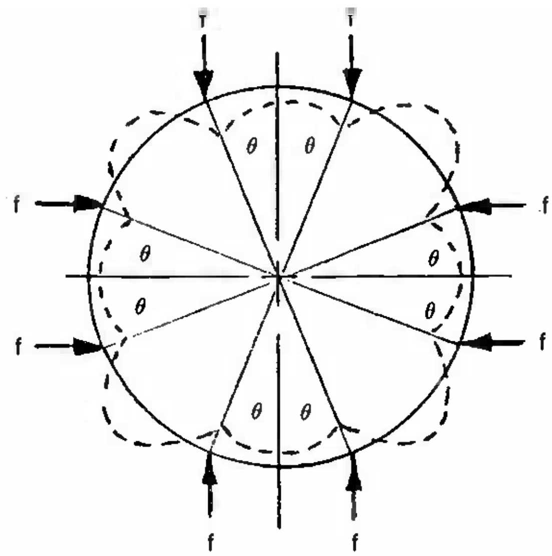
- *Maximum circumferential stress in shell, σ_ϕ.*
  Compression: in upper ring

$$\sigma_\phi = (-)\frac{P_e R_m}{t} - \frac{T_c}{A_1}$$

Tension: in lower ring

$$\sigma_\phi = \frac{PR_m}{t} + \frac{T_T}{A_2}$$

- *Maximum bending stress in shell.*



	At Loads		Between Loads	
	K_r	C_r	K_r	C_r
1°	+0.254	−1.018	−0.143	−1.411
2°	+0.238	−1.040	−0.143	−1.41[illegible]
3°	+0.221	−1.050	−0.142	−1.409
4°	+0.206	−1.066	−0.140	−1.408
5°	+0.194	−1.079	−0.136	−1.407
6°	+0.178	−1.095	−0.135	−1.406
7°	+0.165	−1.108	−0.133	−1.405
8°	+0.153	−1.117	−0.130	−1.40[illegible]
9°	+0.141	−1.130	−0.124	−1.397
10°	+0.130	−1.141	−0.119	−1.393
15°	+0.090	−1.183	−0.093	−1.366
20°	+0.069	−1.204	−0.056	−1.329
25°	+0.069	−1.204	−0.008	−1.282
30°	+0.090	−1.183	+0.049	−1.225
35°	+0.132	−1.141	+0.115	−1.158
40°	+0.194	−1.079	+0.190	−1.083
45°	+0.273	−1.000	+0.273	−1.000

Figure 4-24. Coefficients for rings. (Signs in the table are for loads as shown. Reverse signs for loads are in the opposite direction.).

Upper ring:

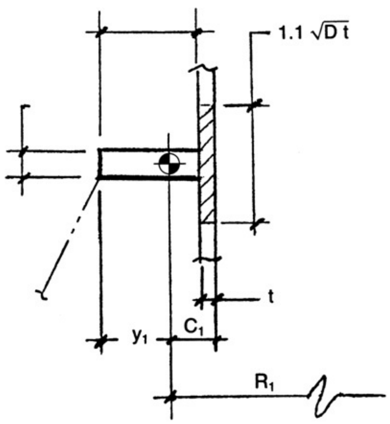
$$\sigma_b = \frac{M_1 C_1}{I_1}$$

Lower ring:

$$\sigma_b = \frac{M_2 C_2}{I_2}$$

- *Maximum bending stress in ring.*
  Upper ring:

$$\sigma_b = \frac{M_1 y_1}{I_1}$$

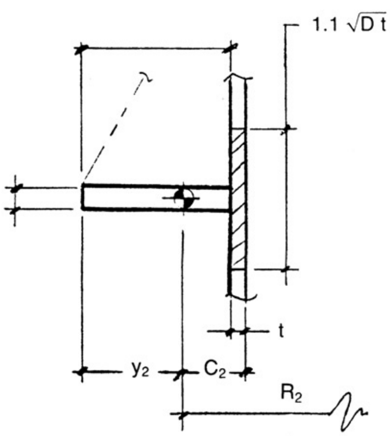


$C_1 = \frac{\Sigma AY}{\Sigma A} =$ $\qquad y_1 =$

$I_1 = \Sigma AY^2 + \Sigma I - C_1 \Sigma AY =$

Item	A	Y	Y^2	AY	AY^2	I
Shell						
Ring						
Σ						

Figure 4-25. Properties of upper ring.



$C_2 = \frac{\Sigma AY}{\Sigma A} =$ $\qquad y_2 =$

$I_2 = \Sigma AY^2 + \Sigma I - C_2 \Sigma AY =$

Item	A	Y	Y^2	AY	AY^2	I
Shell						
Ring						
Σ						

Figure 4-26. Properties of lower ring.

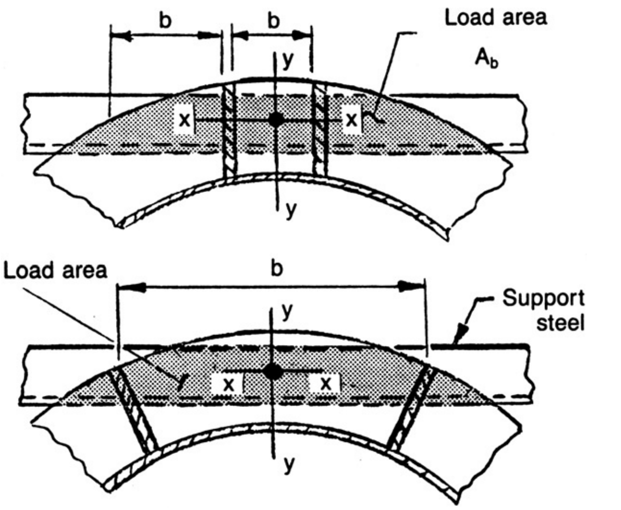


Figure 4-27. Determining the thickness of the lower ring to resist bending.

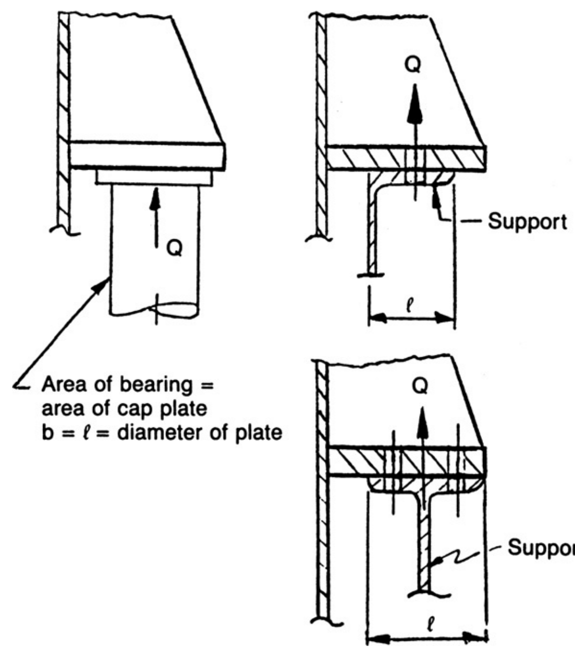


Table 4-16
Maximum bending moments in a bearing plate with gussets

$\frac{\ell}{b}$	$M_x \begin{bmatrix} x = 0.5b \\ y = \ell \end{bmatrix}$	$M_x \begin{bmatrix} x = 0.5b \\ y = 0 \end{bmatrix}$
0	0	(−)0.500 $B_p \ell^2$
0.333	0.0078 $B_p b^2$	(−)0.428 $B_p \ell^2$
0.5	0.0293 $B_p b^2$	(−)0.319 $B_p \ell^2$
0.666	0.0558 $B_p b^2$	(−)0.227 $B_p \ell^2$
1.0	0.0972 $B_p b^2$	(−)0.119 $B_p \ell^2$
1.5	0.1230 $B_p b^2$	(−)0.124 $B_p \ell^2$
2.0	0.1310 $B_p b^2$	(−)0.125 $B_p \ell^2$
3.0−∞	0.1330 $B_p b^2$	(−)0.125 $B_p \ell^2$

Reprinted by permission of John Wiley & Sons, Inc.
From Process Equipment Design, *Table 10.3. (See Note 2.)*

- *Properties of upper ring.*
- *Properties of lower ring.*

Lower ring:

$$\sigma_b = \frac{M_2 y_2}{I_2}$$

- *Thickness of lower ring to resist bending.*

Bearing area, A_b:

$$A_b =$$

Bearing pressure, B_p:

$$B_p = \frac{Q}{A_b}$$

From Table 4-16, select the equation for the maximum bending moment in the bearing plate. Use the greater of M_x or M_y.

$$\frac{\ell}{b} =$$

$$M_b =$$

Minimum thickness of lower ring, t_b:

$$t_b = \sqrt{\frac{6M_b}{S}}$$

Notes

1. Rings may induce high localized stresses in shell immediately adjacent to rings.
2. When $l/b \leq 1.5$, the maximum bending moment occurs at the junction of the ring and shell. When $l/b > 1.5$, the maximum bending moment occurs at the middle of the free edge.
3. Since the mean radius of the rings may be unknown at the beginning of computations, yet is required for determining maximum bending moment, substitute R_m as a satisfactory approximation at that stage.
4. The following values may be estimated:
   - *Ring thickness:* The thickness of each ring is arbitrary and can be selected by the designer. A suggested value is

     $$t_b = 0.3\sqrt[3]{\frac{M_{max}}{S}}$$

   - *Ring spacing:* Ring spacing is arbitrary and can be selected by the designer. A suggested minimum value is

     $$h = B - D$$

   - *Ring depth:* The depth of ring cannot be computed directly, but must be computed by successive approximations. As a first trial,

     $$d = 2.1\sqrt{\frac{M_{max}}{t_r S}}$$

Procedure 4-7: Seismic Design – Vessel on Lugs [5,8–13]

Notation

R_m = center line radius of shell, in.
N = number of equally spaced lugs
W = weight of vessel plus contents, lb
f = radial load, lb
F_h = horizontal seismic force, lb
F_v = vertical seismic force, lb
V_h = horizontal shear per lug, lb
V_v = vertical shear per lug, lb
Q = vertical load on lugs, lb
γ, β = coefficients
M_c = external circumferential moment, in.–lb
M_L = external longitudinal moment, in.-lb
M_ϕ = internal bending moment, circumferential, in.-lb/in.
M_x = internal bending moment longitudinal, in. -lb/in.
N_ϕ = membrane force in shell, circumferential, lb/in.
N_x = membrane force in shell, longitudinal, lb/in.
P = internal pressure, psi
C_h = horizontal seismic factor
C_v = vertical seismic factor
C_c,C_i. = multiplication factors for N_ϕ and N_x for rectangular attachments
K_C,K_1. = coefficients for determining β for moment loads on rectangular areas

K_1, K_2 = coefficients for determining β for radial loads on rectangular areas

K_n, K_b, = stress concentration factors (see Note 5)

σ_ϕ = circumferential stress, psi

σ_x = longitudinal stress, psi

t_s = thickness of shell, in.

t_p = thickness of reinforcing pad, in.

α = coefficient of thermal expansion, in/in/°F

ζ = radial deflection, in

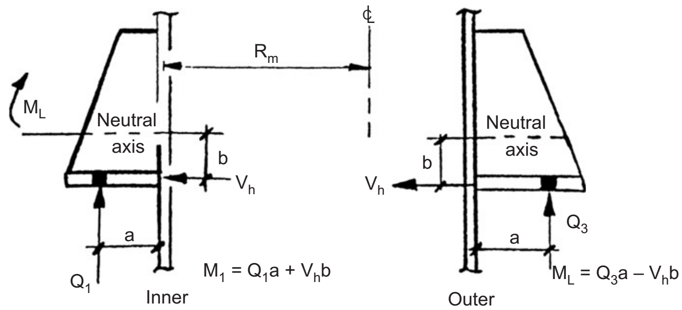


Figure 4-28. Dimensions and forces for support lug.

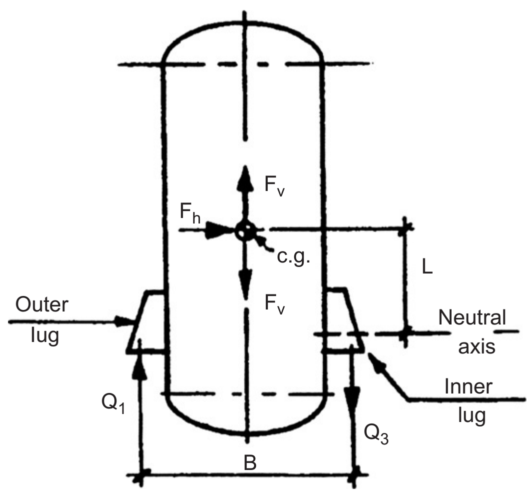


Figure 4-29. Case 1: Lugs below the center of gravity.

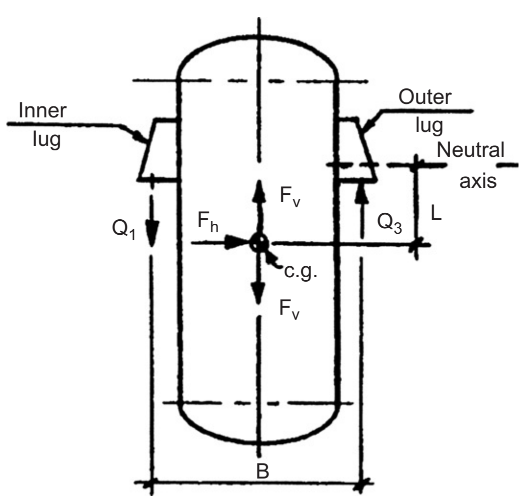


Figure 4-30. Case 2: Lugs above the center of gravity.

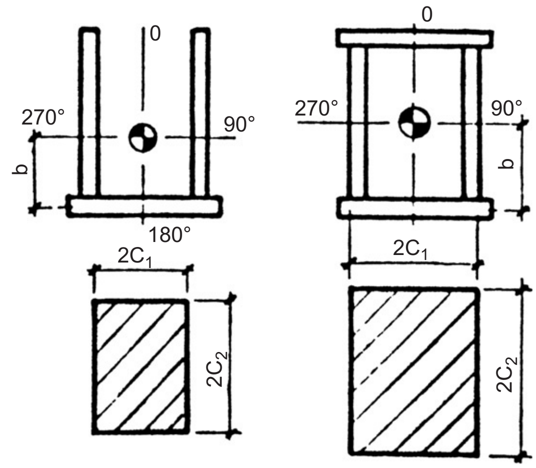


Figure 4-31. Area of loading.

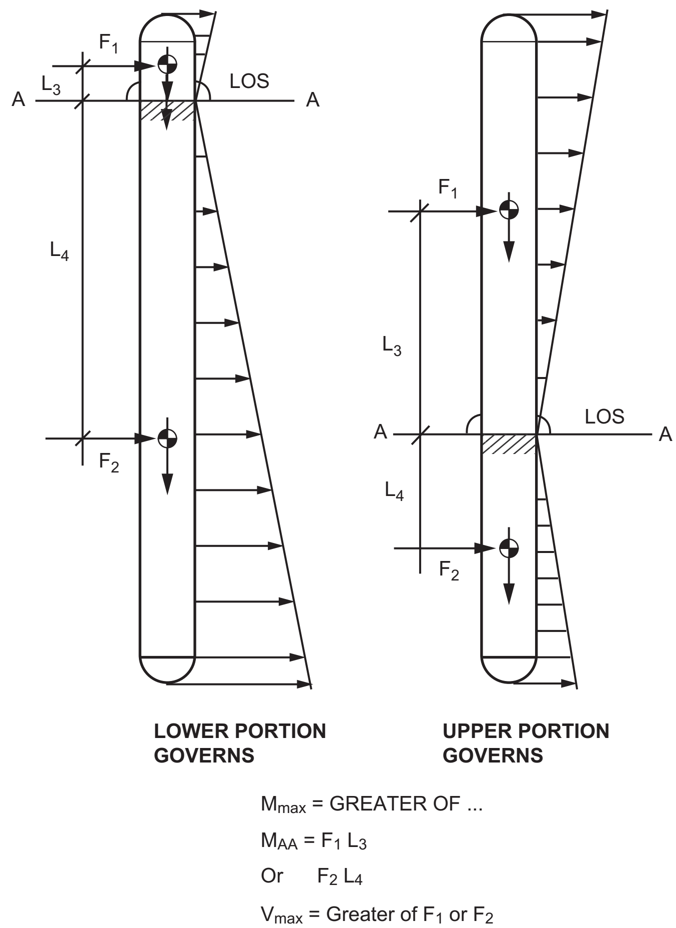
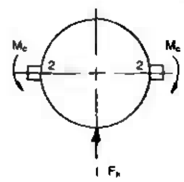
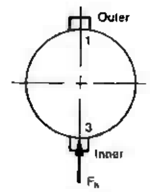
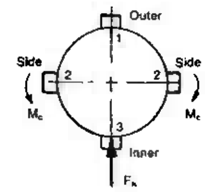


Figure 4-32. Vessel supported on lugs (Influence of support positioning).

Step 1: Compute forces and moments

FORCES			
Lateral force	$F_h = C_h W$		
Horizontal shear per lug	$V_h = F_h/N$		
Vertical force	$F_v = (1 + C_v)W$		
Vertical shear per lug	$V_v = F_v/N$		
LOAD DIAGRAMS			
	Case 1: Two Lugs	**Case 2: Two Lugs**	**Case 3: Four Lugs**
			
VERTICAL LOADS AT LUGS, Q			
Outer		$Q_1 = V_v - \frac{F_h L}{B}$	$Q_1 = V_v - \frac{F_h L}{B}$
Sides	$Q_2 = V_V$		$Q_2 = V_V$
Inner		$Q_3 = V_v + \frac{F_h L}{B}$	$Q_3 = V_v + \frac{F_h L}{B}$
LONGITUDINAL MOMENT, M_L			
Outer		$M_{L1} = Q_1 a - V_h b$	$M_{L1} = Q_1 a - V_h b$
Sides	$M_{L2} = Q_2 a$		$M_{L2} = Q_2 a$
Inner		$M_{L3} = Q_3 a + V_h b$	$M_{L3} = Q_3 a + V_h b$
CIRCUMFERENTIAL MOMENT, M_c			
Sides	$M_c = V_h a$		$M_c = V_h a$

Step 2: Compute geometric parameters

$\gamma = R_m/t$	$\beta_1 = C_1/R_m$	$\beta_2 = C_2/R_m$	β_1/β_2

Step 3: Compute equivalent β values (values of C_L, C_C, K_L, and K_C from Tables 4-17 and 4-18)

β Values for Longitudinal Moment

Values of β		C_L	K_L	β
$\beta_a = \sqrt[3]{\beta_1 \beta_2^2}$	N_ϕ			
$\beta_b = \sqrt[3]{\beta_1 \beta_2^2}$	N_x			
$\beta_c = K_L \sqrt[3]{\beta_1 \beta_2^2}$	M_ϕ			
$\beta_d = K_L \sqrt[3]{\beta_1 \beta_2^2}$	M_x			

β Values for Circumferential Moment

Values of β		C_c	K_c	β
$\beta_e = \sqrt[3]{\beta_1^2 \beta_2}$	N_ϕ			
$\beta_f = \sqrt[3]{\beta_1^2 \beta_2}$	N_x			
$\beta_g = K_c \sqrt[3]{\beta_1^2 \beta_2}$	M_ϕ			
$\beta_h = K_c \sqrt[3]{\beta_1^2 \beta_2}$	M_x			

Step 4: Compute stresses

Forces	Figure	β	Values from Figure	Forces and Moments	Stress
				Longitudinal Moment	
Membrane	7.23A	$\beta_a =$	$\frac{N_\phi R_m^2 \beta}{M_L} = (\ )$	$N_\phi = \frac{(\ )C_L M_L}{R_m^2 \beta} =$	$\sigma_\phi = \frac{K_n N_\phi}{t_s} =$
	7.23B	$\beta_b =$	$\frac{N_x R_m^2 \beta}{M_L} = (\ )$	$N_x = \frac{(\ )C_L M_L}{R_m^2 \beta} =$	$\sigma_x = \frac{K_n N_x}{t_s} =$
Bending	7.24A	$\beta_c =$	$\frac{M_\phi R_m \beta}{M_L} = (\ )$	$M_\phi = \frac{(\ )M_L}{R_m \beta} =$	$\sigma_\phi = \frac{6K_b M_\phi}{t_s^2} =$
	7.24B	$\beta_d =$	$\frac{M_x R_m \beta}{M_L} = (\ )$	$M_x = \frac{(\ )M_L}{R_m \beta} =$	$\sigma_x = \frac{6K_b M_x}{t_s^2} =$
				Circumferential Moment	
Membrane	7.25A	$\beta_e =$	$\frac{N_\phi R_m^2 \beta}{M_c} = (\ )$	$N_\phi = \frac{(\ )C_c M_c}{R_m^2 \beta} =$	$\sigma_\phi = \frac{K_n N_\phi}{t_s} =$
	7.25B	$\beta_f =$	$\frac{N_x R_m^2 \beta}{M_c} = (\ )$	$N_x = \frac{(\ )C_c M_c}{R_m^2 \beta} =$	$\sigma_x = \frac{K_n N_x}{t_s} =$
Bending	7.26A	$\beta_g =$	$\frac{M_\phi R_m \beta}{M_c} = (\ )$	$M_\phi = \frac{(\ )M_c}{R_m \beta} =$	$\sigma_\phi = \frac{6K_b M_\phi}{t_s^2} =$
	7.26B	$\beta_h =$	$\frac{M_x R_m \beta}{M_c} = (\ )$	$M_x = \frac{(\ )M_c}{R_m \beta} =$	$\sigma_x = \frac{6K_b M_x}{t_s^2} =$

Table 4-17
Coefficients for circumferential moment, M_c

β_1/β_2	γ	C_c for N_ϕ	C_c for N_x	K_c for M_ϕ	K_c for M_x
	15	0.31	0.49	1.31	1.84
	50	0.21	0.46	1.24	1.62
0.25	100	0.15	0.44	1.16	1.45
	200	0.12	0.45	1.09	1.31
	300	0.09	0.46	1.02	1.17
	15	0.64	0.75	1.09	1.36
	50	0.57	0.75	1.08	1.31
0.5	100	0.51	0.76	1.04	1.16
	200	0.45	0.76	1.02	1.20
	300	0.39	0.77	0.99	1.13
	15	1.17	1.08	1.15	1.17
	50	1.09	1.03	1.12	1.14
1	100	0.97	0.94	1.07	1.10
	200	0.91	0.91	1.04	1.06
	300	0.85	0.89	0.99	1.02
	15	1.70	1.30	1.20	0.97
	50	1.59	1.23	1.16	0.96
2	100	1.43	1.12	1.10	0.95
	200	1.37	1.06	1.05	0.93
	300	1.30	1.00	1.00	0.90
	15	1.75	1.31	1.47	1.08
	50	1.64	1.11	1.43	1.07
4	100	1.49	0.81	1.38	1.06
	200	1.42	0.78	1.33	1.02
	300	1.36	0.74	1.27	0.98

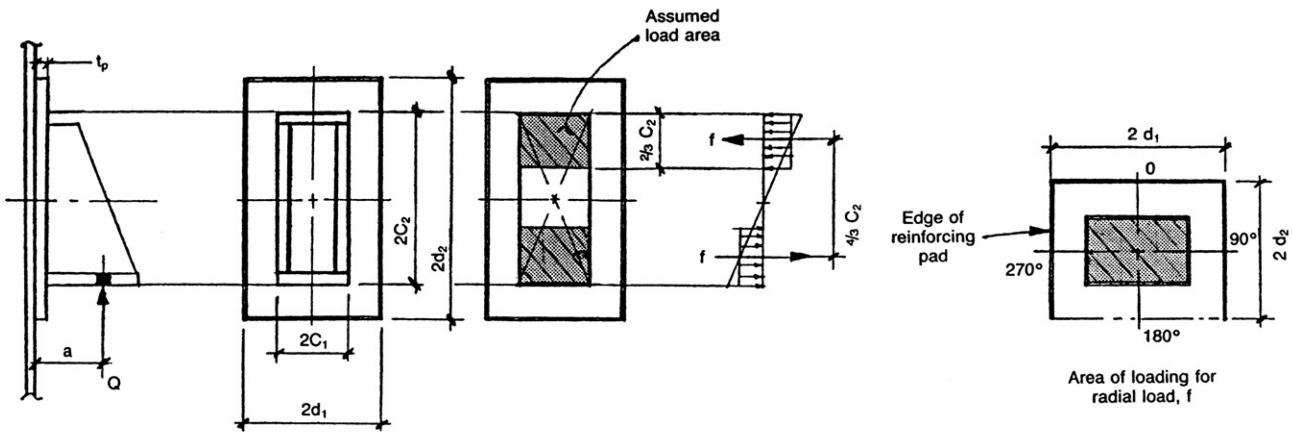
Reprinted by permission of the Welding Research Council.

Table 4-18
Coefficients for longitudinal moment, M_L

β_1/β_2	γ	C_L for N_ϕ	C_L for N_x	K_L for M_ϕ	K_L for M_x
	15	0.75	0.43	1.80	1.24
	50	0.77	0.33	1.65	1.16
0.25	100	0.80	0.24	1.59	1.11
	200	0.85	0.10	1.58	1.11
	300	0.90	0.07	1.56	1.11
	15	0.90	0.76	1.08	1.04
	50	0.93	0.73	1.07	1.03
0.5	100	0.97	0.68	1.06	1.02
	200	0.99	0.64	1.05	1.02
	300	1.10	0.60	1.05	1.02
	15	0.89	1.00	1.01	1.08
	50	0.89	0.96	1.00	1.07
1	100	0.89	0.92	0.98	1.05
	200	0.89	0.99	0.95	1.01
	300	0.95	1.05	0.92	0.96
	15	0.87	1.30	0.94	1.12
	50	0.84	1.23	0.92	1.10
2	100	0.81	1.15	0.89	1.07
	200	0.80	1.33	0.84	0.99
	300	0.80	1.50	0.79	0.91
	15	0.68	1.20	0.90	1.24
	50	0.61	1.13	0.86	1.19
4	100	0.51	1.03	0.81	1.12
	200	0.50	1.18	0.73	0.98
	300	0.50	1.33	0.64	0.83

Reprinted by permission of the Welding Research Council.

Analysis when Reinforcing Pads are Used



$$2d_{11}\ \text{max} = \frac{4d_1C_2}{3C_1} =$$

Figure 4-33. Dimensions of load areas for radial loads.

Step 1: Compute radial loads f

	Case 1	Case 2	Case 3
Outer		$f_1 = \dfrac{3M_{L1}}{4C_2}$	$f_1 = \dfrac{3M_{L1}}{4C_2}$
Sides	$f_2 = \dfrac{3M_{L2}}{4C_2}$		$f_2 = \dfrac{3M_{L2}}{4C_2}$
Inner		$f_3 = \dfrac{3M_{L3}}{4C_2}$	$f_3 = \dfrac{3M_{L3}}{4C_2}$

Step 2: Compute geometric parameters.

At Edge of Attachment	At Edge of Pad
$R_m = \dfrac{\text{I.D.} + t_s + t_p}{2}$	$R_m = \dfrac{\text{I.D.} + t_s}{2}$
$t = \sqrt{t_s^2 + t_p^2}$	$t = t_s$
$\gamma = R_m/t$	$\gamma = R_m/t$
$\beta_1 = C_1/R_m$	$\beta_1 = d_1/R_m$
$\beta_2 = 4C_2/3R_m$	$\beta_2 = d_2/R_m$
β_1/β_2	β_1/β_2

Step 3: Compute equivalent β values.

Table 4-19

Four values of β are computed for use in determining N_ϕ, N_x, M_ϕ, and M_x as follows. The values of K_1 and K_2 are taken from Table 4-19.		**Values of coefficient K_1 and K_2**		
$\beta_1/\beta_2 \geq 1$	**β**		**K_1**	**K_2**
$\beta = [1 - \frac{1}{3}(\frac{\beta_1}{\beta_2} - 1)(1 - k_1)]\sqrt{\beta_1\beta_2}$	β_a for $N_\phi =$ β_b for $N_x =$	$N\phi$ N_x	0.91 1.68	1.48 1.2
$\beta_1/\beta_2 < 1$				
$\beta = [1 - \frac{4}{3}(1 - \frac{\beta_1}{\beta_2})(1 - k_2)]\sqrt{\beta_1/\beta_2}$	β_c for $M_\phi =$ β_d for $M_x =$	$M\phi$ M_x	1.76 1.2	0.88 1.25

Reprinted by permission of the Welding Research Council.

Step 4: Compute stresses for a radial load.

Radial Load	Figure	β	Values from Figure	Forces and Moments	Stress
Membrane	7-21A	$\beta_a =$	$\frac{N_\phi R_m}{f} = (\ )$	$N_\phi = \frac{(\ )f}{R_m} =$	$\sigma_\phi = \frac{K_n N_\phi}{t} =$
	7-21B	$\beta_b =$	$\frac{N_x R_m}{f} = (\ )$	$N_x = \frac{(\ )f}{R_m} =$	$\sigma_x = \frac{K_n N_x}{t} =$
Bending	7-22A	$\beta_c =$	$\frac{M_\phi}{f} = (\ )$	$M_\phi = (\ )f =$	$\sigma_\phi = \frac{6K_b M_\phi}{t^2} =$
	7-22B	$\beta_d =$	$\frac{M_x}{f} = (\ )$	$M_x = (\ )f =$	$\sigma_x = \frac{6K_b M_x}{t^2} =$

COMBINING STRESSES

WITHOUT REINFORCING PAD

Stress Due To			σ_x 0°	σ_x 90°	σ_x 180°	σ_x 270°	σ_ϕ 0°	σ_ϕ 90°	σ_ϕ 180°	σ_ϕ 270°
Longitudinal moment, M_L	Membrane	N_ϕ					+		−	
		N_x	+		−					
	Bending	M_ϕ					+		−	
		M_x	+		−					
Circumferential moment, M_c	Membrane	N_ϕ						+		−
		N_x		+		−				
	Bending	M_ϕ						+		−
		M_x		+		−				
Internal pressure, P	$\sigma_\phi = \frac{PR_m}{t_s}$						+	+	+	+
	$\sigma_x = \frac{PR_m}{2t_s}$		+	+	+	+				
Total	$\sum$									

WITH REINFORCING PAD

Stress Due To			σ_x 0°	σ_x 90°	σ_x 180°	σ_x 270°	σ_ϕ 0°	σ_ϕ 90°	σ_ϕ 180°	σ_ϕ 270°
Radial load, f	Membrane	N_ϕ					+	+	+	+
		N_x	+	+	+	+				
	Bending	M_ϕ					+	+	+	+
		M_x	+	+	+	+				
Internal pressure, P	$\sigma_\phi = \frac{PR_m}{t_s}$						+	+	+	+
	$\sigma_x = \frac{PR_m}{2t_s}$		+	+	+	+				
Total	$\sum$		+	+	+	+	+	+	+	+

NOTES

1. Make sure to remain consistent by lug, that is, that all loadings are from the same lug. This may require several trials to determine the worst case.
2. The calculations for combining stresses with a reinforcing pad should be completed for stresses at the edge of attachment as well as at the edge of the pad. For thinner shells the stress at the edge of the pad will usually govern.

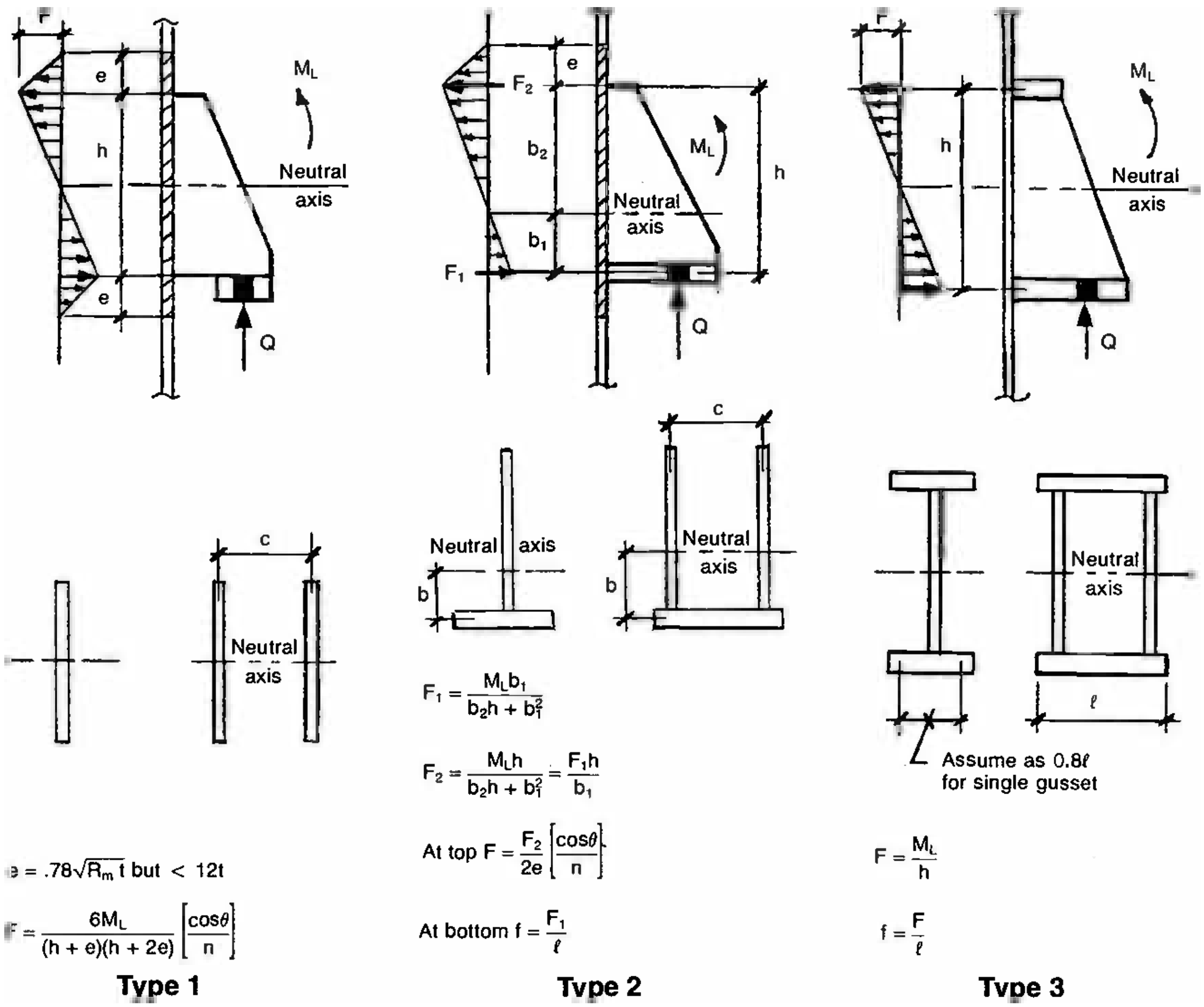
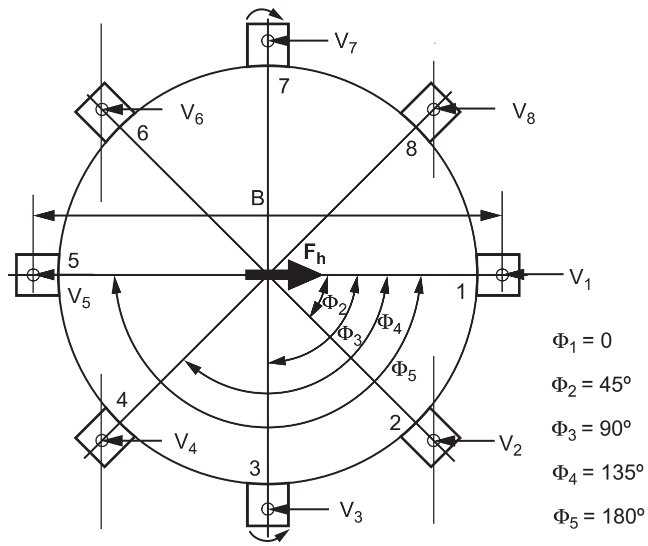
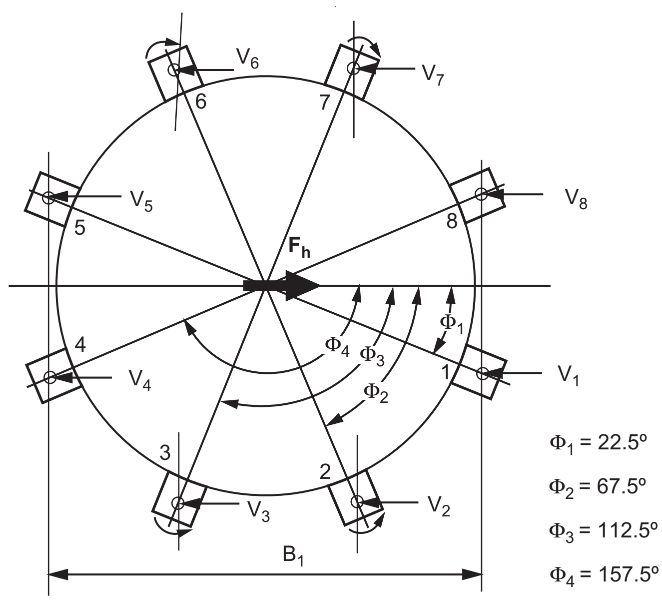


Figure 4-34. Radial loads F and f.



Case 1: Load through lugs



Case 2: Load between lugs

Figure 4-35. Vessel supported on (8) lugs.

Design of Vessel Supported on (8) Lugs

Item	Formula	Calculation
Lateral Force	$F_h = C_h\ W$	
Horizontal Shear/Lug	$V_h = F_h / N$	
Vertical Force	$F_V = (1 + C_V)\ W$	
Overturning Moment	$M = F_h\ L$	
Dead Load/Lug	$V_V = F_V / N$	
Worst Case Vertical Live Load per Lug	$F_L = 4\ M / N\ B$	
VERTICAL LIVE LOAD ON EACH LUG, F_{Ln}		
LUG	**CASE 1**	**CASE 2**
1	$-\ F_L$	$-\ .924\ F_L$
2	$-\ .707\ F_L$	$-\ .383\ F_L$
3	0	$+\ .383\ F_L$
4	$+.707\ F_L$	$+\ .924\ F_L$
5	$+\ F_L$	$+\ .924\ F_L$
6	$-\ .707\ F_L$	$+\ .383\ F_L$
7	0	$-\ .383\ F_L$
8	$+\ .707\ F_L$	$-\ .924\ F_L$
Formulas in Table are based on the following equations;		
CASE 1	$F_L = 4\ M \cos\phi_n / N\ B$	
CASE 2	$F_L = 4\ M \cos\phi_n / N\ B_1$	
Vertical Load on any Lug	$Q_n = V_V + F_{Ln}$	
Worst Case Load per Lug	$Q = V_V + F_L$	
Longitudinal Moment for any Given Lug	$M_{Ln} = Q_n\ a +/- V_h\ b$	
Longitudinal Moment for any Worst Case	$M_L = Q\ a +/- V_h\ b$	

Design of Vessel Supported on (8) Lugs (Example)			Data
ITEM	FORMULA	CALCULATION	$C_h = .1$
Lateral Force	$F_h = C_h\ W$	$F_h = .1\ (141^K) = 14.1^K$	$C_V = .2$
Horizontal Shear/Lug	$V_h = F_h / N$	$V_h = 14.1^K / 8 = 1.76^K$	$W = 141^K$
Vertical Force	$F_V = (1 + C_V)\ W$	$F_V = (1 + .2)\ 141^K = 169.2^K$	$L = 84"$
Overturning Moment	$M = F_h\ L$	$M = 14.1^K\ (84") = 1184$ In-Kips	$a = 12"$
Dead Load/Lug	$V_V = F_V / N$	$V_V = 169.2^K / 8 = 21.15^K$	$b = 6.24"$
Worst Case Vertical Live Load per Lug	$F_L = 4\ M / N\ B$	$F_L = 4(1184"^K) / (8)\ 160.25" = 3.69^K$	$B = 160.25"$
			$B_1 = 113.31$
VERTICAL LIVE LOAD ON EACH LUG, F_{Ln}			
LUG	**CASE 1**	**CASE 2**	
1	$-\ F_L$	$-\ .924\ F_L$	
2	$-\ .707\ F_L$	$-\ .383\ F_L$	
3	0	$+\ .383\ F_L$	
4	$+.707\ F_L$	$+\ .924\ F_L$	
5	$+\ F_L$	$+\ .924\ F_L$	
6	$-\ .707\ F_L$	$+\ .383\ F_L$	
7	0	$-\ .383\ F_L$	
8	$+\ .707\ F_L$	$-\ .924\ F_L$	
Formulas in Table are based on the following equations;			
CASE 1	$F_L = 4\ M \cos\phi_n / N\ B$		
CASE 2	$F_L = 4\ M \cos\phi_n / N\ B_1$		
Vertical Load on any Lug	$Q_n = V_V + F_{Ln}$		
Worst Case Load per Lug	$Q = V_V + F_L$	$Q = 21.15 + 3.69 = 24.84^K$	
Longitudinal Moment for any Given Lug	$M_{Ln} = Q_n\ a +/- V_h\ b$		
Longitudinal Moment for any Worst Case	$M_L = Q\ a +/- V_h\ b$	$M_L = 24.84^K\ (12") + 1.76^K\ (6.24") = 309$ in-Kips	

Check lug for radial thermal expansion, ζ_r

$$DT = \text{Design temperature, } ^\circ F$$
$$R = \text{Radius} = B / 2$$
$$\alpha = \text{Coefficient of thermal expansion, in/in/ } ^\circ F$$
$$\Delta T = \text{Change in temperature from 70 } ^\circ F$$
$$\zeta_r = \alpha \, \Delta T \, R =$$

Example;

$$R = 80.125 \text{ in}$$
$$DT = 925^\circ F$$
$$\alpha = 7.9\left(10^{-6}\right) \text{ in/in/}^\circ F$$
$$\Delta T = 925 - 70 = 855^\circ F$$
$$\zeta_r = 7.9\left(10^{-6}\right) 855 \left(80.125\right) = .541 \text{ in}$$

Use slotted holes!

Size of anchor bolts Required, A_r

Due to Overturning Moment

$$A_r = [(4\,M/B) - W][1/(N_b\,S_b)]$$

N_b = Number of anchor bolts
If A_r is negative, there is no uplift.

Due to Shear

Shear / lug, f_S

$$f_S = F_h/N_b$$
$$A_r = f_S/F_S$$

Use minimum size of anchor bolts of 0.75 in diameter.

Notes

1. A change in location of the c.g. for various operating levels can greatly affect the moment at lugs by increasing or decreasing the "L" dimension. Different levels and weights should be investigated for determining worst case (i.e., full, half-full, empty, etc.)
2. This procedure ignores effects of sliding friction between lugs and beams during heating/cooling cycles. These effects will be negligible for small-diameter vessels, relatively low operating temperatures, or where slide plates are used to reduce friction forces. Other cases should be investigated.
3. Since vessels supported on lugs are commonly located in structures, the earthquake effects will be dependent on the structure as well as on the vessel. Thus horizontal and vertical seismic factors must be provided.
4. If reinforcing pads are used to reduce stresses in the shell or a design that uses them is being checked, then Bijlaard recommends an analysis that converts moment loadings into equivalent radial loads. The attachment area is reduced about two-thirds. Stresses at the edge of load area and stresses at the edge of the pad must be checked. See "Analysis When Reinforcing Pads are Used."
5. Stress concentration factors are found in the procedure on local stresses.
6. To determine the area of attachment, see "Attachment Parameters." Please note that if a top (compression) plate is not used, then an equivalent rectangle that is equal to the moment of inertia of the attachment and whose width-to-height ratio is the same must be determined. The neutral axis is the rotating axis of the lug passing through the centroid.
7. Stiffening effects due to proximity to major stiffening elements, though desirable, have been neglected in this procedure.
8. Assume effects of radial loads as additive to those due to internal pressure, even though the loadings may be in the opposite directions. Although conservative, they will account for the high discontinuity stresses immediately adjacent to the lugs.
9. In general, the smaller the diameter of the vessel, the further the distribution of stresses in the circumferential direction. In small diameter vessels, the longitudinal stresses are confined to a narrow band. The opposite becomes true for larger-diameter vessels or larger R_m/t ratios.
10. If shell stresses are excessive, the following methods may be utilized to reduce the stresses:
    a. Add more lugs.
    b. Add more gussets.
    c. Increase angle θ between gussets.
    d. Increase height of lugs, h
    e. Add reinforcing pads under lugs.

f. Increase thickness of shell course to which lugs are attached.
g. Add top and bottom plates to lugs or increase width of plates.
h. Add circumferential ring stiffeners at top and bottom of lugs.

Procedure 4-8: Seismic Design – Vessel on Skirt [1,2,3]

Notation

T = period of vibration, sec
S_I = code allowable stress, tension, psi
H = overall height of vessel from bottom of base plate, ft
h_x = height from base to center of section or e.g. of a concentrated load, ft
h_i = height from base to plane under consideration, ft
α, β, γ = coefficients from Table 4-20 for given plane based on h_x/H
W_x = total weight of section, kips
W = weight of concentrated load or mass, kips
W_o = total weight of vessel, operating, kips
W_h = total weight of vessel above the plane under consideration, kips
w_x = uniformly distributed load for each section, kips/ft
F_x = lateral force applied at each section, kips
V = base shear, kips
V_x = shear at plane x, kips
M_x = moment at plane x, ft-kips
M_b = overturning moment at base, ft-kips
D = mean shell diameter of each section, ft or in.
E = modulus of elasticity at design temperature, 10^6 psi
E_l = joint efficiency
t = thickness of vessel section, in.
P_i = internal design pressure, psi
P_e = external design pressure, psi
$\Delta\alpha, \Delta\gamma$ = difference in values of α and γ from top to bottom of any given section
l_x = length of section, ft
σ_{xt} = longitudinal stress, tension, psi
σ_{xc} = longitudinal stress, compression, psi
R_o = outside radius of vessel at plane under consideration, in.
A = code factor for determining allowable compressive stress, B
B = code allowable compressive stress, psi
F = lateral seismic force for uniform vessel, kips
C_h = horizontal seismic factor

Cases

Case 1: Uniform Vessels. For vessels of uniform cross section without concentrated loads (i.e., reboilers, packing, large liquid sections, etc.) weight can be assumed to be uniformly distributed over the entire height.

W_o =

H =

D =

t =

$$T = 0.0000265\left(\frac{H}{D}\right)^2\sqrt{\frac{W_oD}{Ht}}$$

Note: P.O.V. may be determined from chart in Figure 4-6 H and D are in feet; t is in inches.

$$V = C_hW_o$$

$$F = V$$

$$M_b = 2/3(FH)$$

Moment at any height h_j

$$M_a = F\left(\frac{2H}{3} - h_i\right)$$

Case 2: Nonuniform Vessels

Procedure for finding period of vibration, moments, and forces at various planes for nonuniform vessels.

A "nonuniform" vertical vessel is one that varies in diameter, thickness, or weight at different elevations. This procedure distributes the seismic forces and thus base shear, along the column in proportion to the weights of

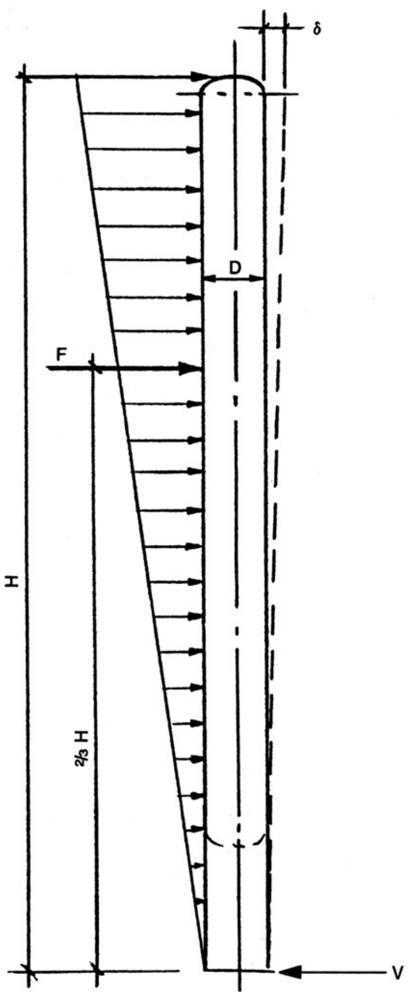


Figure 4-36. Typical dimensional data, forces, and loadings on a uniform vessel supported on a skirt ($\delta =$ deflection).

each section. The results are a more accurate and realistic distribution of forces and accordingly a more accurate period of vibration. The procedure consists of two main steps:

Step 1: *Determination of period of vibration (P.O.V.), T.* Divide the column into sections of uniform weight and diameter not to exceed 20% of the overall height. A uniform weight is calculated for each section. Diameter and thicknesses are taken into account through factors α and γ. Concentrated loads are handled as separate sections and not combined with other sections. Factor β will proportion effects of concentrated loads. The calculation form is completed for each section from left to right, then totaled to the bottom. These totals are used to determine T (P.O.V.) and the P.O.V. in turn is used to determine V and F_t.

Step 2: *Determination of forces, shears, and moments.* Again, the vessel is divided into major sections as in Step 1; however, longer sections should be further subdivided into even increments. For these calculations, sections should not exceed 10% of height. Remember, the moments and weights at each plane will be used in determining what thicknesses are required. It is convenient to work in 8 to 10 foot increments to match shell courses. Piping, trays, platforms, insulation, fireproofing, and liquid weights should be added into the weights of each section where they occur. Overall weights of sections are used in determining forces, not uniform weights. Moments due to eccentric loads are added to the overall moment of the column.

Notes for nonuniform vessels

1. Combine moments with corresponding weights at each section and use allowable stresses to determine required shell and skirt thicknesses at the elevation.
2. $\sum \omega \, \Delta \, \alpha$ and $W\beta/H$ are separate totals and are combined in computation of P.O.V.
3. $(D/10)^3$ is used in this expression if kips are used. Use $(D)^3$ if lb are used.
4. For vessels having a lower section several times the diameter of the upper portion and where the lower portion is short compared to the overall height, the P.O.V. can more accurately be determined bv finding the P.O.V. of the upper section alone (see Figure 4.38a).
5. For vessels where R/t is large in comparison to the supporting skirt, the P.O.V, calculated by this method may be overly conservative. More accurate methods may be employed (see Figure 4.38b).
6. Make sure to add moment due to any eccentric loads to total moment

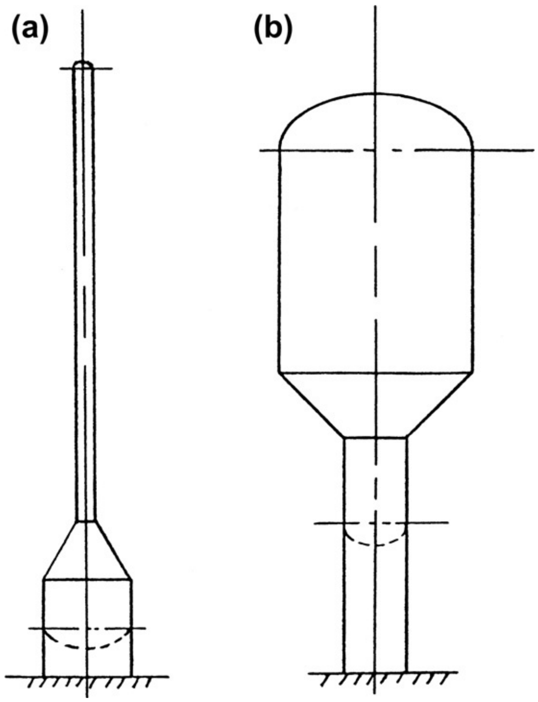
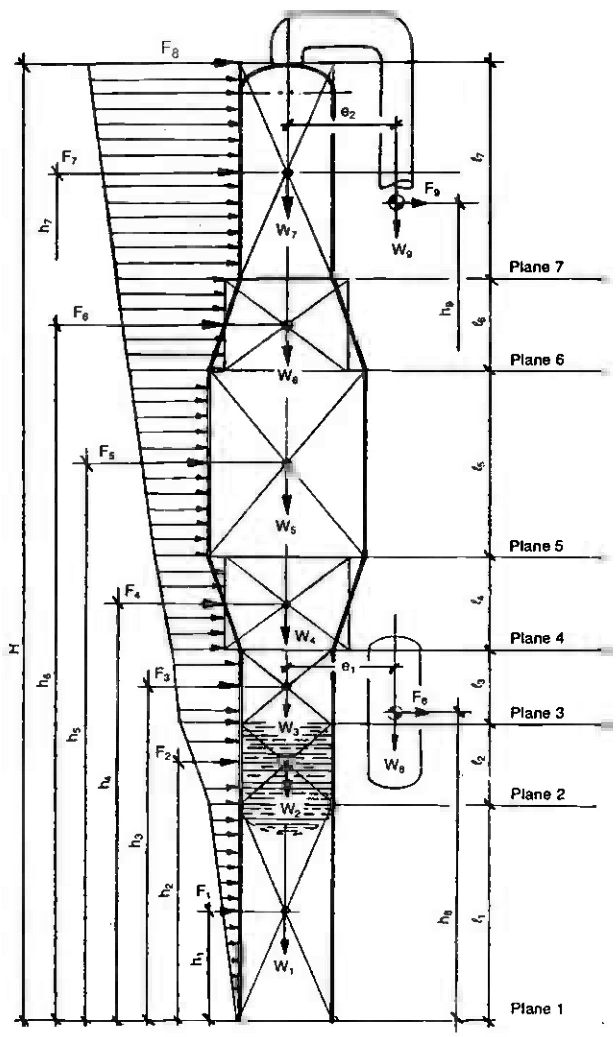


Figure 4-37. Nonuniform vessel illustrating a) Note 4 and b) Note 5.



$$h_x = h_{1,2,3,4,5\cdots}$$

$$W_x = W_{1,2,3,4,5\cdots}$$

$$F_x = F_{1,2,3,4,5\cdots}$$

$$w_x = w_{1,2,3,4,5\cdots}$$

$$M_x = M_{1,2,3,4,5\cdots}$$

$$\ell_x = \ell_{1,2,3,4,5\cdots}$$

$$w_x = \frac{W_x}{\ell_x}$$

$$F_x = \frac{V}{\Sigma W_x h_x}(W_x h_x)$$

$$M_i = \Sigma F_x(h_x - h_i)$$

Figure 4-38. Typical dimensional data, forces, and loadings on a nonuniform vessel supported on a skirt.

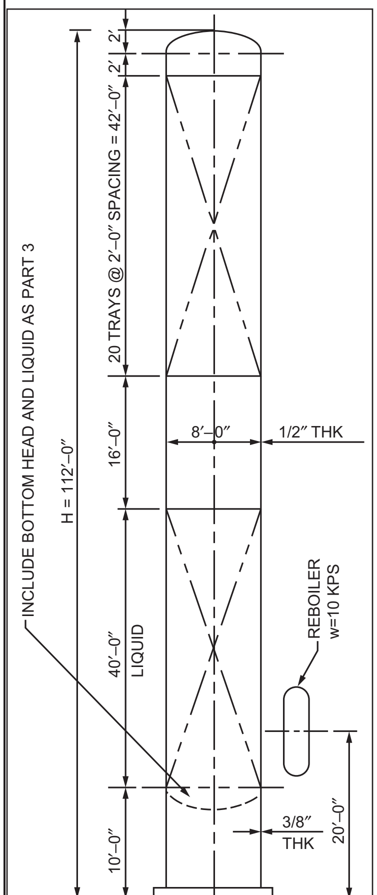
Step 1: PERIOD OF VIBRATION

	Part	ω or W k/ft	hx/H	α	Δα or β	ωΔα or Wβ/H	γ	Δγ	E(D/10)3tΔγ Note 3
			1.0	2.103			1.0		
			0	0			0		
						Σ =			Σ =

$$T = \left(\frac{H}{100}\right)^2 \sqrt{\frac{\Sigma\,\omega\Delta\alpha + \Sigma\,W\beta/H}{\Sigma E(D/10)^3 t\Delta\gamma}}$$

See Notes 2 and 3

Step 1: PERIOD OF VIBRATION EXAMPLE



Part	ω or W k/ft	hx/H 1.0	α 2.103	Δα or β	ωΔα or Wβ/H	γ 1.0	Δγ	E(D/10)3tΔγ Note 3
5	1.18			1.874	2.211		0.017	0.126
		0.59	0.229			0.983		
4	0.59			0.161	0.095		0.052	0.386
		0.45	0.068			0.931		
3	2.78			0.068	0.189		0.598	4.440
2	10 kips			0.024	0.002			
		0.09	0.00003			0.333		
1	0.875			0.00003	2.625E-05		0.333	1.854
		0	0			0		
					Σ = 2.497			Σ = 6.806

$$T = \left(\frac{H}{100}\right)^2 \sqrt{\frac{\sum \omega \Delta\alpha + \sum W\beta/H}{\sum E(D/10)^3 t \Delta\gamma}} \qquad = \left(\frac{112}{100}\right)^2 \sqrt{\frac{2.492 \quad +.002}{6.807}} = 0.759 \cdot \text{sec}$$

See Notes 2 and 3

Step 2: SHEAR AND MOMENTS

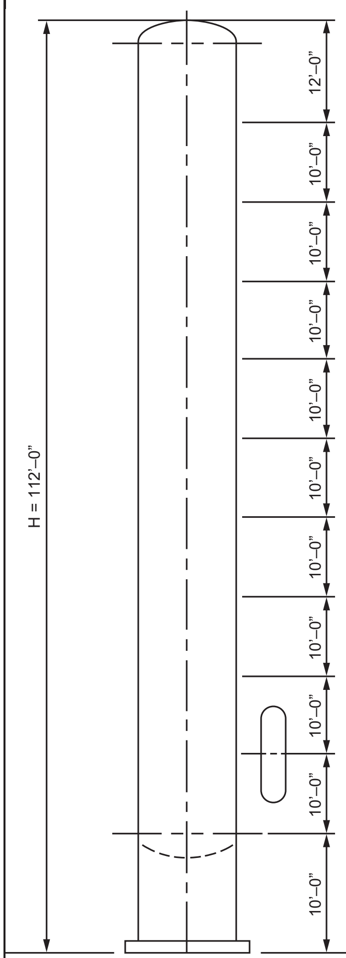
h_i (ft)	Part	W_x (kips)	h_x (ft)	W_xh_x (ft-kips)	F_x (kips)	V_i @ btm (kips)	M_i (kips)
		Σ = 0		Σ = 0			

$$F_x = \frac{V}{\sum W_x h_x^k}\left(W_x h_x^k\right)$$

$$M_i = F_x(h_x - h_i) + V_{i+1}(h_{i+1} - h_i) + M_{i+1}$$

k = 1 for structures with periods of 0.5 seconds or less
k = 2 for structures with periods of 2.5 seconds or more
k shall be linearly interpolated with perdiods between 0.5 and 2.5 seconds

Step 2: SHEAR AND MOMENTS EXAMPLE



h_i (ft)	Part	W_x (kips)	h_x (ft)	$W_x h_x$ (ft-kips)	F_x (kips)	V_i @ btm (kips)	M_i (kips)
112			112.0				
	12	14.16	106	1501	7.32		
100						7.32	43.92
	11	11.8	95	1121	5.47		
90						12.79	144.44
	10	11.8	85	1003	4.89		
80						17.68	296.76
	9	11.8	75	885	4.32		
70						21.99	495.11
	8	8.26	65	537	2.62		
60						24.61	728.13
	7	5.9	55	325	1.58		
50						26.19	982.15
	6	27.8	45	1251	6.10		
40						32.29	1274.58
	5	27.8	35	973	4.74		
30						37.04	1621.25
	4	27.8	25	695	3.39		
20	3	10	20	200	0.98		
						41.40	2008.58
	2	27.8	15	417	2.03		
10						43.44	2432.78
	1	8.75	5	44	0.21		
0						43.65	2868.21
		Σ = 194		Σ = 8951			

$$F_x = \frac{V}{\sum W_x h_x^k}\left(W_x h_x^k\right) \qquad F_x = \frac{43.65}{8951}\left(W_x h_x^k\right)$$

$$M_i = F_x(h_x - h_i) + V_{i+1}(h_{i+1} - h_i) + M_{i+1}$$

k = 1 for structures with periods of 0.5 seconds or less
k = 2 for structures with periods of 2.5 seconds or more
k shall be linearly interpolated with perdiods between 0.5 and 2.5 seconds

Longitudinal Stresses

In the following equations, D is in inches. The term "$48M_X$" is used for ft-lb or ft-kips. If in.-lb or in.-kips are used, then the term "$4M_X$" should be substituted where "$48M_X$" is used. The allowable stresses S_1E_1 or B may be substituted in the equations for t to determine or verify thickness at any elevation. Compare the stresses or thicknesses required at each elevation against the thickness required for circumferential stress due to internal pressure to determine which one will govern. If there is no external pressure condition, assume the maximum compression will occur when the vessel is not pressurized and the term $P_eD/4t$ will drop out.

$$\sigma_{xt} = \text{tension side} = \frac{P_iD}{4t} + \frac{48M_x}{\pi D^2 t} - \frac{W_h}{\pi Dt}$$

$$\sigma_{xc} = \text{compression side} = (-)\frac{P_eD}{4t} - \frac{48M_x}{\pi D^2 t} - \frac{W_h}{\pi Dt}$$

- *Allowable longitudinal stresses.*

  tension : $S_1E_1 =$

Elevation	M_x	W_n	D	t	Tension		Compression	
					S_1E_1	σ_{xt}	B	σ_{xc}

Table 4-20
Coefficients for determining period of vibration of free-standing cylindrical shells having varying cross sections and mass distribution

$\frac{h_x}{H}$	α	β	γ	$\frac{h_x}{H}$	α	β	γ	$\frac{h_x}{H}$	α	β	γ
1.00	2.103	8.347	1.000000	0.65	0.3497	2.3365	0.99183	0.30	0.010293	0.16200	0.7914
0.99	2.021	8.121	1.000000	0.64	0.3269	2.2240	0.99065	0.29	0.008769	0.14308	0.7776
0.98	1.941	7.898	1.000000	0.63	0.3052	2.1148	0.98934	0.28	0.007426	0.12576	0.7632
0.97	1.863	7.678	1.000000	0.62	0.2846	2.0089	0.98789	0.27	0.006249	0.10997	0.7480
0.96	1.787	7.461	1.000000	0.61	0.2650	1.9062	0.98630	0.26	0.005222	0.09564	0.7321
0.95	1.714	7.246	0.999999	0.60	0.2464	1.8068	0.98455	0.25	0.004332	0.08267	0.7155
0.94	1.642	7.037	0.999998	0.59	0.2288	1.7107	0.98262	0.24	0.003564	0.07101	0.6981
0.93	1.573	6.830	0.999997	0.58	0.2122	1.6177	0.98052	0.23	0.002907	0.06056	0.6800
0.92	1.506	6.626	0.999994	0.57	0.1965	1.5279	0.97823	0.22	0.002349	0.05126	0.6610
0.91	1.440	6.425	0.999989	0.56	0.1816	1.4413	0.97573	0.21	0.001878	0.04303	0.6413
0.90	1.377	6.227	0.999982	0.55	0.1676	1.3579	0.97301	0.20	0.001485	0.03579	0.6207
0.89	1.316	6.032	0.999971	0.54	0.1545	1.2775	0.97007	0.19	0.001159	0.02948	0.5992
0.88	1.256	5.840	0.999956	0.53	0.1421	1.2002	0.96688	0.18	0.000893	0.02400	0.5769
0.87	1.199	5.652	0.999934	0.52	0.1305	1.1259	0.96344	0.17	0.000677	0.01931	0.5536
0.86	1.143	5.467	0.999905	0.51	0.1196	1.0547	0.95973	0.16	0.000504	0.01531	0.5295
0.85	1.090	5.285	0.999867	0.50	0.1094	0.9863	0.95573	0.15	0.000368	0.01196	0.5044
0.84	1.038	5.106	0.999817	0.49	0.0998	0.9210	0.95143	0.14	0.000263	0.00917	0.4783
0.83	0.988	4.930	0.999754	0.48	0.0909	0.8584	0.94683	0.13	0.000183	0.00689	0.4512
0.82	0.939	4.758	0.999674	0.47	0.0826	0.7987	0.94189	0.12	0.000124	0.00506	0.4231
0.81	0.892	4.589	0.999576	0.46	0.0749	0.7418	0.93661	0.11	0.000081	0.00361	0.3940
0.80	0.847	4.424	0.999455	0.45	0.0678	0.8876	0.93097	0.10	0.000051	0.00249	0.3639
0.79	0.804	4.261	0.999309	0.44	0.0612	0.6361	0.92495	0.09	0.000030	0.00165	0.3327
0.78	0.762	4.102	0.999133	0.43	0.0551	0.5872	0.91854	0.08	0.000017	0.00104	0.3003
0.77	0.722	3.946	0.998923	0.42	0.0494	0.5409	0.91173	0.07	0.000009	0.00062	0.2669
0.76	0.683	3.794	0.998676	0.41	0.0442	0.4971	0.90448	0.06	0.000004	0.00034	0.2323
0.75	0.646	3.845	0.998385	0.40	0.0395	0.4557	0.89679	0.05	0.000002	0.00016	0.1966
0.74	0.610	3.499	0.998047	0.39	0.0351	0.4167	0.88884	0.04	0.000001	0.00007	0.1597
0.73	0.576	3.356	0.997656	0.38	0.0311	0.3801	0.88001	0.03	0.000000	0.00002	0.1218
0.72	0.543	3.217	0.997205	0.37	0.0275	0.3456	0.87088	0.02	0.000000	0.00000	0.0823
0.71	0.512	3.081	0.996689	0.36	0.0242	0.3134	0.86123	0.01	0.000000	0.00000	0.0418
0.70	0.481	2.949	0.998101	0.35	0.0212	0.2833	0.85105	0.	0.	0.	0.
0.69	0.453	2.820	0.995434	0.34	0.0185	0.2552	0.84032				
0.68	0.425	2.694	0.994681	0.33	0.0161	0.2291	0.82901				
0.67	0.399	2.571	0.993834	0.32	0.0140	0.2050	0.81710				
0.88	0.374	2.452	0.992885	0.31	0.0120	0.1826	0.80459				

Reprinted by permission of the Chevron Corp., San Francisco.

compression:

$$A = \frac{0.125t}{R_o}$$

B = from applicable material chart of ASME Code, Section II, Part D, Subpart 3.

Note: Joint efficiency for longitudinal seams in compression is 1.0.

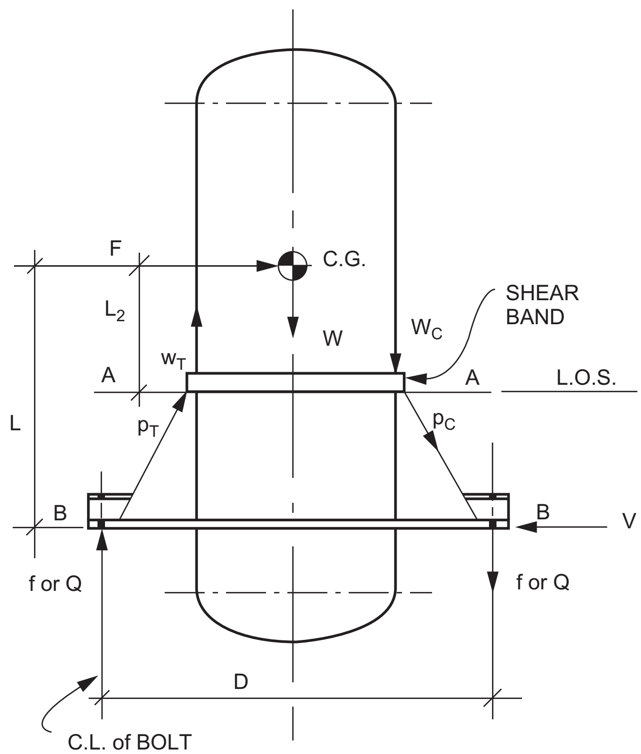
Notes

1. This procedure is for use in determining forces and moments at various planes of uniform and nonuniform vertical pressure vessels.
2. To determine the plate thickness required at any given elevation compare the moments from both wind and seismic at that elevation. The larger of the two should be used. Wind-induced moments may govern the longitudinal loading at one elevation, and seismic-induced moments may govern another.

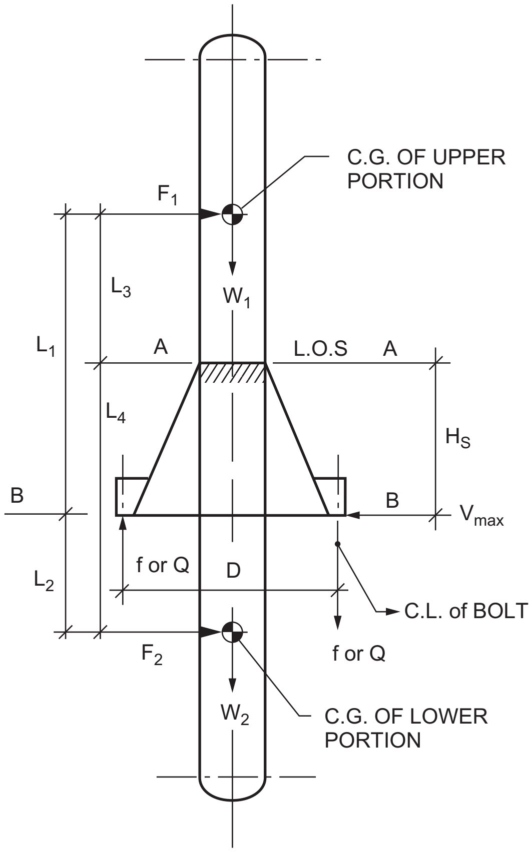
Procedure 4-9: Seismic Design – Vessel on Conical Skirt

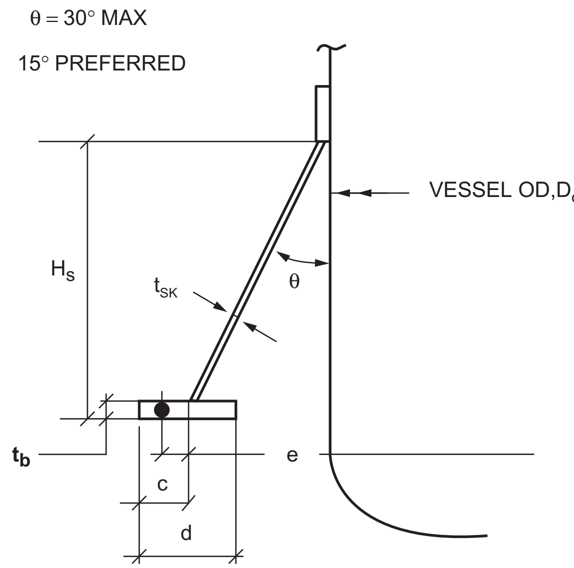
Nomenclature

A = ASME Code strain factor, dimensionless
A_b = Area of base plate supported on steel, in^2
A_t = Area required for one anchor bolt, in^2
A_S = Area of shear band, L_S X t_S, in^2
B_P = Allowable bearing pressure, PSI
D_O = OD of vessel shell, in
D_{SK} = OD of skirt at base plate, in
E = Modulus of elasticity, PSI
F_c = Allowable compressive stress, PSI
f = Load at support points, Lbs
f_P = Bearing pressure, PSI
F_T = Allowable stress, tension, PSI
F_y = Minimum specified yield strength of skirt at design temperature, PSI
F_1 or F_2 = Seismic load for upper or lower portion of vessel
$M_{AA \text{ or } BB}$ = Overturning moment due to earthquake, In-Lbs, at elevation A-A or B-B
M_b = Bending moment, In-Lbs
M_X or M_y = Internal bending moment in base plate, in-lbs
N = Number of support points
N_b = Number of anchor bolts
P = Design pressure, PSIG
p_T, p_C = Load at top of skirt, tension or compression, Lbs/in
Q = Load at support points, Lbs
R_m = Mean radius of shell, in
S = Shell allowable stress, tension, PSI
S_b = Allowable stress, anchor bolts, PSI
t = Thickness of shell, in
t_r = Thickness required, skirt, in
V = Base shear, Lbs
V_{max} = Greater of V_1 or V_2, Lbs
W = Weight, operating, Lbs
W_1 = Weight of vessel, insulation, piping, etc above LOS. Include weight of contents if contents are supported above the LOS. Do not include weight of skirt, Lbs
W_2 = Weight of vessel, insulation, piping, etc., below LOS. Include weight of contents if supported below the LOS. Do not include weight of skirt or base.
w_T, w_C = Uniform load in shell, tension or compression, Lbs/in
ΔT = Temperature differential in skirt; DT – 70° F
λ = Damping Factor
σ_{LT} = Longitudinal tension stress, skirt, PSI
σ_{LC} = Longitudinal compressive stress, skirt, PSI
$\sigma_{\Delta T}$ = Stress in skirt due to ΔT loading, PSI
σ_X = Longitudinal bending stress in shell, PSI
τ_r = Allowable shear stress in shear band, PSI
τ_w = Allowable shear stress in weld, PSI

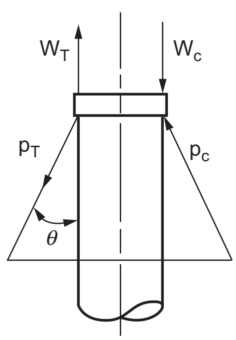


SIMPLE VESSEL DIAGRAM
SEE NOTE 1

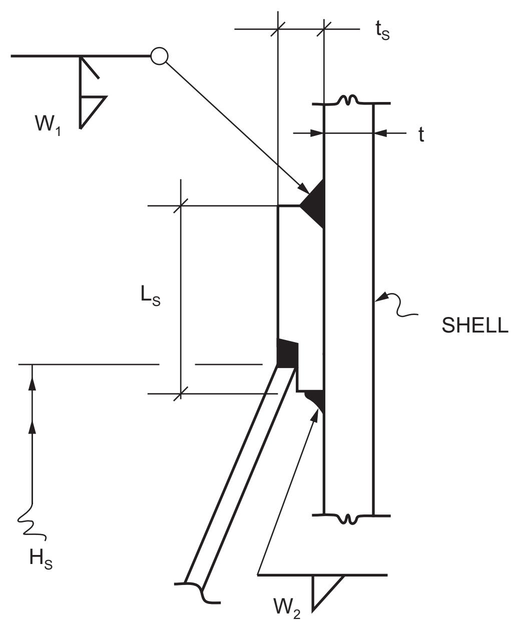




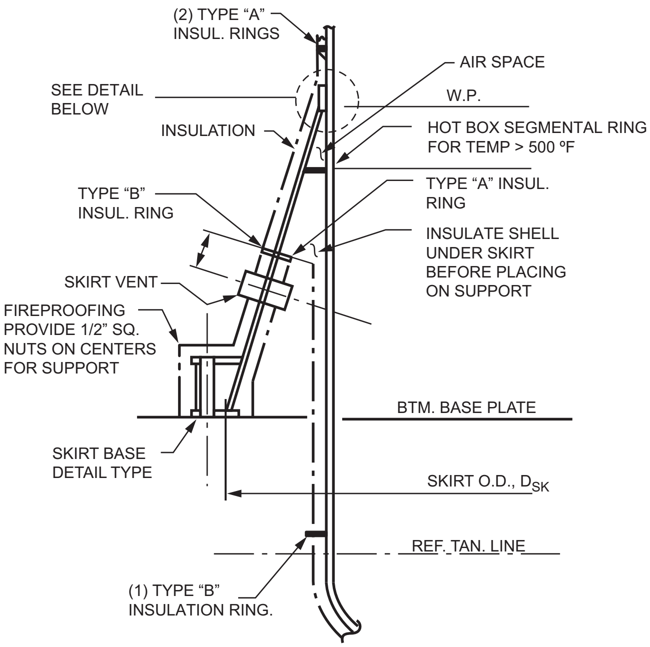
SKIRT DIMENSIONS



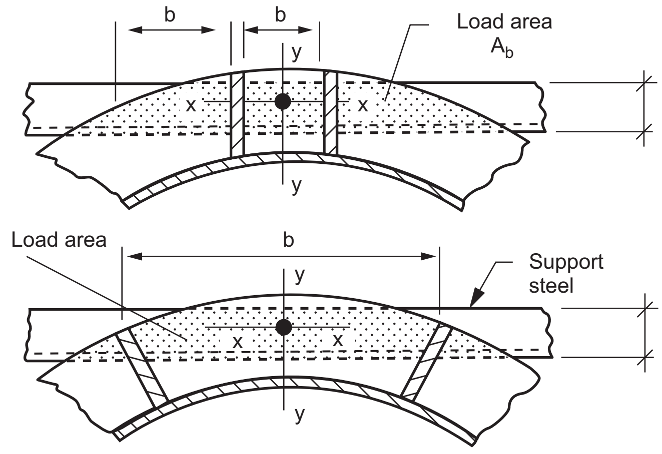
DETAIL OF FORCES



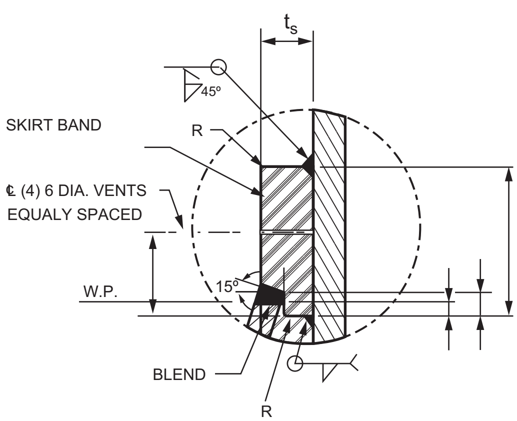
DIMENSIONS OF SHEAR BAND



SKIRT & BASE DETAILS



BASE PLATE LOADING



DETAIL OF SHEAR BAND

Table 4-21
Maximum bending moment in a bearing plate with gussets

a/b	$M_x \begin{bmatrix} x = .5b \\ y = \ell \end{bmatrix}$	$M_y \begin{bmatrix} x = .5b \\ y = 0 \end{bmatrix}$
0	0	(−).500 $B_p\ l^2$
.333	.0078 $B_p\ b^2$	(−).428 $B_p\ l^2$
.5	.0293 $B_p\ b^2$	(−).319 $B_p\ l^2$
.666	.0558 $B_p\ b^2$	(−).227 $B_p\ l^2$
1.0	.0972 $B_p\ b^2$	(−).119 $B_p\ l^2$
1.5	.1230 $B_p\ b^2$	(−).124 $B_p\ l^2$
2.0	.1310 $B_p\ b^2$	(−).125 $B_p\ l^2$
3.0−∞	.1330 $B_p\ b^2$	(−).125 $B_p\ l^2$

Reprinted by permission of John Wiley & Sons, Inc.
From Process Equipment Design, *Table 10.3 (See Note 2.)*

Calculation

Case 1: Simplified Approach (Note 1)

GIVEN:

$D =$ ________ $L =$ ________
$F =$ ________ $L_2 =$ ________
$W =$ ________

Calculate moments;

$M_{AA} = F\,L_2$

$M_{BB} = F\,L$

Case 2: Rigorous Approach (Note 2)

GIVEN:

$H_S =$ ________ $L_3 =$ ________
$L_4 =$ ________ $W_1 =$ ________
$W_2 =$ ________ $F_1 =$ ________
$F_2 =$ ________

$V_{max} =$ Greater of F_1 or F_2

$M_{max} =$ Greater of following;

$M_{AA} = F_1\,L_3$

Or $= F_2\,L_4$

$M_{BB} = (V_{max}\,H_S) + M_{max}$

$W = W_1 + W_2$

Design of Skirt

- Uniform loads in vessel at ELEV A-A;

$$w_T = \left[(-)W/(\pi\,D_O)\right] + \left[\left(4\,M_{AA}/\left(\pi\,D_O^2\right)\right]\right.$$
$$w_C = \left[(-)W/(\pi\,D_O)\right] - \left[\left(4\,M_{AA}/\left(\pi\,D_O^2\right)\right]\right.$$

- Find angle, θ, by layout or calculation;

$$X = .5\,[(D - 2\,e) - (D_O + 2\,t_S)]$$

$$\text{Tan}\,\theta = X/(H_S - t_b)$$

$\theta =$ ________

- Uniform load in skirt at ELEV A-A

$p_T = w_T/\text{Cos}\,\theta$
$p_C = w_C/\text{Cos}\,\theta$

- Allowable stress, skirt;

1. Compression, F_C
   Assume a thickness of skirt and calculate;

$A = (.125\,t_{SK})/(.5\,D_{SK})$
$F_C = (A\,E/2) < .5\,F_y$

2. Tension, F_T

$S =$ from ASME II(D) $< .66\,F_y$

$F_T = 1.2\,S$

- Thickness required, skirt, t_r

Tension; $t_r = p_T/F_T$
Comp; $t_r = p_C/F_C$
Use $t_{SK} =$ ________

- Stress due to ΔT

$$\sigma_{\Delta T} = \left[(48\,\Delta T)/(H_S - t_b)\right][D_O\,t_{SK}]^{1/2}$$

- Longitudinal stress in skirt due to loadings;

Tension; $\sigma_{LT} = (p_T/t_{SK}) + \sigma_{\Delta T}$
Comp; $\sigma_{LC} = (p_C/t_{SK}) + \sigma_{\Delta T}$

Shear Ring

- Allowable shear stresses;

Ring; $\tau_r = .7\,S$
Weld; $\tau_w = .4\,S$

- Minimum length of shear band, L_{min}

$L_{min} = w_C/\tau_r$

- Size fillet welds, w_1 and w_2

$w = w_1 + w_2$
$w = w_C/(.707\,\tau_w)$
Use $w_1 = w_2 =$ ________

- Thickness required for shear band, t_S

$t_S = 2\,w_1$
Use $t_S =$ ________

Base Plate

The base plate thickness depends on how the vessel is supported. The vessel can either have continuous support or partial support. Partial support describes a vessel supported on 4 or 8 points on steel in a structure. Continuous support describes a concrete table top where there is full width, 360° contact between the base plate and the support.

Case A: Full Support

- Maximum load, f
  Note: The maximum loading is assumed to occur at the bolt circle.
  $f = [W/\pi D] \pm [4 M_{BB}/\pi D^2]$
- Bearing pressure, f_P
  $f_P = f/d < B_P$
- Base plate thickness, t_b
  $t_b = C\left[(3 f_P)/(.6 F_y)\right]^{1/2}$

Case B: Partial Support

- Load Q
  $Q = W/N \pm M_{BB}/D$
- Bearing pressure, f_P
  $f_P = Q/A_b$
- Maximum bending moment, M_b, from Table 4-21
  $a/b =$
  $M_b =$ greater of M_X or M_y
- Thickness of base plate, t_b
  $t_b = \left[(6 M_b/.6 F_y)\right]^{1/2}$

Anchor Bolts

- Determine if anchor bolts are required due to uplift
  $N_b A_t = [(48 M_{BB}/D) - W][1/S_b]$

If N_b A_t is negative, then anchor bolts are not required. Use minimum size and maximum spacing for this case.
If N_b A_t is positive then anchor bolts are required.

- Area required, A_t
  $A_t = [(48 M_{BB}/D) - W][1/N_b S_b]$

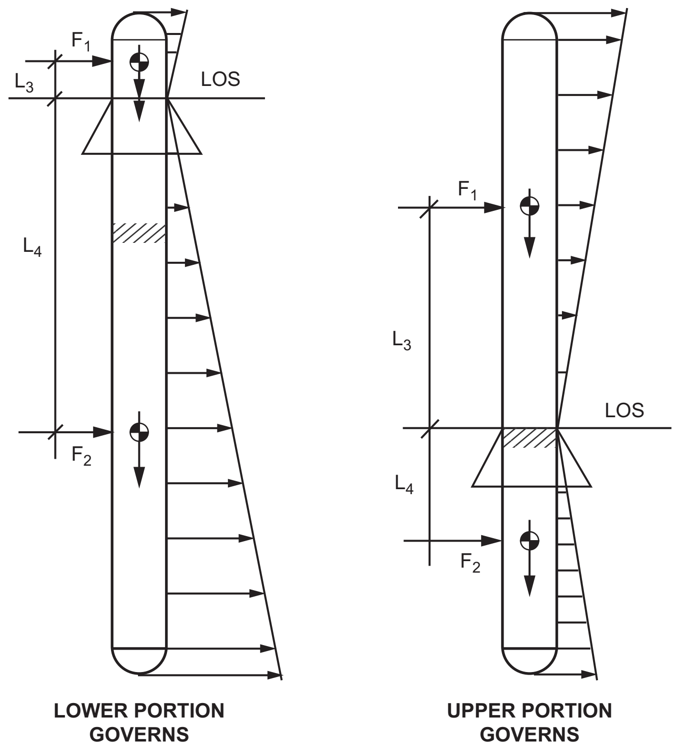
Use N_b = __________
d_b = __________

Longitudinal Stress in Shell due to Shear Band

- Cross sectional area of shear band, A_S
  $A_S = L_S t_S$
- Damping Factor, λ
  $\lambda = 1.285/(R_m t)^{1/2}$
- Bending moment in shell, M
  $M = [P/2\lambda^2][A_S/(A_S + t L_S + 2 t/\lambda)]$
- Longitudinal bending stress in shell, σ_X
  $\sigma_X = 6 M/t^2$

Notes

1. The "Simplified Approach" is valid for average size vessels where L/D < 5 and the support point is near the C.G. of the vessel. The simplified approach applies the full seismic force at the C.G. of the vessel.
2. The "Rigorous Approach" is for vessel where L/D > 5 or the vessel is supported near the top or bottom of the vessel. In such cases the simplified approach may not be adequate. In this case the vessel is divided into two parts; the upper and lower part. The division between the upper and lower part is the line of support.
3. A third approach, not shown here, would be to determine the loadings by determining the shear and moments at each weld plane for each part of the vessel. This procedure is illustrated in Procedure 4-8.
4. The upper weight, W_1, will produce a compressive force in the shell equal to W_1 / A, where A is the cross sectional area of the vessel.
5. The lower weight, W_2, will produce a tensile force in the vessel shell equal to W_2 / A. This would be additive to effects due to internal pressure.
6. The effects of the unbalanced inward (or outward) load on the shell to cone junction should be evaluated for circumferential membrane and bending stresses, as well as longitudinal bending stresses.



M_{max} = GREATER OF ...

$M_{AA} = F_1 L_3$

Or $F_2 L_4$

$M_{BB} = (V_{max} H_s) + M_{max}$

V_{max} = Greater of F_1 or F_2

Figure 4-39. Vessel supported on conical skirt (Influence of support positioning).

Procedure 4-10: Design of Horizontal Vessel on Saddles [1,3,14,15]

Notation

A_r = cross-sectional area of composite ring stiffener, in.^{2}
E = joint efficiency
E_1 = modulus of elasticity, psi
C_h = seismic factor
I_1 = moment of inertia of ring stiffener, in.^{4}
t_w = thickness of wear plate, in.
t_s = thickness of shell, in.
t_h = thickness of head, in.
Q = total load per saddle (including piping loads, wind or seismic reactions, platforms, operating liquid, etc.) lb
W_o = operating weight of vessel, lb
M_1 = longitudinal bending moment at saddles, in.-lb
M_2 = longitudinal bending moment at midspan, in.-lb
S = allowable stress, tension, psi
S_c = allowable stress, compression, psi
S_{1-14} = shell, head, and ring stresses, psi
K_{1-9} = coefficients
F_L = longitudinal force due to wind, seismic, expansion, contraction, etc., lb
F_T = transverse force, wind or seismic, lb
σ_x = longitudinal stress, internal pressure, psi
σ_ϕ = circumferential stress, internal pressure, psi

σ_e = longitudinal stress, external pressure, psi
σ_s = circumferential stress in stiffening ring, psi
σ_h = latitudinal stress in head due to internal pressure, psi
F_y = minimum yield stress, shell, psi
P = internal pressure, psi
P_e = external pressure, psi
K_s = pier spring rate,
μ = friction coefficient
y = pier deflection, in.

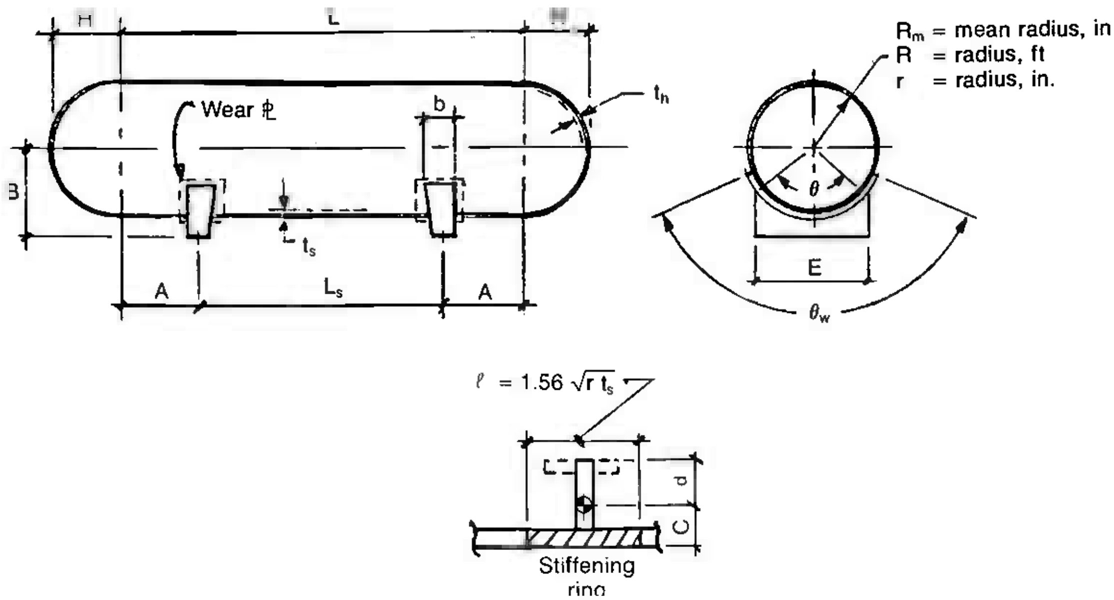


Figure 4-40. Typical dimensions for a horizontal vessel supported on two saddles.

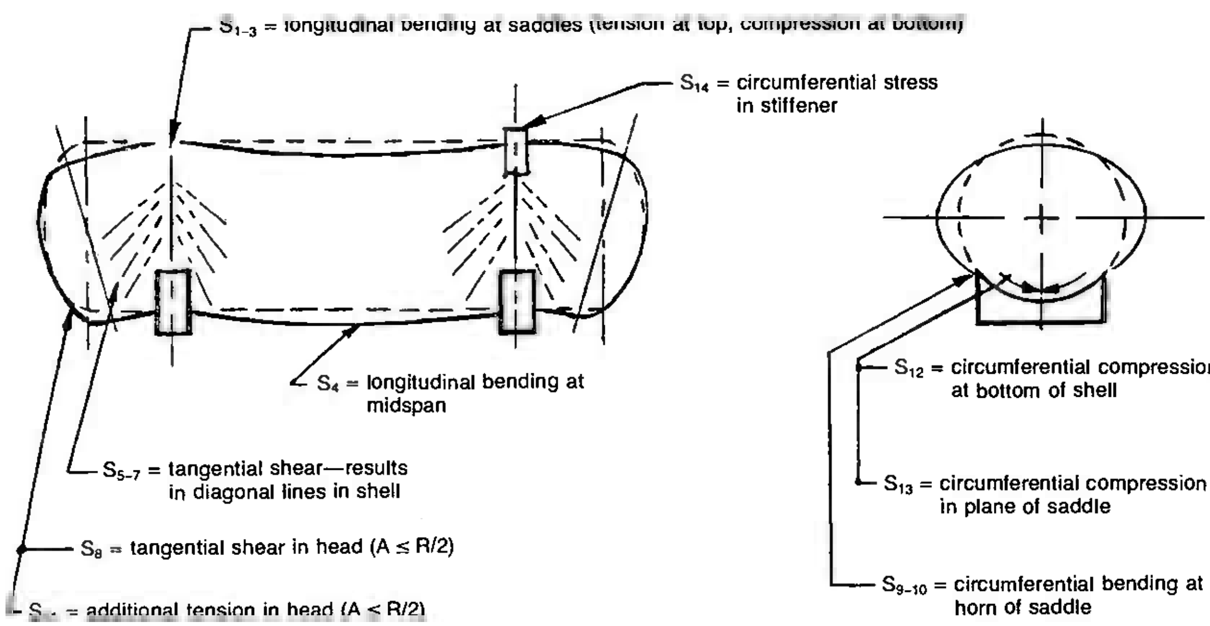
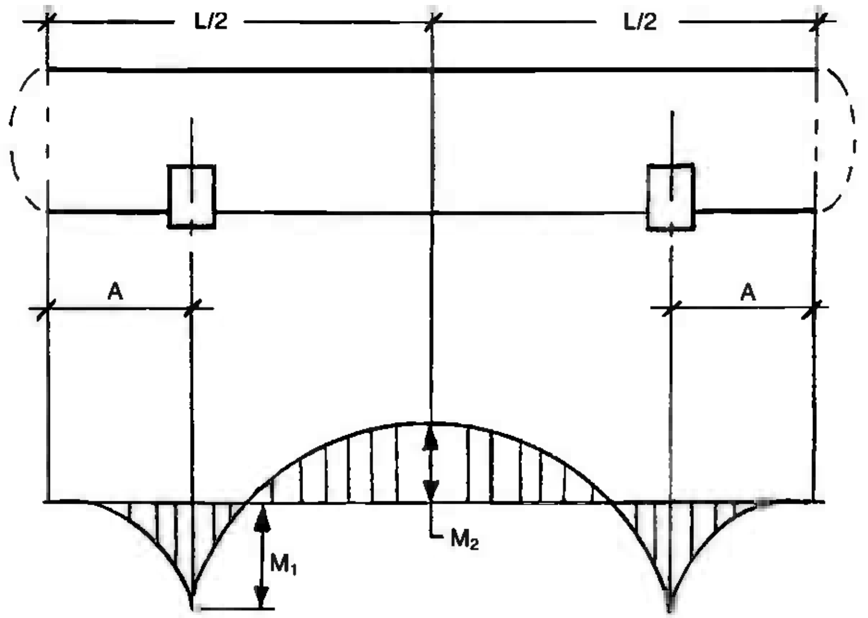


Figure 4-41. Stress diagram.



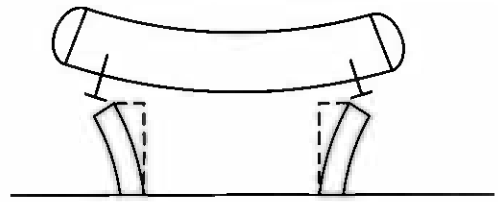
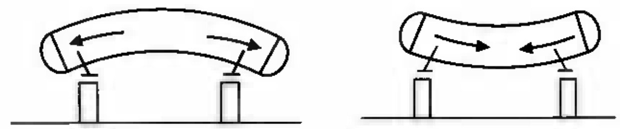
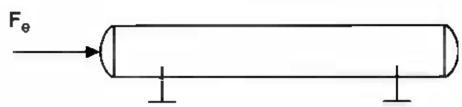
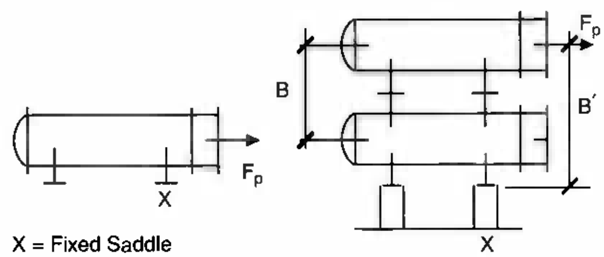
M_2 is negative for

- Hemi-heads.
- If any of the below conditions are exceeded.

M_2 is positive for

- Flat heads where $A/R < 0.707$.
- 100%–6% F&D heads where $A/R < 0.44$
- 2:1 S.E. heads where $A/R < 0.363$.

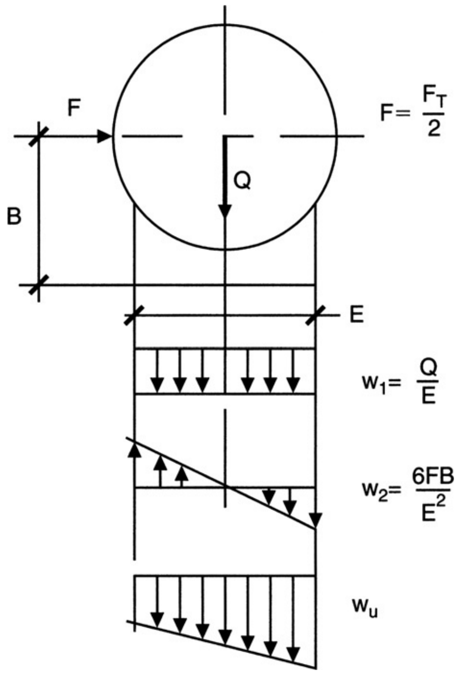
Figure 4-42. Moment diagram.

Longitudinal Forces, F_L	
Case 1: Pier Deflection $F_{L1} = \frac{K_s y}{2}$ $S_a = S$	
Case 2: Expansion/Contraction $F_{L2} = \mu Q_0$ $S_a = S$	
Case 3: Wind $F_{L3} = F_{wL} = A_f C_f G q_z$ $S_a = 1.33S$	
Case 4: Seismic $F_{L4} = F_e = C_h W_o$ $S_a = 1.33S$	
Case 5: Shipping/Transportation F_{L5} (See Chapter 10.) $S_a = 0.9F_y$	
Case 6: Bundle Pulling $F_{L6} = F_p$ $S_a = 0.9F_y$ Full load applies to fixed saddle only!	 X = Fixed Saddle

Note: For Cases 5 and 6. assume the vessel is cold and not pressurized.

Transverse Load: Basis for Equations

Method 1



- *Unit load at edge of base plate, w_u.*

  $$W_u = W_1 + W_2$$

- *Derivation of equation for w_2.*

  $$\sigma = \frac{M}{Z} \qquad M = FB \qquad Z = \frac{E^2}{6}$$

  Therefore

  $$\frac{M}{Z} = \frac{6FB}{E^2}$$

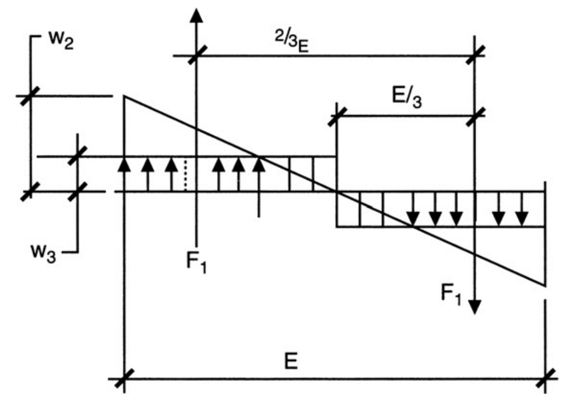
- *Equivalent total load Q_2.*

  $$Q_2 = w_u E$$

This assumes that the maximum load at the edge of the baseplate is uniform across the entire baseplate. This is very conservative, so the equation is modified as follows:

- *Using a triangular loading and 2/3 rule to develop a more realistic "uniform load"*

  $$F_1 = \frac{FB}{(2/3)E} = \frac{3FB}{2E}$$

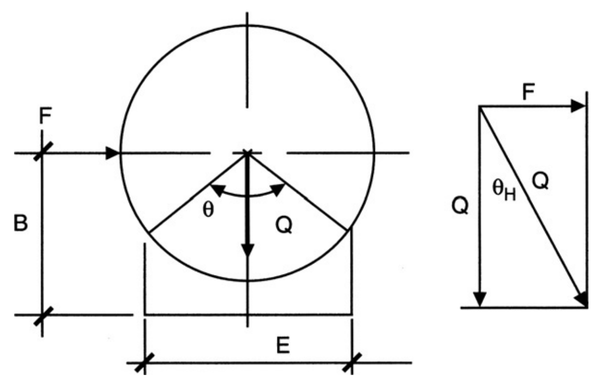


$$w_3 = \frac{3FB}{2E} \div \frac{E}{2} = \frac{6FB}{2E^2} = \frac{3FB}{E^2}$$

Therefore the total load, Q_F, due to force F is

$$Q_F = w_3 E = \frac{3FB}{E^2} E = \frac{3FB}{E}$$

Method 2



This method is based on the rationale that the load is no longer spread over the entire saddle but is shifted to one side.

- *Combined force, Q_2.*

  $$Q_2 = \sqrt{F^2 + Q^2}$$

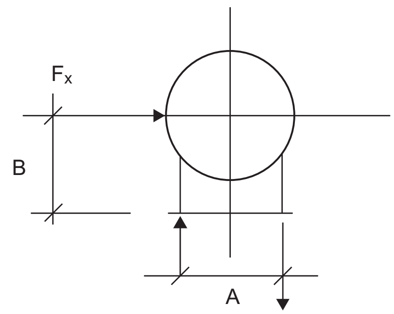
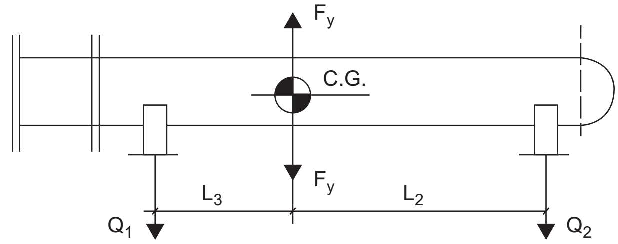
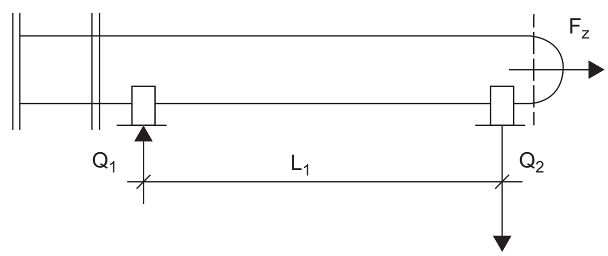
- *Angle, θ_H.*

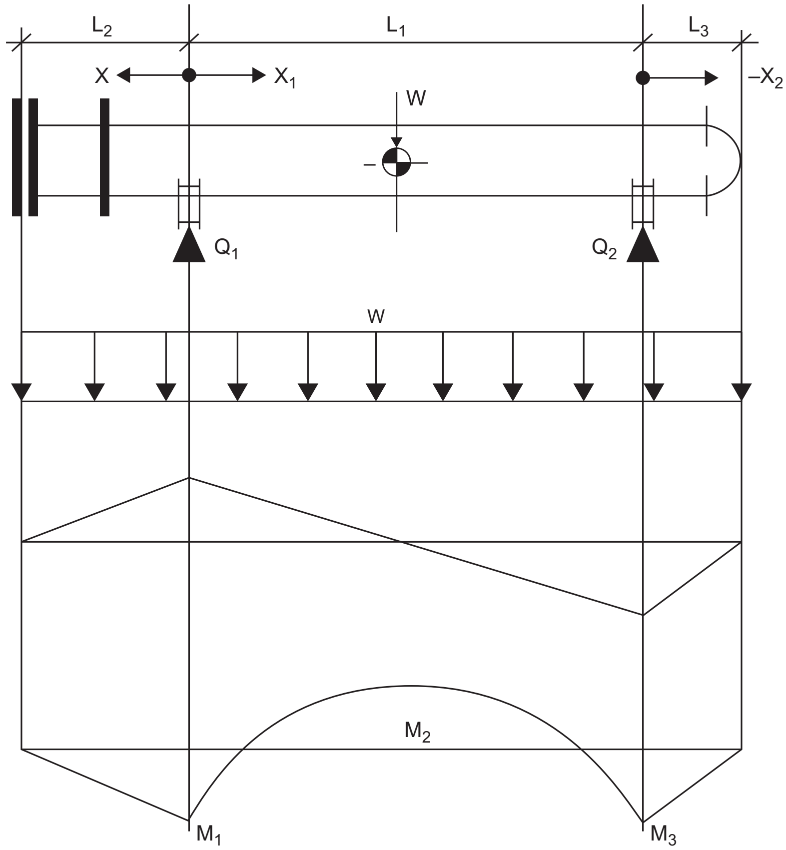
  $$\theta_H = \left(\arctan\right)\frac{F}{Q}$$

- *Modified saddle angle, θ_1.*

  $$\theta_1 = 2\left[\frac{\theta}{2}\right] - \theta_H$$

Saddle Reactions and Moments for Exchangers or Vessels with Offset Saddles

Due to...	Load per Saddle	Diagram
F_x	$Q_1 = \frac{W_s L_2}{L_1} + \frac{F_x B}{2A}$ $Q_2 = \frac{W_s L_3}{L_1} + \frac{F_x B}{2A}$	
F_y	$Q_1 = \frac{(W_s + F_y) L_2}{L_1}$ $Q_2 = \frac{(W_s + F_y) L_3}{L_1}$	
F_z	$Q_1 = \frac{W_S L_2}{L_1} + \frac{F_z B}{L_1}$ $Q_2 = \frac{W_S L_3}{L_1} + \frac{F_z B}{L_1}$	



Note: W = weight of vessel plus any impact factors

$$OAL = L_1 + L_2 + L_3 \quad w = \frac{W}{OAL}$$

$$Q_1 = \frac{w\left[(L_1 + L_2)^2 - L_3^2\right]}{2L_1}$$

$$Q_2 = W - Q_1$$

$$M_1 = \frac{wL_2^2}{2}$$

$$M_2 = Q_1\left(\frac{Q_1}{2w} - L_2\right)$$

$$M_3 = \frac{wL_3^2}{2}$$

$$M_x = \frac{w(L_2 - X)^2}{2}$$

$$M_{x1} = \frac{w(L_2 + X_1)^2}{2} - Q_1X_1$$

$$M_{x2} = \frac{w(L_3 - X_2)^2}{2}$$

Types of Stresses and Allowables

- *S_1 to S_4: longitudinal bending.*

  Tension: S_1, S_3, or $S_4 + \sigma_x < SE$

  Compression: S_2, S_3, or $S_4 - \sigma_e < S_c$

  where S_c = factor "B" or S or $t_sE_1/16r$ whichever is less.

1. Compressive stress is not significant where R_m/t <200 and the vessel is designed for internal pressure only.
2. When longitudinal bending at midspan is excessive, move saddles away from heads; however, do not exceed $A \geq 0.2$ L.
3. When longitudinal bending at saddles is excessive, move saddles toward heads.

4. If longitudinal bending is excessive at both saddles and midspan, add stiffening rings. If stresses are still excessive, increase shell thickness.

- *S_5 to $S_8 < 0.8S$: tangential shear.*
  1. Tangential shear is not combined with other stresses.
  2. If a wear plate is used, t_s may be taken as $t_s + t_w$, providing the wear plate extends R/10 above the horn of the saddle.
  3. If the shell is unstiffened, the maximum tangential shear stress occurs at the horn of the saddle.
  4. If the shell is stiffened, the maximum tangential shear occurs at the equator.
  5. When tangential shear stress is excessive, move saddles toward heads, $A \leq 0.5$ R, add rings, or increase shell thickness.
  6. When stiffening rings are used, the shell-to-ring weld must be designed to be adequate to resist the tangential shear as follows:

$$S_t = \frac{Q}{\pi r} : \frac{\text{lb}}{\text{in. circumference}} < \frac{\text{allowable shear}}{\text{in. of weld}}$$

- *$S_{11} + \sigma_h < 1.25$ SE: additional stress in head.*
  1. S_{11} is a shear stress that is additive to the hoop stress in the head and occurs whenever the saddles are located close to the heads, $A \leq 0.5$ R. Due to their close proximity the shear of the saddle extends into the head.
  2. If stress in the head is excessive, move saddles away from heads, increase head thickness, or add stiffening rings.
- *S_9 and $S_{10} < 1.5$ S and $0.9F_y$: circumferential bending at horn of saddle.*
  1. If a wear plate is used, t_s may be taken as $t_s + t_w$ providing the wear plate extends R/10 above the horn of the saddle. Stresses must also be checked at the top of the wear plate.
  2. If stresses at the horn of the saddle are excessive:
     a. Add a wear plate.
     b. Increase contact angle θ.
     c. Move saddles toward heads, $A < R$.
     d. Add stiffening rings.
- *$S_{12} < 0.5F_y$ or 1.5 S: circumferential compressive stress.*
  1. If a wear plate is used, t_s may be taken as $t_s + t_w$, providing the width of the wear plate is at least $b + 1.56\sqrt{rt_s}$.
  2. If the shell is unstiffened the maximum stress occurs at the horn of the saddle.
  3. If the shell is stiffened the maximum hoop compression occurs at the bottom of the shell.
  4. If stresses are excessive add stiffening rings.
- *$(+)S_{13} + \sigma_\phi < 1.5$ S: circumferential tension stress—shell stiffened.*
- *$(-)S_{13} - \sigma_s < 0.5F_y$: circumferential compression stress—shell stiffened.*
- *$(-)S_{14} - \sigma_s < 0.9F_y$: circumferential compression stress in stiffening ring.*

Procedure for Locating Saddles

Trial 1: Set A = 0.2 L and $\theta = 120°$ and check stress at the horn of the saddle, S_9 or S_{10}. This stress will govern for most vessels except for those with large L/R ratios.

Trial 2: Increase saddle angle θ to 150° and recheck stresses at horn or saddle, S_9 or S_{10}.

Trial 3: Move saddles near heads (A = R/2) and return θ to 120°. This will take advantage of stiffness provided by the heads and will also induce additional stresses in the heads. Compute stresses S_4, S_8, and S_9 or S_{10}. A wear plate may be used to reduce the stresses at the horn or saddle when the saddles are near the heads (A < R/2) and the wear plate extends R/10 above the horn of the saddle.

Trial 4: Increase the saddle angle to 150° and recheck stresses S_4, S_8, and S_9 or S_{10}. Increase the saddle angle progressively to a maximum of 168° to reduce stresses.

Trial 5: Move saddles to A = 0.2L and $\theta = 120°$ and design ring stiffeners in the plane of the saddles using the equations for S_{13} and S_{14} (see Note 7).

Total Saddle Reaction Forces, Q.

Q = greater of Q_1 or Q_2

Longitudinal, Q_1

$$Q_1 = \frac{W_o}{2} + \frac{F_L B}{L_s}$$

Transverse, Q_2

$$Q_2 = \frac{W_o}{2} + \frac{3F_t B}{E}$$

Shell Stresses

There are 14 main stresses to be considered in the design of a horizontal vessel on saddle supports:

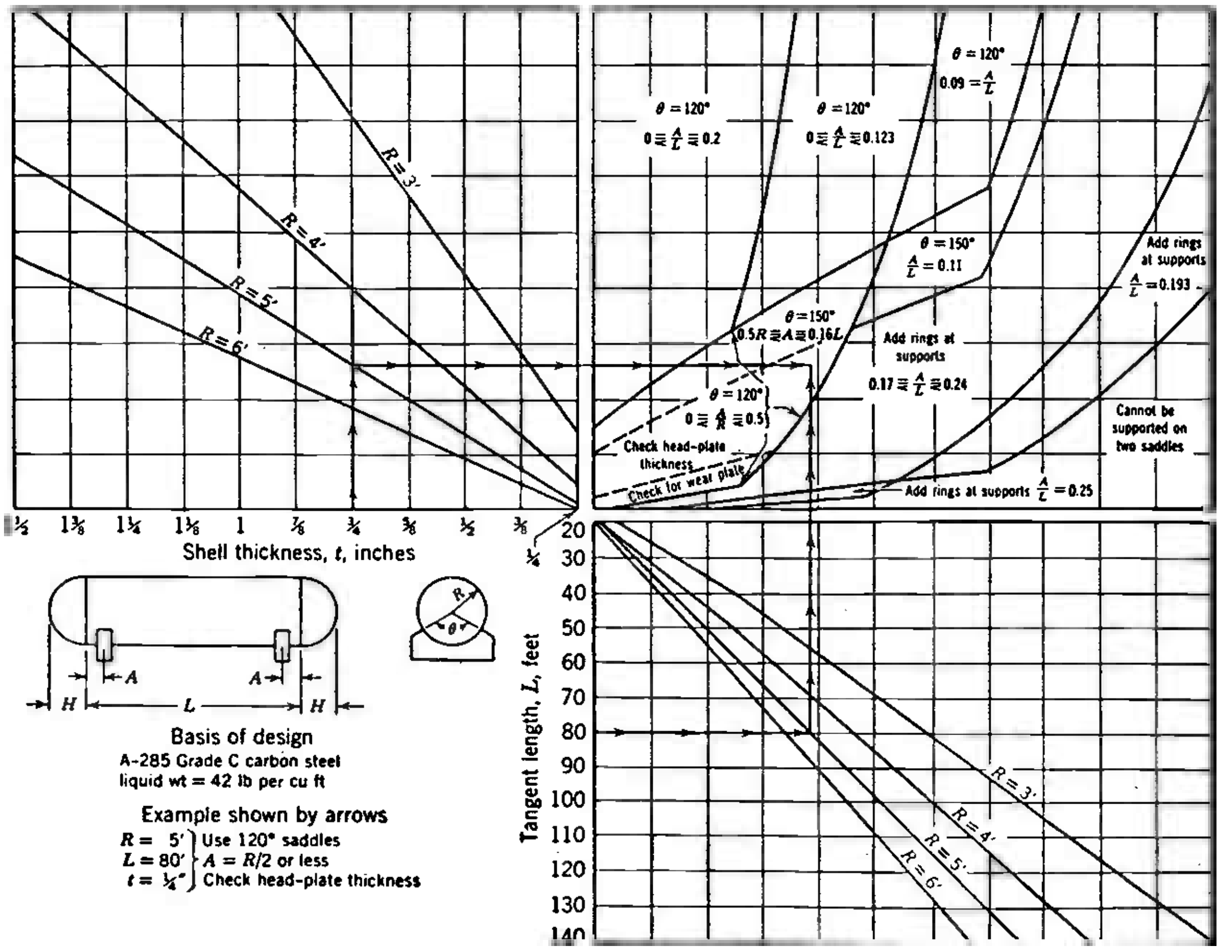


Figure 4-43. Chart for selection of saddles for horizontal vessels. *Reprinted by permission of the American Welding Society.*

S_1 = longitudinal bending at saddles without stiffeners, tension

S_2 = longitudinal bending at saddles without stiffeners, compression

S_3 = longitudinal bending at saddles with stiffeners

S_4 = longitudinal bending at midspan, tension at bottom, compression at top

S_5 = tangential shear—shell stiffened in plane of saddle

S_6 = tangential shear—shell not stiffened, $A > R/2$

S_7 = tangential shear—shell not stiffened except by heads, $A \leq R/2$

S_8 = tangential shear in head—shell not stiffened, $A \leq R/2$

S_9 = circumferential bending at horn of saddle—shell not stiffened, $L \geq 8R$

S_{10} = circumferential bending at horn of saddle—shell not stiffened, $L < 8R$

S_{11} = additional tension stress in head, shell not stiffened, $A \leq R/2$

S_{12} = circumferential compressive stress—stiffened or not stiffened, saddles attached or not

S_{13} = circumferential stress in shell with stiffener in plane of saddle

S_{14} = circumferential stress in ring stiffener

Longitudinal Bending

- *S_1, longitudinal bending at saddles—without stiffeners, tension.*

$$M_1 = 6Q\left[\frac{8AH + 6A^2 - 3R^2 + 3H^2}{3L + 4H}\right]$$

$$S_1 = (+)\frac{M_1}{K_1 r^2 t_s}$$

- *S_2, longitudinal bending at saddles—without stiffeners, compression.*

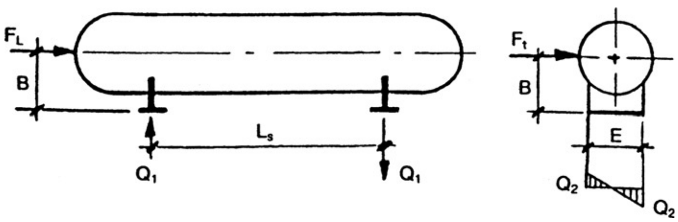


Figure 4-44. Saddle reaction forces.

$$S_2 = (-)\frac{M_1}{K_7 r^2 t_s}$$

- *S_3, longitudinal bending at saddles—with stiffeners.*

$$S_3 = (\pm)\frac{M_1}{\pi r^2 t_s}$$

- *S_4, longitudinal bending at midspan.*

$$M_2 = 3Q\left[\frac{3L^2 + 6R^2 - 6H^2 - 12AL - 16AH}{3L + 4H}\right]$$

$$S_4 = (\pm)\frac{M_2}{\pi r^2 t_s}$$

Tangential Shear

- *S_5, tangential shear—shell stiffened in the plane of the saddle.*

$$S_5 = \frac{Q}{\pi r t_s}\left[\frac{L - 2A}{L + \frac{4}{3}H}\right]$$

- *S_6, tangential shear—shell not stiffened, $A > 0.5R$.*

$$S_6 = \frac{K_2 Q}{r t_s}\left[\frac{L - 2A}{L + \frac{4}{3}H}\right]$$

- *S_7, tangential shear—shell not stiffened, $A \leq 0.5R$.*

$$S_7 = \frac{K_3 Q}{r t_s}$$

- S_8, *tangential shear in head—shell not stiffened, $A \leq 0.5R$.*

$$S_8 = \frac{K_3 Q}{r t_h}$$

Note: If shell is stiffened or $A > 0.5R$, $S_8 = 0$.

Circumferential Bending

- *S_9, circumferential bending at horn of saddle—shell not stiffened ($L \geq 8R$).*

$$S_9 = (-)\frac{Q}{4t_s(b + 1.56\sqrt{rt_s})} - \frac{3K_6 Q}{2t_s^2}$$

Note: $t_s = t_s + t_w$ and $t_s^2 = t_s^2 + t_w^2$ only if $A \leq 0.5R$ and wear plate extends R/10 above horn of saddle.

- *S_{10}, circumferential bending at horn of saddle—shell not stiffened ($L < 8R$).*

$$S_{10} = (-)\frac{Q}{4t_s(b + 1.56\sqrt{rt_s})} - \frac{12K_6 QR}{L t_s^2}$$

Note: Requirements for t_s are same as for S_9.

Additional Tension Stress in Head

- *S_{11}, additional tension stress in head—shell not stiffened, $A \leq 0.5R$.*

$$S_{11} = \frac{K_4 Q}{r t_h}$$

Note: If shell is stiffened or $A > 0.5R$, $S_{11} = 0$.

Circumferential Tension/Compression

- *S_{12}, circumferential compression.*

$$S_{12} = (-)\frac{K_5 Q}{t_s(b + 1.56\sqrt{rt_s})}$$

Note: ts = ts + tw only if wear plate is attached to shell and width of wear plate is a minimum of $b + 1.56\sqrt{rt_s}$.

- *S_{13}, circumferential stress in shell with stiffener (see Note 8).*

$$S_{13} = (-)\frac{K_8 Q}{A_r} \pm \frac{K_9 QrC}{I_1}$$

Note: Add second expression if vessel has an internal stiffener, subtract if vessel has an external stiffener.

- *S_{14}, circumferential compressive stress in stiffener (see Note 8).*

$$S_{14} = (-)\frac{K_8 Q}{A_r} - \frac{K_9 Qrd}{I_1}$$

Pressure Stresses

$$\sigma_x = \frac{PR_m}{2t_s}$$

$$\sigma_\phi = \frac{PR_m}{t_s}$$

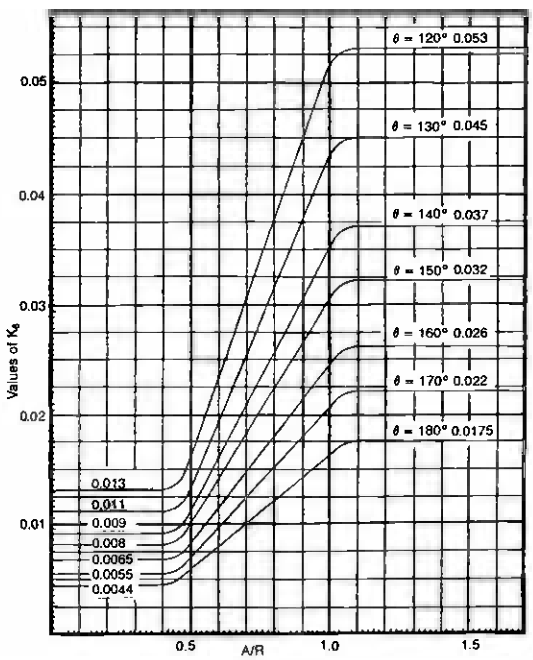
$$\sigma_e = \frac{P_e R_m}{2t_s}$$

$$\sigma_s = \frac{PIR_m}{A_r}$$

$\sigma_h = \sigma_\phi$, maximum circumferential stress in head is equal to hoop stress in shell

Combined Stresses

Tension		Compression	
Stress	**Allowable**	**Stress**	**Allowable**
$S_1 + \sigma_x$	SE =	$-S_2 - \sigma_e$	S_c =
$S_3 + \sigma_x$	SE =	$-S_3 - \sigma_e$	S_c =
$S_4 + \sigma_x$	SE =	$-S_4 - \sigma_e$	S_c =
$S_{11} + \sigma_h$	1.25SE =	$-S_{13} - \sigma_s$	$0.5F_y$ =
$S_{13} + \sigma_\phi$	1.5SE =	$-S_{14} - \sigma_s$	$0.9F_y$ =



Contact Angle θ	K_1*	K_2	K_3	K_4	K_5	K_7	K_8	K_9	Contact Angle θ	K_1*	K_2	K_3	K_4	K_5	K_7	K_8	K_9
120	0.335	1.171	0.880	0.401	0.760	0.603	0.340	0.053	152	0.518	0.781	0.466	0.289	0.669	0.894	0.298	0.03[illegible]
122	0.345	1.139	0.846	0.393	0.753	0.618	0.338	0.051	154	0.531	0.763	0.448	0.283	0.665	0.913	0.296	0.030
124	0.355	1.108	0.813	0.385	0.746	0.634	0.336	0.050	156	0.544	0.746	0.430	0.278	0.661	0.933	0.294	0.028
126	0.366	1.078	0.781	0.377	0.739	0.651	0.334	0.048	158	0.557	0.729	0.413	0.272	0.657	0.954	0.292	0.027
128	0.376	1.050	0.751	0.369	0.732	0.669	0.332	0.047	160	0.571	0.713	0.396	0.266	0.654	0.976	0.290	0.026
130	0.387	1.022	0.722	0.362	0.726	0.689	0.330	0.045	162	0.585	0.698	0.380	0.261	0.650	0.994	0.286	0.025
132	0.398	0.996	0.694	0.355	0.720	0.705	0.328	0.043	164	0.599	0.683	0.365	0.256	0.647	1.013	0.282	0.024
134	0.409	0.971	0.667	0.347	0.714	0.722	0.326	0.042	166	0.613	0.668	0.350	0.250	0.643	1.033	0.278	0.024
136	0.420	0.946	0.641	0.340	0.708	0.740	0.324	0.040	168	0.627	0.654	0.336	0.245	0.640	1.054	0.274	0.023
138	0.432	0.923	0.616	0.334	0.702	0.759	0.322	0.039	170	0.642	0.640	0.322	0.240	0.637	1.079	0.270	0.022
140	0.443	0.900	0.592	0.327	0.697	0.780	0.320	0.037	172	0.657	0.627	0.309	0.235	0.635	1.097	0.266	0.02[illegible]
142	0.455	0.879	0.569	0.320	0.692	0.796	0.316	0.036	174	0.672	0.614	0.296	0.230	0.632	1.116	0.262	0.020
144	0.467	0.858	0.547	0.314	0.687	0.813	0.312	0.035	176	0.0687	0.601	0.283	0.225	0.629	1.137	0.258	0.019
146	0.480	0.837	0.526	0.308	0.682	0.831	0.308	0.034	178	0.702	0.589	0.271	0.220	0.627	1.158	0.254	0.018
148	0.492	0.818	0.505	0.301	0.678	0.853	0.304	0.033	180	0.718	0.577	0.260	0.216	0.624	1.183	0.250	0.017
150	0.505	0.799	0.485	0.295	0.673	0.876	0.300	0.032									

K_1 = 3.14 if the shell is stiffened by ring or head ($A < R/2$)

Figure 4-45. Coefficients.

Table 4-22
Coefficients for Zick's analysis (angles 80° to 120°)

SADDLE ANGLE θ	K_1	K_2	K_3	K_4	K_5	A/R $\leq$ 0.5 K_6	A/R $\geq$ 1.0 K_6	K_7	K_8	K_9
80	0.1711	2.2747	2.0419	0.6238	0.9890	0.0237	0.0947	0.3212	0.3592	−0.0947
81	0.1744	2.2302	1.9956	0.6163	0.9807	0.0234	0.0934	0.3271	0.3592	0.0934
82	0.1777	2.1070	1.9506	0.6090	0.9726	0.0230	0.0922	0.3331	0.3593	0.0922
83	0.1811	2.1451	1.9070	0.6018	0.9646	0.0227	0.0910	0.3391	0.3593	0.0910
84	0.1845	2.1044	1.8645	0.5947	0.9568	0.0224	0.0897	0.3451	0.3593	0.0897
85	0.1879	2.0648	1.8233	0.5877	0.9492	0.0221	0.0885	0.3513	0.3593	0.0885
86	0.1914	2.0264	1.7831	0.5808	0.9417	0.0218	0.0873	0.3575	0.3592	0.0873
87	0.1949	1.9891	1.7441	0.5741	0.9344	0.0215	0.0861	0.3637	0.3591	0.0861
88	0.1985	1.9528	1.7061	0.5675	0.9273	0.0212	0.0849	0.3700	0.3590	0.0849
89	0.2021	1.9175	1.6692	0.5610	0.9203	0.0209	0.0838	0.3764	0.3588	0.0830
90	0.2057	1.8832	1.6332	0.5546	0.9134	0.0207	0.0826	0.3828	0.3586	0.0826
91	0.2094	1.8497	1.5981	0.5483	0.9067	0.0204	0.0815	0.3893	0.3584	0.0815
92	0.2132	1.8172	1.5640	0.5421	0.9001	0.0201	0.0803	0.3959	0.3582	0.0803
93	0.2169	1.7856	1.5308	0.5360	0.8937	0.0198	0.0792	0.4025	0.3579	0.0792
94	0.2207	1.7548	1.4984	0.5300	0.8874	0.0195	0.0781	0.4092	0.3576	0.0781
95	0.2246	1.7247	1.4668	0.5241	0.8812	0.0192	0.0770	0.4160	0.3573	0.0770
96	0.2285	1.6955	1.4360	0.5183	0.8751	0.0190	0.0759	0.4228	0.3569	0.0759
97	0.2324	1.6670	1.4060	0.5125	0.8692	0.0187	0.0748	0.4296	0.3565	0.0748
98	0.2364	1.6392	1.3767	0.5069	0.8634	0.0184	0.0737	0.4366	0.3561	0.0737
99	0.2404	1.6122	1.3482	0.5013	0.8577	0.0182	0.0727	0.4436	0.3557	0.0727
100	0.2445	1.5858	1.3203	0.4959	0.8521	0.0179	0.0716	0.4506	0.3552	0.0716
101	0.2486	1.5600	1.2931	0.4905	0.8466	0.0176	0.0706	0.4577	0.3547	0.0706
102	0.2528	1.5349	1.2666	0.4852	0.8412	0.0174	0.0696	0.4649	0.3542	0.0696
103	0.2570	1.5104	1.2407	0.4799	0.8359	0.0171	0.0686	0.4721	0.3536	0.0686
104	0.2612	1.4865	1.2154	0.4748	0.8308	0.0169	0.0675	0.4794	0.3531	0.0675
105	0.2655	1.4631	1.1907	0.4697	0.8257	0.0166	0.0666	0.4868	0.3525	0.0666
106	0.2698	1.4404	1.1665	0.4647	0.8207	0.0164	0.0656	0.4942	0.3518	0.0656
107	0.2742	1.4181	1.1429	0.4597	0.8159	0.0161	0.0646	0.5017	0.3512	0.0646
108	0.2786	1.3964	1.1199	0.4549	0.8111	0.0159	0.0636	0.5092	0.3505	0.0636
109	0.2830	1.3751	1.0974	0.4500	0.8064	0.0157	0.0627	0.5168	0.3498	0.0627
110	0.2875	1.3544	1.0753	0.4453	0.8018	0.0154	0.0617	0.5245	0.3491	0.0617
111	0.2921	1.3341	1.0538	0.4406	0.7973	0.0152	0.0608	0.5322	0.3483	0.0608
112	0.2966	1.3143	1.0327	0.4360	0.7928	0.0150	0.0599	0.5400	0.3475	0.0599
113	0.3013	1.2949	1.0121	0.4314	0.7885	0.0147	0.0590	0.5478	0.3467	0.0590
114	0.3059	1.2760	0.9920	0.4269	0.7842	0.0145	0.0581	0.5557	0.3459	0.0581
115	0.3107	1.2575	0.9723	0.4225	0.7800	0.0143	0.0572	0.5636	0.3451	0.0572
116	0.3154	1.2394	0.9530	0.4181	0.7759	0.0141	0.0563	0.5717	0.3442	0.0563
117	0.3202	1.2216	0.9341	0.4137	0.7719	0.0139	0.0554	0.5797	0.3433	0.0554
118	0.3251	1.2043	0.9157	0.4095	0.7680	0.0136	0.0546	0.5878	0.3424	0.0546
119	0.3300	1.1873	0.8976	0.4052	0.7641	0.0134	0.0537	0.5960	0.3414	0.0537
120	0.3349	1.1707	0.8799	0.4011	0.7603	0.0132	0.0529	0.6043	0.3405	0.0529
θ SADDLE ANGLE	K_1	K_2	K_3	K_4	K_5	K_6 A/R $\leq$ 0.5	K_6 A/R $\geq$ 1.0	K_7	K_8	K_9

Notes:

1. These coefficients are derived from Zick's equations.
2. The ASME Code does not recommend the use of saddles with an included angle, θ, less than 120°. Therefore the values in this table should be used for very small-diameter vessels or to evaluate existing vessels built prior to this ASME recommendation.
3. Values of K_6 for A/R ratios between 0.5 and 1 can be interpolated.

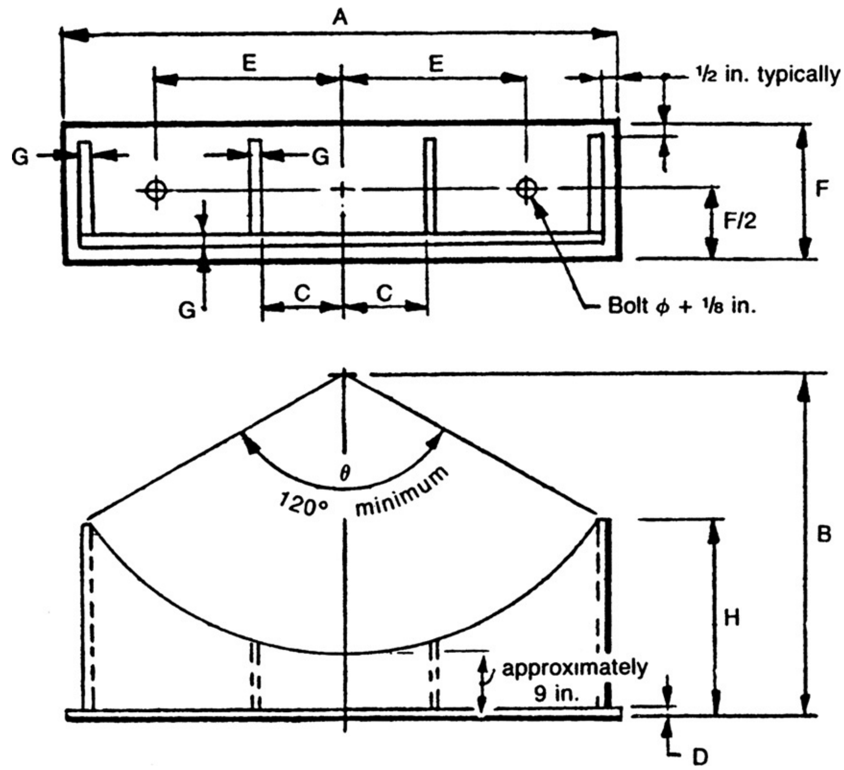


Figure 4-46. Saddle dimensions.

Table 4-23
Slot dimensions

Temperature	Distance Between Saddles				
°F	10ft	20ft	30ft	40ft	50ft
−50	0	0	0.25	0.25	0.375
100	0	0	0.125	0.125	0.250
200	0	0.250	0.375	0.375	0.500
300	0.250	0.375	0.625	0.750	1.00
400	0.375	0.625	0.875	1.125	1.375
500	0.375	0.750	1.125	1.500	1.625
600	0.500	1.00	1.375	1.875	2.250
700	0.625	1.125	1.625	2.125	2.625
800	0.750	1.250	1.625	2.375	3.000
900	0.750	1.375	2.000	2.500	3.375

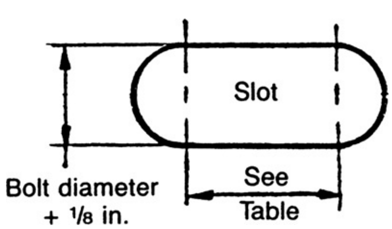


Table 4-24
Typical saddle dimensions*

Vessel O.D.	Maximum Operating Weight	A	B	C	D	E	F	G	H	Bolt Diameter	θ	Approximate Weight/Set
24	15,400	22	21	N.A.	0.5	7	4	0.25	15.2	1	122°	80
30	16,700	27	24			9	4		16.5		120°	100
36	15,700	33	27			12	6		18.8		125°	170
42	15,100	38	30			15			20.0		123°	200
48	25,330	44	33			18			22.3		127°	230
54	26,730	48	36			20			22.7		121°	270
60	38,000	54	39			23			25.0		124°	310
66	38,950	60	42			26			27.2		127°	35D
72	50,700	64	45	10		28		0.375	27.6		122°	420
78	56,500	70	48	11	0.75	31	8		29.8		124°	710
84	57,525	74	51	12		33			30.2		121°	810
90	64,200	80	54	13		36			32.5		123°	880
96	65,400	86	57	14		39			34.7		125°	940
102	94,500	92	60	15		42	10	0.500	37.0	1¼	126°	1,350
108	85,000	96	63	16		44			37.3		123°	1,430
114	164,000	102	66	17		47		0.625	39.6		125°	1,760
120	150,000	106	69	18		49			40.0		122°	1,800
132	127,500	118	75	20		55			44.5		125°	2,180
144	280,000	128	81	22		60			47.0		124°	2,500
156	266,000	140	87	24		66			51.6		126°	2,730

** Table is in inches and pounds and degrees.*

Notes

1. Horizontal vessels act as beams with the following exceptions:
   a. Loading conditions vary for full or partially full vessels.
   b. Stresses vary according to angle θ and distance "A."
   c. Load due to weight is combined with other loads.
2. Large-diameter, thin-walled vessels are best supported near the heads, provided the shell can take the load between the saddles. The resulting stresses in the heads must be checked to ensure the heads are stiff enough to transfer the load back to the saddles.
3. Thick-walled vessels are best supported where the longitudinal bending stresses at the saddles are about equal to the longitudinal bending at midspan. However, "A" should not exceed 0.2 L.
4. Minimum saddle angle $\theta = 120°$, except for small vessels. For vessels designed for external pressure only θ should always $= 120°$. The maximum angle is 168° if a wear plate is used.
5. Except for large L/R ratios or $A > R/2$, the governing stress is circumferential bending at the horn of the saddle. Weld seams should be avoided at the horn of the saddle.
6. A wear plate may be used to reduce stresses at the horn of the saddle *only* if saddles are near heads ($A \leq R/2$), and the wear plate extends R/10 (5.73 deg.) above the horn of the saddle.
7. If it is determined that stiffening rings will be required to reduce shell stresses, move saddles away from the heads (preferable to $A = 0.2$ L). This will prevent designing a vessel with a flexible center and rigid ends. Stiffening ring sizes may be reduced by using a saddle angle of 150°.
8. An internal stiffening ring is the most desirable from a strength standpoint because the maximum stress in the shell is compressive, which is reduced by internal pressure. An internal ring may not be practical from a process or corrosion standpoint, however.
9. Friction factors:

Surfaces	*Friction Factor, μ*
Lubricated steel-to-concrete	0.45
Steel-to-steel	0.4
Lubrite-to-steel	
• Temperature over 500°F	0.15
• Temperature 500°F or less	0.10
• Bearing pressure less than 500 psi	0.15
Teflon-to-Teflon	
• Bearing 800 psi or more	0.06
• Bearing 300 psi or less	0.1

Procedure 4-11: Design of Saddle Supports for Large Vessels [4,15–17,20]

Notation

A_s = cross-sectional area of saddle, in.^{2}
A_b = area of base plate, in.^{2}
A_p = pressure area on ribs, in.^{2}
A_r = cross-sectional area, rib, in.^{2}
Q = maximum load per saddle, lb
$Q_1 = Q_o + Q_R$, lb
$Q_2 = Q_o + Q_L$, lb
Q_o = load per saddle, operating, lb
Q_T = load per saddle, test, lb
Q_L = vertical load per saddle due to longitudinal loads, lb
Q_R = vertical load per saddle due to transverse loads, lb
F_L = maximum longitudinal force due to wind, seismic, pier deflection, etc. (see Procedure 4-10 for detailed description)
F_a = allowable axial stress, psi
F_b = allowable bending stress, psi
F_T = transverse wind or seismic load, lb
N = number of anchor bolts in the fixed saddle
a_t = cross-sectional area of bolts in tension, in.^{2}
Y = effective bearing length, in.
T = tension load in outer bolt, lb
n_1 = modular ratio, steel to concrete, use 10
F_b = allowable bending stress, psi
F_y = yield stress, psi
f_h = saddle splitting force, lb
f_a = axial stress, psi
f_b = bending stress, psi
f_u = unit force, lb/in.
B_p = bearing pressure, psi
M = bending moment, or overturning moment, in.-lb
I = moment of inertia, in.^{4}
Z = section modulus, in.^{3}
r = radius of gyration, in.
K_1 = saddle splitting coefficient
n = number of ribs, including outer ribs, in one saddle
P = equivalent column load, lb
d = distance from base to centroid of saddle arc, in.
W_o = operating weight of vessel plus contents, lb
W_T = vessel weight full of water, lb
σ_T = tension stress, psi
w = uniform load, lb

Forces and Loads

Vertical Load per Saddle

For loads due to the following causes, use the given formulas.

• *Operating weight.*

$$Q_o = \frac{W_o}{2}$$

• *Test weight.*

$$Q_T = \frac{W_T}{2}$$

• *Longitudinal wind or seismic.*

$$Q_L = \frac{F_L B}{L_s}$$

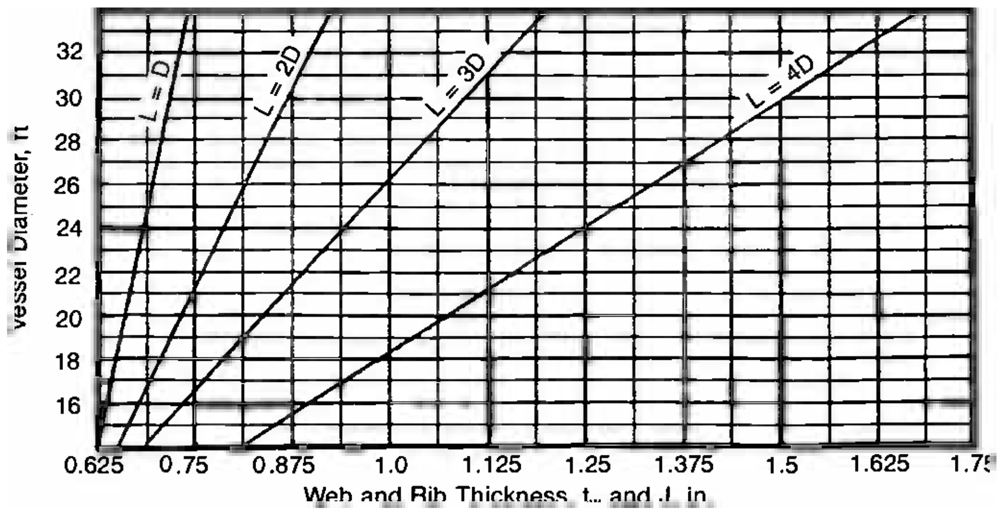


Figure 4-47. Graph for determining web and rib thicknesses.

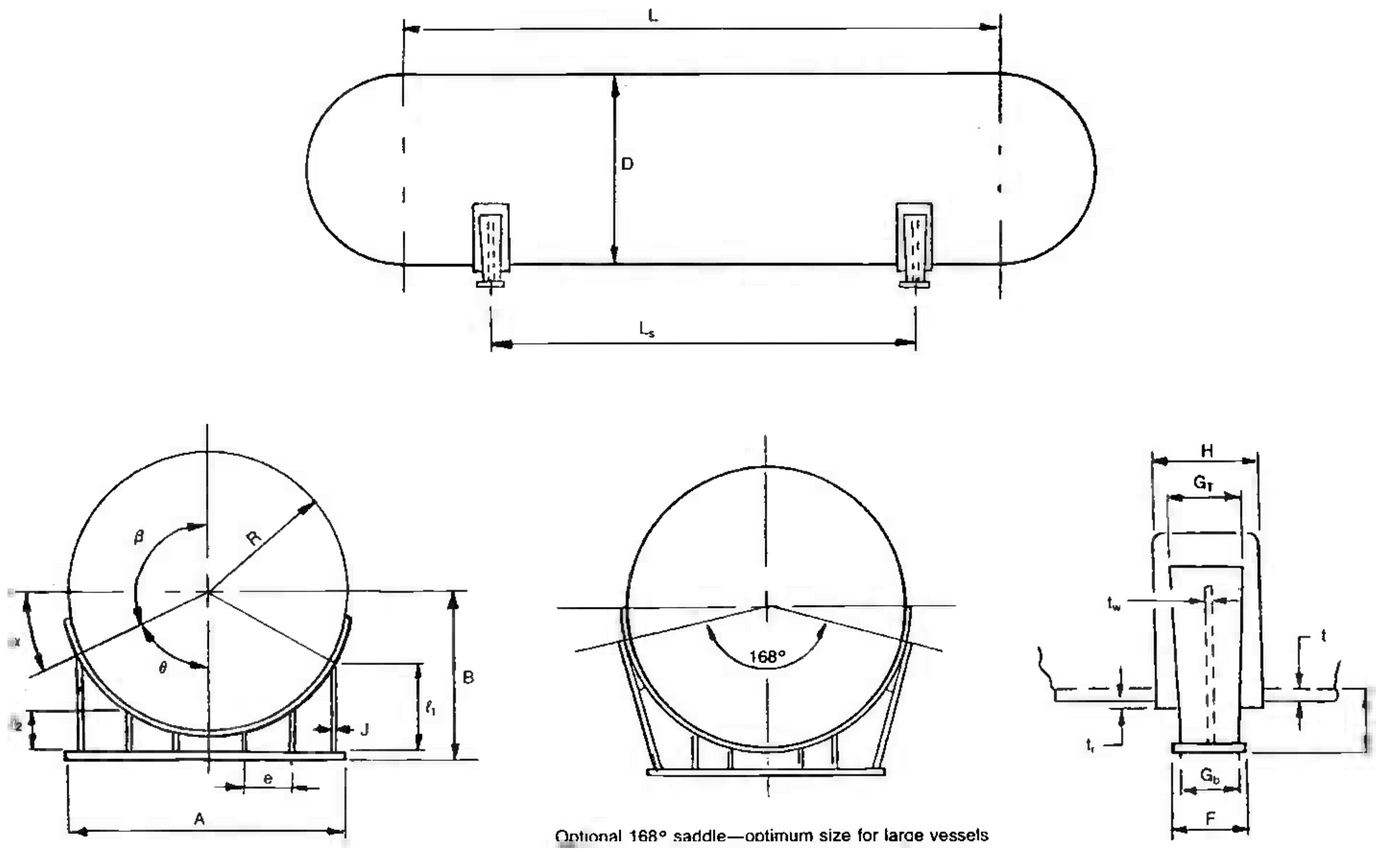


Figure 4-48. Dimensions of horizontal vessels and saddles.

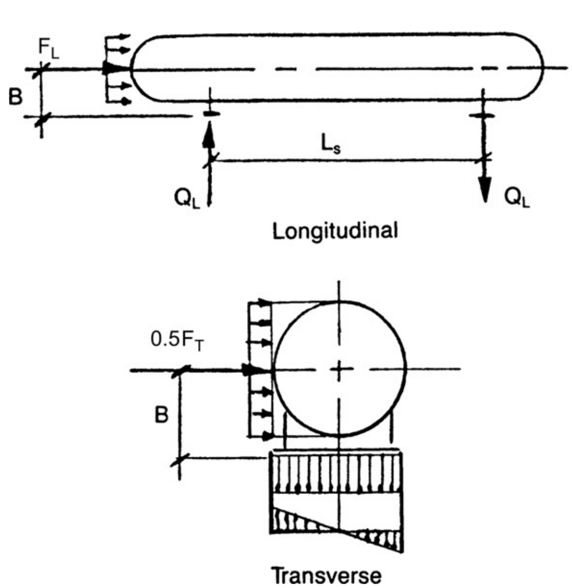


Figure 4-49. Saddle loadings.

- *Transverse wind or seismic.*

$$Q_R = \frac{3F_T B}{A}$$

Maximum Loads

- *Vertical.*
  greater of Q_1, Q_2, or Q_T

$$Q_1 = Q_o + Q_R$$
$$Q_2 = Q_o + Q_L$$

- *Longitudinal.*
  F_L = greater of F_{L1} through F_{L6}

Saddle Properties

- *Preliminary web and rib thicknesses, t_w and J.* From Figure 4-47:

$$J = t_w$$

- *Number of ribs required, n.*

$$n = \frac{A}{24} + 1$$

Round up to the nearest even number.

- *Minimum width of saddle at top, G_T, in.*

$$G_T = \sqrt{\frac{5.012F_L}{J(n-1)F_b}\left[h + \frac{A}{1.96}(1 - \sin\ \alpha)\right]}$$

where F_L and F_b are in kips and ksi or lb and psi, and J, h, A are in inches.

- *Minimum wear plate dimensions.*

Width:

$$H = G_T + 1.56\sqrt{Rt_s}$$

Thickness:

$$t_r = \frac{(H - G_T)^2}{2.43R}$$

- *Moment of inertia of saddle, I.* See Figure 4-50

$$C_1 = \frac{\sum AY}{\sum A}$$
$$C_2 = h - C_1$$
$$I = \sum AY^2 + \sum I_o - C_1 \sum AY$$

- *Cross-sectional area of saddle (excluding shell).*

$$A_s = \sum A - A_1$$

Design of Saddle Parts

Web

Web is in tension and bending as a result of saddle splitting forces. The saddle splitting forces, f_h, are the sum of all the horizontal reactions on the saddle.

- *Saddle coefficient.* See Table 4-25

$$K_1 = \frac{1 + \cos\beta - 0.5\sin^2\beta}{\pi - \beta + \sin\beta\cos\beta}$$

Note: β is in radians.

- *Saddle splitting force.* See Figure 4-51 and 4-52

$$f_h = K_1(Q \text{ or } Q_T)$$

- *Tension stress.*

$$\sigma_T = \frac{f_h}{A_s} < 0.6F_y$$

Note: For tension assume saddle depth “h” as R/3 maximum.

- *Bending moment.*

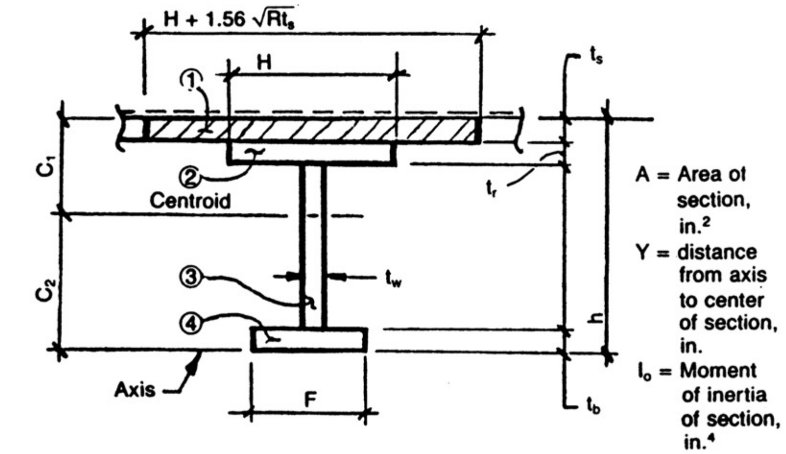
$$d = B - \frac{R\sin\theta}{\theta}$$

θ is in radians.

$$M = f_h d$$

- *Bending stress.*

$$f_b = \frac{MC_1}{I} < 0.66\,F_y$$



	A	Y	AY	AY^2	I_o
①					
②					
③					
④					
Σ					

Note: I_o for rectangles $= \dfrac{bh^3}{12}$

Figure 4-50. Cross-sectional properties of saddles.

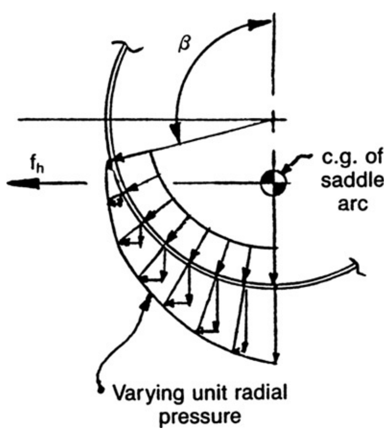
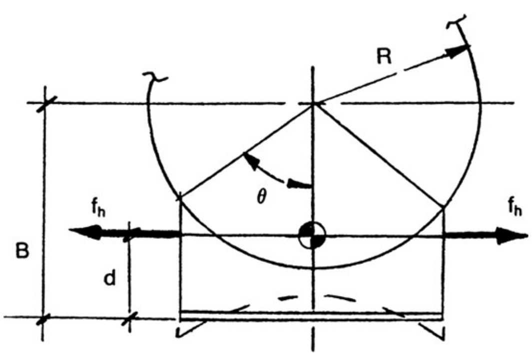


Figure 4-51. Saddle splitting forces.



Note: Circumferential bending at horn is neglected for this calculation.

Figure 4-52. Bending in saddle due to splitting forces.

Base plate with center web see Figure 4-53

- *Area.*

$$A_b = AF$$

- *Bearing pressure.*

$$B_p = \frac{Q}{A_b}$$

- *Base plate thickness.*

$$\text{Now } M = \frac{QF}{8}$$

$$Z = \frac{At_b^2}{6}$$

$$\text{and } f_b = \frac{M}{Z} = \frac{3QF}{4At_b^2}$$

Therefore

Table 4-25
Values of K_1

k_1	2θ
0.204	120°
0.214	126°
0.226	132°
0.237	138°
0.248	144°
0.260	150°
0.271	156°
0.278	162°
0.294	168°

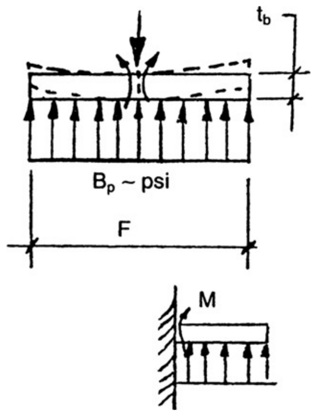


Figure 4-53. Loading diagram of base plate.

$$t_b = \sqrt{\frac{3QF}{4AF_b}}$$

Assumes uniform load fixed in center.

Base plate analysis for offset web (see Figure 4-54)

- *Overall length, $\sum L$.*
  Web $L_w = A - 2d_1 - 2J$
  ribs $L_r = n(G - t_w)$
  $$\sum L = L_w + L_r$$
- *Unit linear load, fu.*
  $$fu = \frac{Q}{\sum L} \text{Ib/linear in.}$$
- *Distances ℓ_1 and ℓ_2.*
  $$\ell_1 = d_2 + t_w + W_w + t_b$$
  $$\ell_2 = F - \ell_1$$
- *Loads / moment.*

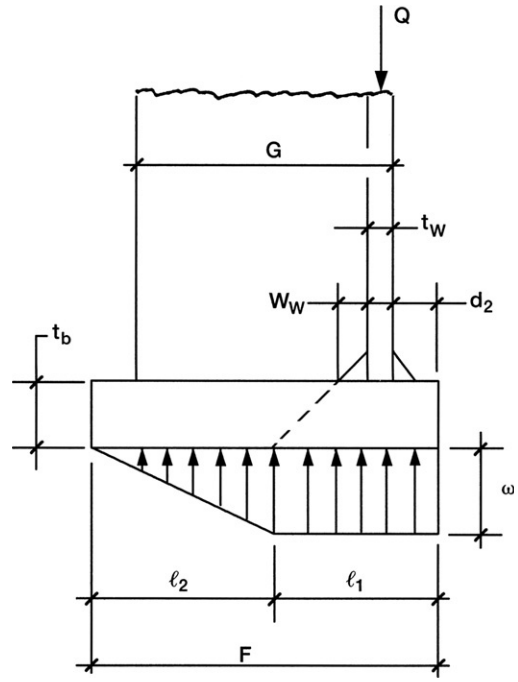


Figure 4-54. Load diagram and dimensions for base plate with an offset web.

$$\omega = \frac{fu}{\ell_1 + 0.5\ell_2}$$

$$M = \frac{\omega \ell_2^2}{6}$$

- *Bending stress, f_b.*
  $$f_b = \frac{6M}{t_b^2}$$

Anchor Bolts

Anchor bolts are governed by one of the three following load cases:

1. *Longitudinal load*: If $Q_o > Q_L$, then no uplift occurs, and the minimum number and size of anchor bolts should be used.
   If $Q_o < Q_L$, then uplift does occur:
   $$\frac{Q_L - Q_o}{N} = \text{load per bolt}$$
2. *Shear*: Assume the fixed saddle takes the entire shear load.
   $$\frac{F_L}{N} = \text{shear per bolt}$$

3. *Transverse load*: This method of determining uplift and overturning is determined from Ref. [20] (see Figure 4-56).

$$M = 0.5\,F_T\,B$$

$$e = \frac{M}{Q_o}$$

If $e < A/6$, then there is no uplift.

If $e \geq A/6$, then proceed with the following steps. This is an iterative procedure for finding the tension force, T, in the outermost bolt.

Step 1: Find the effective bearing length, Y. Start by calculating factors $K_{1\text{-}3}$.

$$K_1 = 3(e - 0.5A)$$

$$K_2 = \frac{6n_1 a_t}{F}(f + e)$$

$$K_3 = (-)K_2\left[\frac{A}{2} + f\right]$$

Step 2: Substitute values of K_{1-3} into the following equation and assume a value of $Y = ⅔\,A$ as a first trial.

$$Y^3 + K_1Y^2 + K_2Y + K_3 = 0$$

If not equal to 0, then proceed with Step 3.

Step 3: Assume a new value of Y and recalculate the equation in Step 2 until the equation balances out to approximately 0. Once Y is determined, proceed to Step 4.

Step 4: Calculate the tension force, T, in the outermost bolt or bolts.

$$T = (-)Q_o\left[\frac{\frac{A}{2} - \frac{Y}{3} - e}{\frac{A}{2} - \frac{Y}{3} + f}\right]$$

Step 5: Select an appropriate bolt material and size corresponding to tension force, T.

Step 6: Analyze the bending in the base plate.

Distance, $x = 0.5A + f - Y$

Moment, $M = Tx$

Bending stress, $f_b = \frac{6M}{t_b^2}$

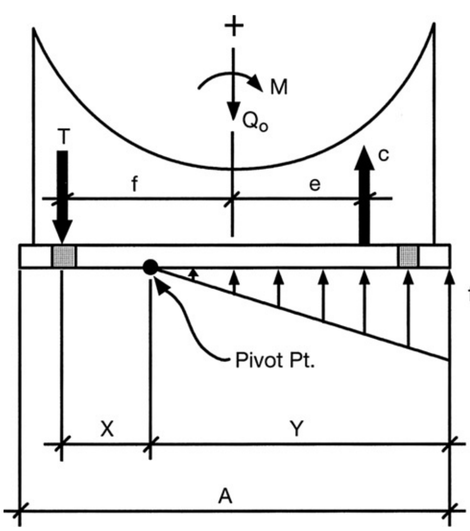


Figure 4-55. Dimensions and loading for base plate and anchor bolt analysis.

Ribs

Outside Ribs

- *Axial load, P.*

$$P = B_pA_p$$

- *Compressive stress, f_a.*

$$f_a = \frac{P}{A_r}$$

- *Radius of gyration, r.*

$$r = \sqrt{\frac{I_1}{A_r}}$$

- *Slenderness ratio, ℓ_1/r.*

$l_1/r =$

$F_a =$

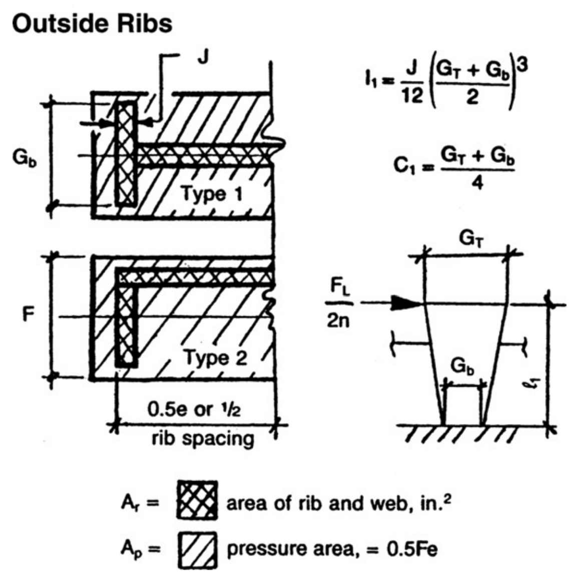


Figure 4-56. Dimensions of outside saddle ribs and webs.

- *Unit force, f_u.*

$$f_u = \frac{F_L}{2A}$$

- *Bending moment, M.*

$$M = 0.5 f_u e l_1$$

- *Bending stress, $F_b = 0.66\ F_y$.*

$$f_b = \frac{MC_1}{I} < F_b$$

- *Combined stress.*

$$\frac{f_a}{F_a} + \frac{f_b}{F_b} < 1$$

Inside Ribs

- *Axial load, P.*

$$P = B_p A_p$$

- *Compressive stress, f_a.*

$$f_a = \frac{P}{A_r}$$

- *Radius of gyration, r.*

$$r = \sqrt{\frac{I_2}{A_r}}$$

- *Slenderness ratio, ℓ_2/r.*

$$l_2/r =$$

$$F_a =$$

- *Unit force, f_u.*

$$f_u = \frac{F_L}{2A}$$

- *Bending moment, M.*

$$M = f_u l_2 e$$

- *Bending stress, f_b.*

$$f_b = \frac{MC_2}{I}$$

- *Combined stress.*

$$\frac{f_a}{F_a} + \frac{f_b}{F_b} < 1$$

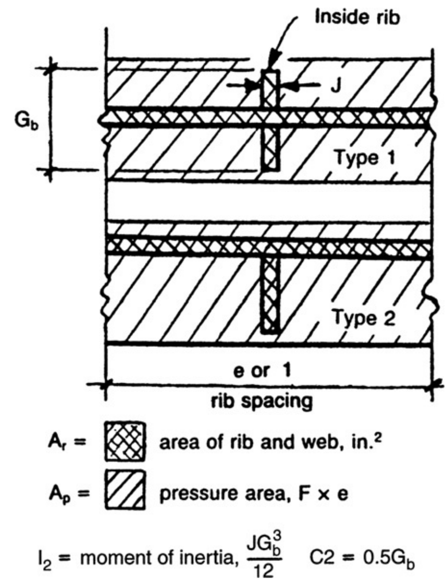


Figure 4-57. Dimensions of inside saddle ribs and webs.

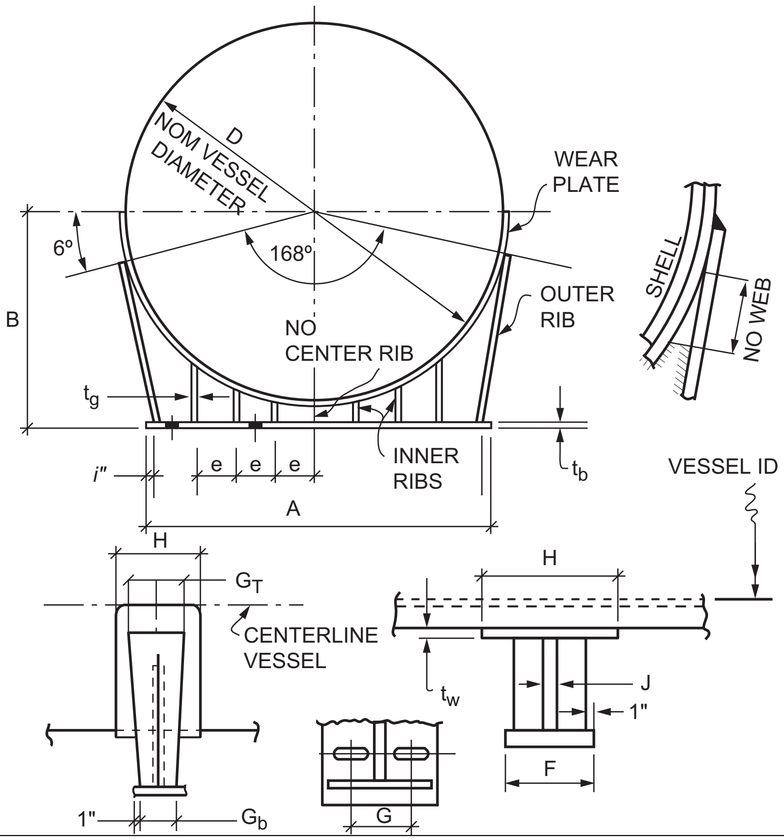
Notes

1. The depth of web is important in developing stiffness to prevent bending about the cross-sectional axis of the saddle. For larger vessels, assume 6 in. as the minimum depth from the bottom of the wear plate to the top of the base plate.
2. The full length of the web may be assumed effective in carrying compressive stresses along with ribs. Ribs are not effective at carrying compressive load if they are spaced greater than 25 times the web thickness apart.
3. Concrete compressive stresses are usually considered to be uniform. This assumes the saddle is rigid enough to distribute the load uniformly.
4. Large-diameter horizontal vessels are best supported with 168° saddles. Larger saddle angles do not effectively contribute to lower shell stresses and are more difficult to fabricate. The wear plate need not extend beyond center lines of vessel in any case or 6° beyond saddles.
5. Assume fixed saddle takes all of the longitudinal loading.

Table 4-26
Allowable tension load on bolts, kips, per AISC

Nominal Bolt Diameter, in		0.625	0.750	0.875	1.000	1.125	1.250	1.375	1.500
Cross-sectional Area, a_b, in^2		0.3068	0.4418	0.6013	0.7854	0.9940	1.2272	1.4849	1.7671
A-307 F_t	22.5	6.9	9.9	13.5	17.7	22.4	27.6	33.4	39.8
A-325 F_t	45.0	13.8	19.9	27.1	35.3	44.7	55.2	66.8	79.5

Table 4-27
Large diameter saddle supports



DIA (Ft)	A	B	D (Note 3)	E	F	G	G_B	G_T	H	J	N	n	t_b	t_g	t_w	Approx wt for 2 Saddles (Kips)
14	155	92	1.375	19	18	9	16	28	34	0.75	8	8	1	0.75	0.75	7
15	171	100	1.375	21	18	9	16	28	34	0.75	8	8	1	0.75	0.75	8
16	183	108	1.375	18	18	9.25	16	28	34	1	10	8	1.125	0.875	0.875	10
17	193	114	1.625	19	21	10.75	19	31	37	1	10	8	1.25	1	1	11
18	207	122	1.625	17	21	11	19	31	37	1.125	12	12	1.375	1.125	1	12
20	207	132	1.625	17	21	11	19	31	37	1.125	12	12	1.375	1.125	1.125	15.5
22	219	144	1.875	18	24	12.5	22	34	40	1.25	12	12	1.5	1.25	1.25	19
24	241	156	1.875	17	24	12.5	22	34	40	1.25	14	12	1.5	1.25	1.25	22
26	255	172	1.875	18	24	12.5	22	34	40	1.375	14	12	1.625	1.25	1.25	26
28	275	184	2.125	17	27	14	25	37	43	1.375	16	16	1.625	1.375	1.25	31
30	308	196	2.125	19	27	14.25	25	37	43	1.5	16	16	1.75	1.5	1.375	37
32	328	208	2.125	18	27	14.25	25	37	43	1.5	18	16	1.75	1.5	1.375	44
34	346	220	2.375	19	31	16	29	41	47	1.75	18	16	2	1.75	1.375	54
36	364	230	2.375	18	31	16	29	41	47	1.75	20	16	2	1.75	1.375	66
38	384	244	2.375	19	32	16.25	30	42	48	2	20	16	2.25	2	1.5	80
40	404	256	2.625	20	34	17.75	32	44	50	2	20	20	2.5	2	1.5	100

Notes:

1. All dimensions are in inches unless noted otherwise
2. All saddles in this size range must be fully designed. The dimensions shown are a starting place or to be used for estimating only!
3. Assume that anchor bolts diameter is d − .125", where d is the diameter of the hole. Assume that slots for sliding saddle are 6 d long.
4. N = Number of ribs
5. n = Number of anchor bolts

Procedure 4-12: Design of Base Plates for Legs [20,21]

Notation

Y = effective bearing length, in.
M = overturning moment, in.-lb
M_b = bending moment, in.-lb
P = axial load, lb
f_t = tension stress in anchor bolt, psi
A = actual area of base plate, in.^{2}
A_r = area required, base plate, in.^{2}
f'_c = ultimate 28-day strength, psi
f_c = bearing pressure, psi
f_1 = equivalent bearing pressure, psi
F_b = allowable bending stress, psi
F_t = allowable tension stress, psi
F_c = allowable compression stress, psi
E_s = modulus of elasticity, steel, psi
E_c = modulus of elasticity, concrete, psi
n = modular ratio, steel-concrete
n' = equivalent cantilever dimension of base plate, in.
B_p = allowable bearing pressure, psi
$K_{1,2,3}$ = factor
T = tension force in outermost bolt, lb
C = compressive load in concrete, lb
V = base shear, lb
N = total number of anchor bolts
N_t = number of anchor bolts in tension
A_b = cross-sectional area of one bolt, in.^{2}
A_s = total cross-sectional area of bolts in tension, in.^{2}
α = coefficient
T_s = shear stress

Calculations

- *Axial loading only, no moment.*
  Angle legs:

$$f_c = \frac{P}{BD}$$

L = greater of m, n, or n'

$$t = \sqrt{\frac{3f_cL^2}{F_b}}$$

Beam legs:

$$A_r = \frac{P}{0.7f'_c}$$

$$m = \frac{D - 0.95d}{2}$$

$$n = \frac{B - 0.8d}{2}$$

$$\alpha = \frac{b - t_w}{2(d - 2t_f)}$$

$$n' = \frac{b - t_w}{2}\sqrt{\frac{1}{1 + 3.2\alpha^3}} \text{ (See Table 4-27)}$$

Pipe legs:

$$m = \frac{B - 0.707W}{2}$$

$$f_c = \frac{P}{A}$$

$$t = \sqrt{\frac{3f_c m^2}{F_b}}$$

- *Axial load plus bending, load condition #1, full compression, uplift, $e \leq D/6$.* (See Figure 4-59)
  Eccentricity:

$$e = \frac{M}{P} \leq \frac{D}{6}$$

Loadings:

$$f_c = \frac{P}{A}\left[1 + \frac{6e}{D}\right]$$

$$f_1 = \frac{P}{A}\left[1 + \frac{6e(D - 2a)}{D^2}\right]$$

Moment:

$$M_b = \frac{a^2B}{6}(f_1 + 2f_c)$$

Thickness:

$$t = \sqrt{\frac{6M_b}{BF_b}}$$

- *Axial load plus bending, load condition #2, partial compression, uplift, $e > D/6$.* (See Figure 4-59)
  Eccentricity:

$$e = \frac{M}{P} > \frac{D}{6}$$

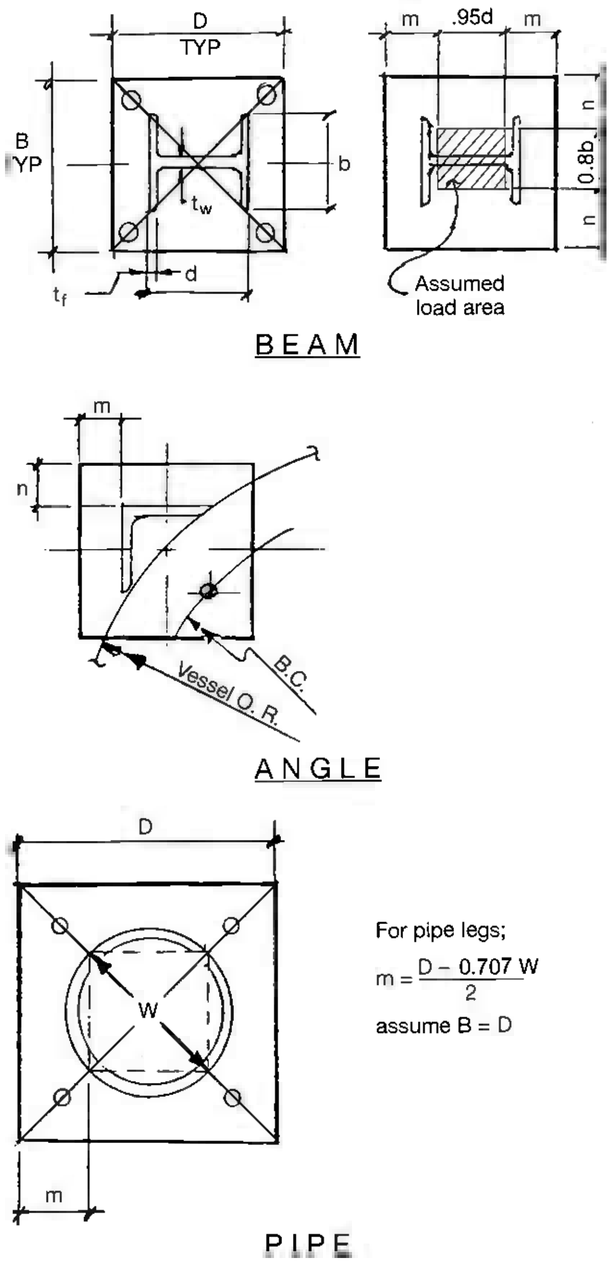


Figure 4-58. Dimensions and loadings of base plates.

Coefficient: (See Table 4-29)

$$n_r = \frac{E_s}{E_c}$$

Dimension:

$$f = 0.5d + z$$

By trial and error, determine Y, effective bearing length, utilizing factors $K_{1\text{–}3}$.

Factors:

$$K_1 = 3\left(e + \frac{D}{2}\right)$$

$$K_2 = \frac{6n_rA_s}{B}(f + e)$$

$$K_3 = (-)K_2(0.5D + f)$$

By successive approximations, determine distance Y. Substitute $K_{1\text{-}3}$ into the following equation and assume an initial value of $Y = \frac{2}{3}A$ as a first trial.

$$Y^3 + K_1Y^2 + K_2Y + K_3 = 0$$

Tension force:

$$T = (-)P\left[\frac{\frac{D}{2} - \frac{Y}{3} - e}{\frac{D}{2} - \frac{Y}{3} + f}\right]$$

Bearing pressure:

$$f_c = \frac{2(P + T)}{YB} < f'_c$$

Moment:

$$x = 0.5D + f - Y$$

$$M_t = Tx$$

$$f_1 = f_c\left(\frac{Y - a}{Y}\right)$$

$$M_c = \frac{a^2B}{6}(f_1 + 2f_c)$$

Thickness:

$$t = \sqrt{\frac{6M_b}{BF_b}}$$

where M_b is greater of M_T or M_c.

- *Anchor bolts.*
  Without uplift: design anchor bolts for shear only.

$$T_s = \frac{V}{NA_b}$$

With uplift: design anchor bolts for full shear and tension force, T.

$$f_t = \frac{T}{N_TA_b}$$

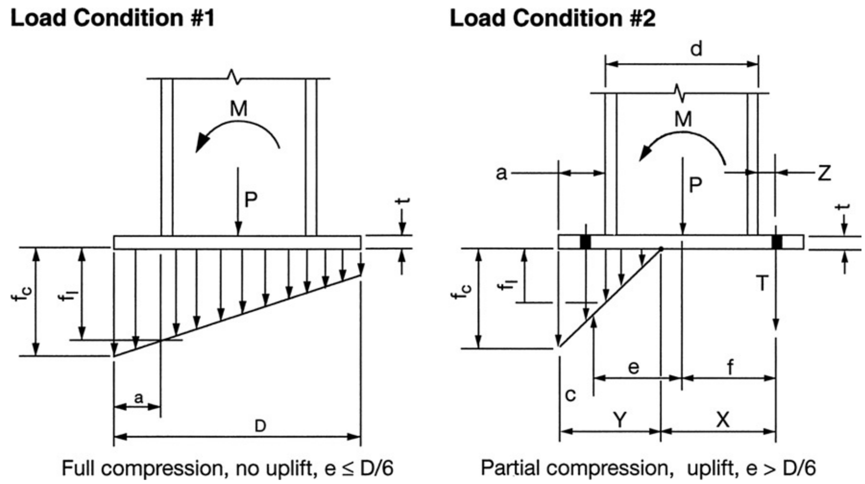


Figure 4-59. Load conditions on base plates.

Table 4-28
Values of n′ for beams

Column Section	n′	Column Section	n′
W14 × 730 – W14 × 145	5.77	W10 × 45 – W10 × 33	3.42
W14 × 132 – W14 × 90	5.64	W8 × 67 – W8 × 31	3.14
W14 × 82 – W14 × 61	4.43	W8 × 28 – W8 × 24	2.77
W14 × 53 – W14 × 43	3.68	W6 × 25 – W6 × 15	2.38
W12 × 336 – W12 × 65	4.77	W6 × 16 – W6 × 9	1.77
W12 × 58 – W12 × 53	4.27	W5 × 19 – W5 × 16	1.91
W12 × 50 – W12 × 40	3.61	W4 × 13	1.53
W10 × 112 – W10 × 49	3.92		

Table 4-29
Average properties of concrete

Water Content/Bag	Ult f'_c 28 -Day Str (psi)	Allowable Compression, F_c (psi)	Allowable B_p (psi)	Coefficient, n_r
7.5	2000	800	500	15
6.75	2500	1000	625	12
6	3000	1200	750	10
5	3750	1400	938	8

Reprinted by permission of John Wiley & Sons, Inc.

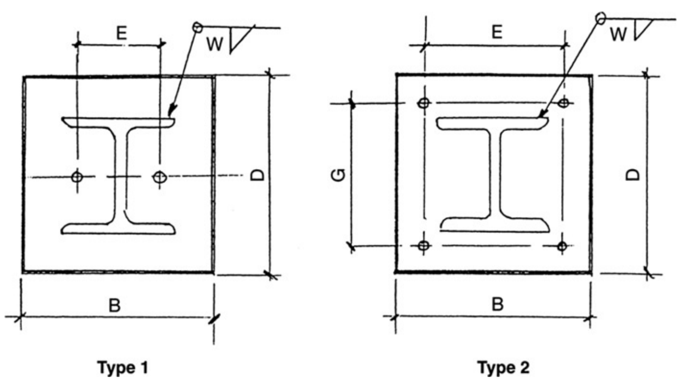


Figure 4-60. Dimensions for base plates-beams.

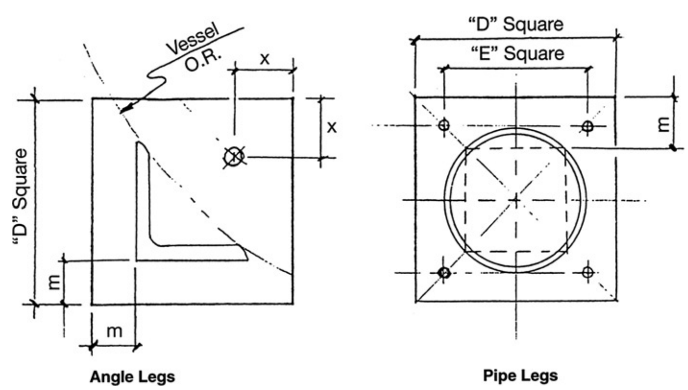


Figure 4-61. Dimensions for base plates—angle/pipe.

Dimensions for Type 1—(2) Bolt Base Plate

Column Size	D, in.	B, in.	E, in.	W, in.	Min Plate Thk, in.	Max Bolt ϕ, in.
W4	8	8	4	¼	⅝	¾
W6	8	8	4	¼	¾	¾
W8	10	10	6	¼	¾	¾
W10–33 thru 45	12	12	6	5/16	¾	1
W10–49 thru 112	13	13	6	5/16	¾	1
W12–40 thru 50	14	10	6	5/16	⅞	1
W12–53 thru 58	14	12	6	5/16	⅞	1
W12–65 thru 152	15	15	8	5/16	⅞	1¼

Dimensions for Type 2—(4) Bolt Base Plate

Column Size	D, in.	B, in.	G, in.	E, in.	W, in.	Min Plate Thk, in.	Max Bolt ϕ, in.
W4	10	10	7	7	¼	⅝	1
W6	12	12	9	9	5/16	¾	1
W8	15	15	11	11	⅜	⅞	1
W10–33 thru 45	17	15	13	11	⅜	⅜	1¼
W10–49 thru 112	17	17	13	13	⅜	⅜	1¼
W12–40 thru 50	19	15	15	11	⅜	1	1½
W12–53 thru 58	19	17	15	13	⅜	1	1½
W12–65 thru 152	19	19	15	15	⅜	1	1½

Dimensions for Angle Legs

Leg Size	D	X	m	Min. Plate Thk
L2 in. × 2 in.	4 in.	1.5	1	½ in.
L2½in. × 2½in.	5 in.	1.5	1.25	½ in.
L3 in. × 3 in.	6 in.	1.75	1.5	½ in.
L4 in. × 4 in.	8 in.	2	2	⅝ in.
L5 in. × 5 in.	9 in.	2.75	2	⅝ in.
L6 in. × 6 in.	10 in.	3.5	2	¾ in.

Dimensions for Pipe Legs

Leg Size	D	E	m	Min. Plate Thk
3 in. NPS	7 ½in.	4 ½ in.	2.5 in.	½ in.
4 in. NPS	8 ½in.	5 ½ in.	2.7 in.	½ in.
6 in. NPS	10 in.	7 in.	2.7 in.	⅝ in.
8 in. NPS	11 ½in.	8 ½ in.	2.7 in.	¾ in.
10 in. NPS	14 in.	10 in.	3.2 in.	⅞ in.
12 in. NPS	16 in.	12 in.	3.5in.	1 in.

Procedure 4-13: Design of Lug Supports

Notation

Q = vertical load per lug, lb
Q_a = axial load on gusset, lb
Q_b = bending load on gusset, lb
n = number of gussets per lug
F_a = allowable axial stress, psi
F_b = allowable bending stress, psi
f_a = axial stress, psi
f_b = bending stress, psi
A = cross-sectional area of assumed column, in.^{2}
Z = section modulus, in.^{3}
w = uniform load on base plate, lb/in.
I = moment of inertia of compression plate, in.^{4}
E_v = modulus of elasticity of vessel shell at design temperature, psi
E_s = modulus of elasticity of compression plate at design temperature, psi
e = log base 2.71
M_b = bending moment, in.-lb
M_x = internal bending moment in compression plate, in.-lb
K = spring constant or foundation modulus
β = damping factor

Design of Gussets

Assume gusset thickness from Table 4-30.

$$Q_a = Q \sin\theta$$

$$Q_b = Q \cos\theta$$

$$C = \frac{b \sin\theta}{2}$$

$$A = t_g C$$

$$F_a = 0.4F_y$$

$$F_b = 0.6F_y$$

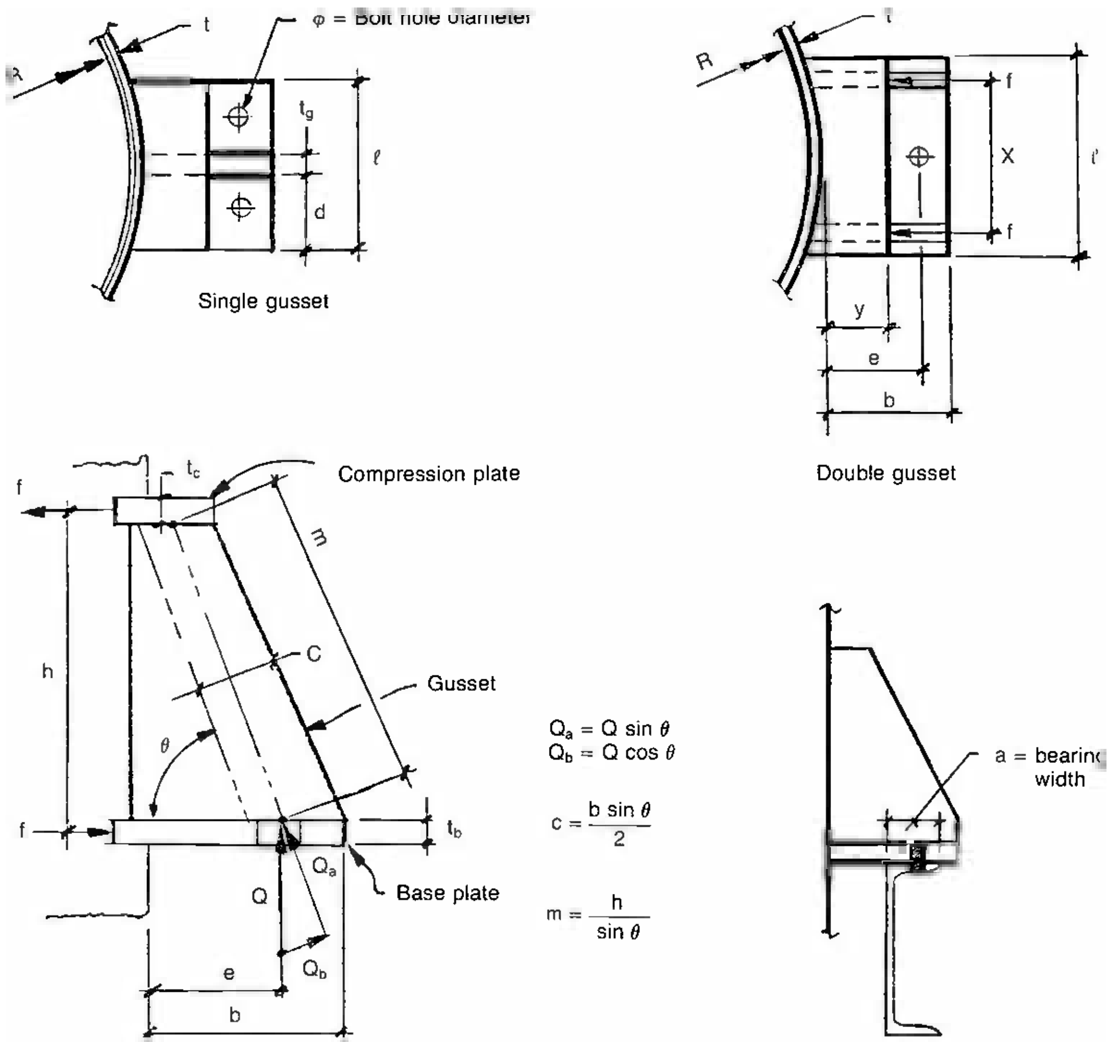


Figure 4-62. Dimensions and forces on a lug support.

$$Z = \frac{t_g C^2}{6}$$

$$M_b = \frac{Q_b m}{n}$$

$$f_a = \frac{Q_a}{nA}$$

$$f_b = \frac{M_b}{Z}$$

Design of Base Plate

Single Gusset

- *Bending.* Assume to be a simply supported beam.

$$M_b = \frac{Ql}{4}$$

- *Bearing.*

$$w = \frac{Q}{al}$$

$$M_b = \frac{wd^2}{2}$$

- *Thickness required base plate.*

$$t_b = \sqrt{\frac{6M_b}{(b - \phi)F_b}}$$

where M_b is greater moment from bending or bearing.

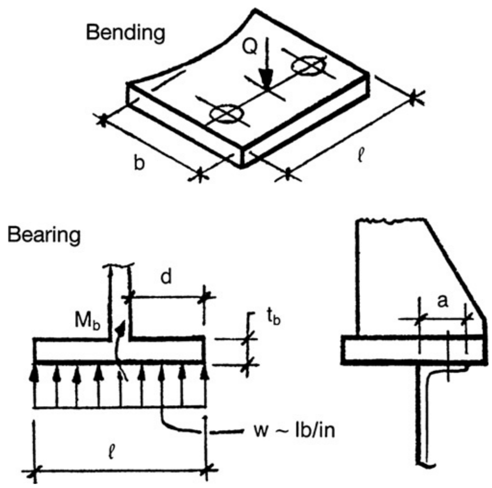


Figure 4-63. Loading diagram of base plate with one gusset.

Double Gusset

- *Bending*. Assume to be between simply supported and fixed.

$$M_b = \frac{Ql}{6}$$

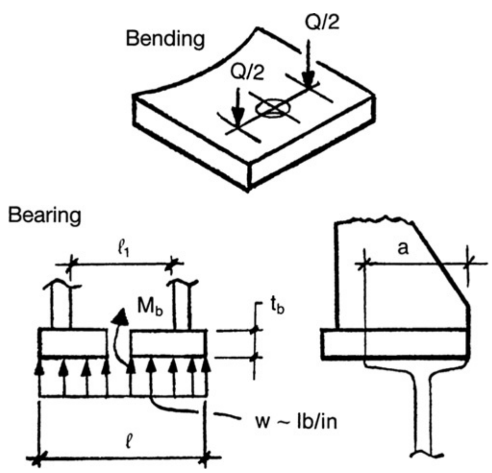


Figure 4-64. Loading diagram of base plate with two gussets.

- *Bearing*.

$$w = \frac{Q}{al}$$

$$M_b = \frac{wl_1^2}{10}$$

- *Thickness required base plate.*

$$t_b = \sqrt{\frac{6M_b}{(b - \phi)F_b}}$$

where M_b is greater moment from bending or bearing.

Compression Plate

Single Gusset

$$f = \frac{Qe}{h}$$

$$K = \frac{E_v t}{R^2}$$

Assume thickness t_c and calculate **I** and **Z**:

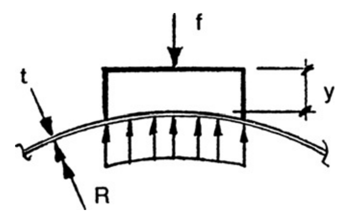


Figure 4-65. Loading diagram of compression plate with one gusset.

$$I = \frac{t_c y^3}{12}$$

$$Z = \frac{t_c y^2}{6}$$

$$\beta = \sqrt[4]{\frac{K}{4E_s I}}$$

$$M_x = \frac{f}{4\beta}$$

$$f_b = \frac{M_x}{Z} \quad < 0.6F_y$$

Note: These calculations are based on a beam on elastic foundation methods.

Double Gusset

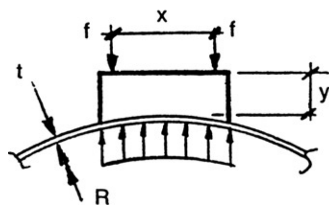


Figure 4-66. Loading diagram of compression plate with two gussets.

$$f = \frac{Qe}{2h}$$

$$K = \frac{E_v t}{R^2}$$

$$I = \frac{t_c y^3}{12}$$

$$Z = \frac{t_c y^2}{6}$$

$$\beta = \sqrt[4]{\frac{K}{4E_s I}}$$

Table 4-30
Standard lug dimensions

Type	e	b	y	x	h	$t_g = t_b$	Capacity (lb)
1	4	6	2	6	6	3/8	23,500
2	4	6	2	6	9	7/16	45,000
3	4	6	2	6	12	½	45,000
4	5	7	2.5	7	15	9/16	70,000
5	5	7	2.5	7	18	5/8	70,000
6	5	7	2.5	7	21	11/16	70,000
7	6	8	3	8	24	3/4	100,000

$$M_x = \frac{f}{4\beta}\left[1 + \left(e^{-\beta x}(\cos \beta x - \sin \beta x)\right)\right]$$

β x is in radians.

$$f_b = \frac{M_x}{Z} < 0.6F_y$$

Procedure 4-14: Design of Base Details for Vertical Vessels – Shifted Neutral Axis Method [4,9,13,17,18]

Notation

A_b = required area of anchor bolts, in.^{2}
B_d = anchor bolt diameter, in.
B_p = allowable bearing pressure, psi
b_p = bearing stress, psi
C = compressive load on concrete, lb
d = diameter of bolt circle, in.
d_b = diameter of hole in base plate of compression plate or ring, in.
F_{LT} = longitudinal tension load, lb/in.
F_{LC} = longitudinal compression load, lb/in.
F_b = allowable bending stress, psi
F_c = allowable compressive stress, concrete, psi
F_s = allowable tension stress, anchor bolts, psi
F_y = minimum specified yield strength, psi
f_b = bending stress, psi
f_c = compressive stress, concrete, psi
f_s = equivalent tension stress in anchor bolts, psi
M_b = overturning moment at base, in.-lb
M_t = overturning moment at tangent line, in.-lb
M_x = unit bending moment in base plate, circumferential, in.-lb/in.
M_y = unit bending moment in base plate, radial, in.-lb/in.
H = overall vessel height, ft
δ = vessel deflection, in.
M_o = bending moment per unit length in.-lb/in.
N = number of anchor bolts
n = ratio of modulus of elasticity of steel to concrete
P = maximum anchor bolt force, lb
P_1 = maximum axial force in gusset, lb

E = joint efficiency of skirt-head attachment weld
R_a = root area of anchor bolt, in.^{2}
r = radius of bolt circle, in.
W_b = weight of vessel at base, lb
W_t = weight of vessel at tangent line, lb
w = width of base plate, in.
Z_l = section modulus of skirt, in.^{3}
S_t = allowable stress (tension) of skirt, psi
S_c = allowable stress (compression) of skirt, psi
G = width of unreinforced opening in skirt, in.
C_c,C_T,J,Z,K = coefficients
γ_1,γ_2 = coefficients for moment calculation in compression ring
S = code allowable stress, tension, psi
E_l = modulus of elasticity, psi
t_s = equivalent thickness of steel shell which represents the anchor bolts in tension, in.
T = tensile load in steel, lb
ν = Poisson's ratio, 0.3 for steel
B = code allowable longitudinal compressive stress, psi

Equivalent Area Method

The "Equivalent Area Method" is also known as the "Shifted Neutral Axis Method". This procedure is in contrast with the "Centered Neutral Axis Method" which assumes that the neutral axis is on the centerline. The Centered Neutral Axis Method is easier to apply but also results in a conservative anchorage design. The Equivalent Area method is more accurate and will result in reduced anchorage requirements. Both methods are used to determine the anchorage requirements and the base plate details of a vertical vessel supported on a skirt.

The Equivalent Area Method is based on reinforced concrete beam design that utilizes a balance between the steel in tension and the concrete in compression. Because of the different properties the neutral axis is shifted from the centerline. This procedure enables the designer to find the exact position of the neutral axis and compute the properties required based on this location.

In order to find the minimum anchor bolt area required that is consistent with a given base ring area and a given working stress in the anchor bolts, it is necessary to resort to a trial and error basis, an iterative procedure. To start, the variables are either given or assumed. The variables in this process are as follows;

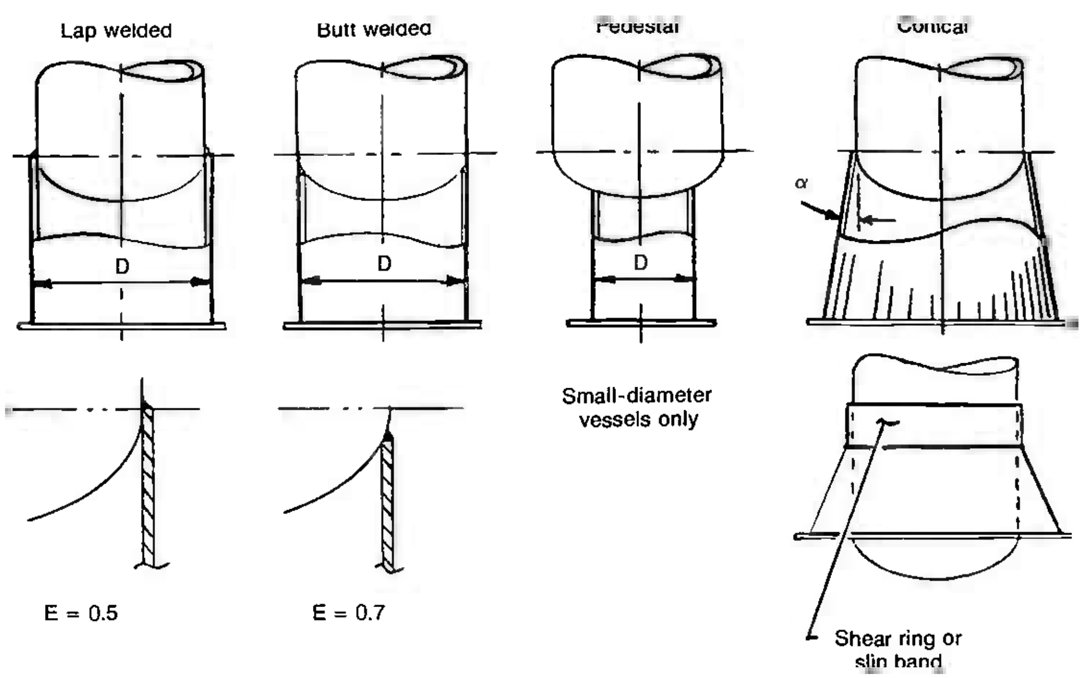


Figure 4-67. Skirt types.

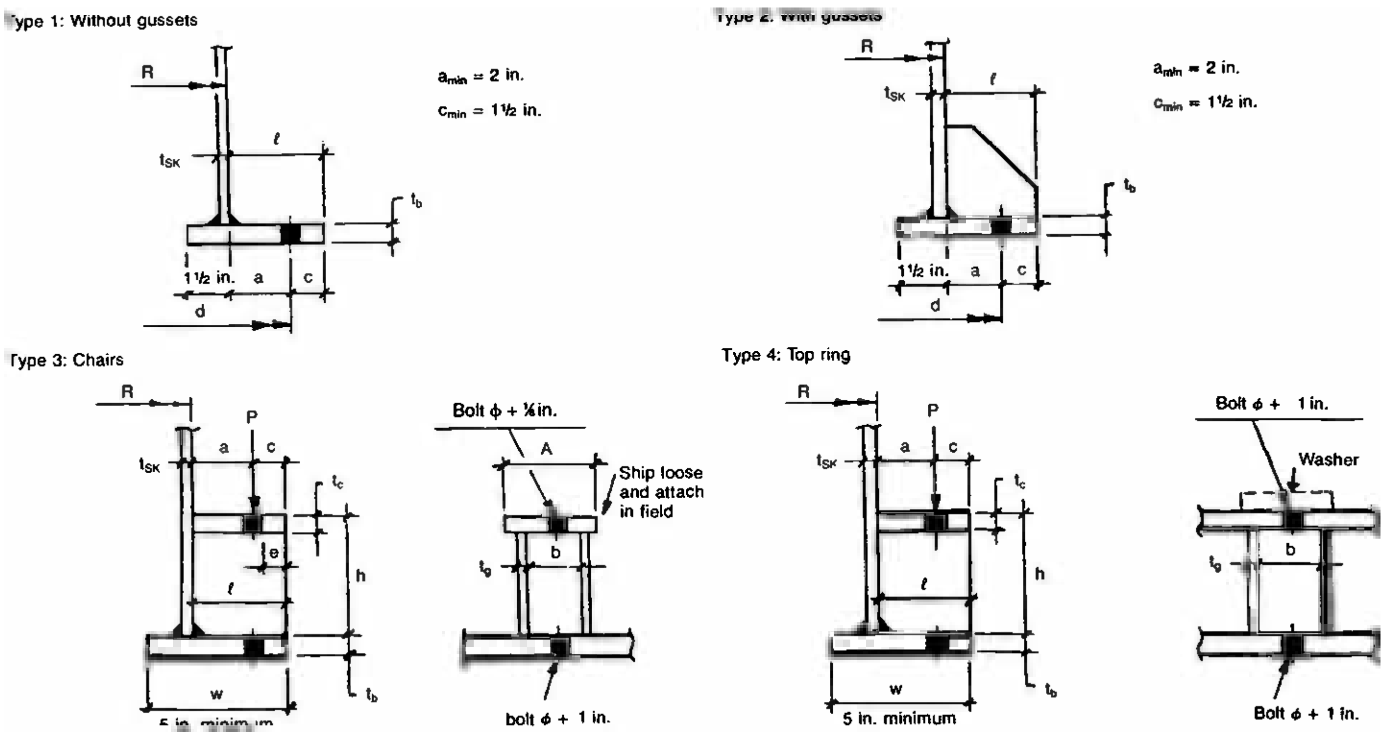


Figure 4-68. Base details of various types of skirt-supported vessels.

Table 4-31
Bolt chair data

Size (in.)	A_{min}	R_a	a_{min}	b	c_{min}
¾–10	5.50	0.302	2	3.50	1.5
⅞–9	5.50	0.419	2	3.50	1.5
1–8	5.50	0.551	2	3.50	1.5
1 ⅛–7	5.50	0.693	2	3.50	1.5
1 ¼–7	5.50	0.890	2	3.50	1.5
1 ⅜–6	5.50	1.054	2.13	3.50	1.75
1 ½–6	5.75	1.294	2.25	3.50	2
1 ⅝–5 ½	5.75	1.515	2.38	4.00	2
1 ¾–5	6.00	1.744	2.5	4.00	2.25
1 7/8–5	6.25	2.049	2.63	4.00	2.5
2–4 ½	6.50	2.300	2.75	4.00	2.5
2¼–4 ½	7.00	3.020	3	4.50	2.75
2½–4	7.25	3.715	3.25	4.50	3
2 ¾–4	7.50	4.618	3.50	4.75	3.25
3–4	8.00	5.621	3.75	5.00	3.50

Table 4-32
Number of anchor bolts, N

Skirt Diameter (in.)	Minimum	Maximum
24–36	4	4
42–54	4	8
60–78	8	12
84–102	12	16
108–126	16	20
132–144	20	24

**See also Table 4-40*

Table 4-33
Allowable stress for bolts, F_s

Spec	Diameter (in.)	Allowable Stress (KSI)
A-307	All	20.0
A-36	All	19.0
A-325	<1-1/2”	44.0
A-449	<1”	39.6
	1-1/8” to 1-1/2”	34.7
	1-5/8” to 3”	29.7

Table 4-34
Average properties of concrete

Water Content/ Bag	Ult 28–Day Str (psi)	Allowable Compression, F_c (psi)	Allowable B_p (psi)	Coefficient, n
7.5	2000	800	500	15
6.75	2500	1000	625	12
6	3000	1200	750	10
5	3750	1400	938	8

**See also Table 4-43*
Reprinted by permission of John Wiley & Sons, Inc.

Table 4-35
Bending moment unit length

ℓ/b	$M_x\begin{pmatrix} x = 0.5b \\ y = \ell \end{pmatrix}$	$M_y\begin{pmatrix} x = .5b \\ y = 0 \end{pmatrix}$
0	0	$-0.5f_c\,\ell^2$
0.333	$0.0078f_cb^2$	$-0.428f_c\,\ell^2$
0.5	$0.0293f_cb^2$	$-0.319f_c\,\ell^2$
0.667	$0.0558f_cb^2$	$-0.227f_c\,\ell^2$
1.0	$0.0972f_cb^2$	$-0.119f_c\,\ell^2$
1.5	$0.123f_cb^2$	$-0.124f_c\,\ell^2$
2.0	$0.131f_cb^2$	$-0.125f_c\,\ell^2$
3.0	$0.133f_cb^2$	$-0.125f_c\,\ell^2$
∞	$0.133f_cb^2$	$-0.125f_c\,\ell^2$

Reprinted by permission of John Wiley & Sons, Inc.

Table 4-36
Constant for moment calculation, γ_1, and γ_2

b/ℓ	γ_1	γ_2
1.0	0.565	0.135
1.2	0.350	0.115
1.4	0.211	0.085
1.6	0.125	0.057
1.8	0.073	0.037
2.0	0.042	0.023
∞	0	0

Reprinted by permission of John Wiley & Sons, Inc.

Table 4-37
Values of constants as a function of K

K	C_c	C_t	J	Z	K	C_c	C_t	J	Z
0.1	0.852	2.887	0.766	0.480	0.55	2.113	1.884	0.785	0.381
0.15	1.049	2.772	0.771	0.469	0.6	2.224	1.765	0.784	0.369
0.2	1.218	2.661	0.776	0.459	0.65	2.333	1.640	0.783	0.357
0.25	1.370	2.551	0.779	0.448	0.7	2.442	1.510	0.781	0.344
0.3	1.510	2.442	0.781	0.438	0.75	2.551	1.370	0.779	0.331
0.35	1.640	2.333	0.783	0.427	0.8	2.661	1.218	0.776	0.316
0.4	1.765	2.224	0.784	0.416	0.85	2.772	1.049	0.771	0.302
0.45	1.884	2.113	0.785	0.404	0.9	2.887	0.852	0.766	0.286
0.5	2.000	2.000	0.785	0.393	0.95	3.008	0.600	0.760	0.270

Reprinted by permission of John Wiley & Sons, Inc.

1. Width of base ring
2. Quantity of anchor bolts
3. Sizes of anchor bolts
4. Strength of anchor bolts
5. Strength of concrete

If the width of the base plate is increased, the neutral axis will be displaced toward the compression side and the stresses in the concrete and steel will be reduced. The maximum compressive stress between base plate and the concrete occurs at the outer periphery of the base plate. When uplift occurs, part of the base plate lifts up, resulting in a shift of the neutral axis toward the compression side.

The value of K represents the location of the neutral axis between the anchor bolts in tension and the concrete in compression. A preliminary value of K is estimated based on a ratio of the "allowable" stresses of the anchor bolts and concrete and a ratio of the modulus of elasticity of the two materials. From this preliminary value, anchor bolt sizes and numbers are determined and actual stresses computed. Using these actual stresses, the location of the neutral axis is found and thus an actual corresponding K value. A comparison of these K values tells the designer whether the location of the neutral axis that was assumed for selection of anchor bolts was accurate. In successive trials, the anchor bolt sizes and quantity and width of base plate can be varied to obtain an optimum design. At each trial a new K is estimated and calculations repeated until the estimated K and actual K are approximately equal. This indicates both a balanced design and accurate calculations.

Rather than apportioning a load to each anchor bolt, the anchor bolt area is assumed as a continuous uniform cylinder whose thickness corresponds to the area of the bolts.

The equations can be manipulated to find the exact width of base plate required, w_r, for the parameters of each case. The equation is;

$$w_r = [W_b + (C_t\,f_S - C_C\,f_C\,n)r\,t_S]/(C_C\,f_C\,r)$$

Example is based on the illustrated case in this procedure;

Trial 1:

$W_b = 194,000$ Lbs

$C_t = 2.113$

$C_C = 1.884$

$n = 10$

$r = 52.5$ in

$t_S = 0.225$ in

$f_S = 13,660$ PSI

$f_C = 449$ PSI

$$w_r = [194,000 + (2.113\ (13,660) - 1.884(449)\ 10)$$
$$\times 52.5(225)/[1.884(449)52.5] = 9.79 \text{ in}$$

Trial 2:

$C_t = 2.355$

$C_C = 1.610$

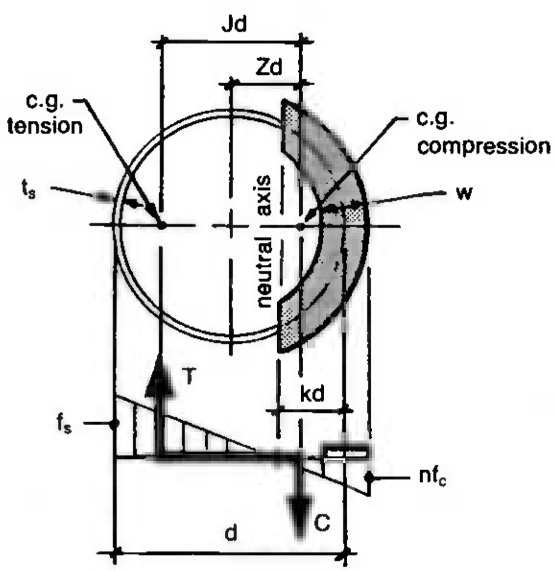
$t_S = 0.225$ in

$f_S = 12,100$ PSI

$f_C = 611$ PSI

$$w_r = [194,000 + (2.355(12,100) - 1.61(611)10)$$
$$\times 52.5(.225)/[1.610(611)52.5] = 8.02 \text{ in}$$

ANCHOR BOLTS: EQUIVALENT AREA METHOD



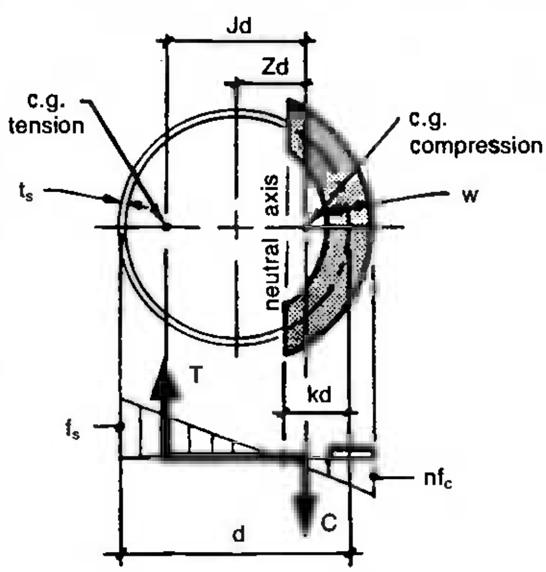
PROCEDURE

1. Calculate preliminary K value based on allowables.
2. Make preliminary selection of anchor bolts and width of base plate.
3. Calculate loads and stresses.
4. Calculate K based on actual stresses and compare with value computed in Step 2.
5. If difference exceeds .01, select a new K between both values and repeat Steps 2–8.

TRIAL 1				TRIAL 2	
1		Data		1 Data	
F_s (Table 4-33)		M_b			
F_c (Table 4-34)		d			
n (Table 4-34)		r			
W_b					
2 Approximate K Using Allowables		**Coefficients**		**2 Approximate K Using Allowables**	
$K = \dfrac{1}{1 + \dfrac{F_s}{nF_c}}$		C_c			
		C_t			
		J			
		Z			
3 Tensile Load in Steel				**3 Tensile Load in Steel**	
$T = \dfrac{M_b - W_b(Zd)}{Jd}$					
4 Number of Anchor Bolts Required				**4 Number of Anchor Bolts Required**	
$A_b = \dfrac{T\pi d}{F_s r C_t}$		R_a (Table 4-31)	in.^{2}		
A_b/N		Use ( )	bolts		
5 Stress in Equivalent Steel Band				**5 Stress in Equivalent Steel Band**	
$t_s = \dfrac{NR_a}{\pi d}$		$f_s = \dfrac{T}{t_s r C_t}$			
6 Compressive Load in Concrete				**6 Compressive Load in Concrete**	
$C = T + W_b$					
7 Stress in Concrete				**7 Stress in Concrete**	
$f_c = \dfrac{C}{[(w - t_s) + nt_s]rC_c}$					
8 Recheck K Using Actual f_s and f_c				**8 Recheck K Using Actual f_s and f_c**	
$K = \dfrac{1}{1 + \dfrac{f_s}{nf_c}}$					

See example of completed form on next page.

ANCHOR BOLTS: EQUIVALENT AREA METHOD EXAMPLE



PROCEDURE

1. Calculate preliminary K value based on allowables.
2. Make preliminary selection of anchor bolts and width of base plate.
3. Calculate loads and stresses.
4. Calculate K based on actual stresses and compare with value computed in Step 2.
5. If difference exceeds .01, select a new K between both values and repeat Steps 2–8.

TRIAL 1				TRIAL 2
1	**Data**			**1 Data**
F_s (Table 4-33)	= 15 KSI	M_b = 3034 FT-KIPS		USE W = 8.25" and K = .34
F_c (Table 4-34)	= 1.2 KSI	d = 8.75' or 105"		
n (Table 4-34)	= 10	r = 4.38' or 52.5"		
W_b	= 194 KIPS			
2 Approximate K Using Allowables		**Coefficients**		**2 Approximate K Using Allowables**
$K = \frac{1}{1 + \frac{F_s}{nF_c}}$	= .444	C_c = 1.884		K = .34
		C_1 = 2.113		C_c = 1.610, J = .7826
		J = .785		C_t = 2.355, Z = .429
		Z = .404		
3 Tensile Load in Steel				**3 Tensile Load in Steel**
$T = \frac{M_b - W_b(Zd)}{Jd}$	$= \frac{3034 - 194(.404)8.75}{.785(8.75)} = 341^K$			336.7^K
4 Number of Anchor Bolts Required				**4 Number of Anchor Bolts Required**
$A_b = \frac{T\pi d}{F_s r C_1} = \frac{341\,\pi\,8.75}{15(4.38)2.113} = 67.5\ in^2$		R_a (Table 4-31)	= 3.715 in.²	$\frac{336.7\,\pi\,8.75}{15(4.36)2.355} = 59.82$; $\frac{59.82}{20} = 2.99\ in^2$
$A_b/N = 67.5/20 = 3.37$		Use (20)	2½" ϕ bolts	Use (20) 2½" ϕ BOLTS
5 Stress in Equivalent Steel Band				**5 Stress in Equivalent Steel Band**
$t_s = \frac{NR_a}{\pi d} = \frac{20(3.715)}{\pi\ 105} = .225$	$f_s = \frac{T}{t_s r C_1} = \frac{341}{.225(52.5)2.113} = 13.66$ KSI < 15 KSI OK			$t_s = .225$; $f_s = \frac{336.7}{.225(52.5)2.355} = 12.10$ OK
6 Compressive Load in Concrete				**6 Compressive Load in Concrete**
$C = T + W_b = 341 + 194 = 535^K$				$C = 336.7 + 194 = 530.7^K$
7 Stress in Concrete				**7 Stress in Concrete**
$f_c = \frac{C}{[(w - t_s) + nt_s]rC_c} = \frac{535}{[(10-.225)+2.25]52.5(1.884)} = .449$ KSI				$f_c = \frac{530.7}{10.275(52.5)1.61} = .611$ KSI
8 Recheck K Using Actual f_s and f_c				**8 Recheck K Using Actual f_s and f_c**
$K = \frac{1}{1 + \frac{f_s}{nf_c}} = \frac{1}{1 + \frac{13.66}{10(.449)}} = .247 \neq .444$ NO GOOD!				$K = \frac{1}{1 + \frac{12.10}{10(.611)}} = .336$ OK $\approx .34$

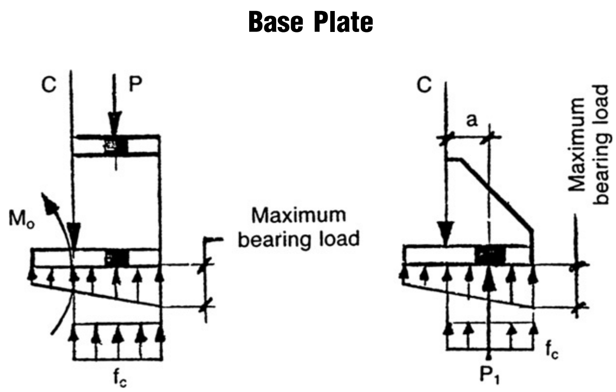


Figure 4-69. Loading diagram of base plate with gussets and chairs.

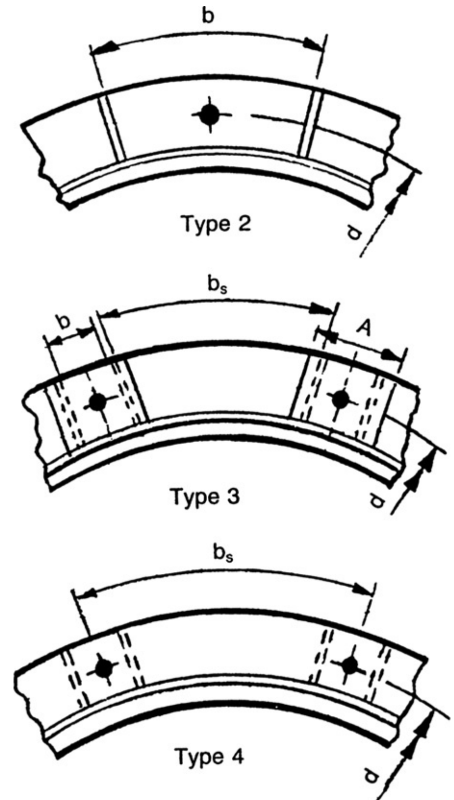


Figure 4-70. Dimensions of various base plate configurations.

Type 1: Without Chairs or Gussets

K = from "Anchor Bolts."

l =

f_c = from "Anchor Bolts."

d =

- *Bending moment per unit length.*

$$M_o = 0.5 f_c l^2$$

- *Maximum bearing load.*

$$b_p = f_c\left(\frac{2Kd + w}{2Kd}\right) < B_p \text{ (see Table 4-34)}$$

- *Thickness required.*

$$t_b = \sqrt{\frac{6M_o}{F_b}}$$

Type 2: With Gussets Equally Spaced, Straddling Anchor Bolts

- *With same number as anchor bolts.*

$$b = \frac{\pi d}{N}$$

$$\frac{l}{b}$$

M_o = greater of M_x or M_y from Table 4-35

$$t_b = \sqrt{\frac{6M_o}{F_b}}$$

- *With twice as many gussets as anchor bolts.*

$$b = \frac{\pi d}{2N}$$

$$\frac{l}{b}$$

M_o = greater of M_x or M_y from Table 4-35

$$t_b = \sqrt{\frac{6M_o}{F_b}}$$

Type 3 or 4: With Anchor Chairs or Full Ring

- *Between gussets.*

$$P = F_s R_a$$

$$M_o = \frac{Pb}{8}$$

$$t_b = \sqrt{\frac{6M_o}{(w - d_b)F_b}}$$

- *Between chairs.*

$\frac{\ell}{b_s}$

M_o = greater of M_x or M_y from Table 4-35

$$t_b = \sqrt{\frac{6M_o}{F_b}}$$

Top Plate or Ring (Type 3 or 4)

- *Minimum required height of anchor chair (Type 3 or 4).*

$$h_{min} = \frac{7.29\delta d}{H} < 18 \text{ in.}$$

- *Minimum required thickness of top plate of anchor chair.*

$$t_c = \sqrt{\frac{P}{F_b e}(0.375b - 0.22d_b)}$$

Top plate is assumed as a beam, e x A with partially fixed ends and a portion of the total anchor bolt force P/3, distributed along part of the span. (See Figure 4-71.)

- *Bending moment, M_o, in top ring (Type 4).*

$\frac{b}{\ell}$

γ_1 = (see Table 4-36)
γ_2 = (see Table 4-36)

1. If $a = \ell/2$ and $b/\ell > 1$, M_y governs

$$M_o = \frac{P}{4\pi}\left[(1+\nu)\log\left(\frac{2\ell}{\pi g}\right) + (1-\gamma_1)\right]$$

2. If $a \neq \ell/2$ but $b/\ell > 1$, M_y governs

$$M_o = \frac{P}{4\pi}\left[(1+\nu)\log\left(\frac{2\ell \sin\frac{\pi a}{\ell}}{\pi g}\right) + 1\right] - \frac{\gamma_1 P}{4\pi}$$

3. If $b/\ell < 1$, invert b/ℓ and rotate axis X-X and Y-Y 90°

$$M_o = \frac{P}{4\pi}\left[(1+\nu)\log\left(\frac{2\ell \sin\frac{\pi a}{\ell}}{\pi g}\right) + 1\right] - \left[(1-\nu-\gamma_2)\frac{P}{4\pi}\right]$$

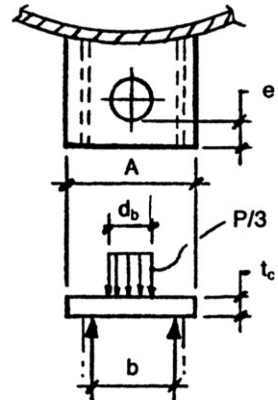


Figure 4-71. Top plate dimensions and loadings.

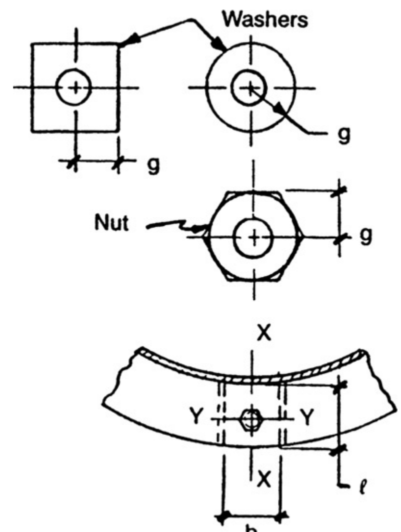


Figure 4-72. Compression plate dimensions.

- *Minimum required thickness of top ring (Type 4).*

$$t_c = \sqrt{\frac{6M_o}{F_b}}$$

Gussets

- *Type 2.* Assume each gusset shares load with each adjoining gusset. The uniform load on the base is f_c, and the area supported by each gusset is $\ell \times b$. Therefore the load on the gusset is

$$P_1 = f_c \ell b$$

Thickness required is

$$t_g = \frac{P_1(6a - 2\ell)}{F_b \ell^2}$$

- Type 3 or 4.

$$t_g = \frac{P}{18{,}000\,\ell} > \frac{3}{8} \text{ in.}$$

Skirt

- *Thickness required in skirt at compression plate or ring due to maximum bolt load reaction.*
  For Type 3:

$$Z = \frac{1.0}{\frac{1.77At_b}{\sqrt{Rt_{sk}}}\left[\frac{t_b}{t_{sk}}\right]^2 + 1}$$

$$S = \frac{Pa}{t_{sk}^2}\left[\frac{1.32Z}{\frac{1.43Ah^2}{Rt_{sk}} + \left[4Ah^2\right]^{0.333}} + \frac{0.031}{\sqrt{Rt_{sk}}}\right] < 25 \text{ ksi}$$

For Type 4:
Consider the top compression ring as a uniform ring with N number of equally spaced loads of magnitude.

$$\frac{Pa}{h}$$

See Procedure 7-1 for details.

The moment of inertia of the ring may include a portion of the skirt equal to 16 t_{sk} on either side of the ring (see Figure 4-74).

- *Thickness required at opening of skirt.*
  Note: If skirt is stiffened locally at the opening to compensate for lost moment of inertia of skirt cross section, this portion may be disregarded.
  G = width of opening, in.

$$f_b = \frac{1}{\pi D - 3G}\left[\frac{48\ M_b}{D} + W_b\right]$$

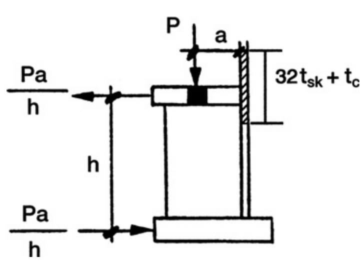


Figure 4-73. Dimensions and loadings on skirt due to load P.

Actual weights and moments at the elevation of the opening may be substituted in the foregoing equation if desired.

Skirt thickness required:

$$t_{sk} = \frac{f_b}{8F_y} \quad \text{or} \quad \sqrt{\frac{f_b}{4{,}640{,}000}}$$

whichever is greater

- *Determine allowable longitudinal stresses.*
  Tension

$$S_t = \text{lesser of } 0.6F_y \text{ or } 1.2\,S$$

Compression

$$S_c = 0.333\ F_y$$

$$= 1.2 \times \text{factor “B”}$$

$$= \frac{t_{sk}E_1}{16\ R}$$

$$= 1.2\,S$$

whichever is less.
Longitudinal forces

$$F_{LT} = \frac{48\ M_b}{\pi D^2} - \frac{W_b}{\pi D}$$

$$F_{LC} = (-)\frac{48\ M_b}{\pi D^2} - \frac{W_b}{\pi D}$$

Skirt thickness required

$$t_{sk} = \frac{F_{LT}}{S_t} \quad \text{or} \quad \frac{F_{LC}}{S_c}$$

whichever is greater.

- *Thickness required at skirt-head attachment due to M_t.*
  Longitudinal forces

$$F_{LT} = \frac{48\ M_t}{\pi D^2} - \frac{W_t}{\pi D}$$

$$F_{LC} = (-)\frac{48\ M_t}{\pi D^2} - \frac{W_t}{\pi D}$$

Skirt thickness required

$$t_{sk} = \frac{F_{LT}}{0.707\ S_tE} \quad \text{or} \quad \frac{F_{LC}}{0.707S_cE}$$

whichever is greater.

Notes

1. Base plate thickness:
   - If $t \leq$ ½ in., use Type 1.
   - If ½ in. $< t \leq$ ¾ in., use Type 2.
   - If $t >$ ¾ in., use Type 3 or 4.
2. To reduce sizes of anchor bolts:
   - Increase number of anchor bolts.
   - Use higher-strength bolts.
   - Increase width of base plate.
3. Number of anchor bolts should always be a multiple of 4. If more anchor bolts are required than spacing allows, the skirt may be angled to provide a larger bolt circle or bolts may be used inside and outside of the skirt. Arc spacing should be kept to a minimum if possible.
4. The base plate is not made thinner by the addition of a compression ring, t_b would be the same as required for chair-type design. Use a compression ring to reduce induced stresses in the skirt or for ease of fabrication when chairs become too close.
5. Dimension "a" should be kept to a minimum to reduce induced stresses in the skirt. This will provide a more economical design for base plate, chairs, and anchor bolts.
6. For heavy-wall vessels, it is advantageous to have the center lines of the skirt and shell coincide if possible. For average applications, the O.D. of the vessel and O.D. of the skirt should be the same.
7. Skirt thickness should be a minimum of R/200.

Procedure 4-15: Design of Base Details for Vertical Vessels – Centered Neutral Axis Method

Notation

E = joint efficiency
E_1 = modulus of elasticity at design temperature, psi
A_b = cross-sectional area of bolts, in.^{2}
d = diameter of bolt circle, in.
W_b = weight of vessel at base, lb
W_T = weight of vessel at tangent line, lb
w = width of base plate, in.
S = code allowable stress, tension, psi
N = number of anchor bolts
F'_c = allowable bearing pressure, concrete, psi
F_y = minimum specified yield stress, skirt, psi
F_s = allowable stress, anchor bolts, psi
f_{LT} = axial load, tension, lb/in.-circumference
f_{LC} = axial load, compression, lb/in.-circumference
F_T = allowable stress, tension, skirt, psi
F_c = allowable stress, compression, skirt, psi
F_b = allowable stress, bending, psi
f_s = tension force per bolt, lb
f_c = bearing pressure on foundation, psi
M_b = overturning moment at base, ft-lb
M_T = overturning moment at tangent line, ft-lb

Allowable Stresses

$$F_T = \text{lesser of} \begin{cases} \bullet 0.6F_y = \\ \bullet 1.2\,S = \end{cases}$$

$$F_c = \text{lesser of} \begin{cases} \bullet\, 0.333F_y = \\ \bullet 1.2 \text{ Factor B} = \\ \bullet \dfrac{t_{sk}E_1}{16\ R} = \\ \bullet 1.2\,S = \end{cases}$$

$$F_b = 0.6\,F_y$$

$$F'_c = 500 \text{ psi for } 2000 \text{ lb concrete}$$
$$750 \text{ psi for } 3000 \text{ lb concrete}$$

$$\text{Factor A} = \frac{0.125t_{sk}}{R} =$$

Factor B = from applicable material chart of ASME Code, Section II, Part D, Subpart 3

Anchor Bolts

- *Force per bolt due to uplift.*

$$f_s = \frac{48M_b}{dN} - \frac{W_b}{N}$$

- *Required bolt area, A_b.*

$$A_b = \frac{f_s}{F_s} =$$

Use ( ) _____ diameter bolts
Note: Use four ¾-in.-diameter bolts as a minimum.

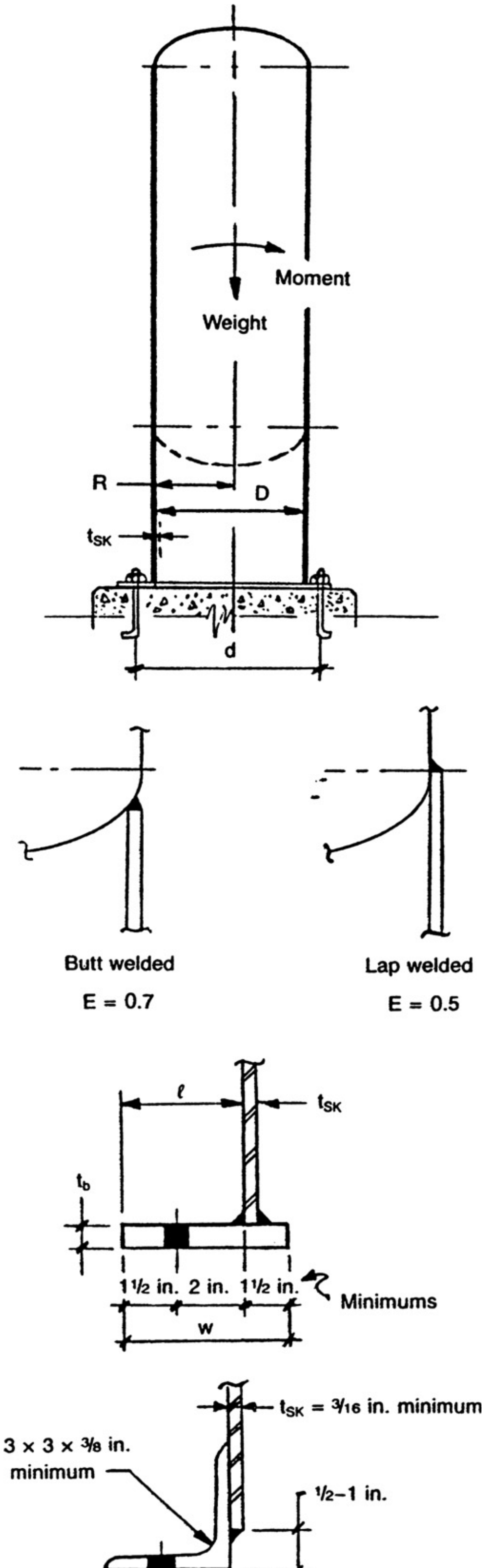


Figure 4-74. Typical dimensional data and forces for a vertical vessel supported on a skirt.

Base Plate

- *Bearing pressure, f_c (average at bolt circle).*

$$f_c = \frac{48M_b}{\pi d^2 w} + \frac{W_b}{\pi dw} =$$

- *Required thickness of base plate, t_b.*

$$t_b = 1\sqrt{\frac{3f_c}{20,000}}$$

Skirt

- *Longitudinal forces, f_{LT} and f_{LC}.*

$$f_{LT} = \frac{48M_b}{\pi D^2} - \frac{W_b}{\pi D}$$

$$f_{LC} = (-)\frac{48M_b}{\pi D^2} - \frac{W_b}{\pi D}$$

- Thickness required of skirt at base plate, t_{sk}.

$$t_{sk} = \text{greater of } \frac{f_{LT}}{F_T} =$$

$$\text{or } \frac{f_{LC}}{F_C} =$$

- Thickness required of skirt at skirt-head attachment.

Longitudinal forces:

$$f_{LT}, f_{LC} = \pm\frac{48M_T}{\pi D^2} - \frac{W_T}{\pi D} =$$

$$f_{LT} =$$

$$f_{LC} =$$

Thickness required:

$$t_{sk} = \text{greater of } \frac{f_{LT}}{0.707\ F_T E} =$$

$$\text{or } \frac{f_{LC}}{0.707\ F_C E} =$$

Notes

1. This procedure is based on the centered neutral axis method and should be used for relatively small or simple vertical vessels supported on skirts.
2. If moment M_b is from seismic, assume W_b as the operating weight at the base. If M_b is due to wind, assume empty weight for computing the maximum value of f_{LT} and operating weight for f_{LC}.

Procedure 4-16: Design of Anchor Bolts for Vertical Vessels

Notation

A_b = Cross sectional area of anchor bolt, in^2
A_r = Area of one anchor bolt required, In^2
D_b = Diameter of bolt circle, Ft
M = Overturning moment due to wind or seismic, Ft-lbs
N = Number of anchor bolts
S_b = Allowable tensile stress, PSI
W = Weight of vessel under consideration. Typically use empty for wind and full for seismic for worst case, Lbs

Formulas

$$N\,A_b = [(48\,M/D_b) - W]\,[1/S_b]$$

- If N A_b is negative, no anchor bolts are required
- If N A_b is positive, than anchor bolts are required
- Size of anchor bolts required is as follows, A_r;

$$A_r = [(48\,M/D_b) - W]\,[1/(N\,S_b)]$$

Notes

1. Values for S_b in table are based on .333 F_U
2. Assumes centered neutral axis method

Table 4-38
Area of anchor bolts, A_b

DIA	A_b	DIA	A_b
¾"–10	.302	1-3/4"–5	1.744
7/8"–9	.419	2"–4-1/2	2.3
1"–8	.551	2-1/2"–4	3.715
1-1/4"–7	.890	2-3/4"–4	4.618
1-1/2"–6	1.294	3"-4	5.621

Table 4-39
Allowable stress, KSI

MATL	DIA	F_y	F_U	S_b
A-36	<4"	36	58	19.14
A-307	<8"	36	60	20
A-193-B7	<2.5"	105	125	41.25
	2.5 –4"	95	115	38
A-449	<1"	92	120	39.6
	1–1.5"	81	105	34.65
	<3"	58	90	29.7

Table 4-40
Recommended quantity and spacing of anchor bolts

Diameter, D		Quantity, N		Spacing, b_S (Ft)	
Ft	In	MIN (1)	MAX (2)	MIN (3)	MAX (4)
2	24	4	4	1.75	6
3	36		4	2.35	
4	48		8	1.57	
5	60		12	1.31	
6	72		12	1.57	
7	84		16	1.37	
8	96		16	1.57	
9	108	8	20	1.41	6
10	120		20	1.57	
11	132		24	1.44	
12	144		24	1.57	
13	156		28	1.46	
14	168		28	1.57	
15	180		32	1.47	
16	192	12	32	1.57	6
17	204		36	1.48	
18	216		36	1.57	
19	228		40	1.49	
20	240		40	1.57	

(*Continued*)

Table 4-40
Recommended quantity and spacing of anchor bolts—cont'd

Diameter, D		Quantity, N		Spacing, b_S (Ft)	
Ft	In	MIN (1)	MAX (2)	MIN (3)	MAX (4)
21	252		44	1.5	
22	264		44	1.57	
23	276	16	48	1.51	6
24	288		48	1.57	
25	300		52	1.51	
26	312		52	1.57	
27	324		56	1.51	
28	336		56	1.57	
29	348		60	1.51	
30	360	20	60	1.57	6
31	372		64	1.52	
32	384		64	1.57	

Notes:

1. Minimum quantity is based on minimum arc spacing of 4' and maximum arc spacing of 6'.
2. Maximum quantity is based on 2D.
3. Minimum spacing of anchor bolts is based on the maximum quantity of anchor bolts, $\pi\ D_{b/N_{max}}$.
4. Maximum spacing is based on 6' max arc spacing as practical limit.
5. Minimum anchor bolt size is 3/4".

Table 4-41
Anchor bolt torque values

Bolt Dia (in)	Tensile Area, R_a	Design Bolt Tension (KIPS) (1) (2)	Torque Bolt Tension (KIPS) (4)	Torque (Ft-Lbs)
		CASE 2: A-449		
0.75 – 10 UNC	0.302	8.5	9.1	85
0.875 – 9 UNC	0.419	11.9	12.8	140
1-8 UNC	.551	15.1	16.8	210
1.25 – 7 UNC	0.89	23.9	27.5	430
1.5 – 6 UNC	1.294	33.8	40.5	760
1.75 – 5 UNC	1.744	42.5	54.9	1200
2 – 4.5 UNC	2.3	53.5	72	1800
2.25 – 4.5 UNC	3.02	69.2	93.5	2630
2.5 -4 UNC	3.715	85.2	115.2	3600
2.75 -4 UNC	4.618	99.3	142.3	4890
3-4 UNC	5.621	113.9	171.7	6440
		CASE 2: A-449		
0.75 – 10 UNC	0.302	22	23.5	220
0.875 – 9 UNC	0.419	29.6	32	350
1-8 UNC	.551	38.8	43.2	540
1.25 – 7 UNC	0.89	53.9	62.1	970
1.5 – 6 UNC	1.294	76	91.2	1710
1.75 – 5 UNC	1.744	68.5	88.2	1930

Table 4-41
Anchor bolt torque values—cont'd

Bolt Dia (in)	Tensile Area, R_a	Design Bolt Tension (KIPS) (1) (2)	Torque Bolt Tension (KIPS) (4)	Torque (Ft-Lbs)
		CASE 2: A-449		
2 – 4.5 UNC	2.3	86.3	116	2900
2.25 – 4.5 UNC	3.02	111.1	150.8	4230
2.5 -4 UNC	3.715	137.2	185.6	5800
2.75 -4 UNC	4.618	159.9	228.9	7870
3-4 UNC	5.621	183.7	276.8	10380

Notes:

1. Values in Table for A-36 and A-307 bolts are based on approximately 25 KSI tensile stress on the tensile area.
2. Values in Table for A-449 bolts are based on .7 F_y tensile stress on the tensile area.
3. The threads and underside of nuts should be waxed prior to installation to reduce friction.
4. Torque bolt tension allows a % increase over bolt tension to allow for loss of pretension due to creep of concrete and bolt material.
5. All torque values result in a tension stress less than .8 F_y.

Procedure 4-17: Properties of Concrete

Notation

f'_C = Ultimate 28 day Compressive Stress, PSI
F_C = Allowable Compressive Stress, PSI
B_P = Allowable Bearing pressure, PSI
E_S = Modulus of elasticity, steel, PSI
E_C = Modulus of elasticity, concrete, PSI
n = Ratio, E_S / E_C

Table 4-42
Soil bearing pressure

Type of Soil	Bearing Pressure, PSF
Rock	4000
Rocky	3000
Gravel	2000
Sandy	1500
Clay	1000

Table 4-43
Allowable stress, concrete

Ultimate 28 Day Compressive Stress, f'_C (PSI)	Allowable Compressive Stress, F_C (PSI) (1)	Allowable Bearing Pressure, B_P (PSI)(2)	Ratio, n
2000	800	500	15
2500	1000	625	12
3000	1200	750	10
3750	1500	938	8
4000	1600	1000	6

Notes:

1. F_C = 40% of f'_C
2. B_P = 25% of f'_C
3. See ACI 318 or AISC Steel construction Manual for F_c based on either ASD or LRFD methods.

References

[1] ASCE/SEI 7-10, Minimum Design Loads for Buildings and Other Structures, American Society of Civil Engineers.
[2] Recommended Practice #11, Wind and Earthquake Design Standards. San Francisco, CA: Chevron Corp.; March 1985.
[3] Bednar HH. Pressure Vessel Design Handbook. Van Nostrand Reinhold Co 1981.
[4] Brownell LE, Young EH. Process Equipment Design. John Wiley and Sons, Inc 1959.
[5] Fowler DW. New Analysis Method for Pressure Vessel Column Supports. Hydrocarbon Processing May 1969.
[6] Manual of Steel Construction. 8th ed. American Institute of Steel Construction, Inc.; 1980. Tables C1.8.1 and 3–36.
[7] Roark RJ. Formulas for Stress and Strain. 4th ed. McGraw Hill; 1971. Table VIII, Cases 1, 8, 9 and 18.
[8] Wolosewick FE. Support for Vertical Pressure Vessels. Petroleum Refiner July 1981:137–40. August 1981, pp. 101–108.
[9] Blodgett O. Design of Weldments. The James F. Lincoln Arc Welding Foundation 1963. Section 4.7.
[10] Local Stresses in Spherical and Cylindrical Shells Due to External Loadings. WRC Bulletin #107, 3rd revised printing April 1972.
[11] Bijlaard PP. Stresses from Radial Loads and External Moments in Cylindrical Pressure Vessels. Welding Journal Research Supplement December 1955:608–17.
[12] Bijlaard PP. Stresses from Radial Loads and External Moments in Cylindrical Pressure Vessels. Welding Journal Research Supplement December 1954:615–23.
[13] Megyesy EF. Pressure Vessel Handbook. 3rd ed. Pressure Vessel Handbook Publishing Co.; 1975. 72–85.
[14] Zick LP. Stresses in Large Horizontal Cylindrical Pressure Vessels on Two Saddle Supports. Welding Research Journal Supplement September 1951.
[15] Moody GB. How to Design Saddle Supports. Hydrocarbon Processing November 1972.
[16] Wolters BJ. Saddle Design—Horizontal Vessels over 13 Feet Diameter. Irvine, CA: Fluor Engineers, Inc.; 1978.
[17] Committee of Steel Plate Producers, Steel Plate Engineering Data, Volume 2, Useful Information on the Design of Plate Structures, American Iron and Steel Institute, Part VII.
[18] Gartner AI. Nomographs for the Solution of Anchor Bolt Problems. Petroleum Refiner, July 1951:101–6.
[19] Manual of Steel Construction. 8th ed. American Institute of Steel Construction, Inc.; 1980. Part 3.
[20] Blodgett O. Design of Welded Structures. The James F. Lincoln Arc Welding Foundation, 7th printing 1975. Section 3.3.

5
Vessel Internals

Procedure 5-1: Design of Internal Support Beds

Process vessels frequently have internal beds that must be supported by the vessel shell. Sand filters, packed columns and reactors with catalyst beds are just a few examples of vessels that have internal beds. The beds are typically supported by a combination of beams, grids and a support ring welded to the vessel shell. The support ring supports the periphery of the grating or tray plates. The support ring can be welded to the vessel wall by fillet welds or full penetration welds for larger loaded members. In some cases they are integrally forged into the vessel wall.

This procedure can be used to design all the various components of a support structure consisting of the following;

1. Beams
2. Support ring
3. Grating
4. Beam seats
5. Beam support clips and bolting
6. Vessel wall

Beam Supports

The beams are typically supported by one of three methods;

1. Beam seats
2. Clips
3. Support ring

Beams should never be welded directly to the vessel wall because of the restraint they would impose on vessel growth. All of the methods listed above will allow for the radial expansion of the vessel due to temperature and pressure. Slotted holes in the clip or for the bolting attaching the beam to a beam seat, allow for expansion.

Beam seats and clips allow the top flange, or top of beam, to be level with the top of the support ring. This is a very convenient feature for this type of support. In effect, it creates a level surface to support the grating or tray plates. Conversely, if the beams are supported by the support ring, then the bottom flange of the beam will sit on top of the support ring. For most applications this would be unacceptable. In order to make the top of the bottom flange of the beam to be on the same plane as the top of the support ring, support blocks are used. Support blocks can be used for solid beams, lattice beams or built up beams. For certain applications, the web of the beam can be extended to create the support surface in lieu of support blocks, or support blocks can be added to each side of the web to reinforce the web locally and provide a larger bearing area.

Beams

There is really no limit to the number of cross beams that may be utilized, however this procedure only illustrates the use of up to eight beams. If access through the center of the vessel is a concern, then only an even number of beams can be used. A center beam would impede access. Beams can be solid members, welded I-Beam type, welded T-type, or open, lattice type structures.

One additional support method, not shown here, is the "Hub Ring" or "Ring Beam" approach. Hub rings are ideal for many applications. They consist of two parallel rings attached to radial beams. The radial beams in turn are supported by the vessel shell or support rings. The upper ring is a compression ring. The lower ring is a tension member. The rings are naturally stiff members that are loaded in their strongest direction, perpendicular to the applied load. In theory they employ all the strength aspects of the arch, and thus are sometimes referred to as "arcuate beams". Design of hub rings with radial beams are not detailed in this procedure. Only the design of straight beams are included.

Lattice beams are desirable, if possible, due to their light weight. Where internals are stainless steel to prevent or minimize corrosion, they will also reduce cost. Unlike solid or built-up beams, the top and bottom members of a lattice beam have distinct functions. The bottom chord is the tension only member. The top chord is a compression only member. This is for beams above the line of support. For beams below the line of support, the reverse would be true.

In a lattice beam, the top chord is a compression member and must be capable of supporting a column axial load without buckling. A T-type compression member is sometimes used to provide extra lateral stability in the compression chord.

Solid and built–up beams must be checked for lateral stability against buckling. In the event this is exceeded, then either the design must be changed or some anti-buckling supports installed. One style of anti-buckling devices is the anti-buckling comb.

Each beam supports the load consisting of one half the distance between each adjacent beam or the support ring, though this would result in a non-uniform loading. However, one simplifying assumption, which is conservative, allows for a rectangular loading for each beam based on the overall length of the beams. This method neglects the contribution of the support ring, and assumes the beams support the entire load. While not completely accurate, it is much easier to apply.

It is most convenient from a design standpoint to have the beams equally spaced. However, this is not always possible because of the clearances, accessibility and other internals penetration. This procedure gives standard loads for uniformly spaced beams, but also describes the procedure for the design of beams that are not uniformly spaced.

Whether the beams extend above or below the support level is matter of application. Both methods have their pros and cons. Basically, the method of support is determined by the designer to accommodate the space inside the vessel as well as the process function of the bed or tray that the beams are supporting. If the beam extends up into the bed, then a certain amount of media is displaced and removal of media is more difficult. Conversely, in the case where the beams support a tray, then having the beams extend above the tray may impose restrictions on flow. Beams for catalyst support grid (CSG) applications almost always extend up into the bed to minimize the length of the vessel.

Loadings

The loading for a bed consists of dead load, liquid load and live load. The dead load consists of the weight of the supports and the media it supports, i.e. catalyst, packing, inert balls, sand, clay, etc. The live load consists of the delta P, differential pressure, developed by the restriction of the bed to downward flow. Typically there is some amount of fouling that occurs in beds that cause a buildup of delta P across the bed during operation. The live load can easily become the major design load for the supports.

The last load is the weight of product. This could be either liquid or solid. It could be the liquid on a tray or the liquid hold-up in the bed itself.

The grating is the support members that span between the beams and support the media. The grating may be covered with screen to prevent egress of small particles. The screen may be installed in a single or multiple layers of mesh. It can also be covered with a layer of wedge wire screen for the same purpose. The grating must be designed to support the total of all loads.

In reactors, the support bed configuration is frequently referred to as a CSG. The CSG consists of beams, grating and screen.

Procedure

The basic design procedure for the beams is as follows;

1. Estimate the dead loads to be supported to include the media and weight of support members.
2. Calculate the live load based on the delta P specified.
3. Calculate the liquid (or solids) loading in the bed.
4. Combine the three load cases to get a total load.
5. The total load divided by the cross sectional area is the uniform design load, p.
6. Calculate the area supported by each beam and multiply this area times the uniform load to get the load supported by each beam.
7. A standard AISC type beam type analysis is then performed to size the beam.

Notation

A_r = Area of bolt required, in^2
A_n = Total area supported by beam, in^2
A_c = Cross sectional area of vessel, in
B = Ratio of actual force to allowable force per inch on weld
C_n = Compressive force in beam, Lbs
DL = Dead load, Lbs
E = Modulus of elasticity, PSI

f_b = Stress, bending, PSI
f_C = Stress, compressive, PSI
f_T = Stress, tension, PSI
F = Total load of bed, Lbs
F_b = Allowable bending stress, PSI
F_{br} = Allowable stress, bearing, PSI
F_C = Allowable stress, compression, PSI
F_S = Allowable stress, shear, PSI
F_y = Minimum specified yield strength at design temperature, PSI
I = Moment of inertia, in^4
K = End connection coefficient per AISC
K_1 = Vertical distance from bottom of beam flange to top of fillet on web, in
LL = Live load, Lbs
M = Moment, in-Lbs
n_b = Number of bolts required
n_g = Number of bearing bars per foot
N_b = Minimum bearing length, in
N = Number of beams
P = Concentrated load, Lbs
PL = Product load, Lbs
P_c = Free area in packing/catalyst, %
p = Uniform load over entire bed, PSI
r = Radius of gyration, in
R = End reactions, Lbs
R_a = Root area of bolt, in^2
S = Allowable stress in shell, tension, PSI
S_U = Minimum specified tensile strength, PSI
t = Thickness, in
T_n = Tension force in beam, Lbs
t_w = Thickness of web, in
W_C = Weight, contents, (catalyst, packing, etc) Lbs
W_e = Weight of entrained liquid, Lbs
W_g = Weight, grating, Lbs
W_b = Weight, beams, Lbs
w_f = Size of fillet weld, in
w_n = Uniform load on beam, Lbs/in
V_c = Volume, catalyst/packing, Ft^3
Z = Section modulus, PSI
δ = Deflection, in
ΔP = Differential pressure loading

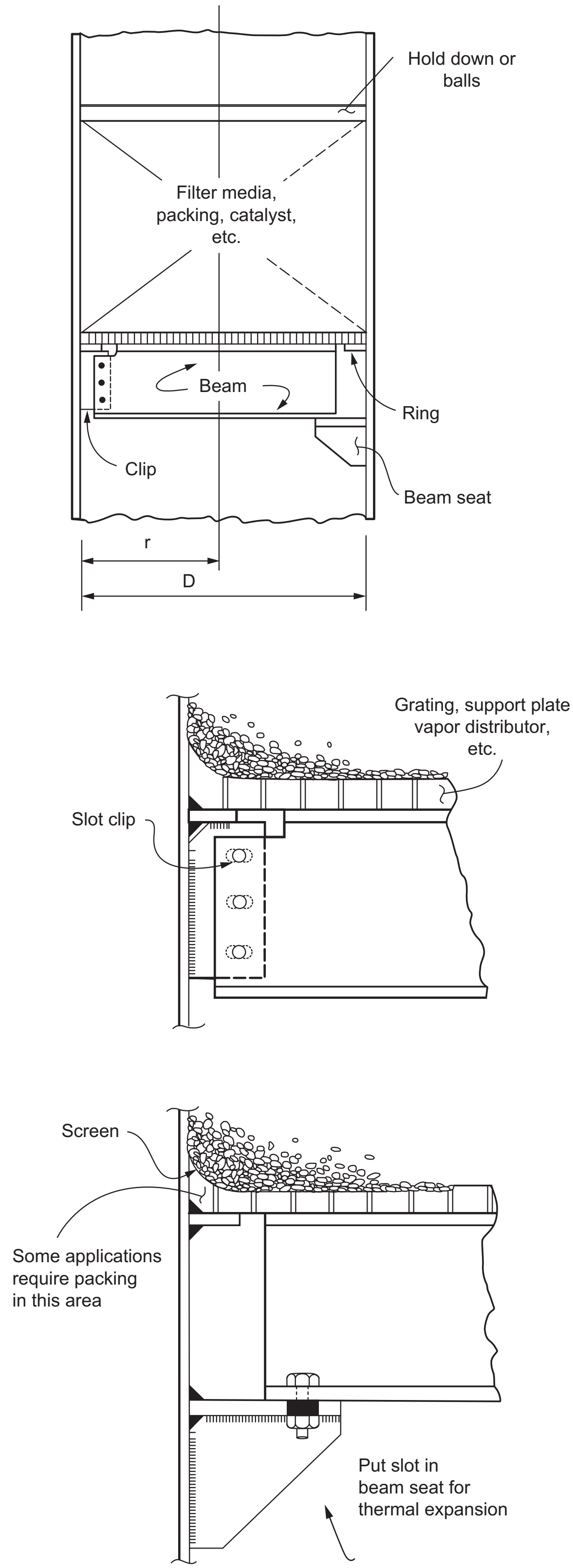


Figure 5-1. Typical support arrangements and details of an internal bed.

CASE 1: SINGLE BEAM	CASE 2: DOUBLE BEAM

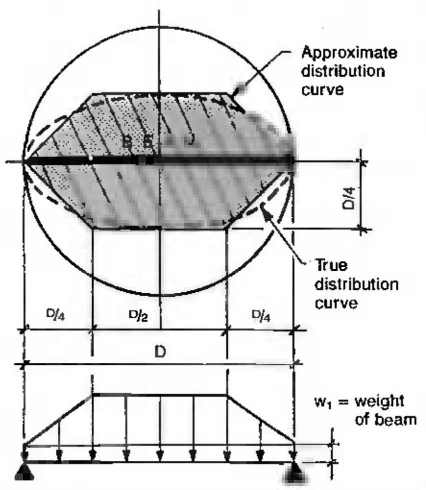


Figure 5.2 Loading diagram of single-beam support. Area of loading = 48%.

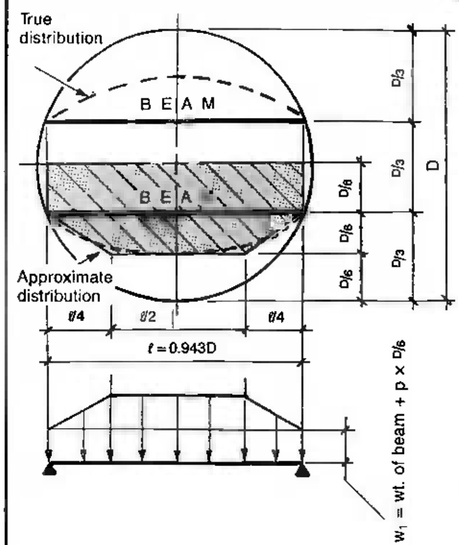


Figure 5.3 Loading diagram of double-beam support. Area of loading = 37% per beam.

BEAM CALCULATIONS

UNIFORM LOAD DUE TO BEAM WEIGHT		UNIFORM LOAD DUE TO BEAM WEIGHT	
$M_1 = \frac{w_1 D^2}{8} =$		$M_1 = \frac{w_1 \ell^2}{8}$	
$R_1 = \frac{w_1 D}{2} =$		$R_1 = \frac{w_1 \ell}{2}$	
CONCENTRATED LOADS ON END SECTIONS		CONCENTRATED LOADS ON END SECTIONS	
$P = p\left(\frac{D}{4}\right)^2 =$		$P = \frac{pD\ell}{48}$	
$R_2 = P =$		$R_2 = P$	
UNIFORM LOAD IN CENTER SECTION		UNIFORM LOAD IN CENTER SECTION	
$M_2 = \frac{PD}{6} =$		$M_2 = \frac{P\ell}{6}$	
$w_2 = \frac{pD}{2} =$		$w_2 = \frac{pD}{6}$	
$R_3 = \frac{w_2 D}{4} =$		$R_3 = \frac{w_2 \ell}{4}$	
$M_3 = \frac{R_3 D}{4} + \frac{R_3^2}{2w_2} =$		$M_3 = \frac{R_3 \ell}{4} + \frac{R_3^2}{2w_2}$	

MOMENT AND REACTION CALCULATIONS

Total moment $M = M_1 + M_2 + M_3$		$Z_{reqd} = \frac{M}{F_b}$	New	
Total end reaction $R = R_1 + R_2 + R_3$			Corroded	

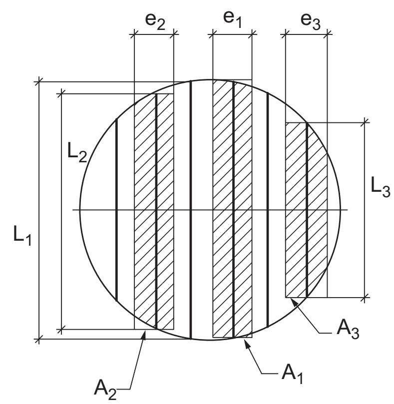
Select beam and add appropriate correction allowance to web and flange

DESIGN LOADS FOR BEAMS				
DATA			LOADINGS	
ITEM	SYMBOL	VALUE	ITEM	VALUE
Material, beams and grating			Dead load, DL	
Material, vessel shell			a. Weight of beams	
Quantity of beams	N		b. Weight of grating	
Vessel ID	D		c. Weight of tray	
Design temperature	DT		d. Weight of screen	
Differential pressure	ΔP		e. Weight of packing/catalyst	
Bed depth	d_C		Product load, PL	
Liquid/contents unit weight	w_c		a. Weight of liquid on tray	
Corrosion allowance	C_a		b. Weight of liquid in bed	
Specific gravity	S_g		c. Weight of liquid above bed	
Liquid holdup (%)			d. Weight of solids	
Free area in bed (%)	P_C		Live load, LL	
Packing/catalyst unit weight	w_P			
Volume of packing/catalyst	V_P		Total load, F, Lbs	
Packing/catalyst total wt	W_C		$F = DL + PL + LL$	
Weight of entrained liquid	W_e			
Weight of grating	W_g		Total cross sectional area, A_c	
Weight of beams	W_b		$A_c = \pi r^2$	
Miscellaneous				
MATERIAL PROPERTIES & ALLOWABLE STRESS			Uniform load, p, PSI	
Shell at design temperature	S		$p = F / A_c$	
	E			
	F_y			
Beams at design temperature	E			
	F_y			
Bending = $.66 F_y$	F_b			
Compression, dependent on KL/r ratio	F_C			
Shear = $.4 F_y$	F_S			
Bearing = $.9 F_y$	F_{br}			

DESIGN OF SUPPORT BEAMS

DIMENSIONS

(6) Beams shown for example only!



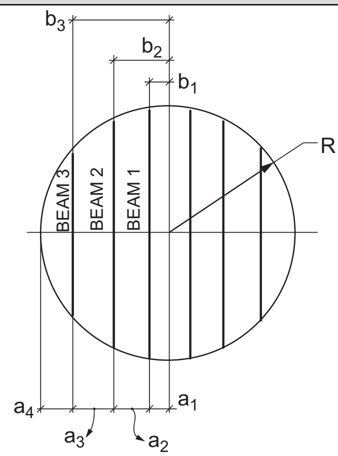
1.0 AREA OF LOADING

BEAM	L_n	e_n	A_n
1			
2			
3			
4			

2.0 LOADS ON BEAMS

BEAM	F_n	R_n	w_n
1			
2			
3			
4			

DIMENSIONS



3.0 MOMENT & FORCES

BEAM	M_n	h_n	T_n or C_n
1			
2			
3			
4			

4.0 BEAM STRESSES

BEAM	f_T	f_C	
1			
2			
3			
4			

5.0 ALLOWABLE STRESS, COMPRESSION

BEAM	L_n	$K\,L_n / r$	F_c
1			
2			
3			
4			

EQUATIONS

$e_1 = a_1 + .5\,a_2$

$e_2 = .5\,a_2 + .5\,a_3$

$e_3 = .5\,a_3 + .5\,a_4$

$e_4 = .5\,a_4 + .5\,a_5$

$L_n = 2\,[R^2 - b_n^2\,]^{1/2}$

$f_T = T_n / A_T$

$f_C = C_n / A_c$

$$C_C = \sqrt{\frac{2\,\pi^2 E}{F_y}}$$

$$F_C = \frac{\left[1 - \frac{(KL/r)^2}{2\,C_C^2}\right] F_y}{5\backslash 3 + \frac{3(KL/r)}{8\,C_C} - \frac{(KL/r)^3}{8\,C_C^3}}$$

$A_n = L_n\ e_n$

$F_n = A_n\ p$

$R_n = w_n\ L_n / 2$

$w_n = F_n / L_n$

$M_n = w_n\ L_n^2 / 8$

$T_n = C_n = M_n / h_n$

Notes:

1. h will vary depending on the type of beam used.
2. Every beam type, solid, built-up or lattice... will all have a tension and compression zone, section or component.

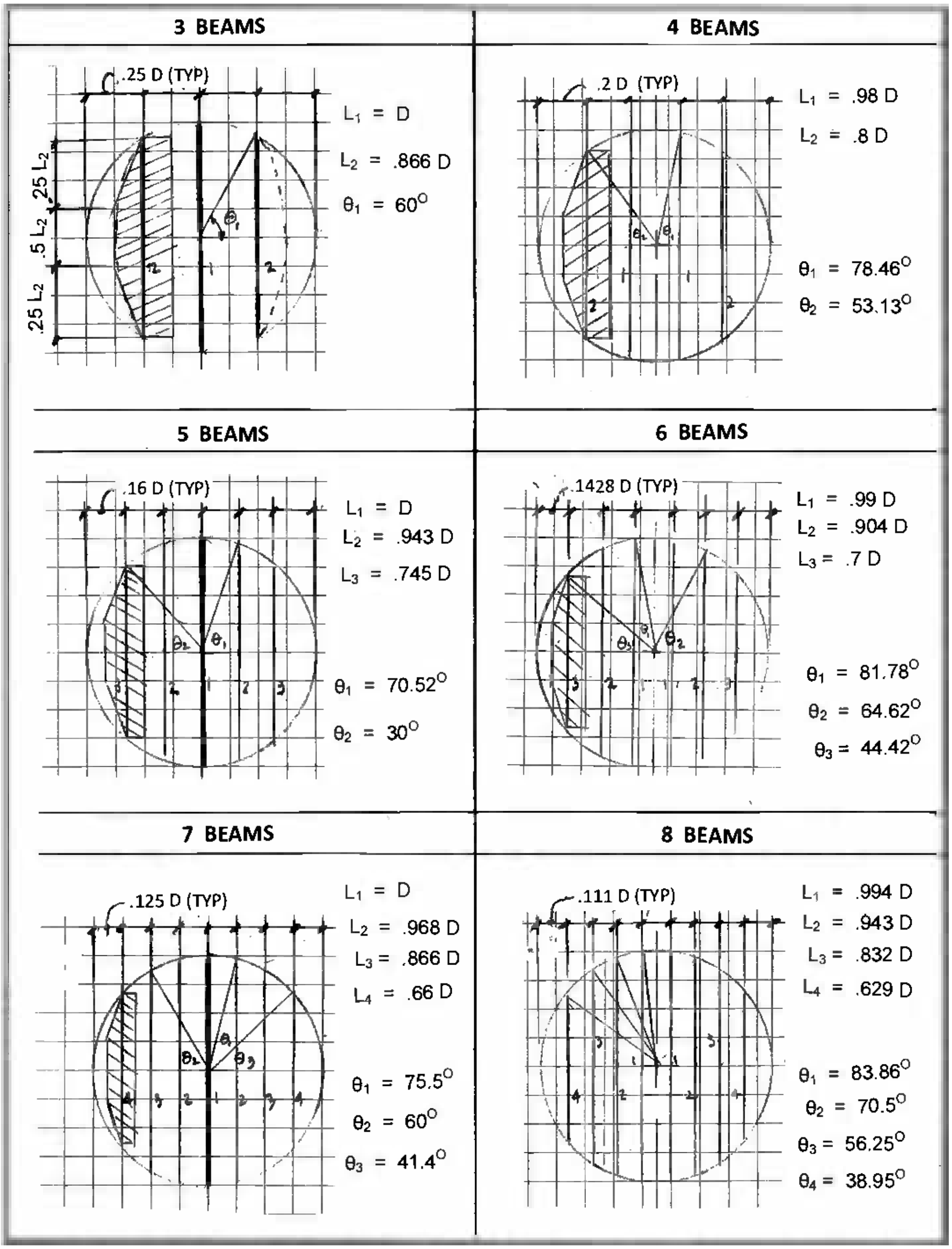

3 BEAMS
.25 D (TYP)
.25 L2
.5 L2
.25 L2
L1 = D
L2 = .866 D
θ1 = 60°
4 BEAMS
.2 D (TYP)
L1 = .98 D
L2 = .8 D
θ1 = 78.46°
θ2 = 53.13°
5 BEAMS
.16 D (TYP)
L1 = D
L2 = .943 D
L3 = .745 D
θ1 = 70.52°
θ2 = 30°
6 BEAMS
.1428 D (TYP)
L1 = .99 D
L2 = .904 D
L3 = .7 D
θ1 = 81.78°
θ2 = 64.62°
θ3 = 44.42°
7 BEAMS
.125 D (TYP)
L1 = D
L2 = .968 D
L3 = .866 D
L4 = .66 D
θ1 = 75.5°
θ2 = 60°
θ3 = 41.4°
8 BEAMS
.111 D (TYP)
L1 = .994 D
L2 = .943 D
L3 = .832 D
L4 = .629 D
θ1 = 83.86°
θ2 = 70.5°
θ3 = 56.25°
θ4 = 38.95°

Table 5-1
Beam supports - Summary of forces and moments

Quantity of Beams	Beam	L_n	A_n	F_n	R_n	M_n	w_n
1	1	D	$.3927\ D^2$	$.3927\ p\ D^2$	$.1864\ p\ D^2$	$.0565\ p\ D^3$	.393 p D
2	1	.943 D	$.2698\ D^2$	$.2698\ p\ D^2$	$.1349\ p\ D^2$	$.0343\ p\ D^3$	.286 p D
3	1	D	$.2333\ D^2$	$.2333\ p\ D^2$	$.1167\ p\ D^2$	$.0311\ p\ D^3$	.233 p D
	2	.866 D	$.1850\ D^2$	$.1850\ p\ D^2$	$.0925\ p\ D^2$	$.0219\ p\ D^3$	.214 p D
4	1	.98 D	$.1925\ D^2$	$.1925\ p\ D^2$	$.0963\ p\ D^2$	$.0240\ p\ D^3$	.196 p D
	2	.8 D	$.1405\ D^2$	$.1405\ p\ D^2$	$.0703\ p\ D^2$	$.0143\ p\ D^3$	.176 p D
5	1	D	$.1655\ D^2$	$.1655\ p\ D^2$	$.0828\ p\ D^2$	$.0208\ p\ D^3$	.166 p D
	2	.943 D	$.1548\ D^2$	$.1548\ p\ D^2$	$.0774\ p\ D^2$	$.0185\ p\ D^3$	.164 p D
	3	.745 D	$.1092\ D^2$	$.1092\ p\ D^2$	$.0546\ p\ D^2$	$.0107\ p\ D^3$	.147 p D
6	1	.99 D	$.142\ D^2$	$.142\ p\ D^2$	$.071\ p\ D^2$	$.0175\ p\ D^3$	.143 p D
	2	.904 D	$.129\ D^2$	$.129\ p\ D^2$	$.0645\ p\ D^2$	$.0146\ p\ D^3$	.143 p D
	3	.7 D	$.085\ D^2$	$.085\ p\ D^2$	$.0425\ p\ D^2$	$.0074\ p\ D^3$	.121 p D
7	1	D	$.125\ D^2$	$.125\ p\ D^2$	$.0625\ p\ D^2$	$.0156\ p\ D^3$	.125 p D
	2	.968 D	$.121\ D^2$	$.121\ p\ D^2$	$.0605\ p\ D^2$	$.0146\ p\ D^3$	.125 p D
	3	.866 D	$.108\ D^2$	$.108\ p\ D^2$	$.0541\ p\ D^2$	$.0117\ p\ D^3$	.125 p D
	4	.66 D	$.0722\ D^2$	$.0722\ p\ D^2$	$.0360\ p\ D^2$	$.0059\ p\ D^3$	.109 p D
8	1	.994 D	$.110\ D^2$	$.110\ p\ D^2$	$.055\ p\ D^2$	$.0137\ p\ D^3$	.111 p D
	2	.943 D	$.105\ D^2$	$.105\ p\ D^2$	$.052\ p\ D^2$	$.0123\ p\ D^3$	.111 p D
	3	.832 D	$.0924\ D^2$	$.0924\ p\ D^2$	$.046\ p\ D^2$	$.0096\ p\ D^3$	.111 p D
	4	.629 D	$.0611\ D^2$	$.0611\ p\ D^2$	$.0031\ p\ D^2$	$.00048\ p\ D^3$	.097 p D

Notes:

1. Table is for uniformly spaced beams only
2. Equations are as follows;

$A_n = L_n\, e_n$ $F_n = A_n\, p$ $R_n = F_n/2$ or $w_n\, L_n/2$ $M_n = w_n\, L_n^2/8$ $w_n = F_n/L_n$

1.0. Design of Support Clip

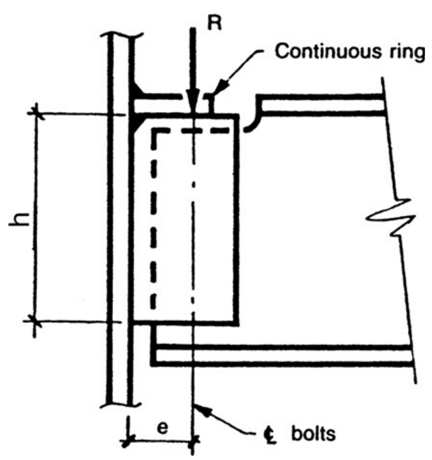


Figure 5-4. Typical clip support.

- Moment in clip, M

$$M = R\ e$$

- Thickness required, t_r

$$t_r = (6\,M)/(h^2\,F_b)$$

- Area required per bolt, A_r

$$A_r = R\ /\ F_s\, n_b$$

Use: n = ____________
Size: ________________
Material: ____________

2.0. Design of Beam Seat

- Thickness required, gusset, t_g

$$T_g = (R(6\,e - 2\,a))/(F_b\, a^2\, \mathrm{Sin}^2\, \phi)$$

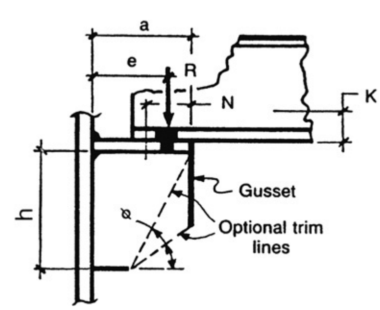


Figure 5-5. Typical beam seat support.

- Bearing length, N_b

$$N_b = [R/(.75\ t_W\ F_y)] - K_1$$

- Ratio B;

For E-60 Welds;

$$B = R/(23040\ W_f)$$

For E-70 Welds;

$$B = R/(26880\ W_f)$$

For other materials use $B = .384\ S_U$

- Required height of gusset, h_r

$$H_r = \left[.5B\left(B + (B^2 + 64\ e^2)^{1/2}\right)\right]^{1/2}$$

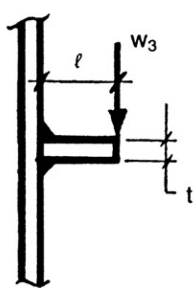


Figure 5-6. Loading diagram of a continuous ring.

3.0. Design of Ring

- Find uniform load, w_3, from table

Qty of Beams	W_3
1	P D / 4
2	P D / 6
3	P D / 8
4	P D /10
5	P D / 12
6	P D / 14
7	P D / 16
8	P D / 18

- Find moment in ring, M_3

$$M_3 = w_3\ L$$

- Thickness required, ring, t_r

$$t_r = (6\ M_3 / F_b)^{1/2}$$

Select appropriate ring size.

4.0. Design of Grating

- Determine maximum span of grating, L_g

$$L_g =$$

- Area of loading for a one foot wide panel, A_g

$$A_g = 12\ L_g$$

- Total load on panel, F_g

$$F_g = pA_g$$

- Uniform loading on one bearing bar, w_g

$$W_g = F_g/L_g\ n_g$$

- Bending moment in one bearing bar, M_4

$$M_4 = w_g L_g/8$$

- Required section modulus of one bearing bar, Z_r

$$Z_r = M_4/F_b$$

- Actual properties of grating;

$$Z = (n_g\ b\ d^2)/6$$

$$I = (n_g\ b\ d^3)/12$$

$$\delta = \left[5\ p\ L_g(12\ L_g)^3\right] / 384\ E\ I$$

Notes

1. Recommended beam ratio, span over depth should be between 15 and 18, 20 maximum.
2. For loading consider packing, catalyst, grating, weight of beams, liquid above packing or filter media, entrained liquid, and differential pressure. The weight of entrained liquid is equal to the volume of bed × the open area × specific gravity × 62.4.
3. Minimum gusset thickness of beam seat should not be less than the web thickness of the beam.
4. Main bearing bars of grating should run perpendicular to the direction of the support beams.
5. Make width of beam seat at least 40% of height.
6. Make fillet weld size no greater than .75 t_w.

BEAMS - LATERAL STABILITY CHECK

SOLID OR BUILT-UP BEAMS

BEAM	L/h	F_c
1		
2		
3		
4		

Determine F_c from the following Table

L/h	F_c
≤7	F_y
≥35	$.66\,[\,(\pi^2 E)\ /\ 12\,(L/h)^2\,]$
7 < L/h < 35	Interpolate between values

NOTES:

1. The allowable buckling stress, F_c, is a function of dimensions L and h.

2. If the allowable buckling stress, F, is exceeded there are two options;

   a. Redesign the beams

   b. Add anti-buckling devices

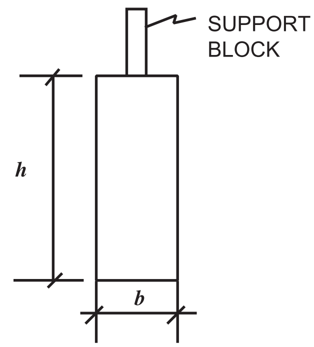
3. Anti-buckling devices consist of two types;

   a. Anti-buckling combs

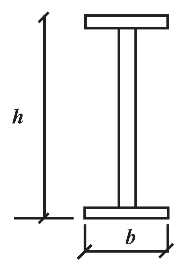
   b. Web stiffeners

4. Anti-buckling devices effectively reduce the "L" dimension and thus increase the allowable stress.

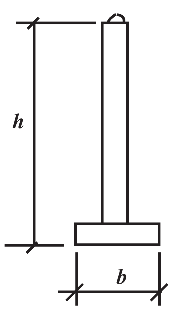
5. Anti-buckling combs are plate devices which fit over the web of multiple beams and reinforce the web.



SOLID BEAM



BUILT-UP BEAM



T-BEAM

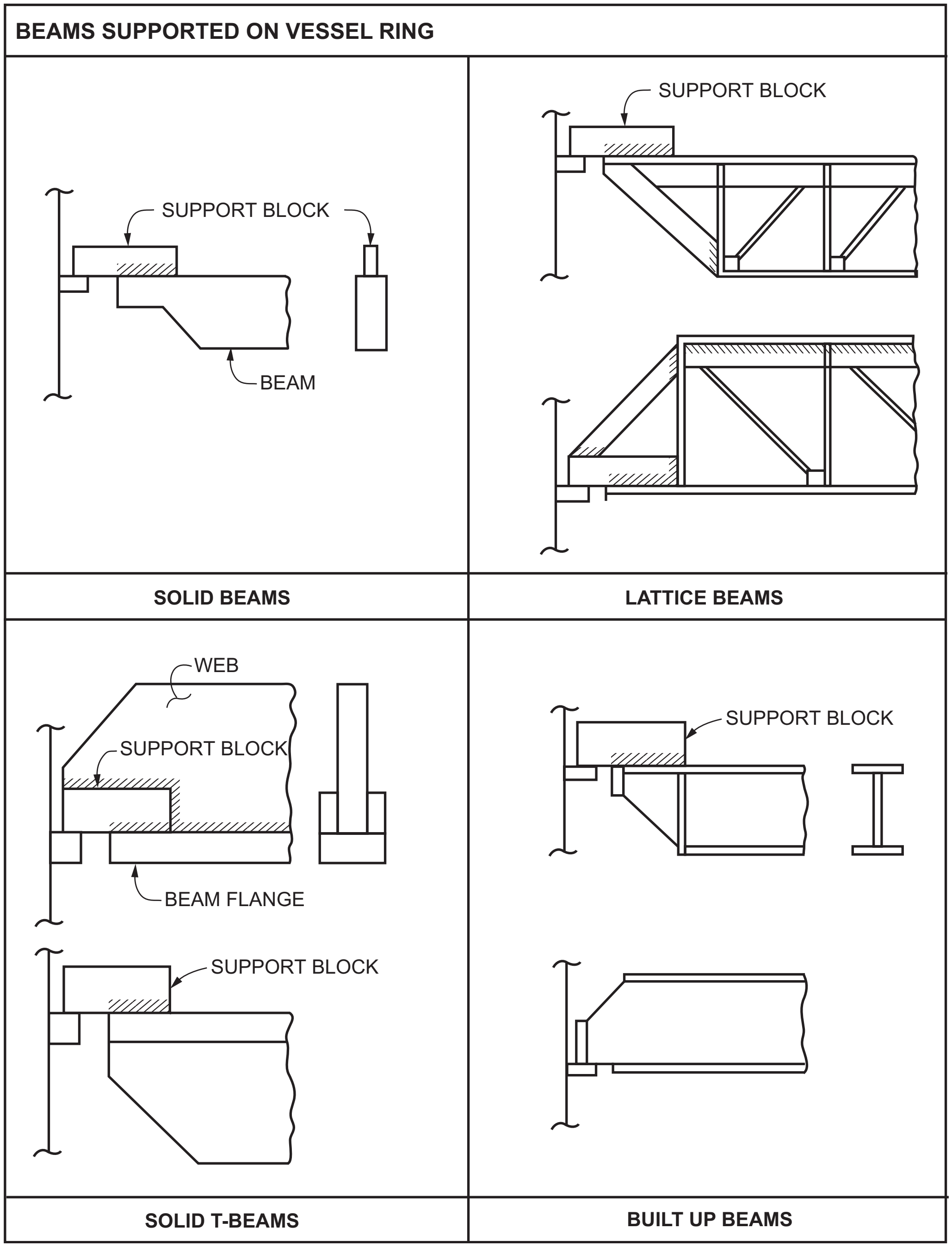
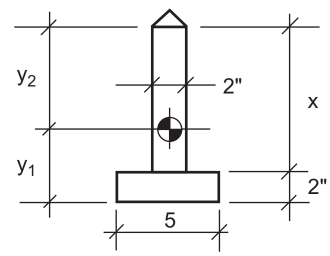

BEAMS SUPPORTED ON VESSEL RING
SUPPORT BLOCK
BEAM
SOLID BEAMS
SUPPORT BLOCK
LATTICE BEAMS
WEB
SUPPORT BLOCK
BEAM FLANGE
SUPPORT BLOCK
SOLID T-BEAMS
SUPPORT BLOCK
BUILT UP BEAMS

Table 5-2
Properties of heavy T-Beams

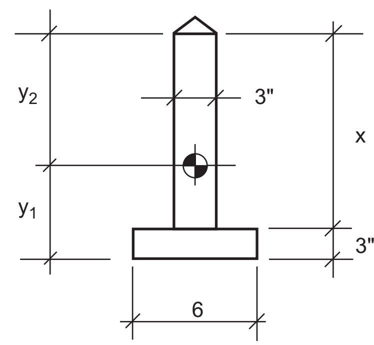
No.	X	A	y_1	y_2	I	Z_T	Z_C
TYPE 1							
1	8	26	4.07	5.93	242.5	59.6	40.9
2	12	34	5.94	8.06	637.2	107.27	79.1
3	16	42	7.86	10.14	1303	165.8	128.5
4	20	50	9.8	12.2	2305	235.2	189
5	24	58	11.75	14.24	3711	315.9	260.6
TYPE 2							
1	8	42	4.64	6.36	452.6	97.55	71.16
2	10	48	5.56	7.44	738.8	132.9	99.3
3	12	54	6.5	8.5	1120.5	172.4	131.8
4	14	60	7.45	9.55	1609.9	216.1	168.6
5	16	66	8.4	10.6	2224	264.7	209.8
6	18	72	9.375	11.625	2960	315.7	254.6
7	20	78	10.34	12.66	3844	371.8	303.6
8	21	81	10.83	13.17	4345	401	329.9
9	24	90	12.3	14.7	6094	495.4	415.6
10	26	96	13.28	15.72	7484	563.6	476.1
TYPE 3							
1	18	100	9.92	12.08	4421	445.6	365.9
2	20	108	10.88	13.12	5701	524	434.5
3	22	116	11.86	14.13	7179	605	508
4	24	124	12.83	15.16	8908	694	588
5	26	132	13.82	16.18	10856	786	671
6	28	140	14.8	17.2	13089	885	761
7	30	148	15.8	18.2	15607	989	857

FIGURES

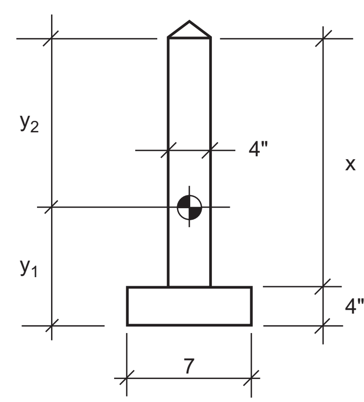
TYPE 1



TYPE 2



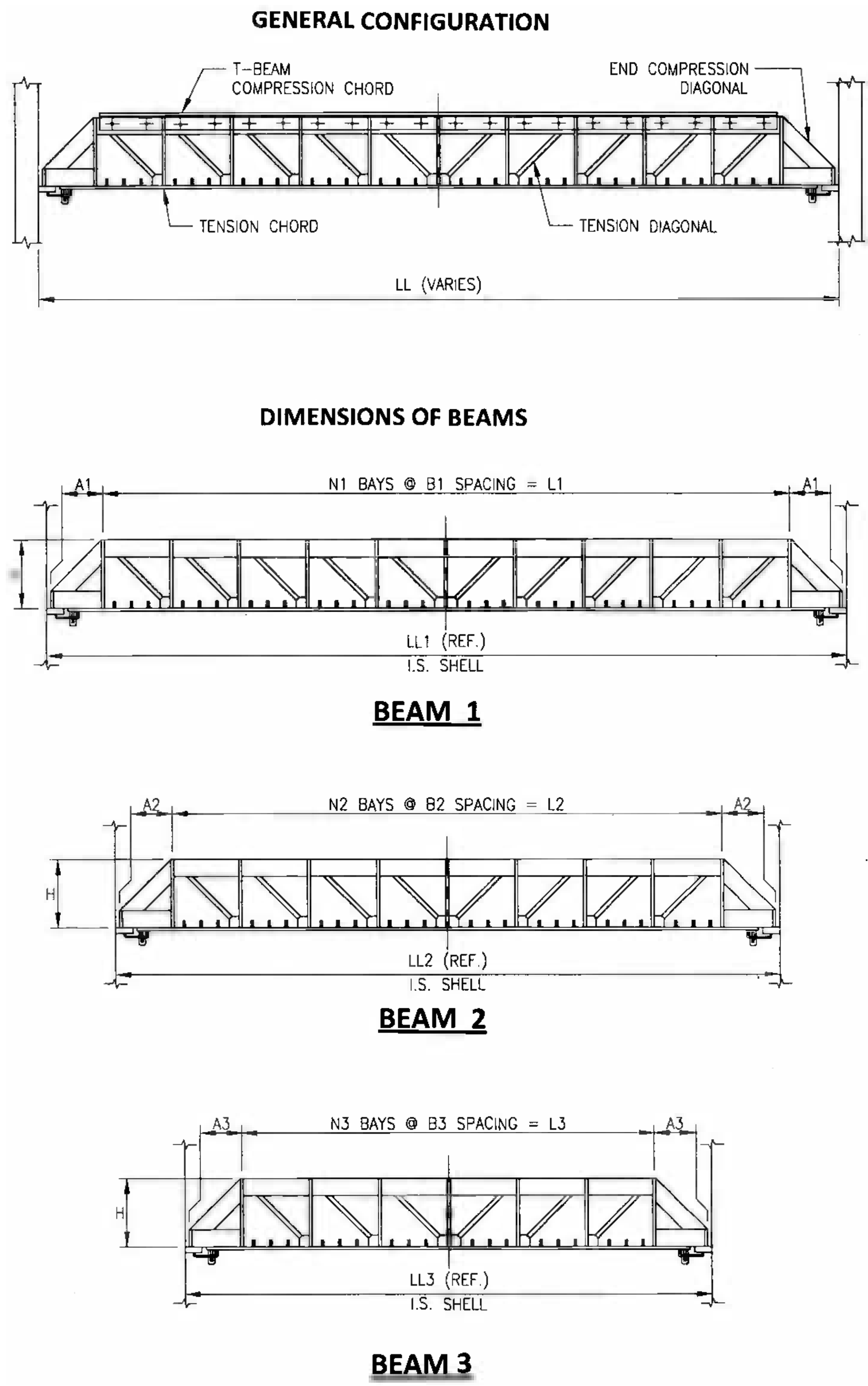
TYPE 3

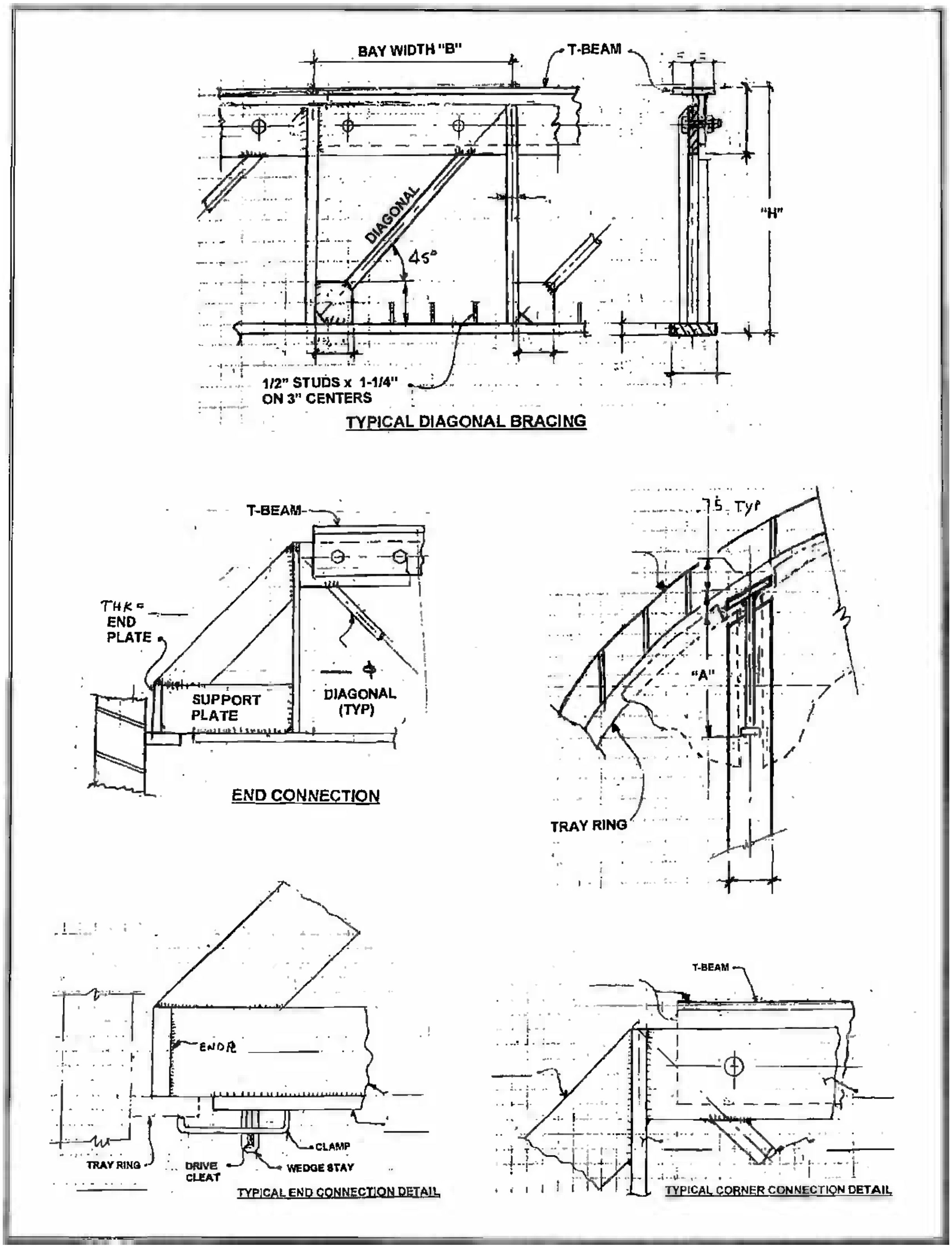


Notes:

1. Z_T = Section Modulus for tension, $= I / y_1$.
2. Z_C = Section Modulus for compression, $= I / y_2$.
3. Designer should review that beams can be inserted through existing manway size.

Procedure 5-2: Design of Lattice Beams



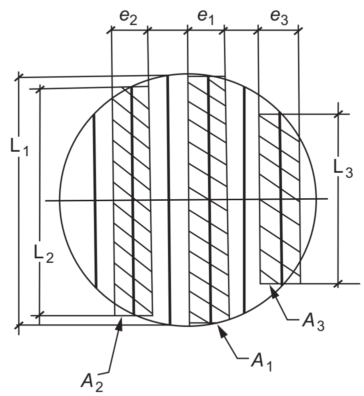

BAY WIDTH "B"
T-BEAM
DIAGONAL
45°
"H"
1/2" STUDS x 1-1/4"
ON 3" CENTERS
TYPICAL DIAGONAL BRACING
T-BEAM
THK =
END
PLATE
SUPPORT
PLATE
DIAGONAL
(TYP)
END CONNECTION
.75 Typ
"A"
TRAY RING
ENDR
TRAY RING
DRIVE
CLEAT
CLAMP
WEDGE STAY
TYPICAL END CONNECTION DETAIL
T-BEAM
TYPICAL CORNER CONNECTION DETAIL

DESIGN LOADS FOR LATTICE BEAMS				
DATA			LOADINGS	
ITEM	SYMBOL	VALUE	ITEM	VALUE
Material, beams and grating			Dead load, DL	
Material, vessel shell			a. Weight of beams	
Quantity of beams	N		b. Weight of grating	
Vessel ID	D		c. Weight of tray	
Design temperature	DT		d. Weight of screen	
Differential pressure	ΔP		e. Weight of packing/catalyst	
Bed depth	d_C		Product load, PL	
Liquid/contents unit weight	w_c		a. Weight of liquid on tray	
Corrosion allowance	C_a		b. Weight of liquid in bed	
Specific gravity	S_g		c. Weight of liquid above bed	
Liquid holdup (%)			d. Weight of solids	
Free area in bed (%)	P_C		Live load, LL	
Packing/catalyst unit weight	w_P			
Volume of packing/catalyst	V_P		Total load, F, Lbs	
Packing/catalyst total wt	W_C		$F = DL + PL + LL$	
Weight of entrained liquid	W_e			
Weight of grating	W_g		Total cross sectional area, A_c	
Weight of beams	W_b		$A_c = \pi r^2$	
Miscellaneous				
MATERIAL PROPERTIES & ALLOWABLE STRESS			Uniform load, p, PSI	
Shell at design temperature	S		$p = F / A_c$	
	E			
	F_y			
Beams at design temperature	E			
	F_y			
Bending = $.66 F_y$	F_b			
Compression, dependent on KL/r ratio	F_C			
Shear = $.4 F_y$	F_S			
Bearing = $.9 F_y$	F_{br}			

DESIGN OF LATTICE BEAMS

DIMENSIONS

(6) Beams shown for example Only!



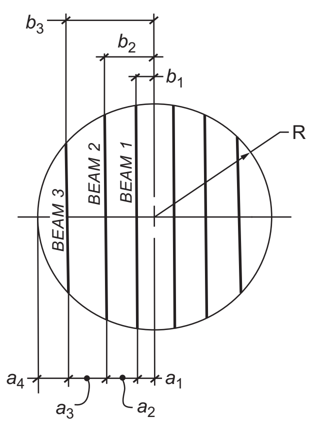
1.0 AREA OF LOADING

BEAM	L_n	e_n	A_n
1			
2			
3			
4			

2.0 LOADS ON BEAMS

BEAM	F_n	R_n	w_n
1			
2			
3			
4			

DIMENSIONS



3.0 MOMENT & FORCES

BEAM	M_n	h_n	T_n or C_n
1			
2			
3			
4			

4.0 BEAM STRESSES

BEAM	f_T	f_C	
1			
2			
3			
4			

5.0 ALLOWABLE STRESS, COMPRESSION

BEAM	L_{cn}	$K\,L_{cn} / r$	F_c
1			
2			
3			
4			

EQUATIONS

$e_1 = a_1 + .5\,a_2$

$e_2 = .5\,a_2 + .5\,a_3$

$e_3 = .5\,a_3 + .5\,a_4$

$e_4 = .5\,a_4 + .5\,a_5$

$L_n = 2\,[R^2 - b_n^2]^{1/2}$

$f_T = T_n / A_T$

$f_C = C_n / A_c$

$$C_C = \sqrt{\frac{2\pi^2 E}{F_y}}$$

$$F_C = \frac{\left[1 - \frac{(KL/r)^2}{2\,C_C^2}\right] F_y}{5\backslash 3 + \frac{3(KL/r)}{8\,C_C} - \frac{(KL/r)^3}{8\,C_C^3}}$$

$A_n = L_n\ e_n$

$F_n = A_n\ p$

$w_n = F_n / L_n$

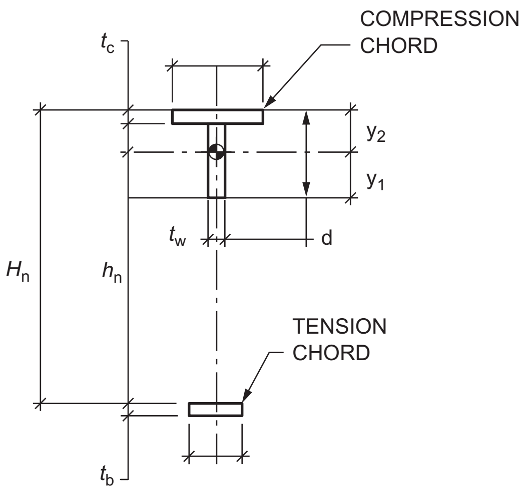
$R_n = F_n / 2$

$M_n = w_n\ L_n^2 / 8$

$T_n = C_n = M_n / h_n$

6.0. Properties of Beam

- Properties of compression chord;



DIMENSIONS OF LATTICE BEAM

Part	A	y	A y	A y^2	I
1					
2					
Σ					

$$y_1 = \Sigma A y / \Sigma A$$
$$y_2 = d - y_1$$
$$I = \Sigma A y^2 + \Sigma I - y_1 \Sigma Ay$$
$$Z = I/y_1 \text{ or } Z_r = M_n/F_n$$
$$r = (I/\Sigma A)^{1/2}$$

7.0. Diagonals

- All diagonals are in tension
- Maximum load in diagonals, f_d

Beam	F_n	f_n	d_n
1			
2			
3			
4			

- Axial load in diagonal, tension, f_n

$$f_n = F_n/2 \sin\theta$$

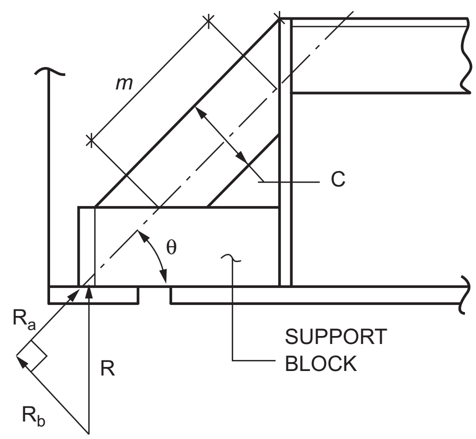
If $\theta = 45^o$, $f_n = .707\ F_n$

- Diameter of diagonal, d_{min}

$$d_{min} = [(4\ f_n)/(\pi\ F_T)]^{1/2}$$

8.0. Beam End Diagonal

- Determine Loads;



DIMENSIONS OF END CONNECTION

Beam	R_n	R_a	R_b
1			
2			
3			
4			

$$R_n = \text{Reaction, Lbs}$$
$$R_a = R_n \sin\theta$$
$$R_b = R_n \cos\theta$$

- Determine minimum gusset thickness, t_g

$$t_g = (6\ R_b\ m)/F_b\ C^2$$

Use ______________

- Determine properties;

Beam	m	Z	M
1			
2			
3			
4			

- Section modulus, Z

  $Z = \left(t_g\ C^2\right)/6$
- Moment, M

  $M = R_b\ m$
- Stress in end beam diagonal

Beam	f_a	f_b	F_C
1			
2			
3			
4			

$f_a = R_a/t_g\ C \le F_C$

$f_b = M/Z \le F_b$

$f_a/F_C + f_b/F_b \le 1$

F_C is based on L/r; L/r = m/r

$r = C/12^{1/2} = .289C$

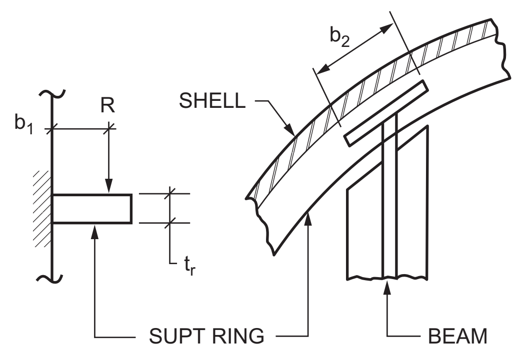
9.0. Stability Check

- The allowable buckling stress for the compression chord, F_{CB}, is dependent on the K L /r ratio (from AISC).

$K\ L_C / r$	Type	F_{CB}
<50	Short	F_y
50 TO 200	Intermediate	Interpolate
>200	Long	Not Recommended

- For $K\ L_C/r$ ratios between 50 and 200, the following apply;
  - If $K\ L_C/r < 4.71\ (E/F_y)^{1/2}$
    Or $F_e \ge .44\ F_y$
    Then $F_{CB} = (.658^{Fy/Fe})\ F_y$
  - If $K\ L_C / r > 4.71\ (E/F_y)^{1/2}$
    Or $F_e < .44\ F_y$
    Then $F_{CB} = .877\ F_e$
  - Where $F_e = (\pi^2\ E\ I) / (K\ L_c)^2$

10.0. Ring Analysis



DIMENSIONS AND LOADS - SUPPORT RING

- Uniform load in support ring, w_S

  $w_S = R/b_2$
- Moment, M_S

  $M_S = w_S\ b_1$
- Bending stress, f_b

  $f_b = \left(6\ M\right)/t_r{}^2 \le F_b$

11.0. Notes

1. The allowable compressive stress in the compression chord should be the lesser of F_C or F_{CB}.
2. See procedure "Design of Internal Support Beds" for terms and definitions not shown.
3. The compression chord can be made without a T-section, with a bolted on T-section or as a built-up T-beam. This is entirely dependent on rigidity of the vertical member. A bolted on T-section is utilized where the assembly of tray plates or grating is difficult with the T-section in place.

Procedure 5-3: Shell Stresses due to Loadings at Support Beam Locations

Notation

A = Bearing area, in^2
D = Flexural rigidity, Lbs-in
E = Modulus of elasticity, PSI
F_y = Minimum specified yield strength at design temperature, PSI
f_{br} = Bearing stress, PSI
F_b = Allowable bending stress in shell, PSI
J_n = Effective width of shell acting at support point, in
M_{Ln} = Longitudinal moment at any point X or Y distance from point of applied load, in-Lbs
M_{rn} = Moment in ring due to uniform load, w, in-Lbs
M_n = Longitudinal moment in shell due to axial load, P, in-Lbs/in
N_n = Minimum bearing length, in
P_n = Uniform compressive load, Lbs/in
R_n = Reaction, Lbs
t_r = Thickness of support ring required, in
w_n = Uniform load in ring, Lbs/in
X = Distance from applied load in axial (longitudinal) direction, in
Y = Distance from applied load in circumferential direction, in
y = Circumferential stress factor ratio
λ = Damping factor
σ_{XM} = Longitudinal membrane stress, PSI
$\sigma_{\phi M}$ = Circumferential membrane stress, PSI
σ_{XB} = Longitudinal bending stress, PSI
$\sigma_{\phi B}$ = Circumferential bending stress, PSI
ν = Poisson's ratio, .3 for steel

Equations

- Angle ϕ_n;

  $\sin \phi_n = m_n/R_i$ therefore $\phi_n =$
- Angle α_n;

  $\alpha_n = 90 - \phi_n$
- Minimum bearing length, N_n;

  $N_n = (R_n \sin \alpha_n)/[(.75\ F_y)\ K_n] < g$
- Effective width of shell, J_n;

  $J_n = [(K_n/\sin \alpha_n)] + 2\ e$
- Distance from centerline of shell to point of applied load, a;

  $a = .5\ t + e$
- Uniform compressive load, P_n, Lbs/in:

  $P_n = R_n/J_n$
- Longitudinal moment in shell due to axial load, P, in-Lbs

  $M_n = P_n\ a$
- Flexural rigidity, D:

  $D = (E\ t^3)/[12(1-\nu^2)] = 0.0916\ E\ t^3$
- Damping factor, λ;

  $\lambda = [3(1-\nu^2)/(R_m{}^2\ t^2)]^{1/4}$

  $\lambda = 1.285(R_m\ t)^{1/2}$
- Longitudinal moment at any point X distance from point of applied load, M_{Ln}, in-Lbs/in

  $M_{Ln} = M_n/2[e^{-\lambda X} \cos^{\lambda X}]$

  Note: At X = 0 and Y = 0, $M_{Ln} = M_n/2$
- Longitudinal membrane stress, σ_{XM} PSI

  $\sigma_{XM} = \pm P_n/t$
- Circumferential membrane stress, $\sigma_{\phi M}$ PSI

  $\sigma_{\phi M} = (y\ E)/R_m$
- Longitudinal bending stress, σ_{XB}, PSI

  $\sigma_{XB} = \pm(6\ M_n)/t^2$
- Circumferential bending stress, $\sigma_{\phi B}$, PSI

  $\sigma_{\phi B} = \pm(6\ \nu\ M_n)/t^2$

  $\sigma_{\phi B} = \pm(1.8\ M_n)/t^2$

  $\sigma_{\phi B} = \pm .3\sigma_{XB}$
- Uniform load in ring, w_n, Lbs/in

  $w_n = R_n\ (\cos \phi_n/K_n)$
- Moment in ring due to uniform load, w, M_{rn}, in-Lbs/in

  $M_{rn} = w_n\ e$
- Thickness of support ring required, t_r, in

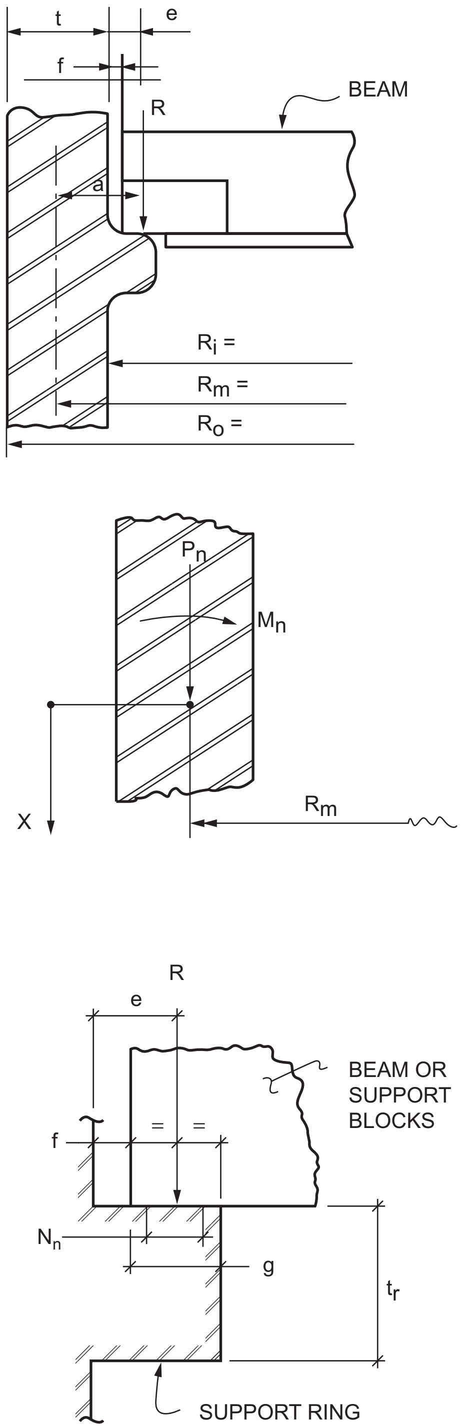
  $t_r = [(6\ M_{rn})/F_b]^{1/2}$

  Use largest value of M_{rn}
- Bearing area, A_b, in^2;

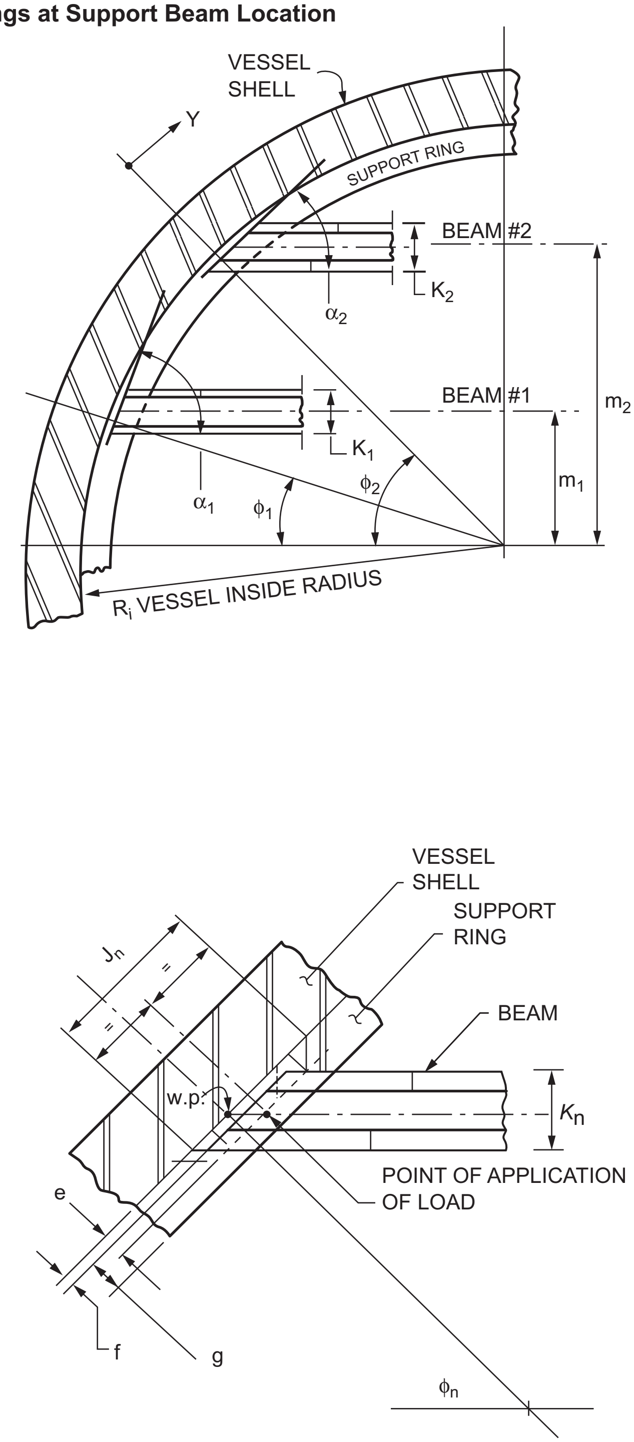
  $A_b = g\ K_n$
- Bearing stress, f_{br}, PSI

  $f_{br} = R_n/A_b$

Shell Stresses due to Loadings at Support Beam Location



CROSS SECTION OF VESSEL SHELL



PLAN VIEW OF VESSEL QUADRANT

LOADS AND STRESSES IN VESSEL SHELL AT SUPPORT POINT LOCATIONS

1.0 FIND ANGLES ϕ AND α

BEAM	m_n	ϕ_n	α_n
1			
2			
3			
4			

2.0 DIMENSIONS & LOADS

BEAM	R_n	N_n	J_n
1			
2			
3			
4			

3.0 LOADS & FACTORS

BEAM	P_n	M_n	y_n
1			
2			
3			
4			

4.0 MOMENTS & MEMBRANE STRESSES

BEAM	M_{Ln}	σ_{XM}	$\sigma_{\phi M}$
1			
2			
3			
4			

5.0 BENDING STRESSES

BEAM	σ_{XB}	$\sigma_{\phi B}$	
1			
2			
3			
4			

6.0 MAX UNIFORM LOAD IN RING

BEAM	R_n	w_n	M_{rn}
1			
2			
3			
4			

7.0 BEARING STRESS

BEAM	K_n	A_{bn}	f_{br}
1			
2			
3			
4			

Notes:

1. This procedure establishes one method for determining the shell stresses resulting from beams supported directly on a support ring, as opposed to supported off beam seats or clips.
2. This procedure assumes the beam is supported off "support blocks".

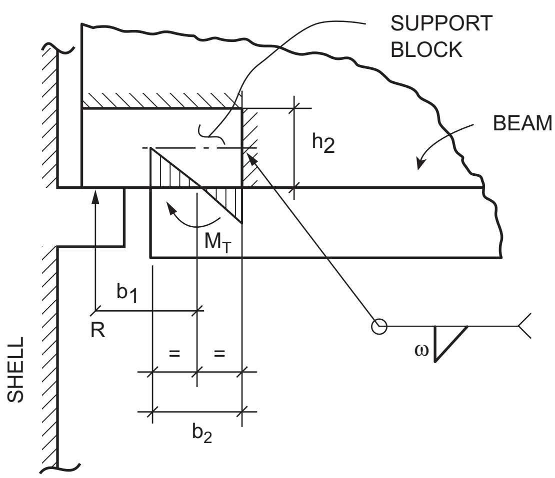
Procedure 5-4: Design of Support Blocks

Often times, when beams are supported on a support ring, the beam is supported by, or reinforced with a support block(s). This procedure shows how to design the support blocks and attachment welds for the various load cases. There are two types of support blocks used.

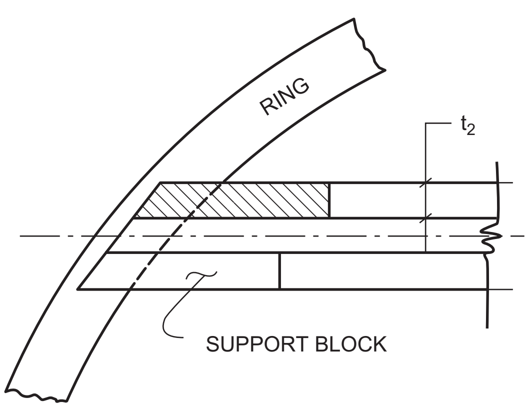
Type 1: Support blocks are welded integral with the web of the beam to expand the bearing area as well as reinforce the web in the area of local load. This type is used primarily with T-Type beams but could also be used with a built-up I beam.

Type 2: This type is used with solid beams where the top of the beam is located level with the top of the support ring to create a uniform support plane. The support block, also called a support plate, is welded to the top of the beam and cantilevers beyond the end of the beam to support the load.

Type 1: Support Blocks Welded Integral with Beam



Type 1-Loads and Dimensions of Support Block



Plan View Beam with Support Blocks

- Moment, M_T

$$M_T = .5\ R\ b_1$$

- Section modulus required, Z_r

$$Z_r = M_T/F_b$$

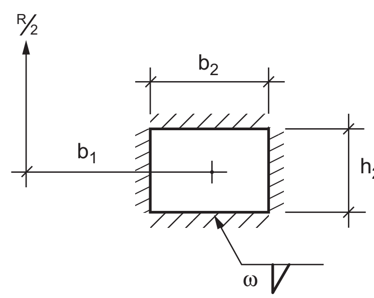
- Actual section modulus, Z

$$Z = \left(t_2\ h_2^2\right)/6$$

- Height of block required, h_r

$$h_r = [(6\ Z_r)/t_2]^{1/2}$$

- Maximum force on weld (treated as a line):



Loads on Welds of Support Blocks

- Polar moment of inertia of weld group, J_w

$$J_w = (b_2 + h_2)^3/6$$

- Shear on weld due to twisting moment in horizontal direction, f_{th}, and vertical direction, f_{tv}

$$f_{th} = M_T\ C/J_w = (M_T\ (0.5\ b_2))/J_w$$

$$f_{tv} = M_T\ C/J_w = M_T\ (0.5\ h_2)/J_w$$

- Area of weld group, A_w

$$A_w = 2b_2 + 2h_2$$

- Vertical shear on weld

$$f_{sv} = 0.5R/A_w$$

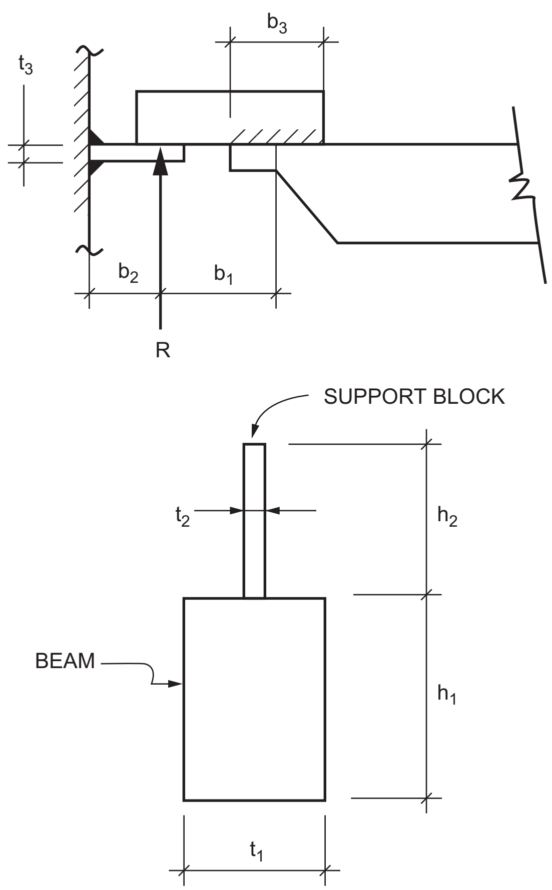
- Resultant shear force on weld, τ_r

$$f_r = \left(f_{th}^2 + (f_{tv} + f_{sv})^2\right)^{1/2}$$

- Size of fillet weld required, w

$$w = f_r/(0.707F_s)$$

Type 2: Support Block Welded to Top of Beam



Type 2 - Loads and Dimensions

- Moment in support block, M_1

$$M_1 = R\ b_1$$

- Required section modulus, Z_r

$$Z_r = M_1/F_b$$

- Actual section modulus, Z

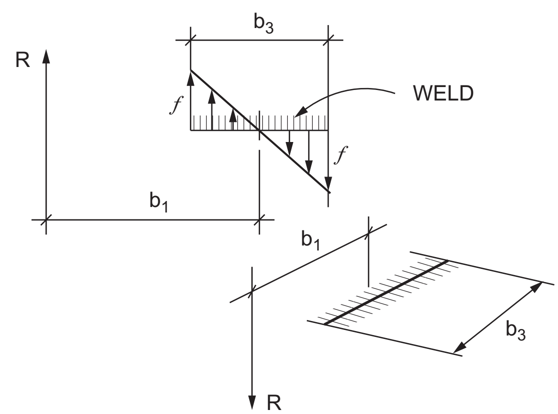
$$Z = \left(t_2\ h_2^2\right)/6$$

- Section modulus of weld group, Z_w

$$Z_w = b_3^2/3$$

- Maximum force on weld group, f

$$f = \left(R\ b_1\right)/\left(b_3^2/3\right)$$



Loads on Welds

- Size of fillet weld required, w

$$w = f/(0.707\ F_s)$$

Notes

1. An end plate can be welded to the end of the support block to stiffen the end of the plate and distribute the load on the ring.
2. See Procedure "Design of Internal Support Beds" for terms and definitions not shown.

Procedure 5-5: Hub Rings used for Bed Supports

The traditional way of supporting beds inside vessels has been to use straight beams. An alternative approach to the conventional straight beam design is to utilize a circular beam, commonly referred to as a "hub ring", supported by equally spaced radial beams. The radial beams are like the spokes of a wheel.

While this application may be new, the technology or method has been in use for many years. This type of support structure has long been recognized as a very efficient support structure. Efficiency in this case is the weight of the structure versus the weight supported. Hub rings can support up to 150 times their weight, while straight beams are in the range of 100.

The loadings demanded by industry have gone up dramatically over the years. As these loadings have gone up, the size of straight support beams has also increased dramatically. The design of the conventional straight beams is in some cases approaching physical limitations due to manway sizes required to insert or remove the beams.

As design loadings have increased, the spacing of beams has decreased. The height, width and thus weight of beams has increased, as well as the cost. This in turn has led to increasing the manway size to accommodate deeper beams while the access between the beams has become smaller and smaller. The hub ring can provide an innovative solution to this problem.

This type of support structure is ideal for pressure vessels because of the convenience of putting a round structure in a round vessel. The hub ring is comprised of two circular rings separated by the radial beams. The natural couple created by the loadings puts the top ring into compression and the lower ring in tension. The steel works great in tension and the circular beam is equally efficient at resisting the compressive loads. The advantage is achieved by utilizing the best properties of the material as well as the shape of the section. It is merely an application of the old adage, "form follows function".

In the hub ring design the radial beams are typically one quarter of the vessel diameter in length. Since the moment formula for a simply supported, uniformly loaded beam is $wl^2/8$, the beams are proportional, not to the length but to the square of the length. By comparison, straight beams have to span practically the whole diameter of the vessel, while radial beams only span ¼ of the diameter.

Another advantage, besides efficiency, weight, and cost, is safety. Not safety in design, but safety during maintenance periods. The design of straight beams drives the designer to increase the quantity of beams with a corresponding decrease in beam spacing. This reduces the loads on any individual beam, but in some cases narrows the passageway between the flanges of the beams to minimal levels. During plant shutdowns and maintenance periods, this reduction in access space can cost valuable time and create a hazard for maintenance personnel. When you consider the array of workers, ladders, lights, power cords, vacuum lines and loading socks that have to pass between these supports, this restriction is more than a major inconvenience. Since the hub ring is open in the center it can easily provide large access space.

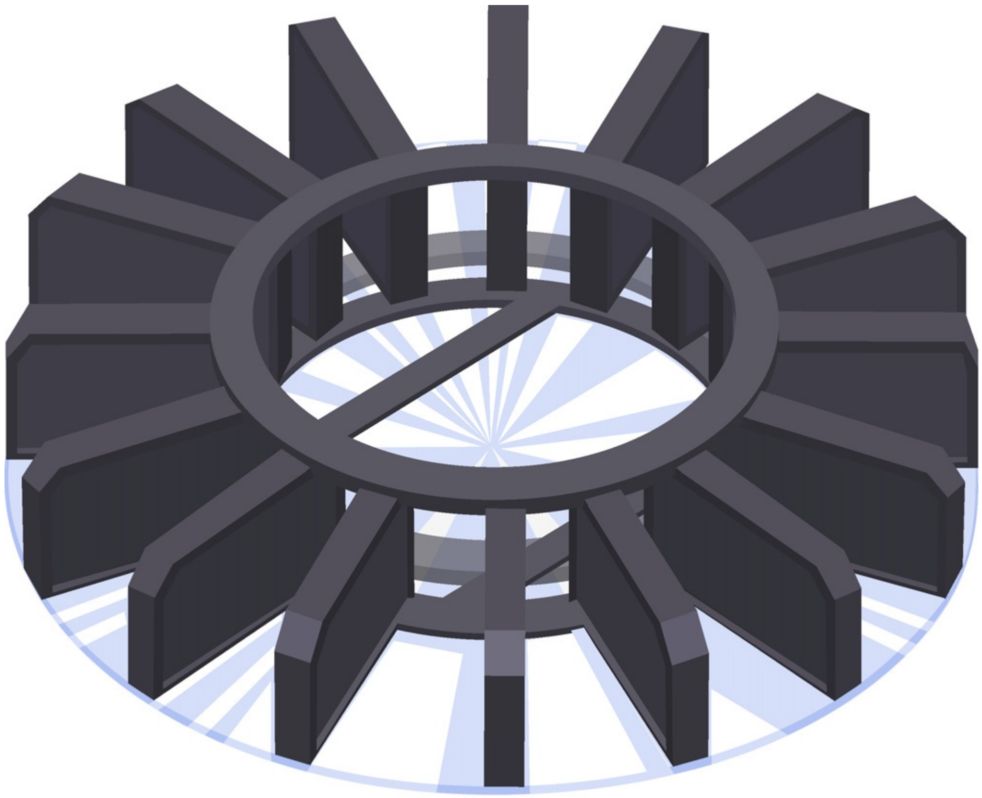
For vessels larger than about 10 feet in diameter, a cross beam, or multiple cross beams, may be required inside the hub ring to support tray decks or grating panels. Once again, these cross beams can be spaced such that more than adequate space is allowed for personnel access.

The ideal proportion for the centerline diameter of the hub rings is $D_m = 0.5$ D.

The design is based on "N" number of equally spaced, radial loads. The upper ring is in compression. The lower ring is in tension.

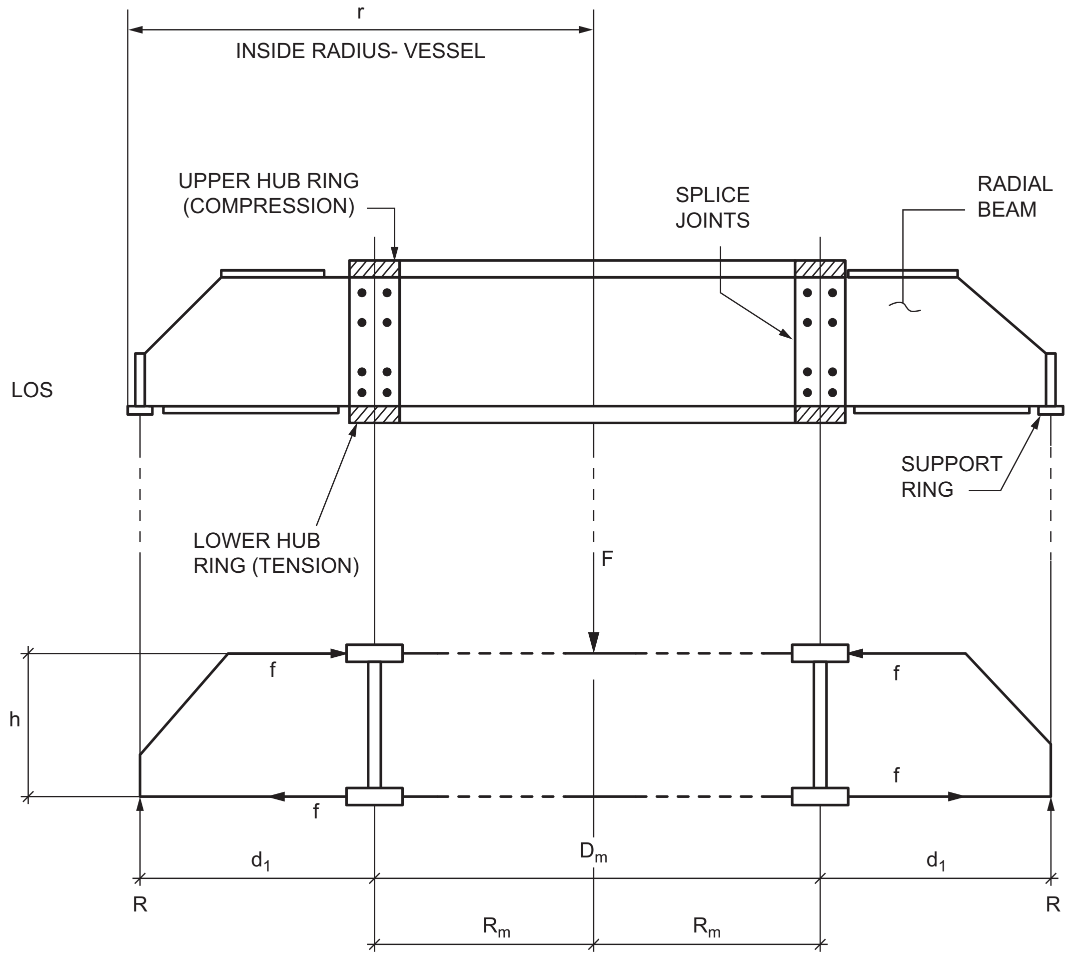
The entire assembly can be welded up and inserted in the vessel prior to installation of the top head or closure seam. Or, the entire thing can be made in pieces to pass through a manway. When made in pieces, the typical splice point is at the radial beams. Thus the quantity of spokes selected is typically a function of what will fit through a given manway size, rather than what loadings the components can handle.

Hub rings really have no limitations. They have been designed to support loads in excess of 1,000,000 pounds and diameters up to 60 feet, but not simultaneously, of course.

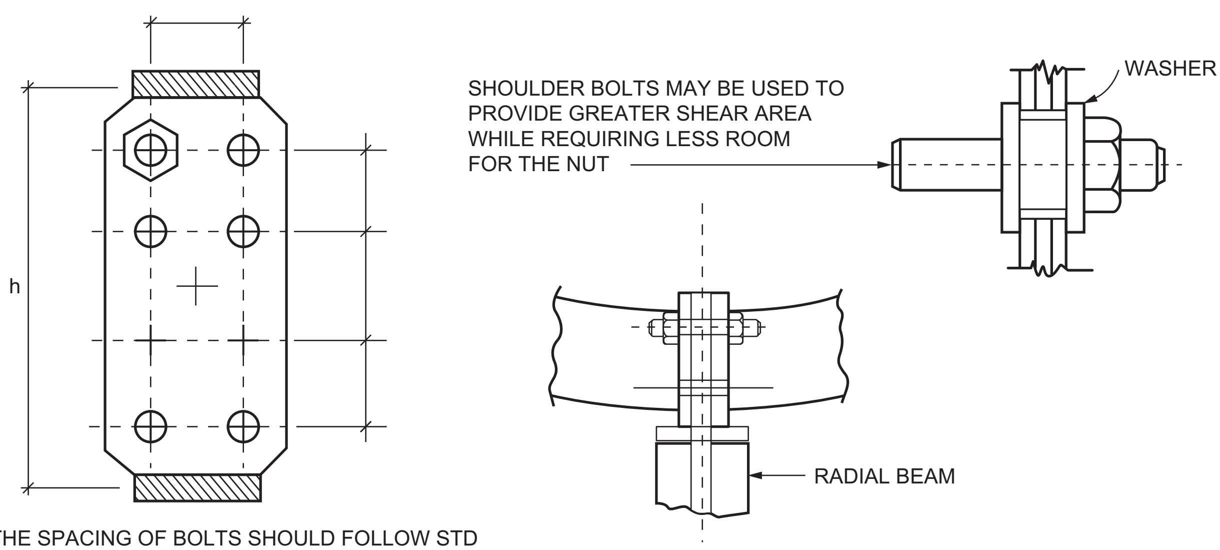


HUB RING WITH RADIAL BEAMS

GENERAL LOADS AND DIMENSIONS

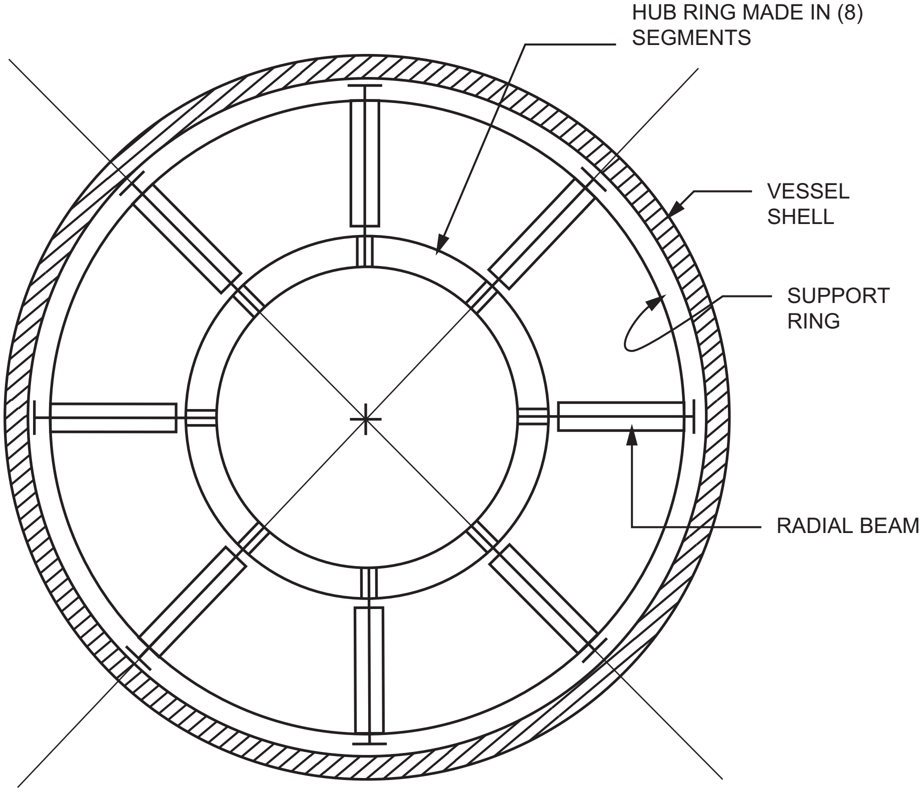


DETAILS OF BOLTED SPLICE CONNECTION

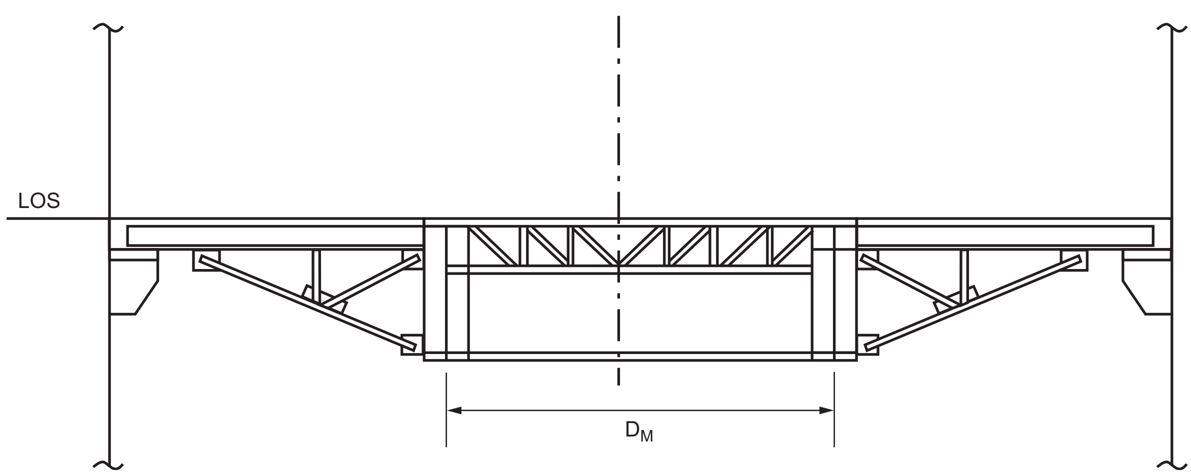


THE SPACING OF BOLTS SHOULD FOLLOW STD
BOLTING CONVENTIONS

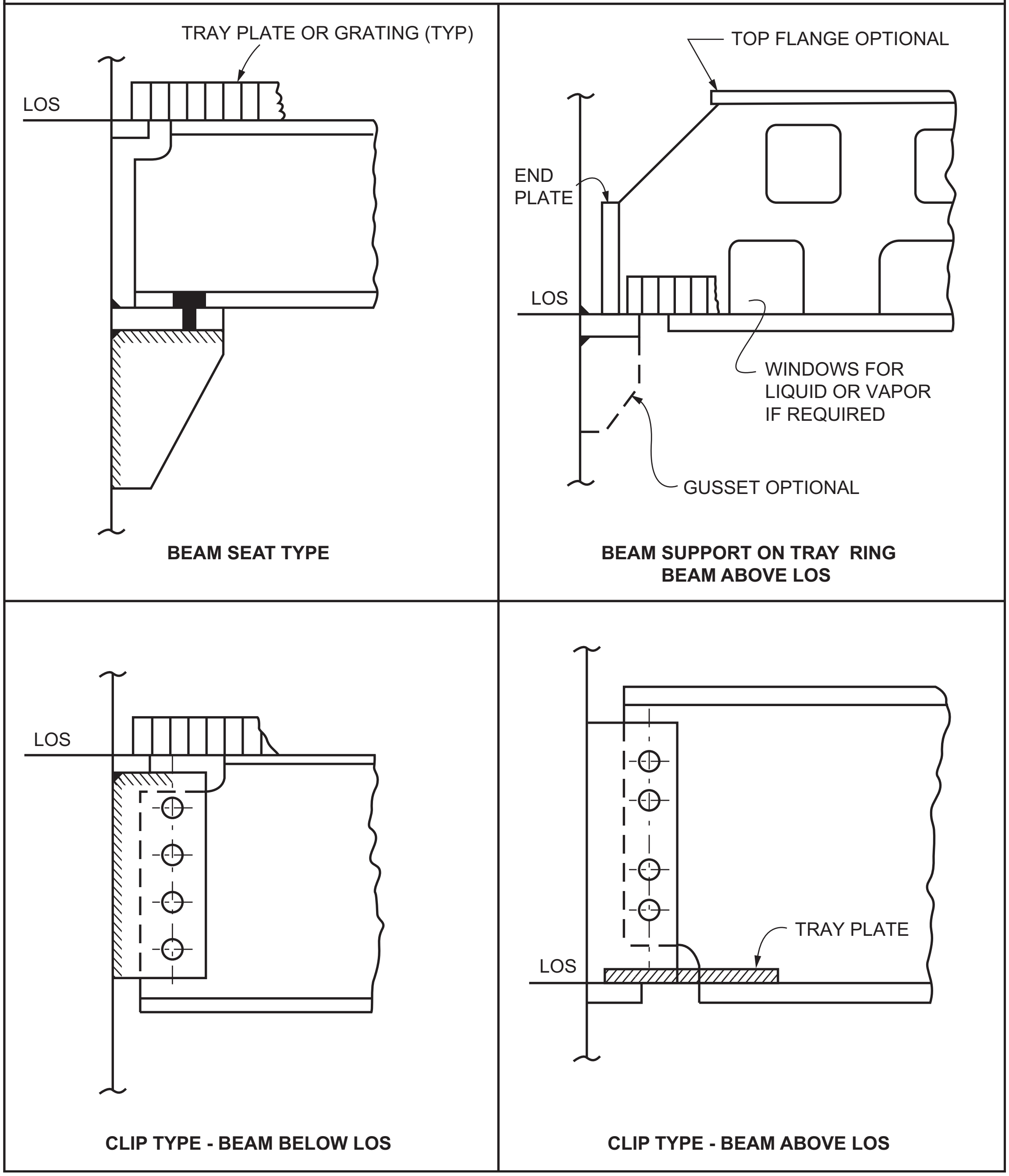
TYPICAL PLAN VIEW
(8) RADIAL BEAMS SHOWN ARE EXAMPLE ONLY



HUB RING SUPPORT FOR LARGE DIAMETER COLUMN
RADIAL BEAMS ARE LATTICE TYPE



SUPPORTS FOR RADIAL BEAMS



BEAM SEAT TYPE

BEAM SUPPORT ON TRAY RING BEAM ABOVE LOS

CLIP TYPE - BEAM BELOW LOS

CLIP TYPE - BEAM ABOVE LOS

Procedure 5-6: Design of Pipe Coils for Heat Transfer [1-9]

This procedure is specifically for helical pipe coils in vessels and tanks. Other designs are shown for illustrative purposes only. Helical coils are generally used where large areas for rapid heating or cooling are required. Heating coils are generally placed low in the tank; cooling coils are placed high or uniformly distributed through the vertical height. Here are some advantages of helical pipe coils.

1. Lower cost than a separate outside heat exchanger.
2. Higher pressures in coils.
3. Fluids circulate at higher velocities.
4. Higher heat transfer coefficients.
5. Conservation of plot space in contrast with a separate heat exchanger.

Manufacture

Helical pipe coils can be manufactured by various means:

1. Rolled as a single coil on pyramid (three-roll) rolling machine. This method is limited in the pitch that can be produced. Sizes to 8 inches NPS have been accommodated, but 3 inches and less is typical. The coil is welded into a single length prior to rolling.
2. Rolled as pieces on a three-roll, pyramid rolling machine and then assembled with in-place butt welds. The welds are more difficult, and a trimming allowance must be left on each end to remove the straight section.
3. Coils can also be rolled on a steel cylinder that is used as a mandrel. The rolling is done with some type of turning device or lathe. The coil is welded into a single length prior to coiling. The pitch is marked on the cylinder to act as a guide for those doing the forming.
4. The most expensive method is to roll the pipe/tubing on a grooved mandrel. This is utilized for very small Dc-to-d ratios, usually followed by some form of heat treatment while still on the mandrel. Grooved mandrels create a very high-tolerance product and help to prevent flattening to some extent.

Coils are often rolled under hydro pressures as high as 85% of yield to prevent excessive ovalling of the pipe or tube. To accomplish this, the hydrotest pump is put on wheels and pulled along during the rolling process. End caps are welded on the pipe to maintain the pressure during rolling.

Stainless steel coils may require solution annealing after forming to prevent "springback" and alleviate high residual stresses. Solution heat treatment can be performed in a fixture or with the grooved mandrel to ensure dimensional stability.

Springback is an issue with all coils and is dependent on the type of material and geometry. This springback allowance is the responsibility of the shop doing the work. Some coils may need to be adjusted to the right diameter by subsequent rolling after the initial forming.

The straight length of pipe is "dogged" to the mandrel prior to the start of the rolling to hold the coil down to the mandrel. Occasionally it may be welded rather than dogged.

Applications for grooved mandrel are very expensive due to the cost of the machining of the mandrel. Mandrels that are solution heat treated with the coil are typically good for only one or two heat treatments due to the severe quench. Thus the cost of the mandrel must be included in the cost of the coil.

Design

There are two distinct aspects of the design of pipe coils for heat transfer. There is the thermal design and the physical design. The thermal design falls into three parts:

1. Determine the proper design basis.
2. Calculating the required heat load.
3. Computing the required coil area.

Physical design includes the following:

1. Selecting a pipe diameter.
2. Computing the length.
3. Determine the type of coil.
4. Location in the tank or vessel.
5. Detailed layout.

To determine the design basis, the following data must be determined:

1. Vessel/tank diameter.
2. Vessel/tank height.

3. Insulated or uninsulated.
4. Indoor or outdoor.
5. Open top or closed top.
6. Maximum depth of liquid.
7. Time required to heat/cool.
8. Agitated or nonagitated.
9. Type of operation.

The type of operation is characterized in the following cases:

1. Batch operation: heating.
2. Batch operation: cooling.
3. Continuous operation: heating.
4. Continuous operation: cooling.

Coils inside pressure vessels may be subjected to the internal pressure of the vessel acting as an external pressure on the coil. In addition, steam coils should be designed for full vacuum or the worst combination of external loads as well as the internal pressure condition. The coil must either be designed for the vessel hydrotest, externally, or be pressurized during the test to prevent collapse.

Pressure Drop

It is important that pressure drop be considered in designing a pipe coil. This will establish the practical limits on the length of pipe for any given pipe size. Large pressure drops may mean the coil is not capable of transmitting the required quantity of liquid at the available pressure. In addition, the fluid velocities inside the coil should be kept as high as possible to reduce film buildup.

There are no set rules or parameters for maximum allowable pressure drop. Rather, an acceptable pressure drop is related to the velocity required to effect the heat transfer. For liquids a minimum velocity of 1-3 feet per second should be considered. For gases "rho-V squared" should be maintained around 4000.

Pressure drop in helical coils is dependent on whether the flow is laminar or turbulent. Typically flows are laminar at low fluid velocities and turbulent at high fluid velocities. In curved pipes and coils a secondary circulation takes place called the "*double eddy*" or *Dean Effect*. While this circulation increases the friction loss, it also tends to stabilize laminar flow, thus increasing the "critical" Reynolds number.

In general, flows are laminar at Reynolds numbers less than 2000 and turbulent when Reynolds numbers are greater than 4000. At Reynolds numbers between 2000 and 4000, intermittent conditions exist that are called the *critical zone*.

For steam flow, the pressure drop will be high near the inlet and decrease approximately as the square of the velocity. From this relationship, combined with the effects of increased specific volume of the steam due to pressure drop, it can be shown that the average velocity of the steam in the coil is three-fourths of the maximum inlet velocity. For the purposes of calculating pressure drop, this ratio may be used to determine the average quantity of steam flowing within the coil.

Heat Transfer Coefficient, U

The heat transfer coefficient, U, is dependent on the following variables:

1. Thermal conductivity of metal, medium, and product.
2. Thickness of metal in pipe wall.
3. Fluid velocity.
4. Specific heat.
5. Density and viscosity.
6. Fouling factor (oxidation, scaling).
7. Temperature differences (driving force).
8. Trapped gases in liquid flow.
9. Type of flow regime (laminar versus turbulent, turbulent being better).

Notes

All of the following apply specifically to helical coils.

1. Overdesign rather than underdesign.
2. The recommended ratio of vessel diameter to pipe diameter should be about 30. However, it has been found that 2 inch pipe is an ideal size for many applications. Pipe sizes of 6 inches and 8 inches have been used.
3. Helical coils are concentric with the vessel axis.
4. Two or more coils may be used, with the recommended distance between the coils of two pipe diameters.
5. Seamless pipe is preferred. Schedule 80 pipe is preferred.

6. Limit maximum pitch to five pipe diameters, with 2 to 2½ recommended. Physical limits should be set between 4 inches minimum and 24 inches maximum.
7. Centerline radius of bends should be 10 times the pipe diameter minimum. (1 inch pipe = 10 inch centerline radius).
8. It is recommended for bend ratios over 5% or fiber elongation greater than 40% that the coils be heat treated after forming. The bend ratio can be computed as follows:

$$\frac{100\ t_p}{R}$$

9. Flattening due to forming should be limited to 10%. Some codes limit ovality to as little as 8%. Ovality may be computed as follows:

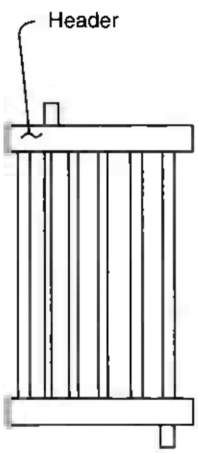
$$100\left(\frac{d_{max} - d_{min}}{d}\right)$$

10. Wall thinning occurs any time a pipe is bent. The inside of the bend gets thicker and the outside of the bend gets thinner. Typically this is not a problem because the outside of the bend that gets thinner will also experience a certain amount of work hardening that can make up for the loss of wall thickness. The tighter the bend, the greater the thinning. Anticipated wall thinning due to forming can be computed as follows:

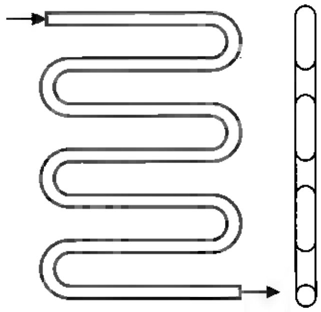
$$t_p\left(1 - \frac{R}{R + 0.5d_0}\right)$$

11. Distance between an internal coil and the side wall or bottom of the tank or vessel is a minimum of 8 inches and a maximum of 12 inches (dimension "c").
12. All coils should be evenly supported at a minimum of three places. Supports should be evenly spaced and allow for thermal expansion of the coil.
13. Coils should be sloped a minimum of ⅛ inch per foot to allow for drainage.
14. Certain flow rates in spiral coils can set up harmonic vibrations that could ultimately be destructive to the coil, supports, etc. In addition, slug flow can cause extreme coil movement. If vibration or movement becomes a problem, then either the flow rate or the coil support arrangement must be changed.
15. Limit velocity to 10 feet per second in coils.
16. The "steady-state" condition requires less coil than any other design condition.
17. If pressure drop is excessive, the coil may be split into multiple coils with manifolds or separate inlets or outlets.

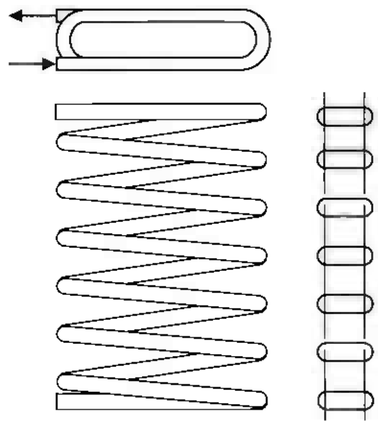
Types of Coils



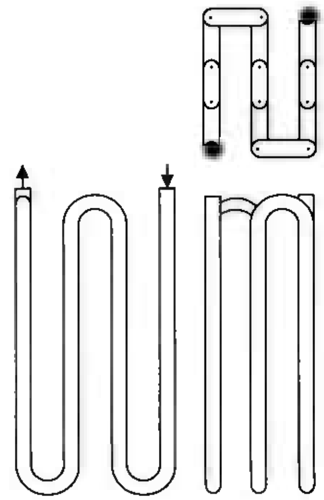
Harp
(flat)



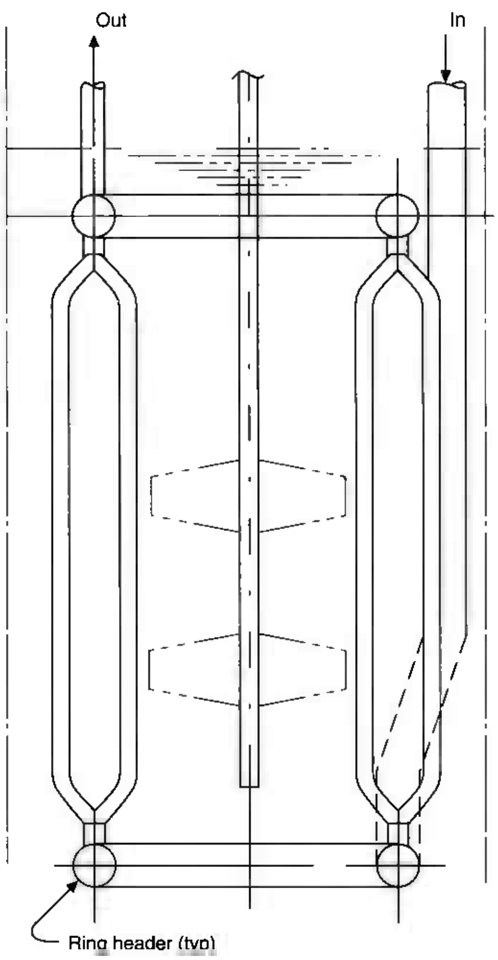
Flat
(serpentine)

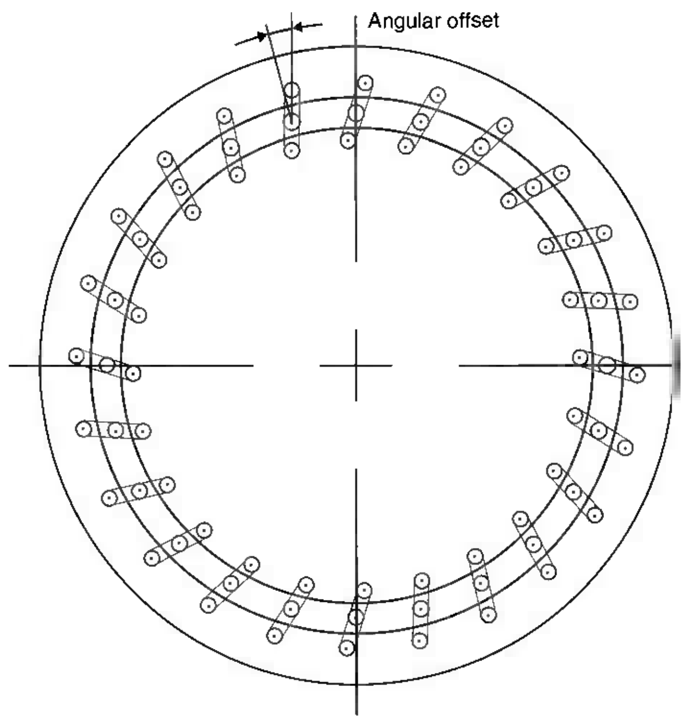


Oval



Box

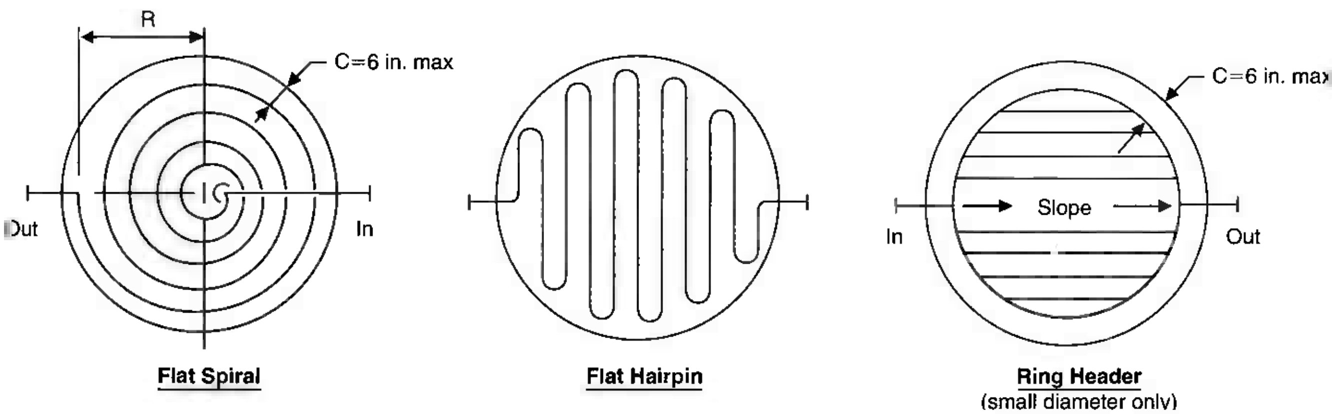




Circular Harp

Note:
Direction of flow will vary depending on the heating or cooling application

Coil Layout for Flat-Bottom Tanks



Developed length of flat spiral coils:

$$L_D = \frac{\pi R^2}{d_0 + C}$$

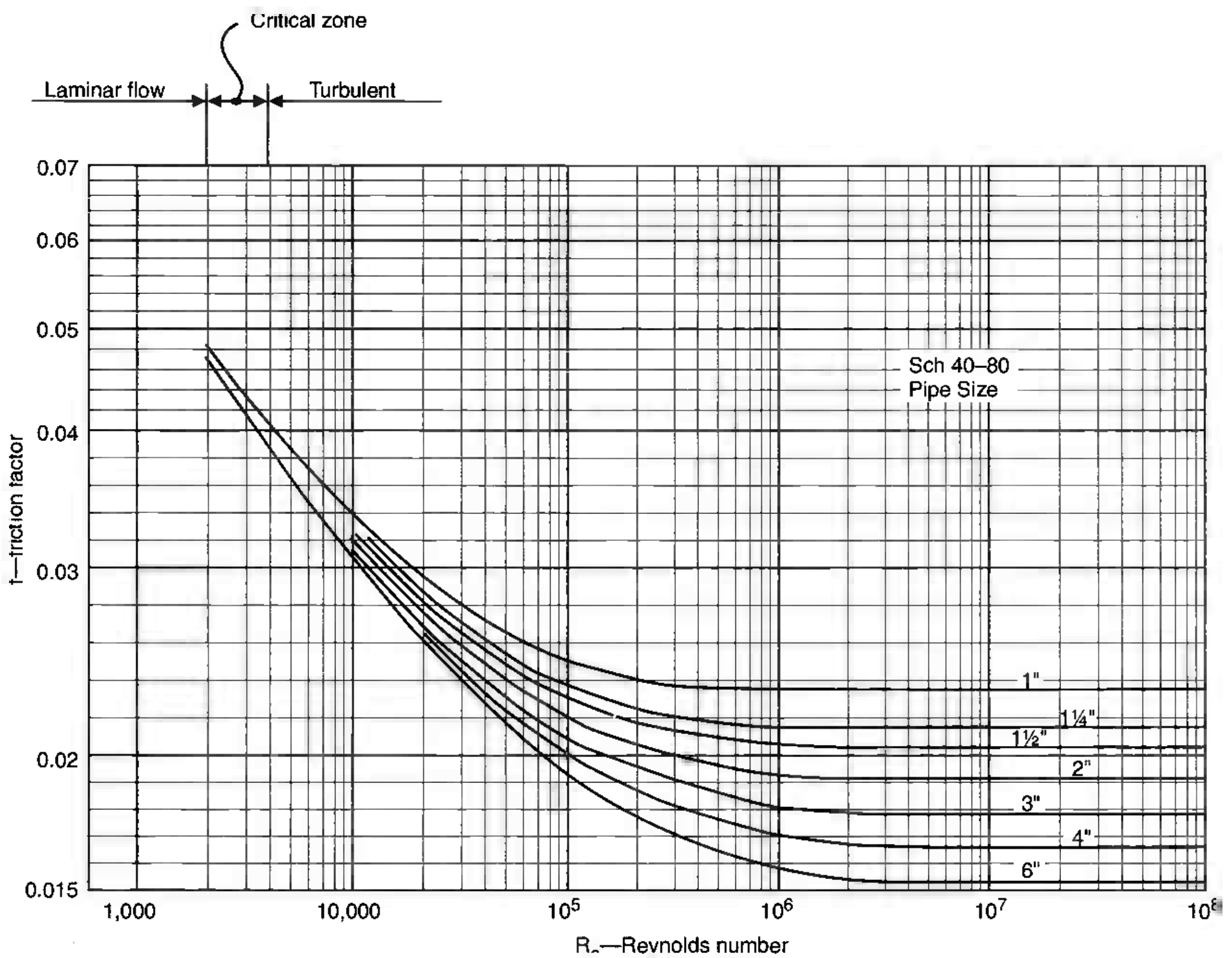
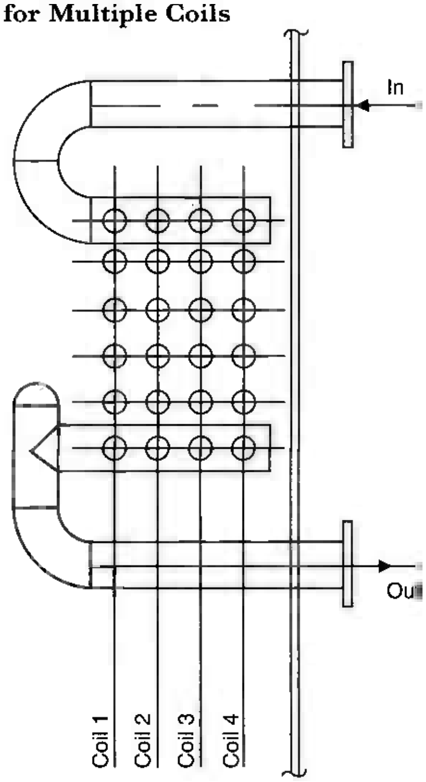


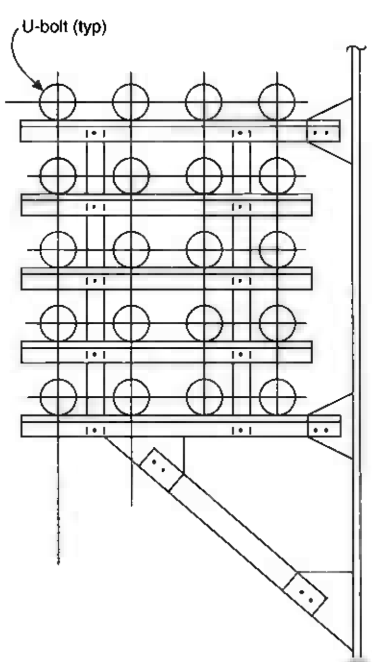
Figure 5-7. Friction factor, f, versus Reynolds number, R_e.

Coil Supports

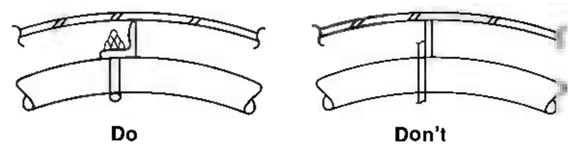
Manifold for Multiple Coils



Support for Multiple Coils

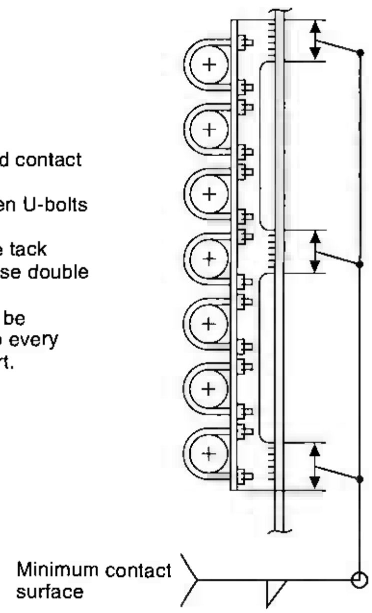


Support for Single Coil

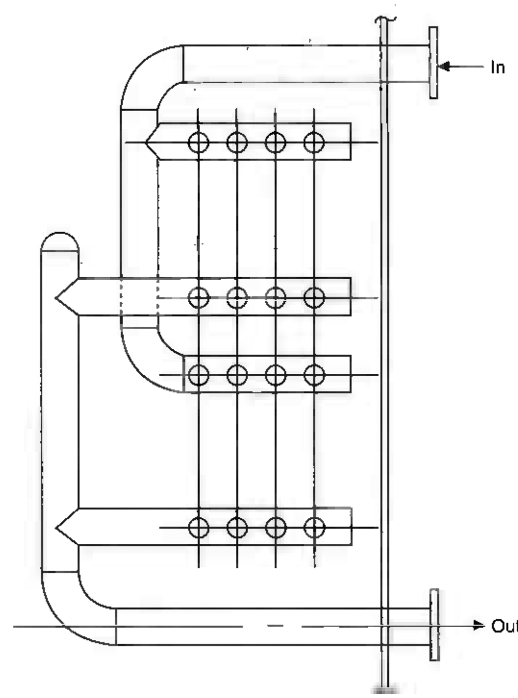


Notes:

1. Provide good contact surface.
2. Do not tighten U-bolts around coil.
3. Nuts may be tack welded, or use double nuts.
4. U-bolts may be alternated to every other support.



Manifold for Multiple Coils, Multiple Series

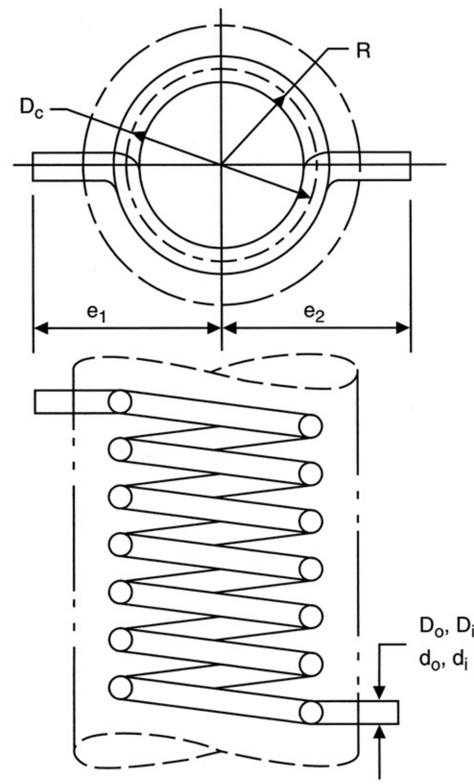


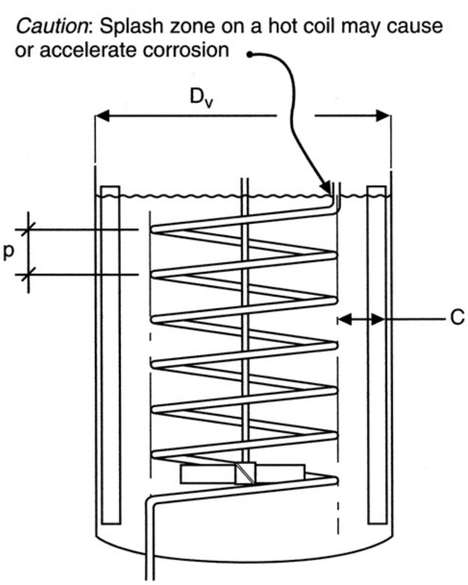
Design of Helical Coils

Notation

A = vessel surface area, ft^2
A_r = surface area of coil required, ft^2
C_p = specific heat of coil or vessel contents, BTU/lb/°F
D_c, d_c = centerline diameter of coil, ft (in.)
D_v = inside diameter of vessel, ft
D_o, D_i = OD/ID of pipe, ft
d_o, d_i = OD/ID of pipe, in.
E = enthalpy, latent heat of evaporation, BTU/lb
f = friction factor
F_{LF} = laminar flow factor
G = rate of flow or quantity of liquid to be heated or cooled, ft^3/hr
GTD = greatest temperature difference, °F
g = acceleration due to gravity, 4.17×10^8 ft/hr^2
g_c = gravitational constant, 32.2 Ibm-ft/lbf-sec^2
h_o, h_i = film coefficients, BTU/hr-ft^2-°F
K = thermal conductivity of pipe, BTU-in/hr/ft^2/°F
L_r = minimum required length of coil, ft
L_a = developed length of coil, ft
LTD = least temperature difference, °F
M = mass flow rate, lb/hr
N = number of turns in coil
NPS = nominal pipe size of coil, in.
P = internal pressure in coil, psig
p = pitch of coil, in.
Q = total heat required, BTU/hr
Q_L = heat loss from vessel shell, BTU/hr
q_L = unit heat loss, BTU/hr
R_e = Reynolds number
S = external pipe surface area, ft^2
S_g = specific gravity of liquid
T = time required to heat or cool the vessel contents, hr
t_p = wall thickness of pipe, in.
t_1 = coil temperature, °F
t_2 = initial temperature of vessel contents, °F
t_3 = final temperature of vessel contents, °F
U = heat transfer coefficient, BTU/hr-ft^2-°F
V = velocity in coil, ft/sec
V_T = volume of vessel contents, ft^3
V_s = specific volume, equal to inverse of density, 1/w, ft^3/lb
W = rate of flow, lb/hr
w = density, lb/ft^3
ΔP = pressure drop, psi
ΔP_L = straight-line pressure drop, psi
ΔT = log mean temperature difference, °F
μ = viscosity, cP

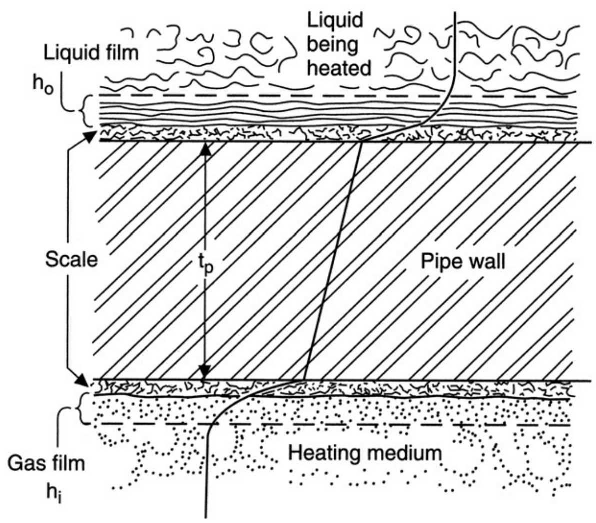
Helical Coil with Baffles and Agitators





Calculations

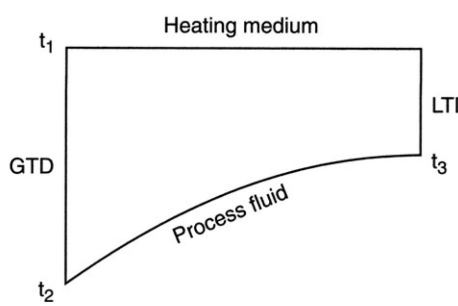
Solving for Heat Transfer Coefficient, U



The value of U can be taken from the various tables or calculated as follows:

$$U = \frac{1}{\frac{1}{h_O} + \frac{t_p}{K} + \frac{1}{h_i}}$$

Heating Applications



- *Determine mass flow rate, M.*

  $M = 62.4GS_g$
- *Determine Δ T.*

  $GTD = t_1 - t_2$

  $LTD = t_1 - t_3$

$$\Delta T = \frac{GTD - LTD}{2.3 \quad \log\left(\frac{GTD}{LTD}\right)}$$

- *Heat required, Q.*

$$Q = MC_p \, \Delta T + Q_L$$

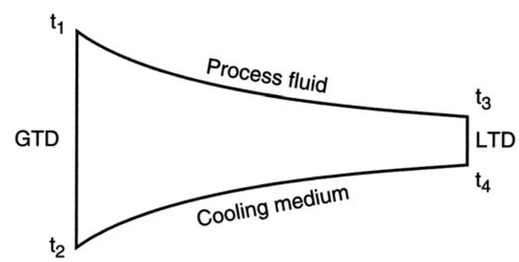
- *Area required, A_r.*

$$A_r = \frac{Q}{U \, \Delta T}$$

- *As an alternative, compute the time required, T.*

$$T = \frac{WC_pGTD}{A_rU\Delta T}$$

Cooling Applications



- *Cooling applications are equivalent to "heat recovery" types of applications. Only the "parallel" type is shown.*
- *Determine mass flow rate, M.*

$$M = 62.4GS_g$$

- *Determine ΔT.*

$$GTD = t_1 - t_2$$

$$LTD = t_3 - t_4$$

$$\Delta T = \frac{GTD - LTD}{2.3 \log\left(\frac{GTD}{LTD}\right)}$$

- *Heat required, Q.*

$$Q = MC_p \, \Delta T - Q_L$$

Subtract heat losses to atmosphere from heat to be recovered.

- *Area required, A_r.*

$$A_r = \frac{Q}{U\,\Delta T}$$

- *As an alternative, compute the time required, T.*

$$T = \frac{WC_pGTD}{A_rU\,\Delta T}$$

Coil Sizing

- *Make first approximate selection of nominal pipe size, NPS.*

$$NPS = \frac{D_v}{30}$$

Preliminary selection:__________

Pipe properties: d_i =__________

D_i =

S =

- *Determine length of coil required, L_r.*

$$L_r = \frac{A_r}{S}$$

- *Check minimum centerline radius, R.*

$$R > 10\ NPS$$

- *Select a pitch of coil, p.* Note: Pitch should be 2 to 2.5 × NPS.

  Use p =________

- *Determine the number of turns required, N.*

$$N = \frac{L_r}{\sqrt{(\pi D_c)^2 + p^2}}$$

  Use N =________

- *Developed length, L_a.*

$$L_a = N\sqrt{(\pi D_c)^2 + p^2}$$

Reynolds Number

- *For steam heating coils.*
  1. Given Q, determine the rate of flow, W:

$$W = \frac{Q}{E}$$

  2. Reynolds number, R_e:

$$R_e = \frac{6.31W}{d_i\mu}$$

- *For other liquids and gases.*
  1. Find velocity in coil, V:

$$V = \frac{0.0509WV_S}{d_i^2}$$

  2. Reynolds number, R_e:

$$R_e = \frac{123.9d_iVw}{\mu}$$

- *Find R_e critical.*

For coils, the critical Reynolds number is a function of the ratio of pipe diameter to coil diameter, computed as follows:

$$R_e\ \text{critical} = 20{,}000\left(\frac{D_i}{D_C}\right)^{0.32}$$

The critical Reynolds number can also be taken from the graph in Figure 5-9.

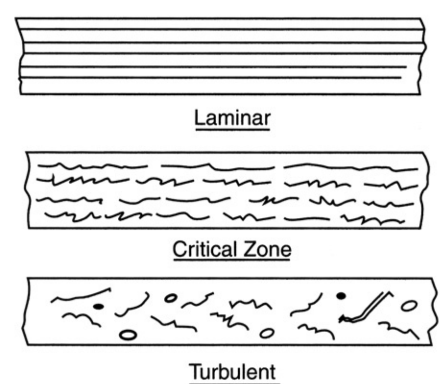


Figure 5-8. Various flow regimes.

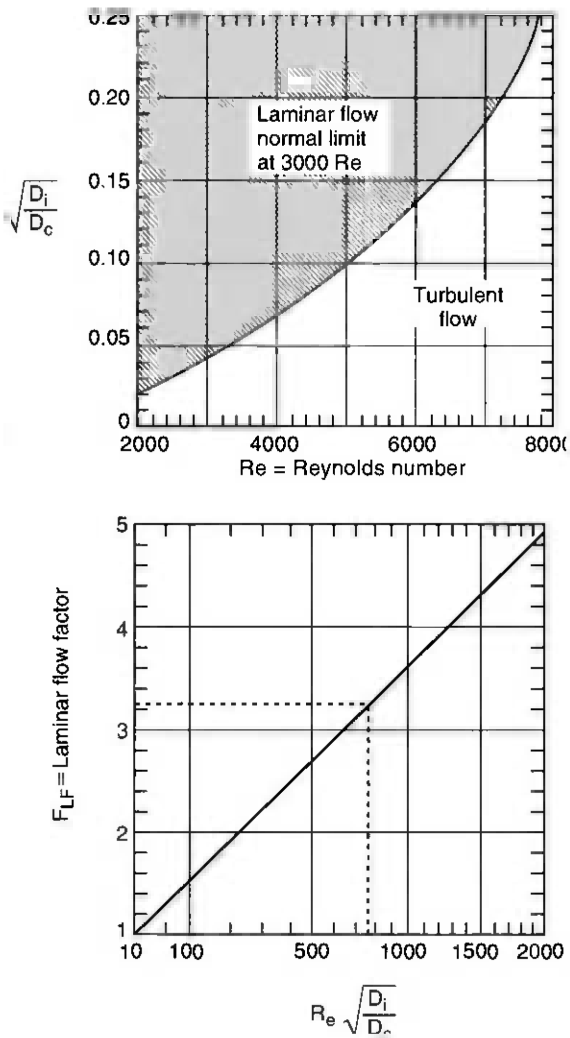


Figure 5-9. Pressure drop factors for flow-through coils. From ASME Transaction Journal of Basic Engineering, Volume 81, 1959, p. 126.

Pressure Drop

- If steam is the heating medium, the pressure drop of condensing steam is;

$$\Delta P = \frac{2fL_aV^2}{3gD_i}$$

The units are as follows;

$f = 0.021$ for condensing steam
L_a is in feet
V is in ft/hr
g is in ft/hr^2
D_i is in ft

- For other fluids and gases;
  a. If flow is laminar,

$$\Delta P_L = \frac{0.00000336fL_aW^2}{d_i^4 w}$$

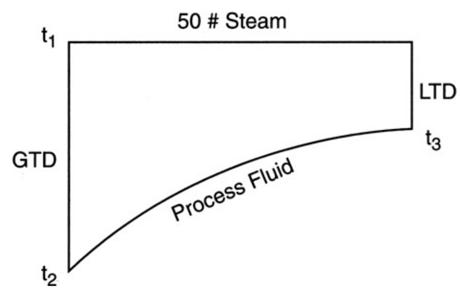
$$\Delta P = \Delta P_L(F_{LF})$$

  b. For turbulent flow,

$$\Delta P_L = \frac{0.00000336fL_aW^2}{d_i^5 w}$$

$$\Delta P = \Delta P_L\sqrt{R_e\left(\frac{d_i}{d_c}\right)^2}$$

Sample Problem 1



Heating Coil: Steam to Oil

- *Batch process.*
- *No agitation (other than natural circulation).*
- *Coil material = carbon steel.*
- *Properties:*

Steam:

$V_s = 6.7$

$E = 912$

$\mu = 0.015$

Oil:

$C_p = 0.42$

$S_g = 0.89$

Vessel:

8 – ft diameter × 30 – ft tan – tan

Liquid height = 15 ft

Volume to liquid height : 700 ft^3 = 5237 gallons

Temperatures:

$t_1 = 300°F$

$t_2 = 60°F$

$t_3 = 200°F$

T = time to heat = 1 hr

- *Log mean temperature difference, ΔT.*

$$GTD = t_1 - t_2 = 300 - 60 = 240$$

$$LTD = t_1 - t_3 = 300 - 200 = 100$$

$$\Delta T = \frac{GTD - LTD}{2.3 \quad \log\left(\frac{GTD}{LTD}\right)} = \frac{240 - 100}{2.3 \quad \log\left(\frac{240}{100}\right)}$$

$$= 160°F$$

- *Quantity of liquid to be heated, G.*

For batch process: $G = \frac{V_T}{T} = \frac{700}{1} = 700\ ft^3/hr$

- *Mass flow rate, M.*

$$M = 62.4GS_g = 62.4(700)0.89 = 38,875\ lb/hr$$

- *Heat required, Q.*

$$Q = MC_p\Delta T + Q_L = 38,875(0.42)160 + 0$$

$$= 2,612,413\ BTU/hr$$

- *Heat transfer coefficient, U.*

U = from Table 5-12: 50–200

from Table 5-13: 20–25
from Table 5-14: 35–60
by calculation: 10–180
Use U = 40.

Table 5-3
Pipe data

Size (in.)	Schedule	d_i (in.)	D_I (ft)	S (ft^2/ft)
1	40	1.049	0.0874	0.344
	80	0.957	0.0797	
1.25	40	1.38	0.115	0.435
	80	1.278	0.1065	
1.5	40	1.61	0.1342	0.497
	80	1.5	0.125	
2	40	2.067	0.1722	0.622
	80	1.939	0.1616	
3	40	3.068	0.2557	0.916
	80	2.9	0.2417	
4	40	4.026	0.3355	1.178
	80	3.826	0.3188	
6	40	6.065	0.5054	1.734
	6	5.761	0.4801	

Table 5-4
Film coefficients

	Medium	Film Coefficient, h_o or h_i
No Change	Water	150–2000
	Gasses	3–50
	Organic solvents	60–500
	Oils	10–120
Condensing	Steam	1000–3000
	Organic solvents	150–500
	Light oil	200–400
	Heavy oil	20–50
Evaporation	Water	1000–2000
	Organic solvents	100–300
	Light oil	200–300
	Heavy oil	100–200

- *Area of coil required, A_r.*

$$A_r = \frac{Q}{U\Delta T} = \frac{(2,612,413)}{40(160)} = 408\ \ ft^2$$

- *Determine the physical dimensions of the coil.*

$$NPS = \frac{D_v}{30} = \frac{96}{30} = 3.2 \qquad \text{Use 3-in. pipe}$$

$C = 12$

Table 5-5
Properties of gases

Material	w	C_p 32°F	C_p 212°F	C_p 932°F
Air	0.0808	0.241	0.242	0.245
Ammonia	0.0482	0.52	0.54	
Benzene		0.22	0.33	0.56
Oxygen	0.0892	0.22	0.225	0.257
Nitrogen	0.0782	0.25	0.25	0.27
Methane	0.0448	0.53	0.6	0.92
Ethane	0.0848	0.4	0.5	0.84
Butane	0.1623	0.375	0.455	0.81
Propane	0.1252	0.38	0.46	0.82
Ethylene	0.0783	0.36	0.45	0.72
CO	0.0781	0.25	0.26	0.27
CO_2	0.1235	0.2	0.21	0.26
Steam			0.453	0.507

Table 5-6
Thermal conductivity of metals, K, BTU-in/hr/ft²/°F

	Temperature, °F								
Material	**200**	**300**	**400**	**500**	**600**	**700**	**800**	**900**	**1000**
Alum—1100-0 annealed	1512	1488	1476	1464	1452	1440	1416		
Alum—6061-0	1224	1236	1248	1260	1272	1272	1272		
Alum—1100 tempered	1476	1464	1452	1440	1416	1416	1416		
Alum—6061-T6	1392	1392	1392	1392	1392	1380	1368		
Carbon steel	360	348	336	324	312	300	288	276	
C-½Mo	348	336	324	312	800	300	288	276	
1Cr–½Mo	324	324	312	300	288	288	276	252	252
2¼Cr–1Mo	300	288	276	276	264	264	252	252	240
5Cr–½Mo	252	252	252	240	240	240	240	228	228
12Cr	168	180	180	180	192	192	192	192	204
18–8 SST	112	118	120	132	132	144	144	156	156
25–20 SST	94	101	107	114	120	132	132	144	144
Admiralty brass	840	900	948	1008	1068				
Naval brass	852	888	924	960	996				
90Cu–10Ni	360	372	408	444	504	564	588	612	636
80Cu–20Ni	264	276	300	324	348	372	408	444	480
70Cu–30Ni	216	228	252	276	300	324	360	396	444
Monel	180	180	192	192	204	216	216	225	240
Nickel	456	432	396	372	348	336	336	348	372
Inconel/incoloy	113	116	119	120	120	132	132	132	144
Titanium	131	128	125	125	126				

Table 5-7
Properties of steam and water

Saturated Steam					Water		
P (PSIG)	**Temp. (°F)**	**V_s (ft³/lb)**	**E (BTU/lb)**	**μ (centipoise)**	**Temp. (°F)**	**V_s (ft³/lb)**	**μ (centipoise)**
5	227	20	961	0.014	32	0.0160	1.753
10	240	16.5	952	0.014	40	0.0160	1.5
15	250	14	945	0.014	50	0.0160	1.299
20	259	12	940	0.015	60	0.0160	1.1
25	267	10.5	934	0.015	70	0.0161	0.95
30	274	9.5	929	0.015	80	0.0161	0.85
35	281	8.5	924	0.015	90	0.0161	0.75
40	287	8	920	0.015	100	0.0161	0.68
45	292	7	915	0.015	150	0.0163	0.43
50	298	6.7	912	0.015	200	0.0166	0.3
75	320	4.9	895	0.016	250	0.0170	0.23
100	338	3.9	881	0.016	300	0.0175	0.18
125	353	3.2	868	0.017	350	0.0180	0.15
150	366	2.7	857	0.018	400	0.0186	0.13
200	388	2.1	837	0.019			
250	406	1.75	820	0.019			
300	422	1.5	805	0.02			

Table 5-8
Properties of liquids

Material	S_g	C_p	w
Water	1	1	62.4
Light oils	0.89	0.42	55.5
Medium oils	0.89	0.42	55.5
Bunker “C”	0.96	0.4	59.9
#6 Fuel oil	0.96	0.4	59.9
Tar/asphalt	1.3	0.4	81.1
Molten sulfur	1.8	0.2	112.3
Molten paraffin	0.9	0.62	56.2

Table 5-9
Viscosity of steam and water, in centipoise, μ

°F	1 psia	2 psia	5 psia	10 psia	20 psia	50 psia	100 psia	200 psia	500 psia	1000 psia	2000 psia	5000 psia	7500 psia	10000 psia	12000 psia
saturated steam	0.667	0.524	0.388	0.313	0.255	0.197	0.164	0.138	0.111	0.094	0.078	…	…	…	…
saturated water	0.010	0.010	0.011	0.012	0.012	0.013	0.014	0.015	0.017	0.019	0.023	…	…	…	…
1500°	0.041	0.041	0.041	0.041	0.041	0.041	0.041	0.041	0.042	0.042	0.042	0.044	0.046	0.048	0.050
1450	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.041	0.041	0.043	0.045	0.047	0.049
1400	0.039	0.039	0.039	0.039	0.039	0.039	0.039	0.039	0.039	0.040	0.040	0.042	0.044	0.047	0.049
1350	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.039	0.041	0.044	0.046	0.049
1300	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.038	0.040	0.043	0.045	0.048
1250	0.035	0.035	0.035	0.035	0.035	0.035	0.035	0.036	0.036	0.036	0.037	0.039	0.042	0.045	0.048
1200	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.035	0.035	0.036	0.038	0.041	0.045	0.048
1150	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.037	0.041	0.045	0.049
1100	0.032	0.032	0.032	0.032	0.032	0.032	0.032	0.032	0.033	0.033	0.034	0.037	0.040	0.045	0.050
1050	0.031	0.031	0.031	0.031	0.031	0.031	0.031	0.031	0.032	0.032	0.033	0.036	0.040	0.047	0.052
1000	0.030	0.030	0.030	0.030	0.030	0.030	0.030	0.030	0.030	0.031	0.032	0.035	0.041	0.049	0.055
950	0.029	0.029	0.029	0.029	0.029	0.029	0.029	0.029	0.029	0.030	0.031	0.035	0.042	0.052	0.059
900	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.029	0.035	0.045	0.057	0.064
850	0.026	0.026	0.026	0.026	0.026	0.026	0.027	0.027	0.027	0.027	0.028	0.035	0.052	0.064	0.070
800	0.025	0.025	0.025	0.025	0.025	0.025	0.025	0.025	0.026	0.026	0.027	0.040	0.062	0.071	0.075
750	0.024	0.024	0.024	0.024	0.024	0.024	0.024	0.024	0.025	0.025	0.026	0.057	0.071	0.078	0.081
700	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.024	0.026	0.071	0.079	0.085	0.086
650	0.022	0.022	0.022	0.022	0.022	0.022	0.022	0.022	0.023	0.023	0.023	0.082	0.088	0.092	0.096
600	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.087	0.091	0.096	0.101	0.104
550	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.019	0.095	0.101	0.105	0.109	0.113
500	0.019	0.019	0.019	0.019	0.019	0.019	0.019	0.018	0.018	0.103	0.105	0.111	0.114	0.119	0.122
450	0.018	0.018	0.018	0.018	0.017	0.017	0.017	0.017	0.115	0.116	0.118	0.123	0.127	0.131	0.135
400	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.131	0.132	0.134	0.138	0.143	0.147	0.150
350	0.015	0.015	0.015	0.015	0.015	0.015	0.015	0.152	0.153	0.154	0.155	0.160	0.164	0.168	0.171
300	0.014	0.014	0.014	0.014	0.014	0.014	0.182	0.183	0.183	0.184	0.185	0.190	0.194	0.198	0.201
250	0.013	0.013	0.013	0.013	0.013	0.228	0.228	0.228	0.228	0.229	0.231	0.235	0.238	0.242	0.245
200	0.012	0.012	0.012	0.012	0.300	0.300	0.300	0.300	0.301	0.301	0.303	0.306	0.310	0.313	0.316
150	0.011	0.011	0.427	0.427	0.427	0.427	0.427	0.427	0.427	0.428	0.429	0.431	0.434	0.437	0.439
100	0.680	0.680	0.680	0.680	0.680	0.680	0.680	0.680	0.680	0.680	0.680	0.681	0.682	0.683	0.683
50	1.299	1.299	1.299	1.299	1.299	1.299	1.299	1.299	1.299	1.298	1.296	1.289	1.284	1.279	1.275
32	1.753	1.753	1.753	1.753	1.753	1.753	1.753	1.752	1.751	1.749	1.745	1.733	1.723	1.713	1.705

Values directly below undescored viscosities are for water.
°Critical point.
Reprinted by permission by Crane Co., Technical Paper No. 410

Table 5-10
Heat loss, Q_L, BTU/hr

ΔT	Surface	Wind			
		Still Air	10 mph	20 mph	30 mph
	Uninsulated	1.8	4.1	5.2	6.1
60°	1-in. insulation	0.18	0.2	0.14	0.21
	2-in. insulation	0.1	0.11	0.11	0.11
	Uninsulated	2.1	4.4	5.7	6.5
100°	1-in. insulation	0.18	0.2	0.21	0.21
	2-in. insulation	0.1	0.11	0.11	0.11
	Uninsulated	2.7	5.1	6.4	7.4
200°	1-in. insulation	0.19	0.21	0.22	0.22
	2-in. insulation	0.11	0.11	0.11	0.11

Table 5-11
Heat transfer coefficient, U, BTU/hr-ft^2-°F

Fluid Giving up Heat	Fluid Receiving Heat	State of Controlling Resistance		Typical Fluid
		Free Convection, U	Forced Convection, U	
Liquid	Liquid	25–60	150–300	Water
		5–10	20–50	Oil
	Gas	1–3	2–10	Water to Air
	Boiling Liquid	20–60	50–150	Water
		5–20	25–60	Oil
Gas	Liquid	1–3	2–10	Air to Water
	Gas	0.6–2	2–6	Gas to Steam
	Boiling Liquid	1–3	2–10	Gas to Boiling Water
Condensing Vapor	Liquid	50–200	150–800	Steam to Water
		10–30	20–60	Steam to Oil
	Gas	1–2	2–10	Steam to Air
	Boiling Liquid	300–800		Steam to Water
		50–150		Steam to Oil

Reprinted by permission by Crane Co., Technical Paper No. 410
Notes:

1. Consider usual fouling for this service.
2. Maximum values of U should be used only when velocity of fluids is high and corrosion or scaling is considered negligible.
3. “Natural convection” applies to pipe coils immersed in liquids under static conditions.
4. “Forced convection” refers to coils immersed in liquids that are forced to move either by mechanical means or fluid flow.
5. The designer should be aware that a natural circulation will arise in the heating mode once the coil is turned on. This natural circulation is not to be confused with forced circulation, which is referred to as “agitated.”

Source: W. H. McAdams, Heat Transmission*, McGraw-Hill Book Co. Inc, 1942.*

Table 5-12
Heat transfer coefficient, U, BTU/hr-ft^2-°F

Liquid	Heating Medium: 150# Steam	10# Steam	180°F Water
Clean fats, oils, etc., 130°F	25	20	17
Clean fats, oils with light agitation	40	40	40
Glycerine, pure, 104°F	40	35	30
Toluene, 80°F	55	47.5	42.5
Methanol, 100°F	70	62	52
Water, soft, 80°F	85	72	66
Water, soft, 160°F	105	82	
Water, soft, boiling	175	108	
Water, hard, 150°F	120	100	

Table 5-13
Heat transfer coefficient, U, BTU/hr-ft^2-°F

Heating Applications		Clean Surface Coefficients		Design Coefficients	
Hot Side	Cold Side	Natural Convection	Forced Convection	Natural Convection	Forced Convection
Steam	Watery solution	250–500	300–550	125–225	150–275
Steam	Light oils	50–70	110–140	40–45	60–110
Steam	Medium lube oils	40–60	100–130	25–40	50–100
Steam	Bunker “C” or #6 fuel oil	20–40	70–90	10–30	60–80
Steam	Tar or asphalt	15–35	50–70	15–25	40–60
Steam	Molten sulfur	35–45	60–80	4–15	50–70
Steam	Molten paraffin	35–45	45–55	25–35	40–50
Steam	Air or gases	2–4	5–10	1–3	4–8
Steam	Molasses or corn syrup	20–40	70–90	15–30	60–80
High temp., hot water	Watery solution	80–100	100–225	70–100	110–160
High temp., heat transf. oil	Tar or asphalt	12–30	45–65	10–20	30–50
Therminol	Tar or asphalt	15–30	50–60	12–20	30–50
Cooling Applications					
Cold Side	Hot Side				
Water	Watery solution	70–100	90–160	50–80	80–140
Water	Quench oil	10–15	25–45	7–10	15–25
Water	Medium lube oils	8–12	20–30	5–8	10–20
Water	Molasses or corn syrup	7–10	18–26	4–7	8–15
Water	Air or gases	2–4	5–10	1–3	4–8
Freon or ammonia	Watery solution	35–45	60–90	20–35	40–60

Notes:

1. Consider usual fouling for this service.
2. Maximum values of U should be used only when velocity of fluids is high and corrosion or scaling is considered negligible.
3. “Natural convection” applies to pipe coils immersed in liquids under static conditions.
4. “Forced convection” refers to coils immersed in liquids that are forced to move either by mechanical means or fluid flow.
5. The designer should be aware that a natural circulation will arise in the heating mode once the coil is turned on. This natural circulation is not to be confused with forced circulation, which is referred to as “agitated.”

Reprinted by permission by Tranter Inc., Platecoil Division, Catalog 5–63

Therefore $D_c = 72$ in.

- *Pipe properties.*

  Assume 3-in. Sch 80 pipe.

$$d_i = 2.9 \text{ in.}$$
$$D_i = 0.2417 \text{ ft}$$
$$S = 0.916 \text{ ft}^2/\text{ft}$$

- *Length of pipe required, L_r.*

$$L_r = \frac{A_r}{S} = \frac{408}{0.916} = 445 \text{ ft}$$

- *Check minimum radius.*

$$\frac{D_c}{2} > 10\ d_i = \frac{72}{2} > 10(3) = 36 > 30$$

- *Determine pitch, p.*

$$P_{max} = 5\text{NPS} = 5(3) = 15$$
$$P_{min} = 2\text{NPS} = 2(3) = 6$$

  Use p = 2.5(3) = 7.5 in.

- *Find number of turns of spiral, N.*

$$N = \frac{L_r}{\sqrt{(\pi D_c)^2 + p^2}} = \frac{445}{\sqrt{[\pi(6)]^2 + 0.625^2}}$$
$$= 23.59$$

  Use (24) turns × 7.5 in. = 180 in.—OK.

- *Find actual length of coil, L_a.*

$$L_a = N\sqrt{(\pi D_c)^2 + p^2}$$
$$L_a = 24\sqrt{[\pi(6)]^2 + 0.625^2} = 486 \text{ ft}$$

- *Rate of flow, W.*

$$W = \frac{Q}{E} = \frac{2,612,413}{912} = 2864 \text{ lb/hr}$$

- *Reynolds number, R_e.*

$$R_e = \frac{6.31W}{d_i\mu} = \frac{6.31(2864)}{2.9(0.015)} = 415,445$$

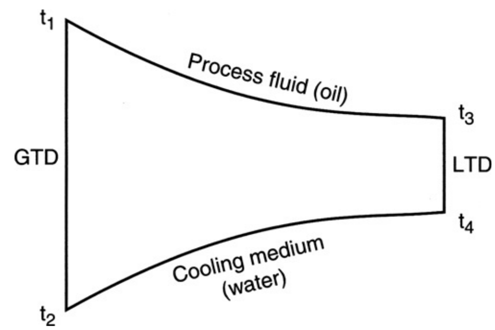
- *Velocity of steam in coil, V.*

$$V = \frac{0.00085WV_s}{d_i^2} = \frac{0.00085(2864)6.7}{2.9^2}$$
$$= 1.94 \text{ ft/sec} = 6982 \text{ ft/hr}$$

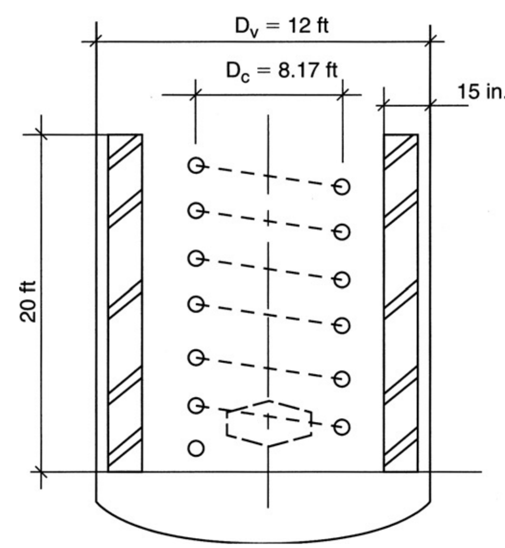
- *Pressure drop, ΔP.*

$$\Delta P = \frac{2fL_aV^2}{3gD_i} = \frac{2(0.021)486\left(6982^2\right)}{3(4.17 \times 10^8)0.2417}$$
$$= 1.88 \text{ psi}$$

Sample Problem 2



Cooling application:	Parallel flow
Process fluid:	Hot oil (vessel contents)
Cooling medium:	Water (coil contents)
Vessel indoors:	$Q_L = 0$
Discharge rate:	3000 GPH
Agitation:	Yes
Baffles:	Yes



Properties

Process fluid

$$C_p = 0.42$$
$$\mu = 13 @ 110°F$$

Coil medium

$\mu = 0.75 @ 90°F$

$V_s = 0.0161$

$w = 62.4$

Temperatures

$t_1 = 140°F$

$t_2 = 60°F$

$t_3 = 110°F$

$t_4 = 90°F$

- *Log mean temperature difference, ΔT.*

$$GTD = t_1 - t_2 = 140 - 60 = 80$$
$$LTD = t_3 - t_4 = 110 - 90 = 20$$
$$\Delta T = \frac{GTD - LTD}{2.3\,\log\left(\frac{GTD}{LTD}\right)} = \frac{80 - 20}{2.3\,\log\left(\frac{80}{20}\right)} = 43°F$$

- *Mass flow rate, M.*

$$M = 3000\frac{gal}{hr}\left(8.33\frac{lb}{gal}\right) = 24,990 \ \ lb/hr$$

- *Heat required, Q.*

$$Q = MC_p\Delta T - Q_L = 24,990(0.42)43 - 0$$
$$= 451,319 \ BTU/hr$$

- *Heat transfer coefficient, U.*

U = from Table 5-14: 10 – 20
Use U = 15.

- *Area of coil required, A_r.*

$$A_r = \frac{Q}{U\Delta T} = \frac{451,319}{15(43)} = 700 \ \ ft^2$$

- *Determine baffle sizes.*

Baffle width, B = 0.083D = 12 in.
Off wall, $B_c = 0.021D = 3$ in.

- *Determine the physical dimensions of the coil.*

$$NPS = \frac{D_v}{30} = \frac{144}{30} = 4.8$$

Use 4-in. pipe.

- *Pipe properties.*

Assume 4-in. Sch 80 pipe:

$d_i = 3.826$ in.

$D_i = 0.3188$ ft

$S = 1.178 \ ft^2/ft$

- *Determine coil diameter, D_c.*

$$D_c = 144 - 2(15) - 2(8) = 98 \text{ in.} \ (8.17 \text{ ft})$$

- *Length of pipe required, L_r.*

$$L_r = \frac{A_r}{S} = \frac{700}{1.178} = 595 \ ft$$

- *Check minimum radius.*

$$\frac{D_c}{2} > 10d_i = \frac{98}{2} > 10(4.5) = 49 > 45$$

- *Determine pitch, p.*

$$P_{max} = 5NPS = 5(4) = 20$$
$$P_{min} = 2NPS = 2(4) = 8$$

Use p = 2.5(4) = 10 in. = 0.833 ft

- *Find number of turns of spiral, N.*

$$N = \frac{L_r}{\sqrt{(\pi D_c)^2 + p^2}} = \frac{595}{\sqrt{[\pi(8.17)]^2 + 0.833^2}}$$
$$= 21.6$$

Use (22) turns × 10 in. = 220 in. < 240 in.—OK

- *Find actual length of coil, L_a.*

$$L_a = N\sqrt{(\pi D_c)^2 + p^2}$$
$$L_a = 22\sqrt{[\pi(8.17)]^2 + 0.833^2} = 565 \ ft$$

- *Rate of flow, W.*

$$W = 24,990 \ lb/hr$$

- *Velocity, V.*

$$V = \frac{0.0509WV_s}{d_i^2} = \frac{0.0509(24,990)0.0161}{3.826^2}$$
$$= 1.4 \ \ ft/sec$$

- *Reynolds number, R_e.*

$$R_e = \frac{123.9d_iV}{V_S\mu} = \frac{123.9(3.826)1.4}{0.0161(0.75)} = 54,961$$

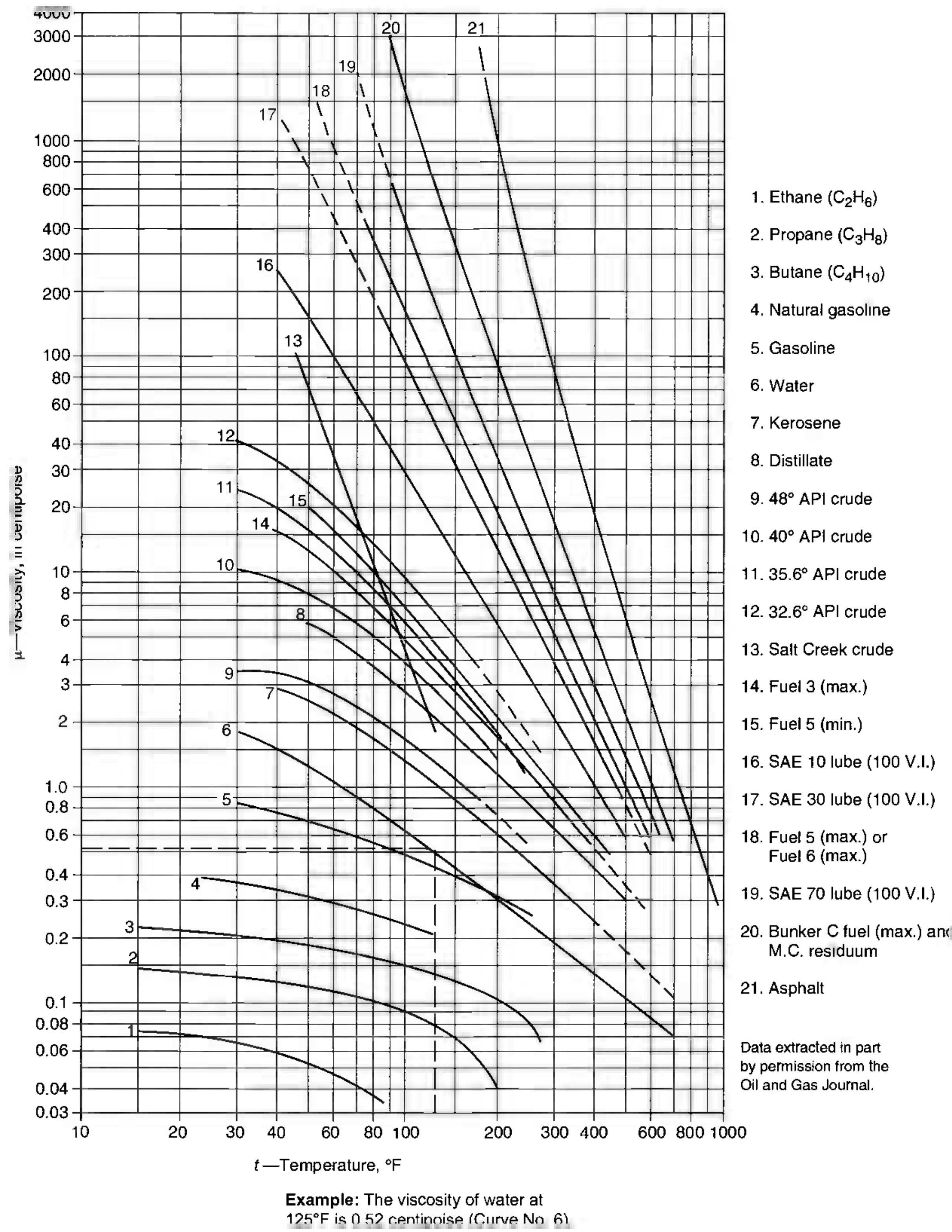


Figure 5-10. Viscosity of water and liquid petroleum products. Reprinted by permission by Crane Co., Technical Paper No. 410

Table 5-15
Effect of metal conductivity on "U" values

Application	Material	Film Coefficients		Thermal Conductivity BTU-in/hr/ft^2/°F	Metal Thickness (in).	U (BTU/hr-ft^2-°F)
		h_o	h_i			
Heating water with saturated steam	Copper	300	1000	2680	0.0747	229
	Aluminum	300	1000	1570	0.0747	228
	Carbon steel	300	1000	460	0.0747	223
	Stainless steel	300	1000	105	0.0747	198
Heating air with saturated steam	Copper	5	1000	2680	0.0747	4.98
	Aluminum	5	1000	1570	0.0747	4.97
	Carbon steel	5	1000	460	0.0747	4.97
	Stainless steel	5	1000	105	0.0747	4.96

Therefore flow is turbulent!

- *Straight-line pressure drop*, ΔP_L.

$$\Delta P_L = \frac{(3.36 \times 10^{-6}) f L_a W^2}{d_i^5 w}$$

$$\Delta P_L = \frac{(3.36 \times 10^{-6}) 0.0218(565)24,990^2}{3.826^5(62.11)}$$

$$= 0.5 \text{ psi}$$

- *Pressure drop,* ΔP.

$$\Delta P = \Delta P_L \sqrt{R_e \left(\frac{d_i}{d_C}\right)^2}$$

$$\Delta P = 0.5\sqrt{54,961\left(\frac{3.826}{98}\right)^2} = 4.57 \text{ psi}$$

Procedure 5-7: Agitators/Mixers for Vessels and Tanks

Mixing is defined as the intermingling of particles that produce a uniform product. Hydraulically, mixers behave like pumps. Mixing applications can be either a batch or a continuous process. Although the terms *agitation* and *mixing* are often used interchangeably, there is a technical difference between the two.

Agitation creates a flow or turbulence as follows:

- Mild agitation performs a blending action.
- Medium agitation involves a turbulence that may permit some gas absorption.
- Violent agitation creates emulsification.

Mechanical mixers are used as follows:

- To mix two or more nonhomogeneous materials.
- To maintain a mixture of materials that would separate if not agitated.
- To increase the rate of heat transfer between materials.

The mechanical mixer usually consists of a shaft-mounted impeller connected to a drive unit. Mechanical mixers can be as small as ¼ hp or as large as 200 hp for some gear-driven units. Power consumption over time determines the efficiency and economy of a mixing process. Top-mounted mixers can be located on center (VOC), off center (VOFC), or angled off center (AOC). Mixers on center require baffles.

If the ratio of liquid height to vessel diameter is greater than 1.25, then multiple impellers are recommended. Ratios of 2:1 and 3:1 are common in certain processes. A common rule of thumb is to use one impeller for each diameter of liquid height.

Mixer applications are designed to achieve one of the following:

- *Blending:* combines miscible materials to form a homogeneous mixture.
- *Dissolving:* the dissipation of a solid into a liquid.
- *Dispersion:* the mixing of two or more nonmiscible materials.
- *Solid suspension:* suspends insoluble solids within a liquid.
- *Heat exchange:* promotes heat transfer through forced convection.
- *Extraction:* separation of a component through solvent extraction.

Mounting

Top-entering units can generally be used on all applications. Side-entering units are usually used for low speed, mild blending, and tank cleaning operations. The most efficient mounting is angled off center (AOC).

Tank Baffles

Antiswirl baffles are required in most larger industrial fluid-mixing operations. Baffles are used for center-shaft, top-mounted mixers to prevent vortexing. Baffles also promote top-to-bottom turnover and represent good mixing practice. The most usual arrangement is to have four baffles spaced at 90°. For viscosities up to 500 centipoise, baffles can be mounted directly to the wall. For use in higher-viscosity material or in any mixing application where solids can build up or where other harmful effects develop when the baffle attaches to the wall, the baffles should be spaced off of the wall. Normal spacing is 25% of the actual baffle width. Above 10,000 cP, baffles should be mounted at least 1½ in. off the wall. Above 20,000 cP, no baffles are typically required. Horizontal tanks do not usually require baffles. Baffles should be selected for the minimum viscosity that will occur during a mixing cycle.

As liquid viscosities go up, the need for baffles—and thus the baffle width—decreases. The industrial use of vessels without baffles is limited because unbaffled systems give poor mixing.

Baffle widths and the wall clearance depend on the viscosity of the liquid being mixed:

Viscosity, cP	Baffle Width, B	Off Wall, Bc
Waterlike	0.083 D	0.021 D
5000	0.056 D	0.014D
10,000	0.042 D	0.011 D
20,000	0.021 D	0.005 D

Impellers

Impellers come in the following types:

- Paddle.
- Propeller.
- Anchor.
- Turbine.
- Ribbon.

Paddle-type impellers are the simplest and lowest cost impellers, but they have small pumping capacity. They have very low axial flow, hence the pitched flat blade version is normally used for low-viscosity materials. The ratio of blade diameter to vessel diameter is usually ⅓ to ⅔. A radial flow impeller is used for high shear.

Propeller types pump liquid. Every revolution of a square pitch prop discharges a column of liquid approximately equal to the diameter of the propeller. The flow is axial. Such pumps are used primarily for high-speed applications and side-entry mixers. Dual propellers are used on vessels with H/D ratios greater than 1. The axial flow decreases mix time. They are heavier and cost more than pitched-blade turbines. Propeller-to-vessel-diameter ratio is usually ⅓. A propeller-type impeller is used for high flow.

Anchor-type impellers rotate slowly and have a large surface area. This makes them ideal for batch applications in higher-viscosity materials.

Turbines are always mounted vertically. They are used at low speed where the application requires greater shear than pumping and higher horsepower per unit volume. There are two basic forms of turbines, the flat-blade radial-discharging type and the pitched-blade axial-thrust type. All others are modifications of these basic types. The ratio of blade diameter to vessel diameter is usually ⅓.

Flat-blade turbines pump liquid outward by centrifugal force. Liquid that is displaced by the blade is replaced by flow from the top and bottom. Suction comes from the center, and delivery is on the circumference of the blade. The primary flow is radial. This is the most widely used type of mechanical agitator. The number of blades vary from 4 to 12. This turbine is used primarily for liquid–liquid dispersion. Turbines with curved blades are used for higher-viscosity materials.

The pitched-blade turbine produces a combination of axial and radial flow. The purpose of pitching the blade is to increase radial flow. Blades can be sloped anywhere from 0° to 90°, but 45° is the commercial standard.

Notes

1. All mixers/agitators rotate clockwise.
2. In general, agitators are sized on the basis of the required torque per unit volume. Other factors that affect size and torque are:
    - Viscosity $>$ 100 cP (viscosity can affect blend times).
    - Critical speeds.
    - Tip speed.
    - Impeller diameter.
    - Required degree of agitation.
3. Each shaft is designed for mechanical loads and critical shaft speed. Motor size and shaft design are related. A larger shaft to take the torque will require more horsepower to eliminate wobble.
4. To prevent solid buildup on the bottom, a radial-blade impeller may be used. If elected, then place the blade one blade width off the bottom.
5. Power consumption:
    - Operating speed is back-calculated to ensure delivery of the proper power for a given impeller diameter.
    - The speed and horsepower define the torque required for the system. The torque in turn sets the shaft size and gear box size.
    - Impeller power consumption determines the horsepower and impeller diameter required for a given mixing process.
6. Mixing parameters:
    - Shaft angle.
    - Time.
    - Impeller type and diameter.
    - RPM (pumping capacity).
    - Power.
    - Viscosity, specific gravity.
7. A steady rest bearing may be utilized at the bottom of the tank if the mixing application allows.
8. Other applicable data:
    - Types of seals or packing.
    - Metallurgy.
    - Drain location.
    - Manway size.
    - Indoor/outdoor.
    - Mixer/agitator run times.
    - Head room required above tank.

Vessels with Agitators or Mixers

Notation

H_p = motor horsepower
N = impeller RPM
D = vessel diameter, in.
d = impeller diameter, in.
B = baffle width, in.
B_c = baffle off-wall distance, in.

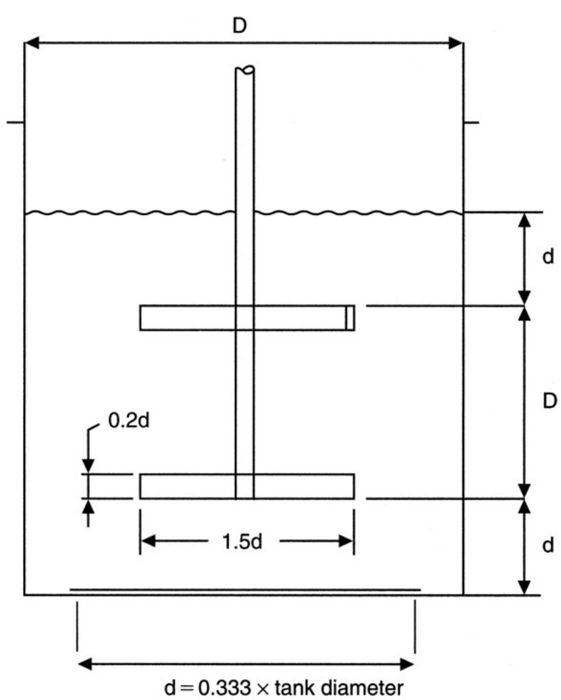
- *Force on baffle, F.*

$$F = \frac{(56,800H_P)}{2N[D - B - (2B_c)]}$$

- *Force per unit area, F_u.*

$$F_u = \frac{F}{Bh}$$

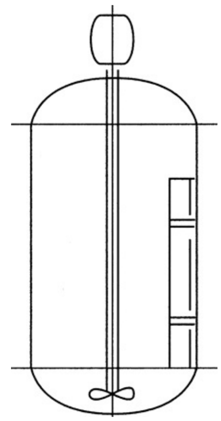
- *Typical ratios.*



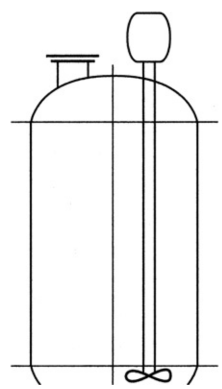
Use wear plate with pitched or propeller blades and suspended solids.

Types of Mixers

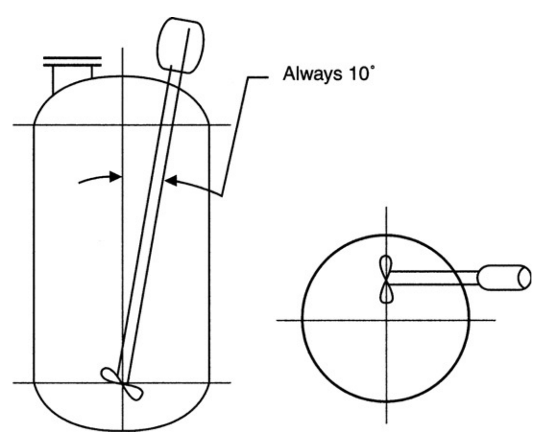
Type 1: Vertical On Center (VOC)
- Requires baffles.



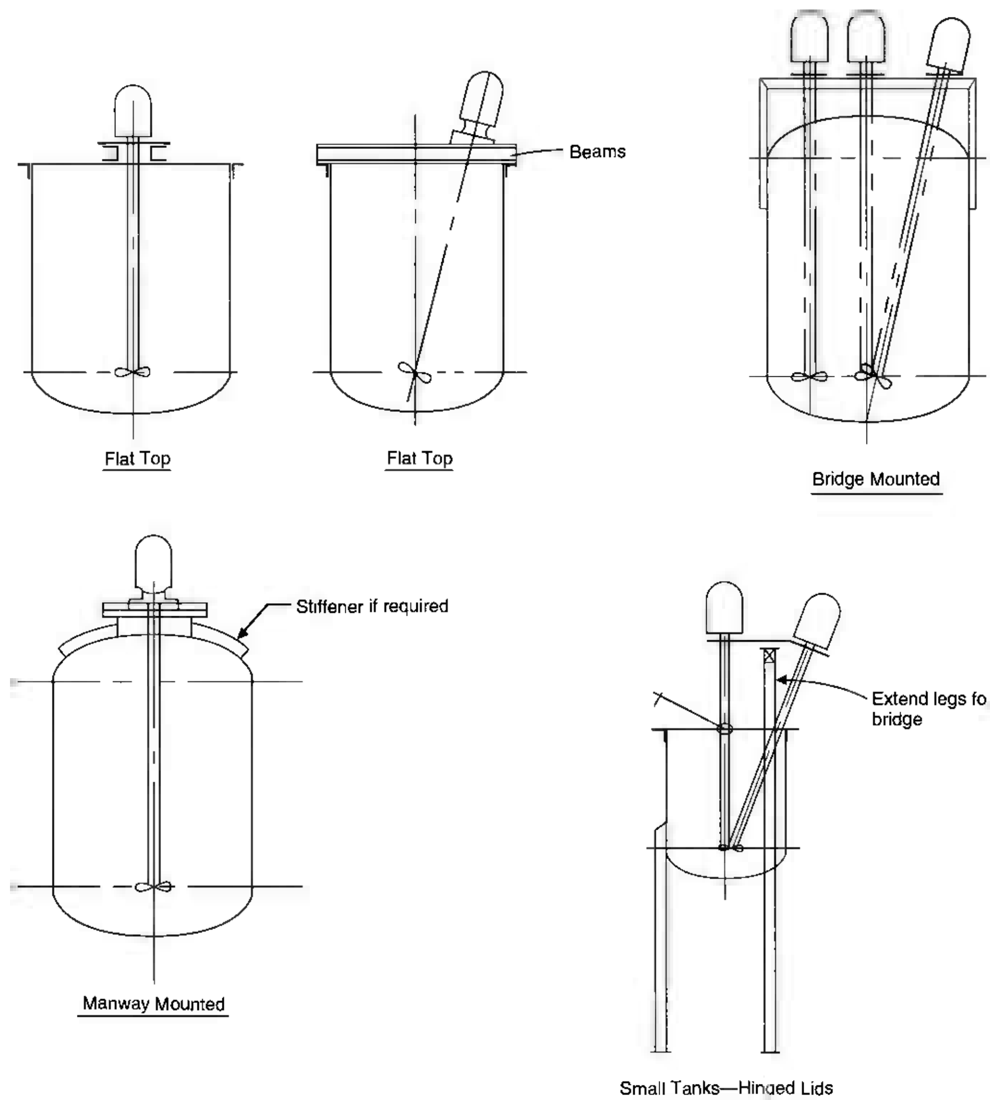
Type 2: Vertical Off Center (VOFC)
- Least effective.
- Poor mixing.



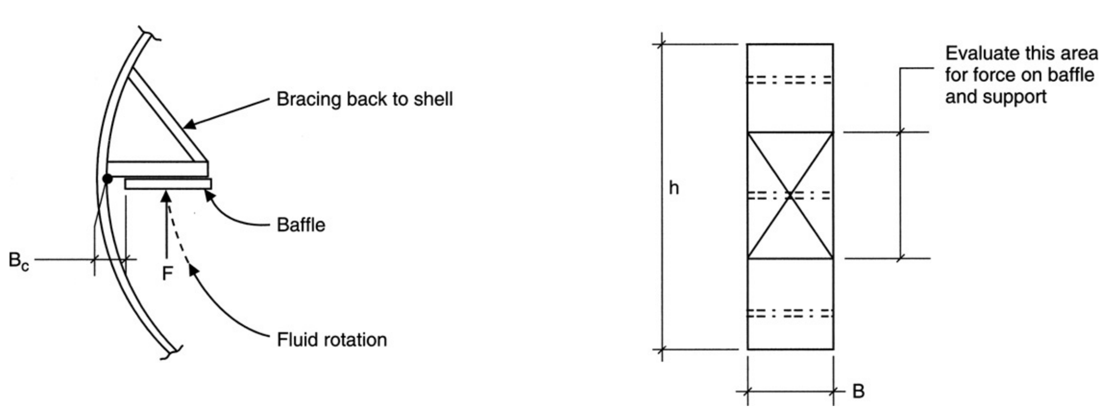
Type 3: Angular Off Center (AOC)



Types of Mounting

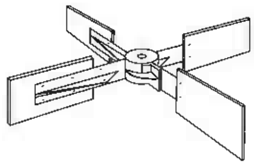


Baffle Supports

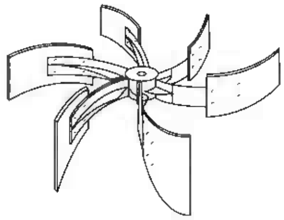


Types of Impellers

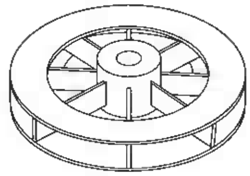
Turbine Impellers



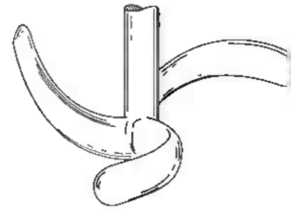
a. Flat blade



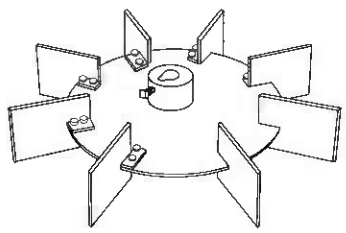
b. Curved blade



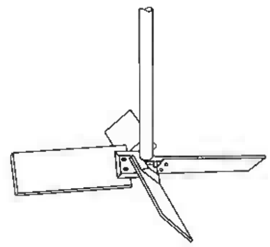
c. Shrouded



d. Retreating blade

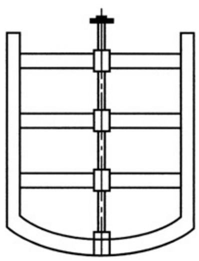


e. Disk flat blade

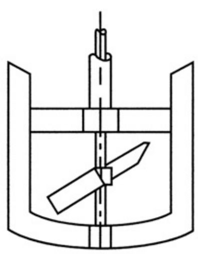


f. Pitched blade

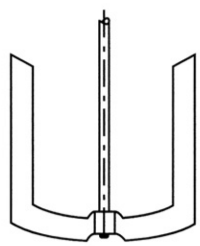
Anchor Impellers



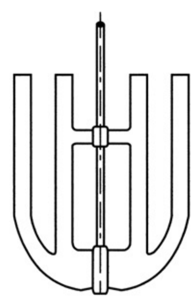
a. Horseshoe with cross members



b. Double-motion horseshoe paddle

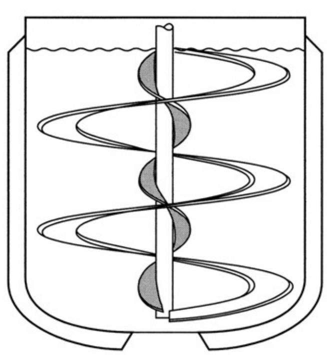


c. Horseshoe

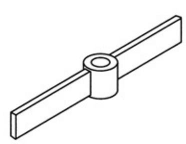


d. Gate type

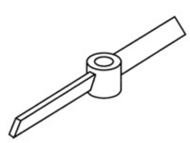
Miscellaneous Impellers



Helical ribbon

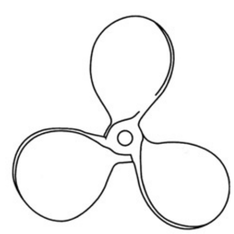


Straight flat blade



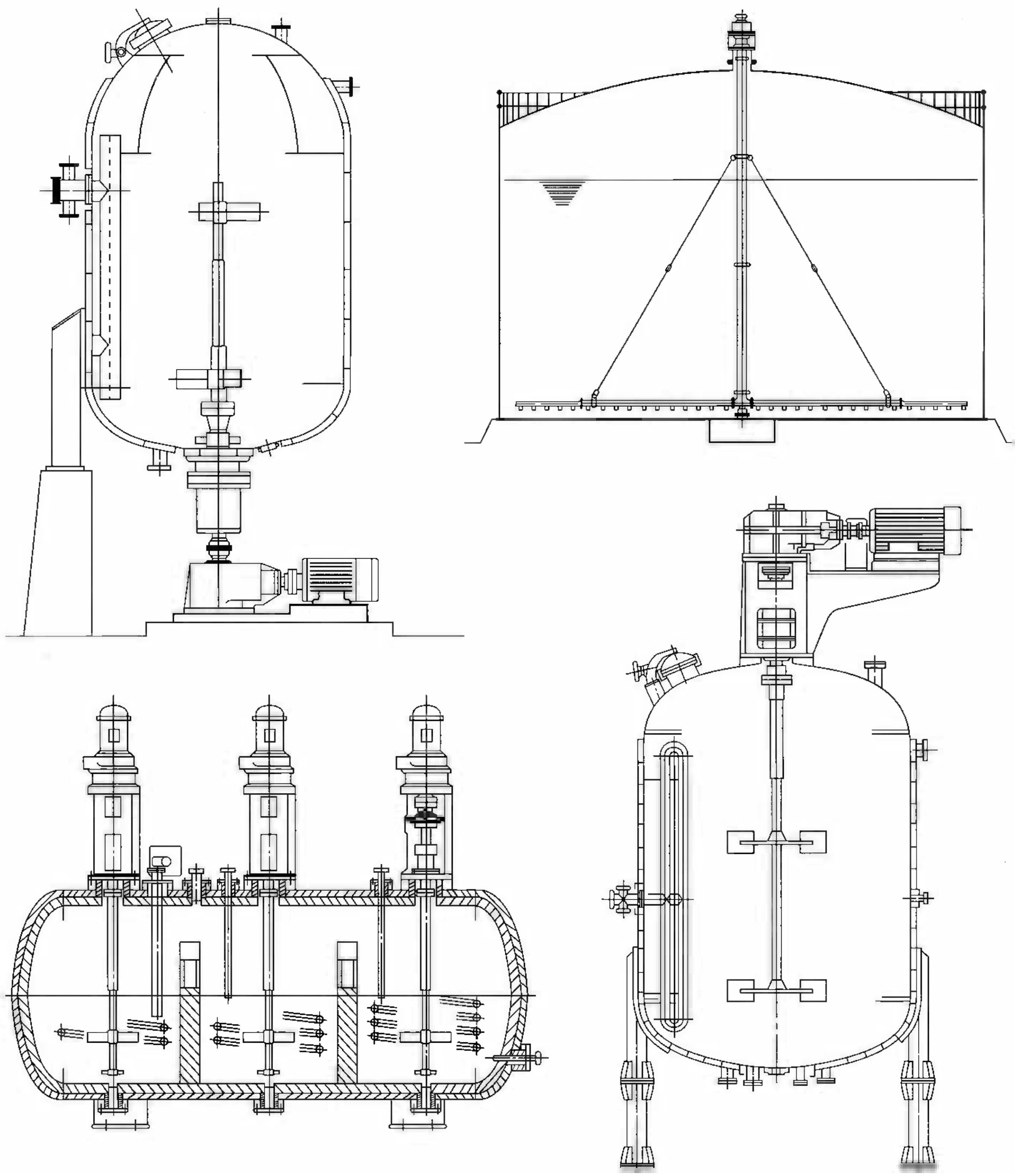
Pitched flat blade

Paddle



Propeller

Typical Applications



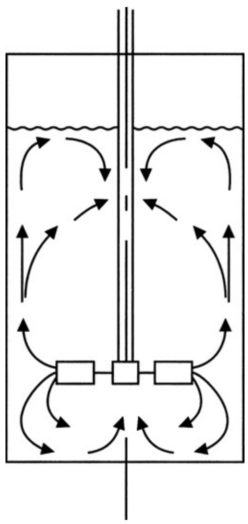
Impeller Actions

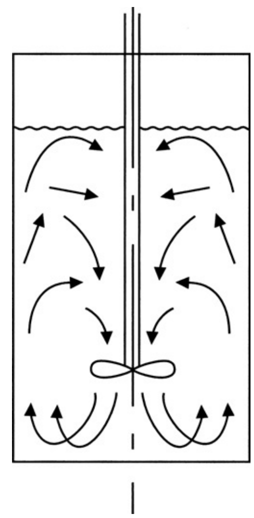
Shear Action

- Break up liquid blobs.
- Use radial-flow impeller such as turbine or paddle types without pitched blades.

Pumping Action

- Lift solids from bottom.
- Good for blending solids and liquids.
- Use propeller or turbine or paddle type with pitched blades.





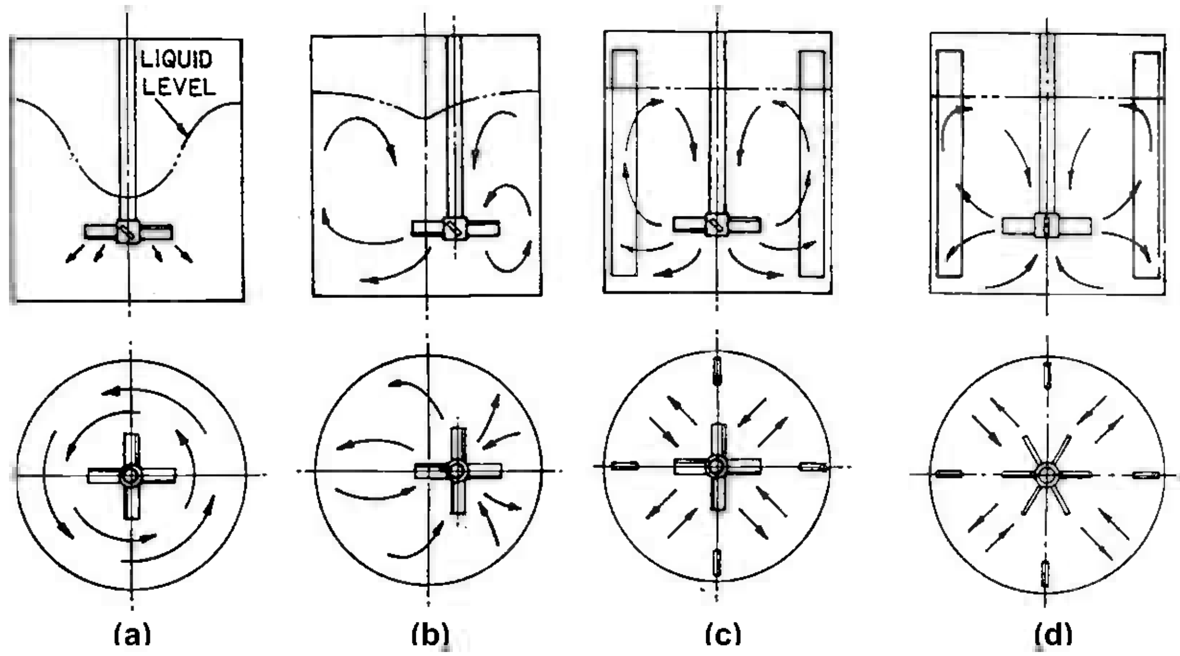


Figure 5-11. Agitator flow patterns. (a) Axial or radial impellers without baffles produce vortexes. (b) Off-center location reduces the vortex. (c) Axial impeller with baffles. (d) Radial impeller with baffles.

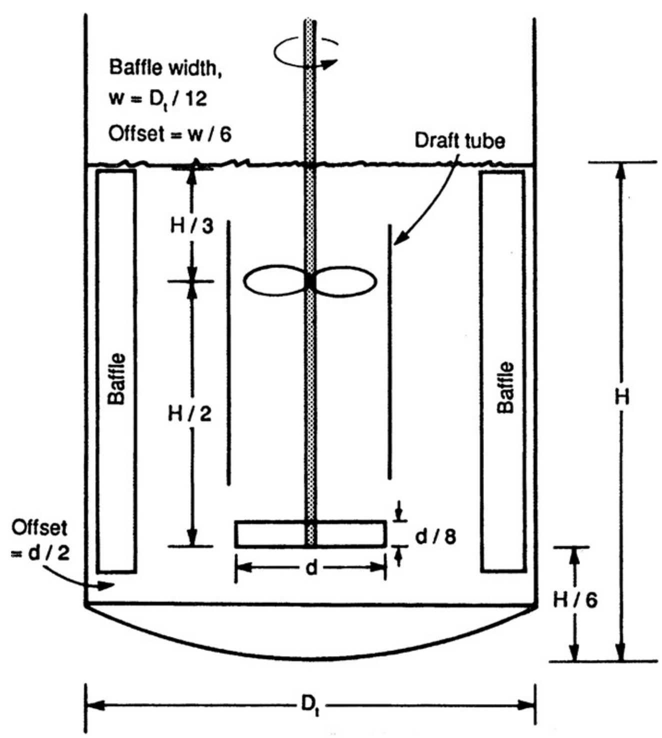


Figure 5-12. Typical proportion of stirred tank design with radial and axial impellers and baffles. The upper axial impeller is housed in a draft tube. For radial impellers, $.3 < d/D_t < .6$

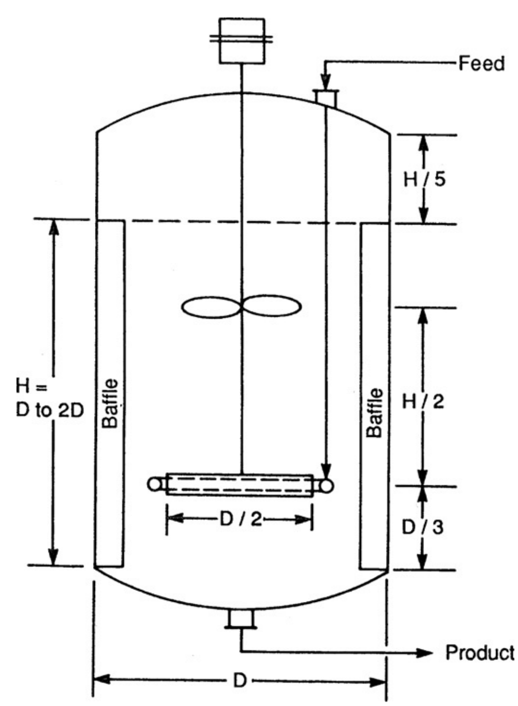


Figure 5-13. Typical proportions of a stirred tank reactor with radial and axial impellers, four baffles, and a sparger feed inlet.

Procedure 5-8: Design of Internal Pipe Distributors

Uniform liquid distribution is essential for efficient operation of chemical processing equipment. To obtain optimum distribution, proper consideration must be given to flow behavior in the distributor itself, flow conditions upstream and downstream of the equipment.

This procedure does not consider flows upstream or downstream of the distributor. However, disturbances upstream of the distributor is dependent on the piping configuration and may increase or decrease the flow to the distribution device.

There are various kinds and types of distributors to handle the following flow regimes;

1. Gas or vapor
2. Liquid
3. Two phase flow

This procedure is only concerned with liquid or two phase flow, and does not consider gas/vapor type distribution devices. There are several types of liquid distributors for packed columns that are also not part of this discussion. These are;

1. Trough distributors
2. Orifice plate distributors

Internal pipe distributors are frequently used for feed or reflux inlets for trayed columns or towers (fractionation or distillation columns). They are used to distribute the inlet stream to a particular point on a tray or uniformly across a tray. The distributor consists of one or more pipes, with or without branches, containing a series of holes, slots or spray nozzles. This procedure addresses the design of these distribution devices.

Internal pipe distributors are also known as "perforated spargers" or "perforated pipe distributors" or headers. They can be designed for either liquid only or two phase flow. A pipe distributor may be considered as liquid service if the volume of vapor is no more than 5%.

Rectangular slots or holes can be used but slots are preferred.

It is not possible to discharge liquid from a distributor pipe with uniform velocities at all discharge points unless the average velocity of the liquid from the holes or slots is double the velocity of the liquid at the inlet. Expressed as an equation, it would look as follows:

Average $V_h \geq 2\,X$ Average V_P

To get uniform volumetric flow where the above criteria cannot be met, it is necessary to calculate the velocity from each orifice, then calculate the area required to get the necessary volumetric flow. To do this it is necessary to calculate the pressure profile in the distributor at all the various points. This would be a very complicated procedure and is seldom done. This method is not discussed here. Only a simplified procedure is presented and is only valid for inlet velocities between 6 and 14 feet per second.

In a simple perforated pipe or sparger, the flow distribution is uniform. This will be the case if the following has been properly considered;

1. Pressure recovery due to kinetic energy or momentum changes
2. Frictional pressure drop along the length of the pipe
3. Pressure drop across the outlet holes

The procedure presented here is only valid if the following criteria are met;

- The distributor pipe is horizontal
- The slots or holes are located either horizontally along the pipe or pointed downward at a maximum of 30° from horizontal.
- The slots (holes) are equally spaced along the pipe.

The following physical criteria should be met for the piping distributor:

- Slots shall meet the dimensional proportions shown in Fig 5.14.
- The minimum hole size is 0.5 inches (12 mm)
- The length of cut metal should not exceed the length of uncut metal in any given row.
- The ratio of total hole area to pipe cross sectional area should be between 1 and 3.
- Tolerance for hole or slot location is ± 0.25 inches (± 6 mm)
- Slot spacing should not exceed 24 inches (610 mm).

Variables for Distributor Design;

1. End condition
   a. Open end
   b. Closed end
   c. Elbows
   d. Tees
   e. Slotted ends

2. Discharge Type
   a. Slots
   b. Holes
   c. Direction of slots
   d. Spacing of holes
   e. Spray nozzles
3. Discharge Location
   a. Center
   b. Sides
   c. Ends
4. Function
   a. Feed
   b. Reflux
5. Passes
   a. 1 pass
   b. 2 pass
   c. 4 pass
   d. Packed bed
6. Distributor Type
   a. T-Type
   b. Ladder Type
   c. H-Type
   d. Single
   e. Multiple
7. Baffle
   a. With
   b. Without
8. Contents
   a. Liquid flow
   b. Two phase flow
9. Support Type
   a. U-bolts with bracket
   b. Clip
   c. Cross diameter angle
   d. Dummy pipe extension

Notes

1. Total slot/hole area shall be 1-3 times as large as the cross sectional area of the feed pipe.
2. The discharge velocity for any given slot or hole should not exceed 9 FPS. Higher velocities could result in excessive splashing or tray turbulence. If the slot velocity exceeds this value, the main header should be enlarged one size and the slot velocity recalculated. Normally, slot velocity will be satisfactory if the inlet velocity of the header is below 14 FPS
3. Normally, the process engineer will have already sized the inlet line before mechanical begins the detailed design of the internals. Therefore this procedure does not include sizing of the main lines.
4. The long axis of the slots should coincide with the long axis of the pipe.
5. The number and size of branches is typically determined by the configuration of the internals. The discharge locations will dictate single or multiple feeds as well as the basic layout.
6. Spray headers are typically used for packed beds in certain process columns. The design of spray headers is not covered in detail in this procedure. However the basic steps are as follows;
   a. Determine total flow
   b. Select spray nozzles that will give the desired flow, coverage and spray patterns.
   c. Layout nozzles to accommodate coverage.
   d. Develop layout and configuration of header configuration.
   e. Calculate flow rates to all spray nozzles
7. The pipe distributor must be capable of supporting itself, or provide for the support. The support design should be based on deflection rather than stress. Deflection should be limited to 0.5 inches. The formulas for deflection are as follows:
   a. Supported on both ends:
   $$\delta = \left(5\ w\ L^4\right) / \left(384\ E\ I\right)$$
   b. Cantilever;
   $$\delta = \left(w\ L^3\right) / \left(8\ E\ I\right)$$

Notation

A_b = Area required for one branch, in^2
A = Total area, all holes or slots, in^2
a = Area of each hole or slot, in^2
A_P = Cross sectional area of pipe, in^2
A_r = Area required, all holes, in^2
C = Orifice discharge coefficient, .6 to .63
d = Diameter of holes or length of slots, in
D_P = ID of pipe, in
f = Fanning friction factor
f_m = Density of 2-phase flow, liquid/vapor, Lbs / Ft^3
G = Specific gravity
g = Acceleration due to gravity = 32.17 Ft/Sec^2
H = Head, Ft

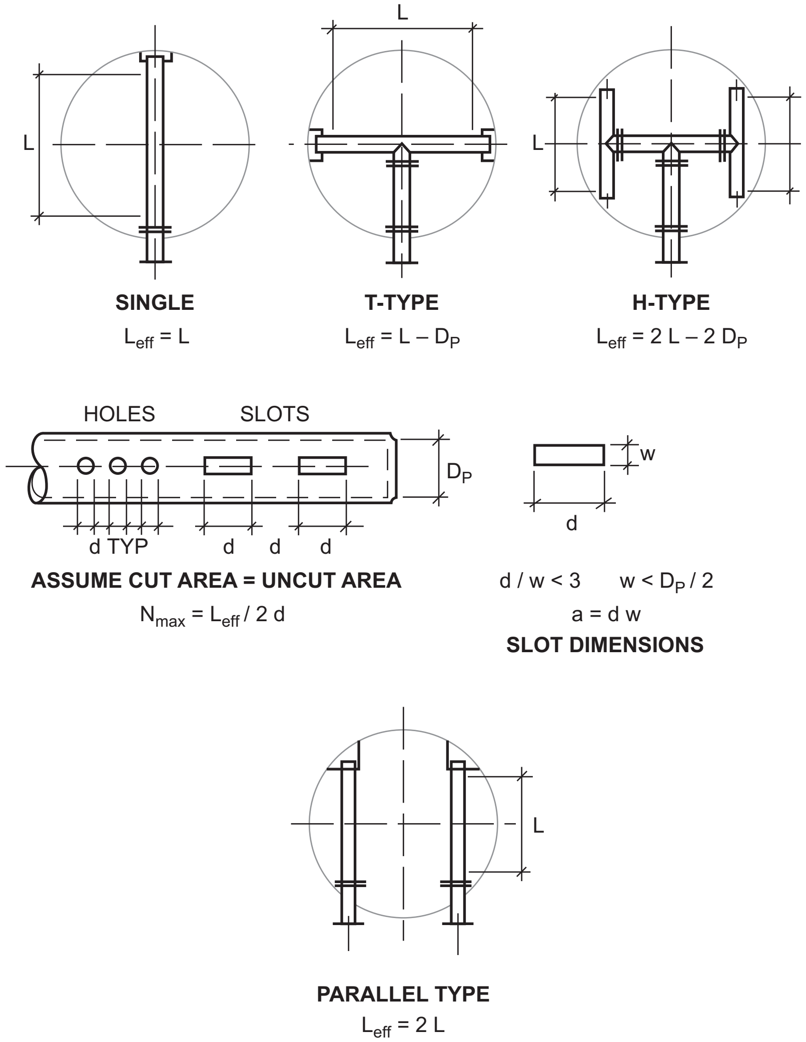


Figure 5-14. Dimensions for piping distributors.

L = Length of pipe or branch where slots or holes can be placed, in
L_{eff} = Total effective length for holes or slots, in
n = Number of branches
N = Number of holes or slots
N_r = Number of holes/slots required
N_{max} = Maximum number of holes/slots that can be placed in the effective length
Q = Flow rate, GPM
R_e = Reynolds number
V = Average inlet velocity, Ft / Sec
V_m = Volumetric 2-phase flow, Ft^3 / Sec
ΔP = Pressure drop, PSI
ρ = Density of liquid, PCF
ρ_m = Density of 2-phase flow, PCF
μ = Viscosity, centepoise

Calculations

Case 1: Liquid Only

- Determine if ΔP is known, assumed or calculated. If assumed, use .25 PSI. It can be calculated as follows;

$$\Delta P = [(4fL_{eff})/(3D_P) - 2][(V^2G)/(4.618g)]$$

- Total area of holes/slots required, A_r

$$A_r = [Q/(38\ C)][G/\Delta P]^{1/2}$$

Note; A_r should be 1-3 times A_P

A_P =

- Assume hole or slot size;

d =

a =

- Determine effective length of pipe available, L_{eff}

L_{eff} = ______

- Determine quantity of holes required, N_r

$$N_r = A_r/a$$

- Determine maximum quantity of holes allowed, N_{max}

$$N_{max} = L_{eff}/2\ d$$

- Compare qty of holes required with the max allowed, N_r to N_{max}

If $N_r \le N_{max}$ the design is OK as is
If $N_r > N_{max}$ then another selection of hole or slot size must be selected

Case 2: 2-Phase Flow

- Area required, A_r

$$A_r = [(1.5\ V_m)/C][\rho m/\Delta P]$$

- Determine quantity of holes required, N_r

$$N_r = A_r/a$$

- Determine maximum quantity of holes allowed, N_{max}

$$N_{max} = L_{eff}/2\ d$$

- Compare qty of holes required with the max allowed, N_r to N_{max}

If $N_r < N_{max}$ the design is OK as is
If $N_r > N_{max}$ then another selection of hole or slot size must be selected

Example

GIVEN;

$G = .523$

$\rho = 62.4\ G = 32.63$ PCF

$Q = 640$ GPM

$L_{eff} = 104''$

$\Delta P = 0.25$ psi

PIPE DATA;

Header Size: 6” Sch 40
Branch Size: None

$A_P = 28.9\ in^2$

$D_P = 6.095'' = .5079'$

CALCULATE;

- Velocity, V, FPS

$$V = (.408\ Q)/D_P^2$$
$$= (.408(640))/6.095^2$$
$$= 7.03\ \text{FPS}$$

- Area of holes required, A_r

$$A_r = [Q/(38\ C)][G/\Delta P]^{1/2}$$
$$= [(640)/(38(.6)][.523/0.25]^{1/2}$$
$$= 40.6\ in^2$$

- Check area proportions;

  $A_r/A_P = 40.6/28.9 = 1.4$

  Between 1 and 3, therefore OK
- Assume hole or slot size;

  $d = 0.5 \text{ in} \quad a = .196 \text{ in}^2$
- Determine quantity of holes required, N_r

  $N_r = A_r/a = 40.6/.196 = 208$
- Determine the max quantity of holes allowed, N_{max}

  $N_{max} = L_{eff}/2\,d = 104/2(.5) = 104$

  No Good!

  TRIAL 2
  Try slots 0.875 in × 2.25 in

  $d = 2.25 \quad a = 1.967 \text{ in}^2$
- Determine quantity of slots required, N_r

  $N_r = A_r/a = 40.6/1.967 = 21$
- Determine max quantity of slots allowed, N_{max}

  $N_{max} = L_{eff}/2\,d = 104/2(2.25) = 23$

  Therefore, design is acceptable.

Useful Conversion Factors

1. Given flow in GPM, find velocity, V, in Ft/Sec

   $V = [GPM(.408)]/D_P{}^2$

2. Given pressure in feet of head, convert to pressure in PSI;

   $P = (H\,G)/2.309$

3. Convert velocity and head;

   $H = V^2/(2\,g)$
   $V = (2\,g\,H)^{1/2}$

4. Given flow in GPM, find velocity in Ft^3/Sec

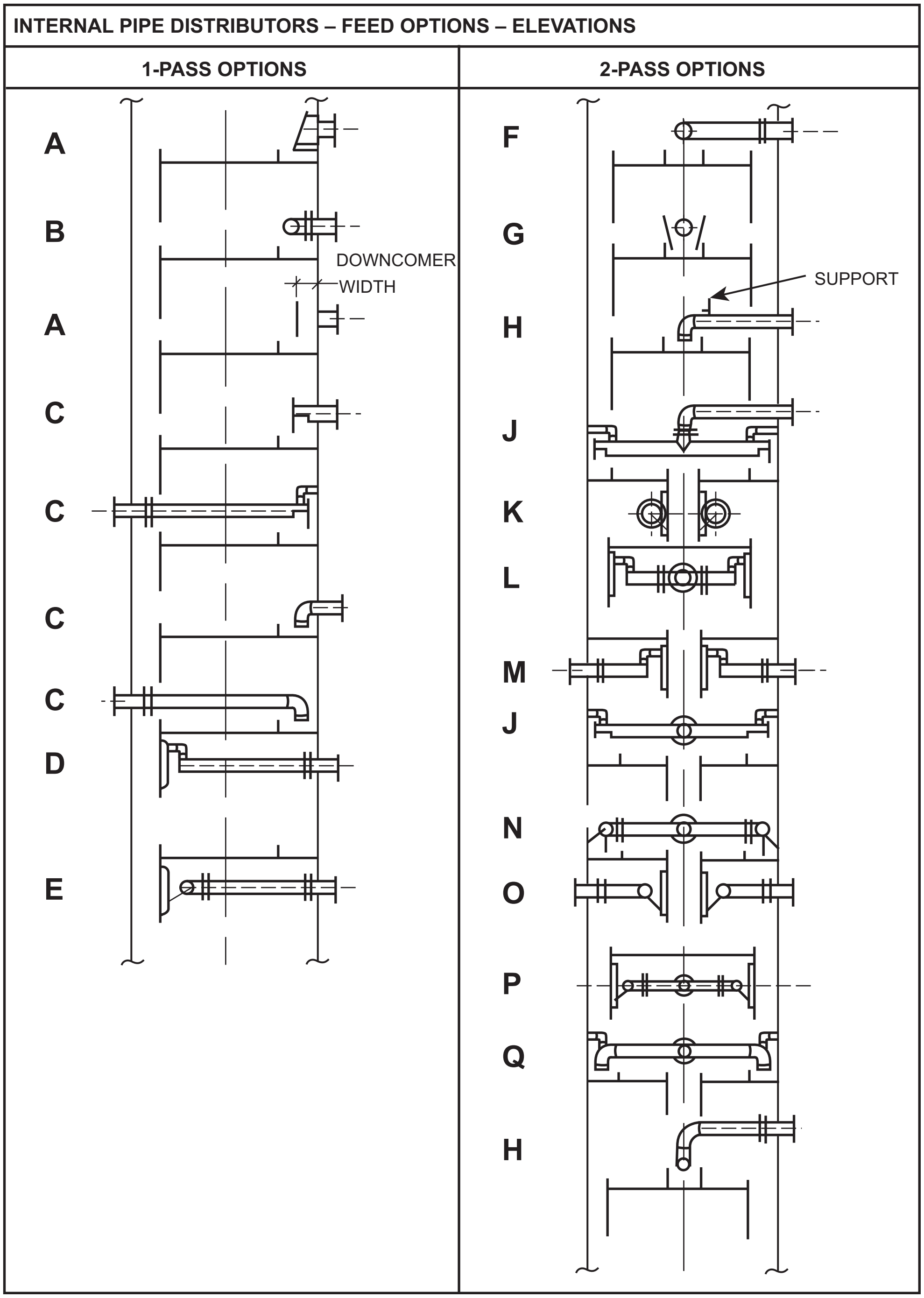
   $V = GPM/449$

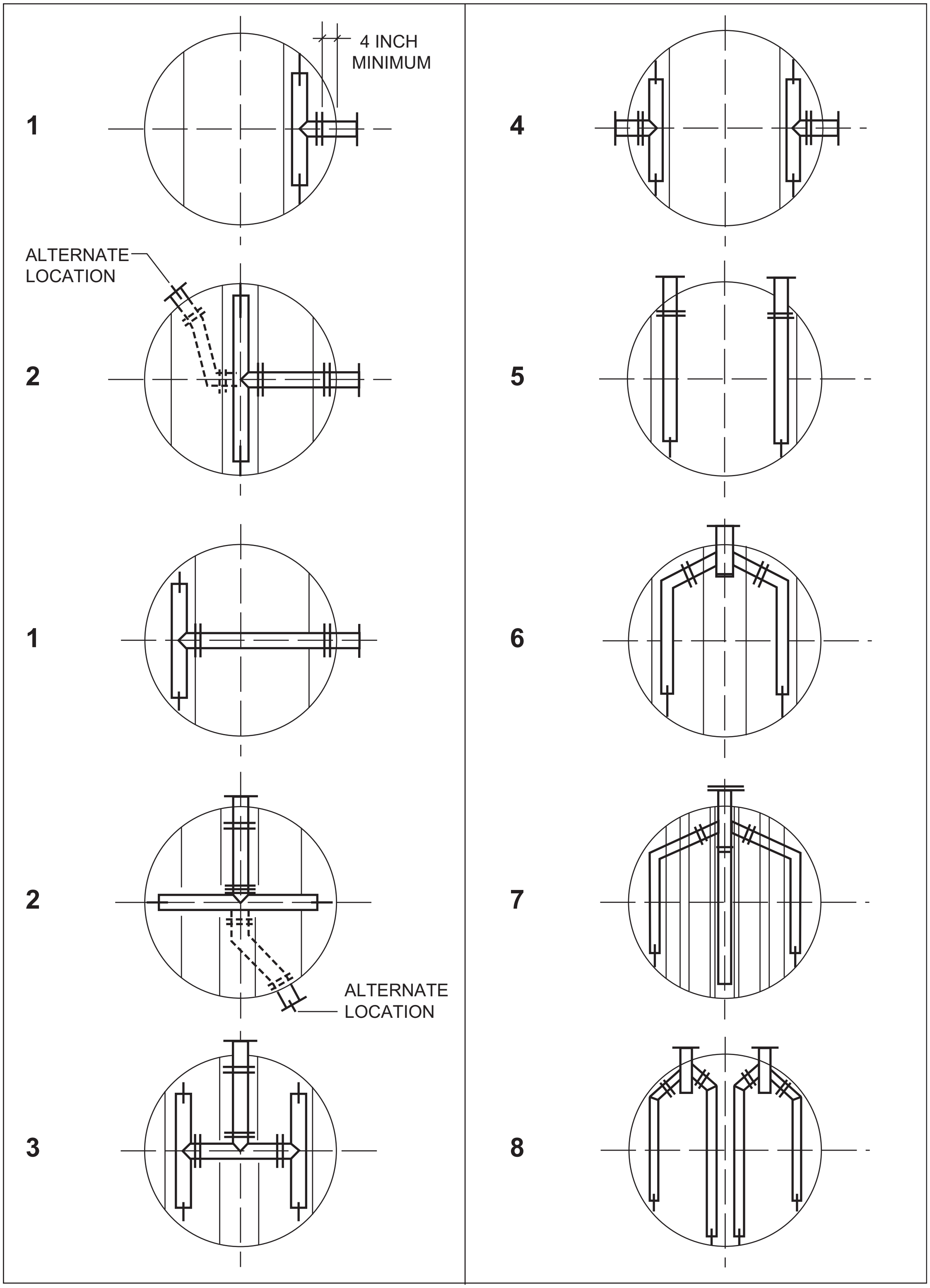
5. GPM equivalents;

   Lbs/Hr/500 G
   Lbs/Min/8.33 G
   Ft^3/Min/7.5
   449 × Ft^3/sec

6. Misc conversions;

   1 gallon = 231 in^3 = 8.345 Lbs H_2O
   1 Ft^3 = 7.482 Gallons
   = 1728 in^3
   = 62.4 Lbs H_2O
   H = .433 PSI/Ft
   H = .433 G for other medium
   1 PSI = 2.309 feet of head
   G = 141.5/(131.5 × API°)

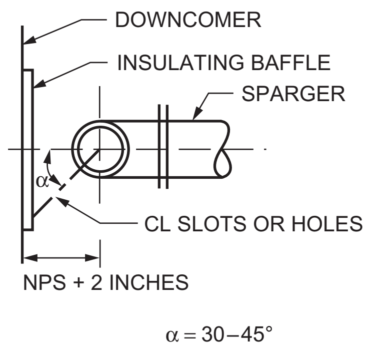

INTERNAL PIPE DISTRIBUTORS – FEED OPTIONS – ELEVATIONS
1-PASS OPTIONS
2-PASS OPTIONS
A
B
A
C
C
C
C
D
E
DOWNCOMER
WIDTH
F
G
H
J
K
L
M
J
N
O
P
Q
H
SUPPORT



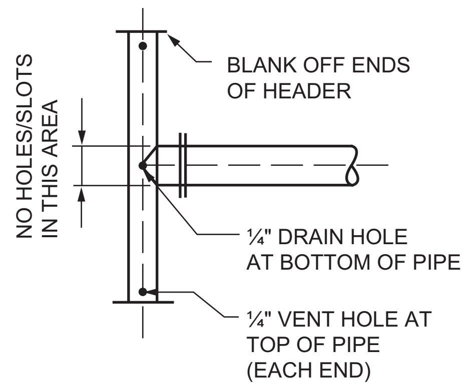
4 INCH
MINIMUM
1
ALTERNATE
LOCATION
2
1
2
ALTERNATE
LOCATION
3
4
5
6
7
8

INTERNAL PIPE DISTRIBUTORS - DESCRIPTIONS			
PLANS		ELEVATIONS	
1	T-TYPE LIQUID FEED FOR 1-PASS TRAY	A	INLET WITH BAFFLE
2	T-TYPE LIQUID FEED FOR 2-PASS TRAY	B	T-TYPE LIQUID FEED/REFLUX FOR 1-PASS TRAY
3	H-TYPE LIQUID/VAPOR FEED FOR 2 OR 4 PASS TRAY	C	SIDE FEED FOR 1-PASS TRAY
4	INDEPENDENT LIQUID/VAPOR FEED FOR 2-PASS TRAY	D	LIQUID/VAPOR FEED FOR 1-PASS TRAY
5	INDEPENDENT DUAL LIQUID/VAPOR FEED FOR 2 OR 4-PASS TRAY	E	T-TYPE LIQUID /VAPOR FEED FOR 1-PASS TRAY
6	COMBINED DUAL LIQUID/VAPOR FEED FOR 2 OR 4 PASS TRAY	F	T-TYPE LIQUID FEED/REFLUX CENTER DISCHARGE FOR 2-PASS TRAY
7	COMBINED TRIPLE LIQUID FEED FOR 4-PASS TRAY	G	SINGLE LINE FEED, CENTER DISCHARGE, WITH BAFFLES FOR 2-PASS TRAY
8	INDEPENDENT DUAL LIQUID/VAPOR FEED FOR 4-PASS TRAY	H	SINGLE LINE, CENTER DISCHARGE, LIQUID, FEED/REFLUX, WITH ELBOW FOR 2-PASS TRAY
		J	T-TYPE, SLOTTED END, LIQUID, FEED/REFLUX, SIDE DISCHARGE FOR 2-PASS TRAY
		K	DUAL, SINGLE LINE, LIQUID/VAPOR FEED AGAINST CENTER DOWNCOMER FOR 2-PASS TRAY
		L	T-TYPE, OPEN END, LIQUID/VAPOR FEED, SIDE DISCHARGE, FOR 2 PASS TRAY
		M	INDEPENDENT, DUAL, OPEN END, LIQUID/VAPOR FEED, CENTER DISCHARGE FOR 2 PASS TRAY
		N	H-TYPE, LIQUID, FEED/REFLUX, SIDE DISCHARGE, FOR 2-PASS TRAY
		O	INDEPENDENT, DUAL, T-TYPE, LIQUID/VAPOR FEED, CENTER DISCHARGE FOR 2-PASS TRAY
		P	H-TYPE, LIQUID/VAPOR FEED, SIDE DISCHARGE FOR 2 PASS TRAY
		Q	T-TYPE, LIQUID FEED/REFLUX, SIDE DISCHARGE WITH ELBOWS FOR 2-PASS TRAY

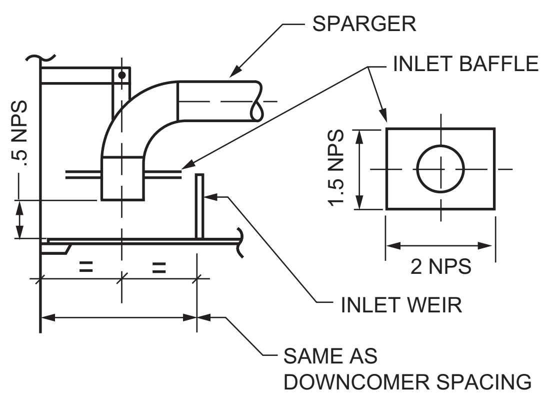
SPARGER DETAILS



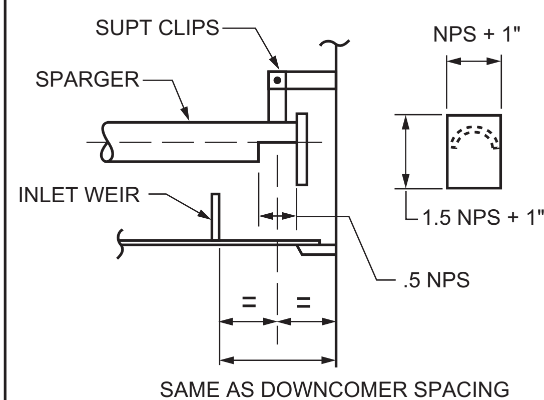
DETAIL 1



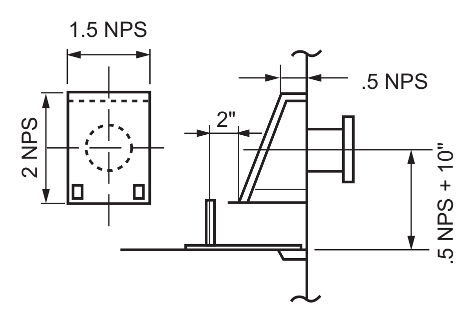
DETAIL 2



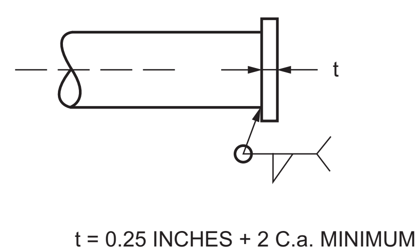
DETAIL 3



DETAIL 4

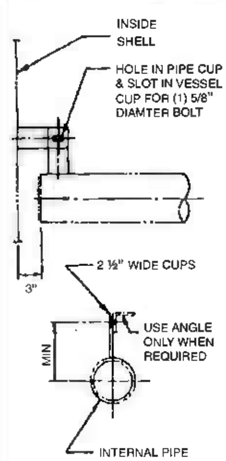


DETAIL 5

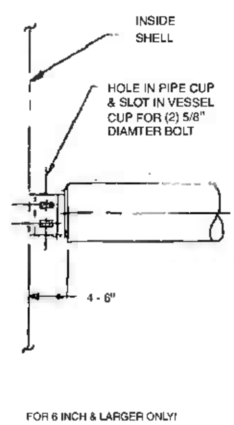


DETAIL 6

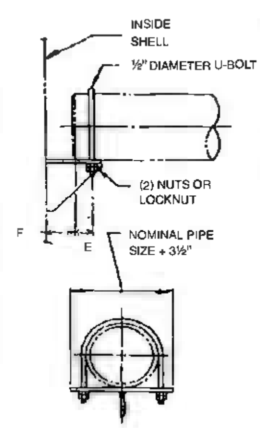
INTERNAL PIPE SUPPORT DESIGN



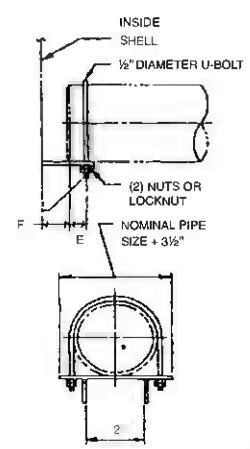
TYPE 1



TYPE 1 ALTERNATE



TYPE 2



TYPE 3

TABLE 1: SUPPORT SELCTION GUIDE

PIPE SIZE	VESSEL DIAMETER 30-48 IN	48-78 IN	78-144 IN	144-240 IN
2 IN	1	1		
3 IN	1	1	1	
4 IN	1	1	1	
6 IN	1	1	1	1
8 IN	1	1	2	2
10 IN		2	2	2
12 IN			2	2
14 IN			2	2
16 IN			2	3
18 IN			3	3
20 IN				3

NOTES

1. All plates are 1/4 inch thick + (2) times C a (3/8 inch minimum)
2. Weld sizes are 1/4 inch + (2) times C.a (3/8 inch minimum)
3. F = 3/16 X pipe length in feet (3 inch minimum) in inches
4. E = 1/8 X pipe length in feet (2 inch minimum) in inches
5. Use Type 1 Alternate as appropriate
6. Table 1 shows recocommendation of pipe support type based on size of pipe and diameter of vessel

Table 5-15
Cross sectional area of pipe, A_p, in^2

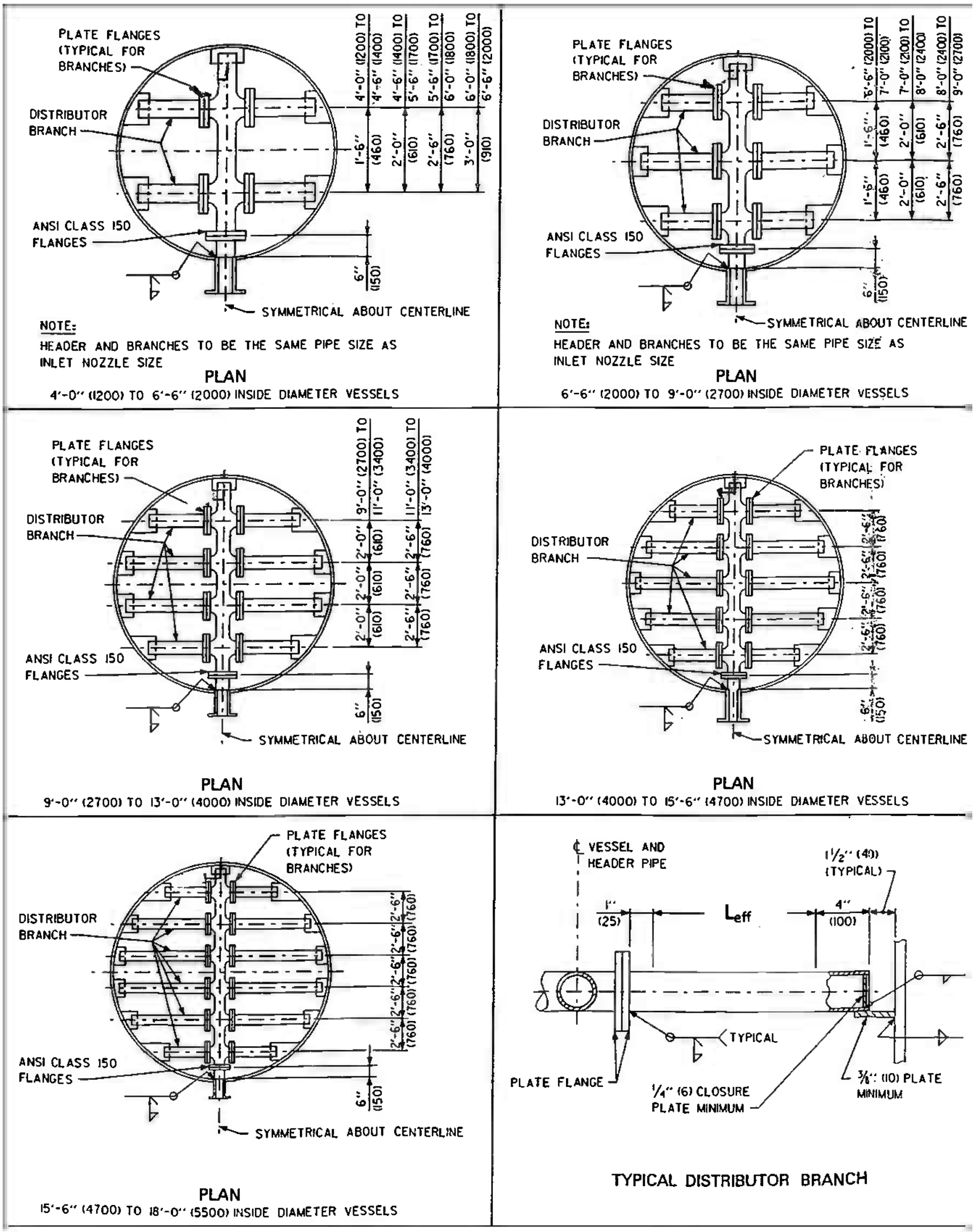
Pipe Size	Schedule			
	10	40	80	160
1	0.945	0.864	0.719	0.522
1.25	1.633	1.496	1.283	1.057
1.5	2.222	2.036	1.767	1.404
2	3.654	3.356	2.953	2.24
2.5	5.45	4.79	4.24	3.55
3	8.35	7.39	6.6	5.41
4	14.25	12.73	11.5	9.28
6	31.7	28.9	26.1	21.1
8	54.5	50	45.7	36.5
10	85.3	78.9	71.8	56.7
12	120.6	111.9	101.6	80.5
14	143.1	135.3	122.7	98.3
16	188.7	176.7	160.9	129
18	240.5	223.7	204.2	163.7
20	298.6	278	252.7	202.7

Table 5-16
Size (ID) of equalizing branches, inches

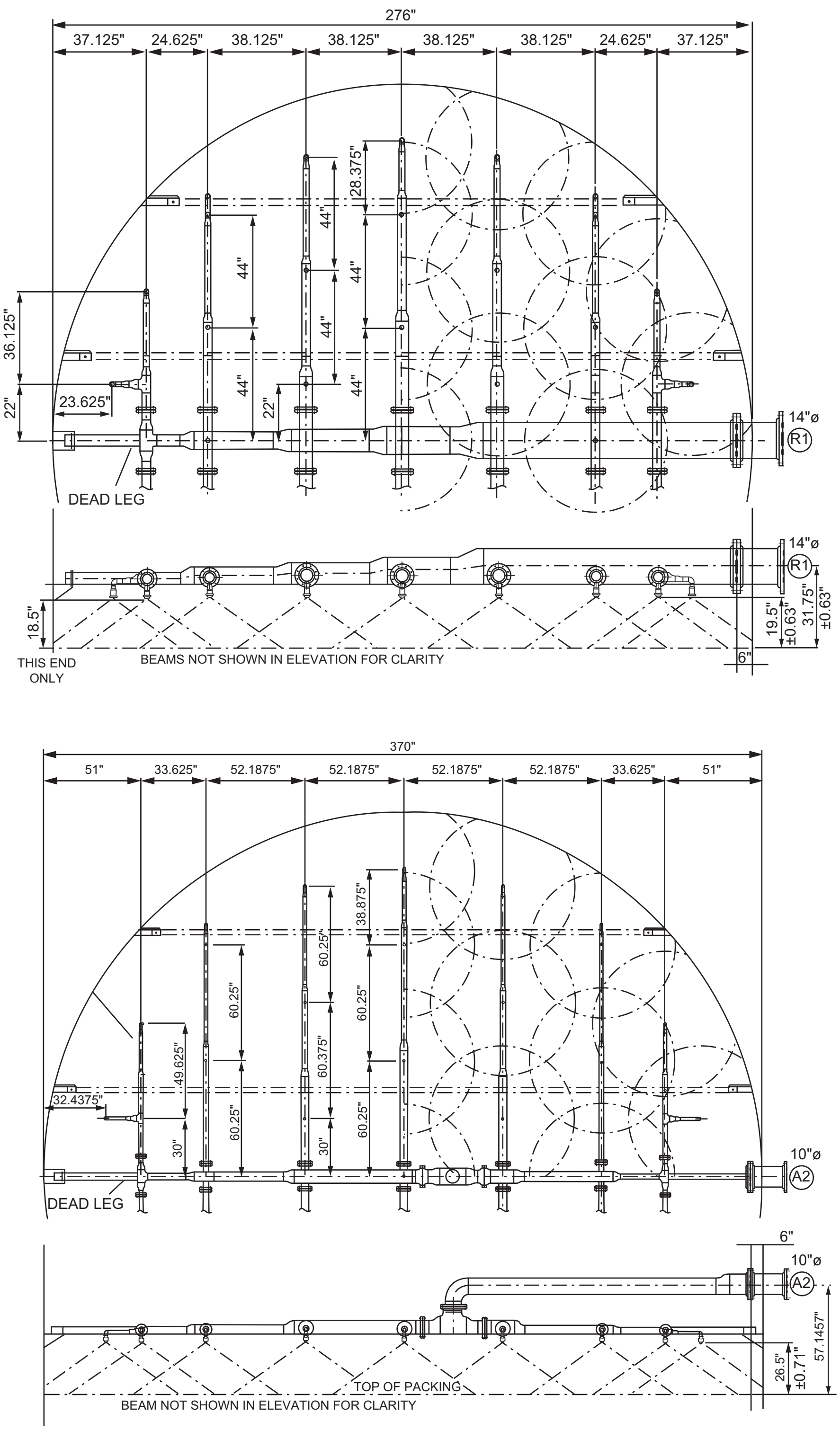
Size of Main Pipe	Area, Ap, in^2 (2)	Quantity of Branches (3)					
		2	4	6	8	10	12
2"	3.356	1.461	1.03				
3"	7.39	2.17	1.53	1.25			
4"	12.73	2.85	2.01	1.64	1.42		
6"	28.9	4.29	3.03	2.48	2.14	1.92	
8"	50	5.64	3.99	3.26	2.82	2.52	
10"	78.9	7.08	5	4.09	3.54	3.17	2.89
12"	111.9	8.44	5.97	4.87	4.22	3.77	3.45
14"	135.3	9.28	6.56	5.36	4.64	4.15	3.79
16"	176.7	10.6	7.5	6.12	5.3	4.74	4.33
18"	223.7	11.93	8.43	6.88	5.97	5.33	4.87
20"	278	13.3	9.4	7.68	7.11	5.95	5.43

Notes:

1. The table lists the exact ID required such that the cross sectional area of the main line and the sum of the cross sectional area of the branches is equal.
2. Assumes that main line is Sch 40.
3. Quantity of branches shown is total. Assume equal quantity for each side.



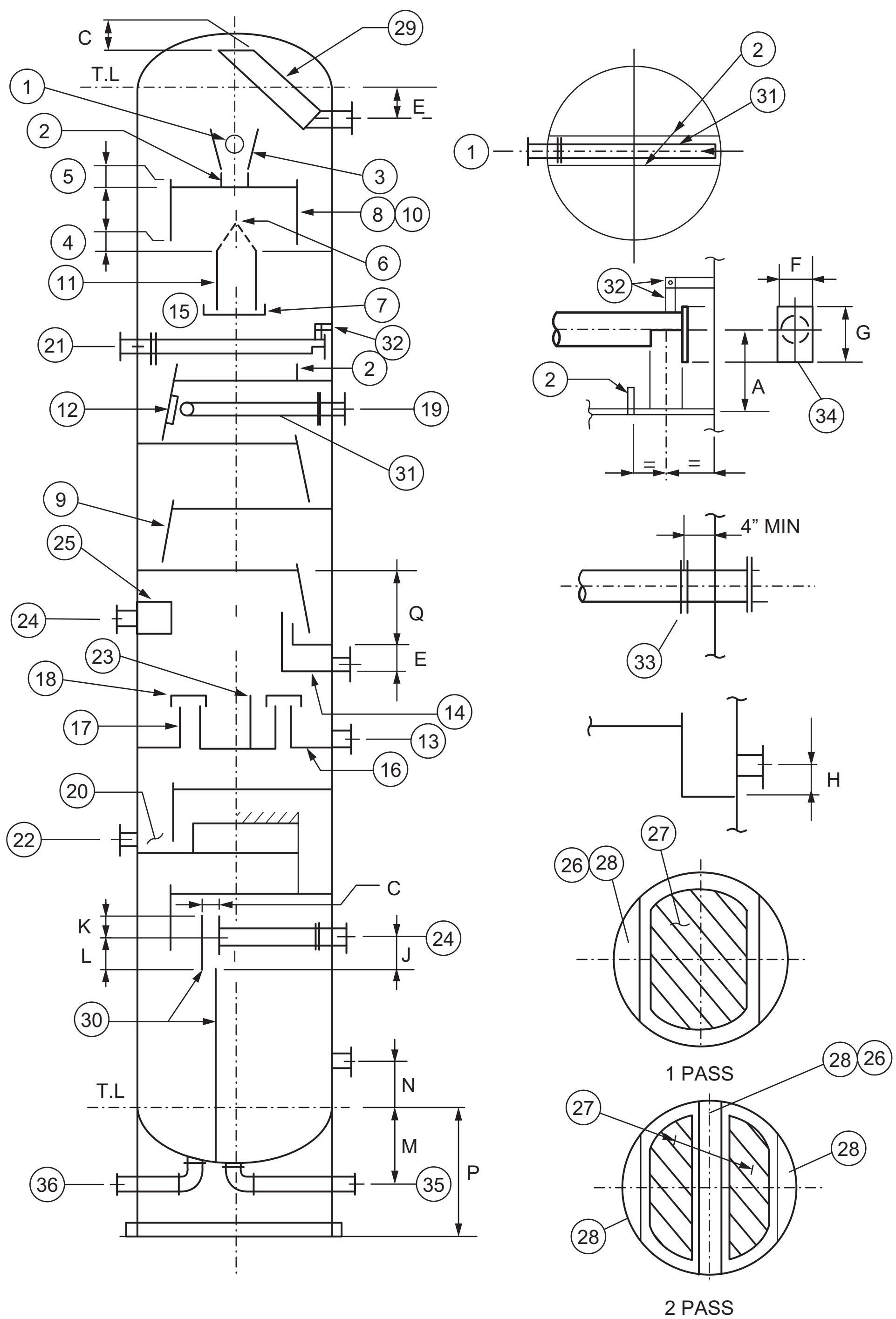
Header types with branches–Examples



SPRAY HEADERS - EXAMPLES

Procedure 5-9: Design of Trays

Typical Tower Dimensions and Nomenclature



Tray Types

1. Valve
2. Bubble Cap
3. Sieve
4. Tunnel
5. Chimney
6. Ripple
7. Shower
8. Shed Rows
9. Disc & Donut
10. Cartridge
11. Jet
12. Kittle
13. Turbogrid
14. Dual Flow
15. Slotted
16. Venturi
17. Cascade
18. Accumulator

Tray Nomenclature

1. Inlet/Feed/Reflux
2. Inlet Weir
3. Inlet Baffle
4. Downcomer Clearance
5. Weir Height
6. Anti-Jump Baffle
7. Seal Pan
8. Straight Downcomer
9. Sloped Downcomer
10. Side Downcomer
11. Center Downcomer
12. Insulating Baffle
13. Draw (Off) Nozzle
14. Draw (Off) Pan
15. Pass Transition
16. Total Draw Chimney Tray
17. Chimney
18. Chimney Hat
19. Return Nozzle
20. Sump
21. Intermediate Feed Nozzle
22. Partial Drawoff Nozzle
23. Partition
24. Vapor Inlet
25. Vapor Inlet Baffle
26. Feed Area
27. Active Tray Area
28. Dowcomer Area (Inactive Tray Area)
29. Vapor Outlet
30. Baffle
31. Internal Pipe
32. Support Clips
33. Internal Flanges
34. End Plate
35. Product Outlet Nozzle
36. Reboiler Draw Nozzle

Dimensions

A = Short = .5 NPS + 4 in

Long = .5 NPS + 10 in

B = Same as middle downcomer

C = .5 NPS

D = Same as weir height

E = NPS + 6 in

F = NPS + 1 in

G = 1.5 NPS + 1 in

H = .5 NPS + 1.5 in

J = 1.5 NPS

K = NPS

L = 2 NPS

M = Minimum with 2:1 S.E. head =

$.25\ D + t_H + 1.5\ NPS + 2\ in$

Minimum with hemi head =

$M = .5\ D + t_H + 1.5\ NPS + 2\ in$

N = Minimum distance from tangent line to nozzle centerline. See Table 5.19

P = Minimum skirt height for 2:1 S.E. Head. See Table 5-20

Q = Minimum seal pan depth = tray spacing + 6 in

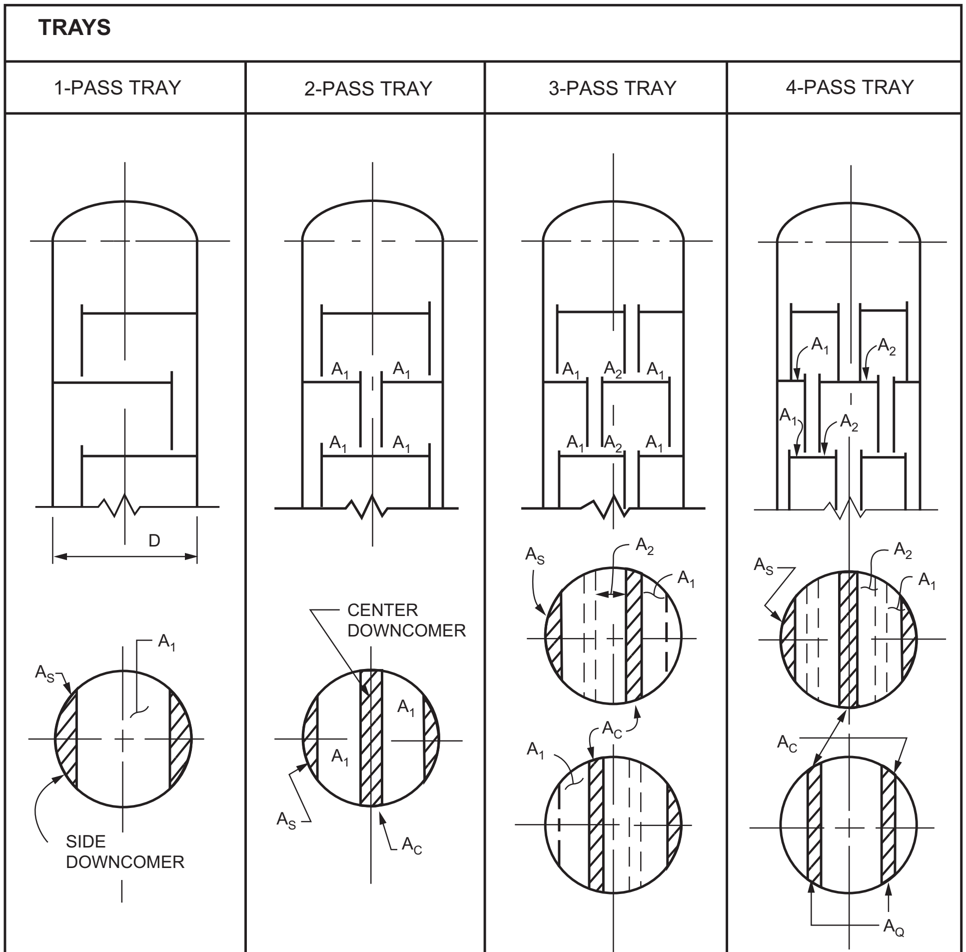
R = Downcomer width

Table 5-18
Minimum Distance From Tangent Line, "N"

Noz Size	N	Noz Size	N
<2 in	5	12 in	16
3 in	7	14 in	18
4 in	8	16 in	20
6 in	10	18 in	24
8 in	12	20 in	27
10 in	14	24 in	30

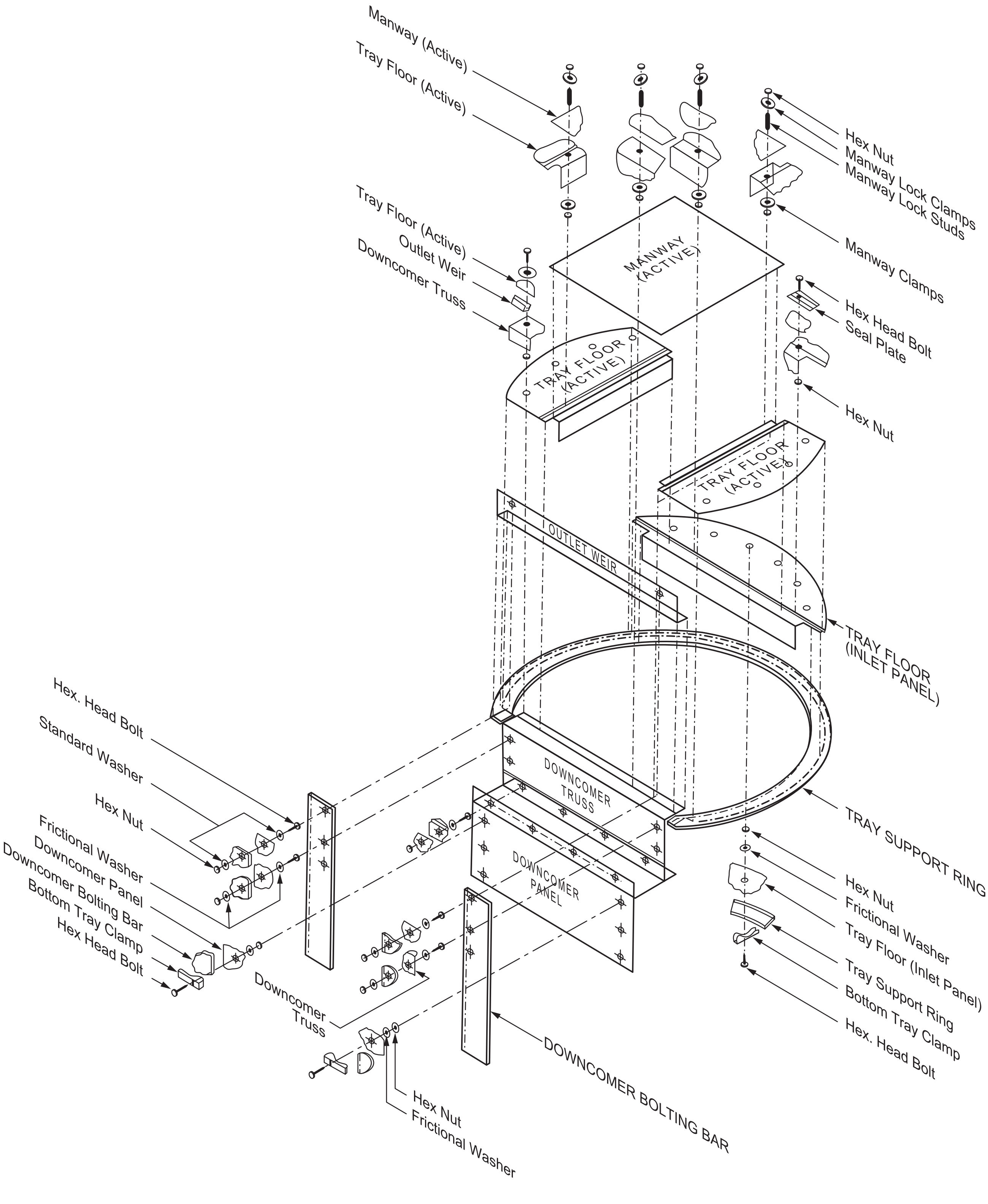
Table 5-19
Minimum Skirt Height, "P"

Dia	P	Dia	P
<24 in	2 ft – 6 in	114–120	5 ft – 6 in
30–36	3 ft – 0 in	126–132	6 ft – 0 in
48–54	3 ft – 6 in	138–144	6 ft – 6 in
54–60	3 ft – 6 in	150–156	7 ft – 0 in
66–72	4 ft – 0 in	162–168	8 ft – 0 in
78–84	4 ft – 6 in	174–180	8 ft – 6 in
90–106	5 ft – 0 in		

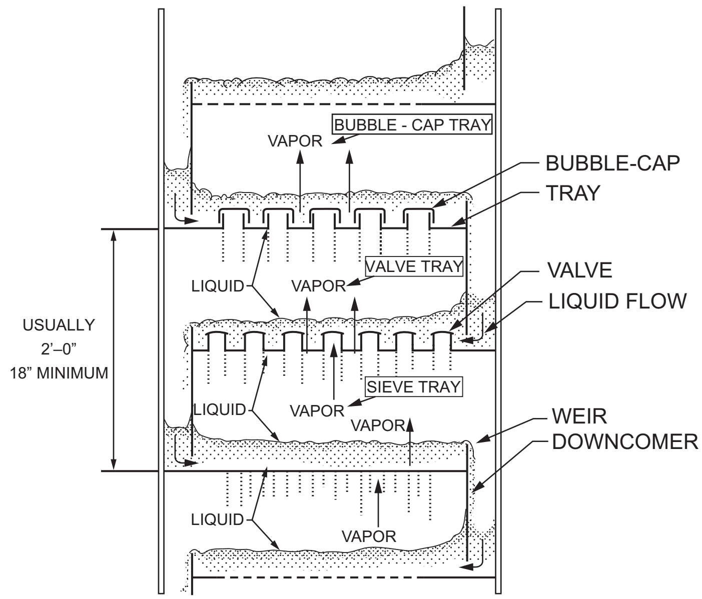


TRAY AND DOWNCOMER AREAS

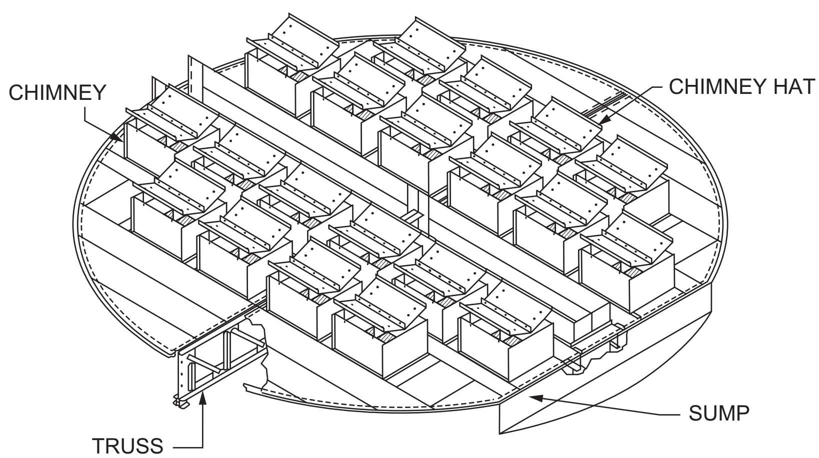
PASSES	TOTAL DC %	A_1	A_2	A_C	A_Q	A_S	MINIMUM DIAMETER	TOTAL WEIR LENGTH
1	8.08	$.6582\ D^2$				$.1268\ D^2$	30 IN	.6834 D
	12.25	$.5927\ D^2$				$.1923\ D^2$	30 IN	.7684 D
	16	$.5337\ D^2$				$.2512\ D^2$	30 IN	.8244 D
2	13.6	$.5715\ D^2$		$.1068\ D^2$		$.1068\ D^2$	48 IN	1.3 D
	17.9	$.5036\ D^2$		$.1407\ D^2$		$.1407\ D^2$	54 IN	1.408 D
	9.21	$.6408\ D^2$		$.0723\ D^2$		$.0723\ D^2$	48 IN	1.156 D
3	17.1	$.2832\ D^2$	$.2389\ D^2$	$.1949\ D^2$		$.0735\ D^2$	78 IN	1.5524 D
	23.65	$.2382\ D^2$	$.1754\ D^2$	$.2646\ D^2$		$.1068\ D^2$	96 IN	1.634 D
4	17.3	$.1712\ D^2$	$.3424\ D^2$	$.0906\ D^2$	$.1359\ D^2$	$.0435\ D^2$	108 IN	3 D
	21.95	$.1561\ D^2$	$.2845\ D^2$	$.1113\ D^2$	$.1724\ D^2$	$.0611\ D^2$	126 IN	3.097 D



TYPICAL TRAY ASSEMBLY
ONE PASS TRAY SHOWN



TYPES OF TRAYS



CHIMNEY/ACCUMULATOR TRAYS

Design of Tray Plates

Stress and Deflection

Case 1: Perforated Plate

Reference: Roark, 5th Edition, Table 26, Case 1A

- Rectangular plate
- Uniformly loaded
- All edges simply supported

DATA

α, β, γ = Coefficients from Table 5-21
E = Modulus of elasticity at design temperature, PSI
t = Corroded thickness of tray, in
C_a = Corrosion allowance, in
n = Hole efficiency
δ = Deflection at center, in
p = Uniform load, PSI
σ = Bending stress, PSI

FORMULAS

- Stress, σ

$$\sigma = (\beta \, p \, b^4)/(n \, t^2)$$

Table 5-20
Coefficients

a/b	β	α	γ
1.0	.2874	.044	.420
1.2	.3762	.0616	.455
1.4	.4530	.0770	.478
1.6	.5172	.0906	.491
1.8	.5688	.1017	.499
2.0	.6102	.1110	.503
3.0	.7134	.1335	.505
4.0	.7410	.1400	.502
5.0	.7476	.1417	.501
∞	.7500	.1421	.500

- Deflection, δ

  $\delta = (\alpha \, p \, b^4)/(E \, n \, t^3)$

- Efficiency of holes for perforated plate, n

  $n = 1 - (.25 \, \pi \, d^2)/(e \, C)$

DIMENSIONAL DATA

a =

b =

t =

a/b =

β =

α =

p =

d =

e =

C =

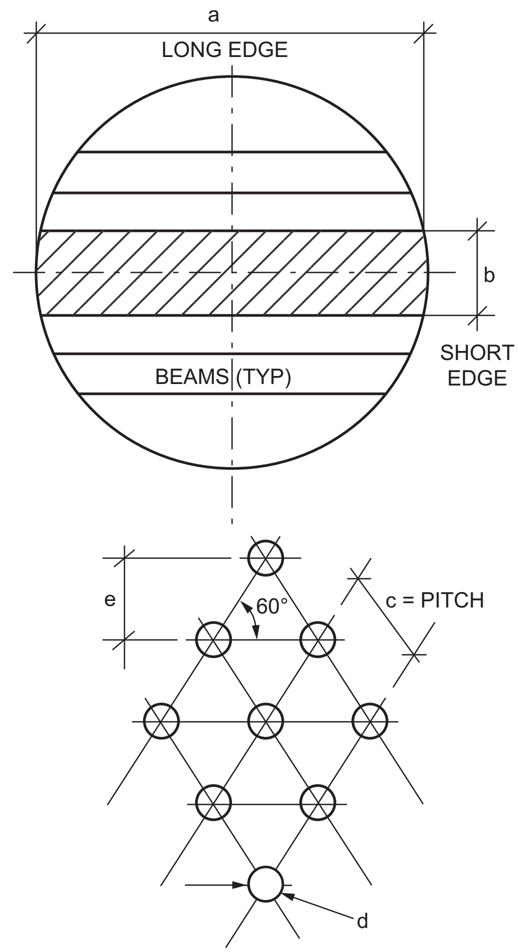
E =

Tray Design

Case 2: Standard tray plates

LOADS

A. DL = Dead load; weight of trays and beams



B. LL = Live load; dynamic load, ΔP

C. L_L = Liquid load; weight of liquid supported

- Total area, A_T

  $A_T = \pi D^2/4 = .7854 D^2$

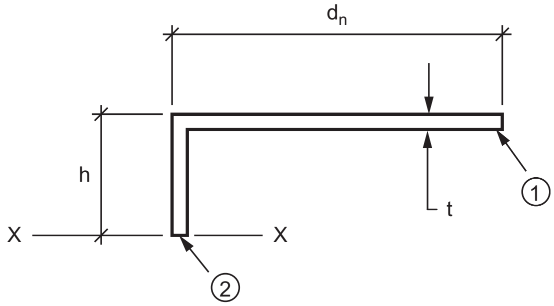
- Total load, P_T

  $P_T = DL + LL + L_L$

- Uniform load, p

  $p = P_T/A_T$

PROPERTIES OF SECTIONS



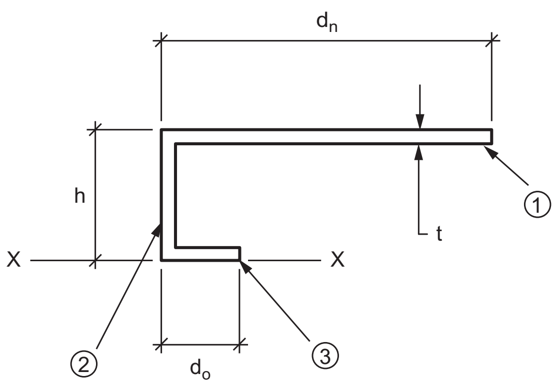
TYPE 1

PART	A	Y	AY	AY^2	I
1					
2					
Σ		-----			

$$C = \Sigma AY/\Sigma A$$

$$I = \Sigma AY^2 + \Sigma I - C\,\Sigma AY$$

$$Z = I/C$$



TYPE 2

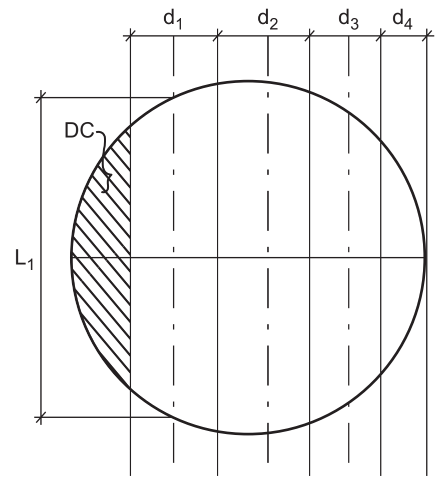
PART	A	Y	AY	AY^2	I
1					
2					
3					
Σ		-----			

$$C = \Sigma AY/\Sigma A$$

$$I = \Sigma AY^2 + \Sigma I - C\,\Sigma AY$$

$$Z = I/C$$

CALCULATIONS



DIMENSIONS OF TRAY PANELS

- Determine panel area, A_P

  $A_P = L_n\, d_n$
- Load on panel, F_P

  $F_P = A_P\, p$

Uniform Load

- Uniform load on beam, w

  $w = F_P/L_n$
- Moment, M

  $M = \left(w\ L_n^2\right)/8$
- Bending stress, f_b

  $f_b = M/Z$
- Deflection, δ

  $\delta = \left(5\,w\,L_n^4\right)/\left(384\,E\,I\right)$

Concentrated Load

- Moment, M

  $M = \left(P\,L_n\right)/4$
- Bending stress, f_b

  $f_b = M/Z$
- Deflection, δ

  $\delta = \left(P\,L_n^3\right)/\left(48\,E\,I\right)$

NOTES

1. Design Loads: Trays, pans, drawoff boxes, or similar intemals, shall be designed using a corroded thickness, to support their own weight plus the following live loads;
   a. Fractionation trays: Design live load shall be the greater of 20 PSF (98 Kg/Sq meter) or the weight of water 2" (50 mm) over the highest weir setting.
   b. Areas under downcomers: Design live load shall be the greater of 64 PSF (314 Kg / Sq meter) or a head of water one half the height of the downcomer.
   c. Pans (accumulator and drawoff pans): Design live load shall be the greater of 1 PSI (700 Kg/ Sq meter) or the weight of water at the maximum operating level of the pan.
   d. Baffles: With no operating liquid level shall use a design live load of 1 PSI (700 Kg/Sq meter) on the projected horizontal area or actual impulse force, whichever is greater.
2. Maintenance Loads: Tray support members (all beams, support clips, etc.) shall be designed for a concentrated load of 300 Lbs (135 Kg) at any point on the installed assembly. The design shall be based on the corroded thicknesses and an allowable stress of .9 Fy. For maintenance loads, stresses in the tray plates need not be considered.
3. Uplift resistance: Typical design of uplift for trays and packing is .25 PSI (17 Millibar). For services where excessive uplift can occur, a higher uplift resistance may be used. Examples of higher uplift factors are as follows;
   a. Crude and FCC side strippers: 1 PSI (70 millibar)
   b. Vacuum tower stripping trays, overflash collector, wash trays or beds: 2 PSI (140 Millibar)
   c. Collectiors and packed beds above the wash section of a vacuum Tower: 1 PSI (70 Millibar)
4. Failure sequence: Tray assemblies, whenever possible shall be designed so that failure will occur in the following order;
   a. Tray manways
   b. Tray deck active areas
   c. Minor beams and downcomers
   d. Major beams (defined as beams 10ft (3 Meters) or longer or beams which extend across a vessel without interruption, regardless of length).
5. Allowable stresses: Allowable unit stress shall be based on yield strength, Fy, at design temperature with AISC factors for tension, bearing, shear, etc.
6. Allowable Deflection: The calculated combined deflection of trays and support beams due to operating loads shall not exceed the lesser of 1/900 of the vessel diameter (in inches) or 3/16" (5 mm)

RECOMMENDED TRAY SPACING & MANWAY SIZE

COLUMN DIA (FT)	TRAY SPACING (IN)	MANWAY SPACING	MIN MANWAY SIZE
2.5 to 16	24	Every 10 trays	18 in
16 to 24	30	Every 8 trays	20 in
24 to 32	36	Every 6 trays	24 in
32 and larger	42	Every 4 trays	30 in

Procedure 5-10: Flow Over Weirs

Notation

b = width, ft
H = static head of liquid, ft
Q = discharge rate, cu ft/sec
V = velocity of approach, ft/sec
H′ = head correction per Table 5-22

Table 5-21
Head correction for velocity of approach

V	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
H’	0.002	0.005	0.01	0.015	0.023	0.03	0.04	0.05	0.062
V	2.2	2.4	2.6	2.8	3.0	3.2	3.4	3.6	3.8
H’	0.075	0.089	0.105	0.122	0.14	0.15	0.179	0.201	0.213

Calculations

Discharge, Q

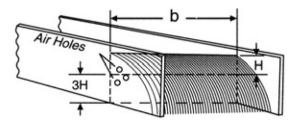
- For a full-length weir (Case 1)
  Q = 3.33b(1.5 H)
- For a contracted weir (Case 2)
  Q = 3.33b(1.5 H)
- For a V-notch weir (Case 3)
  Q = 6.33 H
- For a Cippoletti weir (Case 4)
  Q = 3.367b(1.5 H)

Notes

1. Assumes troughs are level

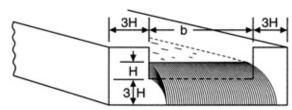
Case 1: Full-Width Weir

V = 1–2 ft/sec at 4 H upstream



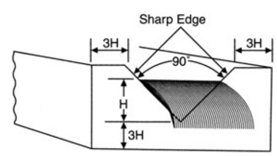
Case 2: Contracted Weir

V = 1–2 ft/sec at 3 H upstream



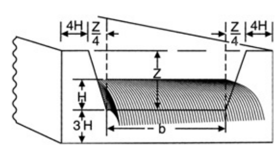
Case 3: V-Notch Weir

V = .5 ft/sec at 5 H upstream



Case 4: Cippoletti Weir

V = 1–2 ft/sec at 4 H upstream



Procedure 5-11: Design of Demisters

Demister pads or mist eliminators, are important internals in process vessels. Anytime there is a continuous two phase flow, vapor and liquid, there is the possibility for liquid entrainment. If it is desirable to separate the liquid and vapor, or prevent liquid carryover in the vapor stream, the velocity must be kept sufficiently low to allow the liquid droplets to fall out of the vapor stream. Demister pads are effective entrainment separators which allow operation at vapor velocities that would otherwise be excessive.

"Vapor disengagement" can be accomplished without a demister, but the vessel must be made much larger to accomplish the same degree of separation. This is known as gravity separation.

Demister pads are used in a wide variety of process vessels. Good performance can be achieved at velocities of 30% to 110% of optimum.

Demister pads can be used in vertical or horizontal vessels. The mesh pad orientation can be either vertical or horizontal in a vertical or horizontal vessel.

A wire mesh demister consists of a knitted wire mat mounted on a light weight support grid. The mesh is made by weaving small diameter metal wire into a mesh. Pads may be made out of metal or plastics. Metal type include carbon steel, stainless steel, monel and others. Plastic materials include PTFE, HDPE and PP.

Pressure drop is usually extremely low, in the range of 1 inch of water column maximum. The pressure drop is normally so low that it is ignored in design.

Vapor disengagement is dependent on the velocity of the stream. The most common equation for determining the allowable vapor velocity for two phase flows is as follows;

$$V_a = K\left[(\rho_L - \rho_V)/\rho_V\right]^{1/2}$$

Where;

V_a = Allowable vapor velocity, Ft/Sec
K = Constant or coefficient
ρ_L = Density of liquid, PCF
ρ_V = Density of vapor, PCF

Notes

1. The mesh pad shall be in one piece or sectional as required for insertion through the vessel manway. Each pad section shall be removable through the vessel manhole.
2. Layers shall not be more than 6 inches thick
3. Where more than one layer is required, all joints shall be staggered.
4. One piece circular pads may be spiral in form.
5. Each section of mesh shall include a grid on one or both sides. The grid, wire and/or fastening device shall be the same material as the mesh pad. Top and bottom grids shall be constructed of ¼ inch diameter rods welded to ⅛ inch by 1 inch bars. The maximum gap between mesh pad and grids shall be ½ inch for diameters less than 84 inches and 1 inch for larger diameters.
6. Estimate grid weights as 3 PSF.
7. Tie down wires shall be 1/16 inch diameter wire thru ¼ inch diameter holes on 4 inch centers. Tie wires should not be pushed through the pad as this could cause leak paths.
8. Pressure drop across mist eliminator shall be assumed as 0.2 PSI for support design.
9. Unless otherwise specified, support rings shall be 2 inches wide for pads 84 inches in diameter and less, and 3 inches wide for larger diameters.
10. The mist eliminator shall be a tight fit against the shell or enclosure as well as between sections. Mesh pads shall be ⅜ inches oversize all around for a force fit against the enclosure. No gaps are allowed.
11. Vessel fabricator shall provide the mesh and grids as well as any stiffeners required.
12. The Mist eliminator shall be specified as either top or bottom removable.
13. "K" is a constant, or coefficient that is a function of the efficiency of the separation. K-values range from .04 to .43, but are generally in the .15 to .35 range. The K-value is affected by pressure, type of pad, size of wire, type of vessel and disengaging height. For pressure applications, the K-value should be decreased by .01 for every 100 PSI of pressure after 100 PSI.
14. Wire diameter for the mesh pad itself vary from 0.003 to 0.016 inches.
15. The free volume of demisters range from 92 to 99.4%
16. The density of the pads ranges from 3 to 33 PCF
17. The NSA (nominal surface area) is the surface area to volume ratio, Ft^2 / Ft^3 and is an important indicator of performance. Ratios range from 50 to 600.
18. To obtain higher separation efficiencies, the wire diameter is decreased and the density and thickness are increased.
19. Typical width of sections for multi-section pads is 12 inches (300 mm) but may vary by manufacturer.

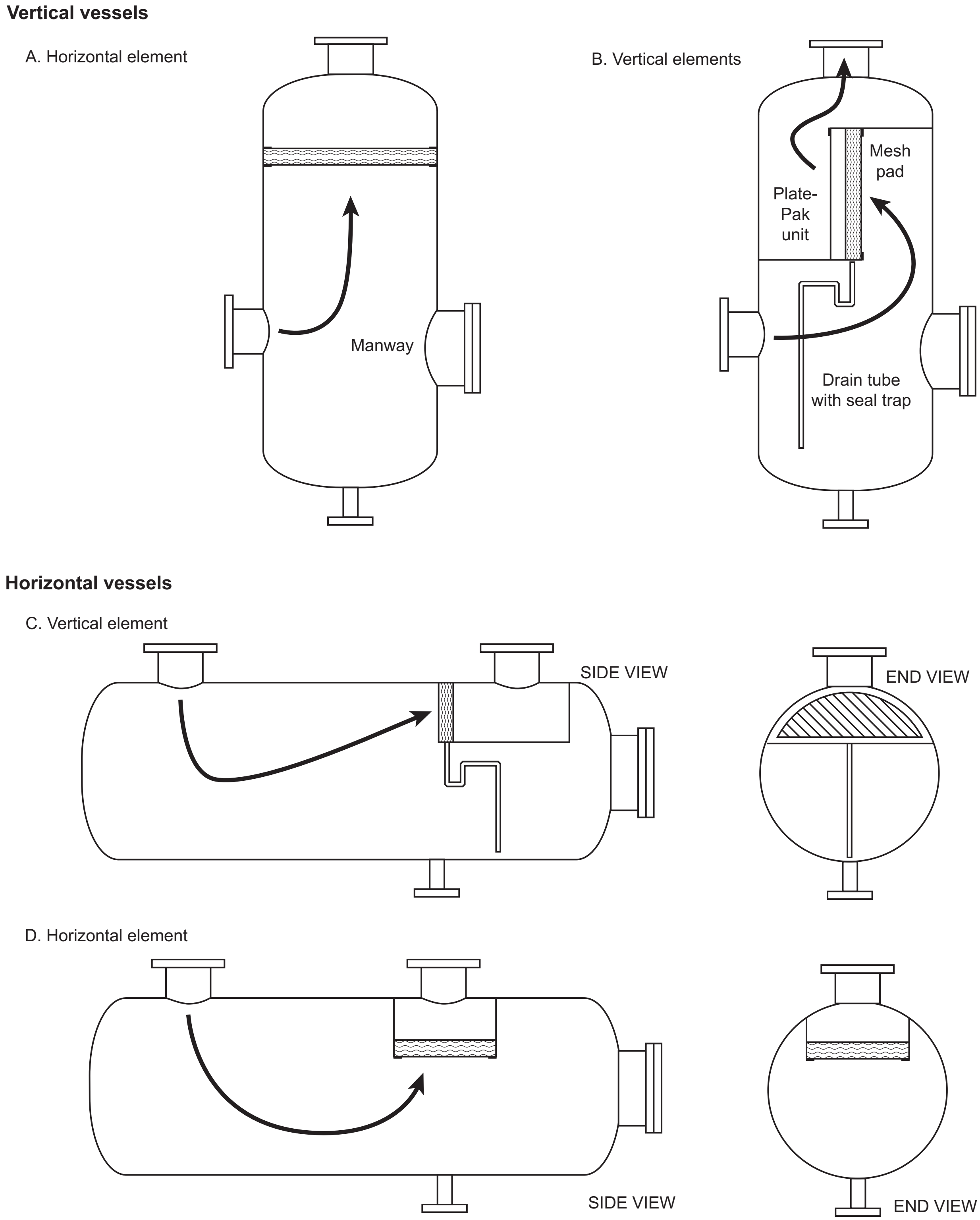


Figure 5-15. Typical demister configurations in vessels.

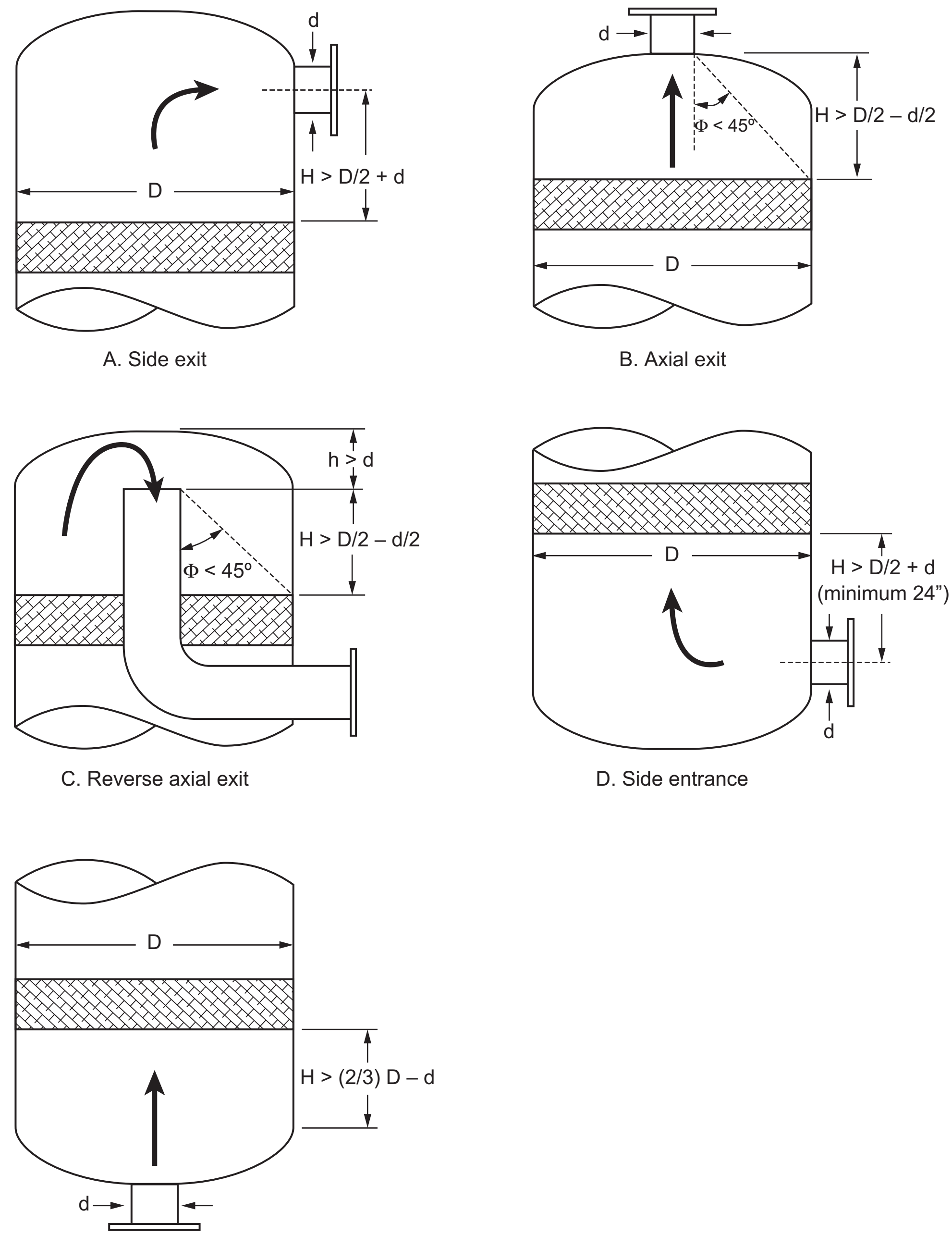


Figure 5-16. Guidelines for maintaining even flow distribution in vessels with axial flow.

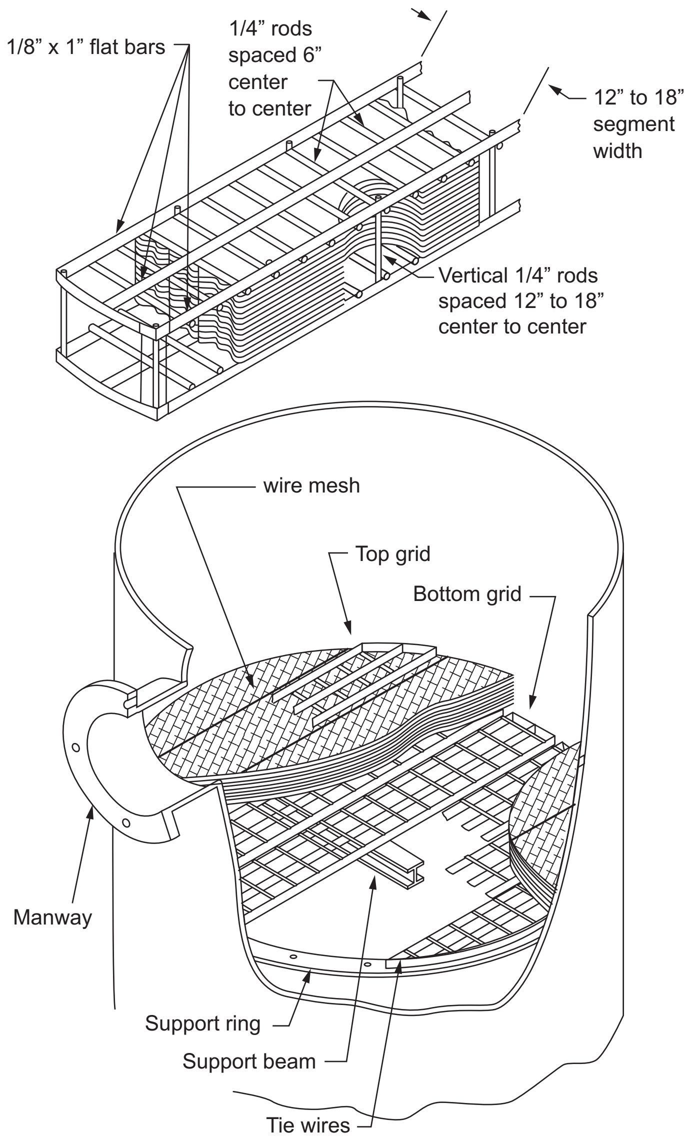


Figure 5-17. Typical mesh pad construction and installation.

Table 5-22
Properties of demister pads

Type	NSA (1) Ft^2 / Ft^3	Density, Pcf	Wire Dia, in	% Free Area	York Style No.	Remarks
A	48	5	0.011	99	931	High Throughput
B	65	7	0.011	98.6	531	Economy, Performance
C	85	9	0.011	98.2	431	Standard - Good All Around
D	110	10.8	0.011	97.7	421	Heavy Duty
E	140	8	0.006	98.4	326	Super High Efficiency - Fine Mist
F	163		0.006	94	371	Liquid - Liquid Coalescer
G	450	20	0.0045	95.9		
H	600	27	0.0045	94.5		

Notes:

1. NSA = Nominal Surface Area

Table 5-23
Values of K (Note 13)

Based on Pressure

Pressure (PSIa)		Vacuum (In Hg)	
Pressure (PSIa)	**K**	**Vacuum (In Hg)**	**K**
15	0.35	30	0.35
50	0.34	20	0.32
100	0.32	10	0.28
200	0.31	5	0.23
300	0.3	1	0.17
500	0.28	<1	0.17
1000	0.27		
>1000	0.27		

Based on Disengaging Height

Disengaging Height, in	K	Disengaging Height, in	K
3	0.12	9	0.32
4	0.15	10	0.35
5	0.19	11	0.38
6	0.22	12	0.4
7	0.25	13	0.42
8	0.29	14	0.43

Based on Application

Vertical Vessels		Horizontal Vessels	
Type	**K**	**Type**	**K**
General	0.35	General	0.35
Compressor Suction Drums	0.25	Steam Drums	0.25
Steam Drums	0.15		

Procedure 5-12: Design of Baffles [10]

Baffles are frequently used in pressure vessels, either vertical or horizontal, to divide the interior volume into different compartments. These compartments may be used to segregate liquids or provide overflow weirs for the separation of liquids. Baffles may be stiffened or unstiffened. When welded across the entire cross section of the vessel, they must be checked that they are not unduly restricting the diametral expansion of the vessel. If the unrestrained radial expansion of the vessel exceeds that of the baffle by more than $^1/_{16}$ in. (⅛ in. on the diameter), then a "flexible" type of connection between the vessel shell and the baffle should be utilized. Various flexible attachment designs are shown within the procedure.

Baffles should always be designed in the corroded condition. It is typical for welded baffles to be designed with a full corrosion allowance on both sides. If the baffle is bolted in, then one-half the full corrosion allowance may be applied to each side, the logic being that a bolted baffle is removable and therefore replaceable.

The majority of baffles are flat and as a result are very inefficient from a strength standpoint. Deflection is the governing case for flat plates loaded on one side. The preference is to have unstiffened baffles, and they should always be the first choice. This will be acceptable for small baffles. However, for larger baffles, as the baffle thickness becomes excessive, stiffeners offer a more economical design. Therefore stiffeners are frequently used to stiffen the baffle to prevent the thickness of the baffle from becoming excessive. The number, size, and spacing of stiffeners are dependent on the baffle thickness selected. There is a continual trade-off between baffle thickness and stiffener parameters.

The design of a baffle with stiffeners is an iterative process. The procedure for the design of the stiffeners is first to divide the baffle into "panel" sections that are rigid enough to withstand the pressure applied on one side. Each individual panel is checked as a flat plate of the dimensions of the panel. The stiffeners are assumed to be strong enough to provide the necessary edge support for the panel.

The stiffeners themselves are designed next. A section of the baffle is assumed as acting with the stiffener and as contributing to the overall stiffness. This combined section is known as the *composite* stiffener. The composite section is checked for stress and deflection. Both vertical and horizontal stiffeners can be added as required.

If required, an alternate design is assumed based on a thicker or thinner baffle and checked until a satisfactory design is found. There is no "right" answer; however, it should be noted that the thinner the baffle, the greater the number of stiffeners. The lightest overall weight is probably the "best" design but may not be the least expensive due to the welding costs in attaching the stiffeners.

One alternative to a flat baffle with stiffeners is to go to a curved baffle. A curved baffle works best as a vertical baffle in a vertical vessel. The curved baffle takes pressure from either side wall. If the pressure is on the concave side the baffle is in tension. If the pressure is on the convex side, the baffle is in compression.

There are various tables given in this procedure for flat plate coefficients. Flat plate coefficients are utilized to determine the baffle thickness or a panel thickness. Each table is specific for a given condition and loading.

Notation

A_p = area of baffle working with stiffener, in.^{2}
A_s = area of stiffener, in.^{2}
C_p = distance from centroid of composite section to panel, in.
C_s = distance from centroid of composite section to stiffener, in.
E = modulus of elasticity, psi
F_b = allowable bending stress, psi
I = moment of inertia, composite, in.^{4}
I_s = moment of inertia, stiffener, in.^{4}
l = length of baffle that works with the stiffener, in.
M = moment, in.-lb
n = number of welds attaching stiffener
P = vessel internal pressure, psig
p = maximum uniform pressure, psi
p_n = uniform pressure at any elevation, a_n, psi
R_m = vessel mean radius, in.
S_g = specific gravity of contents
t = thickness, shell, in.
t_b = thickness, baffle, in.
t_s = thickness, stiffener, in.
V = shear load, lb
w = required fillet weld size, in.
α = thermal coefficient of expansion, in./in./°F
β,γ = flat plate coefficients
ΔT = differential temperature (design temperature minus 70°F), °F
σ_b = bending stress in baffle, psi
σ_s = bending stress in stiffener, psi
$\triangle_n$ = radial expansion, in.
δ = deflection, in.
δ_a = maximum allowable deflection, in.

Baffle Dimensions

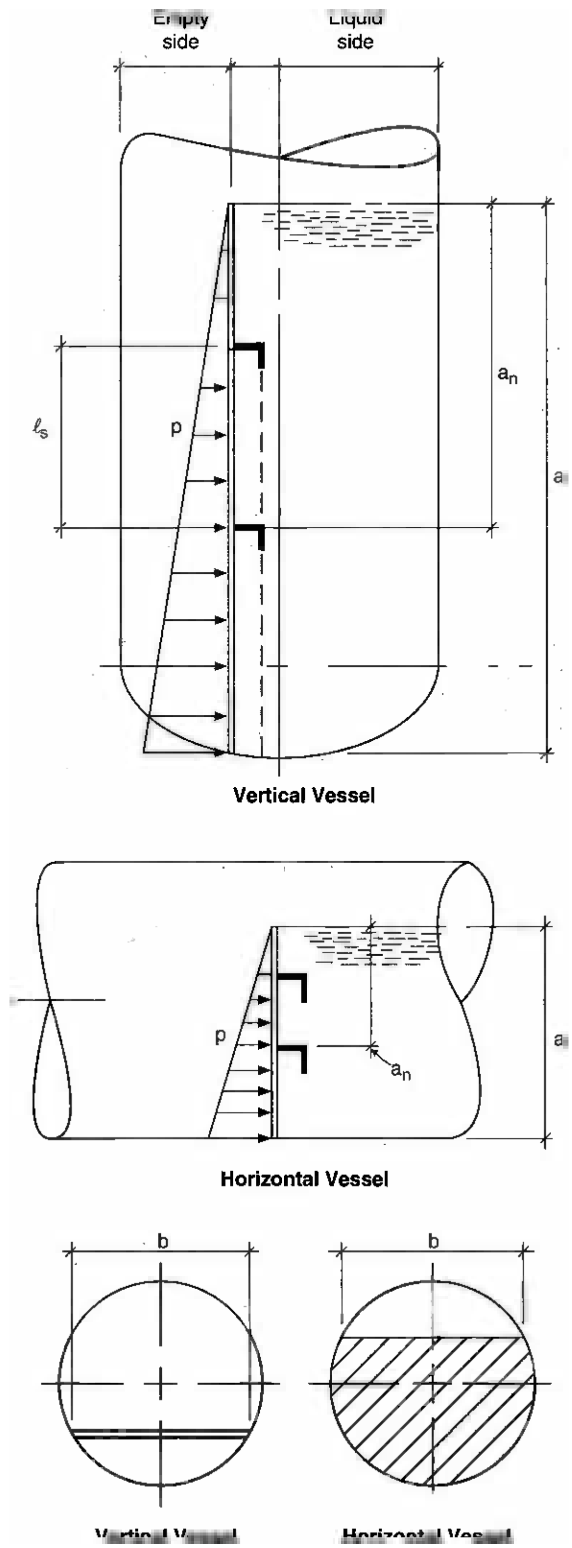


Table 5-24
Flat plate coefficients

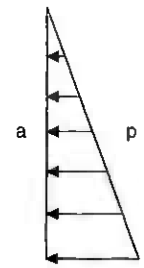
Equations

Case 1: One short edge free, three edges simply supported, uniformly decreasing load to the free edge

a/b / Coefficient	0.25	0.5	0.75	1	1.5
β_1	0.05	0.11	0.16	0.2	0.28
γ_1	0.013	0.026	0.033	0.04	0.05
a/b / Coefficient	**2**	**2.5**	**3**	**3.5**	**4**
β_1	0.32	0.35	0.36	0.37	0.37
γ_1	0.058	0.064	0.067	0.069	0.07

$$\sigma_b = \frac{\beta_1 p b^2}{t^2}$$

$$\delta = \frac{p\gamma_1 b^4}{E t_b^3}$$



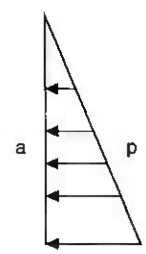
From Ref. 10, Section 6.5-4, Case 4d.

Case 2: All edges simply supported, uniform decreasing load

a/b / Coefficient	0.25	0.5	0.75	1	1.5
β_2	0.024	0.08	0.12	0.16	0.26
γ_2	0	0	0.01	0.02	0.04
a/b / Coefficient	**2**	**2.5**	**3**	**3.5**	**4**
β_2	0.32	0.35	0.37	0.38	0.38
γ_2	0.056	0.063	0.067	0.069	0.07

$$\sigma_b = \frac{\beta_2 p b^2}{t^2}$$

$$\delta = \frac{p\gamma_2 b^4}{E t_b^3}$$

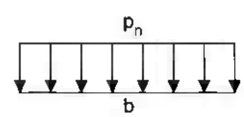


From Ref. 10, Section 6.5-4, Case 4c.

Case 3: All edges simply supported, uniform load

a/b / Coefficient	1	1.25	1.5	1.75	2
β_3	0.287	0.376	0.452	0.569	0.61
γ_3	0.0443	0.0616	0.077	0.1017	0.1106
a/b / Coefficient	**2.5**	**3**	**4**	**5**	**Infinity**
β_3	0.65	0.713	0.741	0.748	0.75
γ_3	0.125	0.1336	0.14	0.1416	0.1422

$$\sigma_b = \frac{\beta_3 p b_n^2}{t^2}$$



$$\delta = \gamma_3\left(\frac{p}{E}\right)\left(\frac{b_n^4}{t_b^3}\right)$$

- Assume p as a uniform load at center of plate.
- $A_n > b_n$

From Ref. 10, Section 6.5-4, Case 4a.

Unstiffened Baffle Check

- *Find load, p.*

$$p = \frac{62.4aS_g}{144}$$

- *Find baffle thickness, t_b.*

$$t_b = \sqrt{\frac{\beta_1 pb^2}{F_b}}$$

- *Find baffle deflection, δ.*

$$\delta = \frac{p\gamma_1 b^4}{Et_b^3}$$

Limit deflection to the smaller of $t_b/2$ or b/360. If deflection is excessive then:

a. Increase the baffle thickness.
b. Add stiffeners.
c. Go to curved baffle design.

If stiffeners are added, the first step is to find the maximum "a" and "b" dimensions that will meet the allowable deflection for a given panel size. This will establish the stiffener spacing for both horizontal and vertical stiffeners. The ultimate design is a balance between baffle thickness, stiffener spacing, and stiffener size.

Thermal Check of Baffle

- Vessel radial expansion due to pressure.

$$\Delta_1 = \frac{0.85PR_m}{tE}$$

- Vessel radial expansion due to temperature.

$$\Delta_2 = R_m \alpha \, \Delta T$$

- Thermal expansion of baffle.

$$\Delta_3 = 0.5b\alpha \, \Delta T$$

- Differential expansion.

$$\Delta_4 = \Delta_1 + \Delta_2 - \Delta_3$$

Stiffener Design

Divide baffle into panels to limit deflection to the lesser of $t_b/2$ or b/360. Deflection is calculated based on the appropriate Cases 1 through 3.

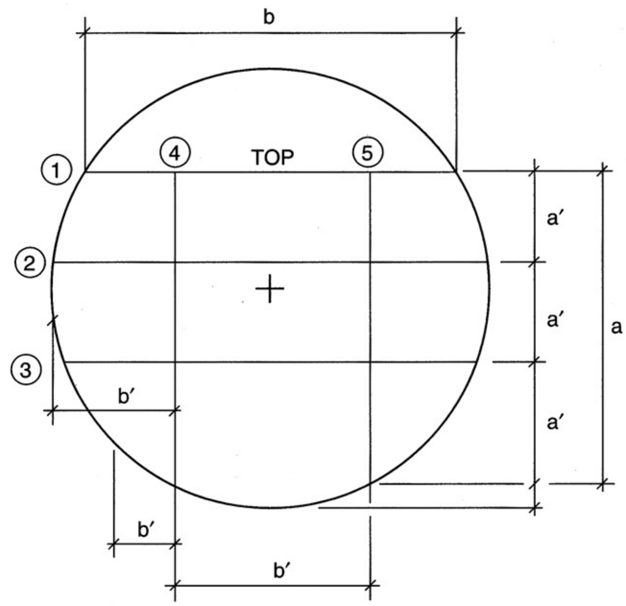
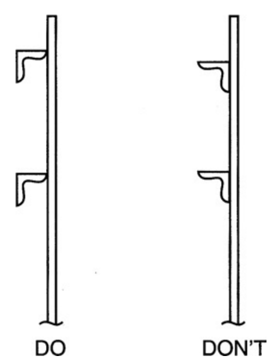


Figure 5-18. Example of stiffener layout.

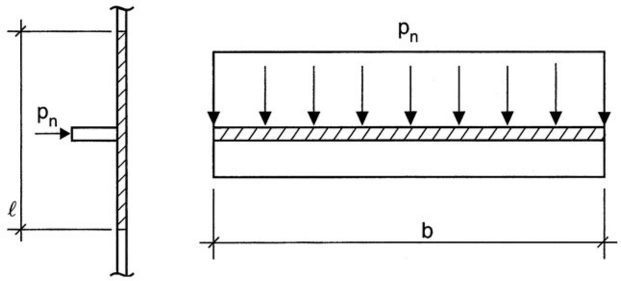
- *Check baffle for panel size $a' \times b'$.*
- *Check stiffener for length a or b.*

Recommendations for attaching stiffeners



Benefits: Provides added stiffness and no corrosion trap.

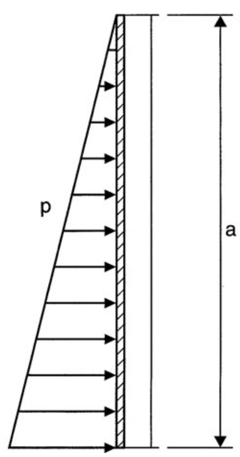
Horizontal Stiffener Design



$$p_n = \frac{a_n 62.4 S_g}{144} \qquad M = \frac{p_n l b^2}{8}$$

$$\delta = \frac{5 p_n l b^4}{384 EI} \qquad V = \frac{p_n l b}{2}$$

Vertical Stiffener Design



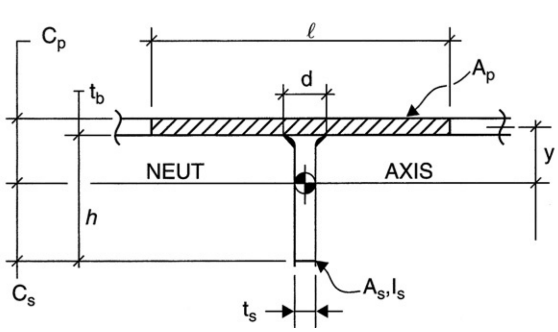
$$p = \frac{a 62.4 S_g}{144}$$

$$\delta = \frac{2.5 p l a^4}{384 EI}$$

$$M = .0642 p l a^2$$

$$V = \frac{p l a}{3}$$

Properties of Stiffener



$$A_s = t_s h$$

$$A_p = t_b l$$

$$I_S = \frac{t_s h^3}{12}$$

$$C_p = \frac{A_s y}{A_s + A_p} + \frac{t_b}{2}$$

$$C_s = (h + t_b) - C_p$$

$$I = I_S + \frac{A_p t_b^2}{12} + \frac{A_s A_p y^2}{A_s + A_p}$$

ℓ = lesser of $32t_b$ or stiffener spacing

Stresses in Baffle/Stiffener

$$\sigma_p = \frac{MC_p}{I}$$

$$\sigma_s = \frac{MC_s}{I}$$

Size Welds Attaching Stiffeners

For E70XX Welds:

$$w = \frac{Vdy}{11,200 In}$$

Table 5-25
Intermittent Welds

Percent of Continuous Weld	Length of Intermittent Welds and Distance Between Centers		
75%		3–4	
66			4–6
60		3–5	
57			4–7
50	2–4	3–6	4–8
44			4–9
43		3–7	
40	2–5		4–10
37		3–8	
33	2–6	3–9	4–12

Reprinted by permission of the James F. Lincoln Arc Welding Foundation.

For E60XX Welds:

$$w = \frac{Vdy}{9600In}$$

Sample Problem

- Given: Horizontal vessel with a vertical baffle

$P = 250$ psig
D.T. $= 500^{\circ}$F
Material = SA-516-70
C.a. $= 0.125$ in.
JE $= 1.0$
$E = 27.3 \times 10^{6}$ psi
$\alpha = 7.124 \times 10^{-6}$ in./in./°F
$F_y = 30.8$ ksi
$D = 240$ in.
$F_b = 0.66\ F_y = 20.33$ ksi
$R_m = 120.938$

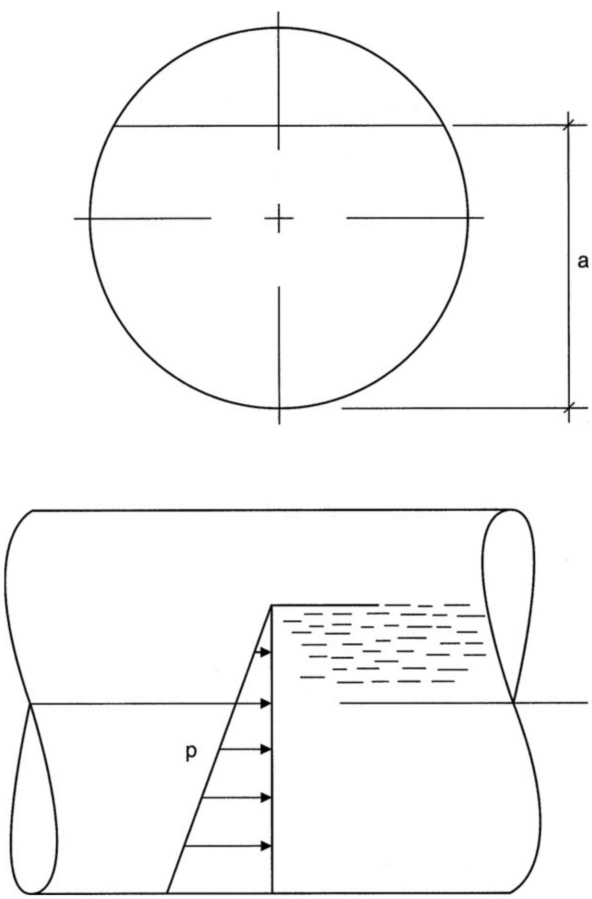


Figure 5-19. Sample problem.

$t_s = 1.75$ in.

$S_g = 0.8$

$a = 15$ ft

$\Delta T = 500 - 70 = 430^{\circ}$F

- *Find baffle thickness without stiffener.*

$$p = \frac{62.4aS_g}{144} = 5.2 \text{ psi}$$

$$\frac{a}{b}\text{ratio} = 15/20 = 0.75$$

from Table 5-25, Case 1:

$$\beta_1 = 0.16 \quad \gamma_1 = 0.033$$

- *Thickness of baffle, t_b.*

$$t_b = \sqrt{\frac{\beta_1 pb^2}{F_b}} = \sqrt{\frac{0.16(5.2)240^2}{20,330}}$$

$$t_b = 1.53 + 0.25 = 1.78$$

No good! Use stiffeners.

- *Assume a suitable baffle thickness and determine maximum panel size.*

$t_b = 0.75$ in. corroded
maximum panel size $= 4\text{ft} \times 4\text{ft}$

- Maximum pressure, p.

$$p = \frac{13(62.4)0.8}{144} = 4.5 \text{ psi}$$

$$\frac{a}{b} = \frac{4}{4} = 1$$

See Table 5-25, Case 3:

$$\beta_3 = 0.287 \quad \gamma_3 = 0.0443$$

$$\sigma_b = \frac{\beta_3 pb_n^2}{t^2}$$

$$= \frac{0.287(4.5)48^2}{0.75^2} = 5290 \text{ psi} < 20,333 \text{ psi}$$

$$\delta = \gamma_3\left(\frac{p}{E}\right)\frac{b_n^4}{t^3} = 0.0443\left(\frac{4.5}{27.3\times10^6}\right)\left(\frac{48^4}{0.75^3}\right)$$

$$= 0.092 \text{ in.} < 0.375 \text{ in.}$$

Balance OK by inspection

- *Assume a layout where the maximum stiffener spacing is 4 ft.*

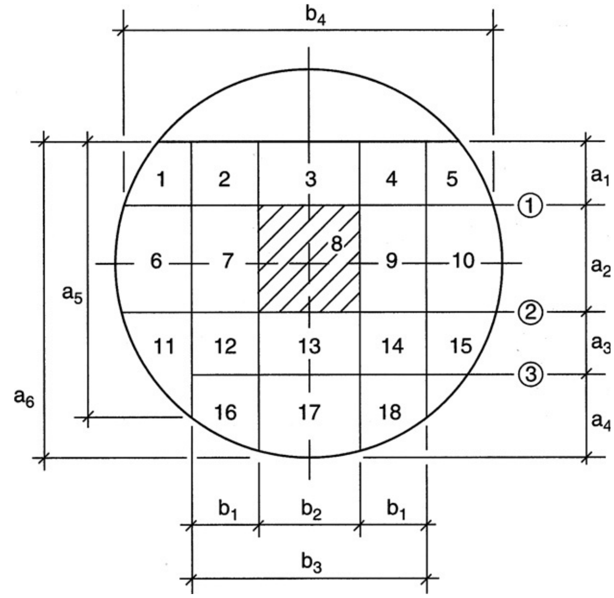


Figure 5-20. Battle layout for sample problem.

(4) horizontal stiffeners
(4) vertical stiffeners
(18) panels

- *Check horizontal stiffeners.*

Dimensions:

$a_1 = 3$ ft $\quad b_1 = 4$ ft

$a_2 = 4$ ft $\quad b_2 = 4$ ft

$a_3 = 4$ ft $\quad b_3 = 12$ ft

$a_4 = 4$ ft $\quad b_4 = 19.6$ ft

$a_5 = 13$ ft

$a_6 = 14.8$ft

- *Assume stiffener size, 1 in. × 4 in.*

$y = 2.375$ in.

$A_s = t_s h = 1(4) = 4 \text{ in.}^2$

$l = 32t_b = 32(0.75) = 24 \text{ in.} < 48 \text{ in.}$

$A_P = t_b l = 0.75(24) = 18 \text{ in.}^2$

$$I_s = \frac{bh^3}{12} = \frac{1(4^3)}{12} = 5.33 \text{ in}^4.$$

$$I = I_s + \frac{A_p t_b^2}{12} + \frac{A_s A_p y^2}{A_s + A_p}$$

$$= 5.33 + 0.633 + 18.46 = 24.42 \text{ in.}^4$$

$$C_p = \frac{A_s y}{A_s + A_p} + \frac{t_b}{2}$$

$$= \frac{4(2.375)}{22} + \frac{0.75}{2} = 0.807 \text{ in.}$$

$$C_s = (h + tb) - C_p$$

$$= 4 + 0.75 - 0.807 = 3.943$$

Check deflections:

Item	b_n	p_n	δ
1	235.2	1.04	1.49
2	235.2	2.43	3.49
3	144	3.81	0.767

Deflections exceed allowable. No good!

- *Assume a larger stiffener size: WT9 × 59.3.*

  $t_f = 1.06 - 0.25 = 0.81$

  $t_w = 0.625 - 0.25 = 0.375$

- *Check corroded thickness to find properties of corroded section.* This section would be equivalent to a WT9 × 30. Properties are:

$A_s = 8.82 \text{ in.}^2$

$I_s = 64.7 \text{ in.}^4$

$C_s = 2.16$ in.

$H = 9$ in.

$C_p = h + t_b - C_s$

$= 9 + 0.75 - 2.16 = 7.59$ in.

Table 5-26
Summary of Results for Stress and Deflection in Composite Stiffeners for Sample Problem

Item	Orientation	a_n	b_n	p_n	M	δ	V	σ_p	σ_s
1	Horiz.	235.2	—	1.04	172,595	0.09	2690	3504	997
2	Horiz.	235.2	—	2.43	403,275	0.21	6287	8186	2330
3	Horiz.	144	—	3.81	237,012	0.045	6035	4811	1370
4	Vert.	—	156	4.50	154,674	0.037	5648	3141	894
5	Vert.	—	177.6	5.13	228,539	0.072	6681	6681	1320

$$y = C_p - \frac{t_b}{2}$$

$$= 7.215 \text{ in.}$$

$$I = I_s + \frac{A_p t_b^2}{12} + \frac{A_s A_p y^2}{A_s + A_p}$$

$$= 64.7 + 0.844 + 308.14 = 373.7 \text{ in.}^4$$

- *Check stresses and deflections.* See results in Table 5-27.
- *Stresses and deflections are acceptable.*
- *Check welds.*

$$d = t_s + 2t_w = 0.375 + 2(0.323) = 1.02$$

$$y = 7.215 \text{ in.}$$

$$I = 373.7 \text{ in.}^4$$

$$n = 2$$

$$w = \frac{Vdy}{11,200In} = \frac{6681(1.02)7.215}{11,200(373.7)2}$$

$$= 0.005 + 0.125 = 0.13 \text{ in.}$$

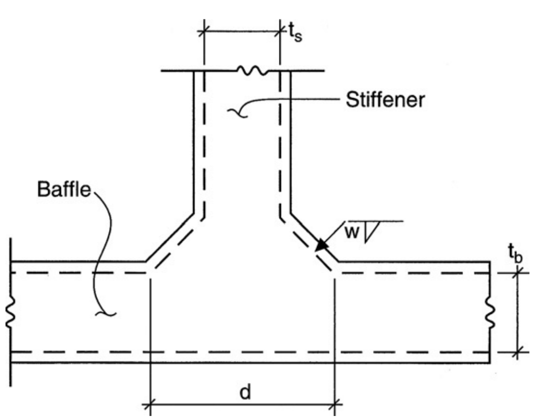


Figure 5-21. Details of weld attaching stiffener.

- *Check thermal expansion of baffle.*

$$\Delta_1 = \frac{0.85PR_m}{tE} = \frac{0.85(250)120.938}{1.75(27.3 \times 10^6)}$$

$$= 0.00054 \text{ in.}$$

$$\Delta_2 = R_m\alpha\ \Delta T = 120.938(7.124 \times 10^{-6})430$$

$$= 0.370 \text{ in.}$$

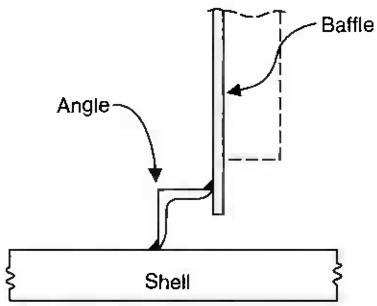
$$\Delta_3 = 0.5b\alpha\ \Delta T = 0.5(240)(7.124 \times 10^{-6})430$$

$$= 0.367 \text{ in.}$$

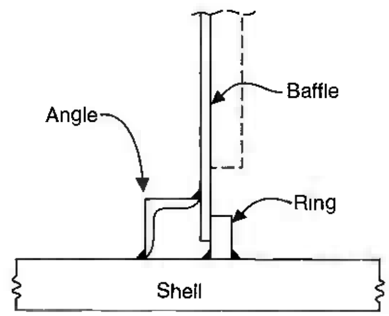
$$\Delta_4 = \Delta_1 + \Delta_2 - \Delta_3 = 0.00054 + 0.370 - 0.367$$

$$= 0.0035 < 0.06 \text{ in.}$$

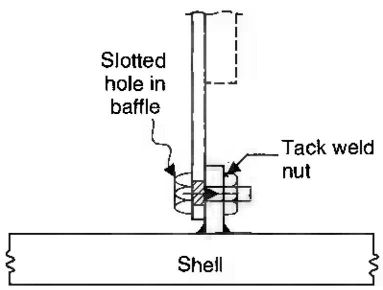
Flexible Baffle Design for Full-Cross-Section Baffles



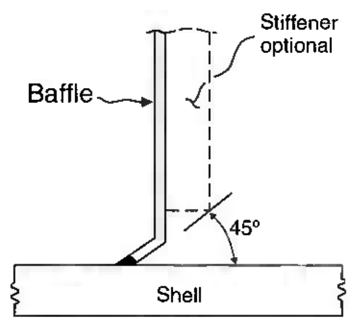
Attached by Angle to Shell



Attached by Angle,
Guided by Ring

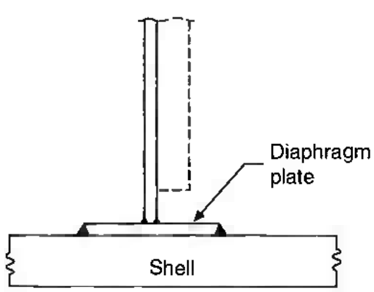


Baffle Bolted to Shell

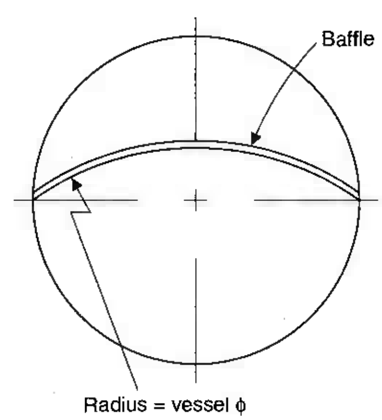


Baffle Welded to Shell

Note: Difficult to fabricate when
t > 3/8" or inside head



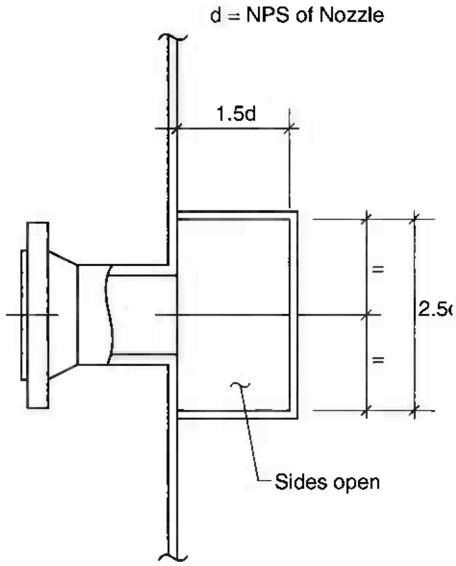
Diaphragm Plate

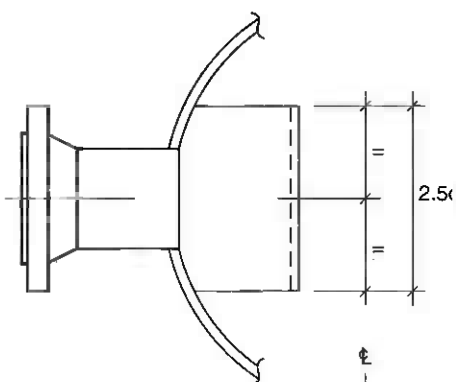


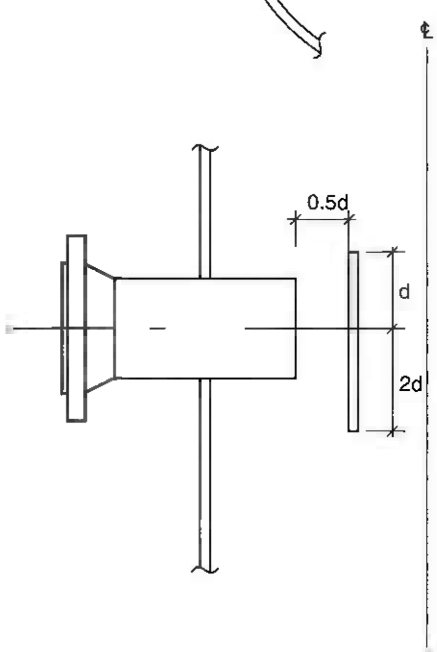
Alternate Construction
Benefits: Easily takes pressure
from either side and good for thermal
expansion

Miscellaneous Baffle Configurations

Vertical Vessels







Horizontal Vessels

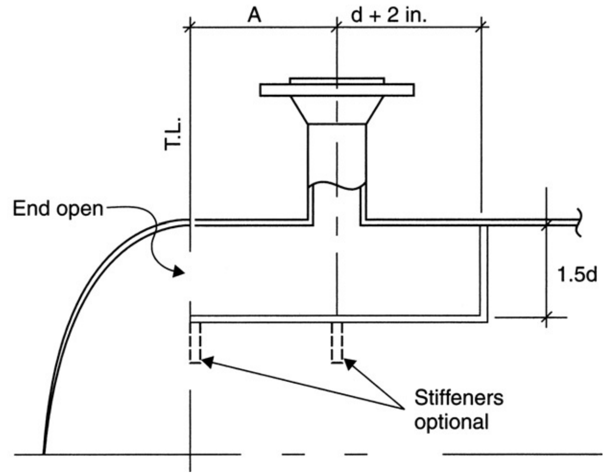
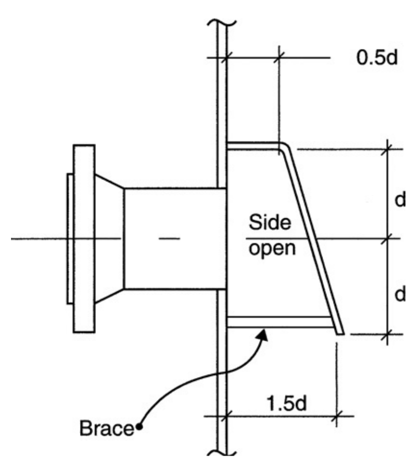


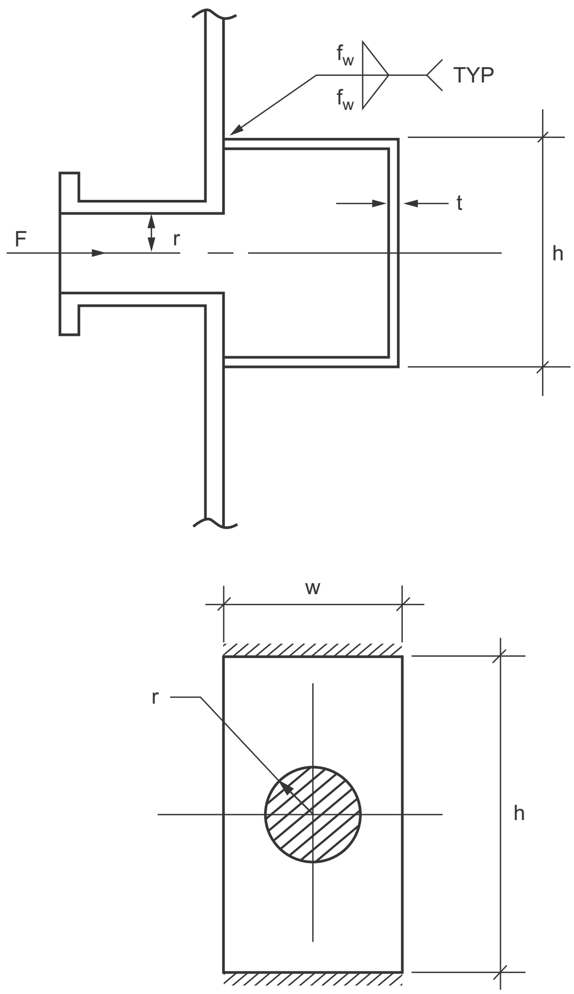
Table 5-27
Dimension "A"

Nozzle Size (in.)	A (in.)	Nozzle Size (in.)	A (in.)
2	5	14	18
3	7	16	20
4	8	18	22
6	10	20	24
8	12	24	28
10	14		
12	16		



Procedure 5-13: Design of Impingement Plates

Notation



Dimensions of baffle impingement plate

Figure 5-22. Dimensions of Baffle Impingement Plate.

A = Cross sectional area of nozzle, in^2
d = Density of liquid, PCF
F = Equivalent static force, Lbs
F_b = Allowable bending stress, PSI
F_y = Minimum specified yield strength, PSI
F_U = Minimum specified tensile strength, PSI
F_W = Allowable shear stress, PSI
f_b = Bending stress, PSI
f_w = Fillet weld size
g = Acceleration due to gravity, 32 Ft/Sec^2
L_W = Length of weld, in
r = Inside radius of nozzle, in
V = Velocity, FPS
τ_S = Shear stress per inch of weld, PSI

Calculation

- Equivalent static force, F

  $F = (V\,A\,d)/g$

  - Maximum bending stress, at center of plate, f_b

    $n = w/h < 1$ (Use 1 for square plate)

    $f_b = (5.3\,F)/(1 + 2.4\,n^2\,t^2)$

  - Allowable stress;

    a. Bending, F_b

    Lesser of… .6 F_y or .3 F_U

    b. Weld in Shear, F_S

    $= .4\,F_y$

  Size of weld required;

  - Length of weld, L_W

    a. Welded from one side only;

    $L_W = 2\,w$

    b. Welded both sides, top and bottom;

    $L_W = 4\,w$

  - Shear per inch of weld, τ_S

    $\tau_S = F/L_W$

  - Size of weld required, f_W

    $f_W = \tau_S/.707\,F_S$

References

[1] Ackley EJ. Film Coefficient of Heat Transfer for Agitated Process Vessels. Chemical Engineering August 22, 1960.

[2] Stuhlbarg D. How to Design Tank Heating Coils. Petroleum Refiner April 1959.

[3] Steve EH. Refine Temperature Control in an Odd-Shaped Vessel. Hydrocarbon Processing July 1999.

[4] Bisi F, Menicatti S. How to Calculate Tank Heat Losses. Hydrocarbon Processing February 1967.

[5] Kumana JD, Kothari SP. Predict Storage Tank Heat Transfer Precisely. Chemical Engineering March 22, 1982.

[6] Kern DQ. Process Heat Transfer. McGraw-Hill; 1950.

[7] Bondy F, Lippa S. Heat Transfer in Agitated Vessels. Chemical Engineering April 4, 1983.

[8] Steam, Its Generation and Use. 40th ed. Babcock and Wilcox Company; 1992. section 3–12.

[9] “Flow of Fluids through Valves, Fittings and Pipe,” Crane Company, Technical Paper No. 410.

[10] Blodgett OW. Design of Welded structures. J.F Lincoln Arc Welding Foundation 1966. Section 5.3.

[11] Manual of Steel Construction. 8th ed. American Institute of Steel Construction, Inc.; 1988.

6
Special Designs

Procedure 6-1: Design of Large-Diameter Nozzle Openings [1]

There are three methods for calculating the strength of reinforcement required for openings in pressure vessels:

1. Area replacement rules per UG-36(b).
2. Analysis per Appendix 1-7.
   a. 2/3 area replacement rule.
   b. Membrane-bending stress analysis.
3. FEA.

The Code defines when and where these methods apply. Reinforcement for large-diameter openings has been in the Code for a long time. The previous rule was simply to move the majority of the area replacement closer to the nozzle neck, also called the *2/3 rule*. Unfortunately, there were a few cases of flange leakage where the flange was located close to the shell. It was discovered that as the opening opened up, the flange was distorted. It was actually bending. In addition, the 2/3 rule did not allow for an accurate way to determine MAWP for the vessel without proof testing.

This issue was addressed in 1979 by McBride and Jacobs. The principle was to calculate stresses in two distinct areas, membrane and bending. Membrane stresses are based on pressure area times metal area. Bending is based on AISC beam formulas. The neck-and-shell section (and sometimes the flange as well) is assumed as bent on the hard axis. This is not a beam-on-elastic-foundation calculation. It is more of a brute-force approach.

This procedure was eventually adopted by the Code and incorporated. Unfortunately, it turned out that the procedure, while good for most cases, was not good for all. Yet it was still superior to what we used before this paper was published. The ASME has now revised the applicability of the procedure to the cases where it has been deemed safe.

Large openings calculated by this procedure are limited to openings less than 70% of the vessel diameter. There are four cases that can be solved for, depending on your nozzle geometry.

Reinforcement for Large-Diameter Openings

Per ASME, Section VIII, Appendix l-7(b)l(b), the rules for “radial nozzles,” not oblique or tangential, must meet strength requirements in addition to area replacement rules. The following lists the parameters for which these additional calculations shall be performed:

a. Exceed the limits of UG-36(b).
b. Vessel diameter > 60 in.
c. Nozzle diameter > 40 in.
d. Nozzle diameter $> 3.4\sqrt{Rt}$.
e. The ratio $R_n/R < 0.7$ (that is, the nozzle does not exceed 70% of the vessel diameter).

Table 6-1 shows the ratio of vessel diameter, D, and shell thickness, t, where the values of $3.4\sqrt{Rt}$ are greater than 40. The heavy line indicates the limits for which 40 is exceeded. For nozzles that exceed these parameters, a finite element analysis (FEA) should be performed.

Table 6-1
Parameters for large-diameter nozzles

t \ D	60	72	84	96	108	120	132	144	156	168	180
1.00											
1.25											
1.5										38.16	39.5
1.75								38.2	39.7	41.2	42.6
2.00							39.1	40.8	42.5		
2.25						39.5	41.4				
2.50					39.5	41.6					
2.75				39.1	41.4						
3.00				40.8							
3.25			39.72	42.5							
3.50			41.22								
3.75		39.5									
4.00	37.2	40.8									
4.25	38.4										
4.50	39.5										
4.75	40.5										

Use $^2/_3$ rule area replacement when $3.4\sqrt{Rt} < 40$

Use membrane-bending analysis when $3.4\sqrt{Rt} > 40$

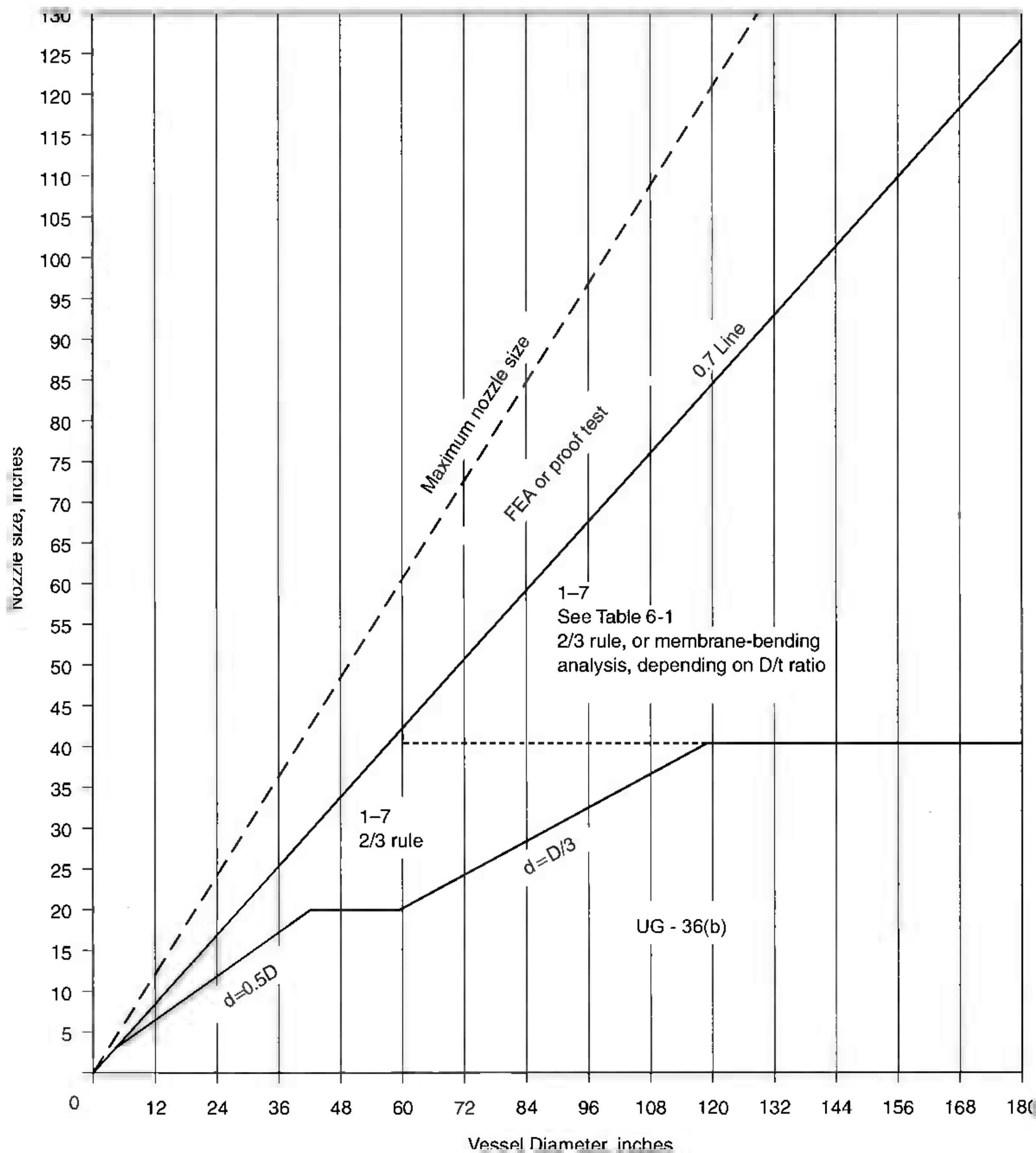


Figure 6-1. Guideline of nozzle reinforcement rules.

Large Openings—Membrane and Bending Analysis

Notation

A_s = area of steel, in.^{2}
A_p = area of pressure, in.^{2}
P = internal pressure, psi (design or test)
r_m = mean radius of nozzle, in.
R_m = mean radius of shell, in.
T = thickness of shell, in.
t = thickness of nozzle, in.
F_y = minimum specified yield strength, ksi
σ = maximum combined stress, psi
σ_b = bending stress, psi
σ_m = membrane stress, psi
I = moment of inertia, in.^{4}
M = bending moment, in.-lb

Procedure

Step 1: Compute boundary limits for bending along shell and nozzle in accordance with Note 3. Limit will be governed by whether material of construction has a yield strength, F_y, less than or greater than 40 ksi.

Along shell =
Along nozzle =

Step 2: Utilizing the appropriate case (Figure 6-3) calculate the moment of inertia, I, and the distance from centroid to the inside of the shell, C.

$I =$
$C =$

Step 3: Compute membrane and bending stresses in accordance with the equations given later.

$\sigma_m =$
$\sigma_b =$

Step 4: Combine stresses and compare with allowable.

$\sigma_m + \sigma_b =$

Calculations

- *Membrane stress, σ_m nozzles with reinforcing pads* (Cases 1 and 3).

$$\sigma_m = P\left[\frac{\left(R_i\left(r_i + t + \sqrt{R_{mT}}\right) + R_i\left(T + T_e + \sqrt{r_m t}\right)\right)}{A_S}\right]$$

- *Membrane stress, S_m nozzles without reinforcing pads* (Cases 2 and 4).

$$\sigma_m = P\left[\frac{\left(R_i\left(r_i + t + \sqrt{R_{mT}}\right) + R_i\left(T + \sqrt{r_m t}\right)\right)}{A_S}\right]$$

- *Bending stress, σ_b.*

$$M = P\left(\frac{r_i^3}{6} + R_i r_i C\right)$$

$$\sigma_b = \frac{MC}{I}$$

- *Allowable stesses.*

$$\sigma_m < S$$
$$\sigma_m + \sigma_b < 1.5S$$

Notes

1. Openings that exceed the limits of UG-36(b)(1) shall meet the requirements of the 2/3 rule.
2. This analysis combines the primary membrane stress due to pressure with the secondary bending stress resulting from the flexure of the nozzle about the hard axis.
3. Boundaries of metal along the shell and nozzle wall are as follows:

	Along Shell	Along Nozzle
Cases 1 and 2	$\sqrt{R_m T}$	$\sqrt{r_m t}$
Cases 3 and 4	16T	16t

4. This procedure applies to radial nozzles only.

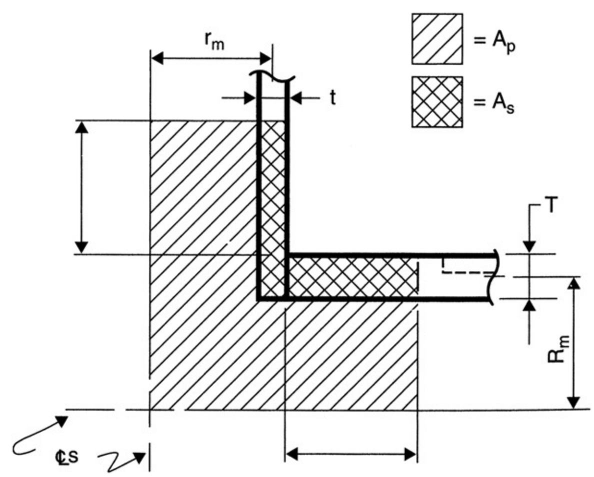
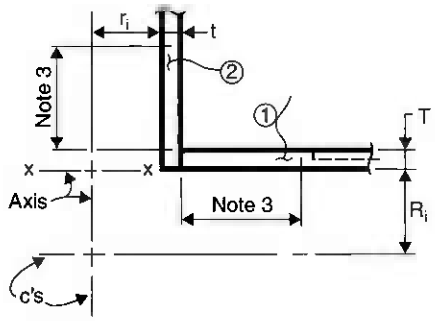


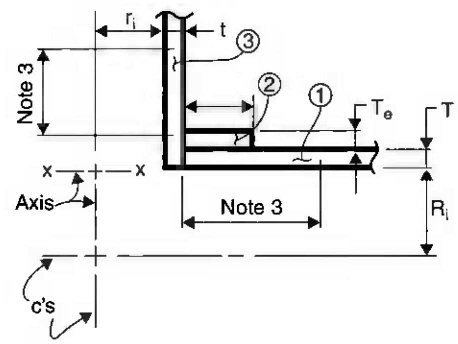
Figure 6-2. Areas of pressure and steel for nozzles.



Moment of Inertia					
Part	A	Y	AY	AY^2	I
1					
2					
$\sum$					

$$C = \frac{\sum AY}{\sum A} \qquad I = \sum AY^2 + \sum I - C\sum AY$$

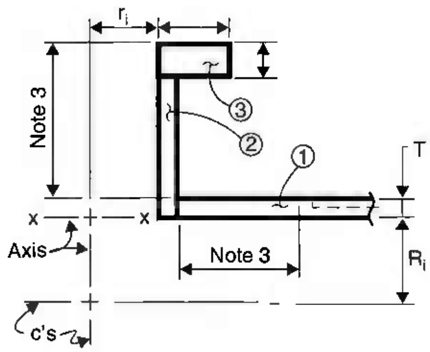
Case 1



Part	A	Y	AY	AY^2	I
1					
2					
3					
$\sum$					

$$C = \frac{\sum AY}{\sum A} \qquad I = \sum AY^2 + \sum I - C\sum AY$$

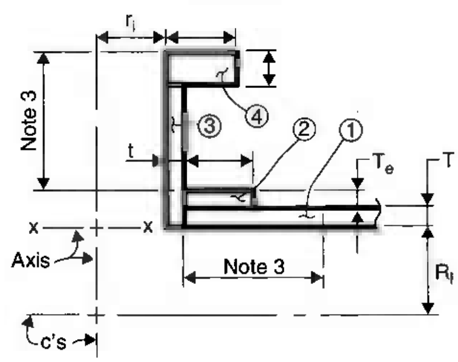
Case 2



Moment of Inertia					
Part	A	Y	AY	AY^2	I
1					
2					
3					
$\sum$					

$$C = \frac{\sum AY}{\sum A} \qquad I = \sum AY^2 + \sum I - C\sum AY$$

Case 3



Part	A	Y	AY	AY^2	I
1					
2					
3					
4					
$\sum$					

$$C = \frac{\sum AY}{\sum A} \qquad I = \sum AY^2 + \sum I - C\sum AY$$

Case 4

Figure 6-3. Calculation form for moment of inertia I and centroid C for various nozzle configurations. Select the case that fits the geometry of the nozzle being considered.

Procedure 6-2: Tower Deflection [3]

Notation

L = overall length of vessel, in.
L_n = length of section, in.
E_n = modulus of elasticity of section, psi
I_n = moment of inertia of section, in.^{4}
W_n = concentrated loads, lb
w = uniformly distributed load, lb/in.
W_{max} = uniformly distributed load at top of vessel, lb/in.
W_{min} = uniformly distributed load at bottom of vessel, lb/in.
X = ratio L_n/L for concentrated loads
δ = deflection, in.

Section n	L_n	L_n^4	I_n	$\frac{L_n^4}{I_n}$	$\frac{L_n^4}{I_{n-1}}$
			$\Sigma=$		

$$\delta = \frac{w}{8E}\left[\sum\frac{L_n^4}{I_n} - \sum\frac{L_n^4}{I_{n-1}}\right]$$

Cases

Case 1: Uniform Vessel, Uniform Load

$$\delta = \frac{wL^4}{8EI}$$

Case 2: Nonuniform Vessel, Uniform Load

- *If E is constant*

$$\delta = \frac{w}{8E}\left[\left(\frac{L_1^4}{I_1}+\frac{L_2^4}{I_2}+\cdots+\frac{L_n^4}{I_n}\right) - \left(\frac{L_2^4}{I_1}+\frac{L_3^4}{I_2}+\cdots+\frac{L_n^4}{I_{n-1}}\right)\right]$$

- *If E is not constant*

$$\delta = \frac{w}{8}\left[\left(\frac{L_1^4}{I_1E_1}+\frac{L_2^4}{I_2E_2}+\cdots+\frac{L_n^4}{I_nE_n}\right) - \left(\frac{L_2^4}{I_1E_1}+\frac{L_3^4}{I_2E_2}+\cdots+\frac{L_n^4}{I_{n-1}E_{n-1}}\right)\right]$$

Case 3: Nonuniform Vessel, Nonuniform Load

$$\delta = \left[\sum\frac{L_n^4}{I_n} - \sum\frac{L_n^4}{I_{n-1}}\right]\left[\frac{w_{min}}{8E}+\frac{5.5(w_{max}-w_{min})}{60E}\right]$$

Section n	L_n	L_n^4	I_n	$\frac{L_n^4}{I_n}$	$\frac{L_n^4}{I_{n-1}}$
			$\Sigma=$		

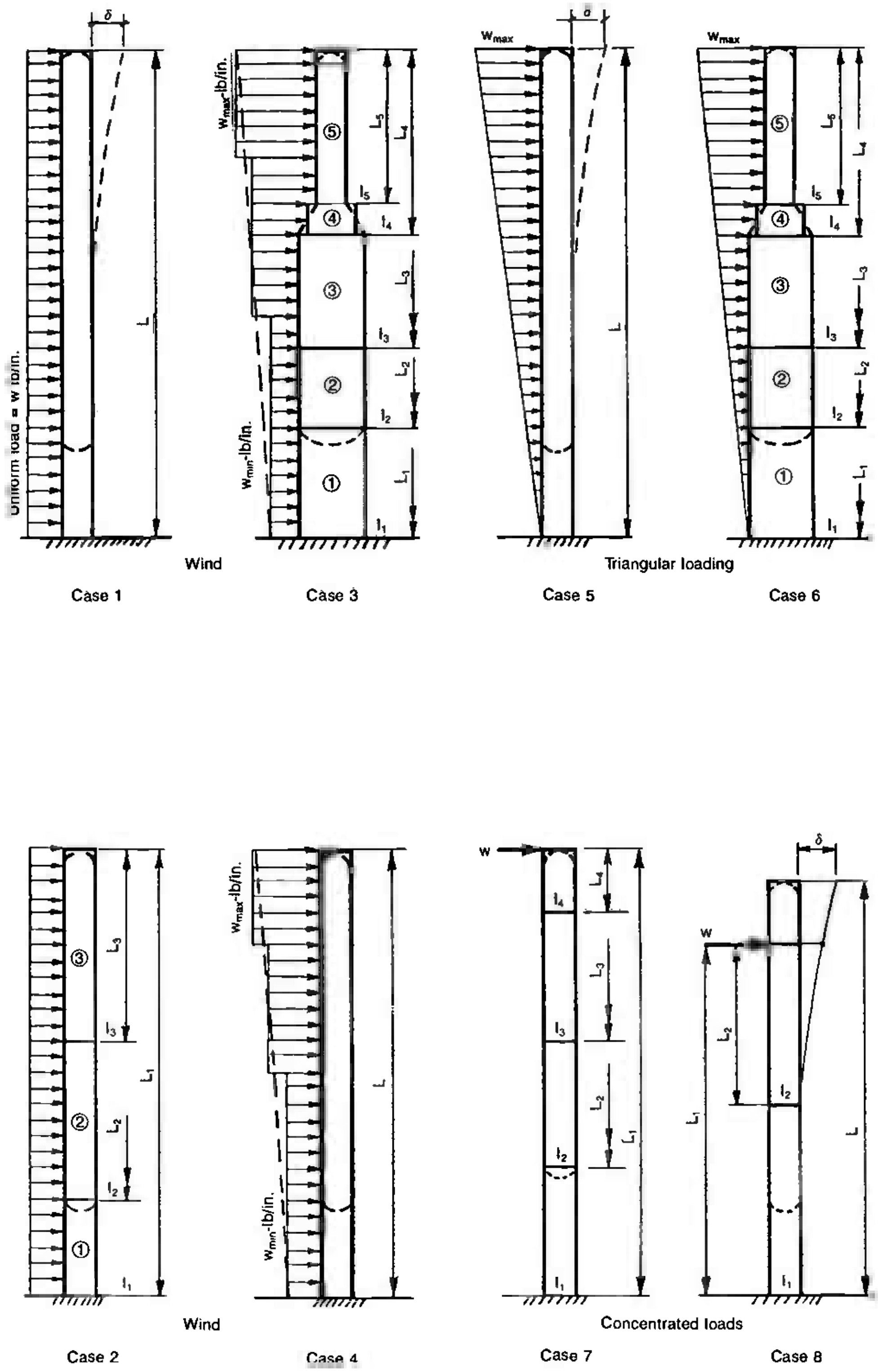


Figure 6-4. Dimension and various loadings for vertical, skirt-supported vessels.

Case 4: Uniform Vessel, Nonuniform Load

$$\delta = \frac{L^4}{I}\left[\frac{w_{min}}{8E} + \frac{5.5(w_{max} - w_{min})}{60E}\right]$$

Case 5: Uniform Vessel, Triangular Load

Load at base = 0
Load at top = w, lb/in.

$$\delta = \frac{5.5wL^4}{60EI}$$

Case 6: Nonuniform Vessel, Triangular Load

$$\delta = \frac{5.5w}{60E}\left[\sum \frac{L_n^4}{I_n} - \sum \frac{L_n^4}{I_{n-1}}\right]$$

Section n	L_n	L_n^4	I_n	$\frac{L_n^4}{I_n}$	$\frac{L_n^4}{I_{n-1}}$
			$\Sigma=$		

Case 7: Concentrated Load at Top of Vessel

- *Uniform vessel*

$$\delta = \frac{WL^3}{3EI}$$

Section n	L_n	L_n^3	I_n	$\frac{L_n^3}{I_n}$	$\frac{L_n^3}{I_{n-1}}$
			$\Sigma=$		

- *Nonuniform vessel*

$$\delta = \frac{W}{3E}\left[\sum \frac{L_n^3}{I_n} - \sum \frac{L_n^3}{I_{n-1}}\right]$$

Case 8: Concentrated Lateral Load at Any Elevation

$$X = \frac{L_1}{L}$$

- *For uniform vessel.*

$$\delta = \frac{WL_1^3}{3EI}\left(\frac{3 - X}{2X}\right)$$

- *For nonuniform vessel.*

$$\delta = \left[\frac{W}{3E}\left(\sum \frac{L_n^3}{I_n} - \sum \frac{L_n^3}{I_{n-1}}\right)\right]\left[\frac{3 - X}{2X}\right]$$

Section n	L_n	L_n^3	I_n	$\frac{L_n^3}{I_n}$	$\frac{L_n^3}{I_{n-1}}$
			$\Sigma=$		

Notes

1. This procedure calculates the static deflection of tall towers due to various loadings and accounts for the following:
   a. Different wind pressures at different elevations.
   b. Various thicknesses, diameters, and moments of inertia at different elevations.
   c. Different moduli of elasticity at different elevations due to a change in material or temperatures.
2. This procedure is not valid for vessels that are subject to wind-induced oscillations or that must be designed dynamically.
3. Deflection should be limited to 6 in. per 100 ft.
4. Deflections due to combinations of various loadings should be added to find the overall deflection.

Procedure 6-3: Design of Ring Girders [4–8]

The circular girder supports the weight of the tank, vessel, or bin; its contents; and any installed plant equipment. The ring beam will take the load from the vessel uniformly distributed over its full circumference, and in turn will be supported on a structural steel framework in at least four places.

The shell of a column-supported tank, vessel, or bin can be considered as a ring beam whether or not there is a special built-up beam structure for that purpose.

Horizontal seismic force is transferred from the shell or short support skirt to the ring beam by tangential shear. The girder performs the function of transmitting the horizontal shear from the tank shell to the rods and posts of the supporting structure.

The girder is analyzed as a closed horizontal ring acted upon by the horizontal shear stresses in the tank shell and by the horizontal components of the stresses in the rods and posts in the top panel of the supporting steel framework.

Maximum girder stresses generally occur when the direction of the earthquake force is parallel to a diameter passing through a pair of opposite posts.

The ring beam (girder) is subjected to compression, bending, and torsion due to the weight of the tank, contents, and horizontal wind or seismic forces. The maximum bending moment will occur at the supports. The torsional moment will be zero at the supports and maximum at an angular distance β away from support points.

This procedure assumes that the rods are tension-only members and connect every adjacent post. It is not valid for designs where the rods skip a post or two!

For cases where the ring beam has additional moment, tangential and/or radial loads (such as sloping columns) these additional horizontal loads may be calculated using ring redundants. See procedure on stresses in circular rings.

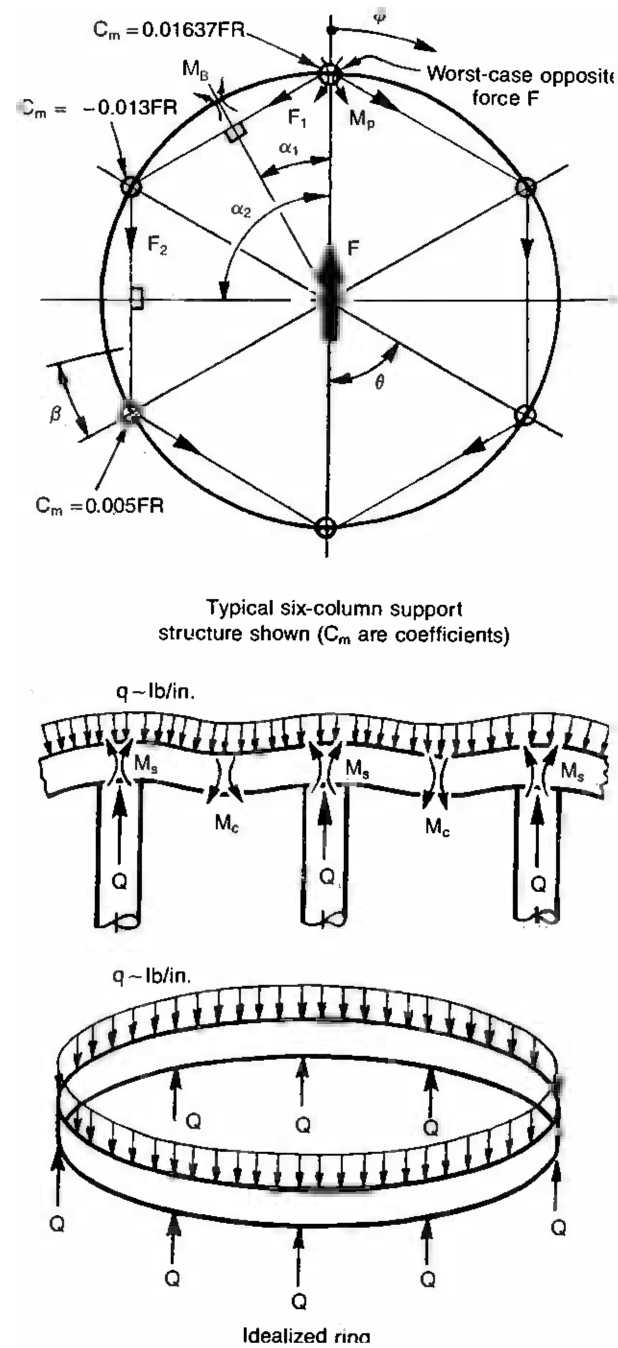


Figure 6-5. Dimension, forces, and loading at a ring girder.

Notation

D = diameter of column circle, in.
F = horizontal wind or earthquake force at plane of girder, lb

Table 6-2
Internal bending moments

No. of Posts	Due to Force Q M_s	M_c	M_T	β	Due to Force F M_P	M_B
4	−0.1366 QR	+0.0705 QR	+0.0211 QR	19°-12′	+0.0683 FR	−0.049 FR
6	−0.0889 QR	+0.0451 QR	+0.0090 QR	12°-44′	+0.0164 FR	−0.013 FR
8	−0.0661 QR	+0.0333 QR	+0.0050 QR	9°-33′	+0.0061 FR	−0.0058 FR
10	−0.0527 QR	+0.0265 QR	+0.0032 QR	7°-37′	+0.0030 FR	−0.0029 FR
12	−0.0438 QR	+0.0220 QR	+0.0022 QR	6°-21′	+0.0016 FR	−0.0016 FR
16	−0.0328 QR	+0.0164 QR	+0.0090 QR	4°-46′	+0.0007 FR	−0.0007 FR

1. Values in table due to force Q are based on Walls, Bins, and Grain Elevators *by M.S. Ketchum, McGraw-Hill Book Co., 1929. Coefficients have been modified for force Q rather than weight W.*
2. Values in table due to force F are based on "Stress Analysis of the Balcony Girder of Elevated Water Tanks Under Earthquake Loads" by W.E. Black; Chicago Bridge and Iron Co., 1941.

$F_{1,2}$ = resisting force in tie rod, panel force, lb
f_b = bending stress, psi
R = radius of column circle, in.
R_t = torsional resistance factor
Q = equivalent vertical force at each support due to dead weight and overturning moment, lb
q = uniform vertical load on ring beam, lb/in.
q_t = tangential shear, lb/in.
W = operating weight, lb
β = location of maximum torsional moment from column, degrees
I_x, I_y = moment of inertia, in.^{4}
τ = torsional shear stress, psi
B_p = bearing pressure, psi
J = polar moment of inertia, in.^{4}
M = bending moment in base plate due to bearing pressure, in.-lb
M_B = horizontal bending moment between posts due to force F, in.-lb
M_c = vertical bending moment between posts due to force Q, in.-lb
M_o = overturning moment of vessel at base of ring beam, in.-lb
M_p = horizontal bending moment at posts due to force F, in.-lb
M_s = vertical bending moment at posts due to force Q, in.-lb
M_T = torsional moment at distance β from post, in.-lb

Formulas

$$M_s = \frac{WR}{N}\left[\frac{1}{\theta} - \frac{0.5}{\tan\theta/2}\right]$$

$$M_c = M_s \cos\frac{\theta}{2} + \frac{WR}{2N}\left[\sin\frac{\theta}{2} - \frac{2\sin^2\theta/4}{\theta/2}\right]$$

$$M_T = (-)M_s \sin\beta - \frac{WR}{2N}(1-\cos\beta) + \frac{WR\beta}{2\pi}\left(1 - \frac{\sin\beta}{\beta}\right)$$

$$q_t = \frac{F\sin\phi}{\pi R}$$

$$F_{1,2\ldots} = \frac{2F\sin\alpha_n}{N}$$

F_n is maximum where $\alpha = 90°$ since $\sin 90° = 1$.

$$q = (-)\frac{W}{\pi D} \pm \frac{4M_o}{\pi D^2}$$

$$Q = \frac{\pi D q}{N}$$

Load Diagrams

Vertical Forces on Ring Beam

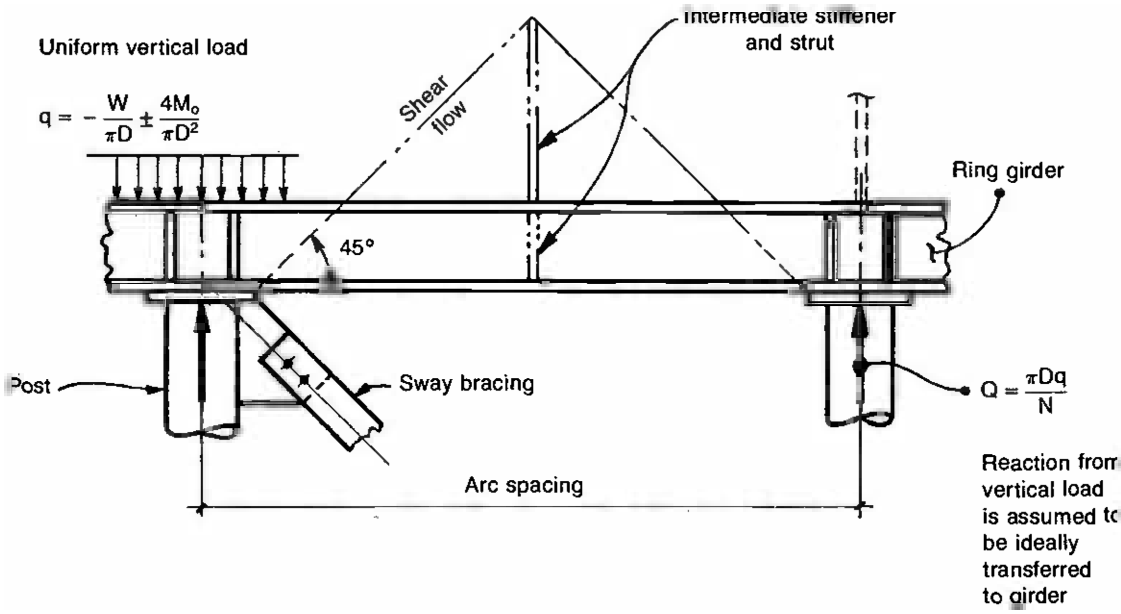


Figure 6-6. Loading diagram for a ring girder: vertical forces on a ring beam.

Horizontal Forces on Ring Beam In the analysis for in-plane bending moment and thrust, the wind or seismic force is assumed to be transferred to the girder by a sine-distributed tangential shear. (See Figure 6-7.) These loads are resisted by the horizontal reaction components of the sway bracing as shown in Figure 6-8.

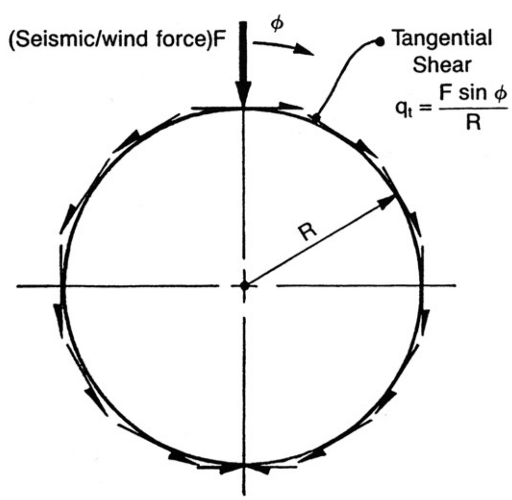


Figure 6-7. Loading diagram for a ring girder: shell to beam.

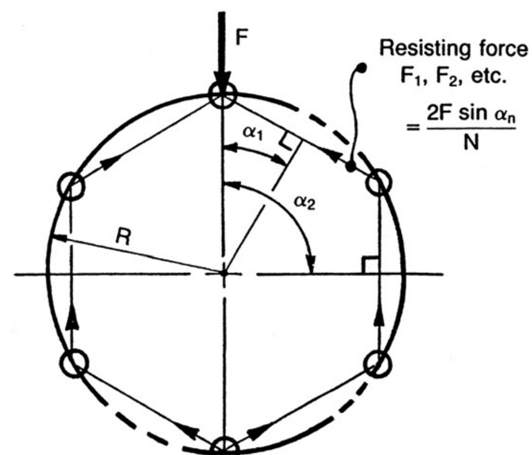


Figure 6-8. Loading diagram for a ring girder: support structure to beam.

Procedure

- *Determine loads q and Q.*

$$q = (-)\frac{W}{D} \pm \frac{4M_o}{\pi D^2}$$

$$Q = \frac{\pi Dq}{N}$$

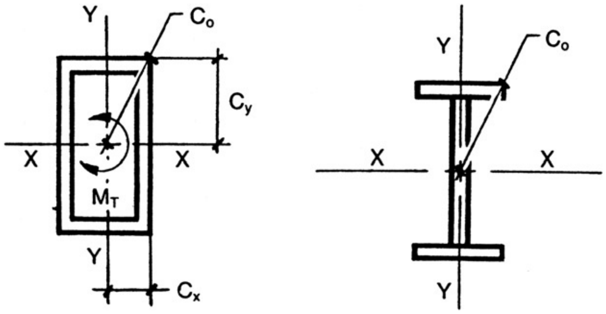


Figure 6-9. Axis and distance of extreme fibers of typical beam sections.

- *Determine bending moments in ring.*
  Note: All coefficients are from Table 6-2.

$$M_s = \text{coefficient} \times QR$$
$$M_c = \text{coefficient} \times QR$$
$$M_T = \text{coefficient} \times QR$$
$$M_P = \text{coefficient} \times FR$$
$$M_B = \text{coefficient} \times FR$$

- *Determine properties of ring.*

For torsion the formula for shear stress, τ, is

$$\tau = \frac{M_T C_o}{J}$$

where J = Polar moment of inertia, in.^{4}
$= I_x + I_y$
C_o = Distance to extreme fiber, in.

Note: Box sections are best for resisting torsion.

An alternate procedure is suggested by Blodgett in *Design of Welded Structures* for substituting a torsional resistance factor, R_t, for the polar moment of inertia in the equation for stress. The torsional resistance factor, R_t, is determined by dividing up the composite section into its component parts, finding the properties of these components, and adding the individual properties to obtain the sum. An example is shown in Figure 6-10.

R_t for any rectangular section $= \gamma bd^3$. See Table 6-3 for γ.

- *Stresses in beam.*

Note: Bending is maximum at the posts. Torsion is maximum at β.

$$f_{bx} = \frac{M_s C_y}{I_x}$$

$$f_{by} = \frac{M_P C_x}{I_y}$$

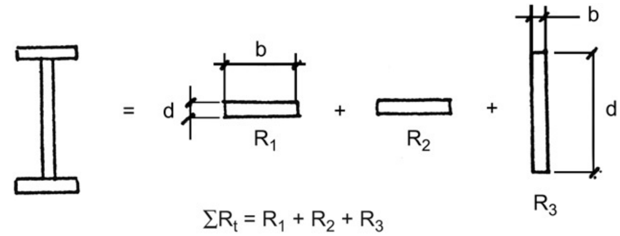


Figure 6-10. Determination of value R_t for typical section.

Table 6-3
Values of coefficient γ

b/d	γ
1.0	0.141
1.5	0.196
1.75	0.214
2.0	0.229
2.5	0.249
3.0	0.263
4.0	0.281
6.0	0.299
8.0	0.307
10.8	0.313
∞	0.333

Reprinted by permission of the James F. Lincoln Arc Welding Foundation.

$$\tau = \frac{M_T C_o}{\sum R_t}$$

- *Additional bending in base plate.*

Additional bending occurs in base plate due to localized bearing of post on ring.

Bearing pressure, B_p, psi

$$B_p = \frac{Q}{A} \pm$$

where A = assumed contact area, area of cap plate or cross-sectional area of post. See Figure 6-11. Assume reaction is evenly distributed over the contact area.

ℓ = Cantilever, in.
L = Semifixed span, in.

Note: Maximum bending is at center of base plate.

- *Moment for cantilever portion.*

$$M = \frac{B_p \ell^2}{2}$$

- *Moment for semifixed span.*

$$M = \frac{B_p L^2}{10}$$

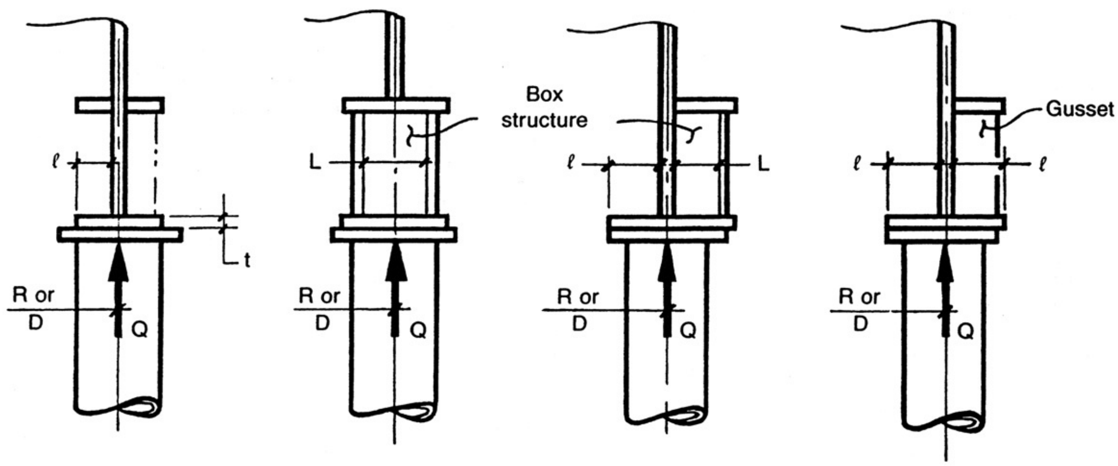


Figure 6-11. Dimensions and loadings for various ring girders.

- *Bending stress, f_b.*

$$f_b = \frac{6M}{t^2}$$

Notes

1. The shell of a column-supported tank, vessel, or bin is considered to be a "circular girder" or "ring beam" uniformly loaded over its periphery and supported by columns equally spaced on the ring circumference.
2. The ring beam (girder) is subjected to compression, bending, and torsion due to the weight of the tank and contents and horizontal wind or seismic force.
3. The maximum bending moment occurs at the supports.
4. The torsional moment M_T will be 0 at the supports and maximum at angular distance β away from supports.

Procedure 6-4: Design of Vessels with Refractory Linings [9–12]

Vessel engineers must eventually become familiar with refractory materials, techniques, strategies and installation in order to properly specify, analyze and check vendor designs for refractory lined vessels, equipment and components. Refractory lining is utilized as a heat barrier, insulator, or as an abrasion resistant lining, or both. Dual component linings are utilized for insulating and abrasion resistance.

Refractory lined equipment and vessels are some of the most interesting applications encountered in the engineering field. During the past 80 years engineers have been perfecting systems where they can connect a layer of ceramic material on steel to protect the steel from effects of high temperature and abrasion. This unique application allows both materials to do what they do best. The alternative, in the case of high temperature, is to specify an exotic alloy. Not only is this more expensive, but even the most exotic alloys available for these applications are only good to about 1500°F. While there are metals that are used up to 2800°F, such as moly used for heat shields, these alloys are not suitable for building vessels, heaters and gasifiers. Above this temperature, you must use some type of refractory barrier to resist the heat.

Refractory materials can be attached to the skin by a variety of anchorage methods. The refractory itself may be put into place by casting, vib-casting, gunniting or ram packed. Anchorage methods, the metal parts that hold the refractory to the metal skin, can be steer bar, wavy V, hex mesh, S bar and various types of punch tabs.

The combination of refractory attached to a steel skin is an unholy alliance, but have some synergies that make the combination work. For example, refractory is good in compression and steel is good in tension. Refractory and metal have drastically different coefficients of thermal expansion. Since the refractory must be connected in some fashion to the steel, the two materials grow at different rates. The result is that the refractory always cracks. This too works to the advantage of the composite system. The cracks in the refractory initiated during cooling, will "seal up" once heated again. The hot face is so much hotter than the cold face that the cracks seal up during operation and prevent the heat from propagating to the skin.

Every refractory has a limited service temperature, or maximum temperature. Above these temperatures brick linings are used. The cutoff of refractory in lieu of brick is about 1800°F. Bricks have the capacity to seal and return to shape after heating without spalling. Solid monolithic linings do not. In addition, bricks can be made in thicker sections than can be cast from refractory.

The cracks formed in monolithic linings above these temperatures are too severe for the material to recover. In the early day of FCC units, the regenerators and reactors were brick lined, like furnaces, because the technology of gunnited and cast linings was not capable of withstanding the service. Brick linings are still utilized in a variety of applications. Shapes are available to conform to cylindrical shapes of containers most commonly used in these vessels and equipment.

In addition to heat resistance and insulating, refractory can provide some excellent abrasion resistance as well. This situation led to the development of the dual component lining used for the past 50 to 60 years. This concept was born because of the properties of the refractory materials themselves. In general, the lighter the refractory, the better its insulating properties but the worse its erosion resistance. The reverse is true for denser, heavier refractories. The denser the refractory, the higher the thermal conductivity, the poorer its insulating capacity but the better its erosion resistance. Early designers quickly arrived at a solution to this problem where insulating and abrasion resistance were both required. The solution was to have a layer of light insulating refractory with a top layer of dense, abrasion resistance refractory.

This combination was usually accomplished by attaching pins and anchors to the skin. The anchors supported the refractory, while the pins extended to the surface where hex mesh was attached to the pins by welding or other mechanical means. The dense, hard abrasion resisting material could be much thinner. The typical application was 3-4" of soft insulating refractory, say 60 PCF, covered with 1" thick of very dense, 165 PCF material in hex mesh. Each material doing what it was best suited for.

But dual component linings had their drawbacks. They were susceptible to degradation due to vibration. Biscuits would break out of the hex mesh and allow the soft layer underneath to be eroded. In short time the steel skin could also be eroded and lead to repairs and/or shut down.

Today, there is a compromise material used for these applications. They use denser refractories, though not as dense as the 165 PCF material through the entire thickness. It has good abrasion resistance properties, and fair insulation refractories. It weighs about 135 PCF and is vib-cast into position. A similar product is made for casting. The material is mixed with stainless steel needles, approximately 2-4 lbs per 60 lbs of refractory, to give the material added strength. This material has a higher thermal conductivity than the insulating castables utilized in the dual component system and therefore you have higher shell temperatures, and greater growth. This additional growth must be designed for in the overall supports and guidance of the system. Shell temperatures are in the range of 100 to 200°F higher than dual component systems.

Vib-cast materials must be cast with an internal form that must be removed later. This is costly but necessary. The materials are cast with very little water so must be vibrated during casting to remove air bubbles to ensure a monolithic lining without voids. This material has been perfected as an alternative to dual component systems without the inherent weaknesses of the dual component system, that is, the loss of biscuits due to vibration and wear.

Lining equipment and vessels with refractory is an economic alternative that will undoubtedly be around for a long time. Thus it behooves the engineers in our industry to familiarize themselves with the properties and techniques used in the industry today, to better serve the needs of industry tomorrow.

The circular cross section of vessels and stacks provide ideal shapes for the supporting and sustaining refractory linings from a stress standpoint. There are a variety of stresses developed in the lining itself as well as stresses induced in the steel containment shell. Compressive stresses are developed in the lining and are a natural result of the temperature gradient. These compressive stresses help to keep the lining in position during operation. This compressive condition is desirable, but must not be so high as to damage the lining.

Several idealized assumptions have been made to simplify the calculation procedure. These are;

1. Assumes steady state conditions
2. Assumes that stress strain relationships are purely elastic
3. Assumes shrinkage varies linearly with temperature
4. Assumes that thermal conductivity and elastic moduli are uniform throughout the lining
5. Circumferential stresses are greater than longitudinal stresses in cylindrical vessels and therefore are the only ones calculated here.

The hot face is in compression during operation and heat-up cycles and in tension during cool-down cycles. The tension and compressive loads vary across the cross-section of the lining during heating and cooling phases. The material at the mean temperature is not necessarily in compression during operation but may be tension or neutral. The hot face stress should always be compressive and is the maximum compressive stress in the lining. If it is not compressive, it can be made to be so either by increasing the thickness of the lining or choosing a refractory with a higher thermal conductivity. Excessive compressive stresses will cause spalling.

The cold face is under tensile stress. This stress often exceeds the allowable tensile stress of the material and cracks must develop to compensate for the excessive tensile stress. The tensile stress is always maximum at the cold face.

Upon cooling of the vessel, the irreversible shrinkage will cause cracks to propogate through the lining. The shrinkage of the hot face amounts to about .001 in/in and crack width at the surface would vary from .01 in to .03 in. These cracks will close early in the reheat cycle and will remain closed under compression at operating temperatures.

Monolithic refractories creep under compressive stress. At stresses much less than the crush strength, the creep rate diminishes with time and approaches zero. Creep occurs under nominally constant stress. When strain instead of stress is held constant, the stress relaxes by the same mechanism that causes creep. Creep rate decreases at lower temperatures and drops off with temperature.

Allowable Refractory Stresses

There is no code or standard that dictates the allowable stresses for refractory materials. Refractory suppliers do not have established criteria for acceptable stress levels. In addition there is very limited experimental information on the behavior of refractory materials under multiaxial stress states.

One criteria that has been used is a factor of safety of 2 based on the minimum specified crush strength of the material at temperature for the allowable compressive stress. The corresponding allowable tensile stress is 40% of the modulus of rupture at 1000°F.

Spalling

Spalling is the technical term used to describe cracking of ceramic materials.

Most materials will crack if rapidly heated and cooled. Metals do not spall because of high ductility and high thermal conductivity. Graphite bricks and crucibles will not spall because of high thermal conductivity. Glass and silica brick spall easily due to high thermal expansion.

However while all refractory materials crack, we hope that they will not spall. Spalling of refractory materials most commonly refers to the fracture of the surface material, usually due to trapped water inside the refractory that converts to steam before the water vapor can escape. This is why strict precautions are taken to dry-out the refractory by baking out any trapped water at a very slow rate. Since refractories are mixed with water to make them viscous enough to cast or gunnite, some amount of water remains trapped in the solidified structure. Like concrete, a certain amount of water is consumed in the chemical reaction of the refractory as it solidifies.

The loose water will become destructive and may cause spalling if not removed. The excess water is removed by slowly baking out the material and allowing the water vapor to slowly dissipate out of the material. This bake out procedure is called the “dry-out” procedure. The material is considered to be safe, and water free once the steel containment reaches 212°F. In effect if the cold face is 212°F, then all of the refractory is at least this temperature as well. This is proof that no free water remains in the material.

Spalling Resistance, R, is given by the following formula;

$R = k\,S / \alpha\,E\,C_p\,p$

k = Thermal conductivity, BTU-in/hr/sq. ft./°F
S = Tensile strength, PSI
α = Coefficient of thermal expansion, in/in/°F $\times 10^{-6}$ from 70°F
E = Modulus of elasticity, PSI
C_p = Specific heat, BTU/Lb/°F
ρ = Density, Lbs/in^3

Refractory Selection

Economy in the use of refractories is governed largely by the selection of the types best suited for a given application. A careful study of equipment design and operating conditions will greatly aid in this selection process.

The type and class of refractories most suitable for a particular case will be determined by such factors as the following:

1. Reason for using refractory;
   a. Thermal protection
   b. Abrasion protection
   c. Both of above
2. Factors related to operation;
   a. Function of the equipment
   b. Properties of contents
   c. Rate of operation
   d. Continuity of operation
   e. Temperature
   f. Rate of change of temperature
   g. Source of heat
   h. Heat release per volume
   i. Rate of heat dissipation
   j. Chemical attack
   k. Velocity of gas stream
   l. Abrasion from dust or particles in moving gas
   m. Impingement of flame for hot spots
   n. Machinery vibration
3. Economic factors;
   a. Delivered cost
   b. Cost of installation
   c. Standard sizes vs special shapes
   d. Service life
   e. Shelf life of product
4. Types of refractory;
   a. Fireclay and high alumina brick
   b. Insulating firebrick
   c. Castable refractory
   d. High alumina plastic refractory
5. Classification of refractory;
   a. Acid
   b. Basic
   c. Neutral
   d. Silica
   e. Special (carbon, Si-C, Zr)
6. Installation;
   a. Brick
   b. Castable
   c. Ramming (plastic)
   d. Vib-cast
   e. Gunnited
7. Anchor type;
   a. Wavy V
   b. Steer bar
   c. "S" bar
   d. Hex mesh
   e. Tyne
   f. Wire
   g. Studs
8. Anchor parameters;
   a. Anchor spacing (1.5 to 3 $\times$ t_L)
   b. Anchor height (2/3 $\times$ t_L)
   c. Anchor pattern (diamond, square, staggered)
   d. Metallurgy
   e. Ceramic
9. Properties of refractory;
   a. Thermal rating
   b. Chemical properties
   c. Abrasion properties
   d. Strength properties
   e. Corrosion/Chemical resistance
   f. Thermal expansion
   g. Thermal shock
   h. Temperature of vitrification
   i. Reversible thermal expansion
   j. Resistance to mechanical stress and impact
   k. Thermal conductivity
   l. Heat capacity
   m. Electrical resistivity
10. Factors relating to design & construction;
    a. Type of equipment
    b. Design and dimensions of walls and arches
    c. Volume for maximum fuel input
    d. Loads imposed upon the lining materials
    e. Conditions of heating
    f. Degree of insulation
    g. Air or water cooling
    h. Type of construction
    i. Methods of bonding and support
    j. Type of bond
    k. Thickness of joints
    l. Nature of bonding material
    m. Provision for thermal expansion

Calculating Heat Loss and Cold Face Temperatures

In any refractory lined system it is required to determine the heat loss and skin temperatures in order to do a proper design of the metal containment (shell) as well as to determine the selection and effectiveness of the refractory itself.

For either case, the designer must first determine the outer skin temperature of the metal containment. This is accomplished by first knowing the following;

1. Refractory material
2. Refractory properties

3. Refractory thickness
4. Internal process temperature
5. Heat transfer coefficients, inside and out
6. Ambient temperature

The heat loss is necessary to validate the selection of the refractory material and thickness. It also will validate the process calculations for heat and material balance.

In any refractory lined system, it is required to determine the thermal expansion of the system in order to determine the stress, support points and restraint points. In order to do this, the designer must accurately determine the skin temperature of the various components. From this data, the thermal expansion of the components is computed, and from this, the stress, depending on support and restraint points.

The structural engineer will need this load data to properly design their foundations and structures to accommodate these loads. The vessel engineer will need this data to determine the local stresses imposed by the installed transfer lines. The piping engineer will need the loads, forces and expansion data to determine if expansion joints are needed in the system. The skin temperatures are needed to determine the allowable stresses of the steel jackets.

The stress in the refractory itself should also be determined based on this analysis to ensure that the refractory material is not overstressed. In any refractory system, the hot face of the refractory expands at a greater rate then the cold wall steel. This keeps the refractory in compression when hot, and seals the cracks. When cooled, cracks will develop in the refractory due to the different temperatures and properties of the steel and refractory.

Before this process can begin, the thermal conductivity (k value) of the selected refractory material must be known. This normally comes from the manufacture of the material and typically published in their literature. However there is considerable variance in test results, depending on how the test is done. Actual results in the field may also vary significantly depending on the installation parameters. Different manufacturers use different testing methods. Testing methods include the following techniques;

1. Hot Wire method (ASTM C1113)
2. Water Calorimeter (ASTM C-201)
3. Water Calorimeter (ASTM C 417)
4. Panel test
5. Dynatech Test

Thermal Conductivity: k

- Units are BTU-in/hr/sq. ft./°F
- A measurement of a materials ability to transmit heat (conduction) or to resist transmitting heat (insulating)
- The lower the value of thermal conductivity, the greater the insulating value. The higher the value, the better conductor.
- Insulators range from .2 to 100. Metals range from 100 to 2700.
- Thermal conductivity for any material increases with temperature. However, the rate of increase, decreases with temperature.
- Thermal conductivity is typically noted as a "k" value.
- Another common term for defining insulating efficiency is the "R" Value.
  $R = t/k$ where t is the thickness of the refractory.

Specific Heat: C_p

- Units are in BTU/Lb/°F
- Definition: The amount of heat energy required to raise 1 Lb of material, 1°F. It is a measure of the amount of heat that can be stored in a material.
- The specific heat of water is 1. Air is .24. Metals range from .03 to .12. Refractory materials are in the range of .2 to .27
- Therefore refractory materials can store much more heat per pound than metals and are about like air in terms of thermal storage.

Refractory Failures and Potential Causes of Hot Spots

The following list are some potential causes of refractory failure, cracking and subsequent hot spots;

1. Refractory spalling; Spalling can be caused by excessive moisture in the material during heating. Too rapid of heat-up or cool down cycles. Too high of thermal gradient across the lining due to improper design, either too thick a lining or too low a thermal conductivity. This case leads to excessive hot face compression.

2. Poor refractory installation
3. Poor refractory material
4. Excessive deflection or flexing of the steel shell due to pressure, surge or thermal stresses
5. Differential expansion
6. Excessive thermal gradient
7. Upsets or excursions leading to rapid heating or cooling rates. These should be limited to about 100°F/hour
8. Upsets or excursions leading to temperatures near or exceeding the maximum service temperature.
9. Poor design details
10. Poor refractory selection
11. Improper curing or dry out rates
12. Poor field joints
13. Temperature differential
14. Incorrect anchorage system
15. Vibration
16. Anchor failure

Refractory Lined Equipment

1. Furnaces
2. Boilers
3. Fired Heaters
4. Incinerators
5. FCC Transfer Lines
6. FCC Flue Gas Lines
7. Expansion Joints
8. Stacks
9. Stills
10. Retorts
11. Dutch Ovens
12. Kilns
13. Sulfur Plants
    a. Converters
    b. Reaction Furnace
    c. Waste Heat Boilers
    d. Hydrogenation Reactor

Refractory Lined Vessels

1. FCC Vessels
2. RFCC Vessels
3. Flexicoker Vessels
4. Fluid Coker Vessels
5. Flexicracking
6. Reactors
7. Secondary Reformers
8. Gasifiers
9. Rheniformer Reactors
10. Third Stage Separators
11. Adiabatic Reformer
12. Orifice Chambers

General Refractory Notes

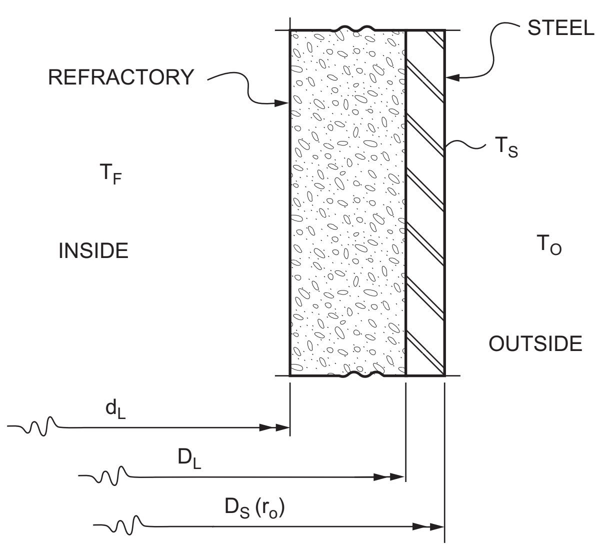
- Once the hot spots have occurred there is obviously a heat leak path to the vessel wall. The subsequent heating of the shell locally also affects the anchors. Since the anchors are made of stainless steel, they grow more then the shell and therefore relax their grip on the refractory. This in turn allows the gap between the shell and refractory to grow.
- Refractory failures are categorized as either tension or compression failures. These failures can result from bending or pure tension/compression loads. In a tension failure the crack is initiated and grows. A "cold joint" is the preferred fix for a tension failure.
- A compression failure will tend to pull the lining away from the wall. A flexible joint with ceramic fiber is the good solution for this type of failure.
- During operation the hot face is in compression varying through the thickness to tension against the steel shell. This is caused by thermal expansion of the material and thermal gradient forces developed internally.
- During the cooling cycle the hot face will be in tension. If the cooling cycle is too rapid or the anchoring too rigid, then the tensile stress of the material becomes critical in resisting cracking.
- Due to low tensile strength, cracking occurs at early stages of load cycles, which ultimately results in load redistribution.
- Temperature loading such as heat-up, cool-down and holding periods at lower temperatures results in stress cycling.
- Refractory properties are non-linear.
- Compressive strength is practically independent of temperature while tensile strength is very dependent on temperature.
- Refractory material undergoes a permanent change in volume due to both loss of moisture during the dryout cycle as well as a change in the chemical structure. The effects of moisture loss as well as chemical metamorphosis are irreversible.

- During initial heating, the steel shell has a tendency to want to pull away from the refractory. The cooler the shell the less the impact on the refractory. The cooler shell tends to hold the refractory in compression longer.
- The use of holding periods during the heat-up and cool-down cycles results in relaxation of compressive stresses due to creep. However this same creep may introduce cracks once the lining is cooled off.
- The two most important effects on refractory linings are creep and shrinkage.
- Optimum anchor spacing is 1.5 – 3 times the thickness of the lining
- Optimum anchor depth is approximately 2/3 of the lining thickness.

Calculate Skin Temperature for Refractory Lined Components

Notation

h = Combined convection and radiation coefficient, outside BTU/Hr/Ft^2/ ^{o}F
K_L = Thermal conductivity of refractory at average temperature of layer BTU-in/Hr/Ft^2/°F
L_n = Log base n
T_s = Skin temperature, outside, ^{o}F
T_F = Internal process temperature, ^{o}F
T_O = Outside ambient temperature, ^{o}F



Cross section of shell

Calculation. Solve the following equation for T_S;

$$(T_S - T_O)hr_o = \frac{(T_F - T_S)K_L}{L_n(D_L/d_L)}$$

Table 6-4
Values of K_L, BTU-in/Hr/Ft^2/°F

Material	1000°F	1500°F	2000°F
RS-3	1.65	1.80	
RS-7	2.35	2.56	
RS-17EC	6.30	6.90	7.70

Sample Problem

$T_O = 20°F$
$T_F = 1425°F$
$h = 4.0$ BTU/Hr/Ft^2/°F
$d_L = 312$ inches
$D_L = 320$ inches
$D_S = 323.5$ inches
$r_o = 161.75$ inches
$K_L = 2.56$ (RS-7 @ 1500^{o}F) BTU-in/Hr/Ft^2/°F
$D_L/d_L = 1.0256$

Solve for T_S …

$$(T_S - 20)4.0(161.75) = ((1425 - T_S)2.56)/L_n\ 1.0256$$

$$16.35(T_S - 20) = 3648 - 2.56\ T_S$$

$$16.35\ T_S - 327.09 = 3648 - 2.56\ T_S$$

$$18.91\ T_S = 3975.09$$

$$T_S = 210°F$$

Notes

1. To determine worst case, assume outside temperature as summer, no wind
2. The value for h, temperature coefficient, is temperature dependent. Therefore an initial temperature must be assumed
3. For multiple layers of refractory, calculate each layer separately.

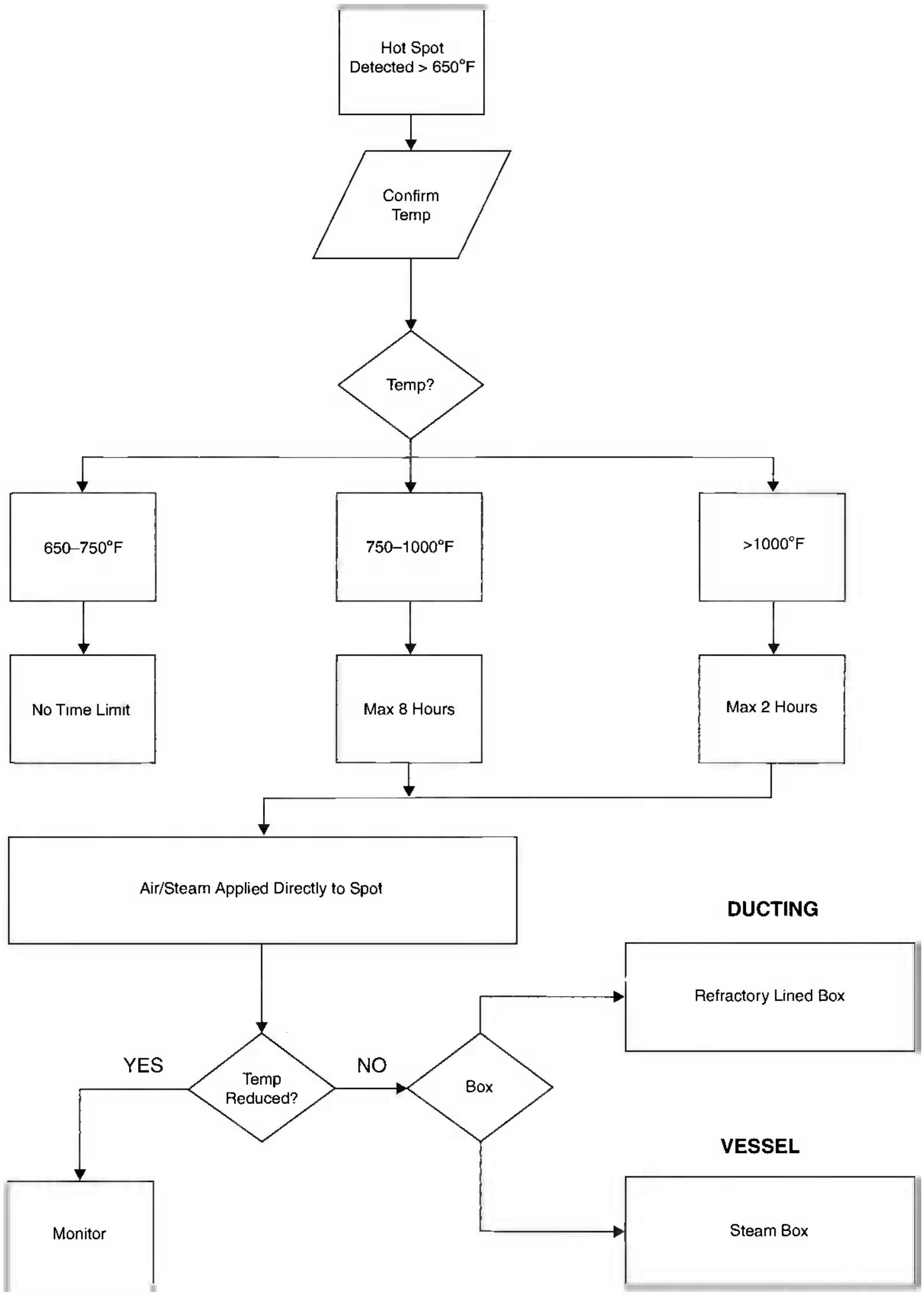


Figure 6-12. Hot spot decision tree.

Notation

Shell Properties

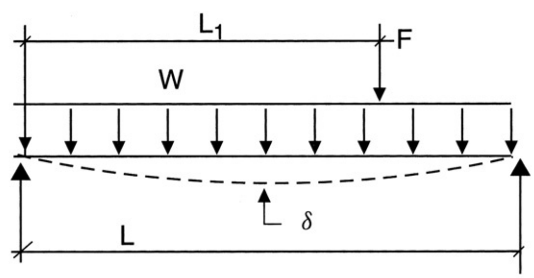
D = shell ID, in.
D_s = shell OD, in.
E_s = modulus of elasticity, shell, psi
I_s = moment of inertia, shell, in.^{4}
K_s = thermal conductivity, shell, BTU-in/hr/ft^2/°F
t_s = thickness, shell, in.
W_s = specific density, steel, pcf
α_s = thermal coefficient of expansion, shell, in./in./°F

Refractory Properties

D_L = refractory OD, in.
d_L = refractory ID, in.
E_L = modulus of elasticity, refractory, psi
F_u = allowable compressive stress, refractory, psi
I_L = moment of inertia, refractory, in.^{4}
K_L = thermal conductivity, refractory, BTU-in/hr/ft^2/°F
S_{TS}, S_{TL} = irreversible shrinkage of lining @ temperatures T_S, T_L
t_L = thickness, refractory, in.
W_L = specific density of refractory, pcf
α_L = thermal coefficient of expansion, refractory, in./in./°F
μ_L = Poisson's ratio, refractory

General

E_{eq} = modulus of elasticity of composite section, psi
h_i, h_o = film coefficients, inside or outside, BTU/hr/ft^2/°F
P = internal pressure, psig
Q = heat loss through wall, BTU/hr/ft^2
T_a = temperature, outside ambient, °F
T_c = temperature, outside ambient during construction, °F
T_L = temperature, refractory, mean, °F
T_{L1} = temperature, lining, inside, °F
T_o = temperature, internal operating, °F
T_s = temperature, shell, mean, °F
T_{s1} = temperature, shell, inside, °F
T_{s2} = temperature, shell, outside, °F
W = overall weight, lb
W_{eq} = equivalent specific density, pcf
δ = deflection, in.
ε_ϕ = circumferential strain due to internal pressure, in./in.
$L1$ = thermal expansion, shell, in./in.
$L2$ = thermal expansion, shell, without lining stress, in./in.
$L3$ = mean thermal expansion, in./in.
$L4$ = mean shrinkage, in./in.
$L5$ = net mean unrestrained expansion, in./in.
$L6$ = net differential circumferential expansion, in./in.
σ_{L1} = mean compressive stress, refractory, due to restraint of shell, psi
σ_{L2} = stress differential from mean, refractory, due to thermal expansion gradient, psi
σ_{L3} = stress differential from mean, refractory, at hot face due to shrinkage, psi
σ_{L4} = circumferential stress in refractory, at hot face, psi
σ_{L5} = circumferential stress in refractory, at cold face, psi
σ_{sc} = circumferential stress in shell caused by the lining, psi
σ_ϕ = circumferential stress due to internal pressure, psi



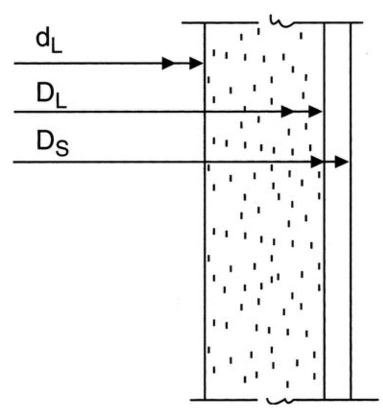


Figure 6-13. Lining dimensions.

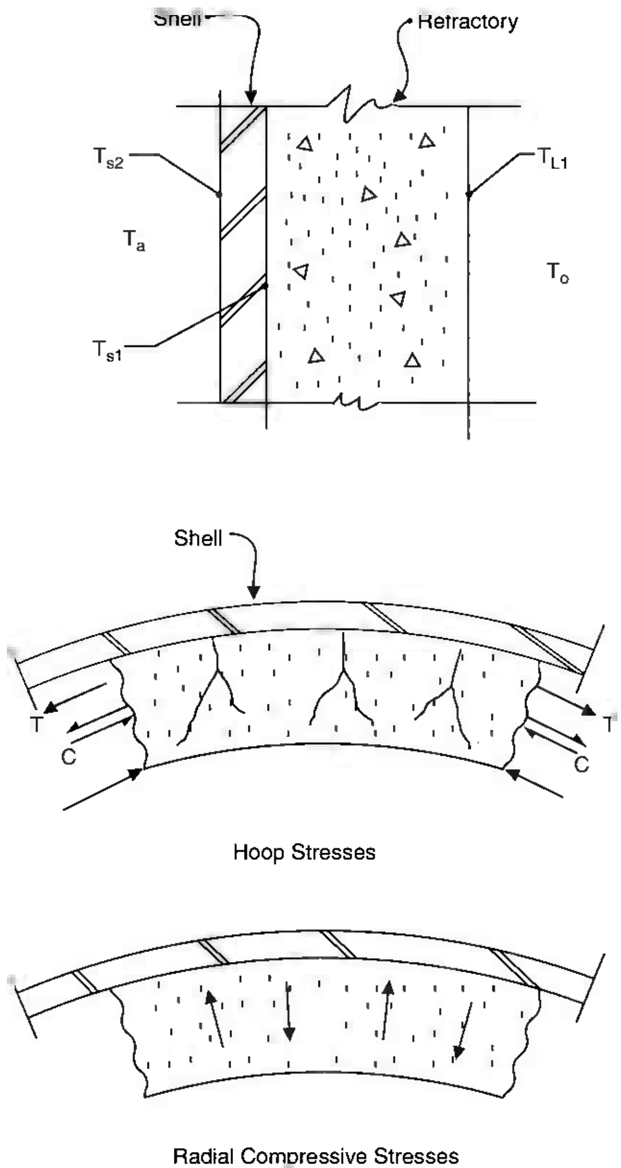


Figure 6-14. Stress/temperatures in wall.

Calculations

Properties of Vessel or Pipe

- *Equivalent specific density, w_{eq}.*

$$w_{eq} = w_s + w_L\left(\frac{D_L^2 - d_L^2}{D_s^2 - D_L^2}\right)$$

- *Moment of inertia.*

Steel: $I_s = \frac{\pi}{64}\left(D_s^4 - D_L^4\right)$

Refractory: $I_L = \frac{\pi}{64}\left(D_L^4 - d_L^4\right)$

Composite: $I = I_s + I_L$

- *Equivalent modulus of elasticity, E_{eq}.*

$$E_{eq} = E_s + \frac{E_L I_L}{I_s}$$

Temperatures

- *Heat loss through wall, Q.*

$$Q = \frac{T_o - T_a}{\frac{1}{h_i} + \frac{t_L}{K_L} + \frac{t_s}{K_s} + \frac{1}{h_o}} \tag{1}$$

- *Outside shell temperature, T_{s1}.*

$$T_{sl} = T_a + Q\left(\frac{1}{h_o}\right) \tag{2}$$

- *Inside shell temperatures, T_{s2}.*

$$T_{s2} = T_{s1} + Q\left(\frac{t_s}{K_s}\right) \tag{3}$$

- *Inside lining temperature, T_{L1}.*

$$T_{L1} = T_{s2} + Q\left(\frac{t_L}{K_L}\right) \tag{4}$$

- *Verification of temperature gradient.*

$$T_o = T_{L1} + Q\left(\frac{1}{h_i}\right) \tag{5}$$

- *Mean shell temperature, T_s.*

$$T_s = 0.5(T_{s1} + T_{s2}) \tag{6}$$

- *Mean lining temperature, T_L.*

$$T_L = 0.5(T_{s1} + T_{L1}) \tag{7}$$

Stresses and Strain

- *Circumferential pressure stress, σ_{ϕ}.*

$$\sigma_{\phi} = \frac{PD}{2t_s} \tag{8}$$

- *Circumferential pressure strain, ε_{ϕ}.*

$$\varepsilon_{\phi} = \frac{0.85\sigma_{\phi}}{E_s} \tag{9}$$

Thermal Expansions

- *Thermal expansion of shell, $\Delta L1$.*

$$\Delta L1 = \alpha_s(T_s - T_c) \tag{10}$$

- *Total circumferential expansion without lining stress, $\Delta L2$.*

$$\Delta L2 = \varepsilon_{\phi} + \Delta L1 \tag{11}$$

- *Mean thermal expansion, $\Delta L3$.*

$$\Delta L3 = \alpha_L(T_L - T_c) \tag{12}$$

- *Mean shrinkage, $\Delta L4A$.*

$$\Delta L4 = 0.5(S_{TS} + S_{TL}) \tag{13}$$

- *Net mean unrestrained expansion, $\Delta L5$.*

$$\Delta L5 = \Delta L3 - \Delta L4 \tag{14}$$

- *Net differential circumferential expansion, $\Delta L6$.*

$$\Delta L6 = \Delta L2 - \Delta L5 \tag{15}$$

Stresses

- *Mean compressive stress in lining due to restraint of shell, σ_{L1}.*

$$\sigma_{L1} = E_L \Delta L6 \left(\frac{E_s t_s}{E_L t_L + E_s t_s} \right) \tag{16}$$

- *Differential stress from mean at hot face and cold face of lining due to thermal expansion, σ_{L2}.*

$$\sigma_{L2} = \frac{(E_L \alpha_L)(T_{L1} - T_{s2})}{2(1 - \mu_L)} \tag{17}$$

- *Differential stress from mean at hot and cold faces of lining due to shrinkage, σ_{L3}.*

$$\sigma_{L3} = \frac{E_L(S_{TL} - S_{TS})}{2(1 - \mu_L)} \tag{18}$$

- *Circumferential stress in lining at hot face, σ_{L4}.*

$$\sigma_{L4} = \sigma_{L1} - \sigma_{L2} + \sigma_{L3} \tag{19}$$

- *Circumferential stress in lining at cold face, σ_{L5}.*

$$\sigma_{L5} = \sigma_{L1} + \sigma_{L2} - \sigma_{L3} \tag{20}$$

- *Circumferential stress in shell caused by lining, σ_{sc}.*

$$\sigma_{sc} = -\sigma_{L1} \left(\frac{t_L}{t_s} \right) \tag{21}$$

Stress and Deflection Due to External loads

- *Uniform load, w.*

$$w = \frac{W}{L}$$

- *Deflection due to dead weight alone, δ.*

$$\delta = \frac{5wL^4}{384E_{eq}I}$$

- *Deflection due to concentrated load, δ.*

$$X = \frac{L_1}{L}$$

$$\delta = \frac{FL_1^3}{3E_{eq}I} \left(\frac{3 - X}{2X} \right)$$

Table 6-5
Properties of refractory materials

Properties	At Temperature (°F)	Material: AA-22S	RS-3	RS-6	RS-7	RS-17EC
Modulus of elasticity, E (10^6 psi)	230	47	2.1	4.1	3.7	18.9
	500	35	1.5	2.94	2.7	16.8
	1000	16.5	0.84	1.62	1.5	15.5
	1500	7.9	0.5	0.93	0.8	14.1
Density, d (pcf)		170	60	75–85	85–95	130–135
Thermal conductivity, K (BTU-in/hr/ft^2/°F)	500		1.7	2.7	2.5	10
	1000	10.3	1.65	2.85	2.8	6.3
	1500	10.4	1.8	3	3.2	6.9
	2000	10.6		3.2	2	7.7
Coefficient of thermal expansion (10^{-6}in./in./°F)		4.7	4.4		4.7	3.5
Poisson's ratio						0.16
Specific heat (BTU/lb/°F)						0.24
% Permanent linear change	1500	−0.1 TO −0.5	−0.3 TO −0.7	−0.1 TO −0.3	−0.2 TO −0.4	−0.1 TO −0.3
	2000	−0.4 TO −1.1	−0.5 TO −1.1	−0.8 TO −1.2	−0.4 TO −0.6	−0.1 TO −0.3
Modulus of rupture (psi)	1000	1400	100	200	200–300	1500–1900
	1500	1400–2200	100–200	200–500	300–700	1400–1800
	2000		150–250	200–500	200–500	
Cold crush strength (psi)	1000	8000–12000	300	1500	600–1000	9000–12000
	1500	7500–10000	300–600	1500–1800	700–1100	8000–11000
	2000	7000–10000	500–800	1200–1600	600–1000	9000–12000
Allowable compressive stress (psi)	1000	4000	150	750	400	5000
Allowable tensile stress (psi)	1000	560	40	80	100	680

Table 6-6
Given input for sample problems

Shell Properties			Refractory Properties		
Item	Case 1	Case 2	Item	Case 1	Case 2
D	360 in.	374 in.	t_L	4 in.	4 in.
t_s	0.5 in.	1.125 in.	E_L	0.6×10^6	0.8×10^6
E_s	28.5×10^6	27.7×10^6	α_L	4.0×10^{-6}	4.7×10^{-6}
α_s	6.8×10^{-6}	7.07×10^{-6}	K_L	4.4	3.2
K_s	300	331.2	μ_L	0.25	0.2
μ_s	0.3	0.3	σ_{ult}	2000 psi	100 psi
T_a	80°F	−20°F	S_{TS}	0.00028	0.002
T_c	60°F	50°F	S_{TL}	0.00108	−0.00025
T_o	1100°F	1400°F	h_i	40	40
P	12 PSIG	25 PSIG	h_o	4	3.5

Table 6-7
Summary of results for sample problems

Equation	Variable	Case 1	Case 2	Equation	Variable	Case 1	Case 2
1	Q	860	908	12	ΔL_3	2.512×10^{-3}	3.53×10^{-3}
2	T_{s1}	295	239	13	ΔL_4	6.8×10^{-4}	1.75×10^{-3}
3	T_{s2}	296	242	14	ΔL_5	1.832×10^{-3}	-4.7×10^{-4}
4	T_{L1}	1079	1377	15	ΔL_6	-1.02×10^{-4}	1.88×10^{-3}
5	T_o	1100	1400	16	σ_{L1}	−52.4 psi	148.9 psi
6	T_s	296	241	17	σ_{L2}	+/−1251 psi	+/−266 psi
7	T_L	688	810	18	σ_{L3}	+/−320 psi	−112.5 psi
8	σ_ϕ	4320	4155	19	σ_{L4}	−983 psi	−229.6 psi
9	ε_ϕ	1.29×10^{-4}	1.275×10^{-4}	20	σ_{L5}	879 psi	427.5 psi
10	ΔL_1	1.6×10^{-3}	1.28×10^{-3}	21	σ_{sc}	419 psi	−530 psi
11	ΔL_2	1.73×10^{-3}	1.41×10^{-3}				

Procedure 6-5: Vibration of Tall Towers and Stacks [13–23]

Tall cylindrical stacks and towers may be susceptible to wind-induced oscillations as a result of vortex shedding. This phenomenon, often referred to as *dynamic instability*, has resulted in severe oscillations, excessive deflections, structural damage, and even failure. Once it has been determined that a vessel is dynamically unstable, either the vessel must be redesigned to withstand the effects of wind-induced oscillations or external spoilers must be added to ensure that vortex shedding does not occur.

The deflections resulting from vortex shedding are perpendicular to the direction of wind flow and occur at relatively low wind velocities. When the natural period of vibration of a stack or column coincides with the frequency of vortex shedding, the amplitude of vibration is greatly magnified. The frequency of vortex shedding is related to wind velocity and vessel diameter. The wind velocity at which the frequency of vortex shedding matches the natural period of vibration is called the *critical wind velocity.*

Wind-induced oscillations occur at steady, moderate wind velocities of 20–25 miles per hour. These oscillations commence as the frequency of vortex shedding approaches the natural period of the stack or column and are perpendicular to the prevailing wind. Larger wind velocities contain high-velocity random gusts that reduce the tendency for vortex shedding in a regular periodic manner.

A convenient method of relating to the phenomenon of wind excitation is to equate it to fluid flow around a cylinder. In fact this is the exact case of early discoveries related to submarine periscopes vibrating wildly at certain speeds. At low flow rates, the flow around the cylinder is laminar. As the stream velocity increases, two symmetrical eddies are formed on either side of the cylinder. At higher velocities vortices begin to break off from the main stream, resulting in an imbalance in forces exerted from the split stream. The discharging vortex imparts a fluctuating force that can cause movement in the vessel perpendicular to the direction of the stream.

Historically, vessels have tended to have many fewer incidents of wind-induced vibration than stacks. There is a variety of reasons for this:

1. Relatively thicker walls.
2. Higher first frequency.
3. External attachments, such as ladders, platforms, and piping, that disrupt the wind flow around the vessel.
4. Significantly higher damping due to:
   a. Internal attachments, trays, baffles, etc.
   b. External attachments, ladders, platforms, and piping.
   c. Liquid holdup and sloshing.
   d. Soil.
   e. Foundation.
   f. Shell material.
   g. External insulation.

Damping Mechanisms

Internal linings are also significant for damping vibration; however, most tall, slender columns are not lined, whereas many stacks are. The lining referred to here would be the refractory type of linings, not paint, cladding, or some protective metal coating. It is the damping effect of the concrete that is significant.

Damping is the rate at which material absorbs energy under a cyclical load. The energy is dissipated as heat from internal damping within the system. These energy losses are due to the combined resistances from all of the design features mentioned, i.e., the vessel, contents, foundation, internals, and externals. The combined resistances are known as the *damping factor*.

The total damping factor is a sum of all the individual damping factors. The damping factor is also known by other terms and expressions in the various literature and equations and expressed as a coefficient. Other common terms for the damping factor are *damping coefficient, structural damping coefficient, percent critical damping*, and *material damping ratio.* In this procedure this term is always referred to either as factor D_F or as β.

There are eight potential types of damping that affect a structure's response to vibration. They are divided into three major groups:

Resistance:

Damping from internal attachments, such as trays.
Damping from external attachments, such as ladders, platforms, and installed piping.
Sloshing of internal liquid.

Base support:

Soil.
Foundation.

Energy absorbed by the shell (hysteretic):

Material of shell.
Insulation.
Internal lining.

Karamchandani, Gupta, and Pattabiraman give a detailed account of each of these damping mechanisms for process towers (trayed columns). They estimate the "percent critical damping" at 3% for empty vessels and 5% for operating conditions. The value actually used by most codes is only a fraction of this value.

Design Criteria

Once a vessel has been designed statically, it is necessary to determine if the vessel is susceptible to wind-induced vibration. Historically, the rule of thumb was to do a dynamic wind check only if the vessel L/D ratio exceeded 15 and the POV was greater than 0.4 seconds. This criterion has proven to be unconservative for a number of applications. In addition, if the critical wind velocity, V_c, is greater than 50 mph, then no further investigation is required. Wind speeds in excess of 50 mph always contain gusts that will disrupt uniform vortex shedding.

This criterion was amplified by Zorrilla, who gave additional sets of criteria. Criterion 1 determines if an analysis should be performed. Criterion 2 determines if the vessel is to be considered stable or unstable. Criterion 3 involves parameters for the first two criteria.

Criterion 1

- If $W/LD_r^2 \leq 20$, a vibration analysis must be performed.
- If $20 < W/LD_r^2 \leq 25$, a vibration analysis should be performed.
- If $W/LD_r^2 > 25$, a vibration analysis need not be performed.

Criterion 2

- If $W\delta/LD_r^2 \leq 0.75$, the vessel is unstable.
- If $0.75 < W\delta/LD_r^2 \leq 0.95$, the vessel is probably unstable.
- If $W\delta/LD_r^2 > 0.95$, the structure is stable.

Criterion 3

This criterion must be met for Criteria 1 and 2 to be valid.

- $L_C/L < 0.5$
- $10{,}000\ D_r < 8$
- $W/W_s < 6$
- $V_c > 50$ mph; vessel is stable and further analysis need not be performed.

Criterion 4

An alternative criterion is given in ASME STS-1-2011, "Steel Stacks". This standard is written specifically for stacks. The criterion listed in this standard calculates a "critical vortex shedding velocity", V_{zcrit}. This value is then compared to the critical wind speed, V_c, and a decision made.

- If $V_c < V_{zcrit}$, vortex shedding loads shall be calculated.
- If $V_{zcrit} < V_c < 1.2\ V_{zcrit}$, vortex shedding loads shall be calculated; however, the loads may be reduced by a factor of $(V_{zcrit}/V_c)^2$.
- If $V_c > 1.2\ V_{zcrit}$, vortex shedding may be ignored.

Equations are given for calculating all of the associated loads and forces for the analysis. This procedure utilizes the combination of two components of β, one β for aerodynamic damping, β_a, and one for steel damping, β_s. The two values are combined to determine the overall β.

Criterion 5

An alternative criterion is also given in the Canadian Building Code, NBC. The procedure for evaluating effects of vortex shedding can be approximated by a static force acting over the top third of the vessel or stack. An equation is given for this value, F_L, and shown in this procedure.

Dynamic Analysis

If the vessel is determined by this criterion to be unstable, then there are two options:

a. The vessel must be redesigned to withstand the effects of wind-induced vibration such that dynamic deflection is less than 6 in./100 ft of height.
b. Design modifications must be implemented such that wind-induced oscillations do not occur.

Design Modifications

The following design modifications may be made to the vessel to eliminate vortex shedding:

a. Add thickness to bottom shell courses and skirt to increase stiffness and raise the natural frequency.
b. Modify the top diameter where possible.
c. For stacks, add helical strakes to the top third of the stack only as a last resort. Spoilers or strakes should protrude beyond the stack diameter by a distance of d/12 but not less than 2 in.
d. Cross-brace vessels together.
e. Add guy cables or wires to grade.
f. Add internal linings.
g. Reduce vessel below dynamic criteria.

Precautions

The following precautions should be taken.

a. Include ladders, platforms, and piping in your calculations to more accurately determine the natural frequency.
b. Grout the vessel base as soon as possible after erection while it is most susceptible to wind vibration.
c. Add external attachments as soon as possible after erection to break up vortices.
d. Ensure that tower anchor bolts are tightened as soon as possible after erection.

Definitions

Critical wind velocity: The velocity at which the frequency of vortex shedding matches one of the normal modes of vibration.

Logarithmic decrement: A measure of the ability of the overall structure (vessel, foundation, insulation, contents, soil, lining, and internal and external attachments) to dissipate energy during vibration. The logarithmic ratio of two successive amplitudes of a damped, freely vibrating structure or the percentage decay per cycle.

Static deflection: Deflection due to wind or earthquake in the direction of load.

Dynamic deflection: Deflection due to vortex shedding perpendicular to the direction of the wind.

Notes

1. See procedure 3-3 to determine a vessel's fundamental period of vibration (POV).
2. See procedure 3-4 to determine static deflection.
3. Vessel should be checked in the empty and operating conditions with the vessel fully corroded.

4. Concentrated eccentric loads can be converted to an additional equivalent uniform wind load.
5. L/D ratios for multidiameter columns can be determined as shown in Note 8.
6. A fatigue evaluation should be performed for any vessel susceptible to vortex shedding. A vessel with a POV of 1 second and subjected to 3 hours per day for 30 years would experience 120 million cycles.
7. This procedure is for cylindrical stacks or vessels only, mounted at grade. It is not appropriate for tapered stacks or vessels. There is a detailed procedure in ASME STS-1 for tapered stacks. Multidiameter columns and stacks can be evaluated by the methods shown. This procedure also does not account for multiple vessels or stacks in a row.
8. L/D ratios can be approximated as follows:

$$\frac{L_1D_1 + L_2D_2 + \ldots + L_xD_x + L_{sk}D_{sk}}{D_x^2}$$

where quantities L_xD_x are calculated from the top down.

Table 6-8
Summary of critical damping

		Case 1: Empty			Case 2: Operating		
			β			β	
Item	Description	δ		%	δ		%
1	Material	0.07	0.011	1.1	0.07	0.011	1.1
2	Insulation	0.0063	0.001	0.1	0.0063	0.001	0.1
3	Soil	0.125	0.02	2	0.125	0.02	2
4	Attachments	0.0063	0.001	0.1	0.0063	0.001	0.1
5	Liquid				0.094	0.015	1.5
	Total	0.208	0.033	3.34	0.302	0.048	4.84

Sources: Ref. 13.

Table 6-9
Logarithmic decrement, δ

		Soil Type		
Type	Description	Soft (1)	Medium (2)	Rock/Piles (3)
1	Steel vessel	0.1	0.05	0.03
2	Tower with internals	0.13	0.08	0.035–0.05
3	Tower internals and operations	0.1	0.05	0.035
4	Tower, refractory lined	0.3	0.1	0.04–0.05
5	Tower, full of water	0.3	0.1	0.07
6	Unlined stack	0.1	0.05	0.035
7	Lined stack	0.3	0.1	0.07

Notes:

1. Soft soils $B_p < 1500$ psi, $\beta_F = 0.07$.
2. Medium soils, 1500 psi $< B_p < 3000$ psi, $\beta_F = 0.03$.
3. Pile foundation, rock, or stiff soils, $\beta_F = 0.005$.

Table 6-10
Values of β

	Standard						
Soil Type	ASCE 7-95	Major Oil Co	ASME STS-1	NBC	Misc. Papers	Gupta	Compress
Soft Medium Rock/Piles	0.005 0.01 0.005	See Table 6-11; 0.004-0.0127	See Note 1 and Table 6-12	Unlined = 0.0016–0.008 Lined = 0.0048–0.0095 See Note 2	See Note 3	See Table 6-8; 0.03–0.05	Default = 2% (0.02)

Table 6-11
β Values per a major oil company

Equipment Description	β
Vessels:	
1. Empty without internals	0.0048
2. Empty with tray spacing > 5 ft.	0.0051
3. Empty with tray spacing 3–5 ft	0.0056
4. Empty with tray spacing < 3 ft	0.0064
5. Operating with tray spacing 5–8 ft	0.0116
6. Operating with tray spacing < 5 ft	0.0127
7. Vessel full of liquid	0.018
Stacks mounted at grade	0.004–0.008

Table 6-12
Values of β_s per ASME STS-1

	Damping Value	
Type of Stack	Rigid Support	Elastic Support
Unlined	0.002	0.004
Lined	0.003	0.006

Notes

1.

$$\beta = \beta_a + \beta_s$$

$$\beta_a = \frac{C_f \cdot \rho \cdot D_r \cdot V_z}{4 \cdot \pi \cdot w_r \cdot f_1}$$

$$\beta_s = \text{from table}$$

2. For lined and unlined stacks only!

$$\beta = \frac{\delta}{2\pi}$$

3.

$$\beta = \frac{w\delta}{D^2} \text{ or } \frac{W\delta}{LD^2}$$

Table 6-13
Coefficient C_f per ASME STS-1

		L/D		
	Surface Texture	1	7	25
$D(q_z)^{0.5} > 2.5$	Smooth	0.5	0.6	0.7
	Rough	0.7	0.8	0.9
	Very rough	0.8	1	1.2
$D(q_z)^{0.5} > 2.5$	All	0.7	0.8	1.2

Table 6-14
Topographic factors per ASME STS-1

Exposure Category	b	α
A	0.64	0.333
B	0.84	0.25
C	1	0.15
D	1.07	0.111

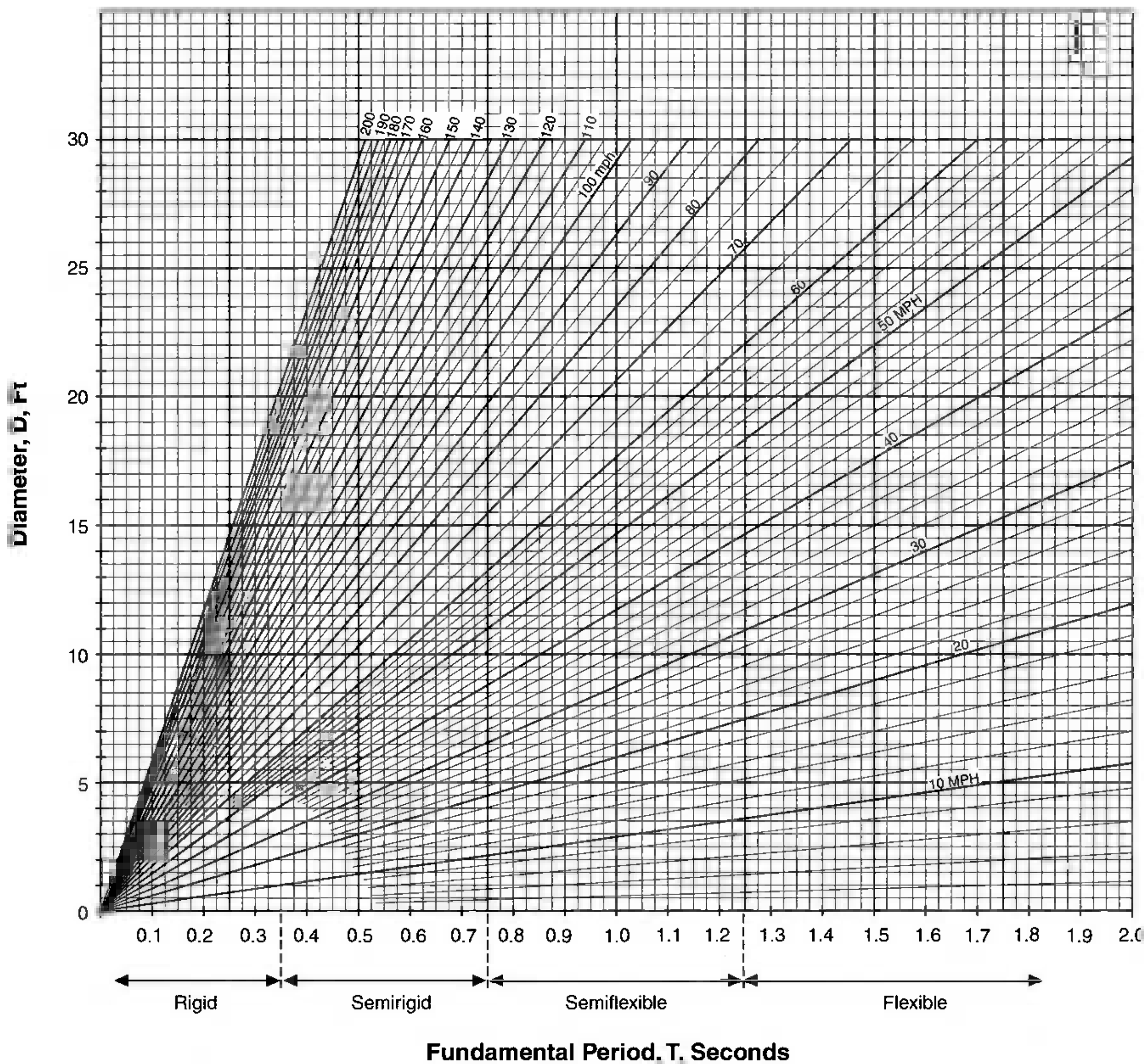


Figure 6-15. Graph of critical wind velocity, V_c.

Notation

B_f = allowable soil bearing pressure, psf
C_f = wind force coefficient, from table
C_1, C_2 = NBC coefficients
D = mean vessel diameter, in.
D_r = average diameter of top third of vessel, ft
E = modulus of elasticity, psi
F_F = fictitious lateral load applied at top tangent line, lb
F_L = equivalent static force acting on top third of vessel or stack, lb
f = fundamental frequency of vibration, Hz (cycles per second)
f_n = frequency of mode n, Hz
f_o = frequency of ovalling of unlined stack, Hz
g = acceleration due to gravity, 386 in./sec^2 or 32 ft/sec^2
I = moment of inertia, shell, in.^{4}
I_F = importance factor, 1.0–1.5
L = overall length of vessel, ft
L_c = Length of cone, ft
M_L = overturning moment due to force F_L, ft-lb
M_S = overturning moment due to seismic, ft-lb
M_R = resultant moment, ft-lb
M_w = overturning moment due to wind, ft-lb
M_{wD} = modified wind moment, ft-lb

q_H = wind velocity pressure, psf, per NBC
q_z = external wind pressure, psf per ASME STS-1
S = Strouhal number, use 0.2
T = period of vibration, sec
t = shell thickness, in.
V = basic wind speed, mph
V_c = critical wind velocity, mph
V_{c1}, V_{c2} = critical wind speeds for modes 1 and 2, mph or fps
V_{co} = critical wind speed for ovalling of stacks, ft/sec
V_r = reference design wind speed, mph, per ASME STS-1
V_z = mean hourly wind speed, ft/sec
V_{zcrit} = mean hourly wind speed at 5/6 L, ft/sec
W = overall weight of vessel, lbs
w = uniform weight of vessel, lb/ft
w_r = uniform weight of top third of vessel, lb/ft
W_s = Weight of base vessel, lbs
α, b = topographic factors per ASME STS-1
β = percent critical damping, damping factor
β_a = aerodynamic damping value
β_f = foundation damping value
β_s = structural damping value
δ = logarithmic decrement
Δ_d = dynamic deflection, perpendicular to direction of wind, in.
Δ_s = static deflection, parallel to direction of wind, in.
ρ = density of air, lb/ft^3 (0.0803) or kg/m^3 (1.2)
λ = aspect ratio, L/D

Miscellaneous Equations

- *Frequency for first three modes, f_n.*

Mode 1: $$f_1 = 0.56\sqrt{\frac{gEI}{wL^4}}$$

Mode 2: $$f_2 = 3.51\sqrt{\frac{gEI}{wL^4}}$$

Mode 3: $$f_3 = 9.82\sqrt{\frac{gEI}{wL^4}}$$

Note: I is in ft^4.

$$I = 0.032D^3t$$

$$f_n = \frac{1}{T}$$

- *Frequency for ovalling, f_o.*

$$f_o = \frac{680t}{D^2}$$

- *Critical wind velocities*:

$$V_c = V_{c1} = \frac{f_1 D}{S} = \frac{D}{ST} = \frac{D}{0.2T}\,(\text{fps})$$

$$V_c = \frac{3.4D}{T}\,(\text{mph})$$

$$V_{c2} = 6.25V_{c1}$$

$$V_{co} = \frac{f_o D}{2S}$$

- *Period of vibration, T, for tall columns and stacks.*

$$T = 1.79\sqrt{\frac{wL^4}{EIg}}$$

where L, D, and t are in feet.

Procedures

Procedure 1: Zorilla Method

Step 1: Calculate structural damping coefficient, β.

$$\beta = \frac{W\delta}{LD_r^2} \text{ or } \beta = \frac{w\delta}{D_r^2}$$

Step 2: Evaluate:

- If $W/LD_r^2 \le 20$, a vibration analysis must be performed.
- If $20 < W/LD_r^2 \le 25$, a vibration analysis should be performed.
- If $W/LD_r^2 > 25$, a vibration analysis need not be performed.
- If $W\delta/LD_r^2 \le 0.75$, the vessel is unstable.
- If $0.75 < W\delta/LD_r^2 \le 0.95$, the vessel is probably unstable.

- If $W\delta/LD_r^2 > 0.95$, the structure is stable.

Step 3: If $\beta < 0.95$, check critical wind velocity, V_c.

$$V_c = \frac{D_r}{TS} = \text{fps}$$

$$V_c = \frac{3.41D_r}{T} = \text{mph}$$

If $V_c > V$, then instability is expected.

Step 4: Calculate dynamic deflection, Δ_d.

$$\Delta_d = \frac{(2.43)(10^{-9})L^5V_c^2}{W\delta D_r}$$

If $\Delta_d < 6$ in./100 ft, then the design is acceptable as is. If $\Delta_d > 6$ in./100 ft, then a "design modification" is required.

Procedure 2: ASME STS-1 Method

Step 1: Calculate damping factor, β.

$$\beta = \beta_a + \beta_S$$

Step 2: Calculate critical wind speed, V_c.

Step 3: Calculate critical vortex shedding velocity, V_{zcrit}.

$$V_{zcrit} = b\left(\frac{Z_{cr}}{33}\right)^{\alpha}\ \frac{22}{15}(V_r)$$

where

$$Z_{cr} = \frac{5L}{6}$$

$$V_r = \frac{V}{I_f}$$

b and α are from table.

Step 4: Evaluate:

- If $V_c < V_{zcrit}$, then vortex shedding loads shall be calculated.
- If $V_{zcrit} < V < 1.2\ V_{zcrit}$, then vortex shedding loads shall be calculated; however, loads may be reduced by a factor of $(V_{zcrit}/V_c)^2$.
- If $V_c > 1.2\ V_{zcrit}$, then vortex shedding may be ignored.

Step 5: To evaluate vortex shedding loads, refer to ASME STS-1, Appendix E.

Procedure 3: NBC

Step 1: Calculate critical wind velocity, V_c. No analysis need be performed if $V_c > V$.

Step 2: Calculate coefficients C_1 and C_2.

- If $\lambda > 16$, then

$$C_1 = 3 \text{ and } C_2 = 0.6$$

- If $\lambda < 16$, then

$$C_1 = \frac{3\sqrt{\lambda}}{4}$$

- If $V_c < 22.37$ mph and $\lambda > 12$, then

$$C_1 = 6 \text{ and } C_2 = 1.2$$

Step 3: If

$\beta > \dfrac{C_2\rho D_r^2}{w_r}$ then no dynamic analysis need be performed.

If

$\beta < \dfrac{C_2\rho D_r^2}{w_r}$ then dynamic analysis should be performed.

Step 4: If a dynamic analysis is required, calculate an equivalent static force to be applied over the top third of the column, F_L.

$$F_L = \frac{C_1 q_H D_r}{\sqrt{\lambda}\sqrt{\dfrac{C_2\rho D_r^2}{w_r}}}$$

Step 5: Determine moment due to force, F_L.

$$M_L = \frac{5F_L L^2}{18}$$

Step 6: Calculate modified wind moment, M_{WD}.

$$M_{WD} = M_W\left(\frac{V_C}{V_W}\right)^2$$

Step 7: Calculate resultant moment, M_R.

$$M_R = \sqrt{M_L^2 + M_{WD}^2}$$

Step 8: If $M_R > M_S$ or M_W, then compute fictitious force, F_F.

$$F_F = \frac{M_R}{L}$$

Step 9: Check vessel with lateral load, F_F, applied at the top tangent line of the vessel. If the stresses are acceptable, the vessel is OK. If the stresses are not acceptable, then the thicknesses must be revised until the stresses are acceptable.

Example No. 1

Given

$$w = 146.5 \text{ kips}$$

$$T = 0.952 \text{ sec}$$

$$S = 0.2$$

$$\delta = 0.08$$

$$D_r = \frac{10 + 6.5}{2} = 8.25 \text{ ft}$$

Soil type: medium.

- *Average weight of top third of column.*

$$\frac{L}{3} = \frac{198}{3} = 66 \text{ ft}$$

$$\frac{W_t}{66} = \frac{35{,}000}{66} = 530 \text{ lb/ft}$$

- *Dynamic check.*

$$\frac{W}{LD_r^2} = \frac{146{,}500}{198(8.25^2)} = 10.87 < 20$$

Therefore an analysis must be performed.

$$\beta = \frac{W\delta}{LD_r^2} = 0.08(10.87) = 0.87$$

Probably stable, proceed.

- *Critical wind speed, V_c.*

$$V_c = \frac{D_r}{TS} = \frac{8.25}{0.952(0.2)} = 43.33 \text{ fps}$$

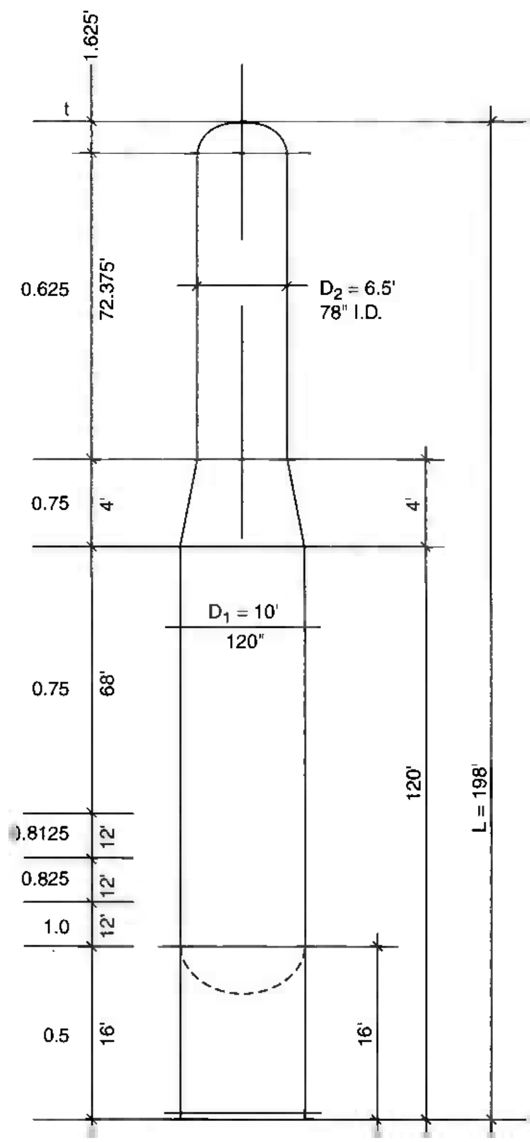
$$43.33 \text{ fps}(0.682) = 29.55 \text{ mph}$$

- *Dynamic deflection, Δ_d.*

$$\Delta_d = \frac{(2.43)\left(10^{-9}\right) L^5 V_c^2}{W\delta D_r}$$

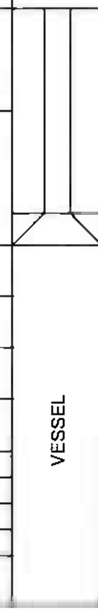
$$\Delta_d = \frac{(2.43)\left(10^{-9}\right) 198^5 \left(29.55^2\right)}{146{,}500\,(0.08)\,8.25} = 6.68 \text{ in.}$$

Dimensions



Example No. 1: Wind Design, Static Deflection, 100 MPH Zone

F_n	A_f	P	H
17,405	395.2	44	160
17,534	416	42.15	120
11,012	280	39.33	100
10,550	280	37.68	80
10,088	280	36.03	60
9430	280	33.68	40
4319	140	30.85	30
2028	70	28.97	25
1962	70	28.03	20
1835	70	26.21	15
4495	180	24.97	0



	H	R_m	t	I
I_7	124	39.313	0.625	119,295
I_6	120	60.375	0.75	518,540
I_5	52	60.375	0.75	518,540
I_4	40	60.4	0.8125	562,450
I_3	28	60.438	0.875	608,844
I_2	16	60.5	1	695,690
I_1	0	60.25	0.5	348,551

Example No. 1: Values for computation of static deflection

Section n	L_n (ft)	L_n (in.)	I_n	L_n^4/I_n	L_n^4/I_{n-1}
1	198	2376	343,551	92,767,217	
2	182	2184	695,690	32,703,540	66,224,596
3	170	2040	606,844	28,539,319	24,894,586
4	158	1896	562,450	22,975,735	21,294,932
5	146	1752	518,540	18,169,967	16,751,453
6	78	936	518,540	1,480,202	1,480,202
7	74	888	119,295	5,212,302	1,199,139
Σ				201,848,283	131,844,908

- Static deflection due to wind, Δ_s.

$$\Delta_s = \left(\sum \frac{L_n^4}{I_n} - \sum \frac{L_n^4}{I_n - 1}\right)\left(\frac{w_{min}}{8E} + 5.5\frac{w_{max} - w_{min}}{60E}\right)$$

$$\Delta_s = (70{,}003{,}375)\left[1.143\left(10^{-7}\right) + 4.44\left(10^{-8}\right)\right] = 11.11 \text{ in.} < 6 \text{ in./100 ft}$$

$$w_{min} = \frac{F_n}{L_n} = \frac{4495}{15} = 300 \text{ lb/ft} = 24.97\text{lb/in.} \qquad E = 27.3\left(10^6\right)\text{psi}$$

$$w_{max} = \frac{17{,}405}{38} = 458 \text{ lb/ft} = 38.2 \text{ lb/in.}$$

Procedure 6-6: Underground Tanks & Vessels

Underground storage tanks or vessels can be buried, or mounted in below ground pits, usually called "sumps". In either case, the design principles are the same. The vessel is subject to flotation if the pit fills with water or the groundwater encroaches on a buried tank. The designer must be sure that the anchorage is strong enough and heavy enough to resist the flotation forces.

The buoyancy of the tank is produced when groundwater encroaches beneath the tank and causes the tank to float. The pressure of flotation (buoyancy) is determined by the displacement of the tank multiplied times the equivalent weight of water of the displacement. This is the same mechanism that allows ships and boats to float. The weight of displacement is greater than the weight of the ship or boat.

The force of flotation is resisted by the weight of the vessel, the overburden (fill), the weight of a deadman (if used) and the weight of any paving over the top of the tank. The vessel itself must be designed to resist the external pressure created by the weight of the overburden. ASME Code, Section VIII, Division 1 method for external pressure should be used for analyzing the shell for the external pressure condition.

This procedure can be used to determine either the depth of burial or the weight of the deadman required to prevent flotation. Underground storage tanks (UST) may be either anchored, or unanchored. If anchored, the anchorage must be sized to resist the buoyancy force. If unanchored, the vessel must be buried deep enough such that the weight of the overburden is greater than the buoyancy force.

Underground storage tanks for gas stations are typically not anchored and do not have foundations. They are simply buried in the ground, resting on a layer of sand. The depth of the overburden over the tank constitutes the greatest force for counteracting flotation. The burial depth of the tank must also allow piping to be sloped to the tank at least 1/8 inch per foot.

Nomenclature

A_r = Area required, in^2
A_S = Area of anchor strap, in^2
A_1 = Area of concrete slab or paving, Ft^2
A_2 = Projected vessel area, Ft^2
A_3 = Area of deadman, Ft^2
A_S = Cross sectional area of strap, in^2
D = Vessel diameter, Ft or in
F = Buoyancy force, Lbs
FS = Factor of Safety
F_T = Allowable tensile stress, PSI
f_T = Stress, tension, PSI
Q = Operating load on one saddle, Lbs
t = Thickness of vessel shell, in
t_d = Thickness of deadman, Ft
t_P = Thickness of concrete pad, Ft
T = Tension force in anchor strap, Lbs
U = Net uplift, Lbs
V_n = Void area in fill for nozzles, Ft^3
V_f = Volume of fill for design, Ft^3
V_{fr} = Volume of fill required, Ft^3
V_V = Volume of vessel, Ft^3
w_C = Uniform weight of concrete, PCF
w_f = Uniform weight of fill, PCF
w_w = Uniform weight of water, PCF
W_{cp} = Weight of concrete slab. Lbs
W_d = Weight of deadman, lbs
W_f = Weight of fill, Lbs
W_{fr} = Weight of fill required, Lbs
W_O = Weight, operating, vessel, Lbs
W_V = Weight, empty, vessel, Lbs
ΣW = Total restraining force, Lbs
$\Sigma W'$ = Total resisting weight, Lbs

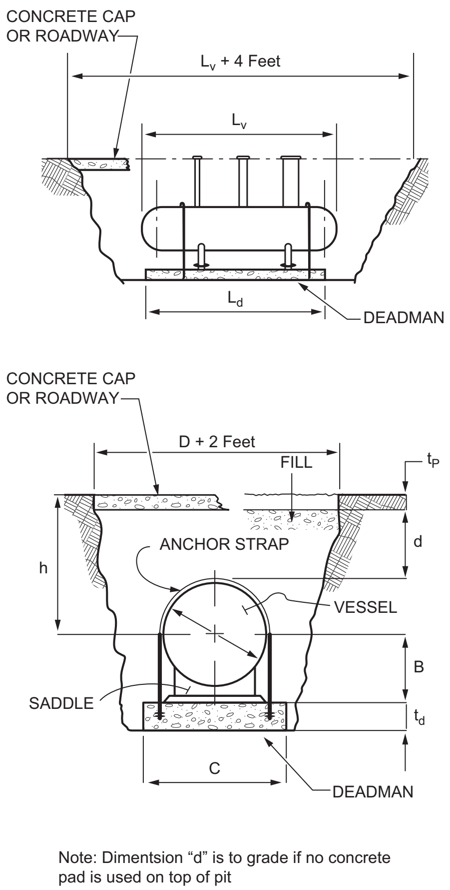


Figure 6-16. Dimensions for buried vessel.

Table 6-15
Weight of materials

Material	Weight (PCF)
Sand & Pea Gravel	60
Reinforced Concrete	90-125
Water	62.4
Clay	40-100
Rock	60-120
Soil	75-85
Steel	497

Buried Vessels

Case 1: With Deadman

Determine the size and weight of the deadman required to counteract the buoyancy forces if required. Determine the size of anchor straps and bolts necessary to resist flotation.

1. Given;

   Volume, $V_V =$
   Weight, $W_V =$
   Uniform weight of fill, $w_f =$
   Diameter, $D =$
   Depth of overburden, $d =$
   Size of concrete slab $=$
   Thickness of concrete slab, $t_P =$
   Area of slab, $A_1 =$
   Projected tank area, $A_2 = D \times L_V$
   Void in overburden, $V_n =$
   $h = .5\,D + d =$

2. Calculate volume of fill, V_f

$$V_f = \left[(0.33\ h)\left(A_1 + A_2 + (A_1A_2)^{1/2}\right)\right] - \left[0.5V_V + V_n\right]$$

3. Calculate weights of fill, W_f

   $W_f = V_f\ w_f$

   and of concrete slab, if applicable

   $W_{cp} = w_c\ A_1\ t_p$

4. Sum of restraining force, $\sum W$

$$\sum W = W_f + W_V + W_{cp}$$

5. Total buoyant force, F

$$F = 62.4(V_V + V_n)$$

6. Determine if deadman is needed;

   If $\sum W > F$ No deadman is needed

   If $\sum W < F$ A deadman is needed

   Weight of deadman required;
   Note: Deadman should be at least 8" thk (.66Ft)

   $W_d = C(t_p)w_C(L_d)$

7. Total resisting weight

$$\sum W' = \sum W + W_d > F$$

Case 2: No Deadman

Determine the depth of burial of a vessel or tank, from the top of the vessel to finished grade, necessary to prevent the tank from floating when empty. Unanchored tank.

- Assume a factor of safety, FS

Recommend a minimum FS of 1.25
Use: _______

- Find weight of fill required, W_{fr}

$$W_{fr} = FS\left(\sum W - W_V - W_{cp}\right)$$

- Find volume of fill required, V_f

$$V_{fr} = W_{fr}/w_f$$

- Find depth of fill required. Solve for h_r:

$$h_r = \left[V_{fr} + \left(0.5\ V_V + V_n\right)\right] \Big/ \left[0.33\left(A_1 + A_2 + (A_1 A_2)^{1/2}\right)\right]$$

- Minimum acceptable burial depth, d

$$d = h_r - .5\ D - t_p$$

Anchor Strap

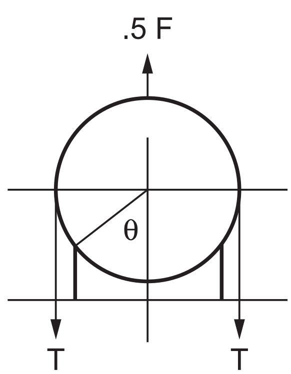


Figure 6-17. Loads on Anchor Strap (Assumes two straps are used.)

- Tension force in anchor strap, T

$$2T = 0.5\ F$$

- The strap size can be determined by either area or stress;

Area; $A_r = T/F_T$
Stress; $f_T = T/A_S$
Use: _______

External Pressure

- External pressure on tank due to fill (overburden), P_X

$$P_X = \left(W_f + W_{cp}\right)/144\ A_2$$

Note: Add P_X to any design external pressure condition and design per ASME Code requirements for the combined condition.

Saddle Design

The saddles must be designed for the operating weight of the vessel plus the weight of fill and concrete pad. Use Zick's analysis to determine the shell stresses.

- Operating load on one saddle, Q

$$Q = .5\left(W_O + W_f + W_{cp}\right)$$

Vessels in Pit

The vessel must be anchored with either anchor bolts in the saddle or anchor straps. Determine the size of anchor straps or bolts necessary to resist flotation.

- Buoyancy force, F

$$F = (V_V + V_n)w_w$$

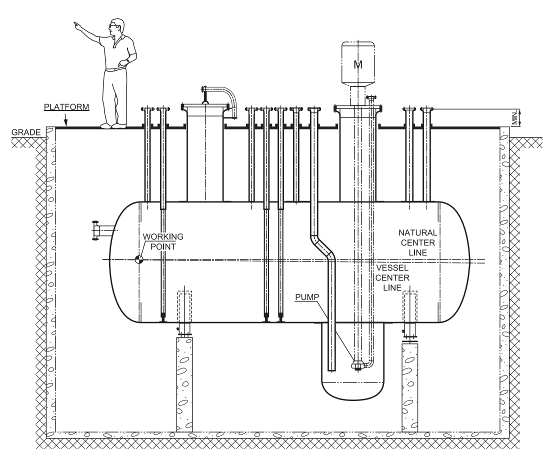


Figure 6-18. Vessel in sump pit

- Net uplift, U

  $U = F\text{-}W_V$

- If $F > U$ then no additional anchorage is required. If $U > F$, then the difference should be added to anchorage required.

Example # 1

Given:

Vessel with 2:1 SE heads, 0.25 in. wall thickness and 24 ft Tan-tan length

$$D = 8 \text{ ft}$$

$$L_V = 28 \text{ ft}$$

$$V_V = 0.262\ D^3 + .25\ \pi\ D^2\ L_{T\text{-}T} = 134 + 1206 = 1340.4\ \text{Ft}^3$$

$$W_V = 9560 \text{ Lbs}$$

$$V_n = (1)30 \text{ in dia} = 12.27 \text{ ft}^3$$

$$(1)12 \text{ in dia} = 1.96 \text{ ft}^3$$

$$(1)6 \text{ in dia} = .98 \text{ ft}^3$$

$$\text{Total} = 15.21 \text{ ft}^3$$

CASE 1: Buried Vessel. Check if a deadman and saddles are required;

$$d = 2.5 \text{ ft}$$

$$h = .5D + d = .5(8) + 2.5 = 6.5 \text{ ft}$$

$$A_1 = (D + 2)(L_V + 4) = 10 \times 32 = 320 \text{ ft}^2$$

$$A_2 = D(L_V) = 8 \times 28 = 224 \text{ ft}^2$$

- Volume of fill, V_f

$$V_f = \left[\left(0.33\ h\right)\left(A_1 + A_2 + (A_1A_2)^{1/2}\right)\right] - \left[0.5V_V + V_n\right]$$

$$V_f = \left[\left(0.33(6.5)\right)(320 + 224 + 267)\right] - \left[0.5(1340) + 1521\right] = 1072 \text{ ft}^3$$

- Weight of fill, W_f

  $W_f = w_f\ V_f = 60 \times 1072 = 64,317$ Lbs

- Weight of concrete pad, W_{cp}

  $W_{cp} = w_c\ A_1\ t_p = 125(320).5 = 20,000$ Lbs

- Total weight, $\sum W$

$$\sum W = W_V + W_f + W_{cp} = 9560 + 64,317 + 20,000 = 93,877 \text{ Lbs}$$

- Buoyancy force, F

$$F = 62.4(V_V + V_n) = 62.4(1340.4 + 15.21) = 84,590 \text{ Lbs}$$

Since the weight of the vessel, fill and concrete pad are heavier than the buoyancy force, no additional restraint is required.

- Factor of Safety, FS

$$FS = \sum W/F = 93,877/84,590 = 1.11$$

If a higher FS is required, than a deadman would have to be added to increase the overall weight of components holding the vessel down.

Example # 2: Vessel in Pit, No Overburden

$$W_V = 7870 \text{ Kg}$$

$$V_V = 21 \text{ m}^3$$

$$w_w = 1001 \text{ Kg/m}^3$$

$$F = V_V\ w_w = 21(1001) = 21,032 \text{ Kg}$$

$$U = F - W_V = 21,032 - 7870 = 13,162 \text{ Kg}$$

Add value of "U" to uplift calculation for design of straps and/or anchor bolts.

Notes

1. Saddle supports are required if the backfill is compacted by tamping, rolling or surface vibration. If saddles are used then a concrete anchorage pad will be required for the saddle to rest on.
2. A saddle is not required if compaction is accomplished by saturation and internal vibration.
3. A full length concrete pad may be used as support for tanks and vessels without saddles or anchorage. In this case the tank should never be placed directly on the concrete pad. At least 6 in and preferably

12 in of clean sand, pea gravel or crushed stone should be placed before the installation of the tank.
4. Underground storage tanks should be retested with air pressure before being covered.
5. Do not use chock blocks under tank to hold in place. These may interfere with the transfer of the load to the backfill.
6. Tanks should be covered with a minimum of 3 feet of earth.
7. When saddles are used, the vessel and saddle must be designed for the weight of the vessel operating as well as the overburden.
8. Nozzles create a void in the overburden and must be accounted for in calculations.
9. Underground storage tanks for storage of flammable products shall comply with the requirements of the following as appropriate;
   a. UL58 (ANSI B137-1971) for steel tanks
   b. NFPA 31 (ANSI Z95.1 – 1969)
   c. NFPA 30
   d. OSHA, Part 1910, Section 106(b)(3)
   e. ASTM D4021-86 for FRP tanks
   f. API RP 1615
10. A tank/vessel should not be anchored unless saddles are used.
11. If the entire area above tank is paved, assume the area as exerting force on the tank as $L_V + 4$ ft $\times$ D + 2 ft
12. Include any installed equipment in the restraining force calculation, for example Submersible pump, piping weights, etc
13. Anchor straps should be made of flat bar in the area with contact with the shell. However the anchor strap must be isolated from contact with the tank shell by use of a gasket.

Procedure 6-7: Local Thin Area (LTA) [2]

Occasionally a vessel shell will sustain damage or be overground in a local area such that the thickness in the damaged area is below the minimum wall thickness. This is known as a “Local Thin Area” or “LTA”. ASME VIII-1 has allowance for such a case as long as certain proportions and guidelines are followed. These guidelines are taken from ASME Section VIII, Division 1, Mandatory Appendix 32 and are presented here.

Nomenclature

C = Projected circumferential length of LTA, in
d = Inside diameter of nozzle, in
L, L_1, L_2 = Projected axial length of LTA, in
R = Inside radius of vessel, in
S = ASME Code allowable stress, PSI
t = Full metal thickness, in
t_L = Minimum thickness of LTA, in
θ = Per Figure 6-19

Case 1: Single LTA

A single LTA is acceptable providing all of the following conditions are met;

1. $t_L / t \geq 0.9$
2. $L \leq \sqrt{Rt}$
3. $C \leq 2\sqrt{Rt}$
4. $t - t_L \leq 3/16$ in.
5. Axial distance at edge of LTA is at least;
   a. $2.5(R\ t)^{1/2}$ from gross structural discontinuity
   b. $d + (R\ t)^{1/2}$ from edge of unreinforced nozzle

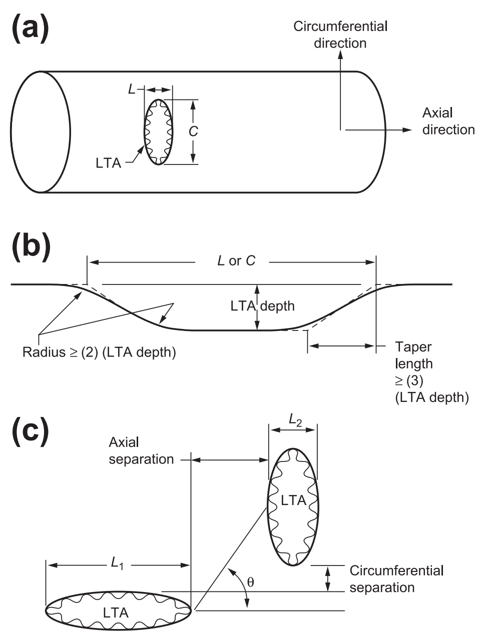


Figure 6-19. Nomenclature.

Table 6-16
Maximum metal temperature

TABLE	TEMPERATURE, ^{o}F
UCS-23	700
UNF-23.1	300
UNF-23.2	150
UNF-23.3	900
UNF-23.4	600
UNF-23.5	600
UHA-23	800
UHT-23	700

6. Taper length ≥ 3
7. Longitudinal stress on LTA from mechanical loads is $\leq 0.3S$
8. Maximum design temperature is less than shown in Table 6-16 for a given material.
9. These rules do not apply to corrosion resistant lining or weld overlay.

Case 2: Multiple LTA

A pair of LTA with finished axial length, L_1 and L_2, providing all of the conditions for a single LTA are met and all of the following:

1. When $\theta \leq 45^o$, the minimum axial separation shall be the greater of the following:
   a. 2 t
   b. $[(1 + 1.5 \text{ Cos } \theta)\ (L_1 + L_2)]\ /\ 2$
2. When $\theta > 45^o$, the minimum axial separation shall be;

   $[(2.91 \text{ Cos } \theta)\ (L_1 + L_2)]/2$

   And the minimum circumferential separation shall be ≤ 2 t
3. Multiple pairs are acceptable providing all pairs meet the conditions above.
4. Multiple LTA's may be combined as a single LTA.

Notes

1. The condition shall be noted on the Manufacturer's Data Report if the LTA occurs prior to ASME Code stamping.

References

[1] McBride WL, Jacobs WS. Design of Radial Nozzles in Cylindrical Shells for Internal Pressure. vol. 2. Journal of Pressure Vessel Technology February 1980.

[2] ASME Boiler and Pressure Vessel Code, Section VIII, Division 1. American Society of Mechanical Engineers; 2010.

[3] Youness A. New Approach to Tower Deflection. Hydrocarbon Processing June 1970.

[4] Lambert FW. The theory and practical design of bunkers. London: British Constructional Steelwork Assoc, Ltd., p. 32–3.

[5] Ketchum MS. Walls, bins, and grain elevators. 3rd ed. New York: McGraw-Hill Book Co.; 1929. p. 206–211.

[6] Blake, Alexander. "Rings and Arcuate Beams," *Product Engineering*, January 7, 1963.

[7] Pirok JN, Wozniak RS. Structural Engineering Handbook. McGraw-Hill Book Co.; 1968. Section 23.

[8] Blodgett O. *Design of Welded Structures*, The James F. Lincoln Arc Welding Foundation, 1975, Section 6.4.

[9] Wygant JF, Crowley MS. Designing Monolithic Refractory Vessel Linings. vol. 43(No. 3). Ceramic Bulletin 1964.

[10] British Ceramic Research Assoc. Thermomechanical Behavior of Refractory Castable Linings. Technical Note No. 320; March 1981.

[11] Buyukozturk O. Thermomechanical Behavior of Refractory Concrete Linings. vol. 65(No. 6). Journal of the American Ceramic Society June 1982.

[12] Truong KT. Improve a FCCU Refractory Lined Piping Design. Hydrocarbon Processing July 1998.

[13] Karamcharandi KC, Gupta NK, Pattabiraman J. Evaluation of Percent Critical Damping of Process Towers. Hydrocarbon Processing May 1982.

[14] Freese CE. Vibration of Vertical Pressure Vessels. Journal of Engineering for Industry February 1959.

[15] Zorilla EP. Determination of Aerodynamic Behavior of Cantilevered Stacks and Towers of Circular Cross Sections. ASME technical paper #71-PET-36; 1971.
[16] DeGhetto K, Long W. Check Stacks for Dynamic Stability. Hydrocarbon Processing February 1966.
[17] Staley CM, Graven GG. The Static and Dynamic Wind Design of Steel Stacks. ASME technical paper #72-PET-30; 1972.
[18] Mahajan KK. Analyze Tower Vibration Quicker. Hydrocarbon Processing May 1979.
[19] Smith JO, McCarthy JN. Dynamic Response of Tall Stacks to Wind Excitation. ASME Technical paper, 63-WA-248; 1964.
[20] Bertolini, I. S., "Wind Action on Towers and Stacks for Petrochemical Plants, " published in *Quaderni Pignone* 17.
[21] Bednar HH. Pressure Vessel Design Handbook. Van Nostrand Reinhold Co; 1981.
[22] Steel Stacks. ASME STS-1–2000. American Society of Mechanical Engineers; 2001.
[23] *National Building Code*, Canada.

7
Local Loads

Stresses caused by external local loads are a major concern to designers of pressure vessels. The techniques for analyzing local stresses and the methods of handling these loadings to keep these stresses within prescribed limits has been the focus of much research. Various theories and techniques have been proposed and investigated by experimental testing to verify the accuracy of the solutions.

Clearly the most significant findings and solutions are those developed by Professor P. P. Bijlaard of Cornell University in the 1950s. These investigations were sponsored by the Pressure Vessel Research Committee of the Welding Research Council. His findings have formed the basis of Welding Research Council Bulletin #107, an internationally accepted method for analyzing stresses due to local loads in cylindrical and spherical shells. The "Bijlaard Curves," illustrated in several sections of this chapter, provide a convenient and accurate method of analysis.

Other methods are also available for analyzing stresses due to local loads, and several have been included herein. It should be noted that the methods utilized in WRC Bulletin #107 have not been included here in their entirety. The technique has been simplified for ease of application. For more rigorous applications, the reader is referred to this excellent source.

Since this book applies to thin-walled vessels only, the detail included in WRC Bulletin #107 is not warranted. No distinction has been made between the inside and outside surfaces of the vessel at local attachments. For vessels in the thick-wall category, these criteria would be inadequate.

Other methods that are used for analyzing local loads are as follows. The designer should be familiar with these methods and when they should be applied.

1. Roark Technical Note #806.
2. Ring analysis as outlined in Procedure 7-1.
3. Beam on elastic foundation methods where the elastic foundation is the vessel shell.
4. Bijlaard analysis as outlined in Procedures 7-4 and 7-5.
5. WRC Bulletin #107.
6. Finite element analysis.

These methods provide results with a varying degree of accuracy. Obviously some are considered "ball park" techniques while others are extremely accurate. The use of one method over another will be determined by how critical the loading is and how critical the vessel is. Obviously it would be uneconomical and impractical to apply finite element analysis on platform support clips. It would, however, be considered prudent to do so on the vessel lug supports of a high-pressure reactor. Finite element analysis is beyond the scope of this book.

Another basis for determining what method to use depends on whether the local load is "isolated" from other local loads and what "fix" will be applied for overstressed conditions. For many loadings in one plane the ring-type analysis has certain advantages. This technique takes into account the additive overlapping effects of each load on the other. It also has the ability to superimpose different types of loading on the same ring section. It also provides an ideal solution for design of a circumferential ring stiffener to take these loads.

If reinforcing pads are used to beef up the shell locally, then the Bijlaard and WRC #107 techniques provide ideal solutions. These methods do not take into account closely spaced loads and their influence on one another. It assumes the local loading is isolated. This technique also provides a fast and accurate method of distinguishing between membrane and bending stresses for combining with other principal stresses.

For local loads where a partial ring stiffener is to be used to reduce local stresses, the beam on elastic foundation method provides an ideal method for sizing the partial rings or stiffener plates. The stresses in the shell must then be analyzed by another local load procedure. Shell stresses can be checked by the beam-on-elastic-foundation method for continuous radial loads about the entire circumference of a vessel shell or ring.

Procedure 7-3 has been included as a technique for converting various shapes of attachments to those which can more readily be utilized in these design procedures. Both the shape of an attachment and whether it is of solid or hollow cross section will have a distinct effect on the distribution of stresses, location of maximum stresses, and stress concentrations.

There are various methods for reducing stresses at local loadings. As shown in the foregoing paragraphs, these will have some bearing on how the loads are analyzed or how stiffening rings or reinforcing plates are sized. The following methods apply to reducing shell stresses locally.

1. Increase the size of the attachment.
2. Increase the number of attachments.
3. Change the shape of the attachment to further distribute stresses.
4. Add reinforcing pads. Reinforcing pads should not be thinner than 0.75 times nor thicker than 1.5 times the

thickness of the shell to which they are attached. They should not exceed 1.5 times the length of the attachment and should be continuously welded. Shell stresses must be investigated at the edge of the attachment to the pad as well as at the edge of the pad.
5. Increase shell thickness locally or as an entire shell course.
6. Add partial ring stiffeners.
7. Add full ring stiffeners.

The local stresses as outlined herein do not apply to local stresses due to any condition of internal restraint such as thermal or discontinuity stresses. Local stresses as defined by this section are due to external mechanical loads. The mechanical loading may be the external loads caused by the thermal growth of the attached piping, but this is not a thermal stress! For an outline of external local loads, see "Categories of Loadings" in Chapter 1.

Procedure 7-1: Stresses in Circular Rings [1–6]

Notation

R_m = mean radius of shell, in.
R_1 = distance to centroid of ring-shell, in.
M = internal moment in shell, in.-lb
M_c = external circumferential moment, in.-lb
M_h = external longitudinal moment (at clip or attachment only), in.-lb
M_L = general longitudinal moment on vessel, in.-lb
F_T = tangential load, lb
F_1,F_2 = loads on attachment, lb
f_a,f_b = equivalent radial load on 1-in. length of shell, lb
f_1 = resultant radial load, lb
P_r = radial load, lb
P = internal pressure, psi
P_e = external pressure, psi
T = internal tension/compression force, lb
K_m,K_T,K_r = internal moment coefficients
C_m,C_T,C_r = internal tension/compression coefficients
$S_{1\text{-}8}$ = shell stresses, psi
Z = section modulus, in.^{3}
t = shell thickness, in.
σ_x = longitudinal stress, psi
σ_ϕ = circumferential stress, psi
e = length of shell which acts with attachment, in.
θ = angular distance between loads or from point of consideration, degrees
W = total weight of vessel above plane under consideration, lb
A = ASME external pressure factor
A_s = metal cross-sectional area of shell, in.^{2}
A_r = cross-sectional area of ring, in.^{2}
B = allowable longitudinal compression stress, psi
E = joint efficiency
E_1 = modulus of elasticity, psi
p = allowable circumferential buckling stress, lb/in.
I = moment of inertia, in.^{4}
S = code allowable stress, tension, psi

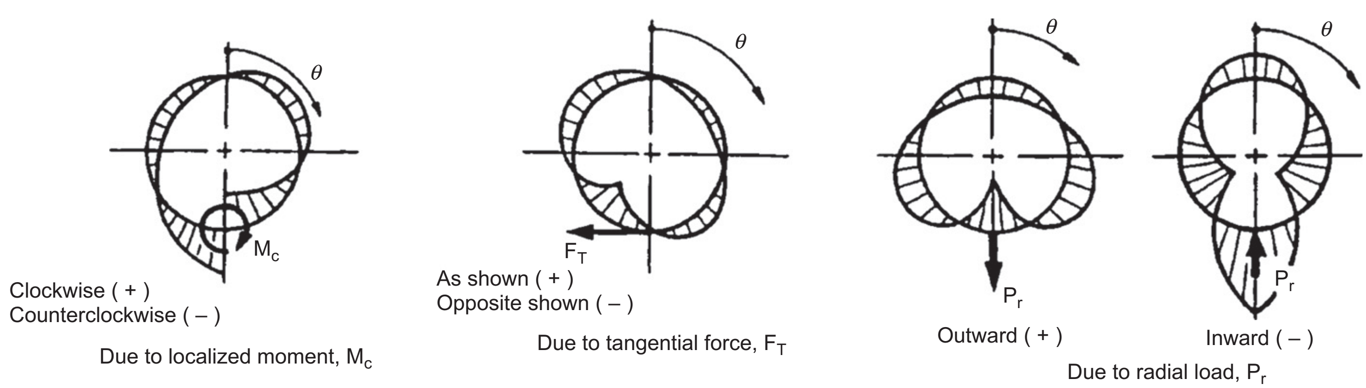
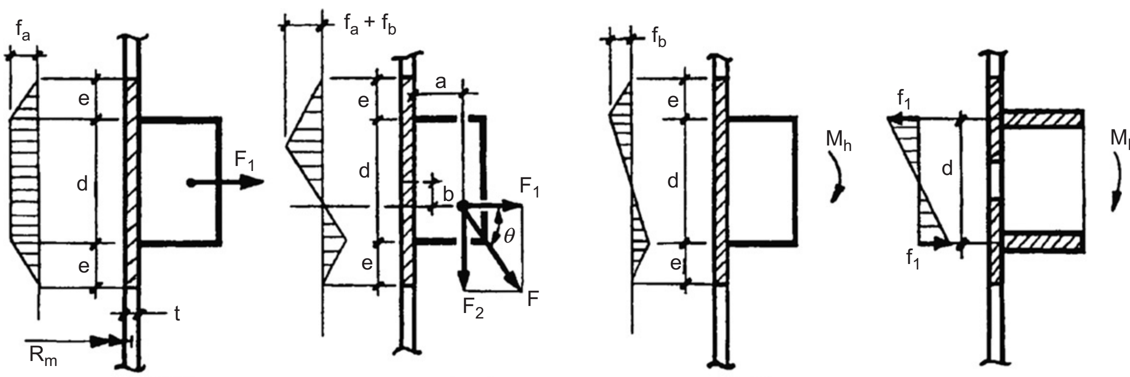


Figure 7-1. Moment diagrams for various ring loadings.

Table 7-1
Moments and forces in shell, M or T

Due to	Internal Moment, M	Tension/Compression Force, T
Circumferential moment, M_c	$M = \sum(K_m M_c)$	$T = \dfrac{\sum(C_m M_c)}{R_m}$
Tangential force, F_T	$M = \sum(K_T F_T)R_m$	$T = \sum(C_T F_T)$
Radial load, P_r	$M = \sum(K_r F_r)R_m$	$T = \sum(C_r F_r)$

Substitute R_1 for R_m if a ring is used. Values of K_m, K_T, K_r, C_m, C_T, and C_r are from Tables 7-4, 7-5, and 7-6.



Case 1:
$e = 0.78\sqrt{R_m t}$
$f_a = \dfrac{F_1}{d + e}$
$f_1 = f_a$

Case 2:
$e = 0.78\sqrt{R_m t}$
$F_1 = F\cos\theta$
$F_2 = F\sin\theta$
$M_h = aF_2 + bF_1$
$f_a = \dfrac{F_1}{d + e}$
$f_b = \dfrac{6M_h}{(d + e)(d + 2e)}$
$f_1 = f_a + f_b$

Case 3:
$e = 0.78\sqrt{R_m t}$
$f_b = \dfrac{6M_h}{(d + e)(d + 2e)}$
$f_1 = f_b$

Case 4:
Continuous rings
$f_1 = \dfrac{M_h}{d}$

Case 1 Case 2 Case 3 Case 4

Figure 7-2. Determination of radial load, f_1, for various shell loadings.

Allowable Stresses

Longitudinal tension: $<1.5SE =$
Longitudinal compression: Factor "B" $=$
Circumferential compression: $<0.5F_y =$
Circumferential buckling: p – lb/in.

$$p = \frac{3E_1 I}{4R^3}$$

(Assumes 4:1 safety factor)
Circumferential tension: $<1.5SE =$
Factor "B"

$$\frac{D_o}{t} = \quad = 0.05 \text{ min}$$

$$\frac{L}{D_o} = \quad = 50 \text{ max}$$

Enter Section II, Part D, Subpart 3, Fig. G, ASME Code

$$A = \quad = 0.1 \text{ max}$$

Enter applicable material chart in ASME Code, Section II:

$$B = \quad \text{psi}$$

For values of A falling to left of material line:

$$B = \frac{AE_1}{2}$$

Table 7-2
Shell stresses due to various loadings

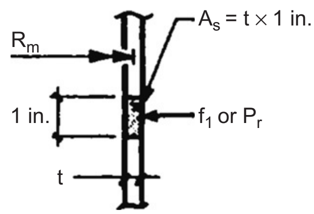
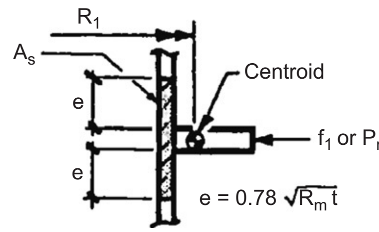
Stress Due To	Stress Direction	Without Stiffener	With Stiffener
			
Internal pressure, P	σ_x	$S_1 = \frac{PR_m}{2t}$	$S_1 = \frac{PR_m}{2t}$
	σ_ϕ	$S_2 = \frac{PR_m}{t}$	$S_2 = \frac{PR_m}{t}\left(\frac{A_s}{A_s + A_r}\right)$
Tension/compression force, T	σ_ϕ	$S_3 = \frac{T}{A_s}$	$S_3 = \frac{T}{A_s + A_r}$
		(+)tension (−)compression	(+)tension (−)compression
Local bending moment, M	σ_ϕ	$S_4 = \frac{6M}{t^2}$	$S_4 = \frac{M}{Z}$
		M can be (+) or (−)	M can be (+) or (−)
External pressure, P_e	σ_x	$S_5 = (-)\frac{P_e R_m}{2t}$	$S_5 = (-)\frac{P_e R_m}{2t}$
	σ_ϕ	$S_6 = (-)\frac{P_e R_m}{t}$	$S_6 = (-)\frac{2P_e R_m e}{A_s + A_r}$
Longitudinal moment, M_L	σ_x	$S_7 = \pm\frac{M_L}{\pi R_m^2 t}$	$S_7 = \pm\frac{M_L}{\pi R_m^2 t}$
Dead load, W	σ_x	$S_8 = (-)\frac{W}{2\pi R_m t}$	$S_8 = (-)\frac{W}{2\pi R_m t}$

Table 7-3
Combined stresses

Type	Tension	Compression
Longitudinal, σ_x	$\sigma_x = S_1 + S_7 - S_8$	$\sigma_x = (-)S_5 - S_7 - S_8$
Circumferential, σ_ϕ	$\sigma_\phi = S_2 + S_3 + S_4$	$\sigma_\phi = (-)S_3 - S_6 - S_4$

Procedure

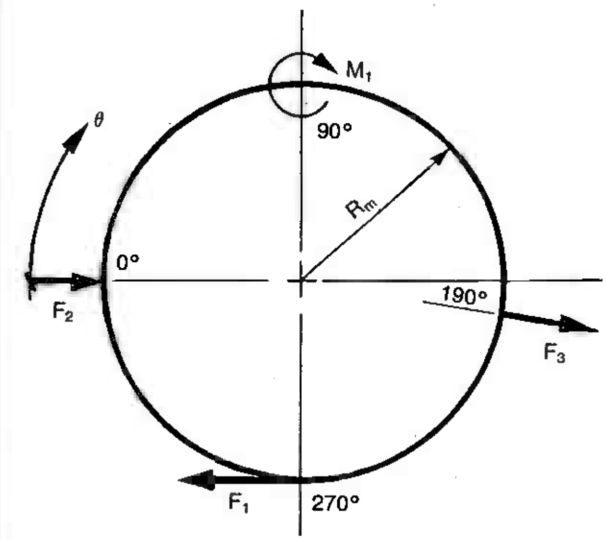
External localized loads (radial, moment, or tangential) produce internal bending moments, tension, and compression in ring sections. The magnitude of these moments and forces can be determined by this procedure, which consists essentially of the following steps:

1. Find moment or tension coefficients based on angular distances between applied loads, at each load from Tables 7–4, 7-5, and 7-6.
2. Superimpose the effects of various loadings by adding the product of coefficients times loads about any given point.

Notes

1. *Sign convention*: It is mandatory that sign convention be strictly followed to determine both the magnitude of the internal forces and tension or compression at any point.
   a. Coefficients in Tables 7-4, 7-5, and 7-6 are for angular distance θ measured between the point

EXAMPLE



GIVEN
$F_1 = (+)$ 1000 lb (would produce clockwise moment)
$F_2 = (-)$ 1500 lb (inward radial load)
$F_3 = (+)$ 500 lb (outward radial load)
$M_1 = (+)$ 800 in.-lb (would produce clockwise moment)
$R_m = 30$ in.
$t = 1$ in.
$A_s = 1 \text{ in.} \times t = 1 \times 1 = 1 \text{ in.}^2$
Since F_2 is the largest load assume the maximum moment will occur there.

Load	θ	Coefficient × Load (÷ Radius)	T
F_2	0°	$C_rF_2 = +0.2387\ (-1{,}500) =$	− 358.1
M_1	90°	$C_mM_1/R_m = -0.3183\ (+800)/30 =$	− 8.5
F_3	190°	$C_rF_3 = -0.2303\ (+500) =$	− 115.2
F_1	270°	$C_TF_1 = 0.0796\ (-1{,}000) =$	− 79.6
		T =	− 561.4 lb

$$\sigma_\phi = \frac{T}{A_s} = \frac{-561.4}{1} = -561 \text{ psi (compression)}$$

Load	θ	Coefficient × Load (× Radius)	M
F_2	0°	$K_rF_2R_m = -0.2387\ (-1500)\ 30 =$	+ 10,742
M_1	90°	$K_mM_1 = -0.0683\ (+800) =$	− 55
F_3	190°	$K_rF_3R_m = -0.0760\ (+500)\ 30 =$	− 1140
F_1	270°	$K_TF_1R_m = -0.0113\ (+1000)\ 30 =$	− 339
		M =	+ 9208 in.-lb

$$\sigma_\phi = \frac{6M}{t^2} = \frac{6(+9208)}{1^2} = +55{,}248 \text{ psi}$$

No good!

$$Z = \frac{M}{S} = \frac{9208}{15000} = 0.614 \text{ in.}^3$$

Use this ring.

Figure 7-3. Sample ring section with various loadings.

Notes *(Cont)*

on the ring under consideration and loads. Signs shown are for θ measured in the clockwise direction only.

b. Signs of coefficients in Tables 7-4, 7-5, and 7-6 are for outward radial loads and clockwise tangential forces and moments. For loads and moments in the opposite direction either the sign of the load or the sign of the coefficient must be reversed.

2. In Figure 7–4 the coefficients have already been combined for the loadings shown. The loads must be of equal magnitude and equally spaced. Signs of coefficients K_r and C_r are given for loads in the direction shown. Either the sign of the load or the sign of the coefficient may be reversed for loads in the opposite direction.
3. The maximum moment normally occurs at the point of the largest load; however, for unevenly spaced or mixed loadings, moments or tension should be investigated at each load, i.e., five loads require five analyses.

(Continued)

Table 7-4
Values of coefficients

	Localized Moment, M_c		Tangential Force, F_T			Localized Moment, M_c		Tangential Force, F_T	
θ	K_m	C_m	K_T	C_T	θ	K_m	C_m	K_T	C_T
0	+0.5	0	0	−0.5	180°	0	0	0	0
5°	+0.4584	−0.0277	−0.0190	−0.4773	185°	+0.0139	+0.0277	−0.0069	−0.0208
10°	+0.4169	−0.0533	−0.0343	−0.4512	190°	+0.0275	+0.0553	−0.0137	−0.0442
15°	+0.3759	−0.0829	−0.0462	−0.4221	195°	+0.0407	+0.0824	−0.0201	−0.0608
20°	+0.3356	−0.1089	−0.0549	−0.3904	200°	+0.0533	+0.1089	−0.0261	−0.0794
25°	+0.2960	−0.1345	−0.0606	−0.3566	205°	+0.0651	+0.1345	−0.0345	−0.0966
30°	+0.2575	−0.1592	−0.0636	−0.3210	210°	+0.0758	+0.1592	−0.0361	−0.1120
35°	+0.2202	−0.1826	−0.0641	−0.2843	215°	+0.0854	+0.1826	−0.0399	−0.1253
40°	+0.1843	−0.2046	−0.0625	−0.2468	220°	+0.0935	+0.2046	−0.0428	−0.1363
45°	+0.1499	−0.2251	−0.0590	−0.2089	225°	+0.1001	+0.2251	−0.0446	−0.1447
50°	+0.1173	−0.2438	−0.0539	−0.1712	230°	+0.1050	+0.2438	−0.0453	−0.1502
55°	+0.0865	−0.2607	−0.0475	−0.1340	235°	+0.1080	+0.2607	−0.0449	−0.1528
60°	+0.0577	−0.2757	−0.0401	−0.0978	240°	+0.1090	+0.2757	−0.0433	−0.1522
65°	+0.0310	−0.2885	−0.0319	−0.0629	245°	+0.1080	+0.2885	−0.0405	−0.1484
70°	+0.0064	−0.2991	−0.0233	−0.0297	250°	+0.1047	+0.2991	−0.0366	−0.1413
75°	−0.0158	−0.3075	−0.0144	+0.0014	255°	+0.0991	+0.3075	−0.0347	−0.1308
80°	−0.0357	−0.3135	−0.0056	+0.0301	260°	+0.0913	+0.3135	−0.0257	−0.1170
85°	−0.0532	−0.3171	+0.0031	+0.0563	265°	+0.0810	+0.3171	−0.0189	−0.0999
90°	−0.0683	−0.3183	+0.0113	+0.0796	270°	+0.0683	+0.3183	−0.0113	−0.0796
95°	−0.0810	−0.3171	+0.0189	+0.0999	275°	+0.0532	+0.3171	−0.0031	−0.0563
100°	−0.0913	−0.3135	+0.0257	+0.1170	280°	+0.0357	+0.3135	+0.0056	−0.0301
105°	−0.0991	−0.3075	+0.0347	+0.1308	285°	+0.0158	+0.3075	+0.0144	−0.0014
110°	−0.1047	−0.2991	+0.0366	+0.1413	290°	−0.0064	+0.2991	+0.0233	+0.0297
115°	−0.1079	−0.2885	+0.0405	+0.1484	295°	−0.0310	+0.2885	+0.0319	+0.0629
120°	−0.1090	−0.2757	+0.0433	+0.1522	300°	−0.0577	+0.2757	+0.0401	+0.0978
125°	−0.1080	−0.2607	+0.0449	+0.1528	305°	−0.0865	+0.2607	+0.0475	+0.1340
130°	−0.1050	−0.2438	+0.0453	+0.1502	310°	−0.1173	+0.2438	+0.0539	+0.1712
135°	−0.1001	−0.2251	+0.0446	+0.1447	315°	−0.1499	+0.2251	+0.0590	+0.2089
140°	−0.0935	−0.2046	+0.0428	+0.1363	320°	−0.1843	+0.2046	+0.0625	+0.2468
145°	−0.0854	−0.1826	+0.0399	+0.1253	325°	−0.2202	+0.1826	+0.0641	+0.2843
150°	−0.0758	−0.1592	+0.0361	+0.1120	330°	−0.2575	+0.1592	+0.0636	+0.3210
155°	−0.0651	−0.1345	+0.0345	+0.0966	335°	−0.2960	+0.1345	+0.0606	+0.3566
160°	−0.0533	−0.1089	+0.0261	+0.0794	340°	−0.3356	+0.1089	+0.0549	+0.3904
165°	−0.0407	−0.0824	+0.0201	+0.0608	345°	−0.3759	+0.0829	+0.0462	+0.4221
170°	−0.0275	−0.0553	+0.0137	+0.0442	350°	−0.4169	+0.0533	+0.0343	+0.4512
175°	−0.0139	−0.0277	+0.0069	+0.0208	355°	−0.4584	+0.0277	+0.0190	+0.4773

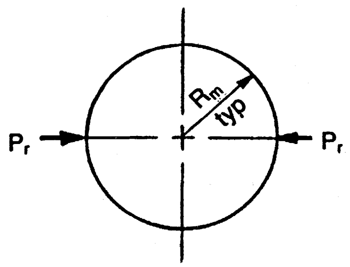
Reprinted by permission R. I. Isakower, Machine Design*, Mar. 4, 1965.*

Table 7-5
Values of coefficient K_r due to outward radial load, P_r

θ	K_r	θ	K_r	θ	K_r	θ	K_r
0–360°	−0.2387	46–314	+0.0533	92–268	+0.0883	138–222	−0.0212
1–359	−0.2340	47–313	+0.0567	93–267	+0.0868	139–221	−0.0237
2–358	−0.2217	48–312	+0.0601	94–266	+0.0851	140–220	−0.0268
3–357	−0.2132	49–311	+0.0632	95–265	+0.0830	141–219	−0.0284
4–356	−0.2047	50–310	+0.0663	96–264	+0.0817	142–218	−0.0307
5–355	−0.1961	51–309	+0.0692	97–263	+0.0798	143–217	−0.0330
6–354	−0.1880	52–308	+0.0720	98–262	+0.0780	144–216	−0.0353
7–353	−0.1798	53–307	+0.0747	99–261	+0.0760	145–215	−0.0382
8–352	−0.1717	54–306	+0.0773	100–260	+0.0736	146–214	−0.0396
9–351	−0.1637	55–305	+0.0796	101–259	+0.0719	147–213	−0.0418
10–350	−0.1555	56–304	+0.0819	102–258	+0.0698	148–212	−0.0438
11–349	−0.1480	57–303	+0.0841	103–257	+0.0677	149–211	−0.0459
12–348	−0.1402	58–302	+0.0861	104–256	+0.0655	150–210	−0.0486
13–347	−0.1326	59–301	+0.0880	105–255	+0.0627	151–209	−0.0498
14–346	−0.1251	60–300	+0.0897	106–254	+0.0609	152–208	−0.0517
15–345	−0.1174	61–299	+0.0914	107–253	+0.0586	153–207	−0.0535
16–344	−0.1103	62–298	+0.0940	108–252	+0.0562	154–206	−0.0553
17–343	−0.1031	63–297	+0.0944	109–251	+0.0538	155–205	−0.0577
18–342	−0.0960	64–296	+0.0957	110–250	+0.0508	156–204	−0.0586
19–341	−0.0890	65–295	+0.0967	111–249	+0.0489	157–203	−0.0602
20–340	−0.0819	66–294	+0.0979	112–248	+0.0464	158–202	−0.0617
21–339	−0.0754	67–293	+0.0988	113–247	+0.0439	159–201	−0.0633
22–338	−0.0687	68–292	+0.0997	114–246	+0.0431	160–200	−0.0654
23–337	−0.0622	69–291	+0.1004	115–245	+0.0381	161–199	−0.0660
24–336	−0.0558	70–290	+0.1008	116–244	+0.0361	162–198	−0.0673
25–335	−0.0493	71–289	+0.1014	117–243	+0.0335	163–197	−0.0686
26–334	−0.0433	72–288	+0.1018	118–242	+0.0309	164–196	−0.0697
27–333	−0.0373	73–287	+0.1019	119–241	+0.0283	165–195	−0.0715
28–332	−0.0314	74–286	+0.1020	120–240	+0.0250	166–194	−0.0719
29–331	−0.0256	75–285	+0.1020	121–239	+0.0230	167–193	−0.0728
30–330	−0.0197	76–284	+0.1020	122–238	+0.0203	168–192	−0.0737
31–329	−0.0144	77–283	+0.1019	123–237	+0.0176	169–191	−0.0746
32–328	−0.0089	78–282	+0.1017	124–236	+0.0145	170–190	−0.0760
33–327	−0.0037	79–281	+0.1013	125–235	+0.0116	171–189	−0.0764
34–326	+0.0015	80–280	+0.1006	126–234	+0.0090	172–188	−0.0768
35–325	+0.0067	81–279	+0.1003	127–233	+0.0070	173–187	−0.0772
36–324	+0.0115	82–278	+0.0997	128–232	+0.0044	174–186	−0.0776
37–323	+0.0162	83–277	+0.0989	129–231	+0.0017	175–185	−0.0787
38–322	+0.0209	84–276	+0.0981	130–230	−0.0016	176–184	−0.0789
39–321	+0.0254	85–275	+0.0968	131–229	−0.0035	177–183	−0.0791
40–320	+0.0299	86–274	+0.0961	132–228	−0.0061	178–182	−0.0793
41–319	+0.0340	87–273	+0.0950	133–227	−0.0087	179–181	−0.0795
42–318	+0.0381	88–272	+0.0938	134–226	−0.0113	180	−0.0796
43–317	+0.0421	89–271	+0.0926	135–225	−0.0145		
44–316	+0.0460	90–270	+0.0909	136–224	−0.0163		
45–315	+0.0497	91–269	+0.0898	137–223	−0.0188		

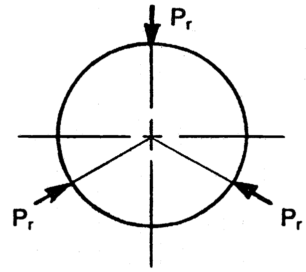
Table 7-6
Values of coefficient C_r due to radial load, P_r

θ	C_r	θ	C_r	θ	C_r	θ	C_r	θ	C_r	θ	C_r
0–360°	+0.2387	31–329	+0.4175	62–298	+0.4010	93–267	+0.2280	124–236	−0.0040	155–205	−0.1870
1–359	+0.2460	32–328	+0.4200	63–297	+0.3975	94–266	+0.2225	125–235	−0.0018	156–204	−0.1915
2–358	+0.2555	33–327	+0.4225	64–296	+0.3945	95–265	+0.2144	126–234	−0.0175	157–203	−0.1945
3–357	+0.2650	34–326	+0.4250	65–295	+0.3904	96–264	+0.2075	127–233	−0.0250	158–202	−0.1985
4–356	+0.2775	35–325	+0.4266	66–294	+0.3875	97–263	+0.2000	128–232	−0.0325	159–201	−0.2025
5–355	+0.2802	36–324	+0.4280	67–293	+0.3830	98–262	+0.1925	129–231	−0.0400	160–200	−0.2053
6–354	+0.2870	37–323	+0.4300	68–292	+0.3790	99–261	+0.1850	130–230	−0.0471	161–199	−0.2075
7–353	+0.2960	38–322	+0.4315	69–291	+0.3740	100–260	+0.1774	131–229	−0.0550	162–198	−0.2110
8–352	+0.3040	39–321	+0.4325	70–290	+0.3688	101–259	+0.1700	132–228	−0.0620	163–197	−0.2140
9–351	+0.3100	40–320	+0.4328	71–289	+0.3625	102–258	+0.1625	133–227	−0.0675	164–196	−0.2170
10–350	+0.3171	41–319	+0.4330	72–288	+0.3600	103–257	+0.1550	134–226	−0.0750	165–195	−0.2198
11–349	+0.3240	42–318	+0.4332	73–287	+0.3540	104–256	+0.1480	135–225	−0.0804	166–194	−0.2220
12–348	+0.3310	43–317	+0.4335	74–286	+0.3490	105–255	+0.1394	136–224	−0.0870	167–193	−0.2240
13–347	+0.3375	44–316	+0.4337	75–285	+0.3435	106–254	+0.1400	137–223	−0.0940	168–192	−0.2260
14–346	+0.3435	45–315	+0.4340	76–284	+0.3381	107–253	+0.1300	138–222	−0.1000	169–191	−0.2280
15–345	+0.3492	46–314	+0.4332	77–283	+0.3325	108–252	+0.1150	139–221	−0.1050	170–190	−0.2303
16–344	+0.3550	47–313	+0.4324	78–282	+0.3270	109–251	+0.1075	140–220	−0.1115	171–189	−0.2315
17–343	+0.3600	48–312	+0.4316	79–281	+0.3200	110–250	+0.1011	141–219	−0.1170	172–188	−0.2325
18–342	+0.3655	49–311	+0.4308	80–280	+0.3150	111–249	+0.0925	142–218	−0.1230	173–187	−0.2345
19–341	+0.3720	50–310	+0.4301	81–279	+0.3090	112–248	+0.0840	143–217	−0.1280	174–186	−0.2351
20–340	+0.3763	51–309	+0.4283	82–278	+0.3025	113–247	+0.0760	144–216	−0.1350	175–185	−0.2366
21–339	+0.3810	52–308	+0.4266	83–277	+0.2960	114–246	+0.0700	145–215	−0.1398	176–184	−0.2370
22–338	+0.3855	53–307	+0.4248	84–276	+0.2900	115–245	+0.0627	146–214	−0.1450	177–183	−0.2375
23–337	+0.3900	54–306	+0.4231	85–275	+0.2837	116–244	+0.0550	147–213	−0.1500	178–182	−0.2380
24–336	+0.3940	55–305	+0.4214	86–274	+0.2775	117–243	+0.0490	148–212	−0.1550	179–181	−0.2384
25–335	+0.3983	56–304	+0.4180	87–273	+0.2710	118–242	+0.0400	149–211	−0.1605	180	−0.2387
26–334	+0.4025	57–303	+0.4160	88–272	+0.2650	119–241	+0.0335	150–210	−0.1651		
27–333	+0.4060	58–302	+0.4130	89–271	+0.2560	120–240	+0.0250	151–209	−0.1690		
28–332	+0.4100	59–301	+0.4100	90–270	+0.2500	121–239	+0.0175	152–208	−0.1745		
29–331	+0.4125	60–300	+0.4080	91–269	+0.2430	122–238	+0.0105	153–207	−0.1780		
30–330	+0.4151	61–299	+0.4040	92–268	+0.2360	123–237	+0.0025	154–206	−0.1825		



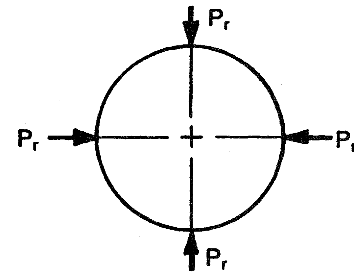
At loads	Between loads
K_r + 0.3183	K_r − 0.1817
C_r + 0	C_r − 0.5

Case 1



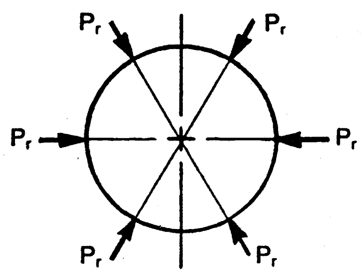
At loads	Between loads
K_r + 0.1888	K_r − 0.1
C_r − 0.2887	C_r − 0.5773

Case 2



At loads	Between loads
K_r + 0.1366	K_r − 0.0705
C_r − 0.5	C_r − 0.707

Case 3



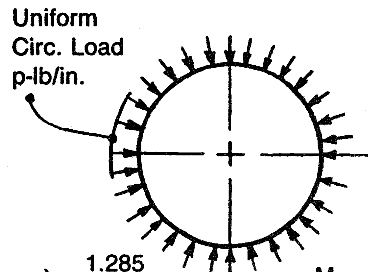
At loads	Between loads
K_r + 0.0889	K_r − 0.045
C_r − 0.866	C_r − 1.0

Case 4

For any number of equally spaced loads
ϕ = ½ angle between loads, radians

- *At loads:* $K_r = 0.5\left[\frac{1}{\phi} - \frac{\cos\phi}{\sin\phi}\right]$
- *Between loads:* $K_r = -0.5\left[\frac{1}{\sin\phi} - \frac{1}{\phi}\right]$
- *Tension force, T:* $T = \frac{P_r}{2}\left[\frac{1}{\sin\phi}\right]$

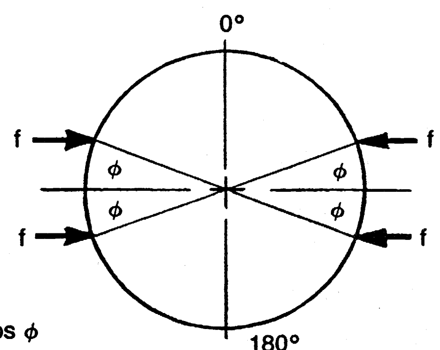
Case 5



$\lambda = \frac{1.285}{\sqrt{R_m t}}$ $M = \frac{p}{4\lambda}$

$\sigma_x = \frac{-3p}{2\lambda t^2}$ $\sigma_\phi = \frac{-pR_m\lambda}{2t}$

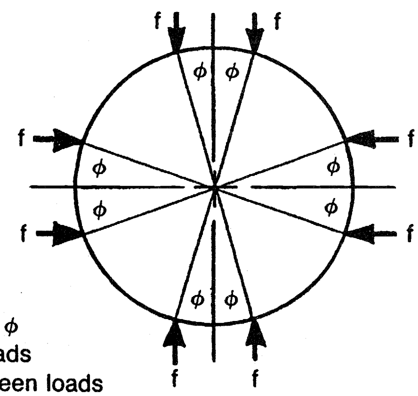
Case 6



$P_r = f\cos\phi$
K_r is at loads
C_r is between loads

ϕ	K_r	C_r	ϕ	K_r	C_r
1°	0.6185	− 1.0	10°	0.4656	− 0.985
2°	0.6011	− 0.999	15°	0.3866	− 0.966
3°	0.5836	− 0.998	20°	0.3152	− 0.940
4°	0.5663	− 0.997	25°	0.2536	− 0.906
5°	0.5498	− 0.996	30°	0.2036	− 0.866
6°	0.5319	− 0.995	35°	0.1668	− 0.819
7°	0.5150	− 0.992	40°	0.1441	− 0.766
8°	0.4980	− 0.990	45°	0.1366	− 0.707
9°	0.4813	− 0.986			

Case 7



$P_r = f\cos\phi$
K_r is at loads
C_r is between loads

ϕ	K_r	C_r	ϕ	K_r	C_r
1°	0.2540	− 1.411	10°	0.1302	− 1.393
2°	0.2375	− 1.410	15°	0.0902	− 1.366
3°	0.2214	− 1.409	20°	0.0688	− 1.329
4°	0.2062	− 1.408	25°	0.0688	− 1.282
5°	0.1918	− 1.407	30°	0.0902	− 1.225
6°	0.1780	− 1.406	35°	0.1324	− 1.158
7°	0.1649	− 1.405	40°	0.1939	− 1.083
8°	0.1525	− 1.404	45°	0.2732	− 1.00
9°	0.1409	− 1.397			

Case 8

Figure 7-4. Values of coefficients K_r and C_r for various loadings.

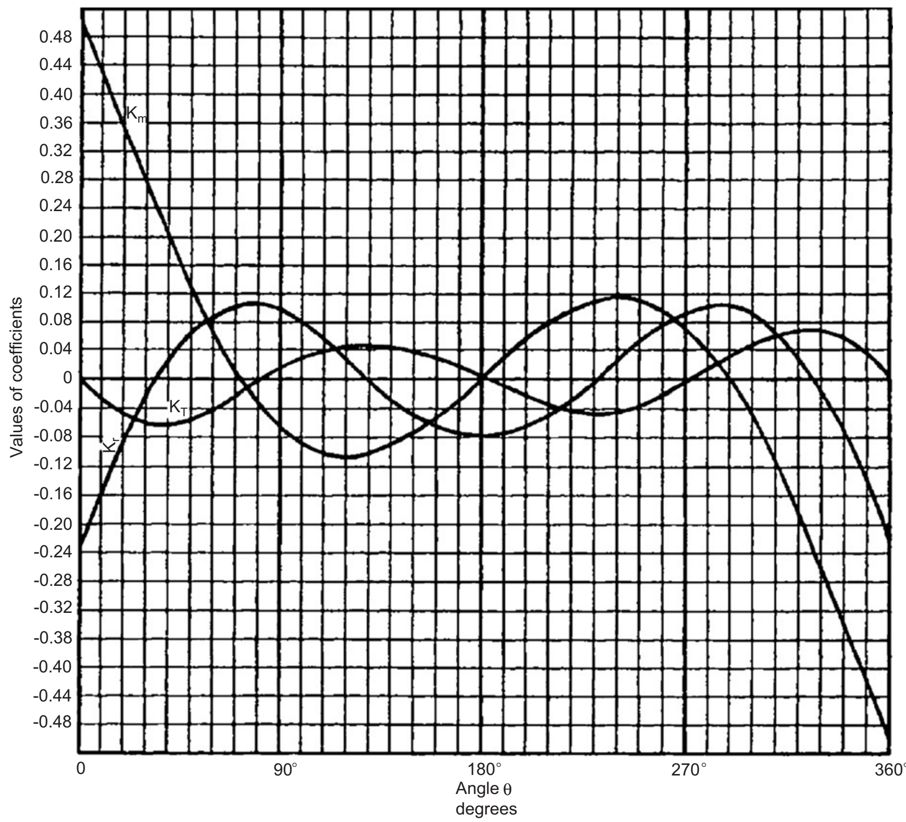


Figure 7-5. Graph of internal moment coefficients K_m, K_r, and K_T.

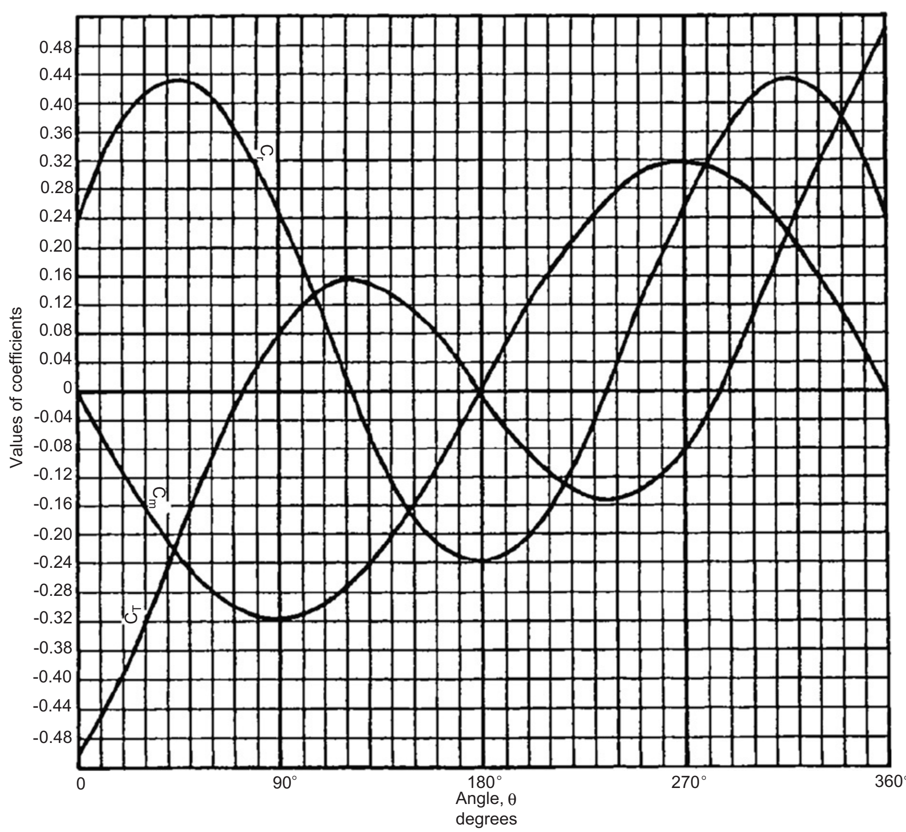


Figure 7-6. Graph of circumferential tension/compression coefficients C_m, C_r, and C_T.

Notes *(Cont)*

4. This procedure uses strain-energy concepts.
5. The following is assumed.
   a. Rings are of uniform cross section.
   b. Material is elastic, but is not stressed beyond elastic limit.
   c. Deformation is caused mainly by bending.
   d. All loads are in the same plane.
   e. The ring is not restrained and is supported along its circumference by a number of equidistant simple supports (therefore conservative for use on cylinders).
   f. The ring is of such large radius in comparison with its radial thickness that the deflection theory for straight beams is applicable.

Procedure 7-2: Design of Partial Ring Stiffeners [7]

Notation

M_L = longitudinal moment, in.-lb
M = internal bending moment, shell, in.-lb
F_b = allowable bending stress, psi
f_b = bending stress, psi
f or f_n = concentrated loads on stiffener due to radial or moment load on clip, lb
F_x = function or moment coefficient (see Table 7-7) = $e^{-\beta x}(\cos \beta x - \sin \beta x)$
E_v = modulus of elasticity of vessel shell at design temperature, psi
E_s = modulus of elasticity of stiffener at design temperature, psi
e = log base 2.71
I = moment of inertia of stiffener, in.^{4}
Z = section modulus of stiffener, in.^{3}
K = "spring constant" or "foundation modulus", lb/in.^{3}
x = distance between loads, in.
β = damping factor, dimensionless
P_r = radial load, lb

Table 7-7
Values Of Function F_x

βx	F_x	βx	F_x
0	1.0	0.55	0.1903
0.05	0.9025	0.6	0.1431
0.1	0.8100	0.65	0.0997
0.15	0.7224	0.7	0.0599
0.2	0.6398	0.75	0.0237
0.25	0.5619	0.8	(−)0.0093
0.3	0.4888	0.85	(−)0.0390
0.35	0.4203	0.9	(−)0.0657
0.4	0.3564	0.95	(−)0.0896
0.45	0.2968	1.0	(−)0.1108
0.5	0.2415		

Formulas

1. *Single load.* Determine concentrated load on each stiffener depending on whether there is a radial load or moment loading, single or double stiffener.

f =

- *Calculate foundation modulus, K.*

$$K = \frac{E_v t}{R^2}$$

- *Assume stiffener size and calculate Z and I.*
  Proposed size: _____

$$I = \frac{bh^3}{12}$$

$$Z = \frac{bh^2}{6}$$

- *Calculate damping factor β based on proposed stiffener size.*

$$\beta = \sqrt[4]{\frac{K}{4E_s I}}$$

- *Calculate internal bending moment in stiffener, M.*

$$M = \frac{f}{4\beta}$$

- *Calculate bending stress, f_b.*

$$f_b = \frac{M}{Z}$$

If bending stress exceeds allowable ($F_b = 0.6F_y$), increase size of stiffener and recalculate I, Z, β, M, and f_b.

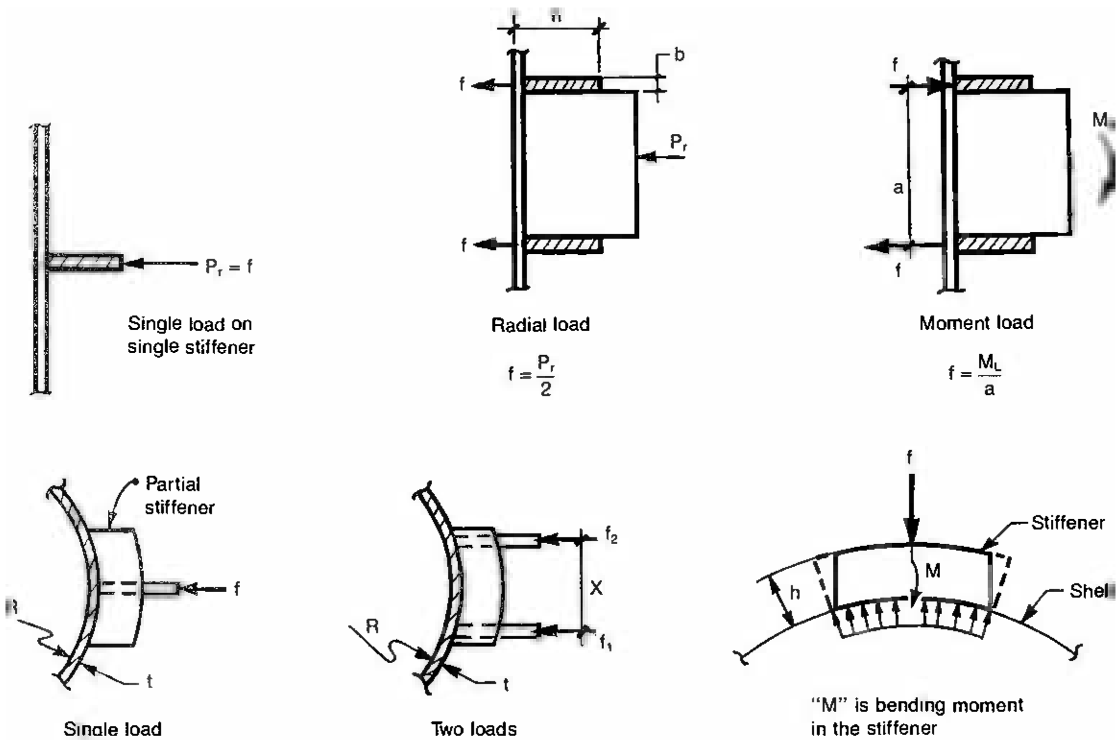


Figure 7-7. Dimensions, forces, and loadings for partial ring stiffeners.

2. *Multiple loads (see Figure 7-8).* Determine concentrated loads on stiffener(s). Loads must be of equal magnitude.

$$f = f_1 = f_2 = \ldots = f_n$$

- *Calculate foundation modulus, K.*

$$K = \frac{E_v t}{R^2}$$

- *Assume a stiffener size and calculate I and Z.*

Proposed size: _____

$$I = \frac{bh^3}{12}$$

$$Z = \frac{bh^2}{6}$$

- *Calculate damping factor β based on proposed stiffener size.*

$$\beta = \sqrt[4]{\frac{K}{4E_s I}}$$

- *Calculate internal bending moment in stiffener.*

Step 1: Determine βx for each load (βx is in radians).

Step 2: Determine F_x for each load from Table 7-7 or calculate as follows:

$$F_x = e^{-\beta x}\left(\cos \beta x - \sin \beta x\right)$$

Step 3: Calculate bending moment, M

$\beta x_0 = 0$	$F_1 = 1$
$\beta x_1 =$ __	$F_2 =$ __
$\beta x_2 =$ __	$F_3 =$ __
$\beta x_n =$ __	$F_n =$ __
	$\sum F_x =$ __

$$M = \frac{f}{4\beta}\left(\sum F_x\right)$$

- *Calculate bending stress, f_b.*

$$f_b = \frac{M}{Z}$$

Notes

1. This procedure is based on the beam-on-elastic-foundation theory. The elastic foundation is the vessel shell and the beam is the partial ring stiffener. The stiffener must be designed to be stiff enough to transmit the load(s) uniformly over its full length.

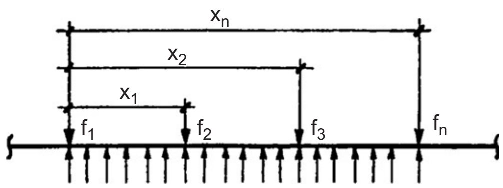


Figure 7-8. Dimensions and loading diagram for beam on elastic foundation analysis.

The flexibility of the vessel shell is taken into account. The length of the vessel must be at least 4.9 $\sqrt{Rt}$ to qualify for the infinitely long beam theory.

2. The case of multiple loads uses the principle of superposition. That is, the effect of each load may be determined independent of the other loads and the total effect may be determined by adding the individual effects.
3. This procedure determines the bending stress in the stiffener only. The stresses in the vessel shell should be checked by an appropriate local load procedure. These local stresses are secondary bending stresses and should be combined with *primary* membrane and bending stresses.

Procedure 7-3: Attachment Parameters

This procedure is for use in converting the area of attachments into shapes that can readily be applied in design procedures. Irregular attachments (not round, square, or rectangular) can be converted into a rectangle which has:

- The same moment of inertia
- The same ratio of length to width of the original attachment

In addition, a rectangular load area may be reduced to an "equivalent" square area.

Bijlaard recommends, for non-rectangular attachments, the loaded rectangle can be assumed to be that which has the same moment of inertia with respect to the moment axis as the plan of the actual attachment. Further, it should be assumed that the dimensions of the rectangle in the longitudinal and circumferential directions have the same ratio as the two dimensions of the attachment in these directions.

Dodge comments on this method in WRC Bulletin 198: "Although the 'equivalent moment of inertia procedure' is simple and direct, it was not derived by any mathematical or logical reasoning which would allow the designer to rationalize the accuracy of the results."

Dodge goes on to recommend an alternative procedure based on the principle of superposition. This method would divide irregular attachments into a composite of one or more rectangular sub-areas.

Neither method is entirely satisfactory and each ignores the effect of local stiffness provided by the attachment's shape. An empirical method should take into consideration the "area of influence" of the attachment which would account for the attenuation length or decay length of the stress in question.

Studies by Roark would indicate short zones of influence in the longitudinal direction (quick decay) and a much broader area of influence in the circumferential direction (slow decay, larger attenuation). This would also seem to account for the attachment and shell acting as a unit, which they of course do.

Since no hard and fast rules have yet been determined, it would seem reasonable to apply the factors as outlined in this procedure for general applications. Very large or critical loads should, however, be examined in depth.

Notes

1. $b = t_c + 2t_w + 2t_s$ where t_w = fillet weld size and t_s = thickness of shell.
2. Clips must be closer than $\sqrt{Rt}$ if running circumferentially or closer than 6 in. if running longitudinally to be considered as a single attachment.

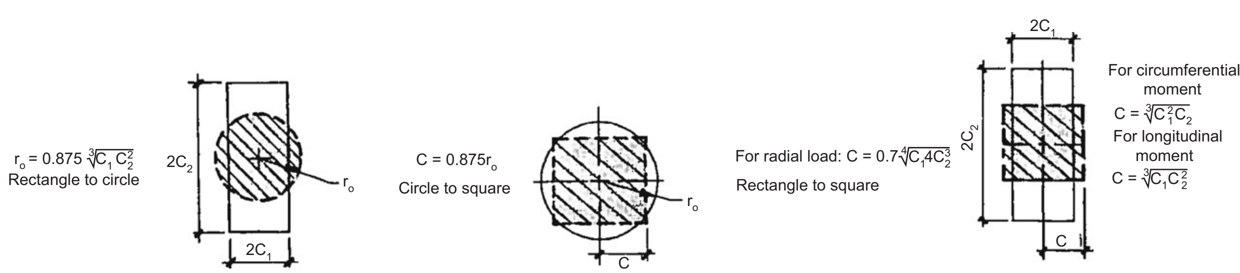
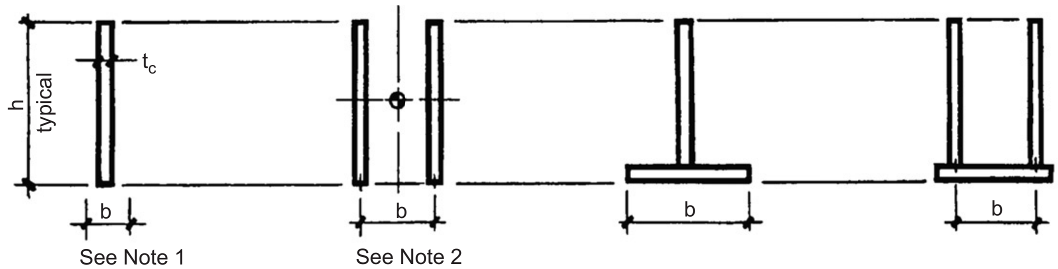
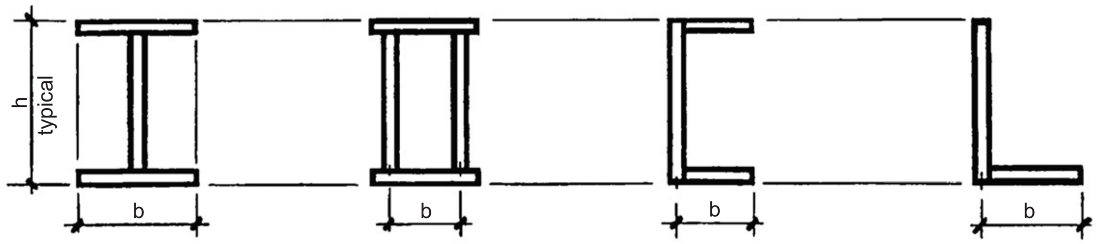


Figure 7-9. Attachment parameters for solid attachments.



C_1	0.5b	0.3b	0.25b	0.3b
C_2	0.4h	0.4h	0.4h	0.4h



C_1	0.4b	0.5b	0.3b	0.2b
C_2	0.5h	0.5h	0.5h	0.4h

Figure 7-10. Attachment parameters for nonsolid attachments.

Procedure 7-4: Stresses in Cylindrical Shells from External Local Loads [7,9,10,11]

Notation

P_r = radial load, lb
P = internal design pressure, psi
M_L = external longitudinal moment, in.-lb
M_c = external circumferential moment, in.-lb
M_T = external torsional moment, in.-lb
M_x = internal circumferential moment, in.-lb/in.
M_ϕ = internal longitudinal moment, in.-lb/in.
V_L = longitudinal shear force, lb
V_c = circumferential shear force, lb
R_m = mean radius of shell, in.
r_o = outside radius of circular attachment, in.
r = corner radius of attachment, in.
K_n,K_b = stress concentration factors

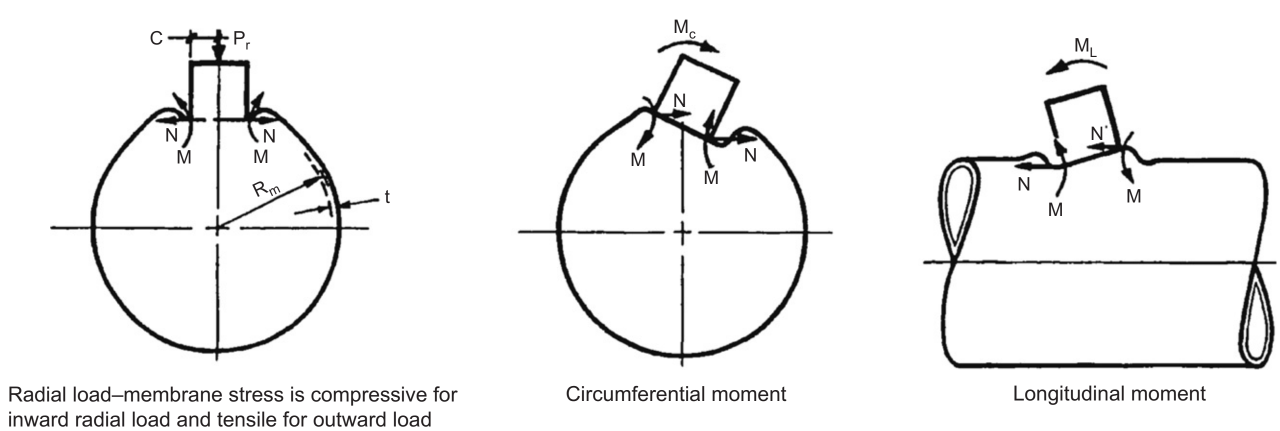


Figure 7-11. Loadings and forces at local attachments in cylindrical shells.

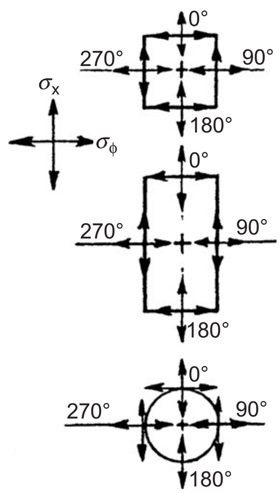


Figure 7-12. Stress indices of local attachments.

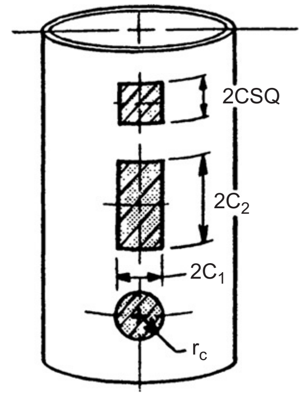
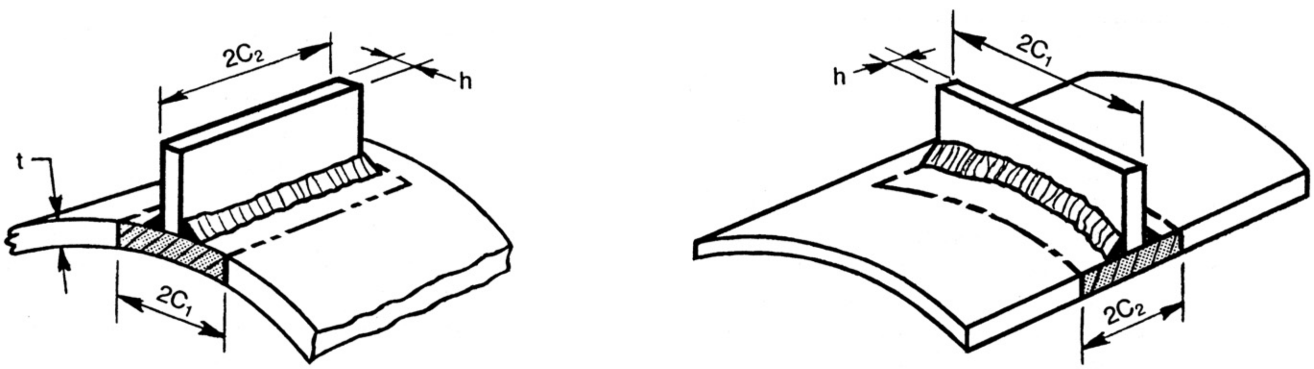


Figure 7-13. Load areas of local attachments. For circular attachments use $C = 0.875r_o$.



$2C_1 = h + 2w + 2t$
w = leg of fillet weld
h = thickness of attachment

$2C_2 = h + 2w + 2t$
Note: Only ratios of C_1/C_2 between 0.25 and 4 may be computed by this procedure.

Figure 7-14. Dimensions for clips and attachments.

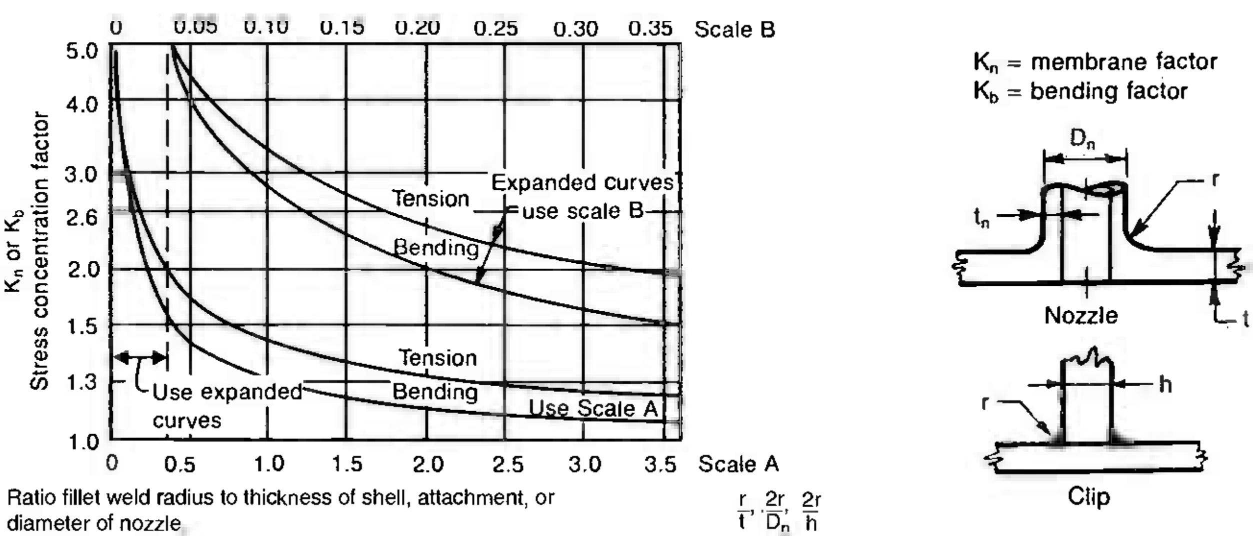


Figure 7-15. Stress concentration factors. (*Reprinted by permission of the Welding Research Council.*)

K_c, K_L, K_1, K_2 = coefficients to determine β for rectangular attachments

N_x = membrane force in shell, longitudinal, lb/in.

N_ϕ = membrane force in shell, circumferential, lb/in.

τ_T = torsional shear stress, psi

τ_s = direct shear stress, psi

σ_x = longitudinal normal stress, psi

σ_ϕ = circumferential normal stress, psi

C = one-half width of square attachment, in.

C_c, C_L = multiplication factors for rectangular attachments

C_1 = one-half circumferential width of a rectangular attachment, in.

C_2 = one-half longitudinal length of a rectangular attachment, in.

h = thickness of attachment, in.

d_n = outside diameter of circular attachment, in.

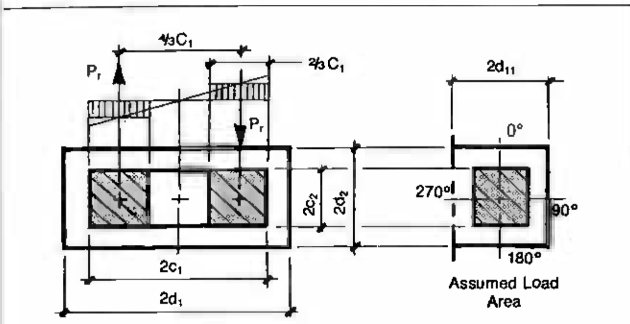
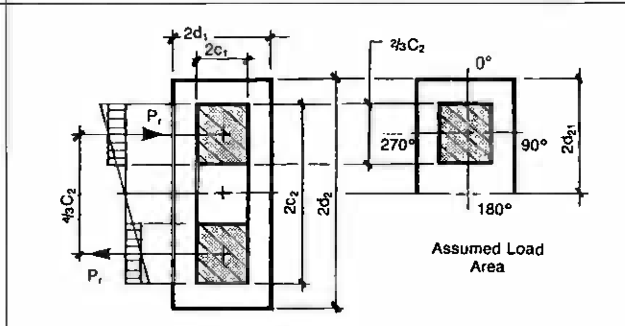
t_e = equivalent thickness of shell and reinforcing, in.

t_p = thickness of reinforcing pad, in.

t = shell thickness, in.

$\gamma, \beta, \beta_1, \beta_2$ = ratios based on vessel and attachment geometry

COMPUTING GEOMETRIC PARAMETERS FOR LOADS ON ATTACHMENTS WITH REINFORCING PADS

CIRCUMFERENTIAL MOMENT				LONGITUDINAL MOMENT			
							
At Edge of Attachment		At Edge of Pad		At Edge of Attachment		At Edge of Pad	
$R_m = \frac{I.D. + t + t_p}{2}$		$R_m = \frac{I.D. + t}{2}$		$R_m = \frac{I.D. + t + t_p}{2}$		$R_m = \frac{I.D. + t}{2}$	
$t_e = \sqrt{t^2 + t_p^2}$		t		$t_e = \sqrt{t^2 + t_p^2}$		t	
$\gamma = \frac{R_m}{t_e}$		$\gamma = \frac{R_m}{t}$		$\gamma = \frac{R_m}{t_e}$		$\gamma = \frac{R_m}{t}$	
$C_1 = \frac{2C_1}{6}$		$C_1 = \frac{2d_{11}}{2}$		$C_1 = \frac{2C_1}{2}$		$C_1 = \frac{2d_1}{2}$	
$C_2 = \frac{2C_2}{2}$		$C_2 = \frac{2d_2}{2}$		$C_2 = \frac{2C_2}{6}$		$C_2 = \frac{2d_{21}}{2}$	
$\beta_1 = \frac{C_1}{R_m}$		$\beta_1 = \frac{C_1}{R_m}$		$\beta_1 = \frac{C_1}{R_m}$		$\beta_1 = \frac{C_1}{R_m}$	
$\beta_2 = \frac{C_2}{R_m}$		$\beta_2 = \frac{C_2}{R_m}$		$\beta_2 = \frac{C_2}{R_m}$		$\beta_2 = \frac{C_2}{R_m}$	
$\frac{\beta_1}{\beta_2}$		$\frac{\beta_1}{\beta_2}$		$\frac{\beta_1}{\beta_2}$		$\frac{\beta_1}{\beta_2}$	
β for N_ϕ		β for N_ϕ		β for N_ϕ		β for N_ϕ	
β for N_x		β for N_x		β for N_x		β for N_x	
β for M_ϕ		β for M_ϕ		β for M_ϕ		β for M_ϕ	
β for M_x		β for M_x		β for M_x		β for M_x	

Geometric Parameters

$$\gamma = \frac{R_m}{t}$$

$$\beta = \frac{C}{R_m}$$

or for circular attachments:

$$\frac{0.875r_o}{R_m}$$

For rectangular attachments:

$$\beta_1 = \frac{C_1}{R_m}$$

$$\beta_2 = \frac{C_2}{R_m}$$

Procedure

To calculate stresses due to radial load P_r, longitudinal moment M_L, and circumferential moment M_c, on a cylindrical vessel, follow the following steps:

Step 1: Calculate geometric parameters:

a. Round attachments:

$$\gamma = \frac{R_m}{t}$$

$$\beta = \frac{0.875r_o}{R_m}$$

b. Square attachments:

$$\gamma = \frac{R_m}{t}$$

$$\beta = \frac{C}{R_m}$$

c. Rectangular attachments:

$$\gamma = \frac{R_m}{t}$$

β values for radial load, longitudinal moment, and circumferential moment vary based on ratios of β_1/β_2. Follow procedures that follow these steps to find β values.

Step 2: Using γ and β values; from Step 1, enter applicable graphs, Figures 7-21 through 7-26 to dimensionless membrane forces and bending moments in shell.

Step 3: Enter values obtained from Figures 7-21 through 7-26 into Table 7-11 and compute stresses.

Step 4: Enter stresses computed in Table 7-11 for various load conditions in Table 7-12. Combine stresses in accordance with sign convention of Table 7-12.

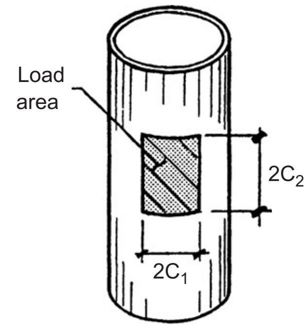


Figure 7-16. Dimensions of load areas.

Computing β Values for Rectangular Attachments

$$\beta_1 = \frac{C_1}{R_m}$$

$$\beta_2 = \frac{C_2}{R_m}$$

$$\frac{\beta_1}{\beta_2}$$

β Values for Radial Load

From Table 7-8 select values of K_1 and K_2 and compute four β values as follows:

If $\frac{\beta_1}{\beta_2} \geq 1$, then β

$$= \left[1 - \frac{1}{3}\left(\frac{\beta_1}{\beta_2} - 1\right)(1 - K_1)\right]\sqrt{\beta_1\beta_2}$$

Table 7-8
β Values of radial loads

	K_1	K_2	β
N_ϕ	0.91	1.48	
N_x	1.68	1.2	
M_ϕ	1.76	0.88	
M_x	1.2	1.25	

Reprinted by permission of the Welding Research Council.

If $\dfrac{\beta_1}{\beta_2} < 1$, then β

$$= \left[1 - \frac{4}{3}\left(1 - \frac{\beta_1}{\beta_2}\right)(1 - \mathrm{K}_2)\right]\sqrt{\beta_1\beta_2}$$

β Values for Longitudinal Moment

From Table 7-9 select values of $\mathrm{C_L}$ and $\mathrm{K_L}$ and compute values of β as follows:

For $\mathrm{N_x}$ and N_ϕ, $\beta = \sqrt[3]{\beta_1\beta_2^2}$

For M_ϕ, $\beta = \mathrm{K_L}\sqrt[3]{\beta_1\beta_2^2}$

For $\mathrm{M_x}$, $\beta = \mathrm{K_L}\sqrt[3]{\beta_1\beta_2^2}$

	$\mathbf{C_L}$	$\mathbf{K_L}$	β
N_ϕ			
$\mathrm{N_x}$			
M_ϕ			
$\mathrm{M_x}$			

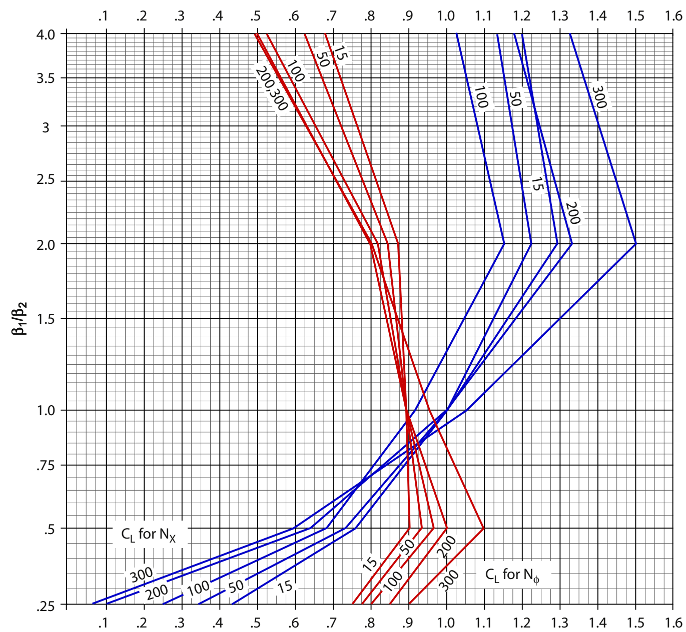


Figure 7-17. Graph of coefficients $\mathrm{C_L}$ for values N_ϕ & $\mathrm{N_X}$ from Table 7-9.

Table 7-9
Coefficients for longitudinal moment, $\mathbf{M_L}$

β_1/β_2	γ	$\mathrm{C_L}$ for N_ϕ	$\mathrm{C_L}$ for $\mathrm{N_x}$	$\mathrm{K_L}$ for M_ϕ	$\mathrm{K_L}$ for $\mathrm{M_x}$
	15	0.75	0.43	1.80	1.24
	50	0.77	0.33	1.65	1.16
0.25	100	0.80	0.24	1.59	1.11
	200	0.85	0.10	1.58	1.11
	300	0.90	0.07	1.56	1.11
	15	0.90	0.76	1.08	1.04
	50	0.93	0.73	1.07	1.03
0.5	100	0.97	0.68	1.06	1.02
	200	0.99	0.64	1.05	1.02
	300	1.10	0.60	1.05	1.02
	15	0.89	1.00	1.01	1.08
	50	0.89	0.96	1.00	1.07
1	100	0.89	0.92	0.98	1.05
	200	0.89	0.99	0.95	1.01
	300	0.95	1.05	0.92	0.96
	15	0.87	1.30	0.94	1.12
	50	0.84	1.23	0.92	1.10
2	100	0.81	1.15	0.89	1.07
	200	0.80	1.33	0.84	0.99
	300	0.80	1.50	0.79	0.91
	15	0.68	1.20	0.90	1.24
	50	0.61	1.13	0.86	1.19
4	100	0.51	1.03	0.81	1.12
	200	0.50	1.18	0.73	0.98
	300	0.50	1.33	0.64	0.83

Reprinted by permission of the Welding Research Council.

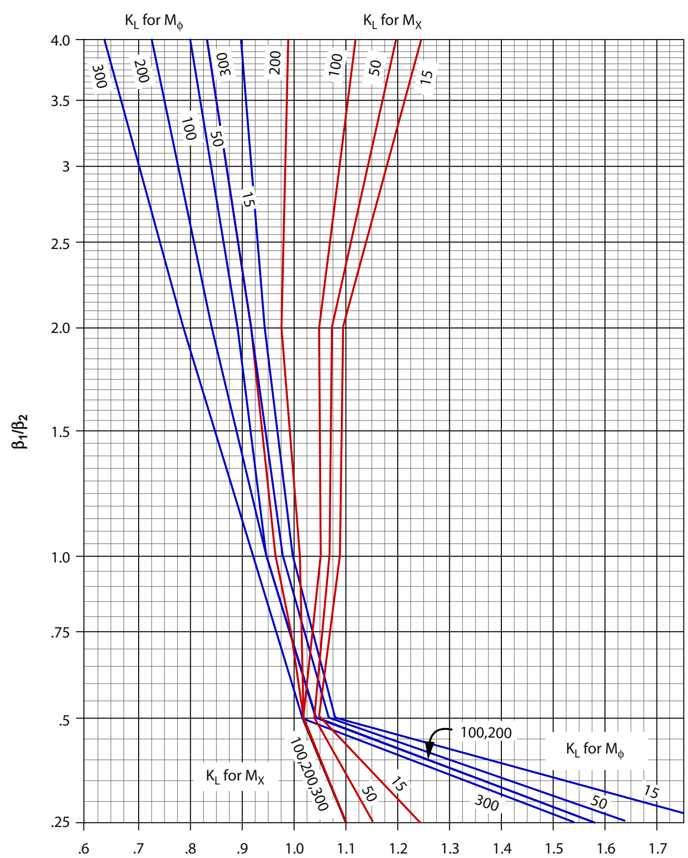


Figure 7-18. Graph of coefficients $\mathrm{K_L}$ for values M_ϕ & $\mathrm{M_X}$ from Table 7-9.

β Values for Circumferential Moment

From Table 7-10 select values of C_c and K_c and compute values of β as follows:

For N_x and N_ϕ, $\beta = \sqrt[3]{\beta_1^2\beta_2}$

For M_ϕ, $\beta = K_c\sqrt[3]{\beta_1^2\beta_2}$

For M_x, $\beta = K_c\sqrt[3]{\beta_1^2\beta_2}$

	C_c	K_c	β
N_ϕ			
N_x			
M_ϕ			
M_x			

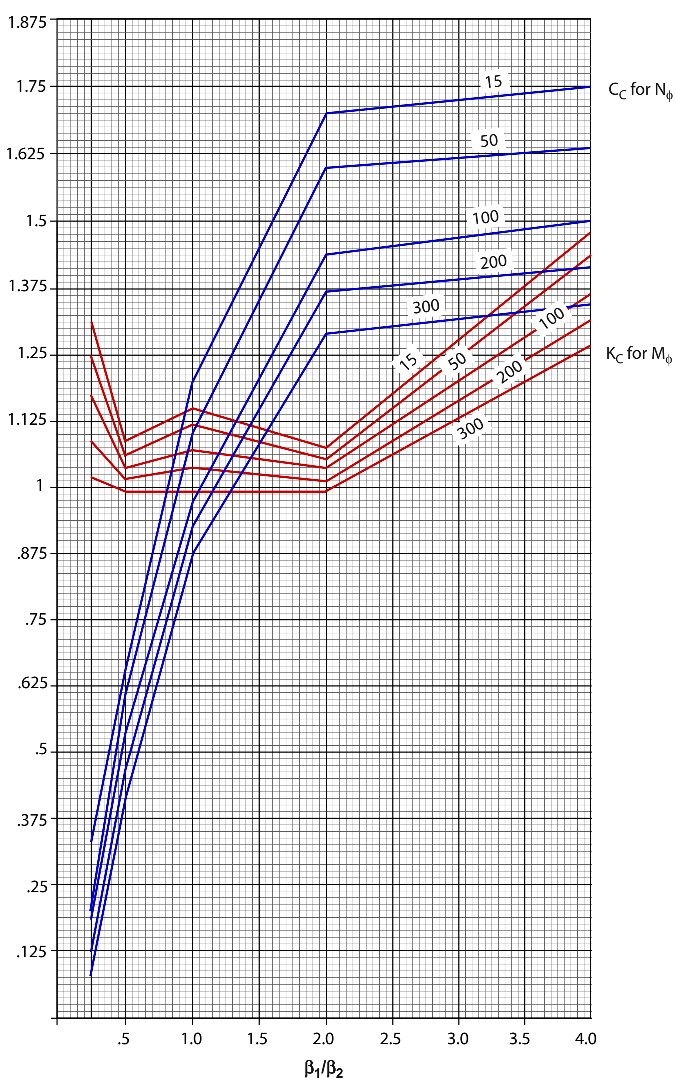


Figure 7-19. Graph of coefficients K_c & C_C for values N_ϕ & M_ϕ from Table 7-10.

Table 7-10
Coefficients for circumferential moment, M_c

β_1/β_2	γ	C_c for N_ϕ	C_c for N_x	K_c for M_ϕ	K_C for M_x
	15	0.31	0.49	1.31	1.84
	50	0.21	0.46	1.24	1.62
	100	0.15	0.44	1.16	1.45
0.25	200	0.12	0.45	1.09	1.31
	300	0.09	0.46	1.02	1.17
	15	0.64	0.75	1.09	1.36
	50	0.57	0.75	1.08	1.31
0.5	100	0.51	0.76	1.04	1.26
	200	0.45	0.76	1.02	1.20
	300	0.39	0.77	0.99	1.13
	15	1.17	1.08	1.15	1.17
	50	1.09	1.03	1.12	1.14
1	100	0.97	0.94	1.07	1.10
	200	0.91	0.91	1.04	1.06
	300	0.85	0.89	0.99	1.02
	15	1.70	1.30	1.20	0.97
	50	1.59	1.23	1.16	0.96
2	100	1.43	1.12	1.10	0.95
	200	1.37	1.06	1.05	0.93
	300	1.30	1.00	1.00	0.90
	15	1.75	1.31	1.47	1.08
	50	1.64	1.11	1.43	1.07
4	100	1.49	0.81	1.38	1.06
	200	1.42	0.78	1.33	1.02
	300	1.36	0.74	1.27	0.98

Reprinted by permission of the Welding Research Council.

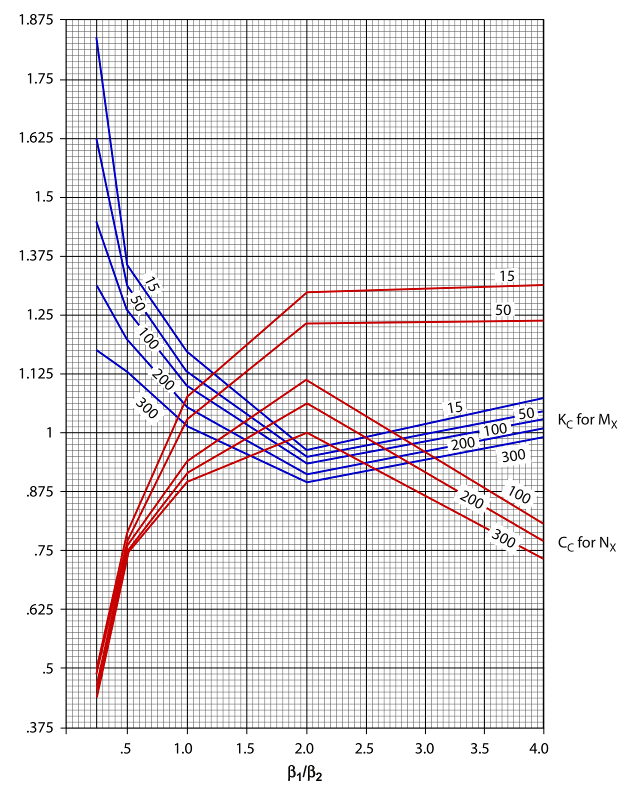


Figure 7-20. Graph of coefficients K_c & C_C for values N_X & M_X from Table 7-10.

Table 7-11
Computing stresses

	Figure	β	Value from Figure	Forces and Moments	Stress
Radial Load					
Membrane	7-21A		$\frac{N_\phi R_m}{P_r} = (\ )$	$N_\phi = \frac{(\ )P_r}{R_m}$	$\sigma_\phi = \frac{K_n N_\phi}{t}$
	7-21B		$\frac{N_x R_m}{P_r} = (\ )$	$N_x = \frac{(\ )P_r}{R_m}$	$\sigma_x = \frac{K_n N_x}{t}$
Bending	7-22A		$\frac{M_\phi}{P_r} = (\ )$	$M_\phi = (\ )P_r$	$\sigma_\phi = \frac{6K_b M_\phi}{t^2}$
	7-22B		$\frac{M_x}{P_r} = (\ )$	$M_x = (\ )P_r$	$\sigma_x = \frac{6K_b M_x}{t^2}$
Longitudinal Moment					
Membrane	7-23A		$\frac{N_\phi R_m^2 \beta}{M_L} = (\ )$	$N_\phi = \frac{(\ )C_L M_L}{R_m^2 \beta}$	$\sigma_\phi = \frac{K_n N_\phi}{t}$
	7-23B		$\frac{N_x R_m^2 \beta}{M_L} = (\ )$	$N_x = \frac{(\ )C_L M_L}{R_m^2 \beta}$	$\sigma_x = \frac{K_n N_x}{t}$
Bending	7-24A		$\frac{M_\phi R_m \beta}{M_L} = (\ )$	$M_\phi = \frac{(\ )M_L}{R_m \beta}$	$\sigma_\phi = \frac{6K_b M_\phi}{t^2}$
	7-24B		$\frac{M_x R_m \beta}{M_L} = (\ )$	$M_x = \frac{(\ )M_L}{R_m \beta}$	$\sigma_x = \frac{6K_b M_x}{t^2}$
Circumferential Moment					
Membrane	7-25A		$\frac{N_\phi R_m^2 \beta}{M_c} = (\ )$	$N_\phi = \frac{(\ )C_c M_c}{R_m^2 \beta}$	$\sigma_\phi = \frac{K_n N_\phi}{t}$
	7-25B		$\frac{N_x R_m^2 \beta}{M_C} = (\ )$	$N_x = \frac{(\ )C_c M_c}{R_m^2 \beta}$	$\sigma_x = \frac{K_n N_x}{t}$
Bending	7-26A		$\frac{M_\phi R_m \beta}{M_c} = (\ )$	$M_\phi = \frac{(\ )M_c}{R_m \beta}$	$\sigma_\phi = \frac{6K_b M_\phi}{t^2}$
	7-26B		$\frac{M_x R_m \beta}{M_c} = (\ )$	$M_x = \frac{(\ )M_c}{R_m \beta}$	$\sigma_x = \frac{6K_b M_x}{t^2}$

Table 7-12
Combining stresses

Stress Due To			σ_x				σ_ϕ			
			0°	90°	180°	270°	0°	90°	180°	270°
Radial load, P_r (Sign is (+) for outward load, (−) for inward load)	Membrane	N_ϕ								
		N_x								
	Bending	M_ϕ								
		M_x								
Longitudinal moment, M_L	Membrane	N_ϕ					+		−	
		N_x	+		−					
	Bending	M_ϕ					+		−	
		M_x	+		−					
Circumferential moment, M_c	Membrane	N_ϕ						+		−
		N_x		+		−				
	Bending	M_ϕ						+		−
		M_x		+		−				
Internal pressure, P	$\sigma_\phi = \frac{PR_m}{t} =$						+	+	+	+
	$\sigma_x = \frac{PR_m}{2t} =$		+	+	+	+				
Total, Σ										

Note: *Only absolute value of quantities are used. Combine stresses utilizing sign convention of table.*

Shear Stresses

- *Stress due to shear loads, V_L or V_c.*
  Round attachments:

$$\tau_s = \frac{V_L}{\pi r_o t}$$

$$\tau_s = \frac{V_C}{\pi r_o t}$$

Square attachments:

$$\tau_s = \frac{V_L}{4Ct}$$

$$\tau_s = \frac{V_C}{4Ct}$$

Rectangular attachments:

$$\tau_s = \frac{V_L}{4C_1 t}$$

$$\tau_s = \frac{V_C}{4C_2 t}$$

- *Stress due to torsional moment, M_T.*
  Round attachments only!

$$\tau_T = \frac{M_T}{2\pi r_o^2 t}$$

Notes

1. Figure 7-15 should be used if the vessel is in brittle (low temperature) or fatigue service. For brittle fracture the maximum tensile stress is governing. The stress concentration factor is applied to the stresses which are perpendicular to the change in section.
2. Subscripts θ and C indicate circumferential direction, X and L indicate longitudinal direction.
3. Only rectangular shapes where C_1/C_2 is between 1/4 and 4 can be computed by this procedure. The charts and graphs are not valid for lesser or greater ratios.
4. Methods of reducing stresses from local loads:
   a. Add reinforcing pad.
   b. Increase shell thickness.
   c. Add partial ring stiffener.
   d. Add circumferential ring stiffener(s).
   e. Kneebrace to reduce moment loads.
   f. Increase attachment size.
5. See Procedure 7-3 to convert irregular attachment shapes into suitable shapes for design procedure.
6. For radial loads the stress on the circumferential axis will always govern.
7. The maximum stress due to a circumferential moment is 2–5 times larger than the stress due to a longitudinal moment of the same magnitude.
8. The maximum stress from a longitudinal moment is not located on the longitudinal axis of the vessel and may be 60°–70° off the longitudinal axis. The reason for the high stresses on or adjacent to the circumferential axis is that, on thin shells, the longitudinal axis is relatively flexible and free to deform and that the loads are thereby transferred toward the circumferential axis which is less free to deform. Figures 7-23 and 7-24 do not show maximum stresses since their location is unknown. Instead the stress on the longitudinal axis is given.
9. *For attachments with reinforcing pads:* This applies only to attachments that are welded to a reinforcing plate that is subsequently welded to the vessel shell. Attachments that are welded through the pad (like nozzles) can be considered as integral with the shell.

Moment loadings for nonintegral attachments must be converted into radial loads. This will more closely approximate the manner in which the loads are distributed in shell and plate. Stresses should be checked at the edge of attachment *and* edge of reinforcing plate. The maximum height of reinforcing pad to be considered is given by:
For radial load:

$$2d_2 \max = \frac{2C_2 d_1}{C_1}$$

For longitudinal moment:

$$2d_{21} \max = \frac{4C_2 d_1}{3C_1}$$

For circumferential moment:

$$2d_{11} \max = \frac{4C_1 d_2}{3C_2}$$

Moments can be converted as follows:

$$P_r = \frac{3M_L}{4C_2}$$

or

$$P_r = \frac{3M_c}{4C_1}$$

10. This procedure is based on the principle of "flexible load surfaces." Attachments larger than one-half the vessel diameter (β>0.5) cannot be determined by this procedure. For attachments which exceed these parameters see Procedure 7-1.

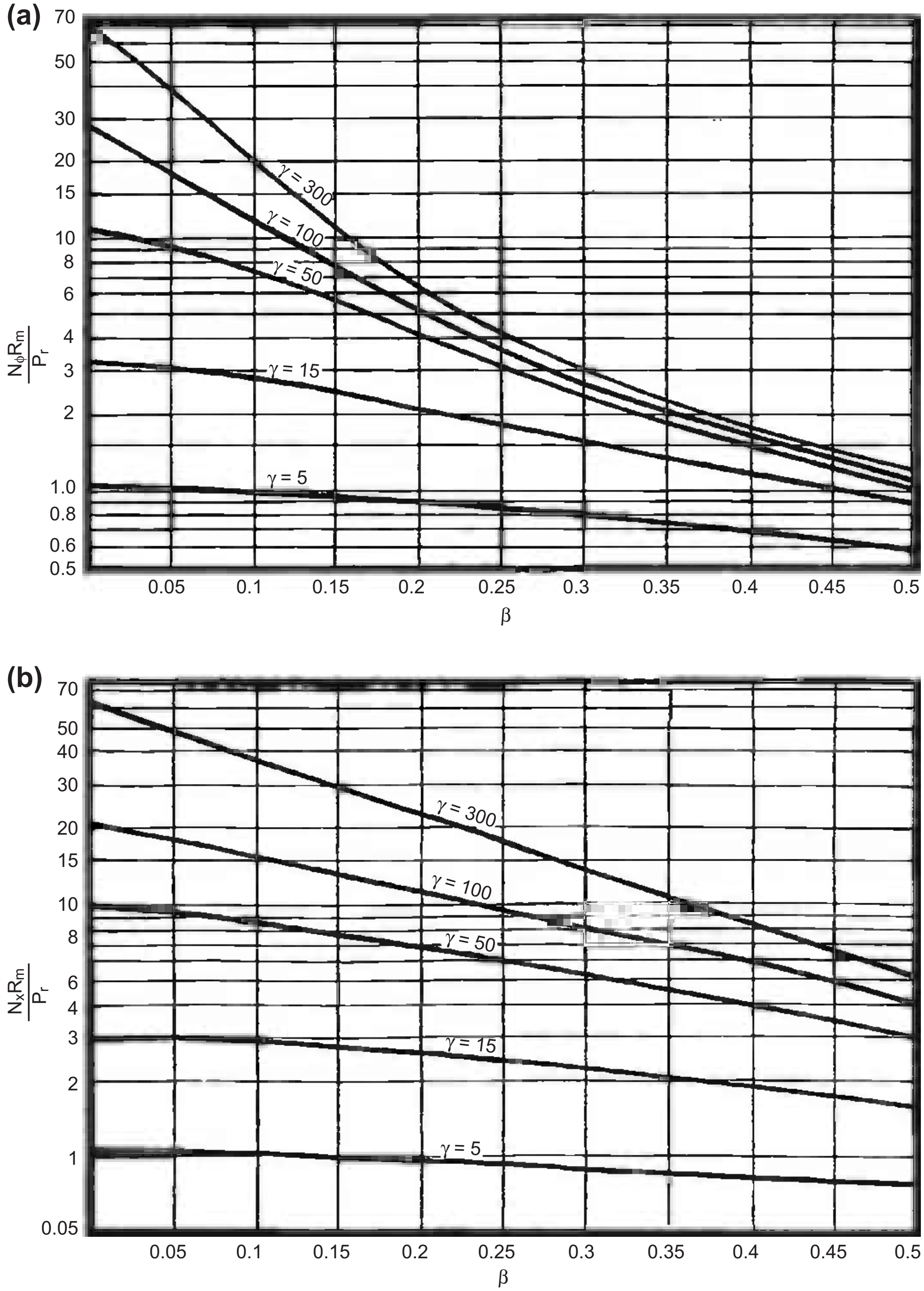


Figure 7-21. Membrane force in a cylinder due to radial load on an external attachment. (*Reprinted by permission from the Welding Research Council.*)

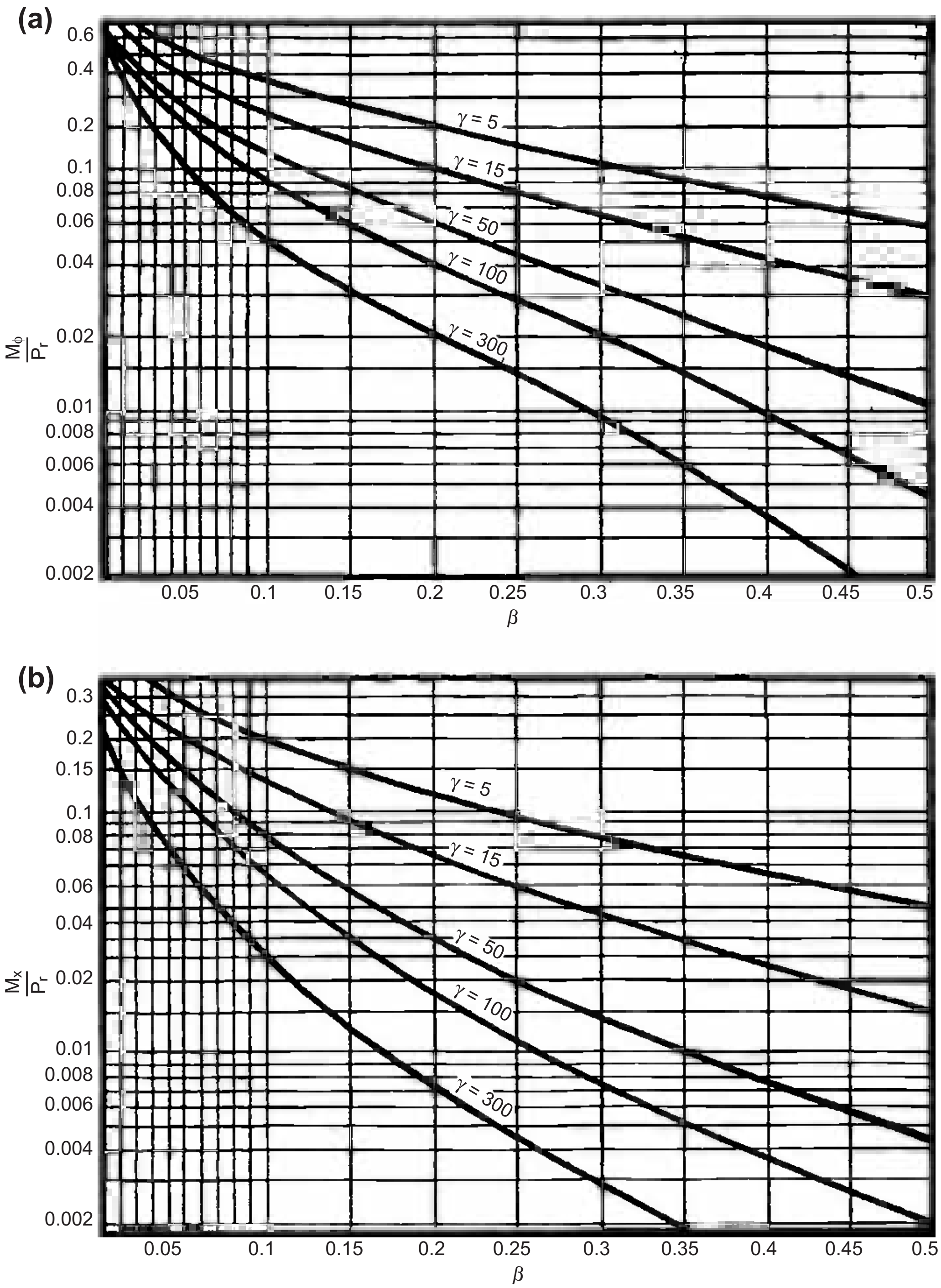


Figure 7-22. Bending moment in a cylinder due to radial load on an external attachment. (*Reprinted by permission from the Welding Research Council.*)

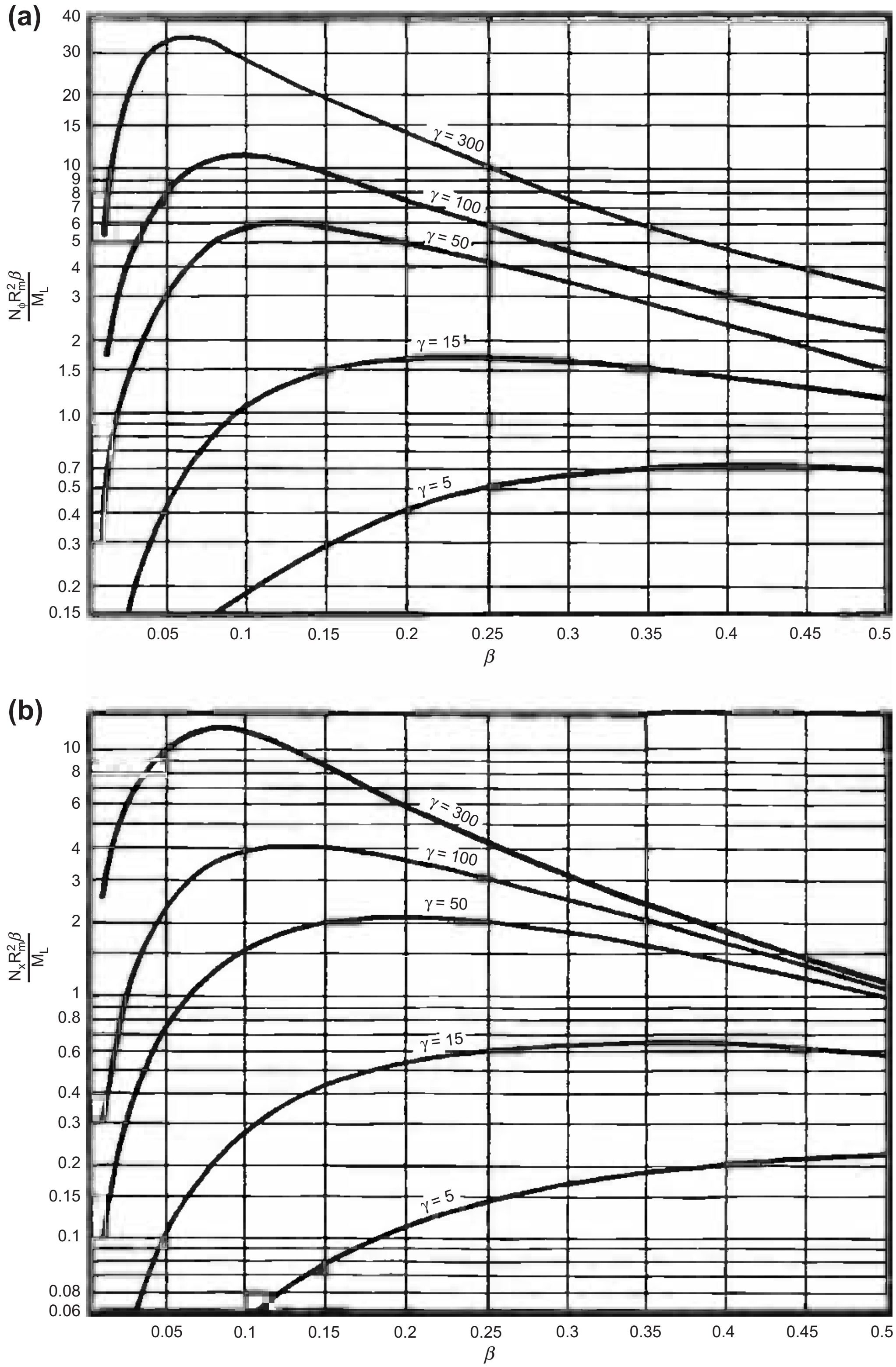


Figure 7-23. Membrane force in a cylinder due to longitudinal moment on an external attachment. (*Reprinted by permission from the Welding Research Council.*)

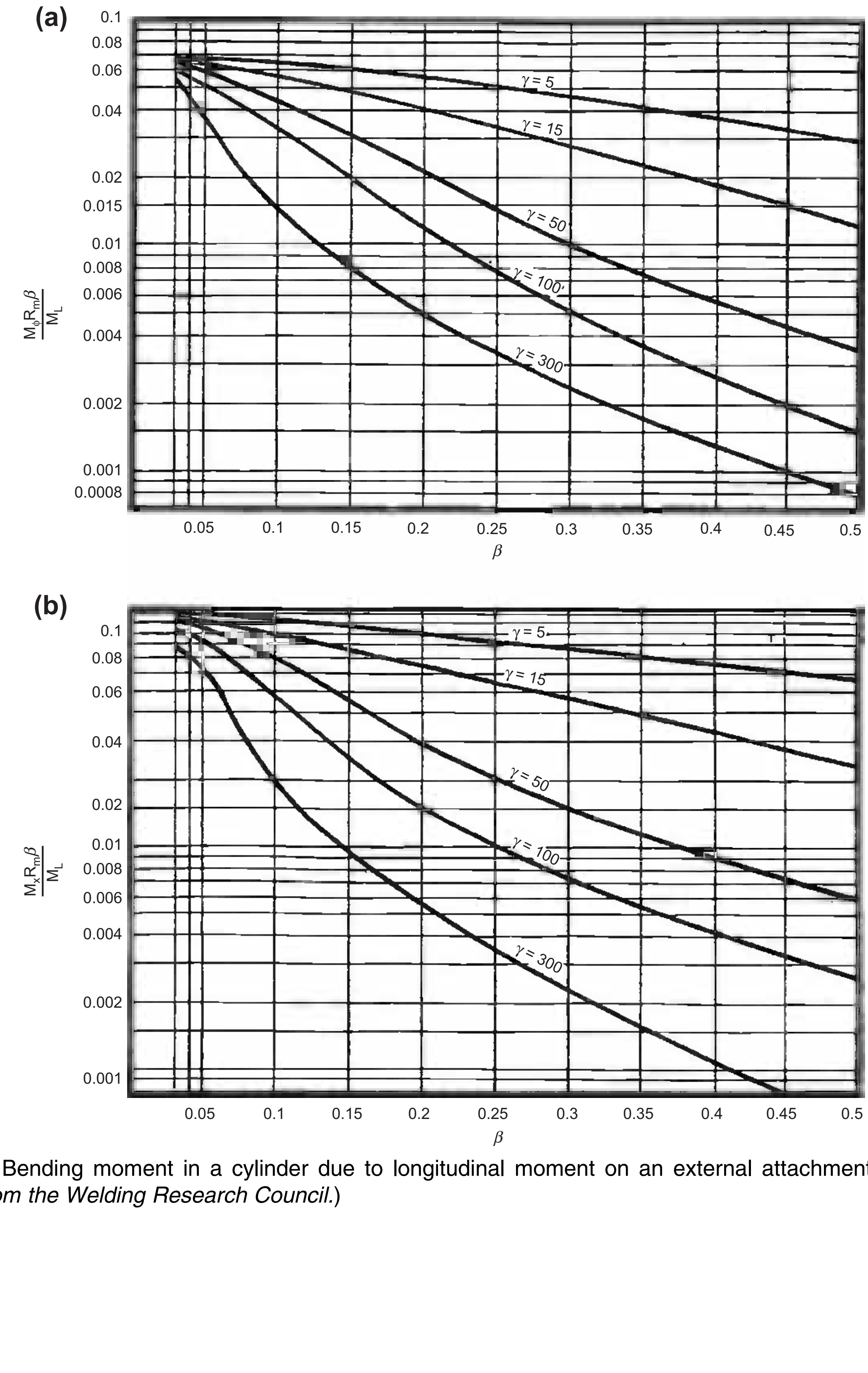


Figure 7-24. Bending moment in a cylinder due to longitudinal moment on an external attachment. (*Reprinted by permission from the Welding Research Council.*)

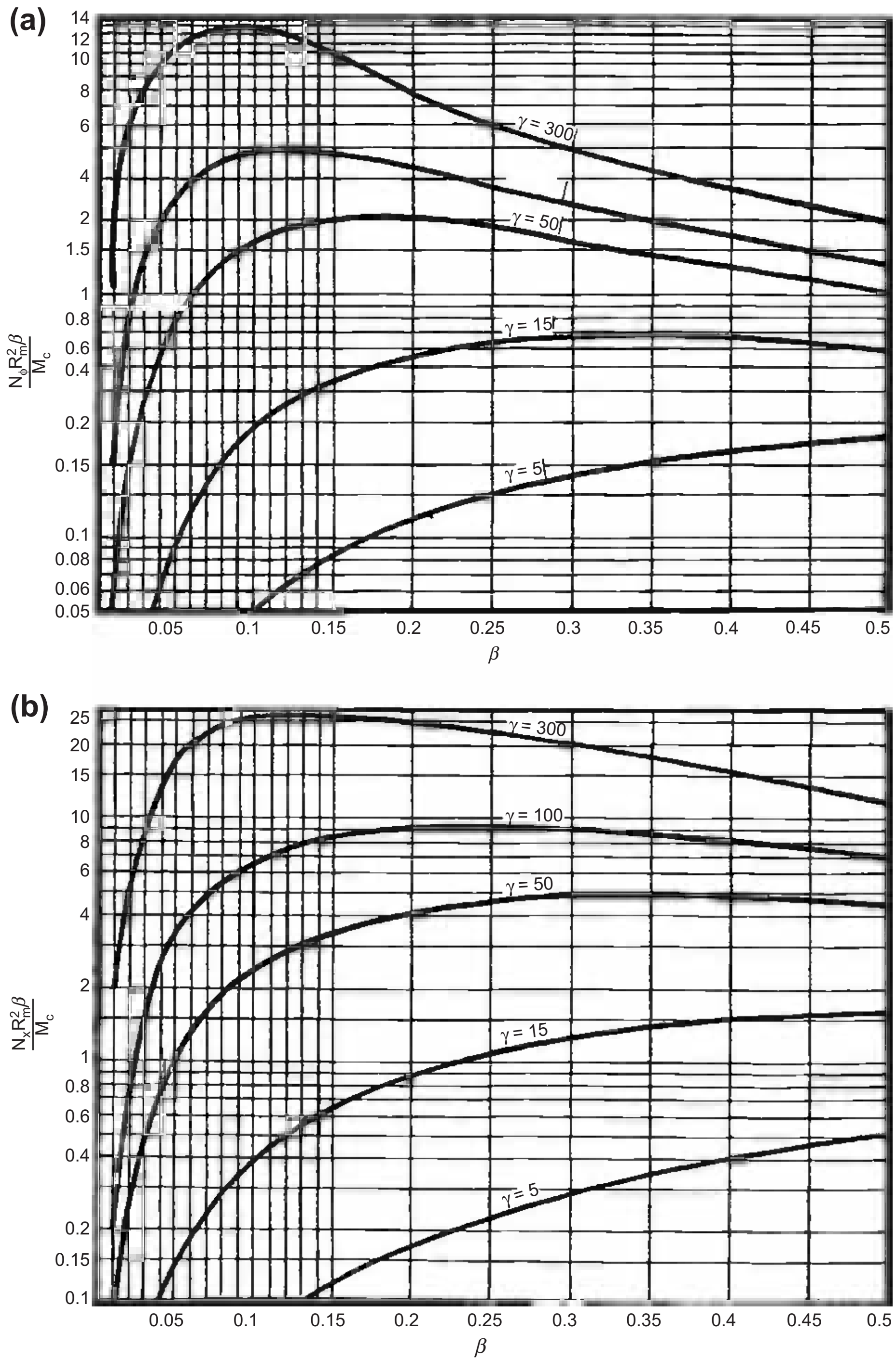


Figure 7-25. Membrane force in a cylinder due to circumferential moment on an external attachment. (*Reprinted by permission from the Welding Research Council.*)

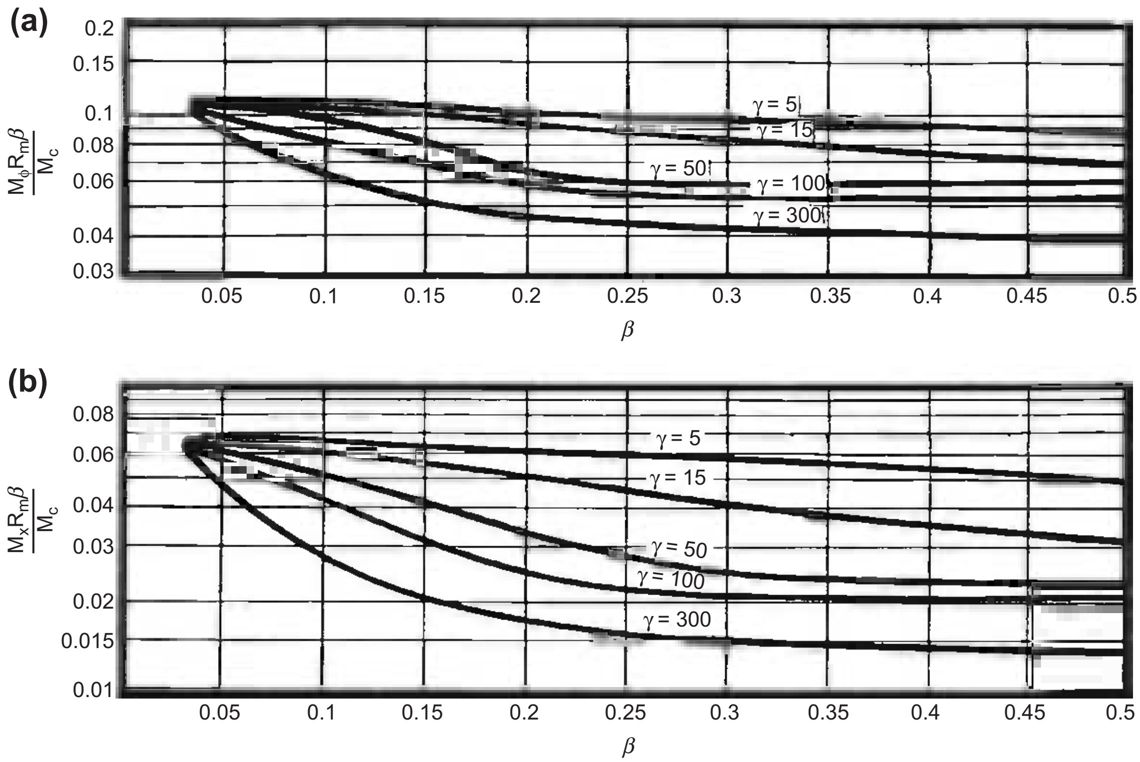


Figure 7-26. Bending moment in a cylinder due to circumferential moment on an external attachment. (*Reprinted by permission from the Welding Research Council.*)

Maximum Allowable Nozzle Loads

This procedure is an alternative work process for developing and analyzing nozzle loads on pressure vessels. It establishes the minimum criteria for design by providing the maximum allowable nozzle load by size and class of flange rating. This is an alternative work process to determining each individual nozzle load after the piping system is designed. This procedure eliminates much of the late design changes that occur when late data on nozzle loads impact either the design of the piping system or the stresses in the vessel shell.

The allowable loads and moments listed in Table 7-14 do not represent any actual loading or a real maximum allowable load or moment. Rather they are "arbitrary" maximum allowable nozzle loads. This procedure does not take into account any of the vessel parameters such as diameter, thickness, material, temperature, allowable stress, internal pressure, etc. Since the basis of Table 7-14 is the "flange rating", the associated nozzle loads are generic only.

There are two reasons for the implementation of this procedure as follows;

1. To provide the vessel fabricator with nozzle loads with which to design the vessel shell or head to which the nozzle is attached, prior to design of the piping systems.
2. To provide the piping designer with guidelines for design of piping that terminates at a vessel nozzle. Therefore, as long as the piping does not exceed the loads in Table 7-14, they automatically know that the vessel shell or head is not overstressed for this condition.

If the piping department cannot design the piping in such a way as to not exceed the values in Table 7-14, then the work process must revert back to the original work process of analyzing each specific, individual nozzle for the actual loads and resultant shell stresses. This would become an iterative work process between the vessel designer and the piping designer where actual loads will dictate the ultimate design.

In the event that the nozzle loads still result in excessive shell stresses after the iterative work process is conducted, the loadings may be reduced by recalculating the piping loads utilizing the "vessel spring rate". Often times the loads determined using the vessel spring rate will be much less than that determined by the normal process of considering the vessel as a "rigid anchor".

In general the following notes apply to this procedure;

1. Each nozzle, including those designated as "spare", but with the exception of manways and instrument nozzles, shall be designed to withstand the forces and moments specified in Table 7-14. The indicated loads are to be considered to act at the shell/head to nozzle intersection and to be true normal and tangential to the shell at that point. The effect on the shell/head shall be analyzed per an acceptable local load procedure such as WRC # 107.
2. With regard to radial load (P_r), calculations shall be made first with the force acting radially outwards in conjunction with the internal pressure and then with the force acting inwards. In the second instance, the internal pressure shall not be used to oppose the compressive stresses due to the force acting radially inwards; for this load condition a null pressure condition is to be considered to exist.
3. Values in Table 7-14 were computed by the coefficients and equations given in Table 7-13. In Table 7-13, the variables shown are "D", the nominal diameter in inches and "β", the value listed against the nozzle flange rating. These variables and equations were used in the development of Table 7-14.
4. Whenever shell or head stresses exceed the allowable stress for local loadings, the vendor shall apply

Table 7-13
Coefficients used for determination of maximum allowable nozzle loads

Flange Rating Class	150	300	600	900	1500	2500
β Value	0.6	0.7	0.8	0.9	1	1.1

Equations:

Longitudinal Bending Moment (Ft- Lbs)	$M_L = \beta \text{ X } 110 \text{ X } D^2$
Circumferential Bending moment (Ft-Lbs)	$M_\varphi = \beta \text{ X } 85 \text{ X } D^2$
Resultant Bending Moment, (Ft-Lbs)	$M_R = [M_L^2 + M_\varphi^2]^{.5} = \beta \text{ X } 140 \text{ X } D^2$
Radial Load (tension or compression) (Lbs)	$P_r = \beta \text{ X } 500 \text{ X } D$

Table 7-14
Maximum allowable nozzle loads

	FLANGE RATING							
	CLASS 150 FLANGES				CLASS 300 FLANGES			
	Force, Lbs	Bending Moment, Ft-Lbs			Force, Lbs	Bending Moment, Ft-Lbs		
NPS (Inches)	Radial Load, P_r	Longitudinal, M_L	Circumferential, M_φ	Resultant, M_R	Radial Load, P_r	Longitudinal, M_L	Circumferential, M_φ	Resultant, M_R
2	600	264	204	336	700	308	238	392
3	900	594	459	756	1050	693	536	882
4	1200	1056	816	1344	1400	1232	952	1568
6	1800	2376	1836	3024	2100	2772	2142	3528
8	2400	4224	3264	5376	2800	4928	3808	6272
10	3000	6600	5100	8400	3500	7700	5950	9800
12	3600	9504	7344	12096	4200	11088	8568	14112
14	4200	12936	9996	16464	4900	15092	11662	19208
16	4800	16896	13056	21504	5600	19712	15232	25088
18	5400	21384	16524	27216	6300	24948	19278	31752
20	6000	26400	20400	33600	7000	30800	23800	39200
24	7200	38016	29376	48384	8400	44352	34272	56448

	FLANGE RATING							
	CLASS 600 FLANGES				CLASS 900 FLANGES			
	Force, Lbs	Bending Moment, Ft-Lbs			Force, Lbs	Bending Moment, Ft-Lbs		
NPS (Inches)	Radial Load, P_r	Longitudinal, M_L	Circumferential, M_φ	Resultant, M_R	Radial Load, P_r	Longitudinal, M_L	Circumferential, M_φ	Resultant, M_R
2	800	352	272	448	900	396	306	504
3	1200	792	612	1008	1350	891	689	1134
4	1600	1408	1088	1792	1800	1584	1224	2016
6	2400	3168	2448	4032	2700	3564	2754	4536
8	3200	5632	4352	7168	3600	6336	4896	8064
10	4000	8800	6800	11200	4500	9900	7650	12600
12	4800	12672	9792	16128	5400	14256	11016	18144
14	5600	17248	13328	21952	6300	19404	14994	24696
16	6400	22528	17408	28672	7200	25344	19584	32256
18	7200	28512	22032	36288	8100	32076	24786	40824
20	8000	35200	27200	44800	9000	39600	30600	50400
24	9600	50688	39168	64512	10800	57024	44064	72576

	FLANGE RATING							
	CLASS 1500 FLANGES				CLASS 2500 FLANGES			
	Force, Lbs	Bending Moment, Ft-Lbs			Force, Lbs	Bending Moment, Ft-Lbs		
NPS (Inches)	Radial Load, P_r	Longitudinal, M_L	Circumferential, M_φ	Resultant, M_R	Radial Load, P_r	Longitudinal, M_L	Circumferential, M_φ	Resultant, M_R
2	1000	440	340	560	1100	484	374	616
3	15000	990	765	1260	1650	1089	842	1386
4	2000	1760	1360	2240	2200	1936	1496	2464
6	3000	3960	3060	5040	3300	4356	3366	5544
8	4000	7040	5440	8960	4400	7744	5984	9856
10	5000	11000	8500	14000	5500	12100	9350	15400
12	6000	15840	12240	20160	6600	17424	13464	22176
14	7000	21560	16660	27440	7700	23716	18326	30184
16	8000	28160	21760	35840	8800	30976	23936	39424
18	9000	35640	27540	45360	9900	39204	30294	49896
20	10000	44000	34000	56000	11000	48400	37400	61600
24	12000	63360	48960	80640	13200	69696	53856	88704

adequate reinforcement and or increase thickness of shell and /or nozzle locally.

5. For nozzles on a formed head or sphere, the resultant bending moment is to be compared with M_R.
6. Shear and torsion effects are omitted from consideration because they have negligible effects on final stress resultants.
7. The loadings computed from these equations shall be considered as caused by 67% thermal and 33% dead weight load.
8. Under vacuum conditions, the deflection. ς, adjacent to the nozzle should be limited to the following;

$\varsigma < .0025$ R where R is the radius of the shell or head.

Procedure 7-5: Stresses in Spherical Shells from External Local Loads [11–13]

Notation

P_r = external radial load, lb
M = external moment, in.-lb
R_m = mean radius of sphere, crown radius of F & D, dished or ellipsoidal head, in.
r_o = outside radius of cylindrical attachment, in.
C = half side of square attachment, in.
N_x = membrane force in shell, meridional, lb/in.
N_ϕ = membrane force in shell, latitudinal, lb/in.
M_x = internal bending moment, meridional, in.-lb/in.
M_ϕ = internal bending moment, latitudinal, in.-lb/in.
K_n,K_b = stress concentration factors (See Note 3)
U,S = coefficients
σ_x = meridional stress, psi
σ_ϕ = latitudinal stress, psi
T_e = thickness of reinforcing pad, in.
τ = shear stress, psi
M_T = torsional moment, in.-lb
V = shear load, lb

Procedure

To calculate stress due to radial load (P_r), and/or moment (M), on a spherical shell or head:

1. Calculate value "S" to find stresses at distance *x* from centerline or value "U" at edge of attachment. *Note:* At edge of attachment, S = U. Normally stress there will govern.
2. From Figures 7-29 to 7-32 determine coefficients for membrane and bending forces and enter values in Table 7-15.
3. Compute stresses in Table 7-15. These stresses are entered into Table 7-16 based on the type of stress (membrane or bending) and the type of load that produced that stress (radial load or moment).
4. Stresses in Table 7-16 are added vertically to total at bottom.

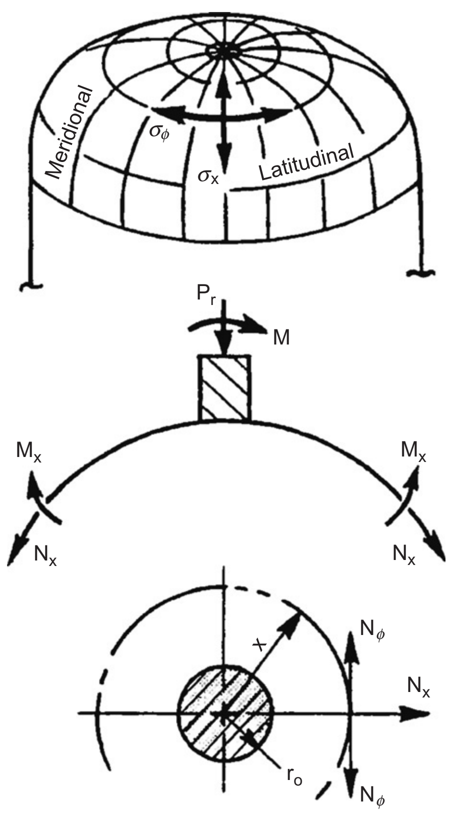


Figure 7-27. Loadings and forces at local attachments in spherical shells.

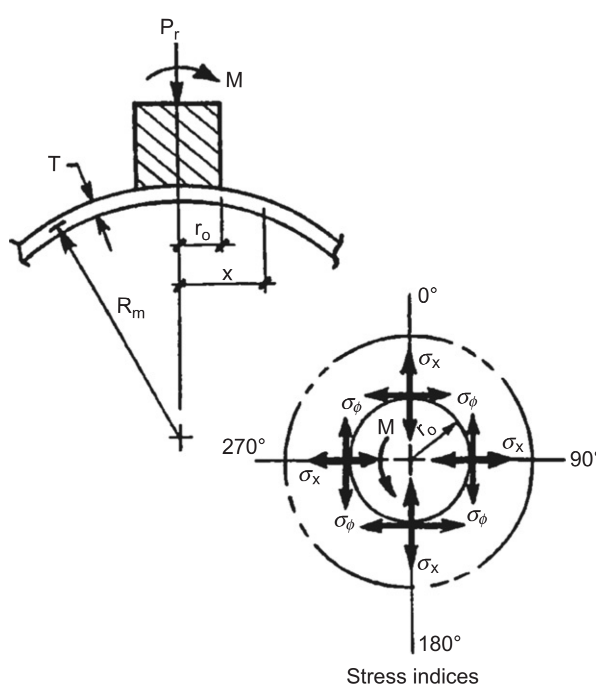


Figure 7-28. Dimensions and stress indices of local attachments.

Table 7-15
Computing stresses

	Figure	Value from Figure	Stresses	
			Radial Load	
Membrane	7-29A	$\frac{N_x T}{P_r} = (\ )$		$\sigma_x = (\ )\frac{K_n P_r}{T^2}$
	7-29B	$\frac{N_\phi T}{P_r} = (\ )$		$\sigma_\phi = (\ )\frac{K_n P_r}{T^2}$
Bending	7-30A	$\frac{M_x}{P_r} = (\ )$		$\sigma_x = (\ )\frac{6K_b P_r}{T^2}$
	7-30B	$\frac{M_\phi}{P_r} = (\ )$		$\sigma_\phi = (\ )\frac{6K_b P_r}{T^2}$
			Moment	
Membrane	7-31A	$\frac{N_x T\sqrt{R_m T}}{M} = (\ )$		$\sigma_x = (\ )\frac{K_n M}{T^2\sqrt{R_m T}}$
	7-31B	$\frac{N_\phi T\sqrt{R_m T}}{M} = (\ )$		$\sigma_\phi = (\ )\frac{K_n M}{T^2\sqrt{R_m T}}$
Bending	7-32A	$\frac{M_x\sqrt{R_m T}}{M} = (\ )$		$\sigma_x = (\ )\frac{6K_b M}{T^2\sqrt{R_m T}}$
	7-32B	$\frac{M_\phi\sqrt{R_m T}}{M} = (\ )$		$\sigma_\phi = (\ )\frac{6K_b M}{T^2\sqrt{R_m T}}$

Table 7-16
Combining stresses

Stress Due To			σ_x 0°	σ_x 90°	σ_x 180°	σ_x 270°	σ_ϕ 0°	σ_ϕ 90°	σ_ϕ 180°	σ_ϕ 270°
Radial load. P_t (Sign is (+) for outward radial load. (–) for inward load)	Membrane	N_x								
		N_ϕ								
	Bending	M_x								
		M_ϕ								
Moment, M	Membrane	N_x	+		–					
		N_ϕ					+		–	
	Bending	M_x	+		–					
		M_ϕ					+		–	
Total		Σ								

Note: *Only absolute values of quantities are used. Combine stresses utilizing sign convention of table.*

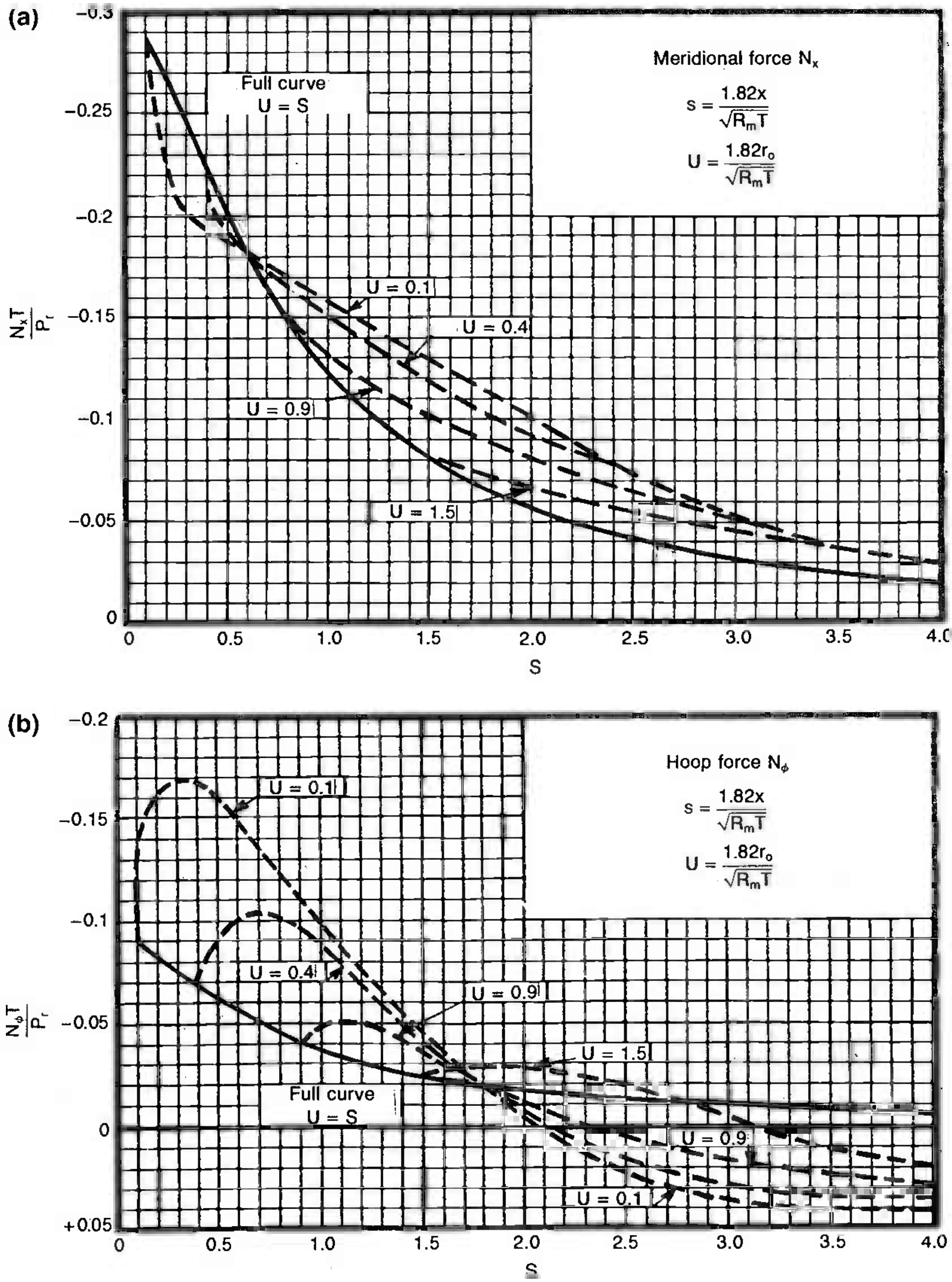


Figure 7-29. Membrane force due to P_r. (*Extracts from BS 5500:1985 are reproduced by permission of British Standards Institution, 2 Park Street, London, W1A 2BS, England. Complete copies can be obtained from national standards bodies.*)

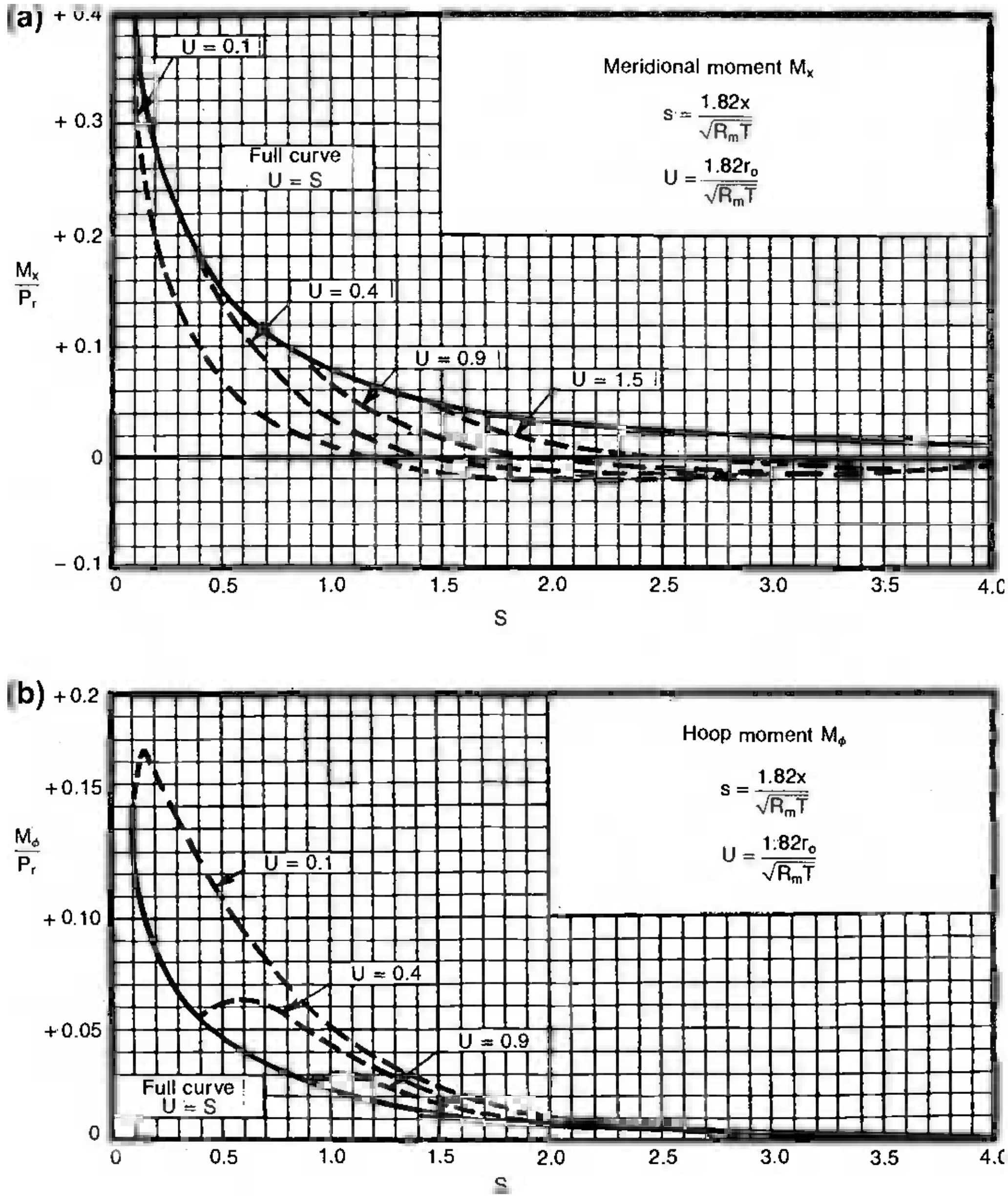


Figure 7-30. Bending moment due to P_r. (*Extracts from BS 5500:1985 are reproduced by permission of the British Standards Institution, 2 Park Street, London, W1A 2BS, England. Complete copies can be obtained from national standards bodies.*)

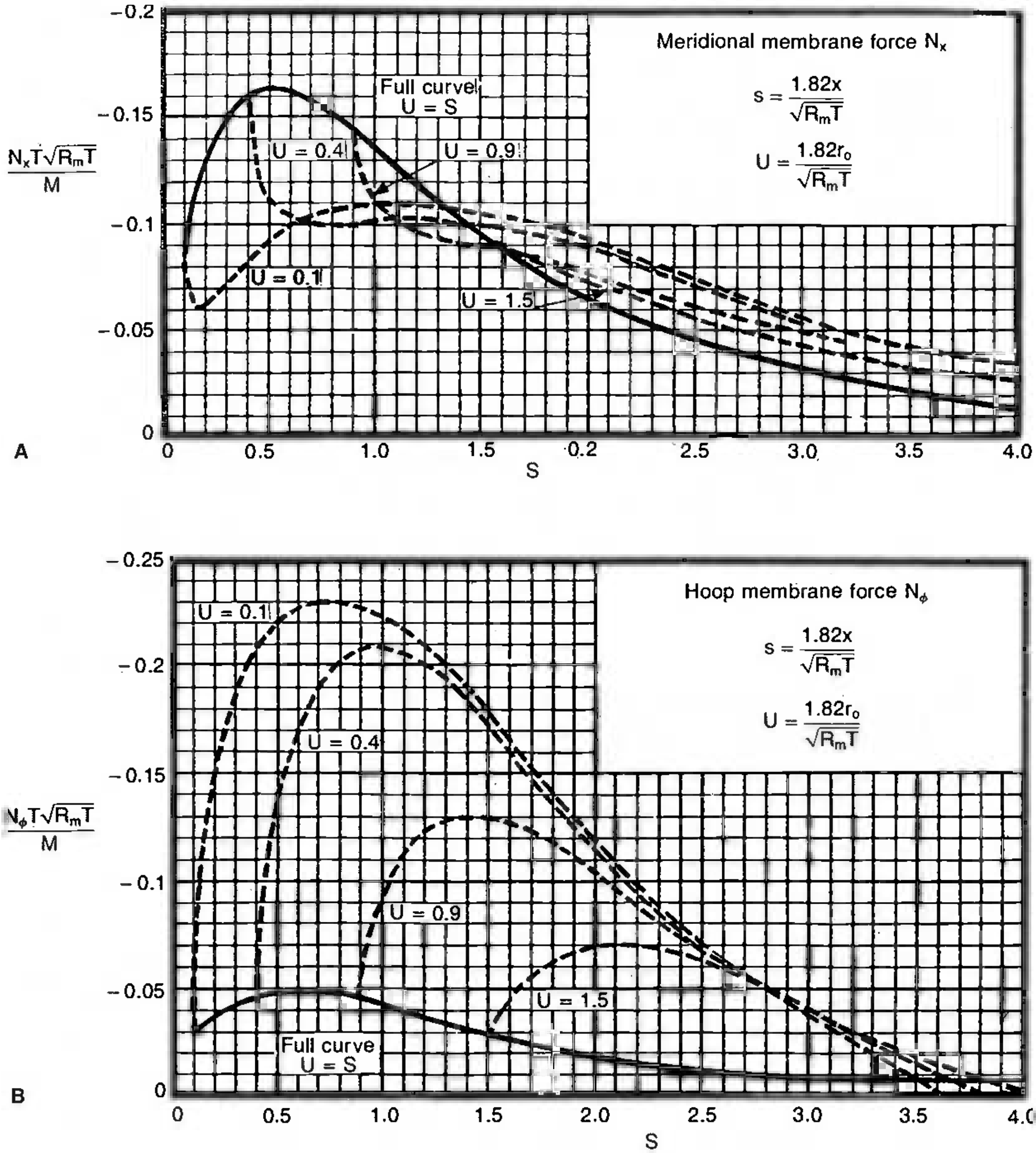


Figure 7-31. Membrane force due to M. (*Extracts from BS 5500:1985 are reproduced by permission of the British Standards Institution, 2 Park Street, London, W1A 2BS, England. Complete copies can be obtained from national standards bodies.*)

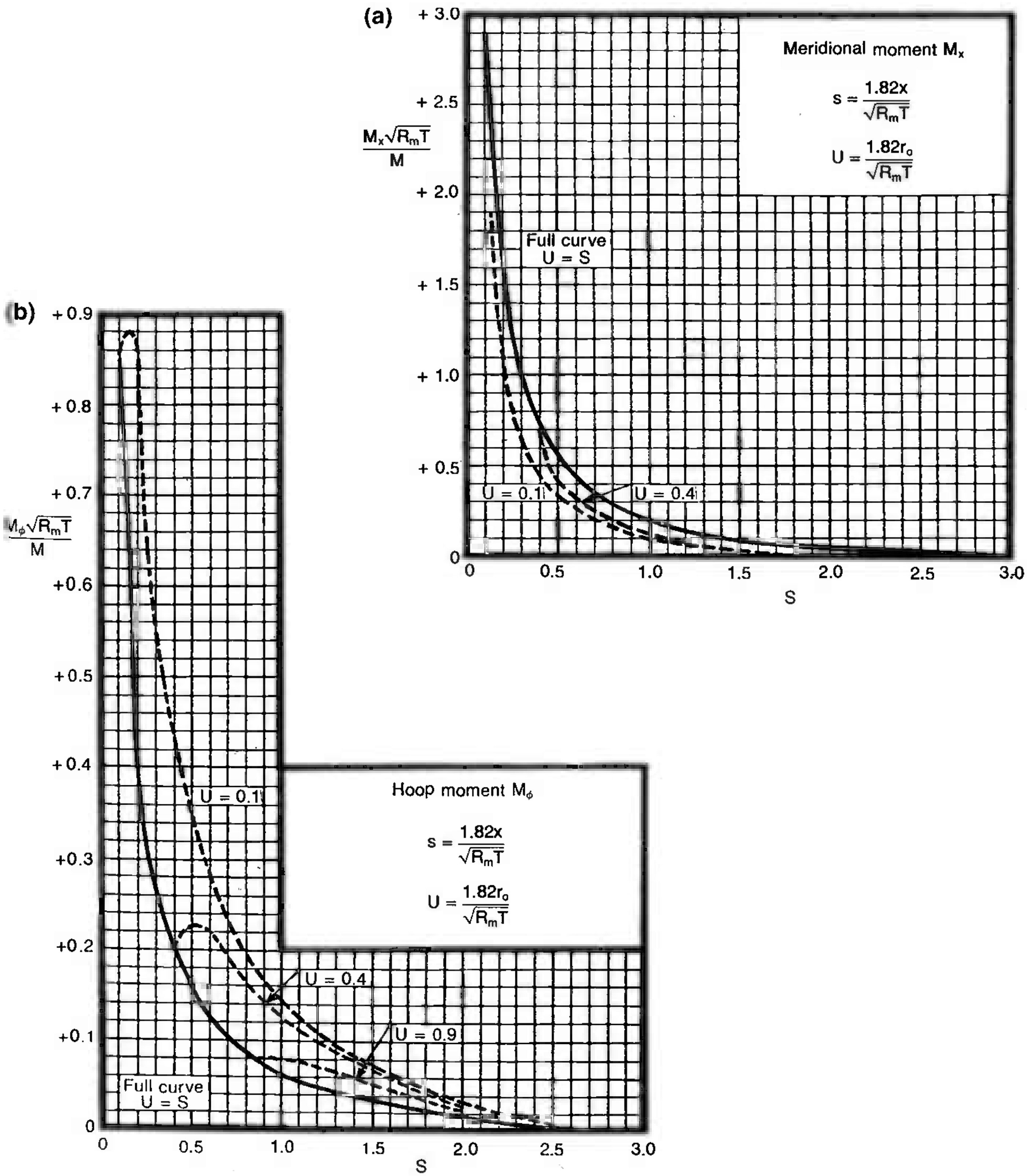


Figure 7-32. Bending moment due to M. (*Extracts from BS 5500:1985 are reproduced by permission of the British Standards Institution, 2 Park Street, London, W1A 2BS, England. Complete copies can be obtained from national standards bodies.*)

Formulas

- *For square attachments.*

$r_o = C$

- *For rectangular attachments.*

$r_o = \sqrt{C_x C_\phi}$

- *For multiple moments.*

$M = \sqrt{M_1^2 + M_2^2}$

- *For multiple shear forces.*

$V = \sqrt{V_1^2 + V_2^2}$

- *General stress equation.*

$\sigma = \frac{N_i}{T} \pm \frac{6M_i}{T^2}$

- *For attachments with reinforcing pads.*

T at edge of attachment $= \sqrt{T^2 + T_e^2}$

T at edge of pad $= T$

- *Shear stresses.*
  Due to shear load

$\tau = \frac{V}{\pi r_o T}$

Due to torsional moment, M_T

$\tau = \frac{M_T}{2\pi r_o^2 T}$

Stress Indices, Loads, and Geometric Parameters

$r_o =$

$R_m =$

$T =$

$K_n =$

$K_b =$

$P_r =$

$M =$

$S = \frac{1.82x}{\sqrt{R_m T}}$

$U = \frac{1.82 r_o}{\sqrt{R_m T}}$

Notes

1. This procedure is based on the "Theory of Shallow Spherical Shells".
2. Because stresses are local and die out rapidly with increasing distance from point of application, this procedure can be applied to the spherical portion of the vessel heads as well as to complete spheres.
3. For "Stress Concentration Factors" see "Stresses in Cylindrical Shells from External Local Loads", Procedure 7-4.
4. For convenience, the loads are considered as acting on a rigid cylindrical attachment of radius r_o. This will yield approximate results for hollow attachments. For more accurate results for hollow attachments, consult WRC Bulletin 107 [11].
5. The stresses found from these charts will be reduced by the effect of internal pressure, but this reduction is small and can usually be neglected in practice. Bijlaard found that for a spherical shell with $R_m/T = 100$, and internal pressure causing membrane stress of 13,000 psi, the maximum deflection was decreased by only 4%–5% and bending moment by 2%. In a cylinder with the same R_m/T ratio, these reductions were about 10 times greater. This small reduction for spherical shells is caused by the smaller and more localized curvatures caused by local loading of spherical shells.

References

[1] Roark RJ. In: Formulas for Stress and Strain. 5th Edition. New York: McGraw-Hill Book Co; 1975.
[2] Isakower RI. Ring Redundants. Machine Design March 1965.
[3] Blake A. Stresses in Flanges and Support Rings. Machine Design September 1974.
[4] Samoiloff A. Investigation of Stress in Circular Rings. Petroleum Refiner July 1947.
[5] Blake A. Rings and Arcuate Beams. Product Engineering January 1963.
[6] Blodgett OW. Design of Welded Structures. The James F. Lincoln Arc Welding Foundation 1966. Section 66–4.
[7] Harvey JF. Theory and Design of Modern Pressure Vessels. 2nd Edition. Van Nostrand Reinhold Co; 1974.
[8] Roark RJ. Stresses and Deflections in Thin Shells and Curved Plates Due to Concentrated and Variously Distributed Loading. Technical Note 806, National Advisory Committee on Aeronautics; 1941.
[9] Tentative Structural Design Basis for Reactor Pressure Vessels and Directly Associated Components. PB 151987, United States Dept. of Commerce; December 1958. pp. 62–81.
[10] Dodge WG. Secondary Stress Indices for Integral Structural Attachments to Straight Pipe. Welding Research Council Bulletin No. 198; September 1974.
[11] Wichman KR, Hopper AG, Mershon JL. Local Stresses in Spherical and Cylindrical Shells Due to External Loadings. Welding Research Council Bulletin No. 107; April 1972.
[12] Bijlaard PP. Computation of the Stresses from Local Loads in Spherical Pressure Vessels or Pressure Vessel Heads. Welding Research Council Bulletin No. 34; 1957.
[13] BS 5500: Specification for Unfired Fusion Welded Pressure Vessels. London: British Standards Institute; 1985.

8
High Pressure Vessels

1.0. General

1.1. Introduction

Designers and engineers of pressure vessels must eventually become acquainted with the design of vessels for high pressure service. The immediate question is, "What constitutes high pressure"? For purposes of discussion here, high pressure begins around 3,000 PSI. This is an arbitrary value as there are no codes or standards that define where high pressure begins. Perhaps a better way to decide if a pressure vessel is high pressure is based on the R/t ratio. Ratios of R/t less than 10 should be considered as high pressure.

There are pressure apparatus that produce pressures up to 1.5 million PSI. Apparatus that produce artificial diamonds and gems operate in this pressure range. They are more technically "presses" and not pressure vessels, per se. Pressures in the range of 20,000 to 150,000 PSI are used for food sterilization. Isostatic pressing operates in the range from 15,000 to 100,000 PSI. Autoclave reactors for hydrothermal crystal growth or production of artificial quartz are used with pressures in the 15,000 to 25,000 PSI range. Polyethylene processes operate in the 20-35,000 PSI range.

So there are a number of industrial processes that use high pressure. This chapter attempts to give the designer and engineer some tools to develop a basic design in the 3,000 to 25,000 PSI range. Vessels that were built in this range would include;

1. Low Density Polyethylene (LDPE) Reactors (Autoclaves)
2. Isostatic Presses (Hot or Cold)
3. Urea Reactors (Autoclaves)
4. Polyethylene Separators
5. Ammonia Converters
6. Sterilization Vessels
7. Hydrothermal Synthesis
8. Autoclaves
9. High Pressure Extraction
10. Water Jet Cutting

High pressure vessels can be built to ASME Section VIII, Divisions 1, 2 or 3. People tend to discount Division 1 as a design code for high pressure vessels but this should not be the case, although Divisions 2 and 3 will produce a more economical design. Remember that Division 2 was not in existence before 1968 and Division 3 before 1997. Many thousands of high pressure vessels were built before these alternative divisions were issued.

High pressure vessels should be designed to either Division 2 or Division 3, Section VIII of the ASME Code. This chapter evaluates the criteria of each division and its impact on the wall thickness of the vessel shell. There are two basic criteria for each division. They are Design by Rules and Design by Analysis. It should be noted that only Division 3 allows credit for pre-stress (autofrettage) for layered construction.

The treatment and discussion of vessels for design of high pressure equipment may appear to be out of the ordinary. This is an illustrated approach, rather than an analytical one. It is a design approach, not an analytical one. This chapter does not focus on higher, more sophisticated, analytical methods such as elastic-plastic or limit analysis. Certainly these methods should be utilized to fine tune the design to reduce wall thickness and weight. The focus of this chapter is to enable the user to develop a design which will work. It may not be the most economical or efficient design, but it will work.

Often in our industry we are asked to come up with designs which can be priced in the marketplace to determine if a particular process is economically feasible. Alternatively, we are asked to determine the ultimate cost so that one process can be compared with another. Without making a research project out of this effort it is possible with the use of material presented here to come up with a design that will suit this work process.

In other words, the design developed by the procedures illustrated here, may not be the most economical design. They will provide a starting place for the "development" of the final design.

Designs with gasketed closures will not reduce the thickness of components. There is a certain force required to seat the gasket and resist the internal pressure. This does not vary with the analytical technique used. In fact, the "brute force" approach is the valid one. On the flip side, the shell design can be greatly reduced by these advanced tools.

For low and medium pressure vessels, the most economical L/D ratio is between 3 and 5. As pressures tend to increase, this ratio goes up to 10, 50 or even 100. This is because the economics will drive the designer to smaller diameters and greater length to achieve the same volume.

For this same reason, most high pressure vessels are built with an end closure to provide access to the inside of the vessel. It is economically unsound, and in many cases impossible to provide manways. In fact, due to the thick shells required, nozzles through the shell wall are discouraged. In high pressure vessels most nozzles go through the end closures, not through the shell.

This is why most high pressure vessels do not use typical pressure vessel heads such as hemispherical or semi ellipsoidal. These heads are impractical in high pressure applications. So the economics and mechanical limits of the materials will determine the ultimate shape of the vessel.

Thus, the focus of this chapter is on end closures and not on design of shells and flat heads. As shown here these are relatively simple calculations.

Diameters are deliberately kept small to keep the wall thickness and metal weight down. However, as pressures and diameters increase, and new industrial processes are developed, it will push open the envelope of our existing technologies. Yet, there is a physical limit to what steels can do. The highest strength for any material in ASME VIII, Division 3, is a forging material, SA-723. However this material cannot be used for welded construction, only machined applications.

1.2. ASME VIII, Division 3

In the mid 1970s it was realized that the current ASME Codes for pressure vessels were not adequate for vessel operating above about 20,000 PSI. ASME Section VIII, Division 2 did not have any allowance for vessels manufactured by the autofrettage technique. In November, 1979 the ASME board on Pressure Technology Codes and Standards established a special working group on high pressure vessels. The group was chartered to come up with a Code that would be applicable for vessels operating from 10,000 to 200,000 PSI. However no upper limit has been established.

This was quite challenging because of the diversity of the equipment required for this pressure range. For example, vessels in the lower end of the range typically are manufactured by welding. However equipment manufactured for the higher end of the range are typically manufactured by machining.

A fracture mechanics analysis is required for all vessels unless a leak before burst mode of failure can be assured. A fatigue analysis is required for all vessels. The determination of the number of cycles allowed by Division 3 is conservative but also allows for the recertification of equipment that has reached its design life.

Division 3 deliberately restricts the materials that are acceptable for use. Only 242 different types and grades of materials have been authorized. Allowable stresses have been based on yield strength. Toughness properties were considered critical for these applications and have required that the materials meet the required CIT values in the weak axis (transverse direction), perpendicular to the direction of rolling, wherever possible. The philosophy for setting high toughness criteria was based on the need for higher strength materials used in this equipment.

There are no tables of allowable stress in Section 2, Part D as there are for other sections of the Code. This is because thicknesses are based on ultimate tensile and yield strength. Properties for those materials acceptable in Division 3 are contained within the code itself in Article KM-4, “Material Design Data”.

Division 3 provides special rules for layered and autofrettage vessels. Rules for wire wound vessels and interlocking strip wound vessels has also been included.

For designs that cannot prove leak-before-burst criteria, the vessel will be subject to fracture mechanics evaluation. Welded construction is permitted for all materials where ductility can be maintained. One material that has found favor for the highest pressure applications is SA-723 forgings. This material has incredible strength but welding is not allowed.

An inner core or liner used for corrosion resistance may or may not be included as part of the strength calculations. Weld overlay or cladding, specifically for corrosion resistance cannot be included in strength calculations.

All nozzles must be integrally reinforced. Reinforcing pads, fillet welds and partial penetration welds are not allowed. Threaded connections must be of the “straight” thread variety. Tapered threads, such as ordinary pipe threads are not allowed.

NDE requirements for Division 3 vessels and components was based on Section III, Class 1 rather than Division 2. UT is the preferred method of examination rather than RT for welded joints. All butt welds shall be 100% UT examined. In addition all plates and forgings must be 100% UT examined, not just those over 4 inches as in Division 2. 100% PT or MT is required for all internal and external surfaces of pressure boundary components.

A hydrotest is required for all vessels and is 1.25 times the design pressure times a stress material ratio, SMR. Unlike other divisions, Division 3 sets an upper limit on the hydrotest pressure.

Documentation and stamping follow ASME convention with regard to nameplate stamping, data reports and ASME third party inspection and witnessing of various hold points.

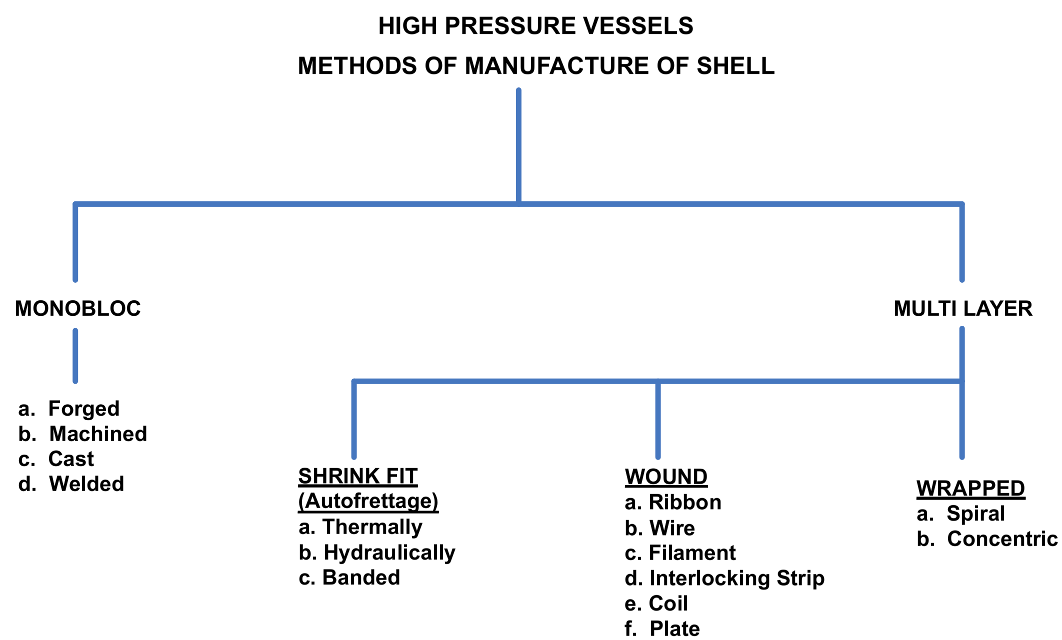
1.3. Types of Construction for High Pressure Vessels

There are two major categories for the manufacture of high pressure vessels as follows;

1. Monobloc
2. Layered
   a. Wound
      i. Wire or filament would
      ii. Interlocking strip wound
      iii. Flat steel ribbon wound
      iv. Plate wound
      v. Coil wound
   b. Autofrettage
      i. Hydraulically expanded type
      ii. Shrink fit
   c. Wrapped
      i. Spiral
      ii. Concentric

A detailed description of the various manufacturing techniques would be as follows;

1. Monobloc: Solid vessel wall by forging or rolled plate
2. Multilayer: Begins with a core of about ½ inches thick and successive layers are applied. Each layer is vented (except the core) and welded individually with no overlapping welds.
3. Multiwall: Begins with a core of 1½ inches to 2 inches thick. Outer layers of the same thickness are successively shrunk fit over the core. This creates compressive stress in the core, which is relaxed during pressurization. The process of compressing layers is called "autofrettage" from the French word meaning, self-hooping.
4. Multilayer autofrettage: Begins with a core about ½ inch thick. Bands of forged or welded rings are slipped over the core, and then the core is expanded hydraulically. The core is stressed into the plastic range but below ultimate strength. The outer rings are maintained at a margin below yield strength. The elastic deformation residual in the outer bands induces compressive stress in the core, which is relaxed during pressurization.
5. Wire Wrapped Vessels: Begin with an inner core of thickness, less than required for internal pressure. Core is wrapped with steel cables in tension until the desired thickness is reached.
6. Coil Wrapped Vessels: Begin with a core that is subsequently wrapped with thin steel sheet until the desired thickness is reached. Only two longitudinal welds are used, one attaching the sheet to the core. The other the final closure weld. Vessels 5 to 6 feet in diameter and pressure up to 5,000 psi have been made in this manner.


HIGH PRESSURE VESSELS
METHODS OF MANUFACTURE OF SHELL
MONOBLOC
a. Forged
b. Machined
c. Cast
d. Welded
MULTI LAYER
SHRINK FIT
(Autofrettage)
a. Thermally
b. Hydraulically
c. Banded
WOUND
a. Ribbon
b. Wire
c. Filament
d. Interlocking Strip
e. Coil
f. Plate
WRAPPED
a. Spiral
b. Concentric

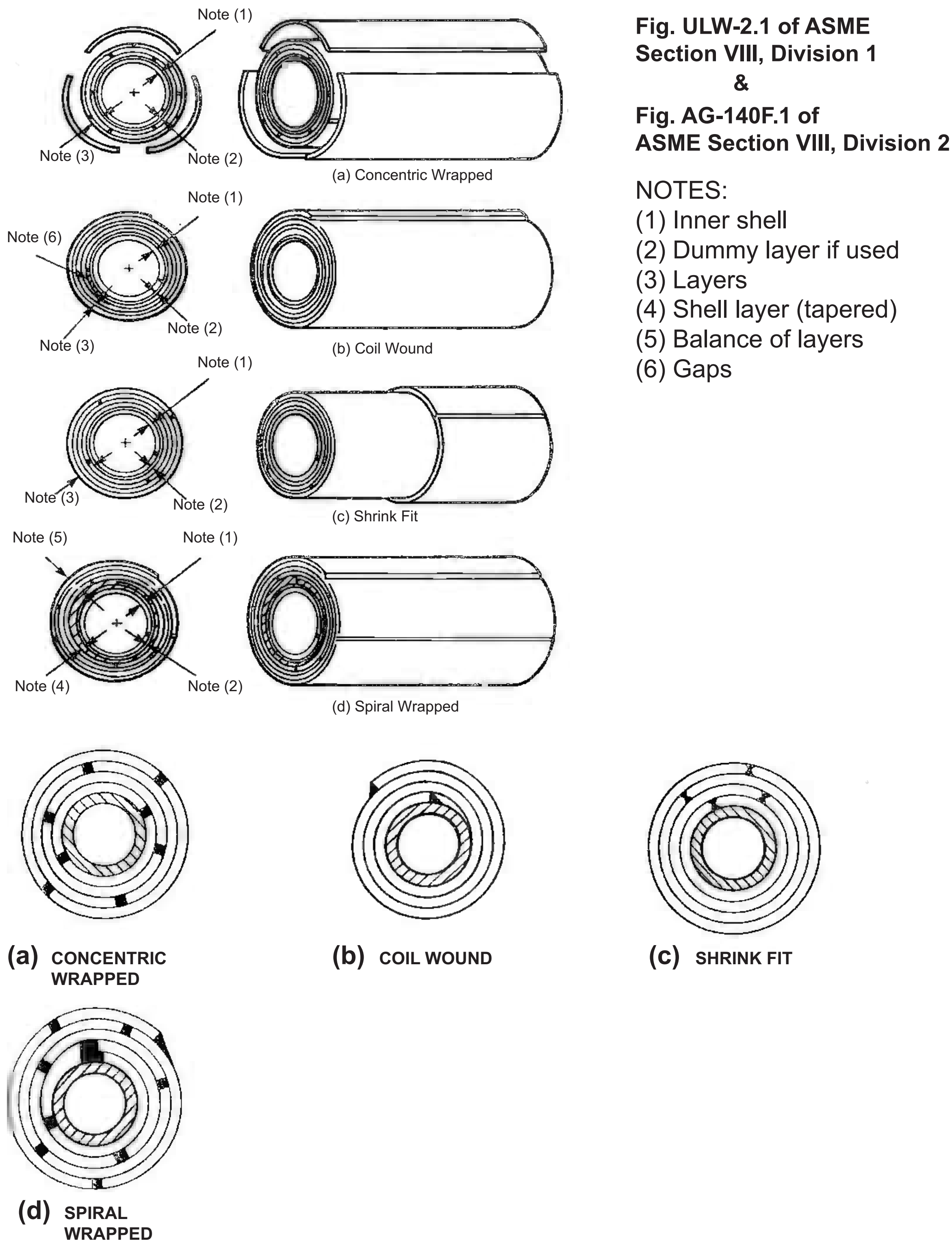


Figure 8-1. Types of layered construction.

1.4. Details of the Ribbon Wound Technique

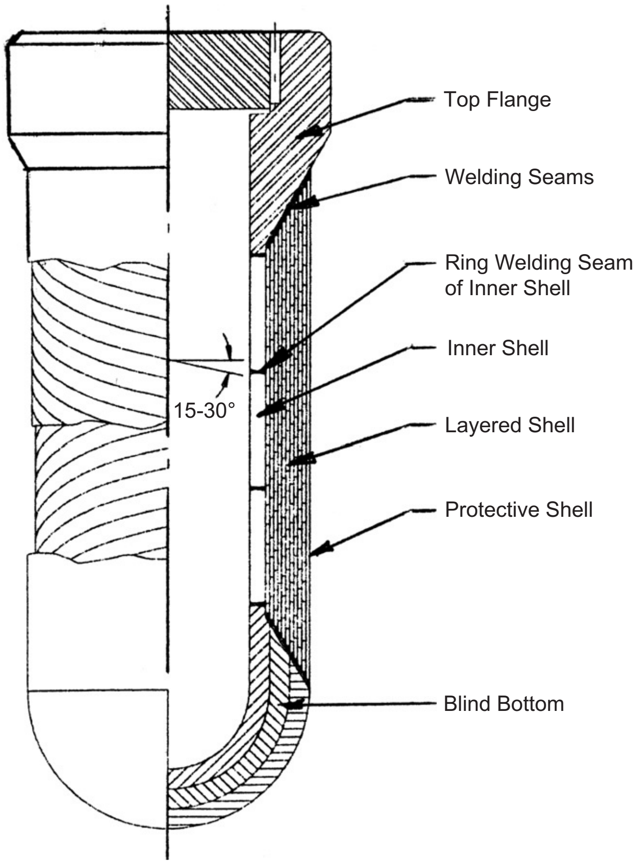


Figure 8-2. Flat steel ribbon wound pressure vessel.

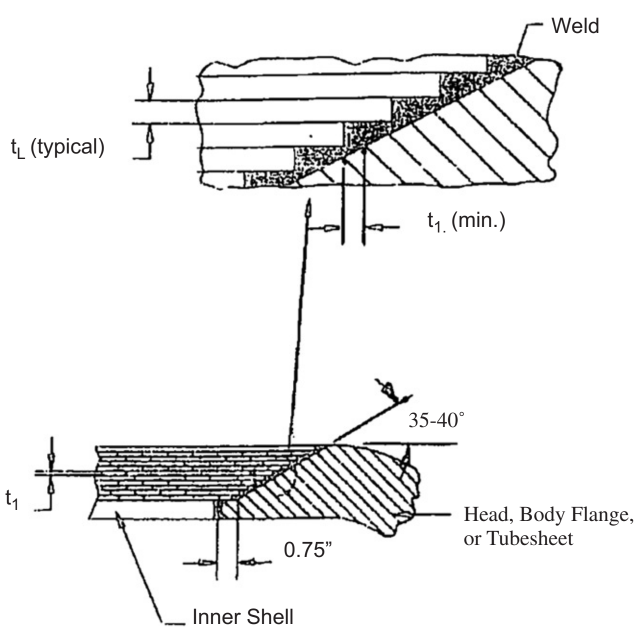


Figure 8-3. Typical head, body flange, or tubesheet attached to layered section.

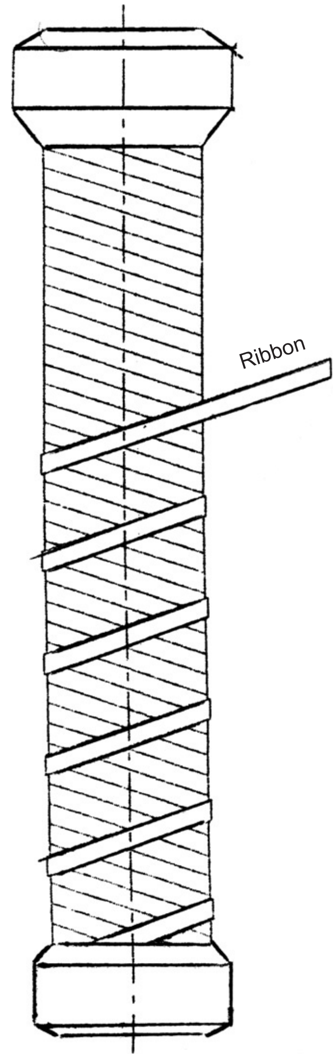


Figure 8-4. Schematic of winding process for the ribbon wound vessel.

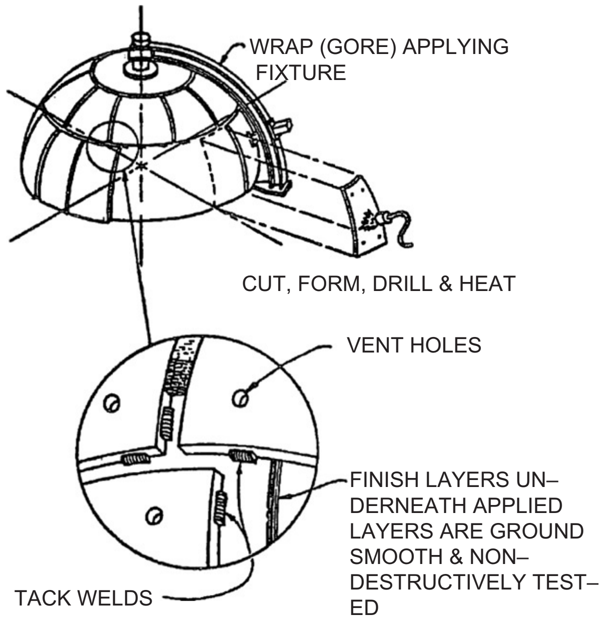


Figure 8-5. Layered hemispherical head fabrication procedure.

Multi-Layer Pressure Vessels. Multi-layer vessels have been known by various names over the years. The most common names are;

1. Multiwall Vessels
2. Banded Vessels
3. Layered Vessels

A multi-layer vessel is a vessel in which the cylindrical portion is made up of two or more contacting bands or layers. The inner shell is the innermost band of a multi-layer vessel and is made in the same manner as any solid wall vessel. The inner shell can be of any suitable material to resist corrosion by the contents. This is one of the unique advantages of a multi-layer vessel. Subsequent layers are added on top of this inner layer by a variety of techniques to achieve the ultimate wall thickness required.

History

The concept for multi-layer construction was first developed in Germany around 1890. However it was not until the 1930s that the concept became practical with the advent of electric arc welding. In the 1930s the A.O. Smith Corporation pioneered the concept of concentric wrapped thin layers and eventually produced more than 10,000 multi-layered pressure vessels. The license was eventually sold to CB&I who produced multi-layer vessels under the patented Multilayer trademark until 1976, when the patents and machinery were subsequently sold to Nooter Corp. Although Nooter is now out of the fabrication business in the U.S.A., multi-layer vessels are still produced in Austria by a Nooter subsidiary, Schoeller-Bleckmann. Nooter marketed the multi-layer vessels under the trade name, Plywall.

In 1962 the autofrettage technique of shrink fitting successive layers was patented by Struthers Wells Corp. Over 800 such vessels were manufactured by the 1980s. Manufacture of these vessels was then transferred via license to Larson & Toubro, India, who has in turn produced more than 50 multi-layered, autofrettage vessels.

A Chinese manufacturing firm developed a ribbon wound technique for multi-layered vessel construction in the 1960s and has subsequently produced over 7,000 ribbon wound vessels. Kobe Steel in Japan, was originally a multi-layer vessel manufacturer and produced approximately 1,000 units of the concentrically wrapped types. They currently do not produce multi-layer vessels any longer but still engage in solid wall, monobloc construction.

In total, it is entirely probable that over 20,000 multi-layer vessels have been fabricated over the past 80 years, many of which are still in operation.

General

Multi-layer vessels have been designed and built within the following ranges;

1. Diameters from 7 inches to 168 inches
2. Pressures from 1000 to 35,000 psi
3. Wall thickness' from 1½ inches to 18 inches
4. Temperatures from (-) 50 to 850°F
5. Lengths up to 100 feet
6. Weights up to 800 tons

In general, the thicker the vessel, and the longer the vessel, the more attractive the multi-layer option becomes. It is usually more economical to design a multi-layer vessel with a large L/D ratio for a given volume. The selection of the multi-layer option is usually determined by economics. However, once the practical manufacturing limits of solid wall construction are exceeded, multi-layer may be the only option.

Multi-layer pressure vessels can be manufactured by a variety of means. The most common categories or types of manufacture are;

1. Wrapped
2. Wound
3. Shrink Fit (autofrettage)

The most common applications;

a. Ammonia Converters
b. Methanol Converters
c. Ethanol Reactors
d. Urea Reactors
e. Urea Autoclaves
f. Storage Vessels for Helium, Nitrogen and Hydrogen
g. Hydrocracking Reactors
h. Polyethylene Reactors
i. High Pressure Separators
j. High Pressure Heat Exchangers
k. Hydraulic Cylinders
l. Accumulators
m. Air Receivers

ASME Code

It was not until January 1979 that layered vessels were included in the ASME Code, almost 50 years after they were first introduced. Also in 1979 the ASME Policy Board, Codes and Standards, approved the establishment of a special working group for high pressure vessels under Section VIII. This effort would ultimately culminate in the issuance of Division 3 of Section VIII in 1998.

Multi-layer vessels can be built to any of the three sections of ASME Section VIII, Divisions 1, 2 or 3. However, only Division 3 allows the designer to take credit for the residual compressive stresses induced by the autofrettage technique. This is a big advantage in the design of the shell. The appropriate ASME Code Sections that apply to multi layer vessels are as follows;

1. Division 1: Part UWL
2. Division 2: Part 4.13
3. Division 3: KD-5 & KD-8

Code Cases 2229 and 2269 were issued in 1996 and 1997 respectively to allow for ribbon wound pressure vessels.

Special Features/Advantages

1. Layered vessels are constructed by successively wrapping thin layers around a center core until the desired wall thickness is achieved. One advantage of this process over monobloc construction, is that the layers each have uniform chemical and mechanical properties. Optimum properties cannot always be achieved across thick sections or maintained during the whole fabrication sequence. With the multi-layer concept, no matter how thick the shell, it does not suffer from lack of material uniformity.
2. For corrosive environments, the inner core cylinder may be made of whatever alloy is necessary to provide corrosion resistance.
3. The use of high tensile material allows for reduced shell thickness.
4. High residual stresses from welding cannot develop across the multiple thin layers in multi-layered vessel shells. Any through thickness welds such as girth seams and nozzle welds do not develop high residual stresses because there is little "shear transfer" between the layers. As such, PWHT is seldom used and not desirable.
5. Vent holes are used in all the layers except the inner core layer. In the event of leakage in the inner cylinder, the leak may be detected without fear of failure of the vessel as a whole.
6. In the case of monobloc construction, the shell thicknesses are limited to ingot sizes, plate roll capacity or forging press capabilities. Layered construction does not have such limitations other than overall weight.

7. Butt welds in the inner core cylinder are radiographed in accordance with the Code. The balance of seams attaching each layer are magnetic particle tested.
8. The longitudinal welds are staggered.
9. Multi-layer vessels can be used in hot hydrogen service for hydrotreating and hydrocracking applications.
10. Multi-layer vessels can be field fabricated. Although this would be unusual, it is possible and has been done before.
11. Multi-layer vessels can be repaired in the field.
12. Multi-layer vessels can be rerated.
13. Nozzles can be welded through the shells of a multi-layer vessel but this is frequently avoided due to the extreme thickness. It is more common for all nozzles to be placed in the heads. In any case nozzles through multi-layered vessels should be minimized.

Disadvantages

1. Through thickness welds (welds that penetrate the multiple layers such as head attachment, circumferential welds between shell courses, or nozzle welds) cannot be easily examined by either UT or radiography.
2. Through thickness welds have different heat conduction properties than the layers to which they are attached. Therefore thermal stresses may create discontinuity stresses at locations already susceptible to higher stress levels.
3. Through thickness welds are not vented like the adjacent layers and therefore may be susceptible to hydrogen embrittlement.

Autofrettage. In thick walled cylindrical vessels, the resistance to yielding, fatigue crack initiation and propagation, and to fracture can be significantly increased by introducing residual compressive tangential stresses into the vessel wall near the inside surface. This effect occurs because the stress distribution in a thick walled cylinder by internal pressure consists of very high stress intensity near the inside surface. This stress intensity decreases rapidly with increasing radius and near the outside radius, the stress intensity is relatively small. This means that much of the material of the wall is not contributing significantly to the strength of the vessel.

Autofrettage is a method for increasing a cylindrical shells ability to withstand pressure by introducing residual compressive stress into the inner portion of the cylinder. Autofrettage is a French word that literally means, "self hooping". It is a method of prestressing the inner radius of a thick walled cylinder. The amount of prestressing causes plastic yielding of the inner membranes of the vessel wall and leaves this area in compression. During pressurization, the residual compression of the bore is initially relaxed before tension stress is achieved. This allows the cylinder to be made thinner since a significant portion of the compressive prestress must be overcome before tension even begins.

Yielding of the inner surface begins when the maximum shear stress is equal to the yield point shear stress. As the pressure, or force, is increased the plastic deformation penetrates farther into the vessel wall until it reaches the outer surface. At that point the entire shell has yielded. If the internal pressure is removed after the cylinder is in plastic state, a residual stress will remain in the wall. This residual stress allows the cylinder to contain more pressure then would be possible to without it.

Various manufacturing techniques have been developed to produce the residual stresses in the bore of thick walled cylinders. These processes are as follows;

1. Autofrettage (bore expansion technique)
   a. Hydraulic swage with tapered plug
   b. Mechanical pressure (swaging)
   c. Pressurized cold working
   d. Explosion technique
   e. Thermal gradient
2. Multiple Wall
   a. Welded layer – weld shrinkage
   b. Shrink fit
3. Wire or ribbon winding

The most common technique used in the manufacture of pressure vessels is the shrink fitting technique. In this process layers are successively shrunk, over an inner core, one cylinder over another. The outer layer is heated which causes it to expand. The inner layer is cooled which causes it to contract. Once the outer cylinder is slipped over the inner cylinder it contracts as it cools, putting the inner wall in compression. Successive layers can be applied which amplify the effect.

Shrink rings are used around other components to achieve the same effect. For example, studded flanges in high pressure applications, frequently have shrink rings placed in the area of high stress around the circumference of the flange to reduce stresses and wall thickness.

Another common practice is to hydraulically expand the inner cylinder beyond the plastic range. This was the first application of autofrettage used with cannon barrels. In this case the inner cylinder or core is expanded beyond the yield range. When the pressure is released the inner core goes into compression. There are several positive effects that result from this process. First, the work hardening that results from the material stretch contributes to the strength of the part. Secondly, cylinders expanded in this fashion are more resistant to fatigue failures then those that are not.

The third method utilized for creating a compressive prestress in the bore, is by wire or ribbon winding of the bore cylinder under tension. Although this is effective, there are limited suppliers who can actually manufacture equipment in this fashion.

Therefore the designer must allow for the various techniques for accomplishing the compressive prestress while allowing the flexibility of the supplier to apply their specialized technology.

History

The history of the autofrettage technique was developed for use in cannon barrels. The first autofrettaged gun barrel was produced in France in 1913. The process quickly was adopted by the U.S.A. and U.K. as well. During WWII, the availability of high strength steels was critical due to shortages of alloying elements and industrial capacity. With autofrettage, low strength steel could be utilized for gun barrels with better efficiency. Expansion of the barrels was accomplished by tapering the barrels and then hydraulically expanding the cylinders into carefully machined outer staves or dies. The expansion of the bore led to the work hardening benefits as well as the residual stresses from the autofrettage. This was known as the "container method".

After WWII, this method was dropped due to availability of alloy materials and the difficulty of expanding alloy barrels. The hydraulic intensifiers worked at an amazing 150,000 PSI. It was noted however that the autofrettage technique had a significant effect on the barrels ability to withstand fatigue. An alternative process was developed that swaged the bore by forcing a mandrel through the bore. The process became known as "hydraulic swage autofrettage" and is still in use today.

Autofrettage Pressure

Autofrettage pressure is the pressure that causes initial overstrain of the bore to occur. It is that pressure that results in plastic deformation, beginning at the inside surface of the cylinder. This is "initial autofrettage pressure" or "minimum autofrettage pressure". The "maximum autofrettage pressure" is the pressure resulting in plastic yielding across the entire wall thickness.

Overstrain Ratio

In order to calculate the residual stress produced by the autofrettage process, a measure of the extent of autofrettage must be known. The measure most frequently used is the "overstrain ratio". This is the ratio of the thickness of the plastically deformed portion of the shell to the total thickness.

Bauschinger Effect

The pressure at which the material of a previously autofrettaged cylinder first undergoes additional plastic deformation is called the re-yield pressure. Theoretically this pressure should be the same as the autofrettage pressure. However studies have shown that the re-yield pressure is actually less than the autofrettage pressure. This is because the unloading from the autofrettage pressure, does not follow linear elastic behavior. The reason for this phenomena has been termed the "Bauschinger Effect". The Bauschinger Effect is defined as a decrease in the yield strength of the material in compression as a result of prior deformation in tension. A Bauschinger Effect Factor (BEF) has been established to account for this condition.

When the yield strength is affected by over-strain, the BEF approaches 0. If the effect is very little, then the BEF approaches 1. High strength steels exhibit a significant Bauschinger Effect. Thus the material near the inner bore does not behave elastically during unloading, but actually reverse yields in compression., which in turn greatly reduces the tangential stress near the surface on the outside of the vessel.

HIGH PRESSURE VESSELS					
DESIGN OPTIONS CHECKLIST					
TYPE OF END CLOSURE	REMOVABLE	NON-REMOVABLE			
TYPE OF REMOVABLE END CLOSURE	THREADED	FLANGED	OTHER		
TYPE OF GASKET	DELTA RING	BRIDGMAN	DOUBLE CONE	LENS RING	OTHER
MATERIAL OF CONSTRUCTION					
TYPE OF SHELL CONSTRUCTION	MONOBLOC	LAYERED	BANDED	WOUND	AUTOFRETTAGE
MATERIAL OF GASKET					
SHRINK SLEEVES	YES	NO			
SLEEVE NUTS	YES	NO			
STUD TIGHTENING METHOD	MECHANICAL	THERMAL	HYDRAULIC		
THREADED CLOSURES - TYPE OF THREADS	ACME	BUTTRESS	MODIFIED	OTHER	
JACK SCREWS	YES	NO			
NOZZLE ATTACHMENTS - Type/Style					
CORROSION					
ORIENTATION - Vertical or Horizontal					
TYPE OF SUPPORT					
TYPE OF OPERATION - Continuous or Batch					
PRESSURE					
TEMPERATURE					

HIGH PRESSURE VESSELS - Checklist of Testing & NDE Requirements						
	Description	Required	Before PWHT	After PWHT	After Hydro	Remarks/Notes
	NDE Requirements (Welds & Base Matl's)					
1	100% RT all Cat A & B butt welds					
2	100% RT all Cat C & D butt welds					
3	100% UT all Cat A & B butt welds					
4	100% UT all Cat C butt welds					
5	100% UT all Cat D joints (butt and groove welds)					
6	Spot RT all back clad areas					
7	MT all plate edges and weld preps					
8	MT/PT all surfaces prior to WOL					
9	PT all attachments welded directly to cladding or WOL					
10	PT/MT all machined surfaces of forgings 100%					
11	PT all WOL/Clad surfaces					
12	UT all WOL/Clad surfaces					
13	MT backgouging of root passes					
14	MT of welds prior to backweld of WOL					
15	MT of main seams					
16	UT/MT of weld buildup after machining					
17	Acoustic emission					
	Heat Treatment Requirements					
1	Plate or forgings for shell and heads Q&T					
2	Plate or forgings for shell and heads N&T					
3	PWHT					
4	Intermediate stress relieve (ISR)					
5	Dehydrogenation heat treatment (DHT)					
6	Step cooling					
7	Multiple HT cycles allowed for (maximum PWHT)					
	Fabrication/Design Requirements					
1	F' Type nozzles required > 3"					
2	All attachments full penetration					

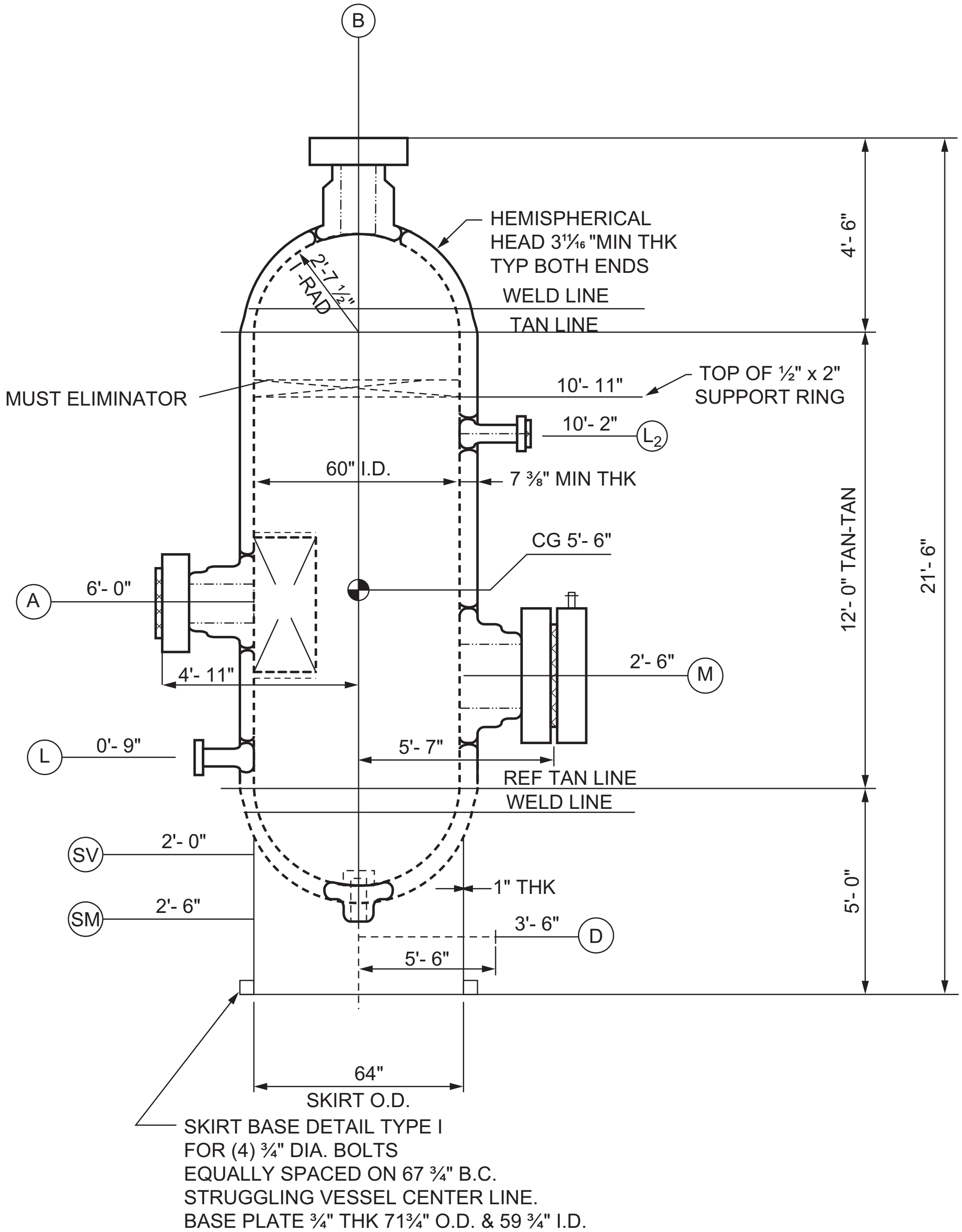
	Description	Required	Before PWHT	After PWHT	After Hydro	Remarks/Notes
	Base Material Requirements					
1	Hot tensile tests					
2	Autoclave testing - disbonding					
3	Shell/head materials to SA-20, paragraphs S1, S2, S5					
4	UT all plate and forgings > 4"					
5	X factor - weld filler metal					
6	J factor - plate, forgings, pipe and fittings					
7	Impact testing of base materials					
8	Production impact testing					
9	Copper sulfate examination - strip back of clad areas					
10	PMI					
11	Hardness					
12	Ferrite check - cladding and WOL					
13	Hot rolling of shell plates					
	Testing Requirements					
1	Hydrotest holding time					
2	Hydrotest water purity					
3	Hydrotest test temperature					
4	Chemical analysis of WOL					
5	Base line UT thickness readings					

Section 1.5: Data for Examples - High Pressure Vessels

Item	Description	Pressure, PSIG	Temp, °F	D, ID, in	Shell Thick. In	T-T OR OAL, Ft	L/D Ratio	R_m, in	R_m /t Ratio	Weight (KIPS)	Shell Type	Matl	Closure Gasket Type/Matl
1	Compressor Suction Drum	4960	250	60	7.375	12	2.4	33.68	4.6	105	Monobloc	CS	NA
2	Receiver	3800	350	22.83	2.283	11.22	5.9	12.56	5.5	11.2	Monobloc	CS	NA
3	Urea Reactor	4000	400	42	5.75	60	17.1	23.875	4.15	214	Multilayer	CS	Bridgman-Alum
4	Ammonia Separator	5800	90	30	6.25	8	3.2	18.125	2.9	42.2		CS	
5	Ammonia Converter	10,000	300	29.75	4.375	37.83	15.3	17.06	3.9			CS	Double Cone-Alum
6	H.P. Separators	10,000	300	29.5	4.375	7.27	3.1	16.94	3.87			CS	Double Cone - Annealed Copper
7	Liquid Phase Converter (1)	10,300	1,000	32	8.75	42.58	16	20.375	2.3	190	Monobloc	Cr-Mo	Delta Ring
8	Ammonia Converter	10,000	300	29.75	4.375	37.83	15.3	17.06	3.9	32.9		Cr-Mo	Double Cone -Alum

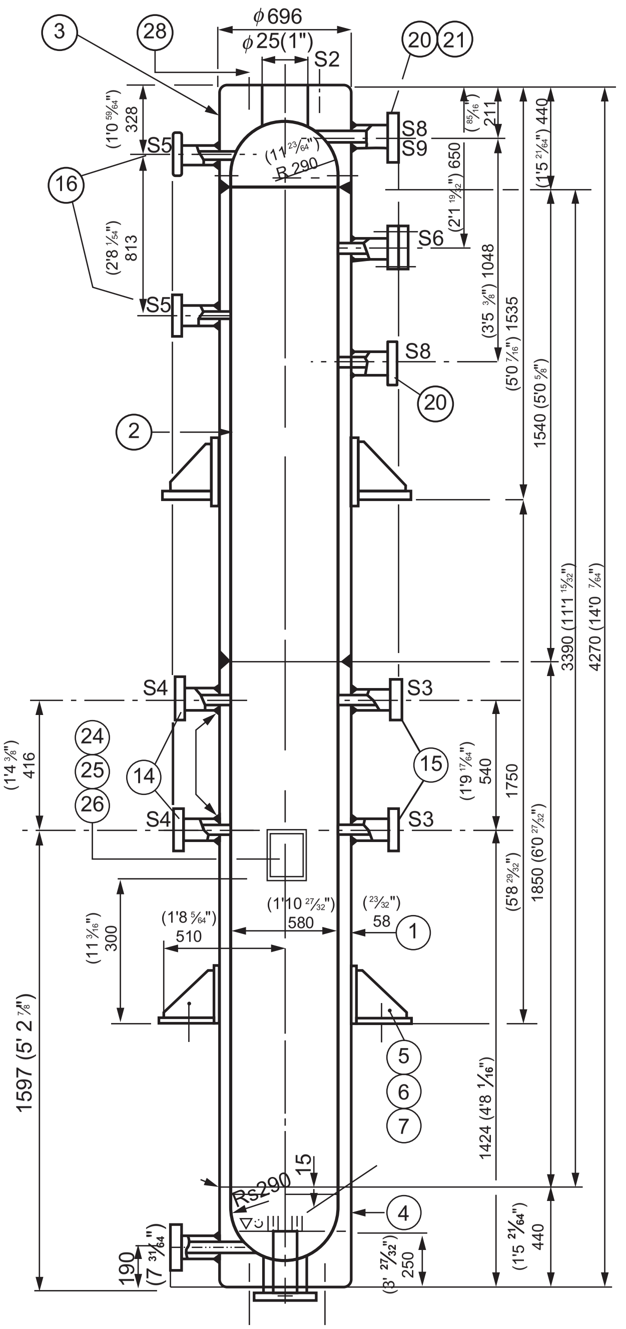
Notes:

1. Vessel is refractory lined. Weight includes refractory

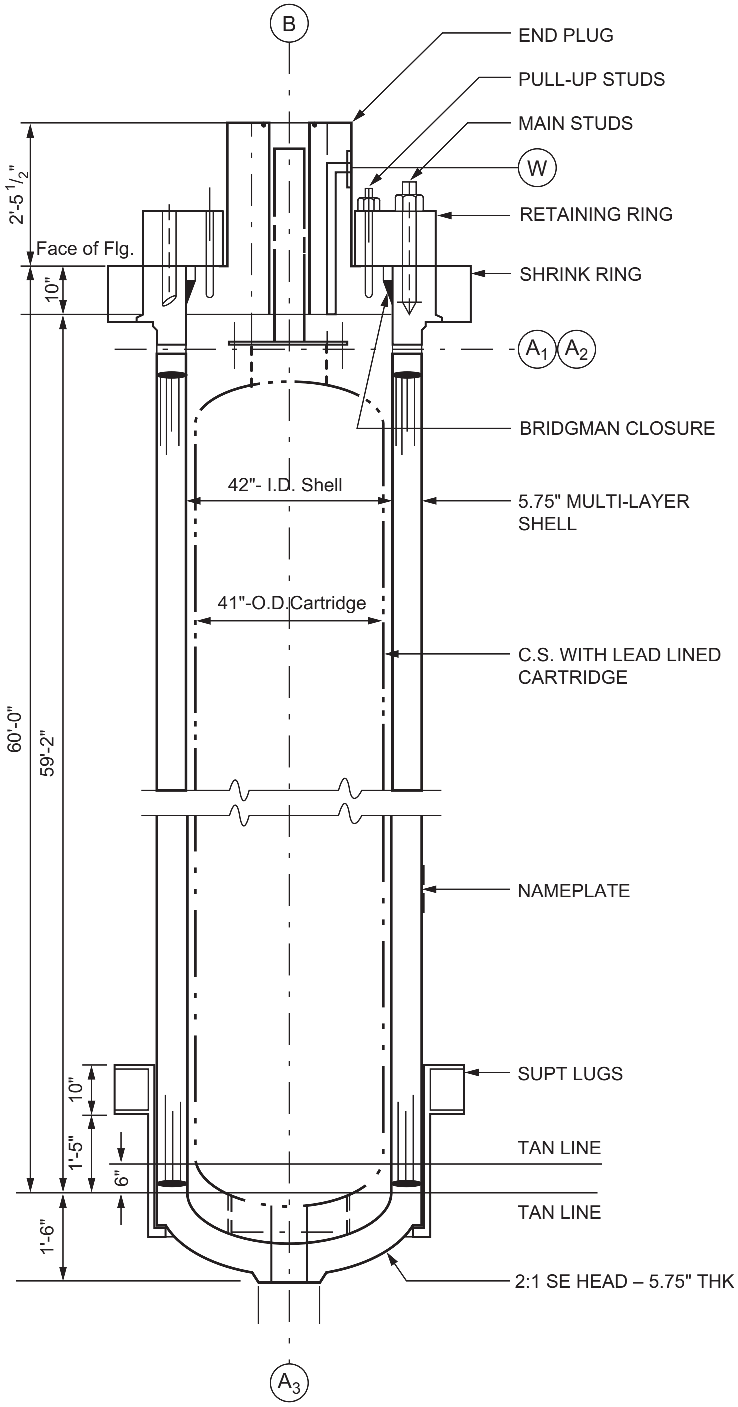


EXAMPLE No.1

Compressor Suction Drum
Matl : SA-105
ASME VIII-2
4960 PSIG @ 250°F

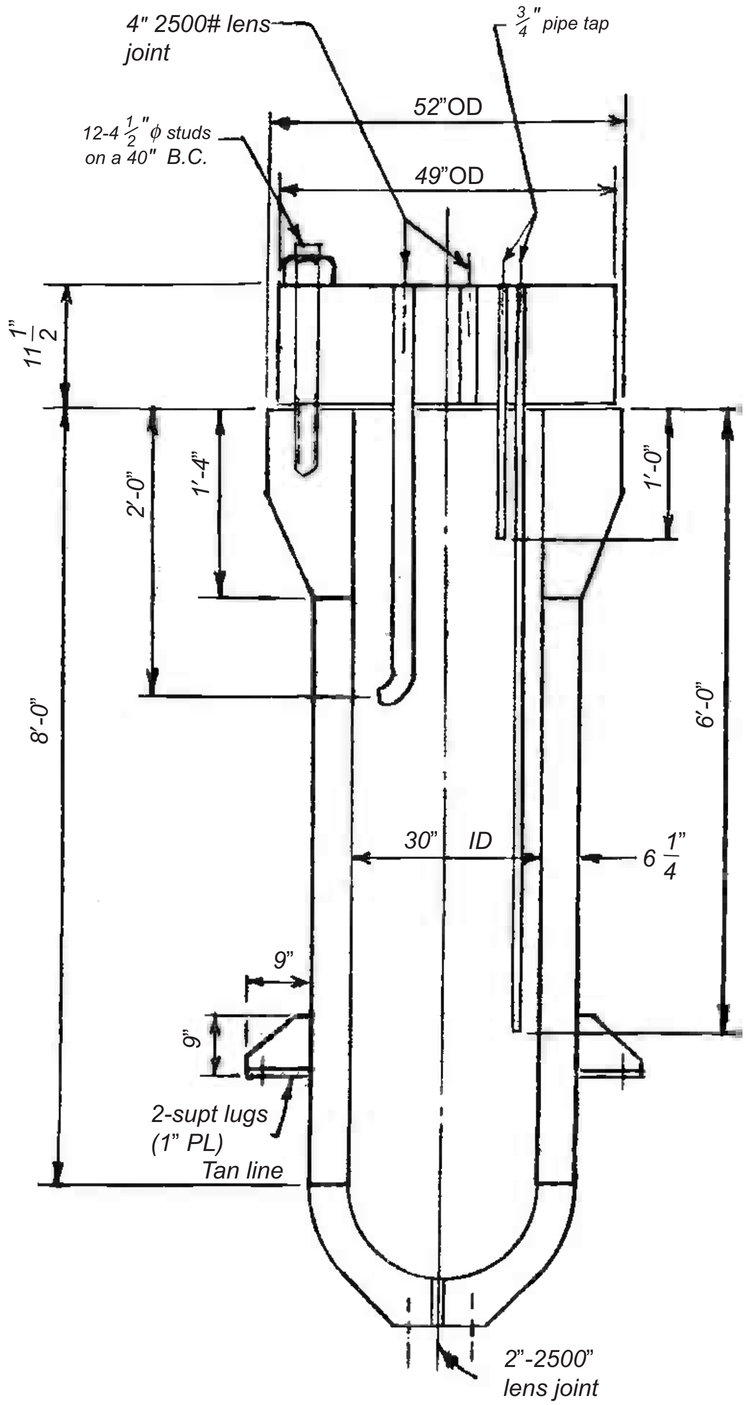


EXAMPLE NO. 2
Matl: SA-105
ASME VIII-1
3800 PSIG @ 350°F



EXAMPLE No. 3

Urea Reactor
Matl: SA-105
4,000 PSIG @ 400ºF

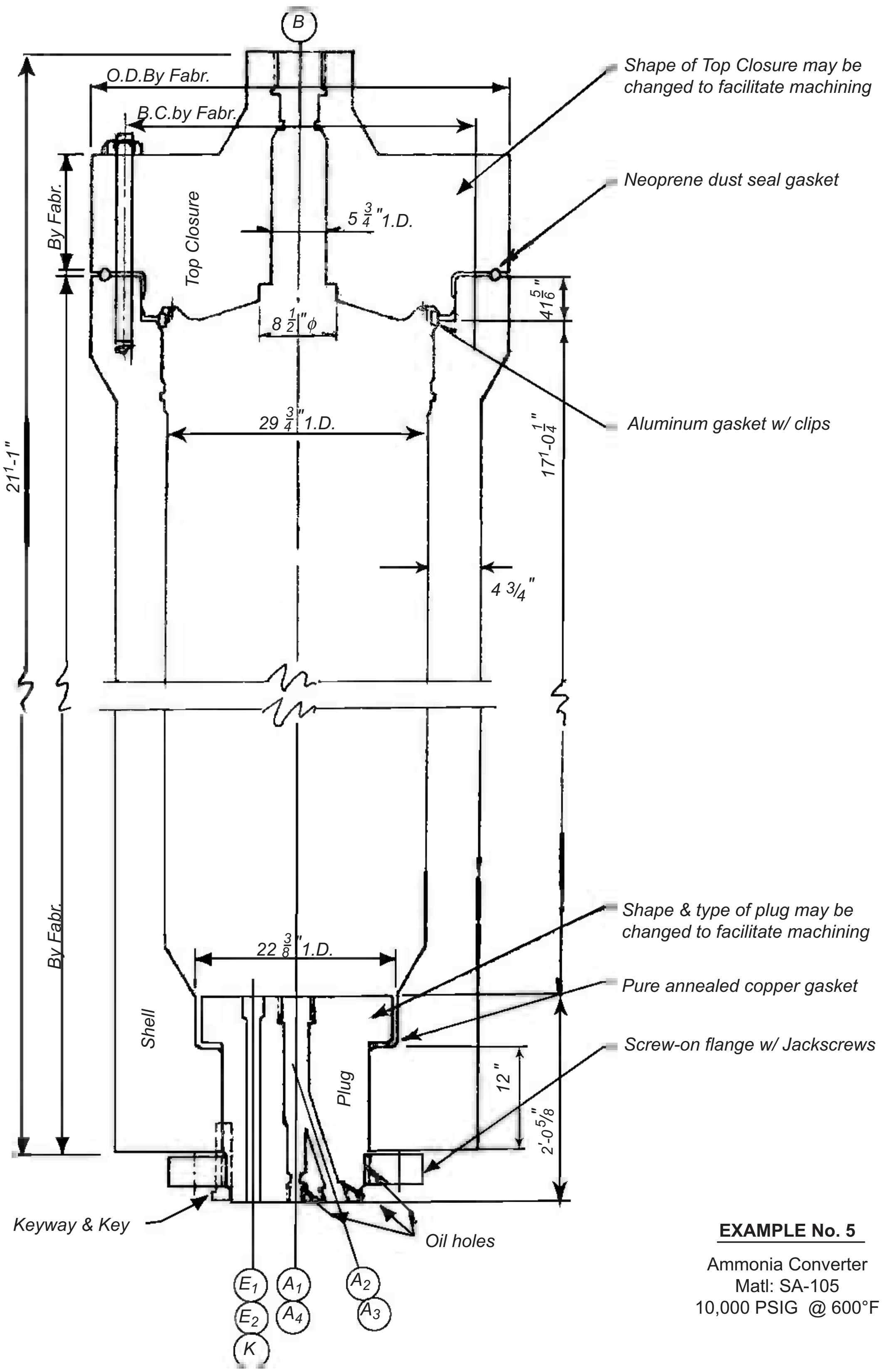


EXAMPLE No. 4

Ammonia Separator

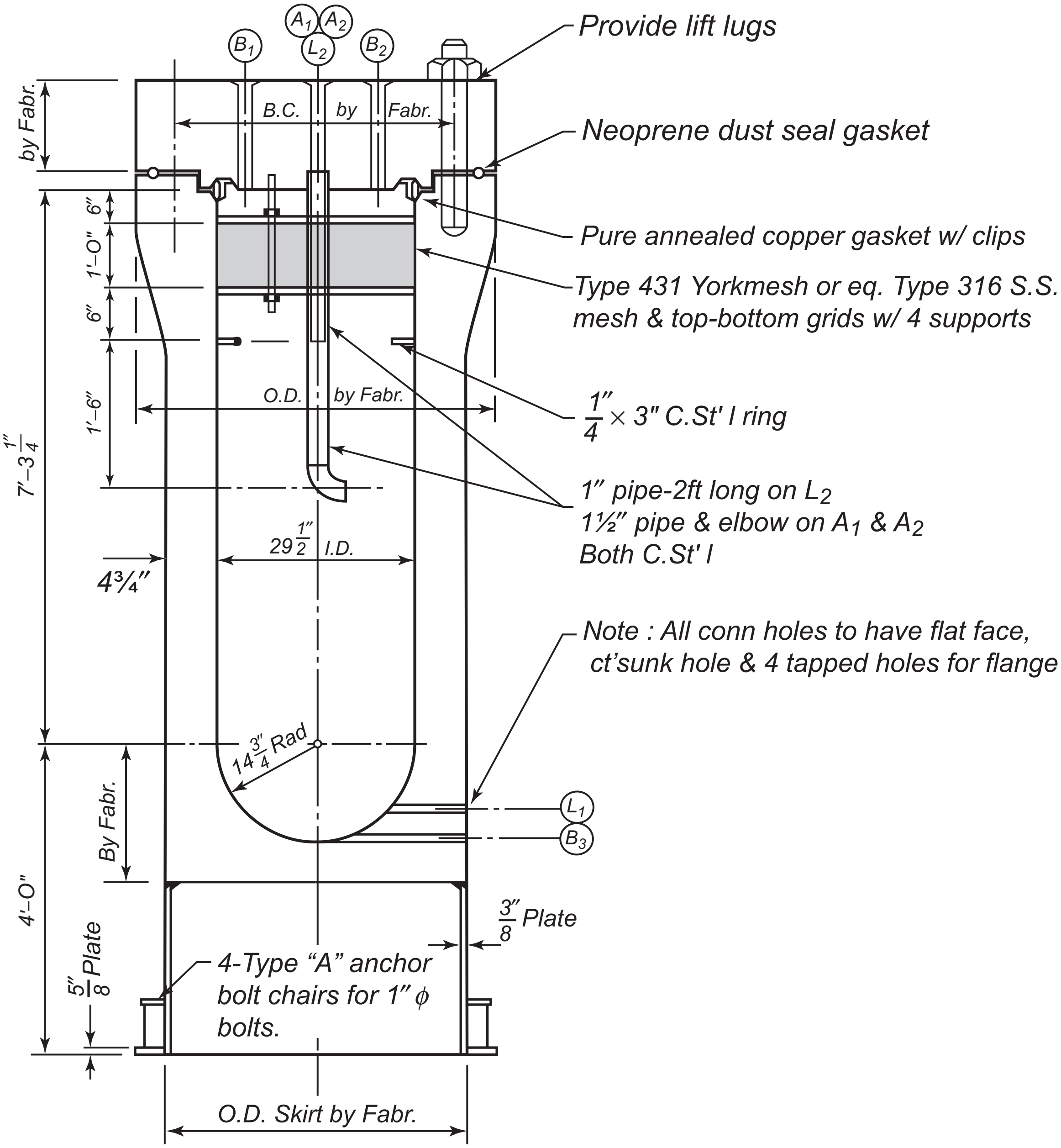
Matl: SA-105

5,800 PSIG @ 90°F



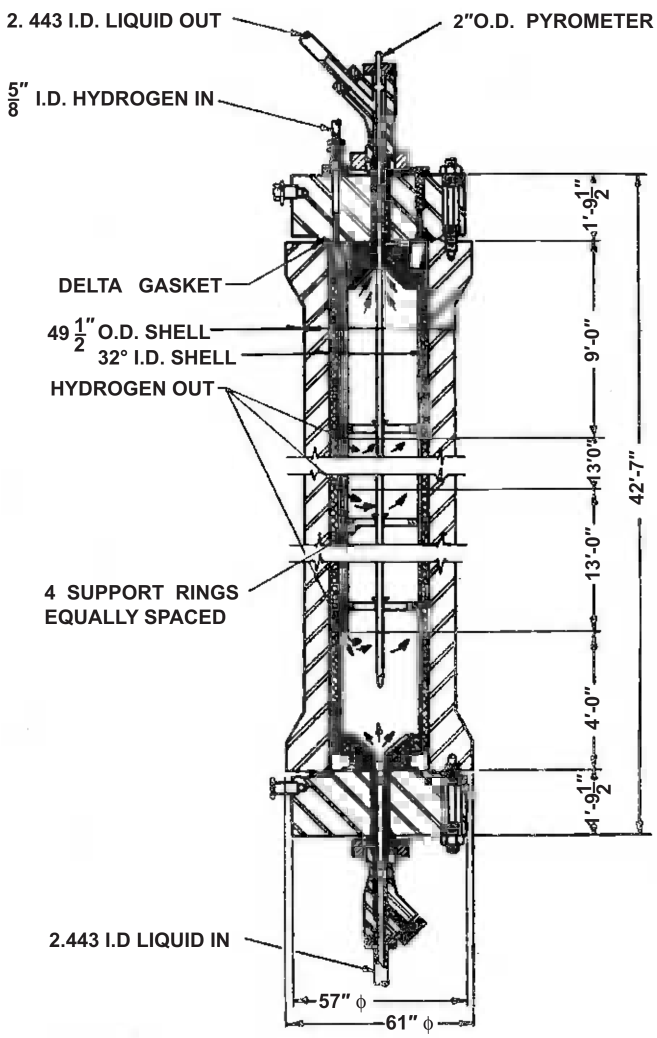
EXAMPLE No. 5

Ammonia Converter
Matl: SA-105
10,000 PSIG @ 600°F



EXAMPLE No. 6

High Pressure Separator
Matl: SA-105
10,000 PSIG @ 300°F



EXAMPLE No. 7

Liquid Phase Converter Assy
10,300 PSIG @ 1,000ºF

Shell material and properties

3% Cr
.65% Ni
.3% Mo

F_U = 100 KSI
F_Y = 55 KSI
S = 17.5 KSI

Shell Thk: 8.75″ solid wall (forging)

Head material

4-6% Cr
1-1.25% Ni
.4-.8% Mo

S= 17.5 KSI

Head Thk: 21.50″ Thk

Head Studs

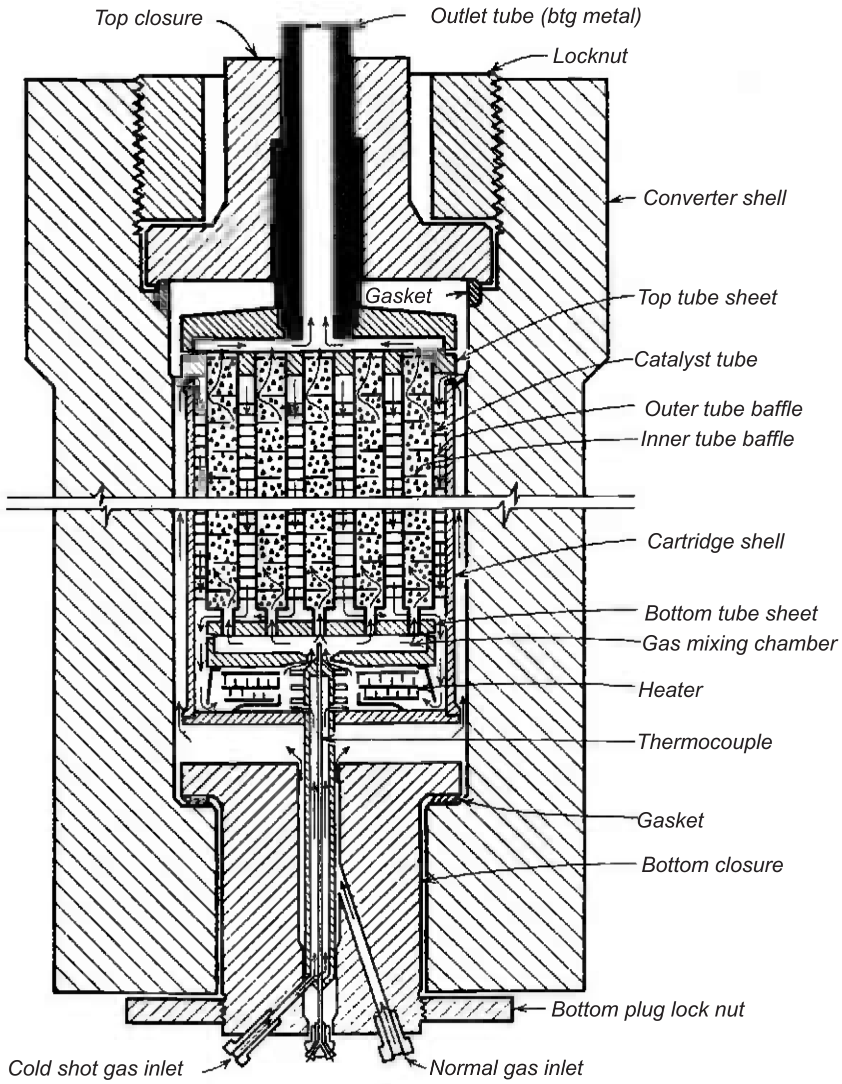
(12) 5.75″ Dia on 45.50″ B.C. SA-193-B16 (AISI 4340)

F_U = 120 KSI
F_Y = 90 KSI
S = 30.5 KSI

Gaskets: Annealed AISI 1020 Steel, Delta Ring

Misc Data:

- Vessel is refractory lined
- Shell Wt: 160 Kips
  Head Wt: 15 K (ea)
  Total Wt: 190 Kips
- No openings in the converter shell



EXAMPLE NO.8

Ammonia Converter
Matl: SA-105
10,000 PSIG @ 600°F

2.0. Shell Design

2.1. Introduction

When the thickness of a cylindrical pressure vessel becomes relatively large in comparison to the diameter ($t > .5R$), the variation of the stresses between the inner surface and outer surface becomes appreciable. At this point the ordinary membrane or "average" stress formulas are not a satisfactory indicator of the stresses.

In addition to the simple membrane stress of the cylinder, the shell is subjected to a radial stress due to the direct application of the pressure against the wall. This is a compressive stress and is insignificant for thin walled pressure vessels when compared to the other principal stresses. But the radial stress becomes more significant as the pressure and thus the thickness is increased.

In a thick walled cylinder, subjected to internal pressure, both circumferential and radial stresses are maximum on the inside surface. However, failure of the shell does not begin at the bore but in fibers along the outside surface of the shell. Although the fibers on the inside surface yield first, they are incapable of failing because they are restricted by the surrounding material, the outer regions of the shell. Above the elastic breakdown pressure, the region of plastic flow, or "overstrain" moves radially outward and causes the circumferential stress to reduce at the inner layers and increase in the outer layers. Thus the maximum hoop stress is reached there first and eventual failure begins there.

In 1833, Lamé developed a series of equations to find the greatest principal stresses in order to determine when the elastic limit was reached. Lamé proposed that elastic failure is considered to occur when the elastic limit of the material is reached. Beyond this limit, the material is permanently deformed or ruptured.

Coulomb and Tresca theorized, that the elastic limit was reached only when the shear stress reached it's maximum value. The basis of their theory was based on the actual failure mode of material. Material stretched beyond its elastic limit actually begins to fail along slip lines at 45 degrees from the applied force. The molecular bonds between adjoining atoms begin to slip along shear planes. The process is called twinning and results in Leuder's lines, visible at the point where failure begins, typically at some point of stress concentration.

Safety Factors for High Pressure Design. Designs below the creep range, are based on the elastic limit of the material. Historically, pressure vessel design for tensile requirements for continuous service was based on the lesser of;

a. ¼ Tensile (UTS)
b. ⅔ Yield
c. 1% strain for 100,000 hours for materials in the creep range

The safety factor for Division 1 was increased from 4:1 tensile to 3.5:1 in 2000. The Division 2 safety factor was historically 3:1 until 2007, when it was changed to 1:2.4. Historically, this translates to safety factors of 1.5 based on yielding or 4 based on bursting.

In the standard wall thickness calculation for ASME Section VIII, Division 1:

$$t = PR/(SE - .6P)$$

as the quantity SE - .6P approaches 0, the thickness approaches infinity. As the pressure increases, the allowable stress must be increased to higher than 60% of the design pressure for the equation to be valid. This becomes impractical for very high pressure applications, and a different theory of failure and design must be used.

Design of cylindrical shells is divided into the following categories;

a. Monobloc
b. Layered
c. Autofrettage

Monobloc cylindrical shells can be designed by one of three methods;

a. Standard method
(KD-220)
b. Elastic-Plastic Analysis
(KD-230)
c. Linear Elastic Analysis
(KD-240)

ASME Section VIII, Division 3 does not give equations for calculating wall thickness as in other sections or divisions of the Code. Instead, an equation is given for maximum design pressure (KD-221.1). This equation can then be used to calculate the required thickness by iteration. A good starting point for determining wall thickness is to use the equation and allowable stresses for ASME Section VIII, Division 2 and then iterate the thickness until the maximum design pressure allowed exceeds the required design pressure.

2.2. Design of Thick Walled Cylinders

Thick walled cylinders can be designed to either ASME Section VIII, Divisions 1, 2 or 3. However, what is different in each of the Divisions is the allowable stress. Using the same allowable stress in the equations for the three divisions will yield approximately the same results. An example has been provided to illustrate this.

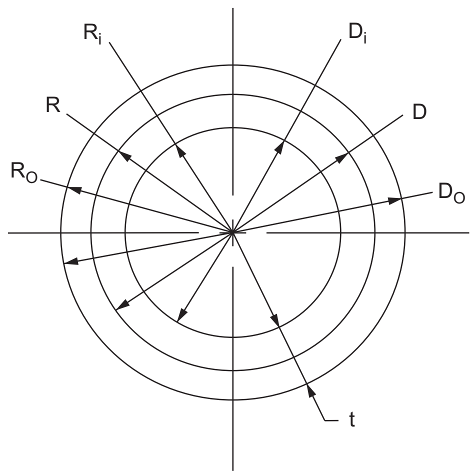
There are advantages to using Divisions 2 or 3 from a purely economic cost standpoint, since these two Divisions will yield lesser wall thickness, and therefore cost less to produce. However, this may not be the only consideration for choosing which Division to use. ASME Section VIII, Division 1 can be used economically up to about 10,000 PSI. However, if the designer wishes to reduce the wall thickness to the maximum extent possible, then there are advanced techniques in Division 2 and 3 that may be utilized. These advanced techniques are not covered in this book at this time.

These advanced techniques are as follows;

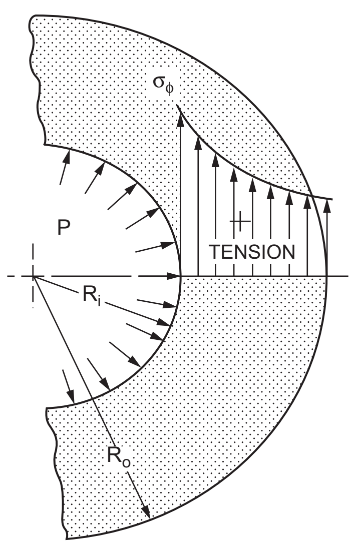
1. Autofrettage technique: Although multi-layer vessels can be built to either Divisions 1, 2 or 3, only Division 3 allows credit for the residual compressive stresses induced by the autofrettage technique.
2. Elastic Analysis
3. Limit Load Analysis
4. Elastic-Plastic

Nomenclature

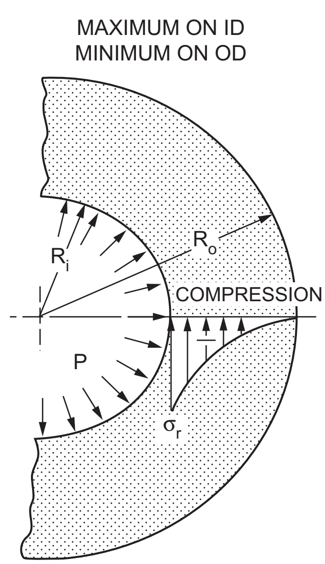
C.a. = Corrosion allowance, in
D = Any intermediate diameter, in
DT = Design temperature, °F
E = Modulus of elasticity, PSI
F_y = Minimum specified yield strength, PSI
P = Design internal pressure, PSIG
P_b = Burst pressure, PSIG
P_f = Pressure at which yielding occurs across the entire thickness, PSIG
P_m = MAWP, PSIG
P_y = Pressure at which yielding begins at the bore, PSIG
R = Any intermediate radius, in
S = Division 2 allowable stress, PSI
S_T = Tensile stress at design temperature, PSI
Y = Ratio, D_o / D_i
Z = Ratio, D_o / D
δ = Radial expansion, in
ε_X = Longitudinal strain, in/in
ε_ϕ = Circumferential strain, in/in
σ_X = Longitudinal stress, PSI
σ_ϕ = Circumferential stress, PSI
σ_r = Radial stress, PSI
σ_{XR} = Longitudinal stress at radius R, PSI
$\sigma_{\phi R}$ = Circumferential stress at radius R, PSI
σ_{rR} = Radial stress at radius R, PSI
τ = Shear stress, PSI
τ_R = Shear stress at radius R, PSI
Z_1 = ASME VIII-1 ratio



Dimensions of Thick Walled Cylinder



(a) Tangential stress distribution



(b) Radial stress distribution

MAXIMUM ON ID
0 AT OUTSIDE

A summary of the requirements of the three Divisions of the ASME Section VIII is as follows;

1.0. ASME VIII-1

Assuming E (Joint efficiency) is one, a thick walled cylinder is defined in Appendix 1-2 as meeting either of the following parameters;

1. $P > .385\ S$
2. $t > .5\ R$

For applications below either of these parameters, the standard equation (UG-27) for thin walled cylinders will yield the same results.

The formulas are;

$$t_r = R_i \left[Z_1^{1/2} - 1 \right] \text{ or}$$

$$t_r = \left[R_O \left(Z_1^{1/2} - 1 \right) \right] / Z_1^{1/2}$$

$$P_m = S(Z_1 - 1)/(Z_1 + 1)$$

$$\text{Where } Z_1 = (S + P)/[(S - P)]$$

$$\text{or } Z_1 = ((R_i + t)/R_i)^2$$

$$\text{or } Z_1 = (R_O / R_i)^2$$

2.0. ASME VIII-2

$$t_r = R_i \left(e^{P/S} - 1 \right)$$

$$P_m = (S\ t)/(R_i + .5\ t)$$

$$\text{or } P_m = (S\ t)/(R_O - .5\ t)$$

3.0. ASME VIII-3

ASME VIII-3 does not give a direct equation for calculating wall thickness. Neither does it give allowable stresses for materials. Instead it gives one simple equation for calculating allowable pressure. By iterative trials, a suitable wall thickness is assumed that will yield a pressure above design. By multiple trials the minimum wall thickness may be determined.

$$P_m = .667\ F_y \text{ Ln } Y$$

$$\text{Where } Y = R_O/R_i \text{ or } D_O/D_i$$

$$\text{If } S = .667\ F_y$$

$$\text{And } Y = D_O/D_i = (D_i + 2\ t)/D_i = 1 + 2\ t/D_i$$

Then; $P_m = S \ Ln \ [1 + (2\ t)/D_i]$

And $e^{P/S} = 1 + 2\ t/D_i$

Solving for t:

$$t_r = .5\ D_i\left(e^{P/S} - 1\right)$$

Therefore the equations for Division 2 and 3 are identical.

Comparison of ASME Sections Viii, Divisions 1, 2 & 3

If the same allowable stress is used it will provide a direct comparison between the various divisions;

Given;
Material: SA-105
P = 8000 PSIG
DT = 400°F
C.a. = 0.25 in
$F_y = 30.8$ KSI
$S = .667\ F_y = 20.53$ KSI
$R_i = 21.25$ in Corr
$D_i = 42.50$ in Corr

ASME VIII – 1

$P/S = 8{,}000/20{,}530 = .3897$

Therefore this vessel qualifies as a "thick" cylindrical shell.

$$Z_1 = (S + P)\ /\ [(S - P)]$$
$$= (20530 + 8000)/(20530 - 8000) = 2.2769$$

$$t_r = R_i\left(Z_1{}^{1/2} - 1\right) = 21.25\left(2.2769^{1/2} - 1\right)$$
$$= 10.815 \text{ in}$$

ASME VIII – 2

$P/S = .3897$

$$t_r = R_i\left(e^{P/S} - 1\right)$$
$$= 21.25\left(e^{.3897} - 1\right) = 10.125 \text{ in}$$

ASME VIII – 3

Assume, t = 10.125 +.25 = 10.375

$D_O = 42 + 2\,(10.375) = 62.75$

$Y = D_O/D_i = 62.75/42.5 = 1.4765$

$$P_m = .667\ F_y\ Ln\ Y = .667(30800)\ Ln\ 1.4765$$
$$= 8001 \text{ PSI}$$

Therefore all three divisions yield very similar results if the allowable stress are the same. This proves that the basis of the equations for determination of wall thickness for all three divisions are practically the same.

Lamé Equations

The equations shown on the following worksheet for determining shell thicknesses, are all based on Lamé equations. Reviewing the equations for various shell stresses show that they are in complete agreement with the ASME equations and yield the exact same results. The following equations serve to illustrate this point;

Lamé equations;

$$\sigma_\phi = \left[\left(P\ R_i{}^2\right)/\left(R_O{}^2 - R_i{}^2\right)\right]\ \left[1 + R_O{}^2/R_i{}^2\right]$$

$$\tau = \left(P\ R_O{}^2\right)/\left(R_O{}^2 - R_i{}^2\right)$$

Alternate equations;

$$\sigma_\phi = \left[P\ \left(Y^2 + 1\right)\right]/\left[Y^2 - 1\right]$$

$$\tau = \left(P\ Y^2\right)/\left(Y^2 - 1\right)$$

WORKSHEET FOR DETERMINING THICKNESS OF CYLINDRICAL SHELL							
GIVEN				STRESSES			
MATERIAL				TERM	EQUATION		VALUE
P				σ_x	$P / (Y^2 - 1)$		
DT				σ_ϕ	$P [(Y^2 + 1) / (Y^2 - 1)]$		
Ca				σ_r	$(-)P$		
F_y				τ	$(P\ Y^2) / (Y^2 - 1)$		
S Div 2				P_y	$.577\ F_y [(Y^2 - 1) / Y^2]$		
D_i				P_f	$1.155\ F_y \operatorname{Ln} Y$		
R_i				P_b	$S_T [(Y^2 + 1) / (Y^2 - 1)]$		
E				σ_{XR}	$[P / (Y^2 - 1)] [1 + Z^2]$		
S_T				$\sigma_{\phi R}$	$[P(Z^2 + 1)] / (Y^2 - 1)$		
CALCULATE				σ_{rR}	$[- P(Z^2 - 1)] / (Y^2 - 1)$		
TERM	EQUATION		VALUE	τ_R	$(P Z^2) / (Y^2 - 1)$		
t_r Div 2 (2)	$R_i [e^{P/S} - 1]$			ε_X	$[1.7\ P] / [E (Y^2 - 1)]$		
t	Actual shell thickness used			ε_ϕ	$[.4\ P] / [E (Y^2 - 1)]$		
D_O	$R_i + 2\ t$			δ	$t_\phi\ R_O$		
Y	D_O / D_i						
Z	D_O / D						
P_m	$.667\ F_y \operatorname{Ln} Y$						
TRIALS TO DETERMINE SHELL THICKNESS PER ASME VIII-3							
TRIAL	1	2	3	4	5	6	7
t							
D_o							
Y							
P_m							

NOTES:

1. Y = Z at the bore, Z = 1 at the outside of the cylinder
2. Use thickness calculated from Division 2 as first trial. Continue trials until P_m is close to P.

WORKSHEET FOR DETERMINING THICKNESS OF CYLINDRICAL SHELL		Example

GIVEN

MATERIAL	SA-723-2-2 (2-3/4 Ni 1-1/2 Cr 1/2 Mo)
P	15,000 PSIG
DT	600°F
Ca	0
F_y	106.7 KSI
S Div 2	56.3 KSI
D_i	36 in
R_i	18 in
E	25.1 X 10^6 PSI
S_T	135 KSI

CALCULATE

TERM	EQUATION	VALUE
t_r Div 2 (2)	$R_i\,[\,e^{P/S} - 1\,]$	5.49
t	Actual shell thickness used	4.25
D_O	$R_i + 2\ t$	44.5
Y	D_O / D_i	1.236
Z	D_O/ D @ R_m	1.106
P_m	$.667\ F_y \ \text{Ln}\ Y$	15,078 PSI

STRESSES

TERM	EQUATION	VALUE
σ_x	$P\ /\ (Y^2 - 1)$	28,425
σ_ϕ	$P\,[\,(Y^2 + 1)\ /\ (Y^2 - 1)\,]$	71,850
σ_r	(-)P	(-) 15,000
τ	$(P\ Y^2)\ /\ (Y^2 - 1)$	43,425
P_y	$.577\ F_y\,[\,(Y^2 - 1)\,/\,Y^2\,]$	21,270
P_f	$1.155\ F_y\ \text{Ln}\ Y$	26,110
P_b	$S_T[\,(Y^2 + 1)\ /\ (Y^2 - 1)\,]$	646,658
σ_{XR}	$[\,P\,/\,(Y^2 - 1)\,]\,[\,1 + Z^2\,]$	28,425
$\sigma_{\phi R}$	$[\,P\,(Z^2 + 1)\,]\,/\,(Y^2 - 1)$	63,196
σ_{rR}	$[\,-P\,(Z^2 - 1)\,]\,/\,(Y^2 - 1)$	(-) 6345
τ_R	$(P\,Z^2)\,/\,(Y^2 - 1)$	34,771
ε_X	$[\,1.7\,P\,]\,/\,[\,E\,(Y^2 - 1)\,]$	0.001925
ε_ϕ	$[\,.4\ P\,]\,/\,[\,E\,(Y^2 - 1)\,]$	0.000453
δ	$t_\phi\ R_O$	

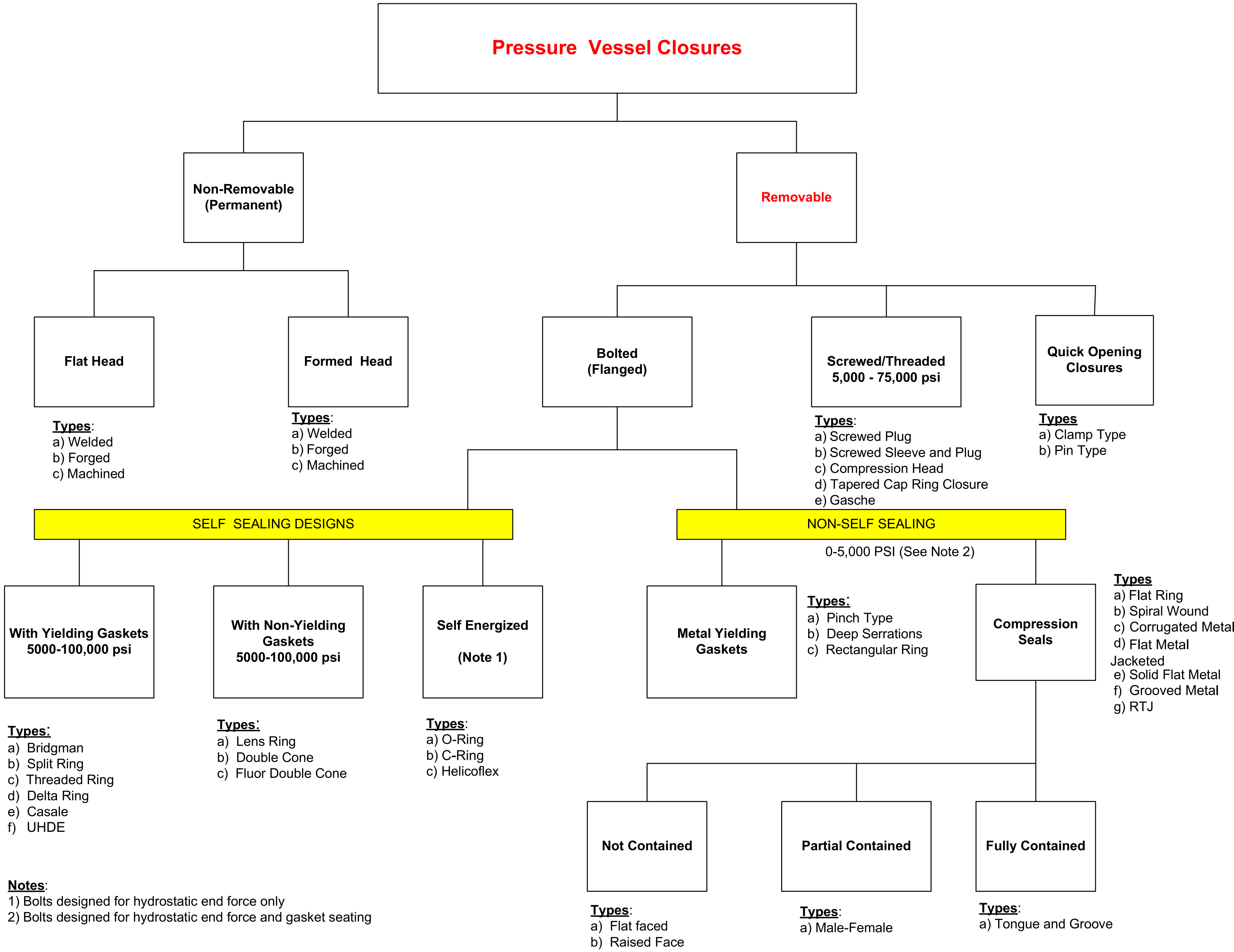
TRIALS TO DETERMINE SHELL THICKNESS PER ASME VIII-3

TRIAL	1	2	3	4	5	6	7
t	5	4.5	4.25				
D_o	46	45	44.5				
Y	1.277	1.25	1.236				
P_m	17,436	15,872	15,078				

NOTES:

1. Y = Z at the bore , Z = 1 at the outside of the cylinder
2. Use thickness calculated from Division 2 as first trial. Continue trials until P_m is close to P.

3.0. Design of Closures



Notes:
1) Bolts designed for hydrostatic end force only
2) Bolts designed for hydrostatic end force and gasket seating

High Pressure Closures

3.1. Introduction

Closures at the ends of vessels may be permanent, removable, screwed or flanged. The type of closure at the end of a pressure vessel depends on the frequency of opening, the size of the vessel, the temperature of the vessel, the corrosive conditions, openings and nozzles through the closure and the type of access provided by the removable head. Quick opening closures are available up to about 20,000 PSI.

Flanged Closures. Conventional flange seals are not particularly efficient as pressure seals. The reason is because as the pressure increases, it results in reduced gasket pressure. In order to maintain the seal, larger bolts are required. Larger bolts in turn result in a larger hub dimensions and thicker flanges to counteract the large bending moments. At some point the law of diminishing returns takes effect and the flange cannot be made large enough to resist the combined effect of bending plus hydrostatic end forces. Thus an alternate strategy is required. This alternate strategy is the principle of "yielding gaskets". Yielding gaskets can be either self-sealing or not self-sealing.

Standard flange design employs the principle of "non-yielding" gaskets. That is, a gasket is designed to have its surface yield, up to the point required to seal the joint. The flange face must be harder than the gasket material. As you approach the point of higher and higher pressure, you reach the point where the gasket and flange face cannot be made any harder. At this point the flanges become outlandishly large to uniformly distribute the gasket load.

For high pressure applications, the flange should be designed around a gasket that will yield. Yielding gaskets create a seal without the extreme force required to seat a gasket. Soft metal gaskets such as aluminum or copper have been used as yielding gaskets. A number of designs and methods have been used to cause a temporary stress concentration on the gasket, thus creating an initial flow of the material and thus a resultant perfect seal. Examples of yielding gaskets include pinch type, deep serrations, etc.

Self-sealing gaskets are those which are sealed by the system pressure. The system pressure acts to force the gasket to deform into a cavity that creates the seal. The seal remains intact by the system pressure.

Gaskets can be yielding or self-sealing or both. For example a delta gasket and Bridgman gasket are both yielding and self-missing sealing. A double cone and lens ring are both self-sealing but are non-yielding gaskets.

There are seven categories of removable end closures based on the type of gasket utilized. Some gasket types fit into more than one category. These are as follows;

1. Compression Seals - Standard Flanged Joints (not typically used for high pressure)
   a. Flat Ring
   b. Spiral Wound
   c. Corrugated Metal
   d. Flat Metal Jacketed
   e. Solid Flat Metal
   f. Grooved Metal
   g. Ring Joint
2. Types of Metal Yielding Gaskets
   a. Pinch Design
   b. Deep Serrations
   c. Delta Ring
   d. Rectangular Ring
   e. Lens Ring
3. Self-Sealing Closures with Yielding Gaskets
   a. Bridgman
   b. Split Ring Type
   c. Threaded Ring Type
   d. Delta Rings
4. Self-Sealing Closures with Non-Yielding Gaskets
   a. Lens Ring
   b. Double Cone
   c. Fluor Double Cone
5. Self Energized Gaskets (Self-Sealing)
   a. O-Ring
      1. Elastomeric
      2. Metal
         a. Plain
         b. Self Energized
         c. Pressure Filled
      3. Helicoflex
   b. Metal C-Ring
      1. For Internal Pressure
      2. For External Pressure
      3. For Axial Pressure
6. Breech Lock Closures
7. Other
   a. Outside Bump Ring
   b. Inside Bump Ring
   c. Gasche Spring Closure
   d. Casale Joint
   e. UHDE Type

Materials. Metal gaskets should be used for all applications over 930°F. A rough rule of thumb has been proposed by J. Harvey [1] that when the product of the pressure times the temperature exceeds 250,000, metal gaskets should be used.

Extreme care must be exercised in the selection of materials for sealing elements. The major considerations are as follows;

a. Ultimate strength of individual components
b. Compatibility of dissimilar metals
c. Difference in degree of component thermal expansion
d. Relative hardness and yield strength of mating components

Flat Ring Gaskets

These are conventional seals or gaskets and are also known as compression seals. There are three methods of utilizing flat ring gaskets:

a. Non-retained
b. Partially retained
c. Fully retained

In high pressure service a non-retained style gasket should never be used. This is a conventional type gasket and the potential to blow out is too great. Spiral wound gaskets have been used up to about 2500 psi service without being fully contained. Examples of partially retained gaskets are male-female flange facings, since they are not confined on the ID. They do however provide sufficient blowout protection.

Ring Joint Gaskets

Ring Joint Gaskets are neither yielding nor self-sealing. The sealing capabilities of RTJs are based on the sheer force required to seat the gasket and maintain a seal during operation. In the past, ring joint gaskets were primarily used for class 1500 or 2500 service. In overlayed, hydrogen service however the threat of cracks developing at the root of the groove has become highly probable and caused considerable concern within operating companies. The propagation of cracks in some cases has gone as deep as 7 inches into the base material. As a result, some owners are no longer specifying RTJ gaskets for new services, and are replacing existing flanges wherever possible. This includes both vessels as well as piping systems.

There are two types of ring joint gaskets. They are the oval ring and the octagonal ring. The oval ring provides a better seal but it is more difficult to produce the dimensional accuracy or surface finish desirable. Conversely, the octagonal ring can be produced more accurately, but requires more torque to cause the plastic flow required for sealing.

Ring type joints are generally considered only for a maximum of 4500 PSI. However API Spec 6A flanges have standard sizes for RTJ flanges to 20,000 PSI service.

Fortunately for ring joint gaskets it is extremely difficult to crush the gasket. Usually the bolts will elongate before the gasket is crushed. Gasket crushing is near the limits of the flanges ability to transmit the load. In addition the gaskets should always be softer than the flange itself. Then gaskets can plastically flow without damaging the surface of the groove.

Reusing of ring joint gaskets is not recommended due the probability of work hardening of the gasket material.

Self-sealing closures with Yielding Types of Gaskets

Most gaskets utilized for high pressure applications are of the self-sealing or self-energized types. The amount of bolting preload required to produce a seal is quite small. However, sufficient bolting force must be applied to resist hydraulic loading. The bolting requirement for gasket seating is never governing for these types of gaskets. This is why these types of gaskets are used. The hydraulic loading, hydrostatic end force, will always be the governing condition for these gaskets. Thus in most calculations, the gasket seating condition is ignored. Types are as follows;

a. Bridgman Closure
b. Delta Ring

Bridgman Closure: The Bridgman Closure is an Axial type of gasket as opposed to a radial type. The pressure reacts on the exposed gasket area in an axial rather than a radial direction. These are generally only considered above 1500 PSI.

Delta Ring: The Radial pressure acting over the height of the gasket produces a sealing force reaction at the tips. Initial seating at the tips is produced by controlled mechanical interferences. These gaskets are used in pressures in the 5,000 to 20,000 psi range. Galling does not occur if the angle of the groove is slightly greater than the angle of the ring. They do not require any initial seating or bolt stress for gasket seating.

Self-Energized Gaskets. O-Rings: Elastomeric o-rings are used for high pressure but low temperature, generally below 400°F. Metal O-Rings have been used for very high temperatures (1500°F) but low pressures. A carefully dimensioned groove or slot is provided for the O-Rings. The mating surfaces are normally separated by .9 of the diameter of the O-Rings. Under pressure, the internal pressure forces the O-Rings to the side of the groove and into the small gap between the mating flanges, thus sealing the joint. Pressures as high as 60,000 psi have been obtained with O-Rings.

Self-Sealing Closures with Non-Yielding Gaskets

Lens Ring: Normally used on small flanges. As the internal pressure increases, it acts on the inside surface of the ring, and forces it to the edge of the cone. Pressures as high as 45,000 psi have been obtained for very small sizes. Often considered one of the fastest opening closures.

Double Cone: Initial seating load is created by the stress to which the main bolts are tightened. The joints and gaskets must be machined to very high tolerances and very smooth finishes.

Design Methods

Lens, delta, double cone and Bridgman gaskets are all exposed to the vessel internal pressure. There are no standard calculations for determining the forces and moment induced into the flange as a result of these loadings. Accordingly, a free-body diagram is required to determine the additional loads put into the flanges and bolting as a result of pressure on the gasket. Detailed procedures are given for determining these effects for the Bridgman gasket. Similar treatments can be developed for the lens and delta types.

Because of the small contact areas and yielding gaskets, the gasket seating forces will never govern design of these flanges and thus can be ignored. The bolting forces due to hydrostatic end force will govern these flanges.

Table 8-1
Closures for high pressure vessels

Closure Type	Pressure Range PSIG	Inside Dia Range, in	Temperature Range, °F	Time for Opening	Ease of Fabrication	Remarks
Solid End - Bored	Unlimited	6	Metal	Permanent	Simple	Suitable for solid alloy or lined construction
Solid End - Forged	Unlimited	6-72	Metal	Permanent	Costly	Solid or electroplated
Flat Welded Head	1,000 - 2,000	6	Low	Permanent	Simple	Solid or thin metal liner
Formed Vessel Head - Welded Hemi or Ellipsoidal	10-15,000	Unlimited	Metal	Permanent	Simple	Solid, clad and weld overlay. Openings for ID access very expensive
Weld Neck Flg w/ Blind	6,000	24	Bolt Metal	Long	Simple	Solid, clad and weld overlay
Screwed Flg W/ Blind	6,000	12	Moderate	Long	Simple	Lining & clad difficult
Studding Flg W/ Blind	10,000	Unlimited	Metal	Long	Simple	Solid, clad and weld overlay
Clamp Type - Quick Opng Closure	25,000	>12	Gasket Matl	Quick	Critical Machining. Many parts	Solid, clad and weld overlay
Screwed Bull Plug	50,000	2	Gasket Matl	Quick	Simple	
Screwed Sleeve End Plug	5,000	3	Gasket Matl	Quick	Careful Machine Work	Satisfactory for high temperatures
Threaded Closure	100,000 (1)	8	Metal	Quick	Careful Machine Work	Solid, clad and weld overlay. Can use any type of self sealing gasket
	50,000 (1)	24	Metal	Quick	Careful Machine Work	Solid, clad and weld overlay
Tapered Cap - Ring Closure	5- 50,000 (1)	6 - 16	Stress in bolts	Quick	Careful Machine Work	Light weight, difficult to clea, economical for solid alloy construction
O-Ring	30,000	12	Gasket Matl	Depends on configuration	Relatively simple	Gasket easily replaced. Tight tolerances on o-ring groove
Bridgman	100,000 30,000 15,000	1- 3 3-6 12- 48	Gasket Matl Gasket Matl Gasket Matl	Long Long Long	Critical machining. Many parts. Tight tolerances and fine finishes	Careful cleaning and inspection after each opening. Use with flanged or threaded closure
Modified Bridgman	Unlimited	12-48	Gasket Matl	Long		
Double Cone/Fluor Double Cone	20,000	12-36	Gasket Matl	Depends on configuration	Lapping of groove required	Use with either threaded or flanged closure
Delta Ring	50,000	2-36	Gasket Matl	Depends on configuration	Tight tolerances and fine finishes required on groove	Use with flanged closure only
Lens Ring	45,000		Gasket Matl	Depends on configuration	Tight tolerances and fine finishes required on gasket & groove	Mainly used on piping systems

Notes:
Reference: Adapted from High Pressure Technology, E.W.Comings, McGraw-Hill Book Co, 1956
1. Pressure limit is for Self Sealing Gasket

3.2. Gaskets

Most gaskets utilized for high pressure applications are of the self-sealing or self-energized types. The amount of bolting preload required to produce a seal is quite small. However, sufficient bolting force must be applied to resist hydraulic loading. The bolting requirement for gasket seating is never governing for these types of gaskets. This is why these types of gaskets are used. The hydraulic loading, hydrostatic end force, will always be the governing condition for these gaskets. Thus in most calculations, the gasket seating condition are ignored.

1. Delta Gaskets: These are self sealing gaskets sealed by internal pressure. They do not require any initial seating or bolt stress. Good for pressures greater than 5,000 PSI.
2. Bridgman Type: Sealed by internal pressure. Used for applications above 1500 PSI.
3. Lens Ring: Self sealing and one of the fastest opening closures.
4. All metal gaskets required where temperatures exceed 930°F.
5. As a general rule, all nozzles should go through the top and bottom closures and not through the shell.
6. Ring type joint gaskets (RTJ) are only considered to be practical to 4500 PSI. RTJ gaskets are not sealed by internal pressure and require large bolt areas for gasket seating. RTJs are not effective in sealing where there is a rapid decrease in temperature. The gasket area is very small in relation to the flanges, and therefore contracts more quickly then the flanges and bolts. This reduces the pre-load and can lead to leakage.

Metal Gaskets. Selection of the material for metal gaskets is based on the following considerations;

- Temperature
- Coefficient of expansion (COE)
- Ductility
- Resistance to corrosion

Table 8-2
Properties of metal gaskets

Material	Max Temp °F	Max Recommended Temp °F	Thermal Conductivity, K	Coe, α
Lead	200	100	240	16.3
Brass	500	250	803-1100	11.6
Copper	600	300	2700	9.8
Aluminum	800	400	840-1500	13.7
304 SST	1000	500	150	9.1
316 SST	1000	500	150	9.1
Soft Iron, Low CS	1000	500	360	8.6
Titanium	1000	500	105	7.1
410 SST	1200	600	170	6.5
Silver	1200	600	2900	10.9
430 SST	1400	700	181	6
Nickel	1400	700	637	9.2
Monel	1500	750	173	9.5
321/347 SST	1600	800	112	10.2
Inconel	2000	1000	104	7.7
Hastelloy	2000	1000	87	7.9

Notes:

1. Units for Conductivity: BTu-in/hr/ft^2/°F
2. Units for coefficient of thermal expansion (COE): in/in/°F $\times 10^{-6}$

As a rough guide to determine whether metallic or nonmetallic gaskets should be used, multiply the operating pressure in psi times the operating temperature in °F. If the result exceeds 250,000, then metallic gaskets should be used. In general, nonmetallic gaskets should not be used above 850°F, or pressures above 1200 psi.

Of the above considerations, the coefficient of thermal expansion is most frequently overlooked. The importance of this consideration is best described by a comparison of the properties of the most common metals used (See Table 8-2). The coefficient of thermal expansion becomes important for applications where rapid temperature fluctuations or thermal gradients are present. Leakage can occur if the thermal coefficient of expansion between the gasket and flange material are too far apart.

In ammonia synthesis, copper was the preferred material for use with steel or low alloy flanges. Copper expands and contracts 9 times faster than steel. Leakages were found due to the large differences of the thermal expansion coefficients between the metals.

Although silver is used for special applications, copper, aluminum, steel and stainless steel are more common. Most alloys of aluminum should be limited to 400°F because the material becomes soft and has a tendency to extrude at under pressure through very small clearances.

Stainless steel has the problem of work hardening.

O-Rings. O-rings are known as self energized gaskets. That is, they use the system pressure to obtain a sealing force. A small amount of excess bolt load is required to prevent gasket blow-out. O-ring types are as follows;

1. Elastomeric
2. Metal
3. Helicoflex

Rubber or Elastomeric O-Rings

Those made of elastomeric materials work on the basis where the gasket is confined in a small groove or recess and the pressure attempts to extrude it out. The tightness of the gasket is dependent on the viscous strength of the seal material and the exit gap created by the machining tolerances and/or a separation of the flange and shoulder. A back-up ring may be used to restrict the extrusion of the softer o-ring material. The back-up ring can be made of nylon or metal. Pressures up to 30,000 PSI have been sealed in this manner. The density of the elastomeric material varies with the pressure being sealed.

Metal O-Rings

Metal o-rings are used in the plated or un-plated variety. Un-plated types are used in many liquid sealing applications. Platings and/or coatings are preferred for most gas applications. Metal o-rings can be used in temperature ranges from cryogenic to 2,000°F. As a rough rule of thumb, metal o-rings should always be used if the product of pressure and temperature is greater than 250,000.

There are three major types of metal o-rings;

1. Plain Type: Used for low to moderate pressures. Made of metal tubing and available in most alloys. These are the most economical and used to a pressure of about 100 PSI.
2. Self Energized: This type is utilized for high pressure applications. The surface exposed to the highest pressure, usually the inner periphery, is vented with small holes or slots. The pressure inside the vessel enters the o-ring and reduces the differential pressure across the seal.
3. Pressure Filled: Pressure filled o-rings are designed for high temperatures, from 800–2,000°F. The rings are filled with an inert gas to 600 PSIG. At high temperatures, the gas pressure increases and offsets the loss of strength of the tubing.

Helicoflex

Helicoflex o-rings are metal gaskets with a helical spring on the inside of the o-ring. This is a variation of the other types of o-rings based upon the plastic deformation of a jacket of greater ductility than the flange material. The close wound helical spring is selected to have a specific compression resistance. During compression, the pressure forces the jacket to yield and fill the flange imperfections. Each coil of the helical spring acts independently and allows the seal to conform to flange imperfections. These o-rings can be used from cryogenic applications up to 1800°F and pressures to 50,000 PSI and higher for special applications.

Jacket materials can be made of aluminum, silver, copper, soft iron, mild steel, nickel, monel, tantalum, stainless steel, inconel or titanium.

3.3. Bolted Flat Covers

Flat, unstayed covers may be integral or loose. If they are welded in place the construction is considered as

Table 8-3
Gasket allowable stresses

Matl			F_y (PSI)	S_g (PSI)	S_s (PSI)
1	Soft Copper or Brass		13,000	19,500	6,500
2	Soft Aluminum		8,800	13,200	4,400
3	Iron or Soft Steel		18,000	27,000	9,000
4	SST	T-304	30,000	45,000	15,000
5		T-304L	25,000	37,500	12,500
6		T-316	30,000	45,000	15,000
7		T-321	30,000	45,000	15,000
8		T-347	30,000	45,000	15,000
9	4-6% Cr		21,800	32,700	10,900
10	Nickel 200		15,000	22,500	7,500
11	Monel 400		21,800	32,700	10,900
12	Inconel 600		35,000	52,500	17,500
13	Incoloy 800		30,000	45,000	15,000
14					
15					

Notes:

1. The values in Table are for ambient conditions only! The gasket material selection is based on the design temperature and the process conditions. The properties for seating and sealing of the gasket should be based on the ambient properties.
2. Verify properties with gasket manufacturer prior to design.
3. $S_g = 1.5\ F_y$ $S_s = .5\ F_y$

integral. These are commonly referred to as flat heads. Conversely, bolted type heads are basically blind flanges.

Flat heads may be circular or non-circular. This section does not cover non-circular heads because this would be a very rare application for a high pressure vessel. The ASME Code, Section VIII, Division 2 has formulas to follow for fixed, non-circular, flat heads.

Heads may also be what is known as an end plug such as is used in Bridgman closures and threaded closures. The end plug is peculiar to these designs and the design of these is handled in their respective sections.

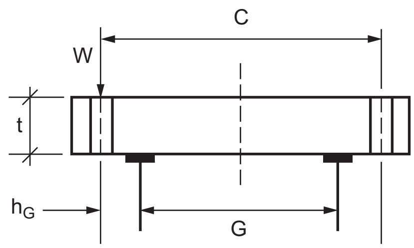
The thickness of flat, bolted heads is governed by bending, and not by tension. Thus, the formulas for the design of these components is the same for Division 1 or Division 2.

For high pressure vessels, the nozzles are typically located in the end closures, and not in the shells. Frequently they have centrally located openings with either integral or non-integral (loose) attachments for nozzles. This procedure provides for the design of both types of construction.

If the centrally located opening exceeds ½ the inside diameter of the vessel, then the design procedure reverts to a standard flange design.

BOLTED FLAT COVERS

CASE 1: NO OPENING



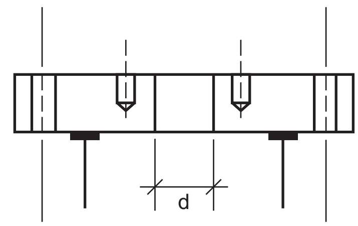
Thickness is greater of the following assuming C_1 attachment factor is equal to .3

$$t_G = G\sqrt{\frac{1.9Wh_G}{S_{fa}G^3}}$$

$$t_o = G\sqrt{\frac{.3P}{S_{fo}} + \frac{1.9W_{m1}h_G}{S_{fo}G^3}}$$

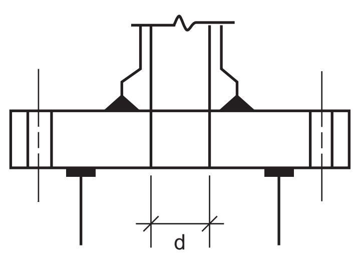
Terms W_{m1}, W, S_{fo} and S_{fa} are defined in the flange design section

CASE 2: CENTRAL OPENING, NON-INTEGRAL CONSTRUCTION, d < .5D



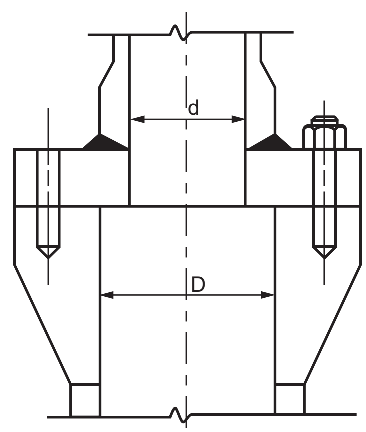
1. If d < .5D then area replacement is .5d (t_r). Increase flange thickness to compensate for area replacement.
2. Alternative design procedures;
   a. Design using C_1 = .6
   b. Increase flange thickness by 1.414

CASE 3: CENTRAL OPENING, INTEGRAL CONSTRUCTION, d < .5D



1. If d < .5D then area replacement is .5d (t_r). Increase flange thickness to compensate for area replacement.
2. Alternative design procedures;
   a. Design using C_1 = .6
   b. Increase flange thickness by 1.414

CASE 4: CENTRAL OPENING, INTEGRAL CONSTRUCTION, d > .5D



When d > .5D design the bolted flat cover as an ordinary weld neck flange.

3.4. Lens Ring Closure

The lens ring is one of the most common types of joints that originated in Germany at about 1920 with ammonia synthesis. The lens ring is used primarily on piping systems with some applications for pressure vessel heads.

The lens ring is a line contact seal for high pressure piping systems. The design is based on an unsupported area or self-sealing principal where the internal pressure acts on the inside surface of the gasket and forces it towards the edge of the cone. As the gasket expands, the sealing surface is forced further into the wedge of the seat. The force is applied radially.

There are many modifications of the basic lens ring. The most popular lens ring has spherical faces and is used between flanges with straight, tapered (20°) faces. The line of contact between the gasket and the flange faces is approximately 1/3 of the way across the flange face.

For hard materials the lens ring is not completely self-sealing like the Bridgman or delta ring, but tends to act more like a ring joint gasket rather than the other self-sealing gaskets. For softer materials like copper or aluminum, it can be considered as fully deformable, and therefore a self sealing type.

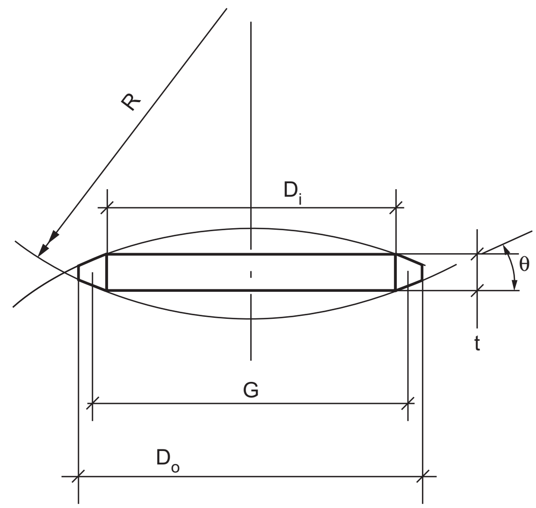
The gasket consists of a lens shaped ring of heat treated low alloy steel or some other metal. The ring should be softer than the flange face. Hardness of the conventional lens ring varies with the metal required for the service condition. This joint is ideal for pipe and tube applications. It can be used for small connections and can be used for pressure vessel closures but is better suited for piping applications.

Much of the problems associated with these joints is the non-uniform take up of the flange bolts as well as misalignment of the piping. The lens ring is one of the fastest opening closures. Lens ring gaskets have been used up to about 45,000 PSI.

Lens rings have been made with stiffening rings added to the basic lens ring, but the stiffening rings seem to be of little value. Hollowed out lens rings, lens rings with a groove cut on the inner periphery, have been used on the theory that internal pressure will "balloon out" the ring and increase its effectiveness. Hollowed out lens rings work satisfactorily, but their tolerances and hardness are very critical.

Types of lens ring gaskets:

a. Standard
b. Bellows
c. Stiffened Lens Ring
d. Modified Type



LENS RING GASKET

Data

A_b = Area of bolts, in^2, $R_a \times n$
F_y = Minimum specified yield strength, PSI
G = Mean gasket diameter, in
n = Number of bolts
N = Gasket seating width, in
R_a = Root area of one bolt, in^2
S_b = Allowable stress, bolt, PSI
θ = Angle of friction, (use 20° for mild steel)

Dimensions

- Gasket OD, D_O

  $D_O = \left[(4\,A_b/\pi) + D_i^2\right]^{1/2}$

- Mean gasket diameter, G

  $G = D_i + [(D_O - D_i)/3]$

- Gasket radius, R

  $R = .5\,G/\sin\theta$

- Minimum seating width, N

  $N = (1.5\,A_b\,S_b)/(3\,\pi\,G\,F_y)$

- Width of gasket, w

  $w = .5(D_O - D_i)$

- Basic gasket seating width, b_O

  b_O = Lesser of .25 N or .125 w

- Effective gasket seating width, b

  $b = b_O$ when $b_O \le 0.25$ in

  $b = .5\,b_O^{1/2}$ when $b_O > 0.25$ in

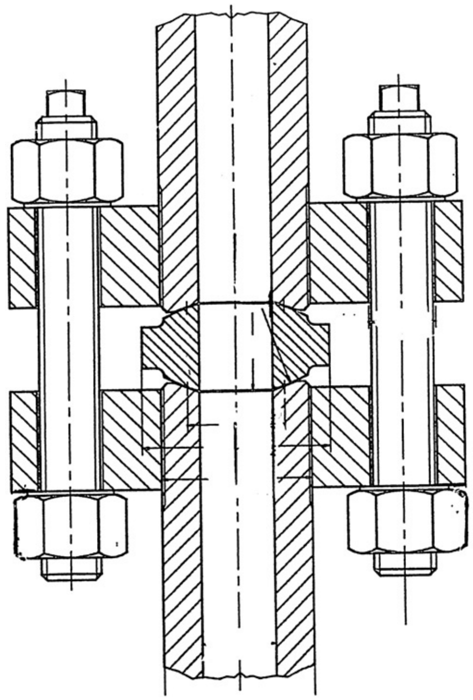


Figure 8-6. Lens ring closure.

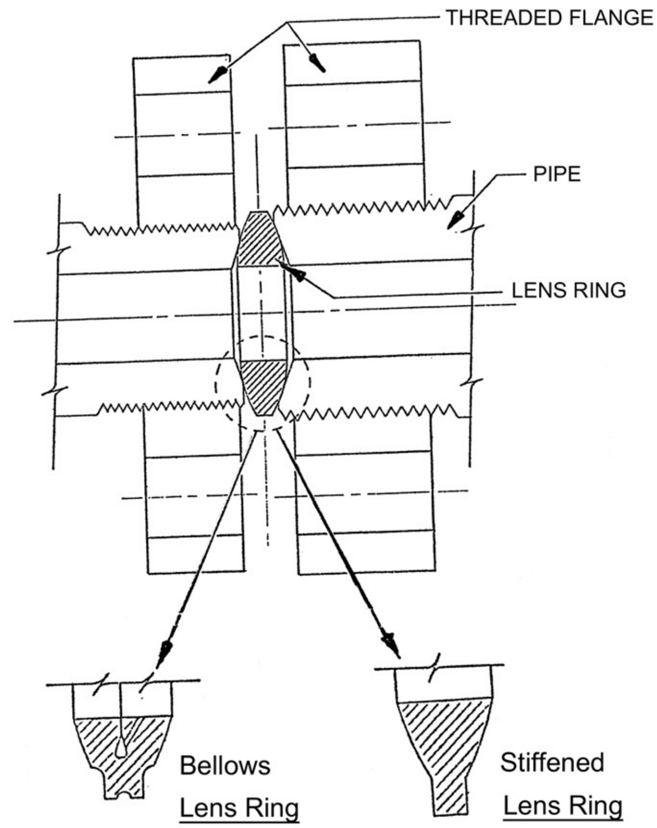


Figure 8-8. Lens ring closure.

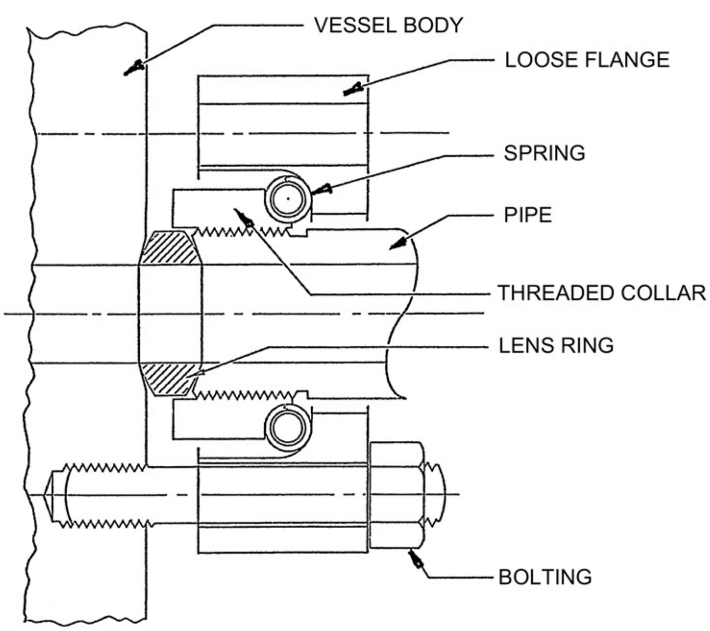
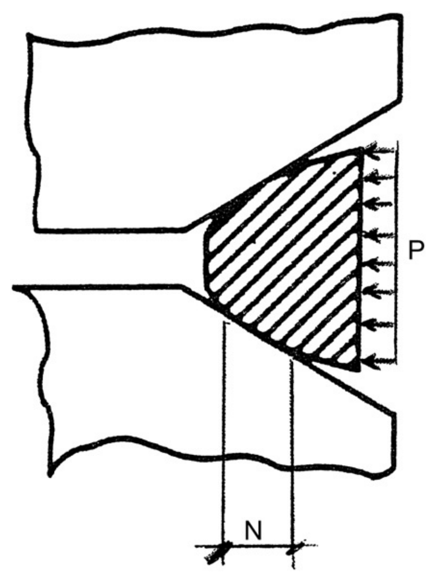


Figure 8-7. Modified lens ring closure.

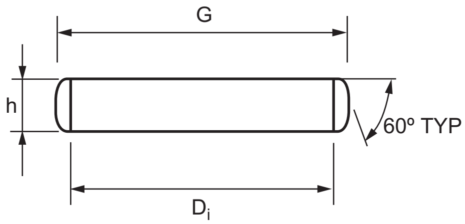


P = Radial Pressure

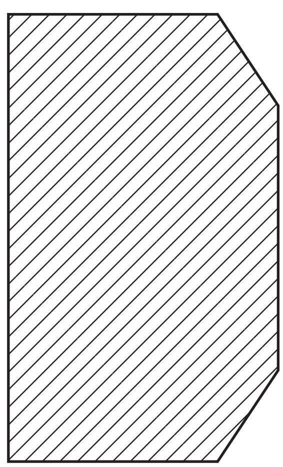
Figure 8-9. Lens ring closure.

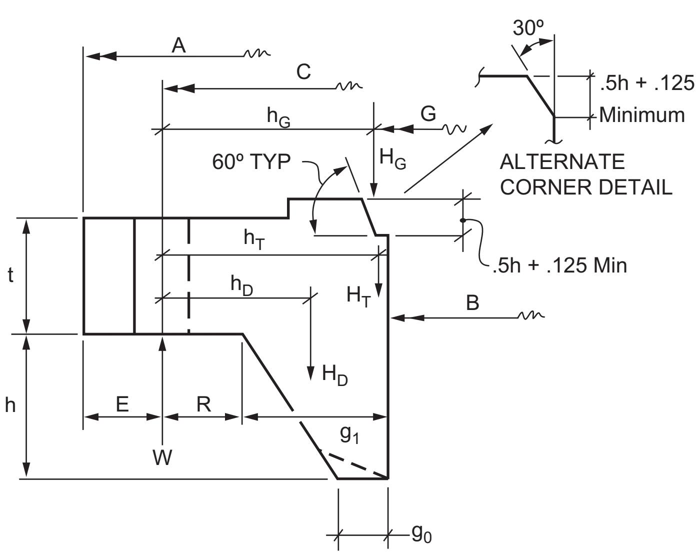
3.5. Double Cone Closures

The double cone ring type seal utilizes the principle of pressure sealing. This gasket type is good for high pressure, high temperature service. The double ring closure has the advantage of providing good seals for larger size vessel openings. As the pressure increases, the pressure sealing characteristics compensate for the effects of bolt elongation. They also allow for relatively easy assembly and disassembly.



DOUBLE CONE GASKET





GASKET DIMENSIONS AND LOADS

Like most high pressure joints, the double cone requires precision machined surfaces. In addition, proper bolt loads and tightening procedures are critical. Double cone gaskets can be used with either a bolted flat head or threaded closure type configuration.

Data

A_b = Area of bolts, in^2, $R_a \times n$
F_y = Minimum specified yield strength, PSI
G = Mean gasket diameter, in
n = Number of bolts
N = Gasket seating width, in
P = Design pressure, PSIG
R_a = Root area of one bolt, in^2
S_b = Allowable stress, bolt, PSI
y = Design seating stress of gasket, PSI

Dimensions

- Gasket OD, D_O

$$D_O = \left[\left(4\ A_b/\pi\right) + D_i^{\,2}\right]^{1/2}$$

- Minimum seating width, N

$$N = \left(1.5\ A_b\ S_b\right)/\left(3\ \mathrm{Cos}\ 60\ \pi\ G\ F_y\right)$$

- Mean gasket diameter, G

$$G = D_O - .5\ N$$

- Gasket radius, R

$$R = .5\ G/\mathrm{Sin}\ \theta$$

- Height of gasket, h

$$h = (y\ N\ \mathrm{Sin}\ 60)/P$$

- Width of gasket, w

$$w = .5\left(D_O - D_i\right)$$

- Basic gasket seating width, b_O

$$b_O = \text{Lesser of } .25\ N \text{ or } .125\ w$$

- Effective gasket seating width, b

$$b = b_O \text{ when } b_O \le 0.25 \text{ in}$$
$$b = .5\ b_O^{1/2} \text{ when } b_O > 0.25 \text{ in}$$

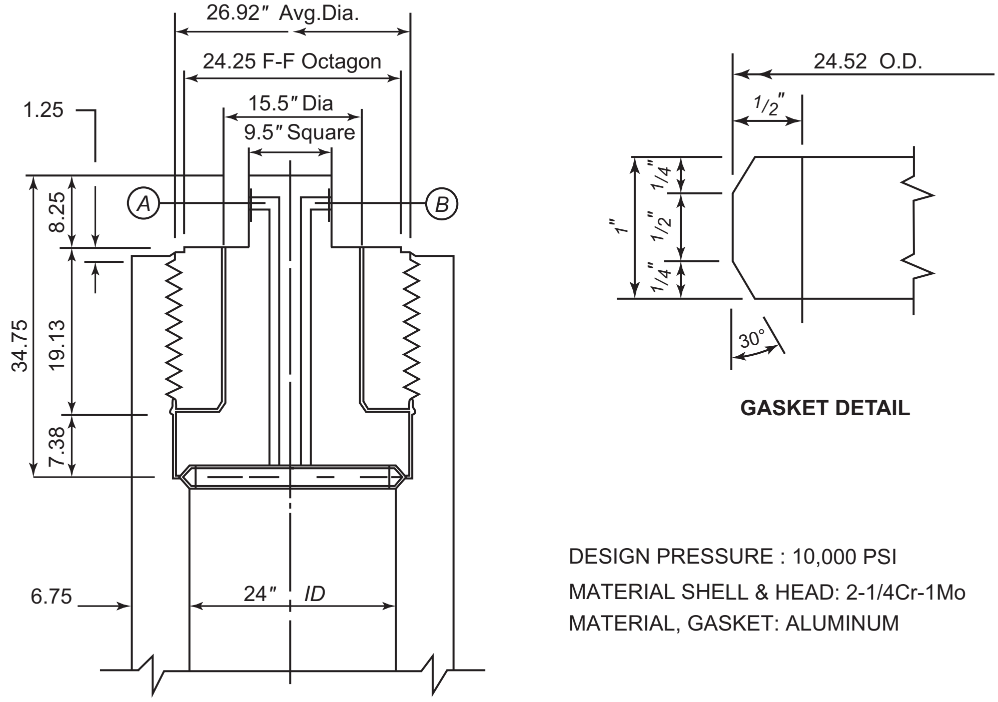


Figure 8-10. Threaded closure with double cone gasket

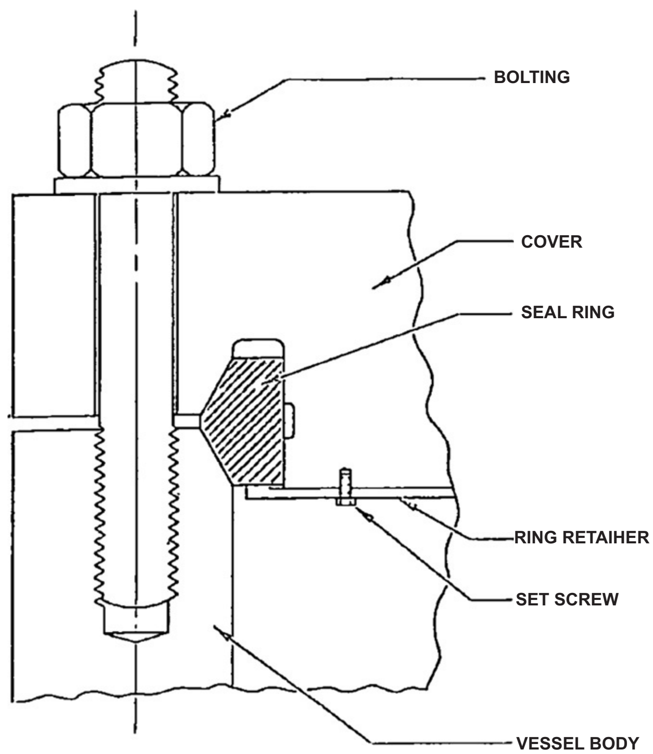


Figure 8-11. Double cone closure

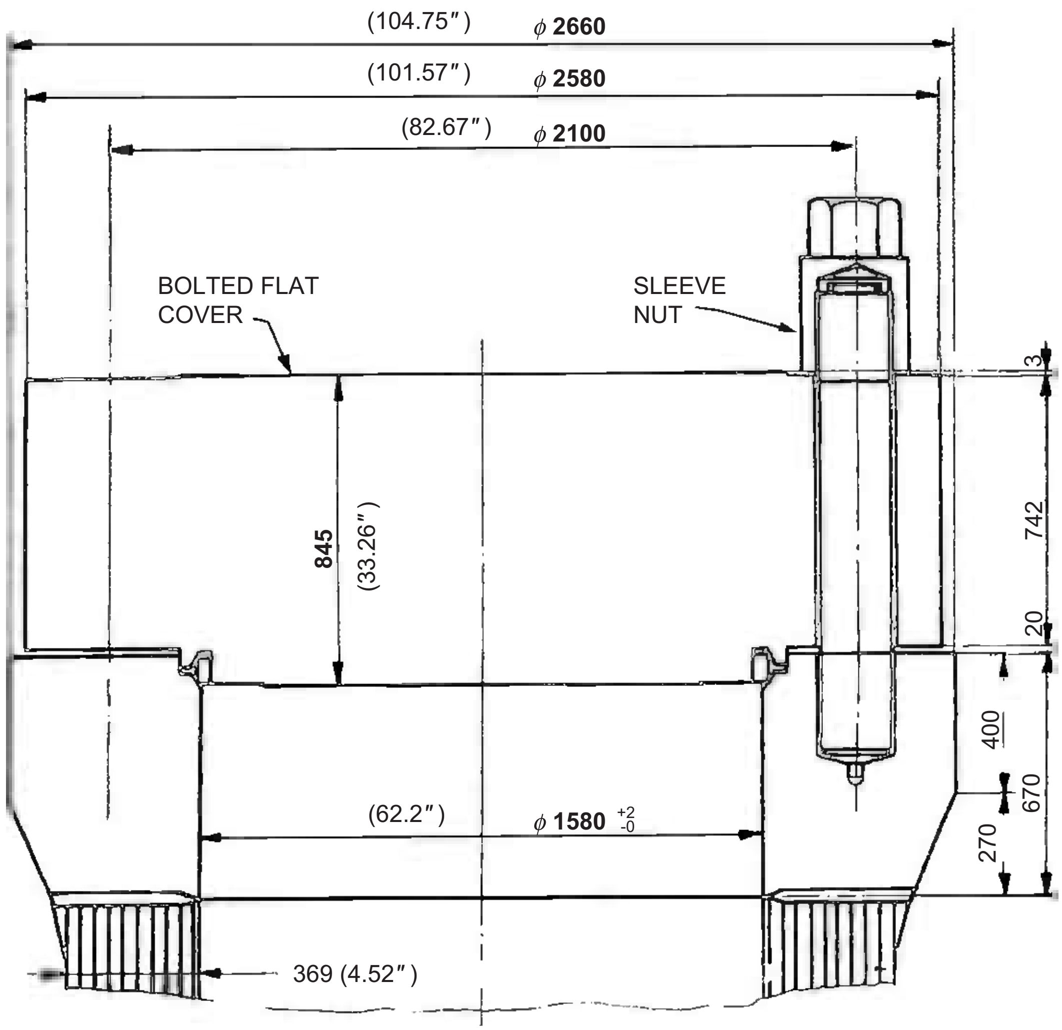


Figure 8-12. Example of double cone closure 10,000 PSI (680 ATM G) @ (-) 27 °F, (-) 33 °C.

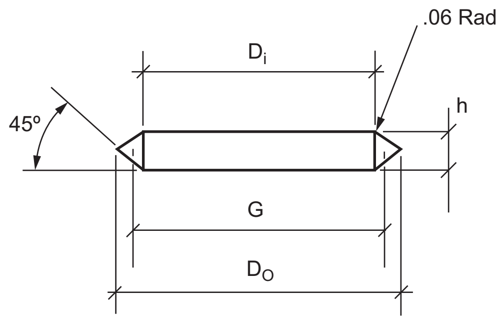
3.6. Delta Ring Closure

Delta rings were first developed in 1938. These gaskets are self-sealing based on the unsupported area principle. The gasket is triangular in cross section and is slightly wider than the combined depth of the gasket grooves. Initial sealing is obtained at the edges of the gasket during bolt-up and further sealing occurs during operation when the internal pressure forces the gasket further into the groove.

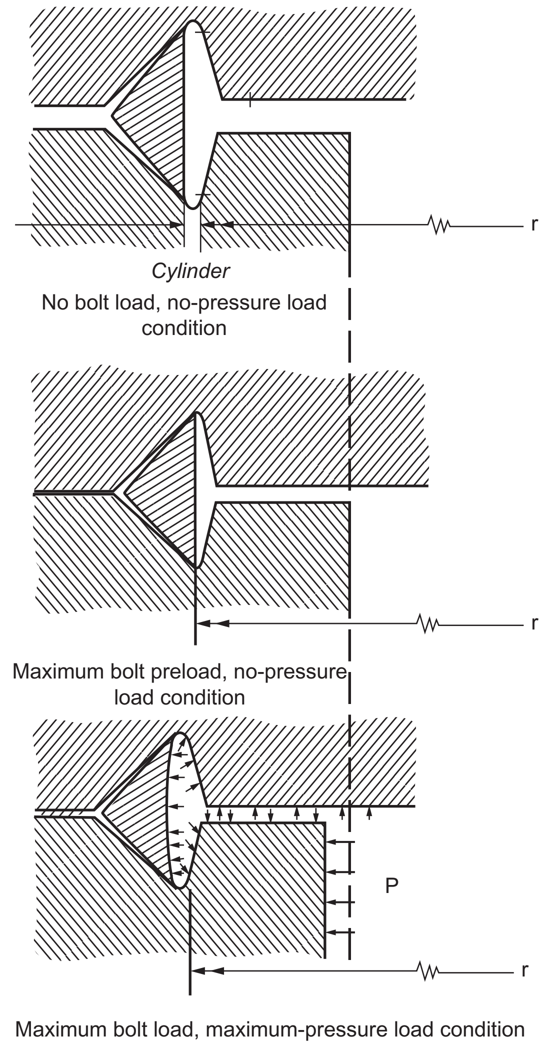
The delta ring gasket may not be suitable for very high cycle loadings such as on rotating machinery. This is due to the relative knife edge of the gasket and the high stresses at this interface point. In addition, the ring and groove dimensional control are too precise and costly to be used as a piping gasket.

These gaskets are used in the range from 2,500 to 30,000 PSI and diameters from 6 inches to 36 inches. This gasket is used more for vessels than on piping. The seating surfaces require very fine finishes, on the order of 15 AARH. Because it is self-sealing, severe tightening of the bolts is unnecessary. This in turn allows for the bolt loads to be used for the hydrostatic end load. It can be used for high temperature and corrosive service as well.

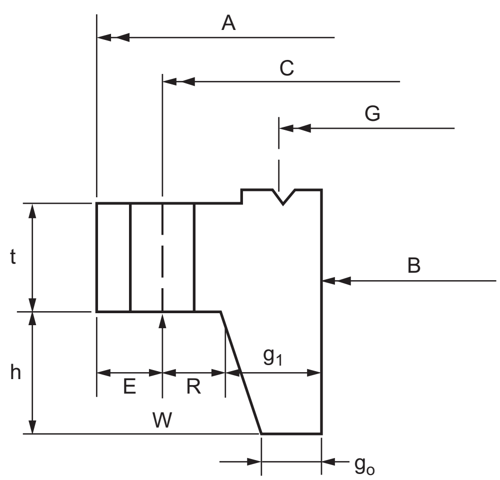
The delta joint has been used on converters, autoclaves, separators, etc.



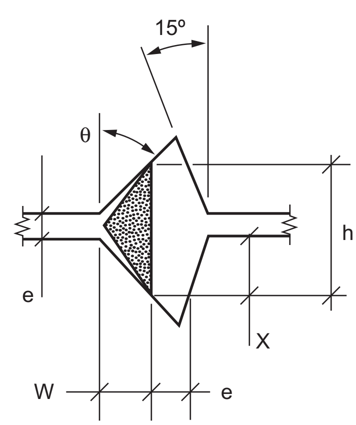
GASKET DIMENSIONS



APPLICATION OF PRESSURE ON GASKET



DIMENSIONS

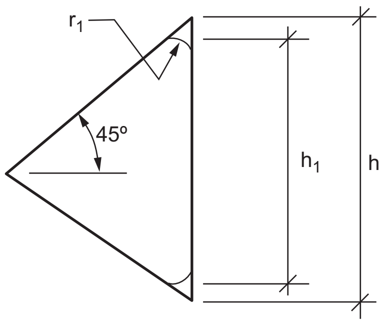


GASKET CAVITY DIMENSIONS

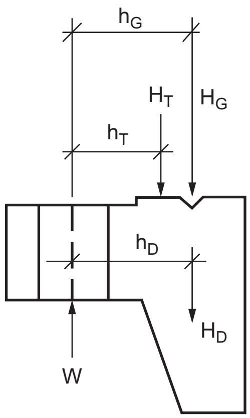
Studded Flange Details

Data

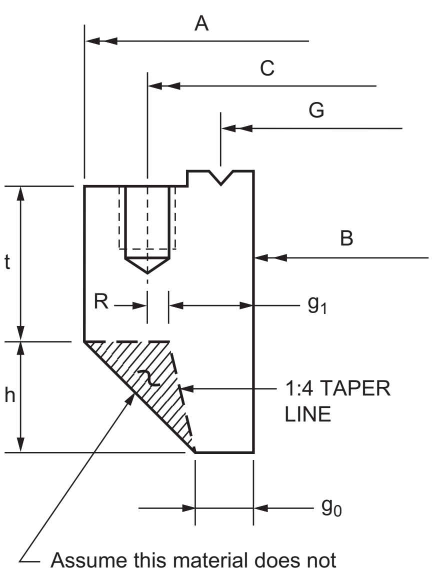
A_b = Area of bolts, in^2, $R_a \times n$
F_y = Minimum specified yield strength, PSI
G = Mean gasket diameter, in
P = Design pressure, PSIG
E = Modulus of elasticity of gasket material, PSI
N = Gasket seating width, in
R_a = Root area of one bolt, in^2
S_b = Allowable stress, bolt, PSI
y = Design seating stress of gasket, PSI



GASKET DIMENSIONS



FORCES



Assume this material does not exist and analyze the flange as an ordinary weld neck flange

$h_G = .5\,(C - G)$
$h_T = .5\,(R + g_1 + h_G)$
$h_D = R + .5g_1$

Table 8-4
Coefficient of friction, μ

Steel - Steel	.8
Steel - Aluminum	.61
Steel - Copper	.53
Steel - Brass	.51
Inconel - Monel	.25 - .4

Table 8-5
Properties of gaskets

MATERIAL	M	y
Aluminum	3.25	5,500
Copper	3.50	6,500
Brass	3.50	6,500
Iron	3.75	7,600
Monel	3.75	9,000
SST	4.25	10,000

Dimensions

- Mean gasket diameter, G
  $G = D_i + .125$
- Width of gasket, w
  $w = .5\ h$
- Basic gasket seating width, b_O
  b_O = Lesser of .25 N or .125 w
- Effective gasket seating width, b
  $b = b_O$ when $b_O \le .25''$
  $b = .5\ b_O^{1/2}$ when $b_O > .25''$

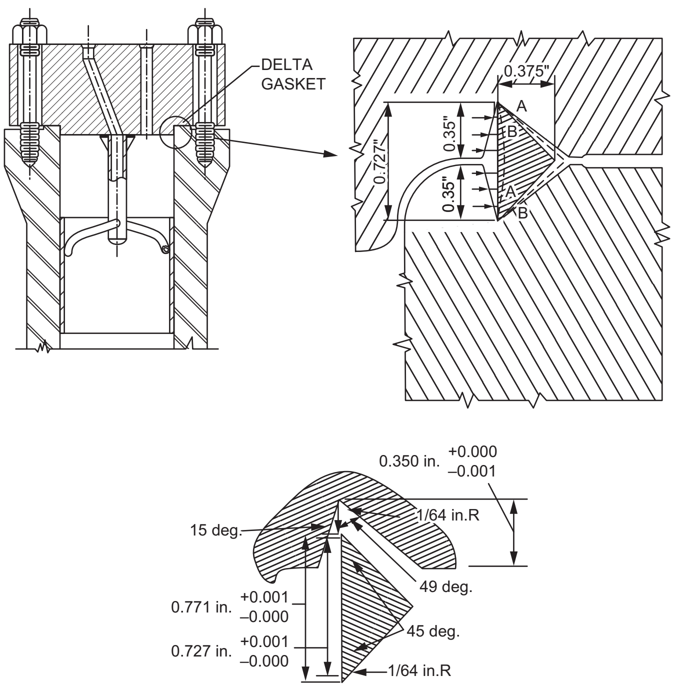


Figure 8-13. Delta gasket assembly.

3.7. Bridgman Closure

This is one of the first self-sealing joints based on the unsupported area principle. This joint was developed by Dr. P.W. Bridgman who was the author of the book, "The Physics of High Pressure", published in 1931. This type of sealing configuration has been used many times since 1929 and is highly reliable. Pressures up to 100,000 PSI have been achieved for small closures and pressures in the range of 25,000 PSI are readily available.

The Bridgman design is an axial type of self-sealing closure. In contrast, the delta ring and the lens ring closures are radial type. In the axial type, the force of the internal pressure acts in the axial direction when applied to the floating head. In the radial types the pressure acts radially to the gasket and the force is transmitted directly into radial contact with the gasket seat by a slight stretch of the ring.

The Bridgman closure can be used in either the bolted or threaded versions. The threaded version utilizes a main nut to provide the initial gasket seating contact before pressure is applied. In the threaded version the threads of the main nut take up the hydrostatic end force. In the bolted type, the main studs take the hydrostatic end force.

In the bolted head version, the pressure load is sustained by a large ring bolted to the top of the vessel. To obtain fluid tightness, the floating head (end plug) is pulled up against the triangular shaped gasket by bolts to get initial sealing. After this, pressure takes over and forces the seal into contact with the seat. It is the internal pressure that seats the gasket, not the force applied by either the main studs or the main nut.

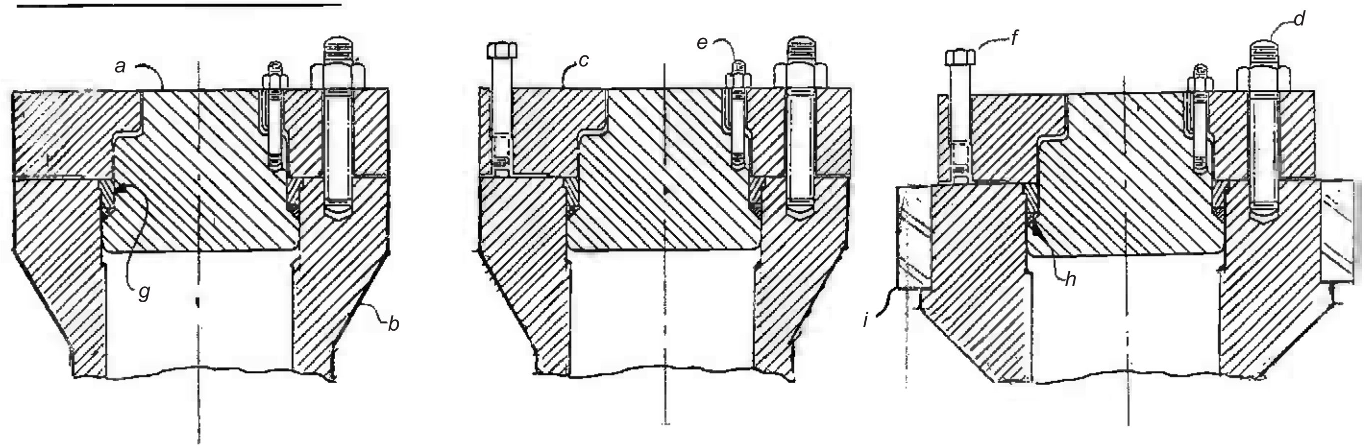
The chief drawbacks of this closure are:

a. High initial cost due to the exact machining tolerances of the gasket and seat.
b. Tendency of the gasket seat to become deformed and require additional machining.
c. Difficulty in adapting it to existing equipment.
d. When the gasket extrudes it causes the closure to stick and complicates removal of the head. Thus jack bolts are frequently utilized to allow for mechanical prying action to dislodge the extruded joint. Brass rings have been used to reduce the amount of extrusion of the gasket.

There are various options when specifying a Bridgman closure. Options include the use of jack bolts and shrink ring. Jack bolts are used for the removal of the end plug from the assembly. Expansion of the gasket during operation can extrude the gasket and make removal difficult. The jack bolts remedy this situation by providing a built-in method for removing the end plug. Jack bolts are not required for all cases. Jack bolts are never applicable for a threaded closure.

A shrink ring is used when the sum of all radial forces beyond the bolt circle cause excessive circumferential stresses in the outer ring section of the flange. A shrink ring provides residual compressive stress to reduce the tensile stress in this outer ring. The decision of whether to add a shrink ring, or increase the diameter of the flange forging is strictly one of economics.

BRIDGMAN CLOSURE



TYPE 1

- For lower pressure applications
- No jack bolts required

TYPE 2

- For medium pressure applications
- Jack bolts required

TYPE 3

- For higher pressure applications
- Jack bolts required
- Shrink ring required

Nomenclature

a. End Plug (Floating Head)
b. Flange
c. Retaining Ring
d. Main Studs
e. Pullup Studs
f. Jack Bolts
g. Follower Ring
h. Gasket
i. Shrink Ring

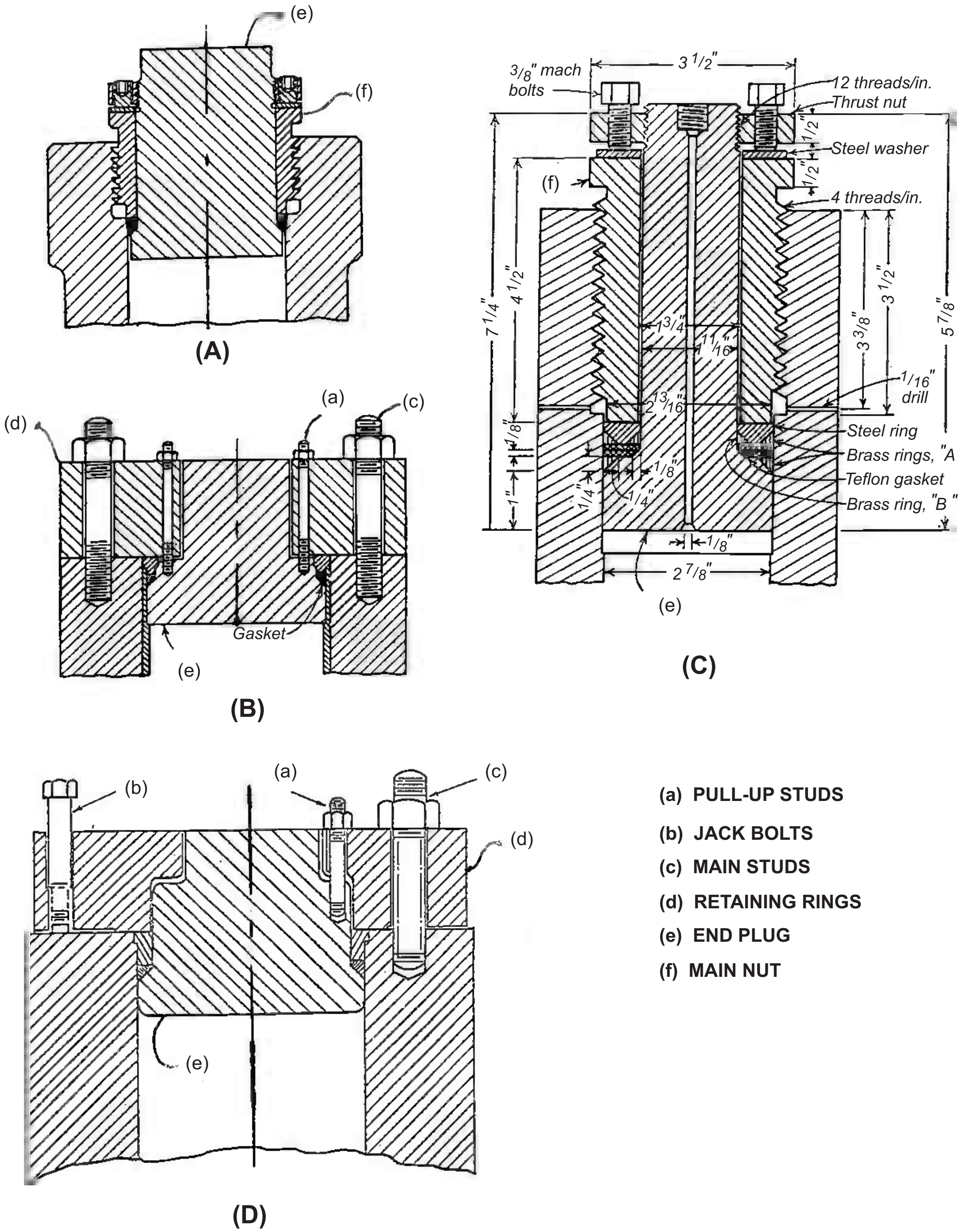


Figure 8-14. Bridgman type closures (A) Threaded plug type (B) Floating head type for large diameter heads (C) Modified type using rings to prevent extrusion of the gasket (D) Studded outlet with floating head.

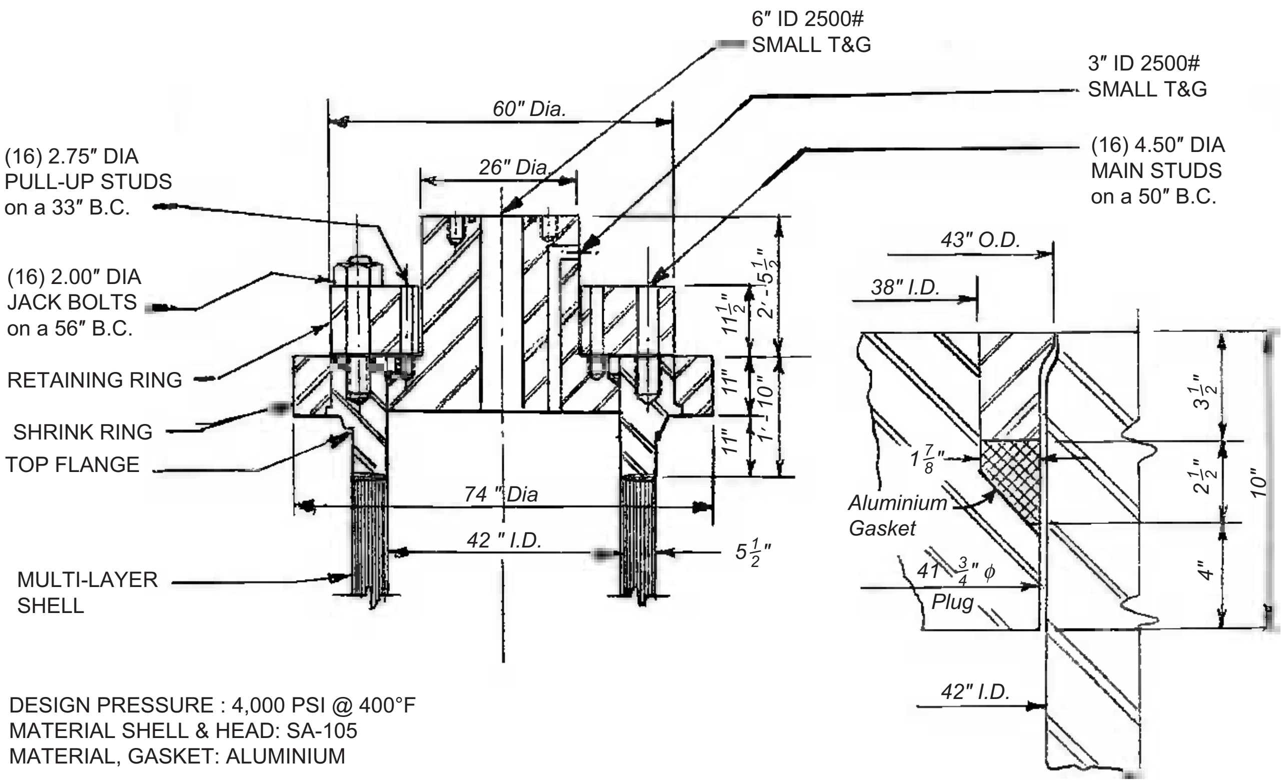


Figure 8-15. Bridgman joint, plug type, with shrink ring.

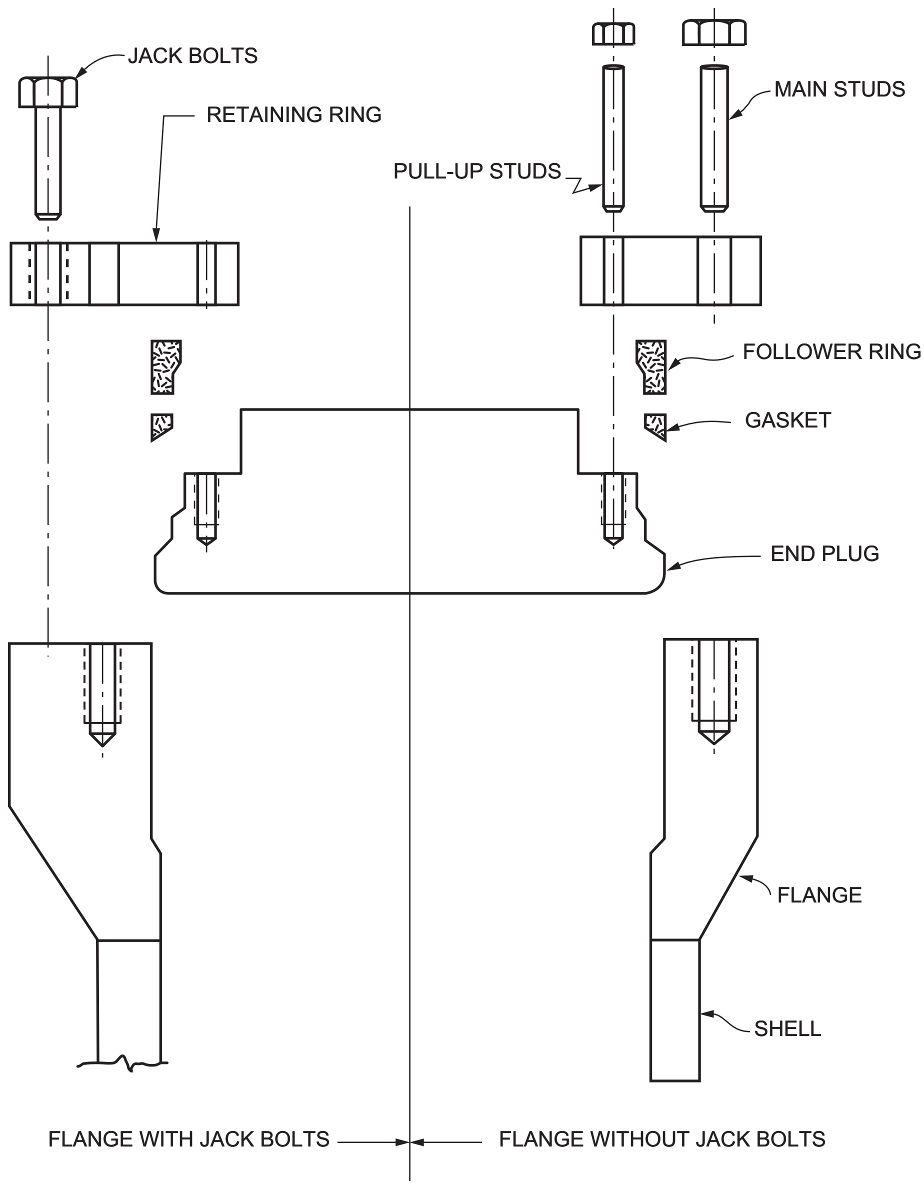


Figure 8-16. Exploded view of typical Bridgman closure.

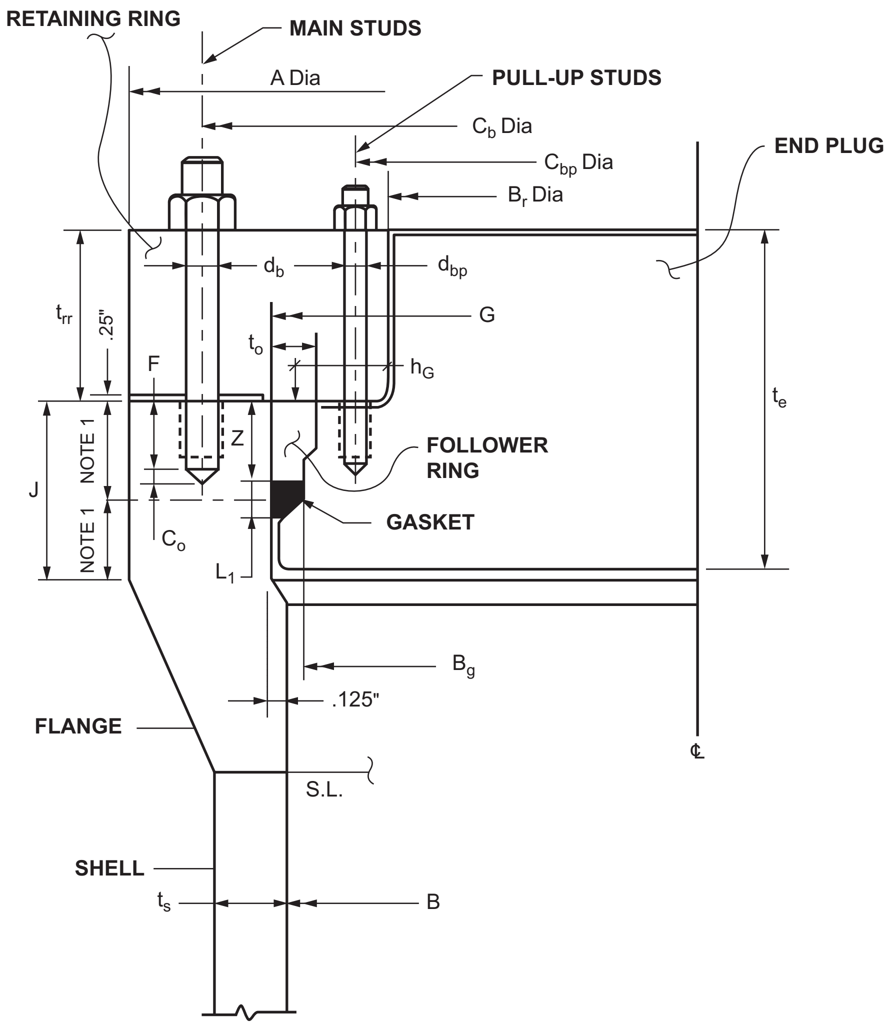
Bridgman Closure

Design Procedure

Notation

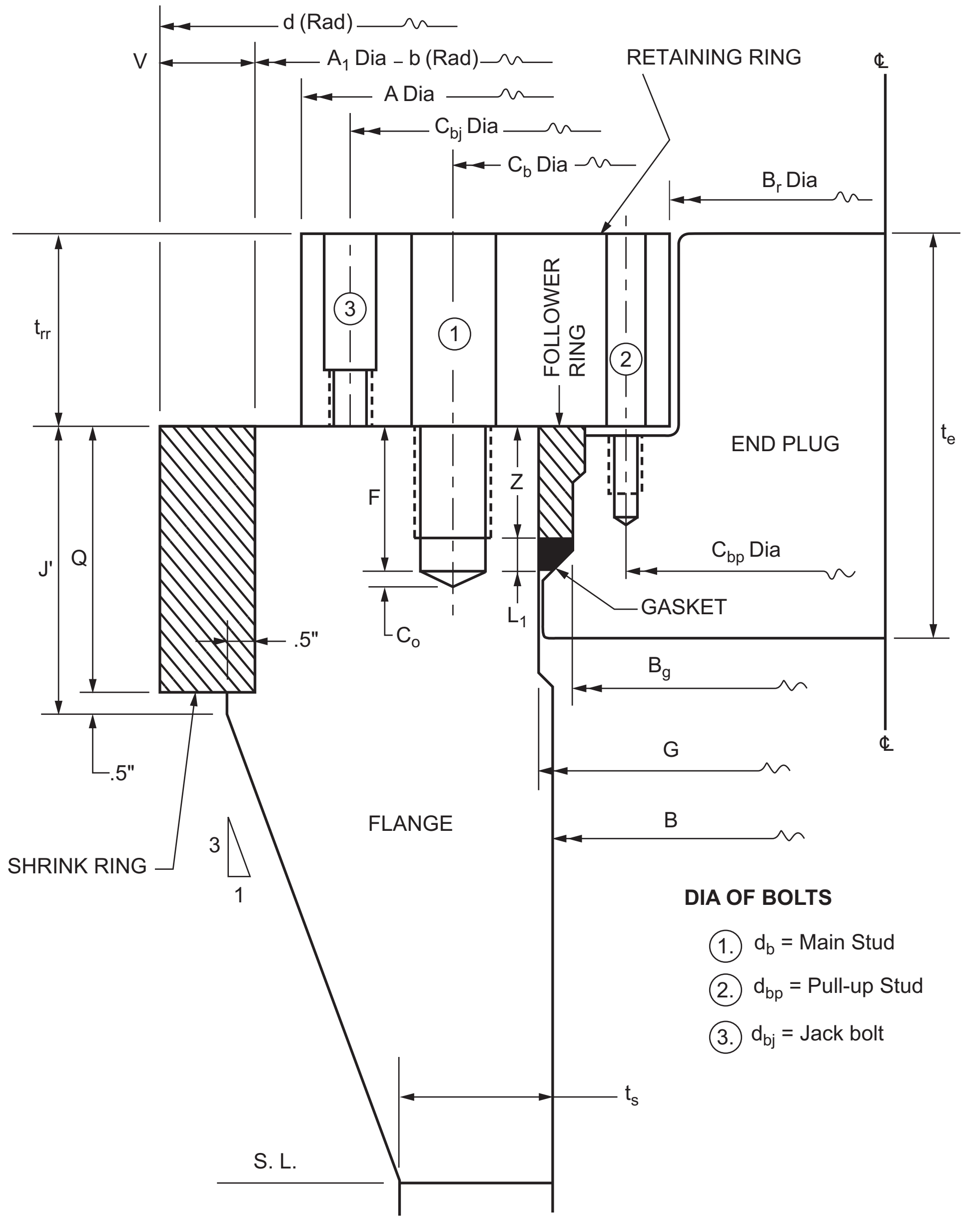
A_b = Area required, main studs, in^2
A_{bj} = Area required, jack bolts, in^2
A_{bp} = Area required, pull up studs, in^2
A_{Fr} = Area required, follower ring, in^2
A_g = Actual gasket area, in^2
A_r = Area required, in^2
B = Vessel ID, in
B_S = Bolt spacing, in
C = Cover factor, .3
C.a. = Corrosion allowance, in
C_W = Wrench clearance, nut, in
DT = Design temperature, °F
D_m = Mean diameter of follower ring, in
d_S = Diameter, main stud, in
d_{bj} = Diameter, jack bolt, in
d_{bp} = Diameter, pull up stud, in
E = Joint efficiency
E_M = Modulus of elasticity, PSI
f_{a1}, f_{a2}, f_{a3}, f_{aT}, f_{an} = Loads in flange, Lbs/in
FS = Factor of safety
F_y = Min specified yield strength, PSI
G = Mean gasket diameter, in
G_O, G_I = Gasket OD/ID, in
H = Hydrostatic end force, Lbs
J = Thickness flange, in
K = Ratio
M_r = Moment in retaining ring, in-Lbs
N = Quantity of bolts or studs
P = Internal pressure, PSIG
P_e = Equivalent internal pressure, PSIG
R = Vessel inside radius, in
R_a = Root area of bolt or stud, in^2
R_1 = Radial load due to gasket reaction, Lbs/in
R_2 = Radial load due to pressure, Lbs/in
R_T = Total combined radial load, Lbs/in
S_{if} = Hoop stress in flange at flange ID, PSI
S_{ri} = Hoop stress in flange at inner surface of main stud hole, PSI
S_{ro} = Hoop stress in flange at outer surface of main stud hole, PSI
S_{of} = Hoop stress at flange OD, PSI
S'_{if} = Corrected hoop stress at flange ID, PSI
t_e = Thickness, end plug, in
t_r = Thickness required, in
t_{rr} = Thickness, retaining ring, in
t_s = Thickness, shell, in
W = Main stud bolt load, Lbs
W_C = Width across corners of nuts, in
W_F = Width across flats of nuts, in
W_G = Pull up stud load, Lbs
W_J = Load on jack bolts, lbs
W_{ga} = Actual gasket load, Lbs
Y_1 = Correction ratio
Y = value of ASME VIII-1, Fig 2-7.1

Allowable Stresses: See Table 8-6



BRIDGMAN CLOSURE W/O SHRINK RING

Note 1: Centerline of gasket should correspond with centerline of flange thickness, "J"



DIA OF BOLTS

1. d_b = Main Stud
2. d_{bp} = Pull-up Stud
3. d_{bj} = Jack bolt

BRIDGMAN CLOSURE W/ SHRINK RING

TABLE 8-6
Summary of materials, stress & allowable stresses

	Symbol	Component	Temp	Basis/Description	Material	F_y (PSI)	Allow Stress (PSI)
1	S_a	Main Studs	Ambient	ASME Code Allowable Stress, Tension			
2	S_b	Main Studs	Design Temp	ASME Code Allowable Stress, Tension			
3	S_c	End Plug	Design Temp	Combined Stress			
4	S_e	End Plug	Design Temp	ASME Code Allowable Stress, Tension			
5	S_f	Flange	Design Temp	ASME Code Allowable Stress, Tension			
6	S_g	Gasket	Design Temp	See Table 8-3			
7	S_r	Retaining Ring	Design Temp	ASME Code Allowable Stress, Tension			
8	S_s	Gasket	Design Temp	Shear Stress, See Table 8-3			
9	S_{bj}	Jack Bolts	Design Temp	Bearing Stress, Use .9 F_y of Flange Material			
10	S_{bp}	Pull-Up Studs	Design Temp	ASME Code Allowable Stress, Tension			
11	S_{rb}	Retaining Ring	Design Temp	Bearing Stress, Use 1.6 X Sr			
12	S_{if}	Flange	Design Temp	Hoop Stress in Flange @ inner surface			
13	S_{ri}	Flange	Design Temp	Hoop Stress in Flange @ inner surface of main stud hole			
14	S_{ro}	Flange	Design Temp	Hoop Stress in Flange @ outer surface of main stud hole			
15	S_{of}	Flange	Design Temp	Hoop Stress in Flange @ outer surface of Flange			
16		Follower Ring	Design Temp				
17	S_{EB}	End Plug	Design Temp	Bending Stress			
18	S_{ES}	End Plug	Design Temp	Shear Stress			

Design Temp = ___________________________

WORKSHEET FOR BRIDGMAN TYPE STUDDING FLANGES

GIVEN	
FLANGE OD, A OR A_1	
FLANGE ID, B	
BOLT CIRCLE, C_b	
HYDROSTATIC END FORCE, H	
GASKET LOAD, W_{ga}	
AREA OF GASKET, A_{ga}	
DIAMETER OF STUDS, d_S	
HEIGHT OF METAL GASKET, L_1	
PRESSURE, P	
THICKNESS, t	

DIMENSIONS	
$a = .5\,(A \text{ or } A_1)$	
$b = .5\,B$	
$r_i = .5\,(C_b - d_S)$	
$r_O = .5\,(C_b + d_S)$	
$F = L + .25"$ or $(1.5\,d_S \text{ min})$	
$C_O = .288\,d_S$	
$J = F + .5\,(a - r_o)/2$	
$Z = .5\,J + .5\,L_1$	

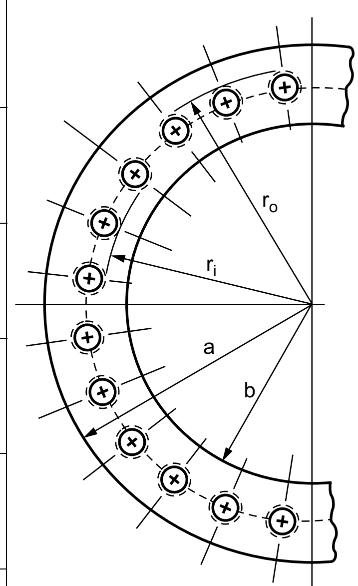
LOADS & STRESSES			
$R_1 = [\,(H + W_{ga})\,L_1\,]\,/A_{ga}$		$f_{an} = f_{at} - [\,(F + .5\,C_O)\,f_{a2}\,/\,J]$	
$R_2 = P\,[\,J - (\,Z + L_1\,)\,]$		$Y_1 = f_{at}\,/\,f_{an}$	
$R_T = R_1 + R_2$		$S'_{if} = Y_1\,S_{if}$	
$P_e = R_T\,/\,J$			
$S_{if} = P_e\,[\,(a^2 + b^2\,)\,/\,(\,a^2 - b^2\,)\,]$			
$S_{ri} = [P_e\,b^2/\,(a^2 - b^2\,)]\,[1 + a^2\,/\,r_i^2\,]$			
$S_{ro} = [P_e\,b^2/\,(a^2 - b^2\,)]\,[1 + a^2\,/\,r_o^2\,]$			
$S_{of} = (2\,P_e\,b^2\,)\,/\,(\,a^2 - b^2\,)$			
$f_{a1} = .5\,(\,r_i - b\,)\,(\,S_{if} + S_{ri}\,)$			
$f_{a2} = .5\,(\,r_o - r_i\,)\,(\,S_{ri} + S_{ro}\,)$			
$f_{a3} = .5\,(\,a - r_o\,)\,(\,S_{ro} + S_{of}\,)$			
$f_{aT} = f_{a1} + f_{a2} + f_{a3}$			

NOTES:

1. If $S'_{if} \le$ Allowable Stress, then the design is OK
2. If $S'_{if} >$ Allowable Stress, then implement a or b below

a. Increase flange proportions

b. Add a shrink ring



Plan View-Dimensions

Bridgman Closure

Calculation Procedure

1.0 GIVEN;

$B = ID =$ _______________

$R = IR =$ _______________

$P =$ Internal Pressure = __________

$DT =$ Design Temperature = _________

Corrosion Allowance = ____________

$FS =$ Factor of Safety = ____

$E =$ Joint Efficiency = _____

$E_m =$ Modulus of Elasticity = _______

$t_S =$ Shell Thickness = __________

$C =$ Cover Factor = ________

2.0 MAIN STUDS;

Material: __________________

$S_b =$ ___________________

a. Mean gasket diameter, G

$$G = B + 0.25 \text{ in}$$

b. Hydrostatic end force, H

$$H = .25\left(\pi G^2 P\right)$$

c. Area required, studs, A_b

$$A_b = H / S_b$$

d. Quantity of studs required, N

Always use multiples of 4, $N = A_b / R_a$

TRIAL	d_b	R_a	N	USE
1				
2				
3				
4				

e. Determine bolt circle, C_b

$$C_b\text{min} = B + d_b + 2 \text{ in}$$

f. Check stud spacing, B_S

$$\text{Max}: (\pi C_b)/N < 2 d_b + J$$

$$\text{Min}: > B_S \text{ from Table 8-7}$$

g. Main stud bolt load, W

$$W = N R_a S_b$$

3.0 GASKET

1. Determine gasket material and properties;
   a. Material _________
   b. S_g from Table 8-3
   c. F_y from Table 8-3
2. Determine gasket proportions;
   a. Area of gasket required;

   $$A_g = \left(2 H\right)/\left(2 S_g - F_y\right)$$

   b. Pull-up stud load required to deform gasket;

   $$W_g = .5 A_g F_y$$

   c. Set gasket OD equal to G;

   $$G_O = G$$

   d. Determine maximum gasket ID;

   $$G_i \text{ max} = \left[G_O{}^2 - \left(4 A_g\right)/\pi\right]^{1/2}$$

   Use $G_i =$ ____________
3. Based on selected gasket dimensions determine;
   a. Actual gasket area;

   $$A_{ga} = \left(.25 \pi G_O{}^2 - .25 \pi G_i{}^2\right)$$

   b. Actual gasket load;

   $$W_{ga} = .5 A_{ga} F_y$$
4. Gasket dimensions;

   $$w = .5 \left(G_O - G_i\right)$$

   $$L_i = w + 0.25 \text{ in Min}$$

4.0 PULL-UP STUDS;

Material: __________

$S_{bp} =$ ___________

a. Required stud area;

$$A_{bp} = W_{ga}/S_{bp}$$

b. Determine quantity of pull-up studs;

Note: The quantity should be the same as the number of main studs if possible.

$$N = A_{bp}/R_a$$

Use: $d_{bp} =$ ________

c. Determine bolt circle;

Minimum:

$$C_{bp} = G_I - 2\left(d_{bp} - 0.125 \text{ in}\right)$$

Note: For pull-up studs less than 1.5 inches diameter, do not subtract 0.125 inches in the above equation.

Maximum:

$C_{bp} = B_r + 3\ d_{bp}$

Use: __________

4.0 FOLLOWER RING

Retainer Ring Material: _________

$S_r =$ ________

- Allowable bearing stress;

$S_{rb} = 1.6\ S_r$

- Minimum area at top of follower ring;

$A_{Fr} = \left(H + W_{ga}\right)/S_{rb}$

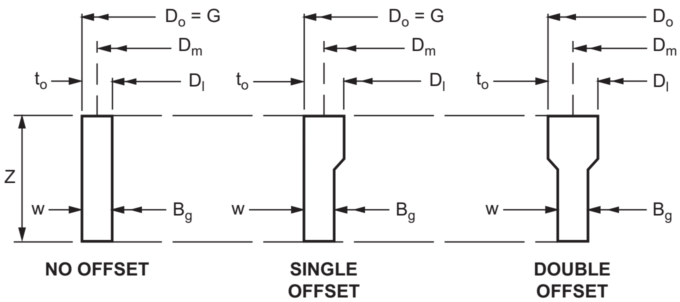
- As a first trial, assume $D_O = G$ and solve for D_i;

$D_i = \left[D_O{}^2 - \left(4\ A_{Fr}\right)/\pi\right]^{1/2}$

Notes for design of Follower Rings;

1. The offset options are determined such that the bearing stress on the retainer ring is not exceeded.
2. The bottom of the follower ring shall have the same ID and OD as the gasket.
3. The maximum offset shall not exceed 25% of width “w”.
4. It is preferable to set the OD of the follower ring as equal to the OD of the gasket.
5. The double offset option should only be utilized when the follower ring width, t_o, exceeds the 25% limit of the gasket width “w”.
6. Width of ring, t_o

   $t_o = .5\ (D_o - D_i)$
7. If $t_o < w$ then design is OK as is.
8. If $t_o > w$ then the designer must decide whether a single or double offset is required.
9. Begin with a single offset and check that offset does not exceed .25 w.
10. If the single offset exceeds .25 w, then a double offset is required.
11. Check the width of ring, t_o, to ensure that neither the inner nor outer offset exceeds the 25% w criteria. This effectively sets a max limit of t_o as lesser than or equal to 1.5 w.
12. If t_o exceeds 1.5 w then either the ring width, w, must be made wider or the material must be changed to increase the allowable bearing strength.



5.0 JACK BOLTS

Notes:

1. The jack bolts are used to remove the end plug in the event that the gasket becomes fused to the inside wall of the flange.
2. For design purposes assume that 65% of the gasket is welded to the flange.
3. The jack bolts must be capable of exerting enough force on the end of the flange to break the weld in shear.
4. A copper gasket is the most easily fused.
5. Use the same quantity of jack bolts as main studs as a starting point.
6. Jack bolts should be a minimum of 1-1/2” diameter.
7. Check wrench clearances between all top nuts and bolts.
8. Stagger jack bolts in between the main studs to obtain adequate wrench clearance.
9. It is common practice to leave the jack bolts in the retainer ring during operation.
10. Maintain at least one bolt diameter between the jack bolt circle and the OD of the retaining ring.

- Design load on jack bolts;

  $W_J = .65\ S_S\ G_O\ \pi\ L_i$
- Allowable bearing stress, flange material;

  $S_{bj} = .9\ F_y$
- Area required, bolts;

  $A_{bj} = W_J/S_{bj}$
- Actual bolting used;

  N = ___________

  d_{bj} = ___________
- Determine bolt circle required;

The bolt circle required for the jack bolts, C_{bj}, is dependent on wrench clearances and spacing between nuts. Recommended wrench clearance and dimensions of nuts are contained in the following Table.

Table 8-7
Nut and wrench clearance

Item	**Bolt Size (in)**				
	1.5	**1.75**	**2**	**2.25**	**2.5**
C_W	3.375	4.125	4.5	4.875	5.375
W_C	2.742	3.175	3.608	4.041	4.474
W_F	2.375	2.75	3.125	3.5	3.875
	Bolt Size (in)				
	2.75	**3**	**3.5**	**4**	**4.5**
C_W	5.875	6.125	7	7.66	8.5
W_C	4.907	5.34	5.928	6.755	8
W_F	4.25	4.625	5.375	6.125	7

Parameters;

$\alpha = .5\ (360\ /\ N)$
$a = .5\ \text{Cos}\ \alpha\ C_b$
$b = .5\ \text{Sin}\ \alpha\ C_b$
$c = .5C_{bj} - a$
$d = (b^2 + c^2)^{1/2}$
$d > .5\ (C_W + W_C)$

For first trial assume $C_{bj} = C_b + C_W$

6.0 RETAINING RING

With jack bolts;

- OD of retaining ring, A_r

  $A_r = C_{bj} + 2\ d_{bj}$

Without jack bolts;

- OD of retaining ring, A_r

  $A_r = C_b + 2\ d_b$
- ID of retaining ring, B_r

  $B_r = C_{bp} - 2d_{bp}$
- Calculate ratio K;

  $K = A_r/B_r$
- Determine value Y from Fig 8-18

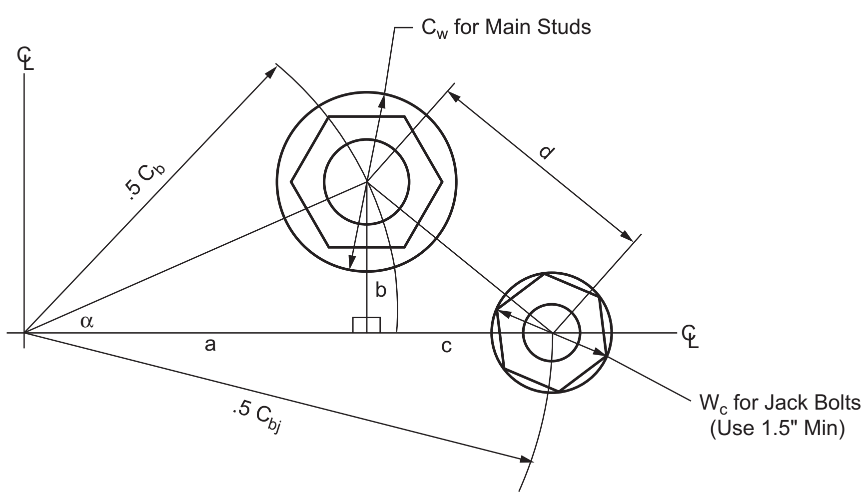


Figure 8-17. Nut-wrench clearances/proportions.

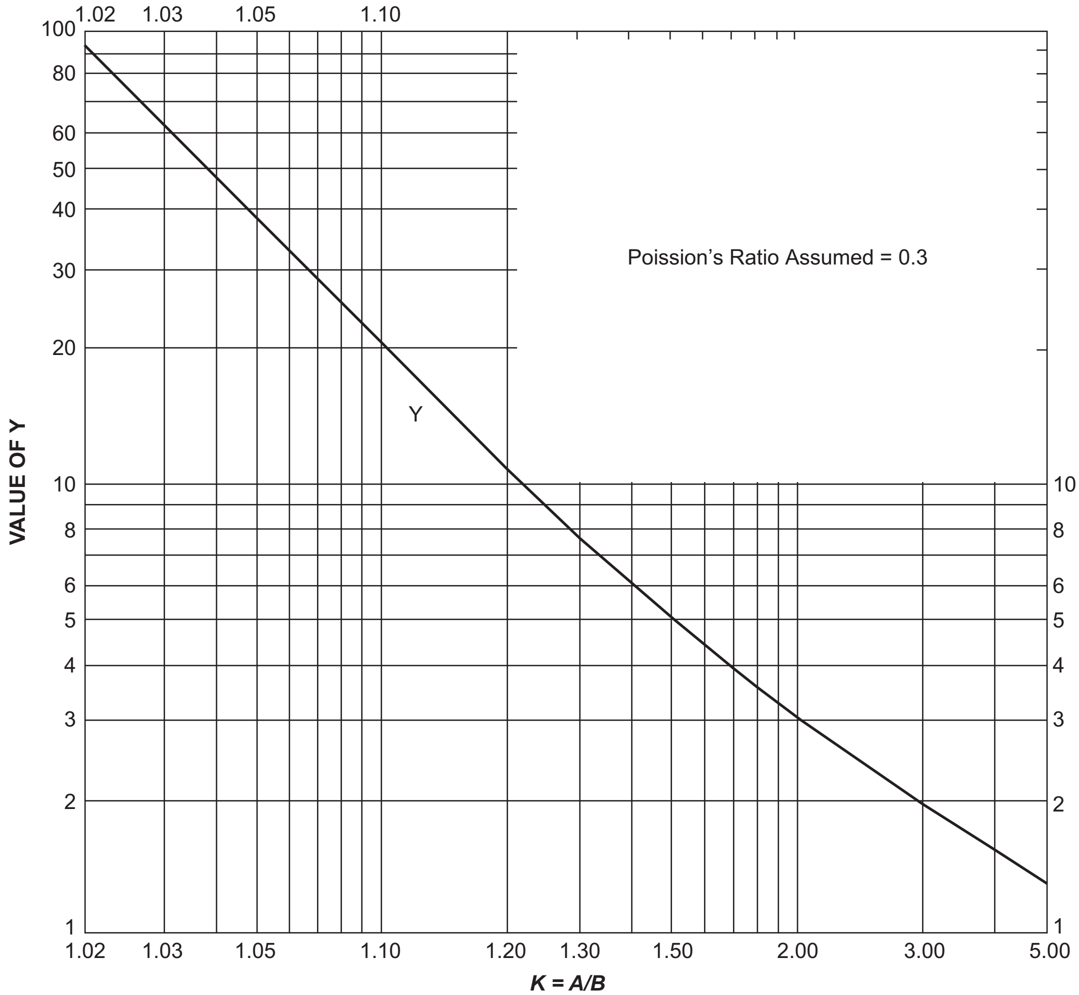


Figure 8-18. Value of Y.

- Mean diameter of follower ring, D_m

$$D_m = .5(D_O - D_i)$$

- Determine lever arm, h_G

$$h_G = .5(C_b - D_m)$$

- Bending moment in retaining ring, M_r

$$M_r = (W\, h_G)/B_r$$

- Thickness required, retaining ring, t_r

$$t_r = [(M_r\, Y)/S_r)] + \text{C.a.}$$

Use t_{rr} = _________

7.0 END PLUG

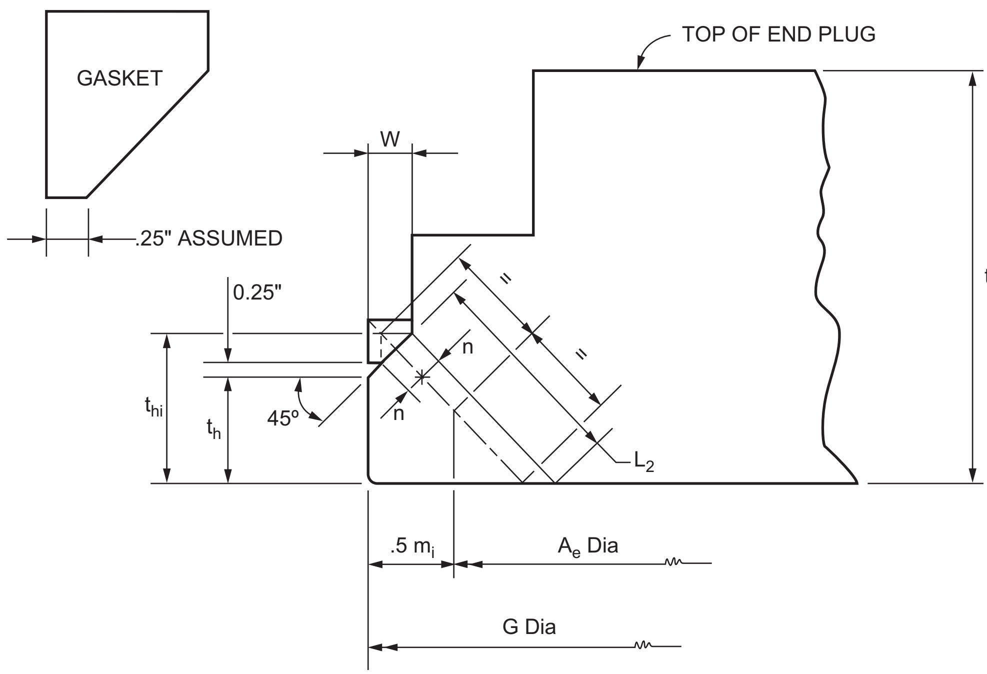


Figure 8-19. Dimensional Proportions of end plug/gasket.

DIMENSIONS;

$t_{hi} = (H + W_{ga})/[.6\,\pi\,S_e(G - 2\,w)]$

Round up to the nearest 0.25 inches.

$t_h = t_{hi} - w$
$n = .707(w - 0.25\text{ in})$
$L_2 = 1.414\,t_{hi}$
$m_i = 2(.354\,L_2 + 0.25\text{ in})$
$A_e = G - m_i$

STRESS;

- Bending stress, S_{EB}

$$S_{EB} = [6(H + W_{ga})n]/[\pi\,A_e\,L_2^2\,\text{Sin}\,45]$$

- Shear stress, S_{ES}

$$S_{ES} = (H + W_{ga})/(\pi\,A_e\,L_2\,\text{Sin}\,45)$$

- Combined stress, S_{comb}

$$S_{comb} = (S_{EB}{}^2 + S_{ES}{}^2)^{1/2}$$

- Allowable stress, S_e

$S_e =$ _________

- Required thickness, end plug, t_r

$$t_r = G\left[(C\,P/S_e)^{1/2}\right] + \text{C.a.}$$

Notes:

1. The top of the end plug should be flush with the top of the retaining ring.
2. If the opening in the end plug is greater than 2″ NPS but less than .5 G, either sufficient integral reinforcement shall be provided or the thickness may be multiplied by 1.414.

- Actual thickness used;

$t_e = t_{rr} + Z + L_1 + X$

Use _____________

8.0 FLANGE

Dimensions;

A = Larger of…

a. $C_b + 2\,(d_b - 0.125\text{ in})$
b. $B + 2\,t_f + 2\,d_b$

$a = .5\ A$
$b = .5\ B$
$r_i = .5\ (C_b - d_b)$
$r_O = .5\ (C_b + d_b)$
$J_{min} = F + .5\ (a - r_O)$
Note: The gasket should be positioned such that the centerline of the gasket corresponds to the centerline of dimension J.

$$Z = .5\,J - .5\,L_1$$

Adjust to the nearest 1/8 inch.

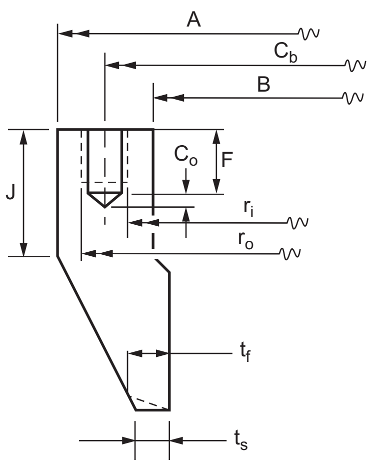


Figure 8-20. Flange dimensions.

Use worksheet for Bridgman type studding flanges to determine all stresses.

Sample Problem

1.0 GIVEN

$B = 24$ in
$R = 12$ in
$P = 3{,}000$ PSI
$DT = 860°F$
$C.a. = 0.1875$ in
$FS = 4$
$E_m = 25 \times 10^6$ PSI
$F_y = 26{,}000$ PSI

2.0 SHELL THICKNESS

$$P/S = 3000/15{,}950 = .188$$

Since this is less than .385, thin wall formulas for VIII-1 apply.

ASME VIII-1

Thin Wall formula…

$$t = (P\,R)/(SE - .6\,P)$$
$$= (3000)(12.1875)/(15950 - 1800)$$
$$= 2.58 + .1875 = 2.77 \text{ in}$$

Thick Wall formula…

$$Z_1 = (S + P)/(S - P)$$
$$= (15950 + 3000)/(15950 - 3000) = 1.46$$
$$t = R_i\left[(Z_1)^{1/2} - 1\right] = 12.1875\left(1.46^{1/2} - 1\right)$$
$$= 2.55 + .1875 = 2.743 \text{ in}$$

ASME VIII-2

$$t = R_i\left[e^{P/S} - 1\right) = 12.1875\left[e^{.188} - 1\right]$$
$$= 2.52 + .1875 = 2.71 \text{ in}$$

ASME VIII-3

Assume $t = 2.5$ new
$t = 2.5 - .1875 = 2.3125$ in
$D_O = 24 + 2\ (2.5) = 29$ in
$Y = D_O / D_i = 29 / 24.375 = 1.189$

$$P_m = .667\ F_y \text{Ln } Y = .667\ \left(26{,}000\right) \text{ Ln } 1.189$$
$$= 3011 \text{ PSI}$$

In summary….

ASME VIII–1 – Thin:	2.77
ASME VIII–1 – Thick:	2.743
ASME VIII–2:	2.71
ASME VIII–3:	2.5
Use ________	

3.0 MAIN STUDS

Matl: SA-193-B16

$$S_b = 22{,}900 \text{ PSI}$$

- Mean gasket diameter, G

$$G = B + 0.25 \text{ in}$$
$$= 24 + .25 = 24.25 \text{ in}$$

- Hydrostatic end force, H

$$H = .25\left(\pi\, G^2\, P\right) = .25\,\left(\pi\, 24.25^2\left(3000\right)\right.$$
$$= 1,385,589 \text{ Lbs}$$

- Area required, studs, A_b

$$A_b = H/S_b = 1,385,589/22900 = 60.5 \text{ in}^2$$

- Quantity of studs required, N

Always use multiples of 4, $N = A_b / R_a$

Trial	d_b	R_a	N	Use
1	2.5	3.715	16.28	20
2	2.25	3.02	20.03	24
3	2	2.3	26.3	28
4	2.25-8	3.423	17.67	20

Use Trial 4,

- Determine bolt circle, C_b

$$C_b\text{min} = B + d_b + 2 \text{ in} = 24 + 2.25 + 2$$
$$= 28.25 \text{ in}$$

- Check stud spacing, B_S

$$\text{Max}: (\pi\, C_b)\ /\ N\ <\ 2\, d_b\ +\ J\, \pi\, 28.25\ /\ 20$$
$$= 4.43 \text{ in}$$

$\text{Min}: > B_S$ from Table 8-7 $= 4.75$ in

Increase C_b to accommodate minimum bolt spacing;

$C_b = B_S\, N/\pi = 4.75\,(20)/\pi = 30.23$ in

Use 30.25 in

- Main stud bolt load, W

$$W = N\, R_a\, S_b = 20\,(3.423)\,22,900$$
$$= 1,567,734 \text{ Lbs}$$

4.0 GASKET

Matl: T-347 SST

$F_y = 30,000$ PSI

$S_g = 45,000$ PSI

$S_S = 15,000$ PSI

- Determine gasket proportions;
  a. Area of gasket required;

$$A_g = \left(2\, H\right)/\left(2S_g - F_y\right)$$
$$= \left(2\left(1,385,589\right)\right)/\left(2\left(45,000\right) - 30,000\right)$$
$$= 46.19 \text{ in}^2$$

- Pull-up stud load required to deform gasket;

$$W_g = .5\, A_g\, F_y = .5\left(46.19\right)30,000$$
$$= 692,850 \text{ Lbs}$$

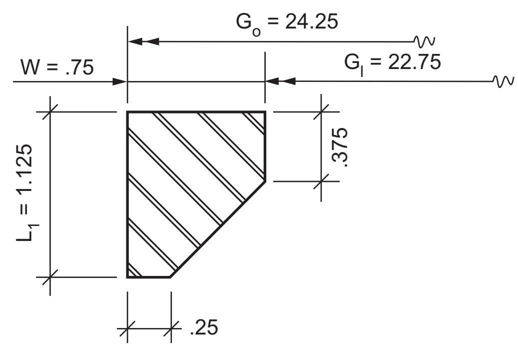
- Set gasket OD equal to G;

$G_O = G = 24.25$ in

- Determine maximum gasket ID;

$$G_i \text{ max} = \left[G_O{}^2 - \left(4\, A_g\right)/\pi\right]^{1/2}$$
$$= \left[24.25^2 - \left(4\right)\,\left(46.19\right)/\pi\right]^{1/2}$$
$$= 23.00 \text{ in}$$

Use 22.75 in



BRIDGMAN GASKET

- Based on selected gasket dimensions determine;
  a. Actual gasket area;

$$A_{ga} = .25\, \pi\, G_O{}^2 - .25\, \pi\, G_i{}^2\big) = 55.37 \text{ in}^2$$

  b. Actual gasket load;

$$W_{ga} = .5\, A_{ga}\, F_y = .5\left(55.37\right)30,000$$
$$= 830,550 \text{ Lbs}$$

$\gg W_g$ OK

5.0 PULL-UP STUDS

Matl: SA-193 B7

$S_{bp} = 25,000$ PSI

- Required stud area;

$$A_{bp} = W_{ga}/S_{bp} = 830,558/25,000 = 33.22 \text{ in}^2$$

- Determine quantity of pull-up studs;
  Note: The quantity should be the same as the number of main studs if possible.

$N = 20$

$A_r = A_{bp}/N = 33.22/20 = 1.66 \text{ in}^2$

Use : $d_{bp} = 1.625\ R_a = 1.68 \text{ in}^2$

- Determine bolt circle;

Minimum: $C_{bp} = G_I - 2(d_{bp} - 0.125 \text{ in})$

$= 22.75 - 2(1.625) = 19.5$

Maximum: $C_{bp} = B_r + 3d_{bp}$

$= 17 + 3(1.625) = 21.875$ in

Use: $C_{bp} = 20$ in

6.0 FOLLOWER RING
Ring material: SA-182–F21

$S_r = 15,950$ PSI

- Allowable bearing stress, S_{rb}

$S_{rb} = 1.6\ S_r = 1.6\ (15,950) = 25,520$ PSI

- Minimum area required at top of follower ring, A_{Fr}

$A_{Fr} = (H + W_{ga})/S_{rb}$

$= (1,385,587 + 830,558)/25,520$

$= 86.84\ \text{in}^2$

- Assume D = G = 24.25
Find D_i

$D_i = [D_O{}^2 - (4A_{Fr})/\pi]^{1/2}$

$= [24.25^2 - (4)(86.84)/\pi]^{1/2} = 21.85$ in

- Width of ring, t_o

$t_o = .5\ (D_o - D_i) = .5(24.25 - 21.85) = 1.12$ in

Therefore a double offset is required.

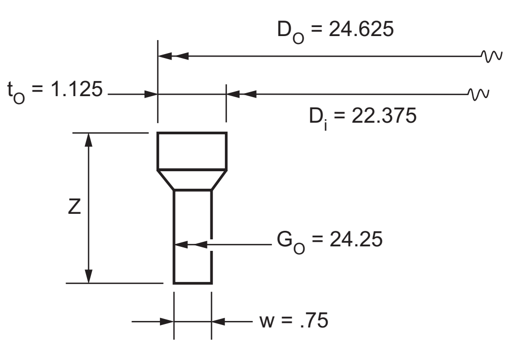
- Max offset allowed;

$.25\ w = .25\ (75) = .188$ in

- Check width of ring for double offset;

$w = t_O - 2 \times \text{offset}$

$= 1.125 - 2(.188) = 0.75$ in OK



FOLLOWER RING

- Find length of ring required, Z

$Z = 4w = 4\ (.75) = 3$ in minimum

7.0 JACKBOLTS

- Design load on jackbolts;

$W_J = .65\ S_S\ G_O\ \pi\ L_i$

$= .65\ (15,000)\ 24.25\ (\pi)\ (1.125)$

$= 835,639$ Lbs

- Allowable bearing stress, flange material;

$S_{bj} = .9F_y = .9\ (26,000) = 23,400$ PSI

- Area required, bolts;

$A_{bj} = W_J/S_{bj} = 835,639/23,400 = 35.71\ \text{in}^2$

- Actual bolting used;

$N = 20$

$d_{bj} = 1.75$ in

$R_a = 1.98\ \text{in}^2$

- Determine bolt circle required;
Given;

$C_b = 30.25$ in

$C_{WM} = 4.875$ in

$C_{WJ} = 4.125$ in

$W_{CJ} = 3.175$ in

Calculate;

$\alpha = .5(360/N) = .5\ (360/20) = 9°$

$a = 0.5\ \text{Cos}\ \alpha\ C_b = 0.5\ \text{Cos}\ 9°\ (30.25) = 14.94$ in

$b = 0.5\ \text{Sin}\ \alpha\ C_b = 0.5\ \text{Sin}\ 9°(30.25) = 2.37$ in

$X = .5(C_{WM} + C_{CJ}) = .5\ (4.875 + 3.175)$

$= 4.025$ in

Assume values for bolt circle, C_{bj}, and calculate corresponding distances "c" and "d". Select bolt circle, C_{bj}, where $d > x$

$c = .5C_{bj} - a$

$d = (b^2 + c^2)^{1/2}$

C_{bj}	c	d
36.00	3.06	3.87
36.25	3.185	3.97
36.50	3.31	4.07
36.75	3.435	4.17
37.00	3.56	4.27

Use $C_{bj} = 36.50$ in

8.0 RETAINING RING

- OD of retaining ring, A

$$A = C_{bj} + 2\ d_{bj} = 36.5 + 2(1.75) = 39.75 \text{ in Dia}$$

- ID of retaining ring, B_r

$$B_r = C_{bp} - 2\ d_{bp} = 20 - 2(1.625)$$
$$= 16.75 \text{ in Dia}$$

- Calculate ratio K;

$$K = A/B_r = 39.75/16.75 = 2.37$$

- Determine value Y from Fig 8-18

$$Y = 2.47$$

- Mean diameter of follower ring, D_m

$$D_m = .5(D_O - D_i) = .5\ (24.625 + 22.375)$$
$$= 23.5 \text{ in Dia}$$

- Determine lever arm, h_G

$$h_G = .5(C_b - D_m) = .5(30.25 - 23.5) = 3.375 \text{ in}$$

- Bolt load, W

$$W = N\ R_a\ S_b = 20(3.423)22,900$$
$$= 1,567,734 \text{ Lbs}$$

- Bending moment in retaining ring, M_r

$$M_r = (W\ h_G)/B_r = (1,567,734(3.375))/16.75$$
$$= 315,887 \text{in} - \text{Lbs}$$

- Thickness required, retaining ring, t_r

$$t_r = [\ (M_r Y)/S_r)] + \text{C.a.}$$
$$= [\ (315,887(2.47))/17,100] + .1875 = 6.94 \text{ in}$$

Use $t_{rr} = 7$ in

9.0 END PLUG

- Required thickness, end plug, t_r

$$t_r = G\ [(C\ P/S_e)]^{1/2} + \text{C.a.}$$
$$= 24.25\left[.3\left(3000\right)\big/17,100\right] + .1875 = 5.75 \text{ in}$$

Given;

$H = 1,385,589$ Lbs

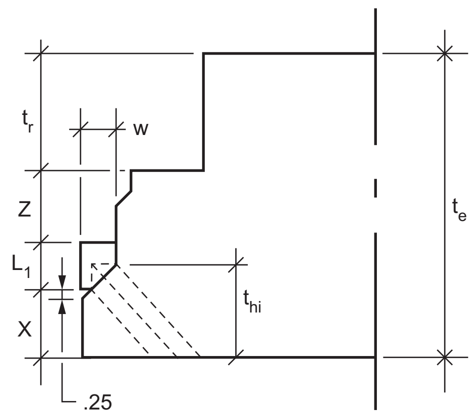
$W_{ga} = 830,558$ Lbs

$G = 24.25$ in

$w = .75$ in

- Calculate distance, t_{hi}

$$t_{hi} = (H + W_{ga})/[.6\ \pi\ S_e(G - 2w)]$$
$$= (1,385,589 + 830,558)/$$
$$\times\ (.6\ \pi\ 17,100(24.25 - 1.5))$$
$$= 3.022(\text{Round to nearest } 1/4 \text{ inch}) = 3.25 \text{ in}$$



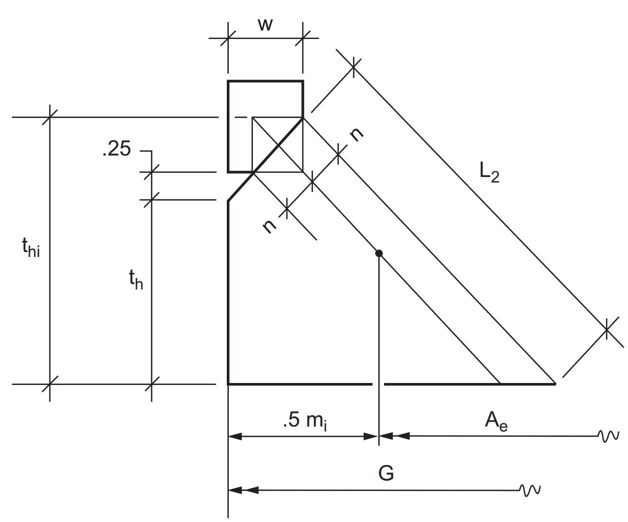
END PLUG AND GASKET

- Distance X

$$X = t - w + 0.25 = 3.25 - .75 + .25 = 2.75 \text{ in}$$

- Actual thickness, t_e

$$t_e = t_{rr} + Z + L_1 + X\ 7 + 3 + 1.125 + 2.75$$
$$= 13.875 \text{ in}$$



DIMENSIONAL PROPORTIONS

$t_h = t_{hi} - w = 3.25 - .75 = 2.5$ in

$n = .707(w - .25) = .707(.75 - .25) = .3535$ in

$L_2 = 1.414 t_{hi} = 1.414\ (3.25) = 4.596$ in

$m_i = 2(.354\ L_2 + .25) = 2(.354(4.596) + .25)$
$= 3.75$ in

$A_e = G - m_i = 24.25 - 3.75 = 20.5$ in

- Bending stress, S_{EB}

$$S_{EB} = \left[6\left(H + W_{ga}\right)n\right] / \left[\pi A_e\ L_2^2\ \text{Sin}\ 45\right]$$
$$= \left[6\left(1,385,589 + 830,558\right)3535\right] / \left[\pi(20.5)\left(4.596^2\right)\text{Sin}\ 45\right]$$
$$= 4896\ \text{PSI}$$

- Shear stress, S_{ES}

$$S_{ES} = \left(H + W_{ga}\right) / \left(\pi\ A_e\ L_2\ \text{Sin}\ 45\right)$$
$$= \left(1,385,589 + 830,558\right) / \times \left[\pi(20.5)\ \left(4.596^2\right)\ \text{Sin}\ 45\right]$$
$$= 10,588\ \text{PSI}$$

- Combined stress, S_{comb}

$$S_{comb} = \left(S_{EB}{}^2 + S_{ES}{}^2\right)^{1/2}$$
$$= \left[4896^2 + 10,588^2\right]^{1/2} = 11,661\ \text{PSI}$$

- Allowable stress, S_e

$S_e = 17,100$ PSI

10.0 FLANGE DESIGN

See worksheet

Table 8-8
Summary of materials, stress & allowable stresses - Example

	Symbol	Component	Temp	Basis/Description	Material	F_y (PSI)	Allow Stress (PSI)
1	S_a	Main Studs	Ambient	ASME Code Allowable Stress, Tension	SA-193-B16		25 KSI
2	S_b	Main Studs	Design Temp	ASME Code Allowable Stress, Tension	SA-193-B16		22.9 KSI
3	S_c	End Plug	Design Temp	Combined Stress	SA-182-F21	26 KSI	17.1 KSI
4	S_e	End Plug	Design Temp	ASME Code Allowable Stress, Tension	SA-182-F21	26 KSI	17.1 KSI
5	S_f	Flange	Design Temp	ASME Code Allowable Stress, Tension	SA-182-F21	26 KSI	17.1 KSI
6	S_g	Gasket	Design Temp		T-347		45 KSI
7	S_r	Retaining Ring	Design Temp	ASME Code Allowable Stress, Tension	SA-182-F21	26 KSI	17.1 KSI
8	S_s	Gasket	Design Temp	Shear Stress	T-347		15 KSI
9	S_{bj}	Jack Bolts	Design Temp	Bearing Stress, Use .9 F_y of Flange Material	SA-193-B7		23.4 Ksi
10	S_{bp}	Pull-Up Studs	Design Temp	ASME Code Allowable Stress, Tension	SA-193-B7		25 KSI
11	S_{rb}	Retaining Ring	Design Temp	Bearing Stress, Use 1.6 X S_r	SA-182-F21	26 KSI	27.36 KSI
12	S_{if}	Flange	Design Temp	Hoop Stress in Flange @ inner surface	SA-182-F21	26 KSI	17.1 KSI
13	S_{ri}	Flange	Design Temp	Hoop Stress in Flange @ inner surface of main stud hole	SA-182-F21	26 KSI	17.1 KSI
14	S_{ro}	Flange	Design Temp	Hoop Stress in Flange @ outer surface of main stud hole	SA-182-F21	26 KSI	17.1 KSI
15	S_{of}	Flange	Design Temp	Hoop Stress in Flange @ outer surface of Flange	SA-182-F21	26 KSI	17.1 KSI
16		Follower Ring	Design Temp		SA-182-F21	26 KSI	17.1 KSI
17	S_{EB}	End Plug	Design Temp	Bending Stress	SA-182-F21	26 KSI	17.1 KSI
18	S_{ES}	End Plug	Design Temp	Shear Stress	SA-182-F21	26 KSI	17.1 KSI

Design Temp = 860°F

WORKSHEET FOR BRIDGMAN TYPE STUDDING FLANGES EXAMPLE

GIVEN	
FLANGE OD, A OR A_1	44.25
FLANGE ID, B	24.25
BOLT CIRCLE, C_b	30.25
HYDROSTATIC END FORCE, H	1,385,589
GASKET LOAD, W_{ga}	830,558
AREA OF GASKET, A_{ga}	55.37
DIAMETER OF STUDS, d_S	2.25
HEIGHT OF METAL GASKET, L_1	1.125
PRESSURE, P	3000
THICKNESS, t	

DIMENSIONS	
$a = .5\ (A \text{ or } A_1)$	22.125
$b = .5\ B$	12.125
$r_i = .5\ (C_b - d_S)$	14
$r_O = .5\ (C_b + d_S)$	16.25
$F = L + .25"$ or $(1.5\ d_S \text{ min})$	3.375
$C_O = .288\ d_S$	0.648
$J = F + (a - r_o)/2$	6.3125
$Z = .5\ J + .5\ L_1$	3

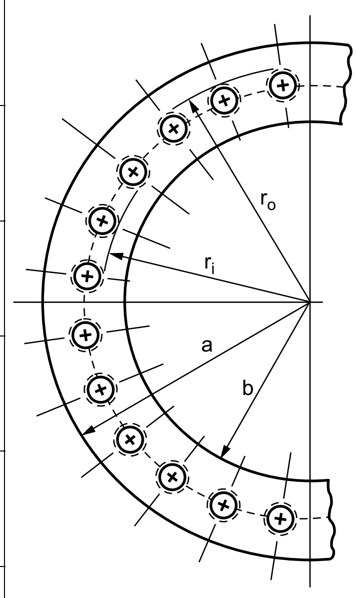
LOADS & STRESSES			
$R_1 = [(H + W_{ga})\ L_1]\ /A_{ga}$	45,027	$f_{an} = f_{at} - [(F + .5\ C_O)\ f_{a2}\ /\ J]$	86,138
$R_2 = P\ [J - (Z + L_1)]$	6,563	$Y_1 = f_{at}\ /\ f_{an}$	1.17
$R_T = R_1 + R_2$	51,590	$S'_{if} = Y_1\ S_{if}$	17,778
$P_e = R_T\ /\ J$	8,173		
$S_{if} = P_e\ [(a^2 + b^2)\ /\ (a^2 - b^2)]$	15,189		
$S_{ri} = [P_e\ b^2/(a^2 - b^2)]\ [1 + a^2\ /\ r_i^2]$	12,269		
$S_{ro} = [P_e\ b^2/(a^2 - b^2)]\ [1 + a^2\ /\ r_o^2]$	10,012		
$S_{of} = (2\ P_e\ b^2)\ /\ (a^2 - b^2)$	7,016		
$f_{a1} = .5\ (r_i - b)\ (S_{if} + S_{ri})$	25,741		
$f_{a2} = .5\ (r_o - r_i)\ (S_{ri} + S_{ro})$	25,066		
$f_{a3} = .5\ (a - r_o)\ (S_{ro} + S_{of})$	50,020		
$f_{aT} = f_{a1} + f_{a2} + f_{a3}$	100,827		

NOTES:

1. If $S'_{if} \leq$ Allowable Stress, then the design is OK
2. If $S'_{if} >$ Allowable Stress, then implement a or b below

a. Increase flange proportions

b. Add a shrink ring



Plan View-Dimensions

3.8. Threaded Closures

Threaded end closure is the nomenclature used to describe configurations where the end plug is secured in place with a threaded main nut. The main nut is threaded into the body of the vessel or end flange as applicable. In other styles of end closures, the end plug is secured with a retainer ring that is connected to the body of the vessel or end flange with studs. The studs are threaded into the end of the shell or flange. To open a threaded closure the entire end plug must be unscrewed out of the body of the vessel using the main nut.

From a design standpoint, the total hydrostatic end force, H, is taken by the threads in a threaded closure, or by the studs in other designs. Various types of gaskets can be used with either design but all are the self-sealing type such as Bridgman, delta or double cone.

Extremely tight fits of the threads should be avoided, and the pitch of the male and female threads should be the same within very close limits. Threads should be generously lubricated with graphite paste for lower temperatures or moly-sulfide for higher temperatures. Coarse threads do not show as much tendency to gall as fine threads. A difference in hardness of mating parts will help to prevent seizure.

Notation

A_m = Area between OD and pitch diameter, in^2
A_P = Equivalent chamber area, in^2
C_f = Body flexibility factor
C_T = Thread flexibility factor
D_i = Inside diameter, in
D_L = Loading diameter, in
D_o = Outside diameter, in
D_r = Diameter at root of thread, in
e = Tooth thickness, in
F = End load per inch at pitch diameter, Lbs/in
f_b = Bending stress, PSI
F_{avg} = Average load on threads, Lbs
F_i = Load on any given thread, Lbs
F_{sum} = Cumulative load on threads, Lbs
F_y = Minimum specified yield stress, PSI
h = Length from root of last thread to end of vessel, in
h_t = Height of thread tooth, in
K_n, K_T, K_f = Stress concentration factors
M_X = Longitudinal bending moment, in-Lbs
n = Number of threads
N_R = Neutral radius, in
P = Internal design pressure, PSI
P_t = Thread pitch, in
R_L = Loading radius, in
R_o = radius, outside, in
R_r = Radius at root of thread, in
S_{bL} = Combined bending and longitudinal stress at root of thread, PSI
S_T = Combined stress intensity at root of thread, PSI
S_{tt} = Thread tooth stress intensity, PSI
σ_C = Combined stress, PSI
σ_X = Average longitudinal stress, PSI
σ_{tt} = Thread tooth bending stress, PSI
ν = Poissons ratio
β = Damping factor

Design Procedure

1.0. Stress in Shell

- Average longitudinal stress across vessel wall, σ_X

$$\sigma_x = \left(D_i^2\, P\right)/\left(D_o^2 - D_r^2\right)$$

- Damping factor, β

$$B = 1.285/(R_r\, t)^{1/2}$$

- Load diameter, D_L

$$D_L = .5(D_r + D_i)$$

$$R_L = .5\, D_L$$

- Neutral radius, N_R

$$N_R = \left[.5\left(R_o^2 + R_r^2\right)\right]^{1/2}$$

- End load, F, Lbs/in at pitch diameter

$$F = \left(D_i^2 P\right)/\left(4 D_L\right)$$

- Longitudinal bending moment, M_X

$$X = \left[\frac{1}{1 + \frac{\beta h}{2} + \frac{1 - v^2}{2\beta_r \beta}\left(\frac{h}{t}\right)^3 \ln\left(\frac{R_o}{R_r}\right)}\right]$$

$$M_X = F(N_R - R_L)X$$

- Bending stress, f_b

$$f_b = (6M_X)/t^2$$

- Combined bending and longitudinal stress at root of thread, S_{bL}

$$S_{bL} = K_T(\sigma_X + f_b)$$

2.0. Thread Load Distribution

- Area between OD and pitch diameter, A_m

$$A_m = .25\ \pi\left(D_o{}^2 - D_L{}^2\right)$$

- Equivalent chamber area at pitch diameter, A_P

$$A_P = .25\ \pi\ D_L{}^2$$

- Body flexibility factor, C_f

$$C_f = p_t\left(1/A_m + 1/A_P\right)$$

- Thread flexibility factor, C_t

$$C_t = 2/D_L$$

- Calculate ratio C_f / C_t

$$C_f/C_t =$$

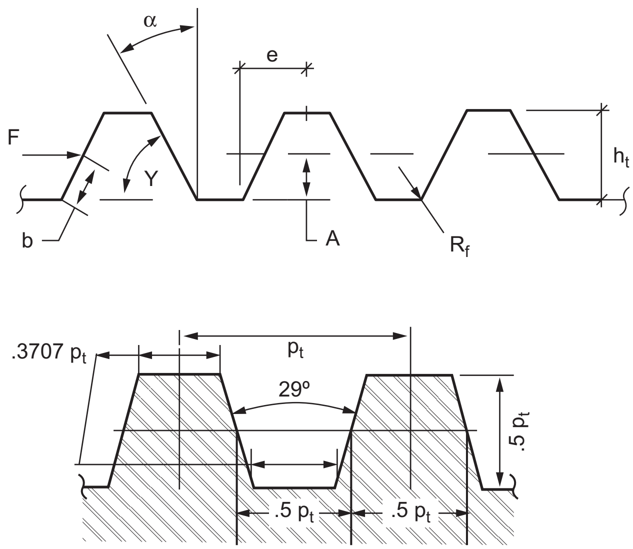
And complete Table below….

Thread Load Distribution			
Thread	**F_i**	**F_{sum}**	**%**
1			
2			
3			
4			
5			
6			
7			
8			
Σ			100

Where;

$$F_i = F_{i-1} + C_f/C_t\left(F_{sum}\right)$$

And % = F_i / F_{sum} (Innermost thread)



ACME THREADS - DIMENSIONS

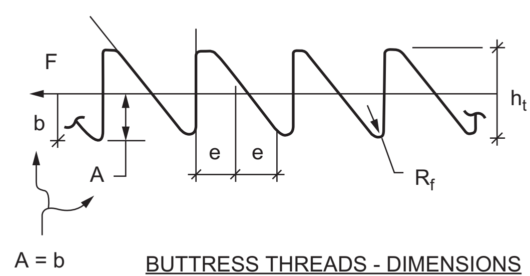
$\gamma = 90° - \alpha$

$$h_t = .5\ p_t$$

$$A = .25\ p_t$$

$$e = (.5\ p_t\ \mathrm{Tan}\ \alpha) + .1853\ p_t$$

$$b = A/\cos\ \alpha$$



BUTTRESS THREADS - DIMENSIONS

$\gamma = 90°$

3.0. Thread Root Tooth Bending Stress

- Thread tooth bending stress, σ_{tt}

$$\sigma_{tt} = \frac{F}{n}\left[\frac{1.5A}{e^2} + \frac{\cos\gamma}{2e} + \frac{.45}{\sqrt{b\ e}}\right]$$

- Stress concentration factor, K_n

$$K_n = \left[1 + .26\ (e/Rf)^{.7}\right]$$

- Highest load distribution factor, K_f

$$K_f = n\left(\%\ Max/100\right)$$

- Thread tooth intensity, S_{tt}

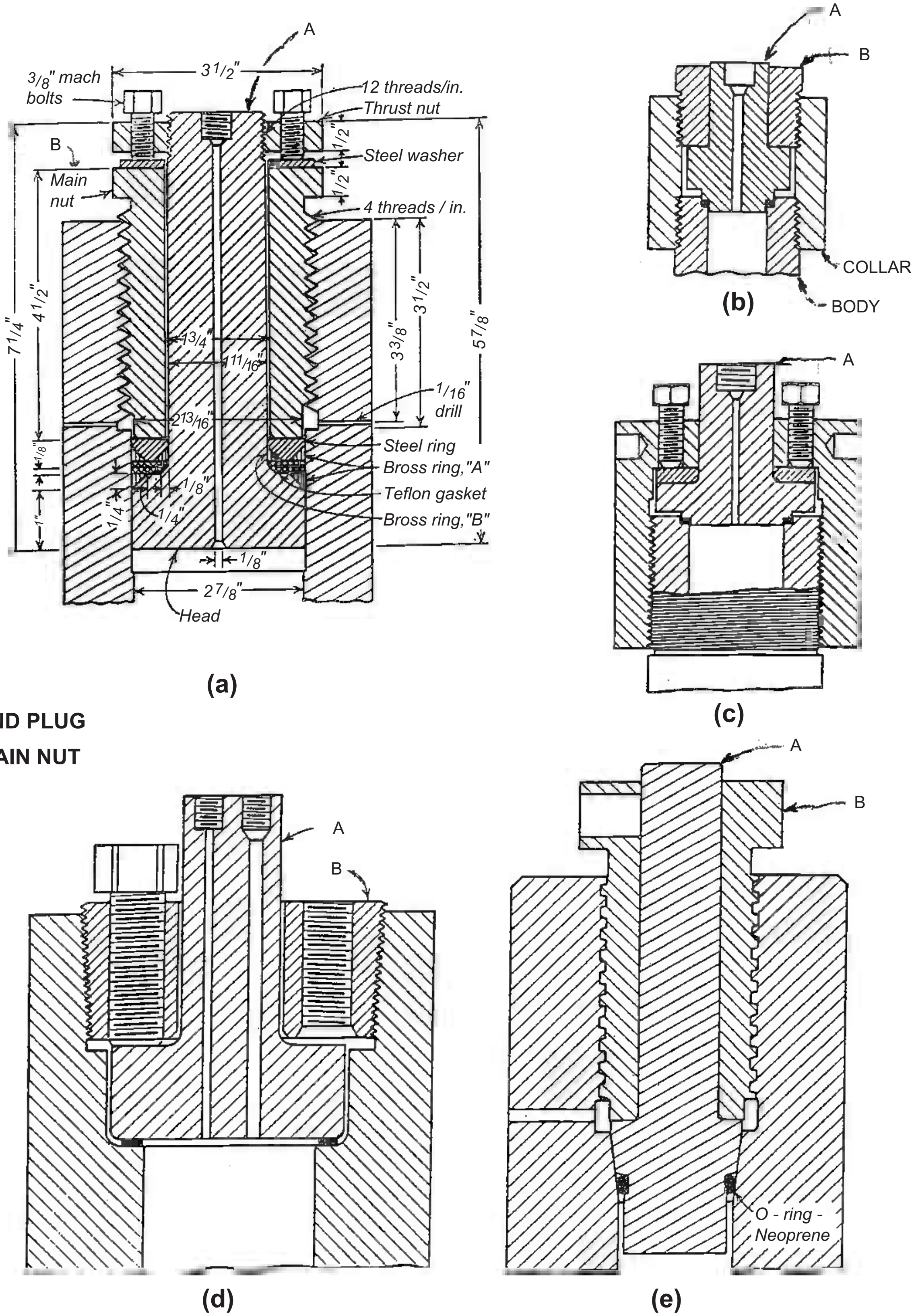
$$S_{tt} = K_f\ K_T\ \sigma_{tt}$$

- Combined stress, S_T

$$S_T = S_{bL} + S_{tt}$$

Notes:

1. Threads are not uniformly loaded
2. The threads closest to the inside (innermost) are the most highly loaded.
3. The maximum load is typically about 2 to 4 times the "average stress load"
4. The practical limit on the number of engaged threads is 10. Threads beyond this will carry negligible load.
5. The innermost threads will carry approximately 20% of the load and the first 4 threads will carry approximately 50%. Conversely, the outermost threads carry only approximately 3 % of the total load.
6. Total stress for fatigue evaluation should be based on the most highly stressed region of the vessel. If a threaded closure is utilized, this worst stress case is at the root of the most highly loaded thread. This stress should be combined with bending and longitudinal shell stresses at the same location.
7. Load distribution depends on a number of factors such as;
   a. Form of threads
   b. Thickness of walls supporting the threads
   c. Pitch of threads
   d. Number of engaged threads
   e. Boundary conditions
8. Although the last threads carry very little load, their contribution in increasing height, h, is significant in reducing the bending moment in the shell carrying the threads.



THREADED CLOSURES

(a) Modified Bridgman Closure using rings to prevent extrusion of gasket (b) Screwed sleeve end plug (c) Compression Head (d) Tapered Cap –Ring Closure (e) O-ring Closure

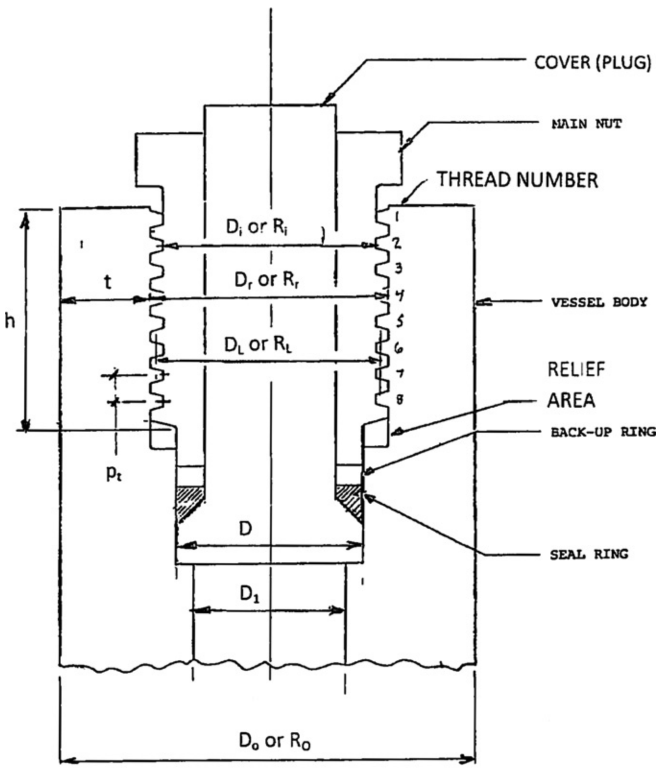


Figure 8-21. Bridgman closure, threaded with ACME threads.

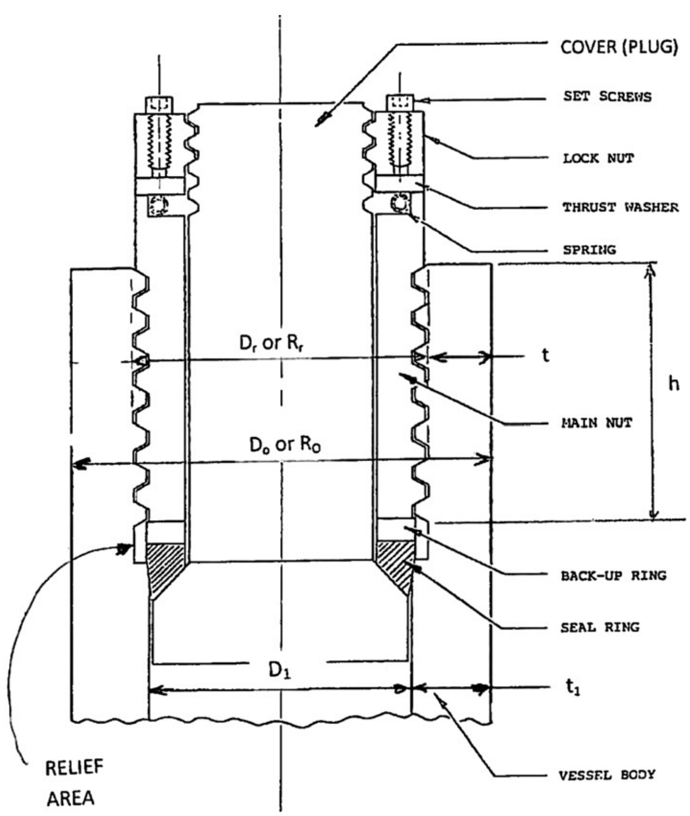


Figure 8-22. Modified Bridgman closure, threaded, with ACME threads.

Threaded Closure With Bridgman Gasket

EXAMPLE No. 1

1.0 GIVEN

$D_i = 17$ in

$D_O = 36$ in

$D_r = 18$ in

$F_y = 54$ KSI

$h = 12$ in

$K_T = 1.25$

$n = 12$

$P = 25{,}000$ PSI

$p_T = 1.00$ in

$R_f = 0.125$ in

$R_o = 18$ in

$R_r = 9$ in

$t = R_o - R_r = 9$ in

$\alpha = 14.5^o$

$\gamma = 90 - 14.5 = 75.5^o$

$\nu = .3$

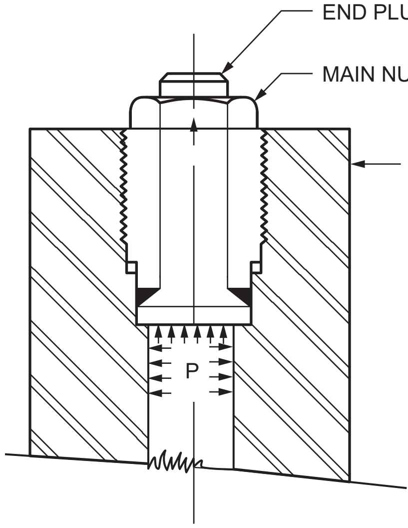
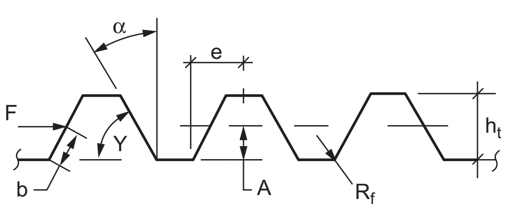


Figure 8-23. Simple threaded type closure with a Bridgman closure.



ACME THREADS - DIMENSIONS

$h_t = .5\ p_t = .5\ (1) = 0.5$ in

$A = .25\ p_t = .25\ (1) = 0.25$ in

$e = (.5\ p_t \operatorname{Tan} \alpha) + .1853\ p_t = 0.3146$ in

$b = A / \operatorname{Cos} \alpha = .25 / \operatorname{Cos} 14.5^o = 0.258$ in

2.0 STRESS IN SHELL

- Average longitudinal stress across vessel wall, σ_X

$$\sigma_X = \left(D_i^2\ P\right)/\left(D_o{}^2 - D_r{}^2\right)$$
$$= \left[17^2\ (25{,}000)\right]/\left[36^2 - 18^2\right] = 7{,}433 \text{ PSI}$$

- Damping factor, β

$$B = 1.285/(\ R_r\ t)^{1/2} = 1.285/(9\ (9))^{1/2} = .143$$

- Load diameter, D_L

$$D_L = .5\ (D_r + D_i) = .5\ (18 + 17) = 17.5 \text{ in}$$
$$R_L = .5\ D_L = .5(17.5) = 8.75 \text{ in}$$

- Neutral radius, N_R

$$N_R = \left[.5\left(R_o{}^2 + R_r{}^2\right)\right]^{1/2} = \left[.5\left(18^2 + 9^2\right)\right]^{1/2}$$
$$= 14.23 \text{ in}$$

- End load, F, Lbs/in at pitch diameter

$$F = \left(D_i^2\ P\right)/\left(4\ D_L\right) = \left(17^2(25{,}000)\right)/\left(4(17.5)\right)$$
$$= 103{,}214 \text{ Lbs/in}$$

- Longitudinal bending moment, M_X

$$X = \left[\frac{1}{1 + \frac{\beta h}{2} + \frac{1 - v^2}{2R_f\beta}\left(\frac{h}{1}\right)^3 \ln\left(\frac{R_o}{R_1}\right)}\right]$$

$$\left[\frac{1}{1 + \frac{.143(12)}{2} + \frac{91}{2(9).143}\left(\frac{12}{9}\right)^3 \ln\left(\frac{18}{9}\right)}\right]$$

$$\left[\frac{1}{1 + .858 + .353(2.37)\log 2}\right] = .474$$

$$M_X = F(N_R - R_L)\,X$$
$$= 103,214\ (14.23 - 8.75)\ .474$$
$$= 268,000 \text{ in-Lbs}$$

- Bending stress, f_b

$$f_b = (6\ M_X)/t^2 = (6\ (268,100))/9^2$$
$$= 19,860 \text{ PSI}$$

- Combined bending and longitudinal stress at root of thread, S_{bL}

$$S_{bL} = K_T(\sigma_X + f_b) = 1.25\ (7433 + 19,859)$$
$$= 34,115 \text{ PSI}$$

3.0 Thread Load Distribution

- Area between OD and pitch diameter, A_m

$$A_m = .25\ \pi\ (D_o{}^2 - D_L{}^2) = .25\ \pi\ (36^2 - 17.5^2)$$
$$= 777.34 \text{ in}^2$$

- Equivalent chamber area at pitch diameter, A_P

$$A_P = .25\ \pi\ D_L{}^2 = .25\ \pi\ (17.5^2) = 240.52 \text{ in}^2$$

- Body flexibility factor, C_f

$$C_f = p_t(1/A_m + 1/A_P)$$
$$= 1\ (1/777.34 + 1/240.52) = .00544$$

- Thread flexibility factor, C_t

$$C_t = 2/D_L = 2/17.5 = .11429$$

- Ratio C_f / C_t

$$C_f/C_t = .005/.114 = .0476$$

Thread Load Distribution			
Thread	**F_i**	**F_{sum}**	**%**
1	1.00	1.00	3.24
2	1.0476	2.0476	3.39
3	1.1451	3.1927	3.71
4	1.2971	4.4898	4.2
5	1.5108	6.00	4.89
6	1.7964	7.7964	5.81
7	2.1675	9.9639	7.01
8	2.6418	12.6057	8.55
9	3.2418	15.8475	10.49
10	3.9961	19.8436	12.93
11	4.9407	24.7843	15.99
12	6.1204	30.9047	19.8
Σ	30.9053	NA	100

Where;

$$F_i = F_{i-1} + C_f/C_t(F_{sum})$$

And % = F_i / F_{sum} (Innermost thread)

- For the example shown, F_n =

Thread	Equation	Result
F_2	F_1 +.0476 (1.00)	1.0476
F_3	F_2 +.0476 (2.0476)	1.1451
F_4	F_3 +.0476 (3.1927)	1.2971
F_5	F_4 +.0476 (4.4898)	1.5108

- For example shown, %

Thread	Equation	Result
1	1.00 / 30.9053	.0324
2	1.0476 / 30.9053	.0339
3	1.1451/ 30.9053	.0371
4	1.2971 / 30.9053	.0420

4.0 THREAD ROOT TOOTH BENDING STRESS

- Thread tooth bending stress, σ_{tt}

$$\sigma_{tt} = \frac{F}{n}\left[\frac{1.5\ A}{e^2} + \frac{\cos\gamma}{2e} + \frac{.45}{\sqrt{be}}\right]$$
$$= 103,214/12(3.789 + .3979 + 1.579)$$
$$= 49,593 \text{ PSI}$$

- Stress concentration factor, K_n

$$K_n = \left[1 + .26\ (e/R_f)^{.7}\right]$$
$$= \left[1 + .26(.3146/.125)^{.7}\right] = 1.496$$

- Highest load distribution factor, K_f

$$K_f = n(\%\,\text{Max}/100) = 19.8/100 \times 12 = 2.376$$

- Thread tooth intensity, S_{tt}

$$S_{tt} = K_f\ K_T\ \sigma_{tt} = 2.376(1.496)\ (49,593)$$
$$= 176,278$$

- Combined stress, S_T

$$S_T = S_{bL} + S_{tt}$$
$$= 33,054 + 176,278 = 209,332 \text{ PSI}$$
$$< 4\ F_y = 216,000 \text{ PSI}$$

3.9. Studs and Nuts

General. As pressures get higher and higher, the studs and nuts that secure the end closures get larger and larger. This results in a corresponding increase in the bolt circle to accommodate the increased number and size of studs given the spacing of the studs required. As the bolt circle increases, the flange gets thicker to resist the bending moment. Eventually the designer reaches the point of diminishing returns.

The typical bolt spacing for standard flange design is based on wrench clearances. And the wrench clearances end up governing the design.

One of the principles of good flange design is to reduce the bolt circle to a minimum. This will minimize the size of all the components and result in the most efficient design.

Designers faced with this dilemma in the past have come up with special nuts for high pressure applications. They are called sleeve nuts.

Sleeve nuts can be made much smaller in diameter than nuts for the same diameter. In addition the spacing between studs will not be based on wrench clearances, but on whatever tightening device is utilized.

The spacing of sleeve type nuts should be 1.5 times the stud diameter, as opposed to 2 to 2-1/2 times the stud diameter for conventional bolting.

Studs. Studs used for high pressure applications are integral to the overall design, the performance of the joint, and functioning of the vessel. For these reasons, it is imperative that every design detail receive the appropriate attention.

Typically the threaded shank of the stud creates a stress concentration. Numerous tests have shown that this is the most likely area of failure. To preclude this, and minimize stress concentrations, a generous radius is machined into the shank of the stud at the location of the first thread.

This is critical for cyclic service or high temperature operation. Any time high temperature or fatigue are involved, the studs should have a generous radius machined at the root of the first thread.

Sleeve Nuts. Sleeve type nuts are utilized whenever normal spacing of nuts based on wrench clearances cannot be met or is not desirable.

Sleeve nuts are a smooth sleeve of metal that is internally threaded. The nuts may have a nut machined to the top of the cylinder to accommodate hand or wrench tightening. However, they are not designed for manual tightening.

Most high pressure applications do not utilize wrench tightening anyway. So why base the bolt spacing based on wrench clearances?

In this case, the nut is not there to create the stud tension, but only used to secure the elongation of the stud. Sleeve nuts achieve this goal with a minimum of metal. Sleeved nuts may have holes drilled in them to facilitate the turning of the nut during stud elongation.

Sleeve nuts have longer threads to develop full strength as opposed to nuts. Sleeve nuts should have a minimum of 10 threads to ensure full strength.

Washers. Washers also become instrumental in good joint design for critical applications. Ordinary hardened washers will transmit any eccentric loading due to unlevelness of the bearing surface. Any out of line loading will cause a bending moment in the stud. In order to negate the effects of an unlevel or non-perpendicular surface, spherical washers are used.

We are all aware of the compound effect that bending will have when combined with tension. However, in the design of the bolting, perfect alignment and pure tension are assumed. Our designs are based on the stud being stressed perfectly in tension without any bending. Without spherical washers, this may not be the case.

Typically spot-facing is done to preclude this condition but may not be 100% effective.

The eccentric surface can be caused by several conditions;

a. The back of the flange is not spotfaced
b. The spotfacing was not done properly
c. The studs are not mounted precisely perpendicular to the flange face.

Spherical washers are used in pairs and carefully machined to nest together. In this manner, whatever misalignment occurs, the spherical washers automatically account for the misalignment and keep the stud in pure tension.

Stud Pretightening Methods

a. Mechanical pre-tightening
b. Thermal pre-tightening
c. Hydraulic pre-tightening

Achieving the optimum tightening

For high temperature connections, there are concerns with over-tightening. Over-tightening can cause the bolts to exceed yield after heat up, thus reducing the gasket load. The required torque range should be between the following limits;

a. Above the value to seat the gasket at ambient condition
b. Below the value where the bolts will be overstressed due to high temperature

Mechanical Pretightening. Mechanical pretightening is accomplished with wrenches. This method may result in torquing the stud rather than tightening the nut. The accuracy of torque is based on the lead angle of the threads and the frictional characteristics of the overall assembly. These coefficients are variable and depend on a number of factors.

Because of all the factors involved, this method is not considered appropriate for large, critical flange joints.

Thermal Pretightening. In this method, the required elongation of the bolt is achieved by thermal expansion of the heated stud. It is essential that the temperature required does not alter the mechanical properties of the stud.

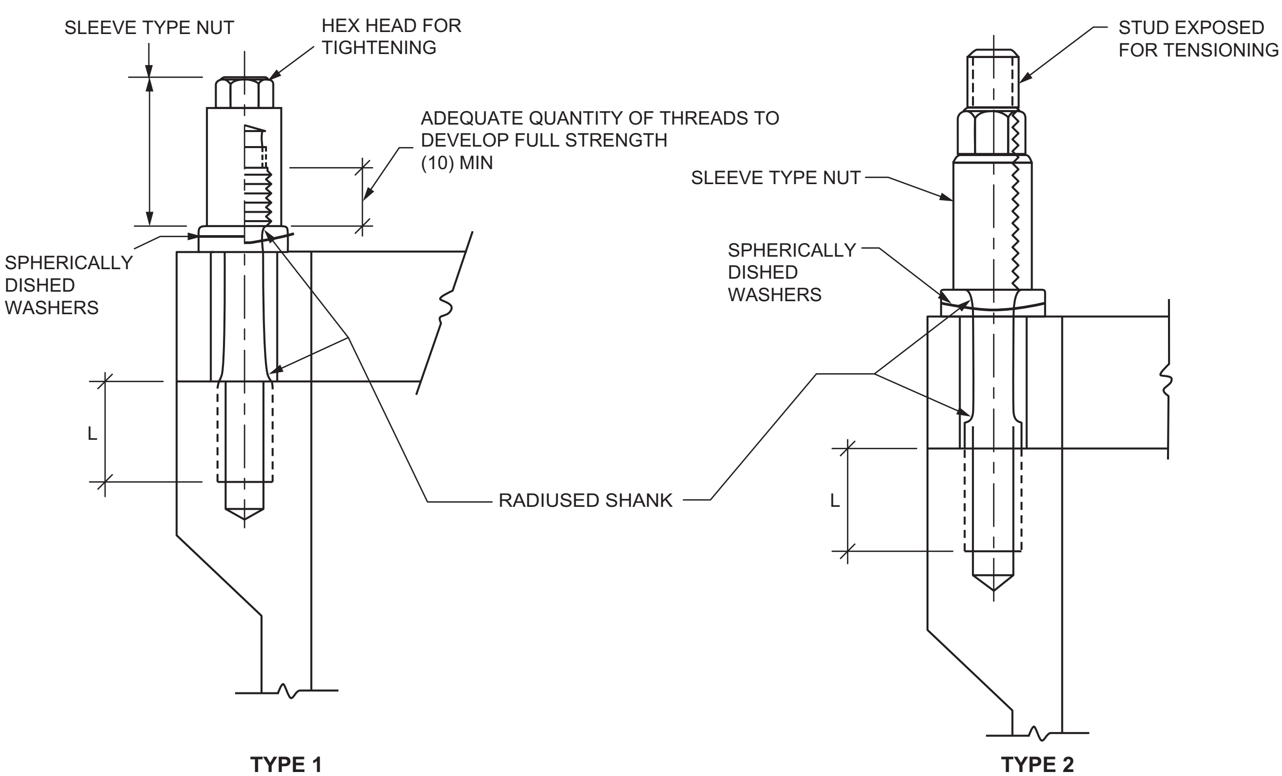
Thermal pretightening is achieved by putting electric resistance wires down a pilot hole in the stud. As the stud is heated, it elongates. Since the thermal coefficient and length are known, it is a simple task to determine the temperature necessary to accomplish a precise elongation.

Hydraulic Pretightening. In this case, the elongation of the stud is accomplished by means of a hydraulic jack. The stud is elongated, and then the nut is hand tightened to secure the degree of elongation desired.

Usually four jacks are used simultaneously so that the joint experiences uniform pressure. All four jacks are rotated in turn to the next set of studs.

Tightening is normally done in stages, typically in three increments of 1/3 total elongation required.

SLEEVE TYPE NUTS



NOMENCLATURE

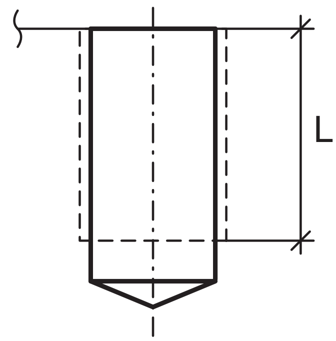
F_{ys} = Yield strength, stud, at design temperature, PSI

F_{yf} = Yield strength, flange, at design temperature, PSI

d_s = Root diameter of stud, in

R_a = Root area, in^2

S_b = Average primary stress in stud, PSI



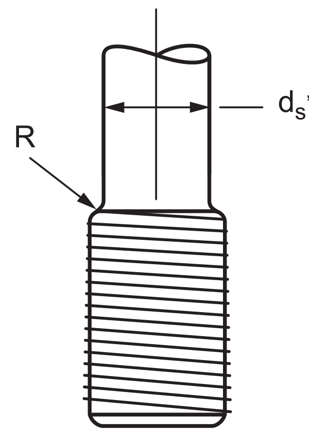
TYPICAL TAPPED HOLE

CALCULATIONS

- Maximum stress in stud, S_b
  $S_b = F_{ys} / 1.8$

- Maximum stress in reduced Shank stud, S_b
  $S_b = F_{ys} / 1.5$

- Minimum length of thread Engagement, L
  $L > .75\ d_s\ (F_{ys} / F_{yf}) > d_s$



REDUCED SHANK STUD
$d_s' < .9\ d_s$

REF: ASME VIII-3, ARTICLE KD-6

Figure 8-24. Tapped hole and stud requirements.

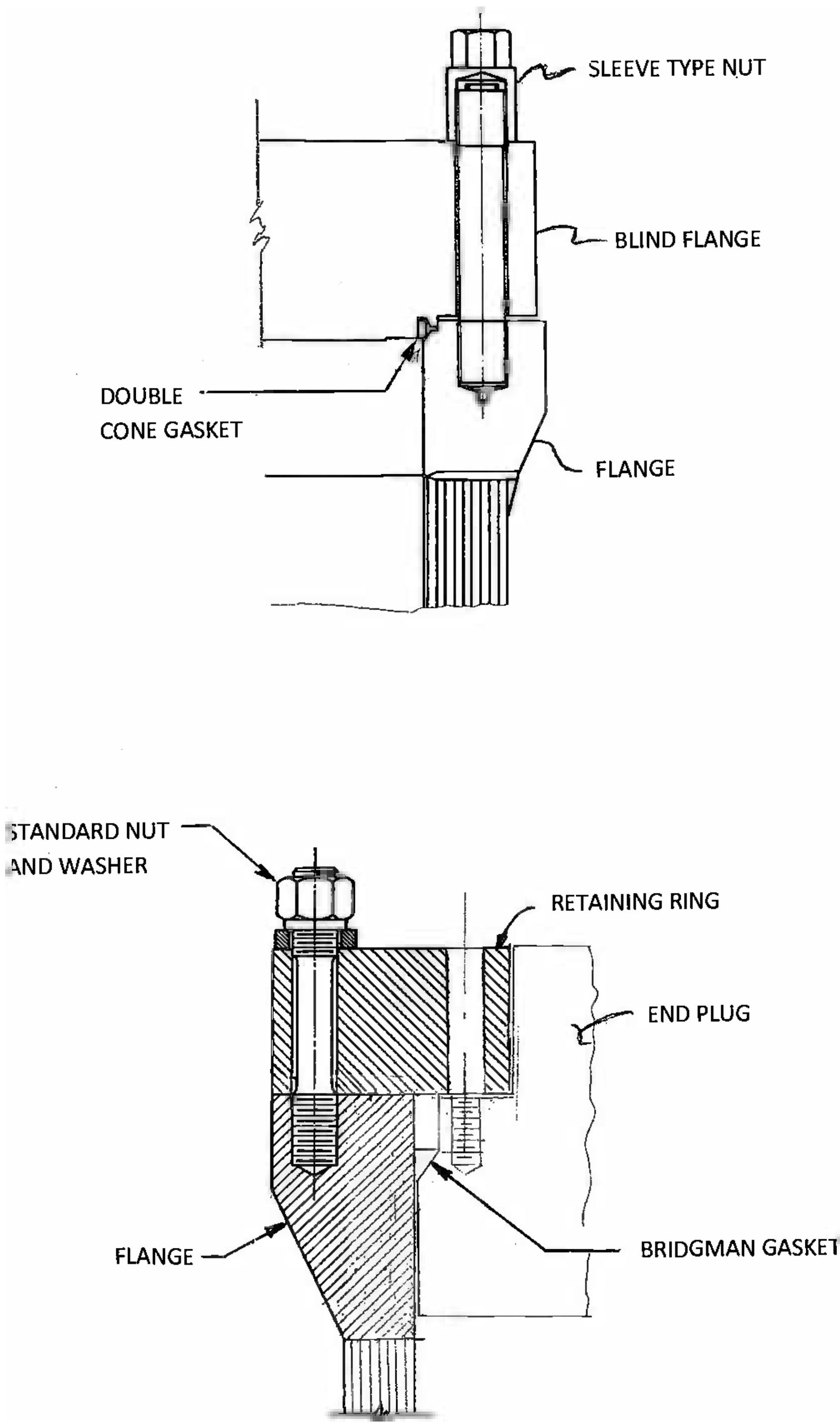


Figure 8-25. Examples of studded flanges.

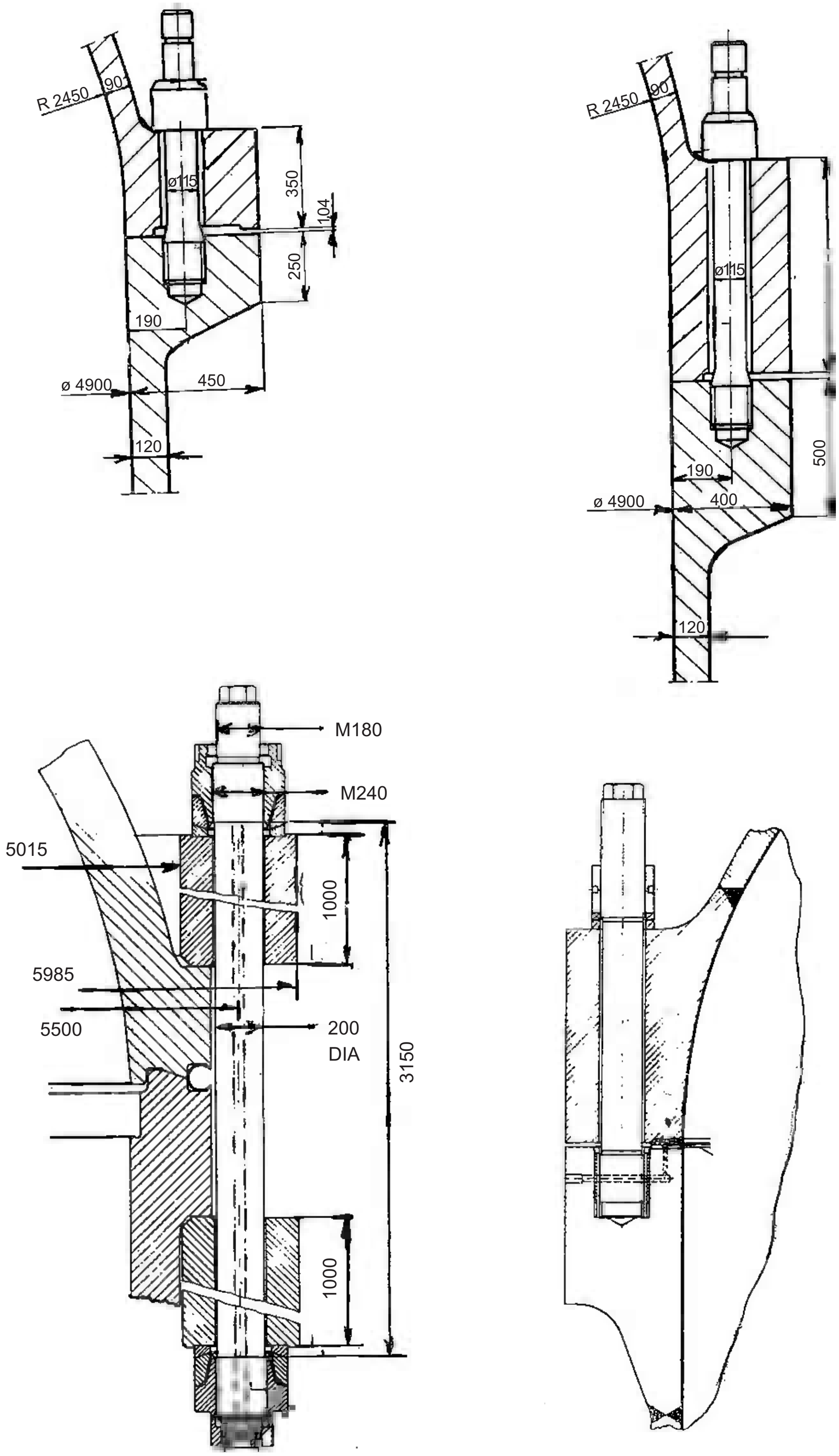


Figure 8-26. Examples of studded flanges & head closures.

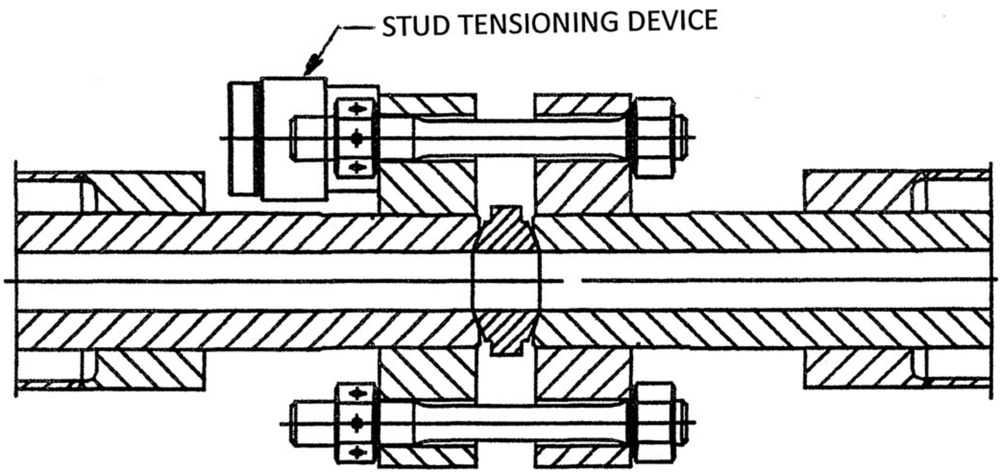


Figure 8-27. Piping sub-assy showing, tapered studs, lens ring gasket and clearances for stud tensioning equipment.

4.0. Nozzles

Nozzle attachments to high pressure vessels are almost always made through the heads, or end closures, rather than through the shell as is done in common pressure vessels. This is primarily due to the thickness of the shells and the fact that multilayered vessels are common in high pressure systems. Nozzles through multilayer vessels, while not impossible, are discouraged.

Openings through the end closures are typically bolted in place with the flange facing machined onto the outside face of the closure. End closures can be very thick (2-3 feet thick) so welding through the head thickness is not an option. Nozzles can be of the set on type but welding can be difficult or create problems such as lamellar tearing. Bolted on connections typically have tapped holes in the closure for attaching the external flanges. Any facing configuration can be machined into the closure.

Nozzle penetrations through the end closures can also be screwed into tapped connections in the head. These type of connections should be avoided if the vessel is in cyclic service.

Nozzles are attached by one of the following methods;

1. Screwed
2. Bolted
3. Welded
4. Socket Welded

Of these, the first two are the most common. Screwed connections can easily be machined into the closure and are strong enough for applications up to about 60,000 PSI. Flanged connections to the end closure are also used up to about 20,000 PSI. The problem with flanged connections at high pressure is the bending moments created by the extreme end force as well as the large bolting forces and local discontinuities.

Due to the undesirability of leaks in high pressure systems, connections are deliberately kept to small sizes, probably less than 4" NPS. Most connections are smaller than that. Screwed connections up to 1¼ inch may be used up to 20,000 PSI and smaller connections up to 60,000 PSI.

It is not uncommon to machine flats for branch connections to main nozzles on a distribution forging. A distribution forging can be designed to accommodate multiple nozzles. In this manner a heavier forged nozzle neck can be machined to accommodate the size and shape of the attached connection. These attachments should be provided by the vessel fabricator and the connections tested with the vessel during the standard shop hydrotest.

High pressure connections should avoid gasket configurations which are not contained. Ring type joint, male-female, and tongue and groove should be utilized wherever possible.

Nozzle reinforcement:

Nozzle reinforcement is not an issue for smaller nozzles through the end plug or closure. However when the size of the opening exceeds ½ of the nominal vessel diameter, special analysis is required. Utilize standard ASME Code reinforcement rules for all other openings. All nozzles shall be integrally reinforced. No reinforcing pads are allowed.

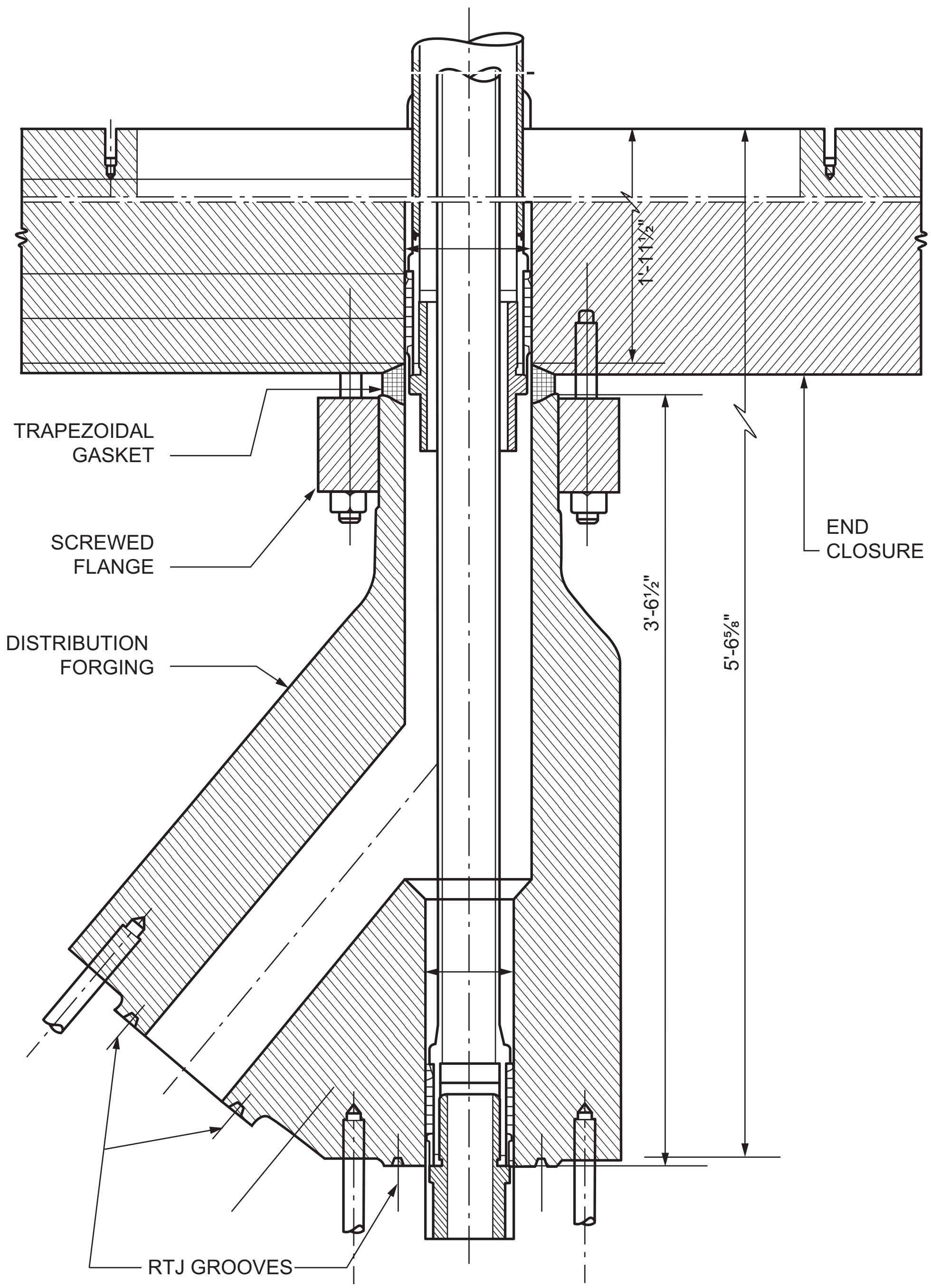
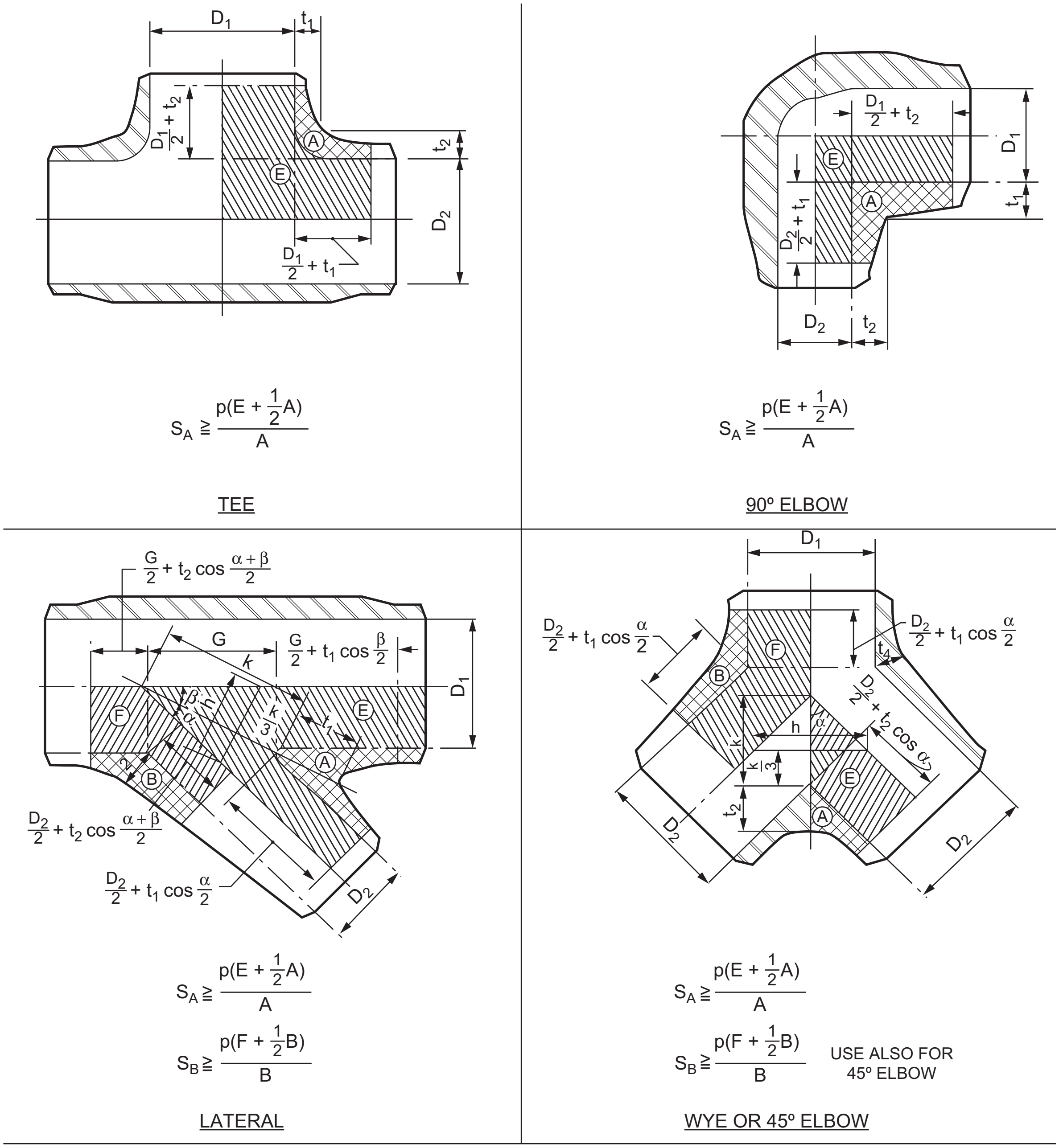


Figure 8-28. Example of a distribution forging.



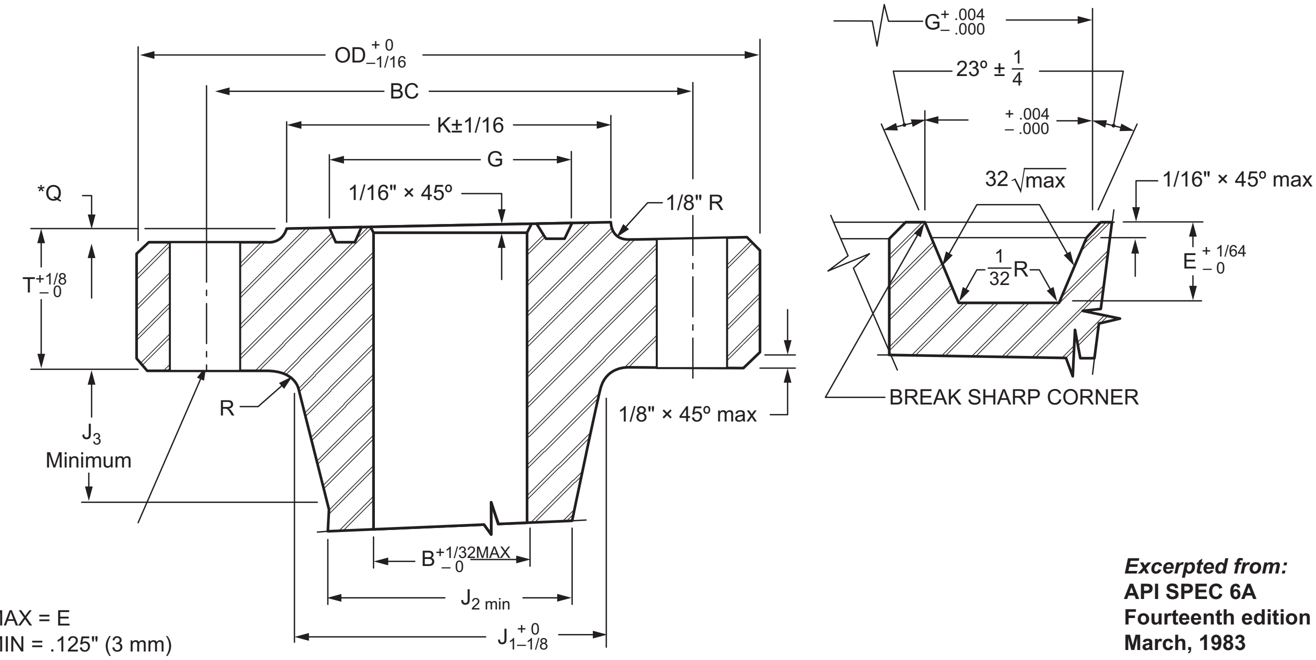
NOMENCLATURE

- A, B - METAL AREA, (SQ.IN.)
- D_1, D_2 - INSIDE DIAMETER OF FITTINGS, (IN.)
- E, F - INDICATED PRESSURE AREA, (SQ.IN.)
- G,h,k - INDICATED LENGTHS, (IN)
- p - DESIGN PRESSURE, AT DESIGN TEMPERATURE, (PSIG)
- S_A, S_B - ALLOWABLE STRESS AT DESIGN TEMPERATURE, (PSI)
- t_1, t_2 - INDICATED METAL THICKNESS, (IN.)
- t_3 - AVERAGE METAL,THICKNESS OF FLAT SURFACE, (IN.)
- α, β - INDICATED ANGLES.

Figure 8-29. Special heavy wall fittings: check of reinforcement for internal pressure.

High Pressure Flanges

5,000 to 20,000 PSI



Q MAX = E
Q MIN = .125" (3 mm)

Excerpted from:
API SPEC 6A
Fourteenth edition
March, 1983

API Type 6BX High Pressure Flanges

NOM SIZE	B	OD	T	J_1	J_2	J_3	R	BC	N	d_b	d_{bh}	L_s	K	G	W	E	Ring No.
5000 PSI																	
1-1/2"	1.69	7	1.5					4.875	4	1	1.12	5.5					20
2"	2.06	8.5	1.813					6.5	8	0.875	1	6					24
2-1/2"	2.56	9.63	1.938					7.5	8	1	1.12	6.5					27
3"	3.12	10.5	2.188					8	8	1.125	1.25	7.25					35
4"	4.06	12.25	2.438					9.5	8	1.25	1.38	6					39
5"	5.12	14.75	3.188					11.5	8	1.5	1.62	10					44
6"	7.06	15.5	3.63					12.5	12	1.375	1.5	10.75					46
8"	9	19	4.06					15.5	12	1.625	1.75	12					50
10"	11	23	4.688					19	12	1.875	2	13.75					54
13"	13.63	26.5	4.438	18.938	16.68	4.5	0.625	23.25	16	1.625	1.75	12.5	18	16.063	0.786	563	
16"	16.38	30.38	5.5	21.875	20.75	3	0.75	28.63	16	1.875	2	14.5	21.06	18.832	0.705	0.328	
18"	18.75	35.63	6.53	26.56	23.56	6	0.625	31.63	20	2	2.12	17.5	24.68	22.185	1.006	0.718	
21"	21.25	39	7.12	29.875	26.75	6.5	0.68	34.88	24	2	2.12	18.75	27.63	24.904	1.071	0.75	
10,000 PSI																	
1-1/2"	1.69	7.188	1.656	3.31	2.4	1.85	0.375	5.56	8	0.75	0.88	5	4	2.893	0.45	0.218	BX-150
2"	2.06	7.875	1.73	3.94	2.938	2.03	3.75	6.25	8	0.75	0.88	5.25	4.375	3.395	0.498	0.234	BX-152
2-1/2"	2.56	9.125	2.01	4.75	3.625	2.25	0.375	7.25	8	0.875	1	6	5.188	4.046	0.554	0.265	BX-153
3"	3.12	10.625	2.21	5.59	4.34	2.5	0.375	8.5	8	1	1.125	6.75	6	4.685	0.606	0.297	BX-154
4"	4.06	12.438	2.76	7.188	5.75	2.875	0.375	10.188	8	1.125	1.25	8	7.25	5.93	0.698	0.328	BX-155
5"	5.12	14.063	3.12	881	7.188	3.188	0.375	11.813	12	1.125	1.25	8.75	8.7	6.955	0.666	0.375	BX-169
6"	7.06	18.875	4.06	11.938	10	3.75	0.625	15.875	12	1.5	1.625	11.25	11.875	9.521	0.921	0.438	BX-156
8"	9	21.75	4.875	14.75	12.875	3.688	0.625	18.75	16	1.5	1.625	13	14.125	11.774	1.039	0.5	BX-157
10"	11	25.75	5.563	17.75	15.75	4.06	0.625	22.25	16	1.75	1.875	15	16.438	14.064	1.149	0.563	BX-158
13"	13.63	30.25	6.63	21.75	19.5	4.5	0.625	26.5	20	1.875	2	17.25	20.38	17.033	1.279	0.625	BX-159
16"	16.38	34.312	6.63	25.813	23.813	3	0.75	30.563	24	1.875	2	17.5	22.688	18.832	0.705	0.328	BX-162
18"	18.75	40.938	8.78	29.63	26.583	6.125	0.625	36.438	24	2.25	2.375	22.5	27.438	22.752	1.29	0.719	BX-164
21"	21.25	45	9.5	33.38	30	6.5	0.613	40.25	24	2.5	2.625	24.5	30.75	25.507	1.373	0.75	BX-166
15,000 PSI																	
1-1/2"	1.69	7.625	1.75	3.688	2.688	1.875	0.375	6	8	0.75	0.875	5.25	3.813	2.893	0.45	0.218	BX-150
2"	2.06	8.188	1.78	3.84	2.813	1.875	0.375	6.313	8	0.875	1	5.5	4.188	3.062	0.466	0.218	BX-151
2-1/2"	2.56	8.75	2	4.375	3.25	2.125	0.375	6.875	8	0.875	1	6	4.5	3.395	0.498	0.234	BX-152
3"	3.12	10	2.25	5.0625	3.94	2.25	0.375	7.875	8	1	1.125	6.75	5.25	4.046	0.554	0.266	BX-153
4"	4.06	11.313	2.53	6.063	4.813	2.5	0.375	9.06	8	1.125	1.25	7.5	6.06	4.685	0.606	0.297	BX-154
5"	5.12	14.188	3.09	7.69	6.25	2.875	0.375	11.438	8	1.375	1.5	9.25	7.625	5.93	0.698	0.328	BX-155
6"	7.06	19.875	0.468	12.81	10.875	3.625	0.625	16.875	16	1.5	1.625	12.75	12	9.521	0.921	0.438	BX-156
8"	9	25.5	5.75	17	13.75	4.875	0.625	21.75	16	1.875	2	15.75	15	11.774	1.039	0.5	BX-157
10"	11	32	7.375	23	16.81	9.29	0.625	28	20	2	2.125	19.25	17.875	14.064	1.149	0.563	BX-158
13"	13.63	34.875	8.06	23.438	20.813	4.5	1	30.375	20	2.25	2.375	21.25	21.313	17.033	1.279	0.625	BX-159
18"	18.75	45.75	10.06	32	28.75	6.125	1	40	20	3	3.125	26.75	28.438	22.752	1.29	0.719	BX-164
20,000 PSI																	
1-1/2"	1.813	10.125	2.5	5.25	4.313	1.938	0.375	8	8	1	1.125	7.5	4.625	3.062	0.466	0.218	BX-151
2"	2.06	11.313	2.813	6.063	5	2.063	0.375	9.063	8	1.125	1.25	8.25	5.188	3.395	0.498	0.234	BX-152
2-1/2"	2.56	12.813	3.125	6.813	5.688	2.3125	0.375	10.313	8	1.25	1.375	9.25	5.938	4.046	0.554	0.266	BX-153
3"	3.12	14.063	3.375	7.563	6.313	2.5	0.375	11.313	8	1.375	1.5	10	6.75	4.685	0.606	0.297	BX-154
4"	4.06	17.563	4.1875	9.563	8.125	2.875	0.375	14.063	8	1.75	1.625	12.25	8.625	5.93	0.698	0.328	BX-155
7"	7.06	25.813	6.5	15.188	13.313	3.813	625	21.813	16	2	2.125	17.5	13.875	9.521	0.921	0.438	BX-156
8"	9	31.688	8.06	18.938	196.88	4.25	1	27	16	2.5	2.625	22.375	17.375	11.774	1.039	0.5	BX-157
10"	11	34.75	8.813	22.313	20	4.063	1	29.5	16	2.75	2.875	23.75	19.875	14.064	1.149	0.563	BX-158
13"	13.63	45.75	11.5	27.313	24.75	5.25	1	40	20	3	3.125	30	24.188	17.003	1.279	0.625	BX-159

5.0. References

1. Fryer DM, Harvey JF. High Pressure Vessels. Chapman and Hall; 1998.
2. Comings EW. High Pressure Technology. McGraw Hill Book Co; 1956.
3. Brownell LE, Young EH. Process Equipment Design. John Wiley & Sons; 1959. Chapters 14 & 15.
4. Crum ASD. The distribution of load over threads of a threaded closure, including temperature effects and local imperfections. High Pressure Technology, Material Design Stress Analysis and Applications, ASME PVP-Vol. 148; 1988.
5. Mraz GJ. Development of design criteria for a high pressure vessel construction code. ASME Journal of Pressure Vessel Technology 1987;Vol. 109:256–9.
6. Boyer CB. The development of a high pressure idea. High Pressure Technology, Codes, Analysis and Applications, PVP-Vol. 263; 1993.
7. Coopey W. Holding High Pressure Joints. Petroleum Refiner May, 1956.
8. Antalffy LP, Chaku PN. The New Division 3 High Pressure Vessel Code. High Pressure Technology, ASME 1997.
9. Kendall DP. A short history of high pressure technology from Bridgman to Division 3. Journal of Pressure Vessel technology August, 2000;Vol. 122. ASME.
10. Ludwig EE. How to specify high pressure vessels. Petroleum Refiner October, 1956;Vol. 35(No 10).
11. Faupel JH. Proven pressure vessel design methods. Petroleum Refiner Sept, 1959;Vol. 38(No. 9).
12. Meyer R. Sky high process pressure demand massive vessels. The Refining Engineer January, 1957.
13. Kalnins A, Updike DP. Limit pressures of cylindrical and spherical shells. Journal of Pressure Vessel Technology, Transactions of the ASME August, 2001; Vol. 123.
14. Motoihaga H, Tahara T. Manufacturing monobloc ultra high pressure vessels. High Pressure Technology, PVP-Vol. 344, ASME; 1997.
15. Korner P, Vardag T. A new design method for ultra high pressure vessels. High Pressure Technology, PVP-Vol. 297, ASME; 1995.
16. Barake N, Chaaban A. Effect of cracks on load distribution in threaded end closures and fracture detection. High Pressure Technology, PVP-Vol. 281, ASME; 1994.
17. Chaaban A, Muzzo U. On the design of threaded-end closures for high pressure vessels. High Pressure – Codes, Analysis and Applications, PVP-Vol. 263, ASME; 1993.
18. Grewal AS, Sabbaghian M. Load distribution between threads in threaded connections. Journal of Pressure Vessel Technology February, 1997;Vol. 119.
19. Jawad MH, Farr JR. Structural Analysis & Design of Process Equipment. Second Edition. John Wiley & Sons; 1989. Chapter 15.

9
Related Equipment

Procedure 9-1: Design of Davits [1,2]

Notation

C_v = vertical impact factor, 1.5–1.8
C_h = horizontal impact factor, 0.2–0.5
f_a = axial stress, psi
f_b = bending stress, psi
f_h = horizontal force, lb
f_v = vertical force, lb
F_a = allowable axial stress, psi
F_b = allowable bending stress, psi
F_r = radial load, lb
F_{re} = equivalent radial load, lb
F_y = minimum specified yield stress, psi
M_1 = bending moment in mast at top guide or support, in.-lb
M_2 = maximum bending moment in curved davit, in.-lb
M_3 = bending moment in boom, in.-lb
M_x = longitudinal moment, in.-lb
M_ϕ = circumferential moment, in.-lb
W_1 = weight of boom and brace, lb
W_D = total weight of davit, lb
W_L = maximum rated capacity, lb
α,β,K = stress coefficients
P = axial load, lb
I = moment of inertia, in.^{4}
A = cross-sectional area, in.^{2}
Z = section modulus, in.^{3}
r = least radius of gyration, in.
t_p = wall thickness of pipe davit, in.
a = outside radius of pipe, in.

Moments and Forces in Davit and Vessel

• *Loads on davit.*

$$f_v = C_V W_L$$

$$f_h = C_h W_L$$

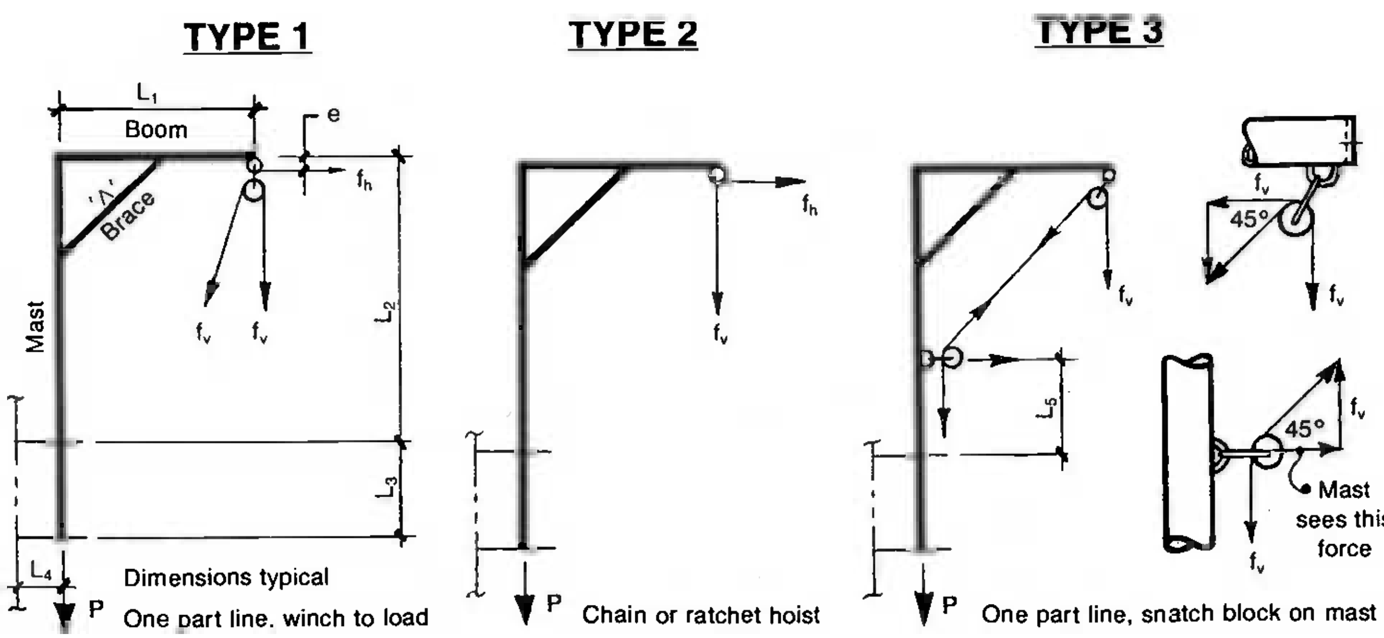


Figure 9-1. Types of rigging.

- *Bending moment in davit mast, M_1.*

Type 1: $M_1 = 2f_vL_1 + 0.5W_1L_1 + f_hL_2$

Type 2: $M_1 = f_vL_1 + 0.5W_1L_1 + f_hL_2$

Type 3: $M_1 = f_v(2L_1 + L_5 - L_2) + 0.5W_1L_1 + f_hL_2$

- *Radial force at guide and support, F_r.*

$$F_r = \frac{M_1}{L_3}$$

F_r is maximum when davit rotation ϕ is at 0°, for other rotations:

$$F_r = F_r \cos\phi$$

- *Circumferential moment at guide and support, M_ϕ.*

$$M_\phi = F_rL_4$$

M_ϕ is maximum when davit rotation ϕ is at 90°, for other rotations:

$$M_\phi = F_rL_4 \sin\phi$$

- *Axial load on davit mast, P.*

Type 1 or 3: $P = 2f_v + W_D$

Type 2: $P = f_v + W_D$

- *Longitudinal moment at support, M_x.*

$$M_x = PL_4$$

Stresses in Davit

Mast Properties

$I =$

$A =$

$Z =$

$r =$

$t_p =$

$a =$

Slenderness ratio:

$$\frac{2.1L_2}{r} =$$

$F_a =$

$F_b = 0.6F_y$

Type A Davit

- *Axial stress—mast.*

$$f_a = \frac{P}{A}$$

- *Bending stress—mast.*

$$f_b = \frac{M_1}{Z}$$

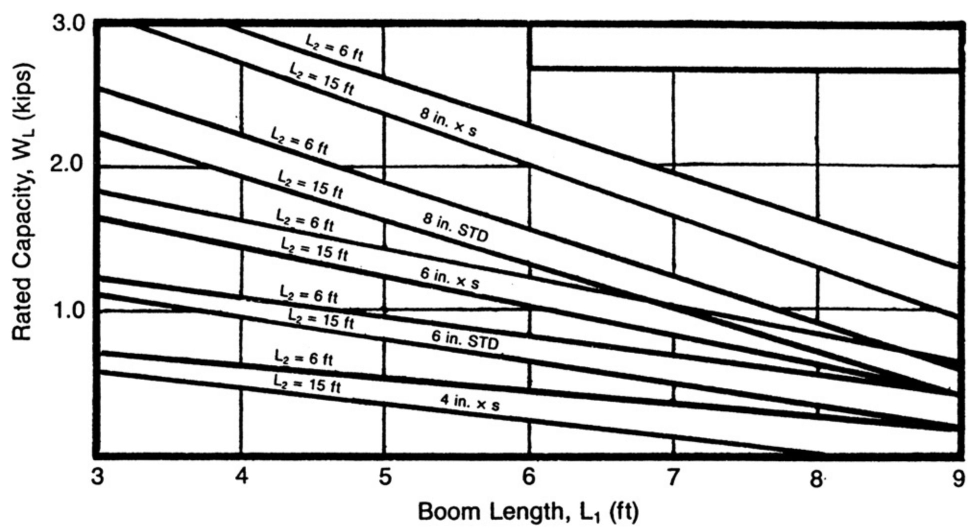


Figure 9-2. Davit selection guide.

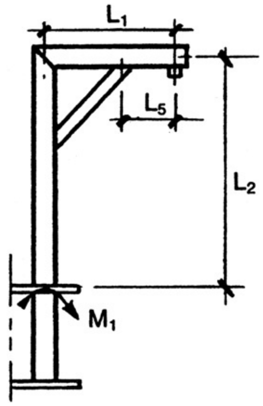


Figure 9-3. Type A davit.

- *Combined stress—mast.*

$$\frac{f_a}{F_a} + \frac{f_b}{F_b} \le 1$$

- *Bending stress—boom.*

$$\text{Type 1: } f_b = \frac{2 f_v L_5}{Z}$$

$$\text{Type 2 or 3: } f_b = \frac{f_v L_5}{Z}$$

Type B Davit

- *Axial stress.*

$$f_a = \frac{P}{A}$$

- *Bending moment, M_2.*

$$M_2 = \frac{M_1(L_2 - R)}{L_2}$$

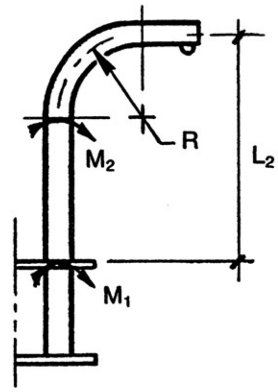
- *Bending stress.*

$$\text{At } M_1, f_b = \frac{M_1}{Z}$$

$$\text{At } M_2, f_b = \frac{M_2 a}{I}\left(\frac{2}{3K\sqrt{3\beta}}\right)$$

- *Combined stress.*

$$\frac{f_a}{F_a} + \frac{f_b}{F_b} \le 1$$



Coefficients

$$\alpha = \frac{t_p R}{a^2}$$

$$\beta = \frac{6}{5 + 6\alpha^2}$$

$$K = 1 - \frac{9}{10 + 12\alpha^2}$$

Figure 9-4. Type B davit.

Finding Equivalent Radial Load, F_{re}

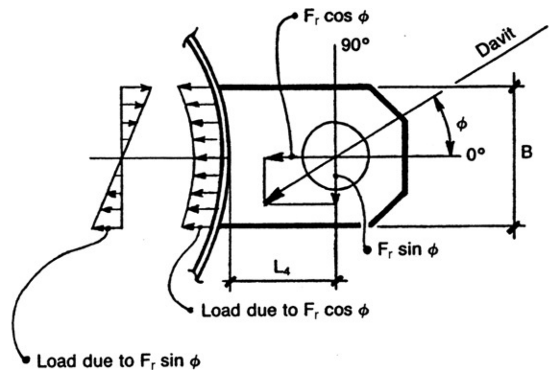


Figure 9-5. Forces in davit guide.

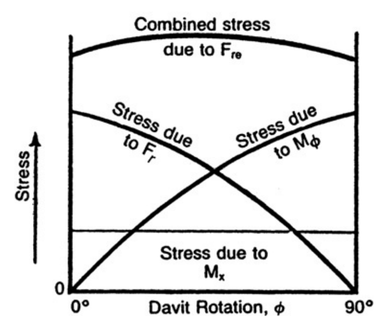


Figure 9-6. Graph of combined stress for various davit rotations.

- *Equivalent unit load, w, lb/in.*

$$w = \frac{F_r \cos\phi}{B} + \frac{6F_r \sin\phi \ L_4}{B^2}$$

- *Equivalent radial load, F_{re}, lb.*

$$F_{re} = \frac{wB}{2}$$

- *Calculate F_{re} for various angles of davit rotation.*

ϕ	W	F_{re}

Shell Stresses (See Note 1)

At Support: Utilizing the area of loading as illustrated in Figure 9-8, find shell stresses due to loads M_x, M_ϕ, and F_r by an appropriate local load procedure.

At Guide: Utilizing the area of loading as illustrated in Figure 9-9, find shell stresses due to loads M_ϕ and F_r by an appropriate local load procedure.

Note: F_{re} may be substituted for M_ϕ and F_r as an equivalent radial load for any rotation of davit other than 0° or 90°.

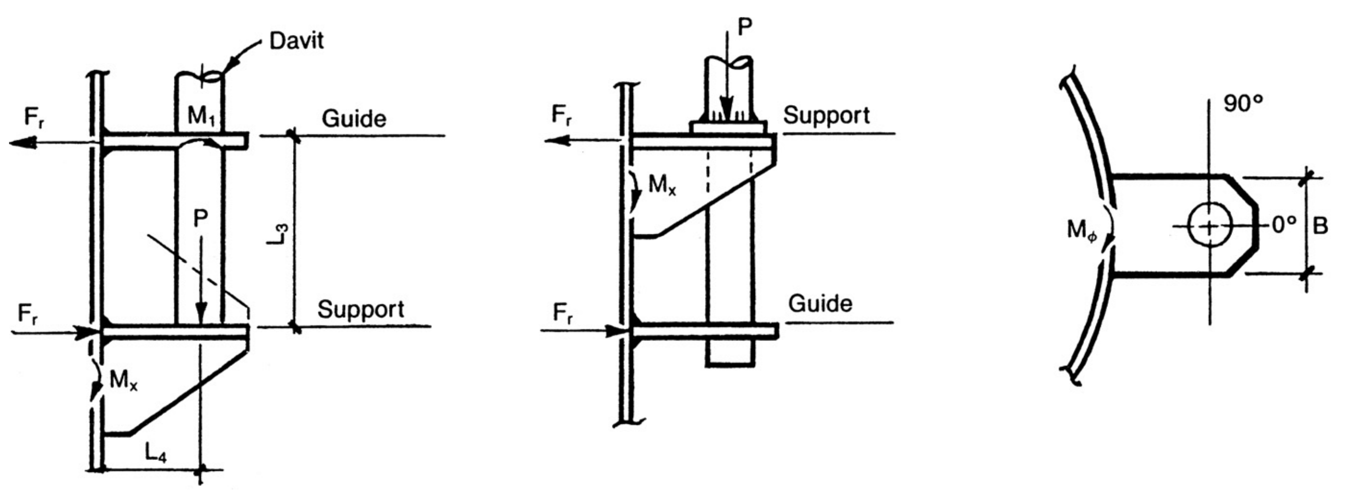


Figure 9-7. Dimensions of forces at davit support and guide.

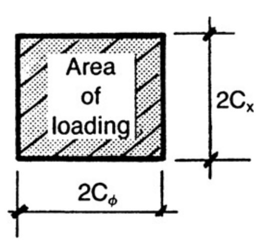


Figure 9-8. Area of loading at davit support.

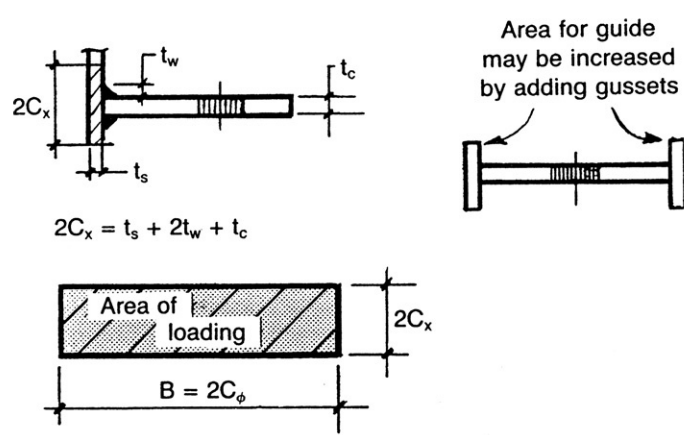
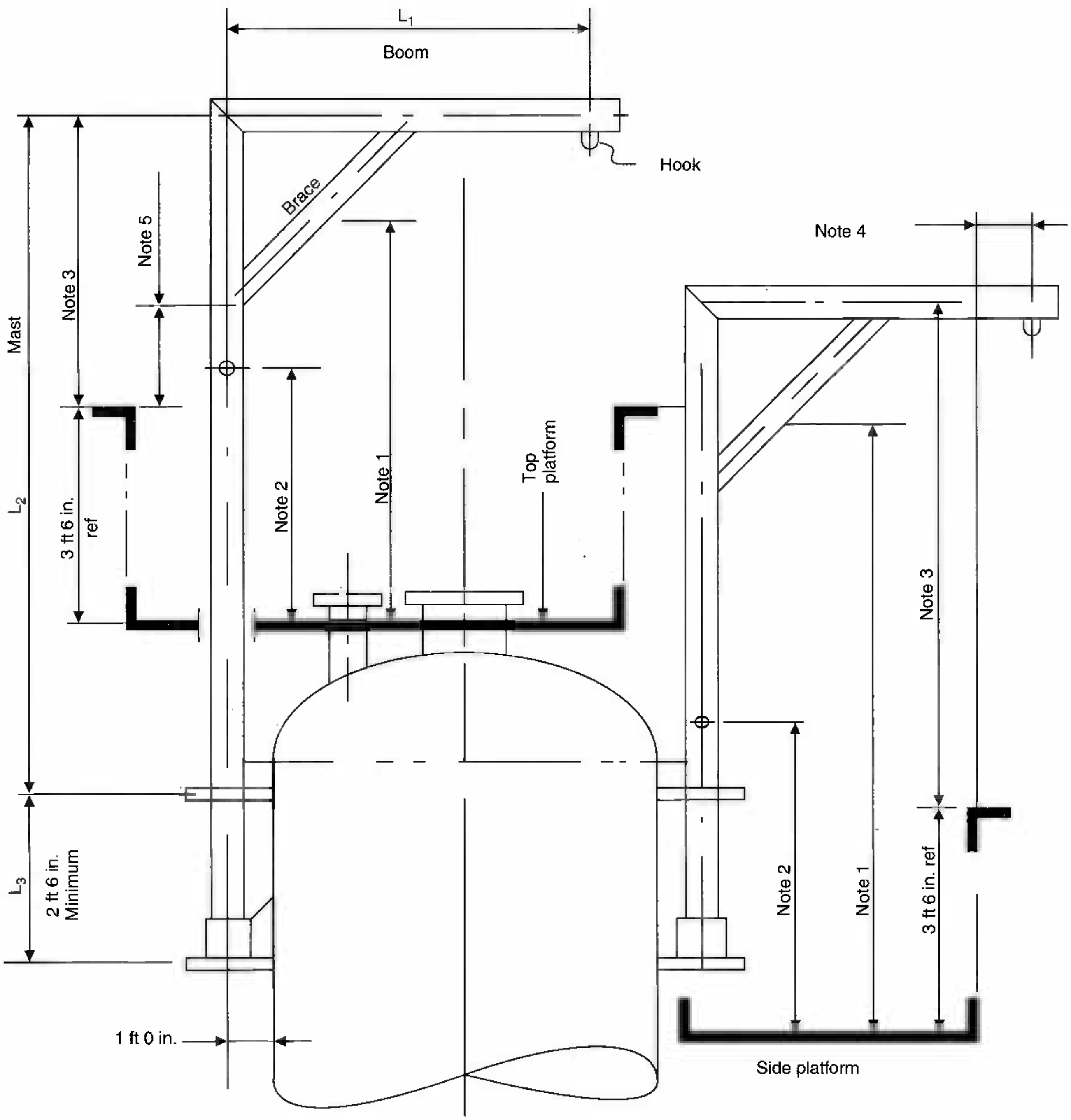


Figure 9-9. Area of loading at davit guide.

Davit Arrangement



Notes:

1. Check head clearance to middle of brace, 7 ft 0 in. minimum.
2. Set location of turning handle, 4 ft 0 in. minimum.
3. Check that equipment handled plus any rigging gear will clear handrail, 3 ft 0 in. minimum. As an alternative, the handrail may be made removable.
4. Check hook clearance to outside of platform, 9 in. minimum.
5. Check clearance between bottom of brace and handrail, 6 in. minimum.

Notes

1. Figure 9-6 illustrates the change in the total combined stress as the davit is rotated between 0° and 90°. As can be seen from the graph the stress due to M_x is constant for any degree of davit rotation. This stress occurs only at the support. The stress due to F_r varies from a maximum at 0° to 0 at 90°. The stress due to M_ϕ is 0 at 0° and increases to a maximum at 90°. To find the worst combination of stress, the equivalent radial load, F_{re} must be calculated for various degrees of davit rotation, ϕ. At the guide shell stresses should be checked by an appropriate local load procedure for the maximum equivalent radial load. At the support shell stresses should be checked for both F_{re} and M_x. Stresses from applicable external loads shall be combined. Remember the force F_{re} is a combination of loads F_r and M_ϕ at a given davit orientation. F_r and M_ϕ are maximum values that do not occur simultaneously.
2. Impact factors account for bouncing, jerking, and swinging of loads.

Procedure 9-2: Design of Circular Platforms

Notation

$$\text{Area} = \frac{\left(R^2 - r^2\right)\pi\phi}{360}$$

$$\text{Arc length,} \quad \ell = \frac{\pi R\theta}{180}$$

$$\text{Angle,} \ \theta = \frac{180\,\ell}{\pi R}$$

$$X = \sqrt{R^2 - A^2} - \sqrt{r^2 - A^2}$$

$$Y = L - \sqrt{r^2 - A^2}$$

f = dead load + live load, psf
f_b = bending stress, beam, psi
f_a = axial stress, psi
$f_{x,y,r}$ = bolt loads, lb
F = total load on bracket, lb
A = load area, ft^2
A' = cross-sectional area of kneebrace, $in.^2$
M_1 = moment at shell, ft-lb
M_2 = moment at bolts, ft-lb or in.-lb
C = distance to C.G. of area, ft
K = end connection coefficient, use 1.0
r' = radius of gyration, in.
P = axial load on kneebrace, lb
Z = section modulus of beam, $in.^3$

Table 9-1
Values of ladder spacing and for given diameters

Diameter (ft)	α
2	23°
4	17°
6	14°
8	11.5°
10	10°
12	9°
14	8°
16	7°
18	6°
20	5.5°

Note: Values in table are approximate only for estimating use.

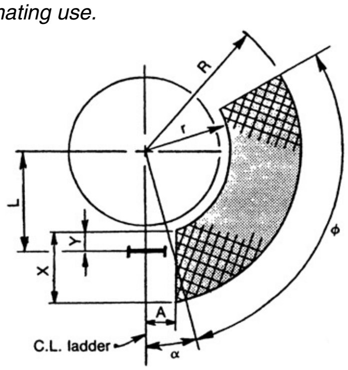


Figure 9-10. Dimensions of a typical circular platform.

AREA OF PLATFORMS				
Platform	ϕ	R	r	Area

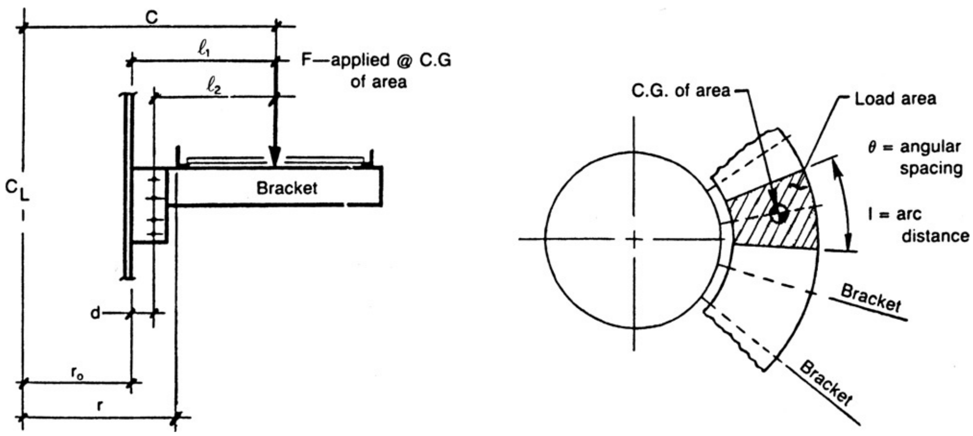


Figure 9-11. Dimensions, force, and local area for circular platforms.

COMPUTING MOMENTS IN SHELL AND BOLT LOADS														
Moments in shell:										**Type Clip**	**Bolt Loads**			
Platform	θ	**R**	**r**	**A**	**F**	**C**	l_1	l_2	M_1		M_2	f_x	f_y	f_r

Formulas for Chart

$$A = \frac{(R^2 - r^2)\pi\theta}{360}$$

$$F = fA$$

$$C = \frac{38.197(R^3 - r^3)\sin\theta/2}{(R^2 - r^2)\theta/2}$$

$$l_1 = C - r_o$$

$$l_2 = l_1 - d$$

$$M_1 = l_1 F$$

$$M_2 = l_2 F$$

Table 9-2
Allowable Shear loads in bolts (kips)

	Size (in.)				
Material	**⅝**	**¾**	**⅞**	**1**	**1–⅛**
A-307	3.68	5.30	7.21	9.42	11.9
A-325	7.36	10.6	14.4	18.8	23.8

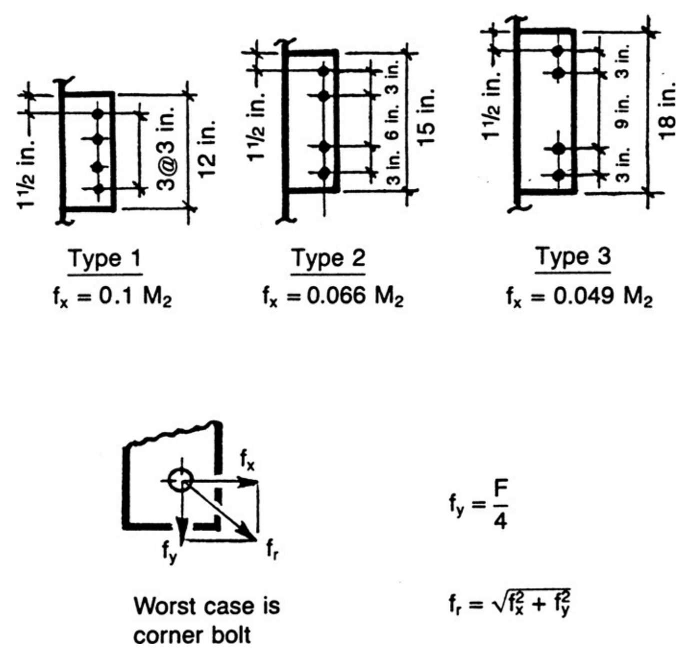


Figure 9-12. Bolt load formulas for various platform support clips. (See Figure 9-16 for additional data.)

Design of Kneebrace

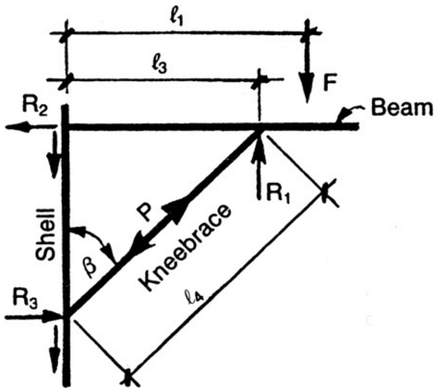


Figure 9-13. Dimensions, forces, and reactions of kneebrace support.

- *Reaction, R_1.*

$$\sum M_{R_2} = l_1F - l_3R_1 = 0 \qquad \therefore \quad R_1 = \frac{l_1F}{l_3}$$

- *Shear load on bolts/radial load on shell.*

$$R_2 = R_3 = R_1 \tan\beta$$

- *Bending stress in beam.*

$$f_b = \frac{|l_1 - l_3|F}{Z}$$

- *Axial load in kneebrace.*

$$P = \frac{R_1}{\cos\beta}$$

- *Axial stress.*

$$f_a = \frac{P}{A'}$$

- *Slenderness ratio/allowable stress.*

$$\frac{Kl_4}{r'} = F_a$$

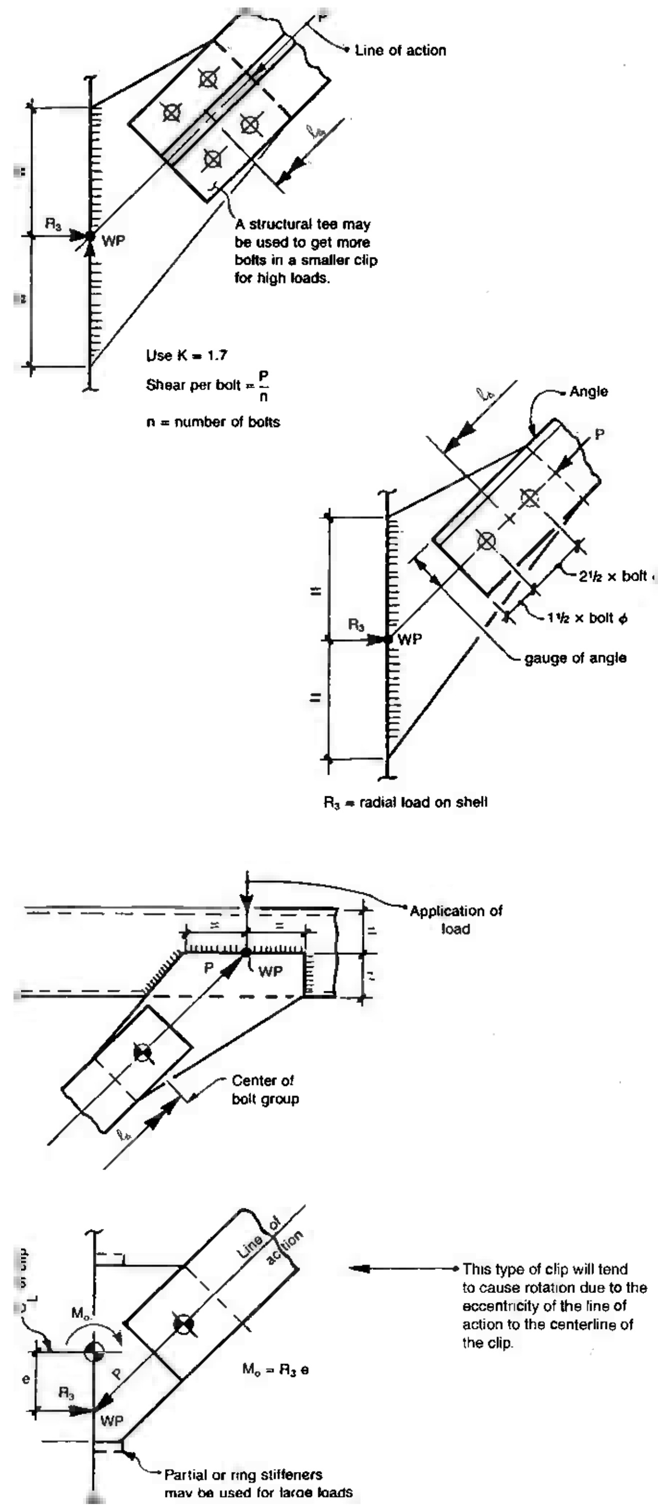


Figure 9-14. Typical bolted connections for kneebrace supports.

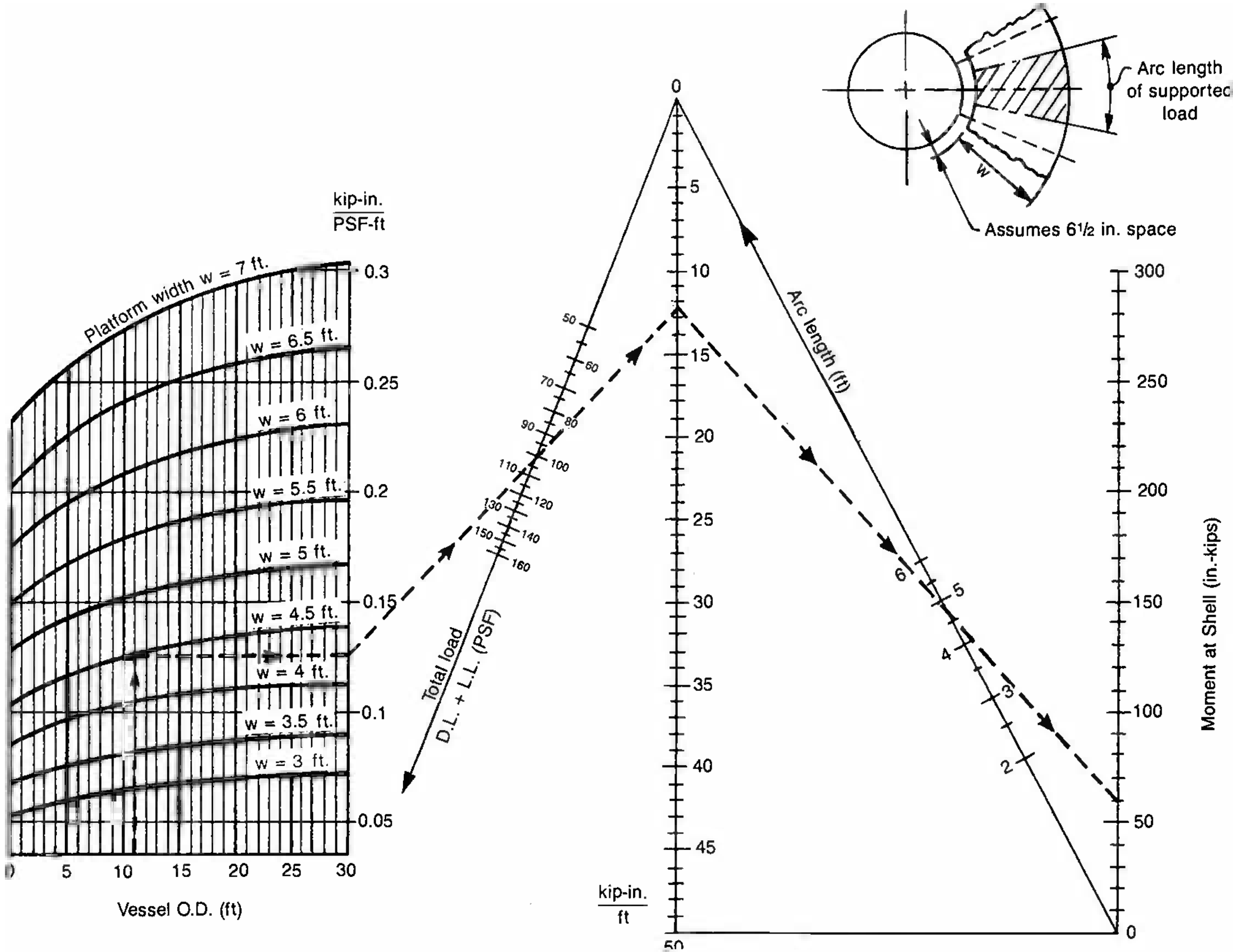


Figure 9-15. Nomograph to find moment at shell due to platform loads.

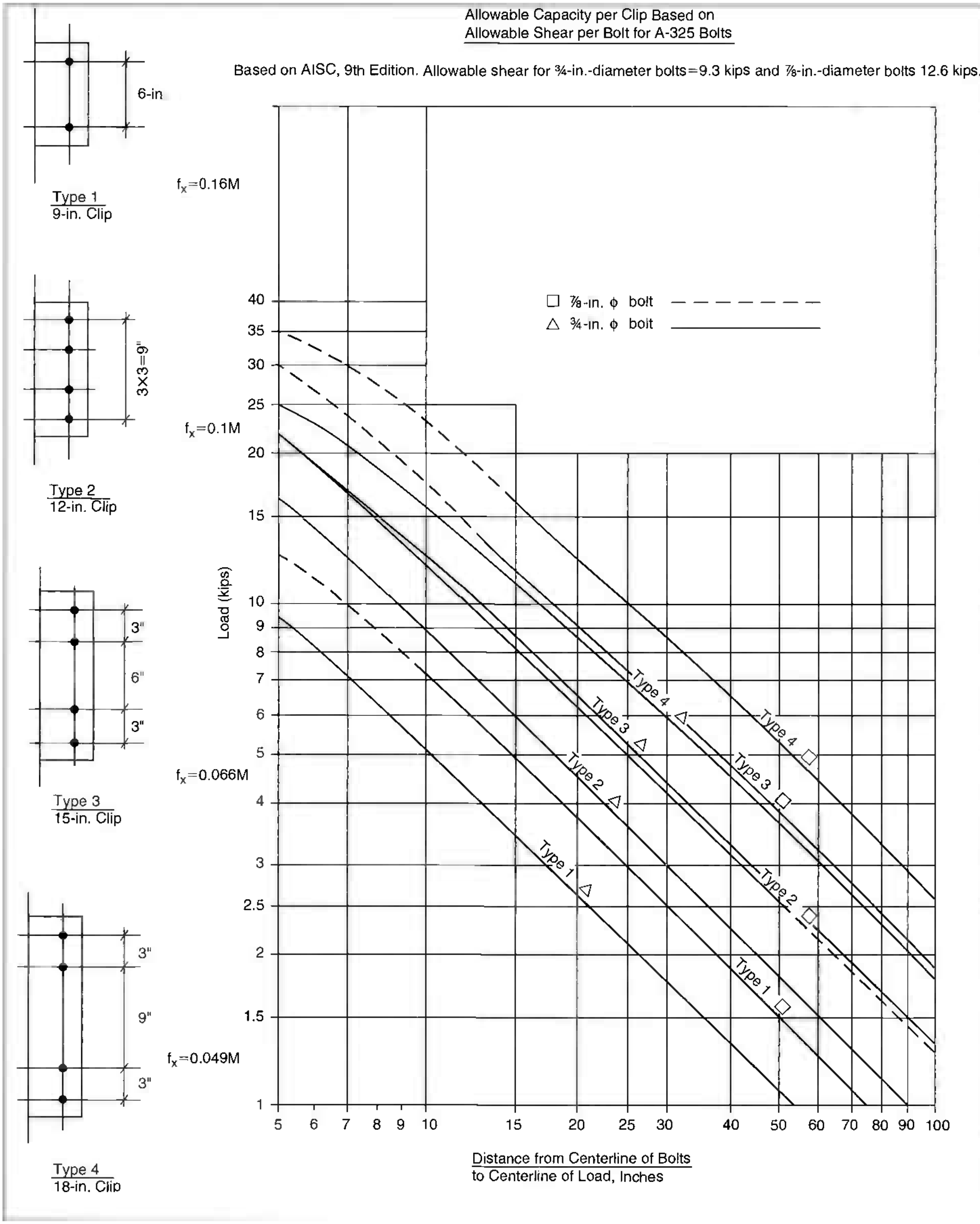
Table 9-3
Grating: Allowable live load based on fiber stress of 18,000 psi

Main Bar Size	Sec. Mod./ft width	Weight lb/sq ft	Bearing Bars at 1 ⅜ Center to Center—Cross Bars at 4 in. Span (ft-in.)											
			Type*	1-0	1-6	2-0	2-6	3-0	3-6	4-0	4-6	5-0	5-6	6-0
1 × ¼	0.380	9.0	U	4562	2029	1142	731	506	372	286	224			
			C	2283	1522	1142	912	762	653	571	506			
1 × 5/16	0.474	11.9	U	5687	2529	1423	910	633	465	355	282			
			C	2845	1898	1423	1139	947	812	712	633			
1¼ × ¼	0.594	10.9	U	7126	3169	1782	1141	793	583	446	353	286	236	196
			C	3564	2376	1782	1426	1186	1019	892	792	713	648	595
1¼ × 5/16	0.741	14.3	U	8888	3948	2221	1423	986	726	555	438	355	295	246
			C	4445	2963	2221	1778	1482	1268	1112	986	889	808	742
1½ × ¼	0.856	12.9	U	10265	4564	2567	1641	1142	836	641	506	412	339	286
			C	5132	3423	2567	2052	1712	1468	1282	1140	1027	932	856
1½ × 5/16	1.066	16.7	U	12796	5689	3198	2048	1423	1045	798	632	512	422	355
			C	6396	4266	3198	2558	2133	1826	1599	1422	1279	1163	1066
1½ × ⅜	1.276	19.6	U	15312	6806	3829	2451	1702	1251	958	758	613	506	425
			C	7654	5105	3829	3063	2553	2188	1914	1702	1532	1393	1276
1¾ × ¼	1.164	14.8	U	13963	6206	3492	2233	1553	1140	875	691	559	463	386
			C	6981	4656	3492	2792	2326	1996	1745	1552	1396	1270	1165
1¾ × 5/16	1.451	19.1	U	17411	7738	4352	2788	1936	1422	1087	861	696	576	484
			C	8708	5805	4352	3483	2903	2488	2176	1935	1742	1583	1452
1¾ × ⅜	1.737	22.5	U	20842	9262	5210	3336	2315	1702	1302	1029	834	688	579
			C	10420	6949	5210	4169	3473	2978	2604	2315	2085	1895	1738
2 × ¼	1.520	16.7	U	18242	8107	4562	2918	2026	1489	1141	902	730	604	507
			C	9121	6082	4562	3648	3040	2608	2281	2027	1825	1858	1521
2 × 5/16	1.895	21.5	U	22740	10102	5686	3637	2526	1858	1422	1123	910	753	633
			C	11371	7581	5686	4547	3791	3248	2842	2529	2275	2067	1895
2 × ⅜	2.269	25.4	U	27224	12098	6808	4356	3026	2223	1702	1344	1088	900	758
			C	13613	9073	6808	5446	4536	3888	3401	3026	2723	2476	2269

** C—concentrated*
U—uniform

Table 9-4
Floor plate: Allowable live load based on fiber stress of 20,000 psi

Long Span (ft-in.)	Nominal Thickness (in.)	Short Span (ft-in.)							
		2-6	3-0	3-6	4-0	4-6	5-0	5-6	6-0
Supports on Four Sides									
2-6	¼	656							
	5/16	1026							
3-0	¼	514	452						
	5/16	806	708						
3-6	¼	441	366	328					
	5/16	691	573	515					
4-0	¼	393	316	274	249				
	5/16	617	496	431	391				
4-6	¼	366	284	239	210	195			
	5/16	575	446	376	331	307			
5-0	¼	350	262	215	185	167	156		
	5/16	550	411	338	291	264	246		
5-6	¼	340	248	198	168	148	135	126	
	5/16	532	391	312	265	234	214	201	
6-0	¼	330	240	187	154	134	120	111	104
	5/16	517	377	293	244	213	191	173	166
6-6	¼			178	145	124	109	96	93
	5/16			281	230	197	174	158	140
7-0	¼			173	138	116	101	91	83
	5/16			273	218	184	162	145	135
7-6	¼			170	133	111	95	84	76
	5/16			268	210	175	152	135	122
8-0	¼					106	90	79	71
	5/16					168	143	127	114
8-6	¼					102	86	75	67
	5/16					163	137	120	106
9-0	¼							72	63
	5/16							114	101
Supports on Two Sides									
∞	¼	255	174	125	93	71	55		
∞	5/16	402	275	198	148	114	90	72	58



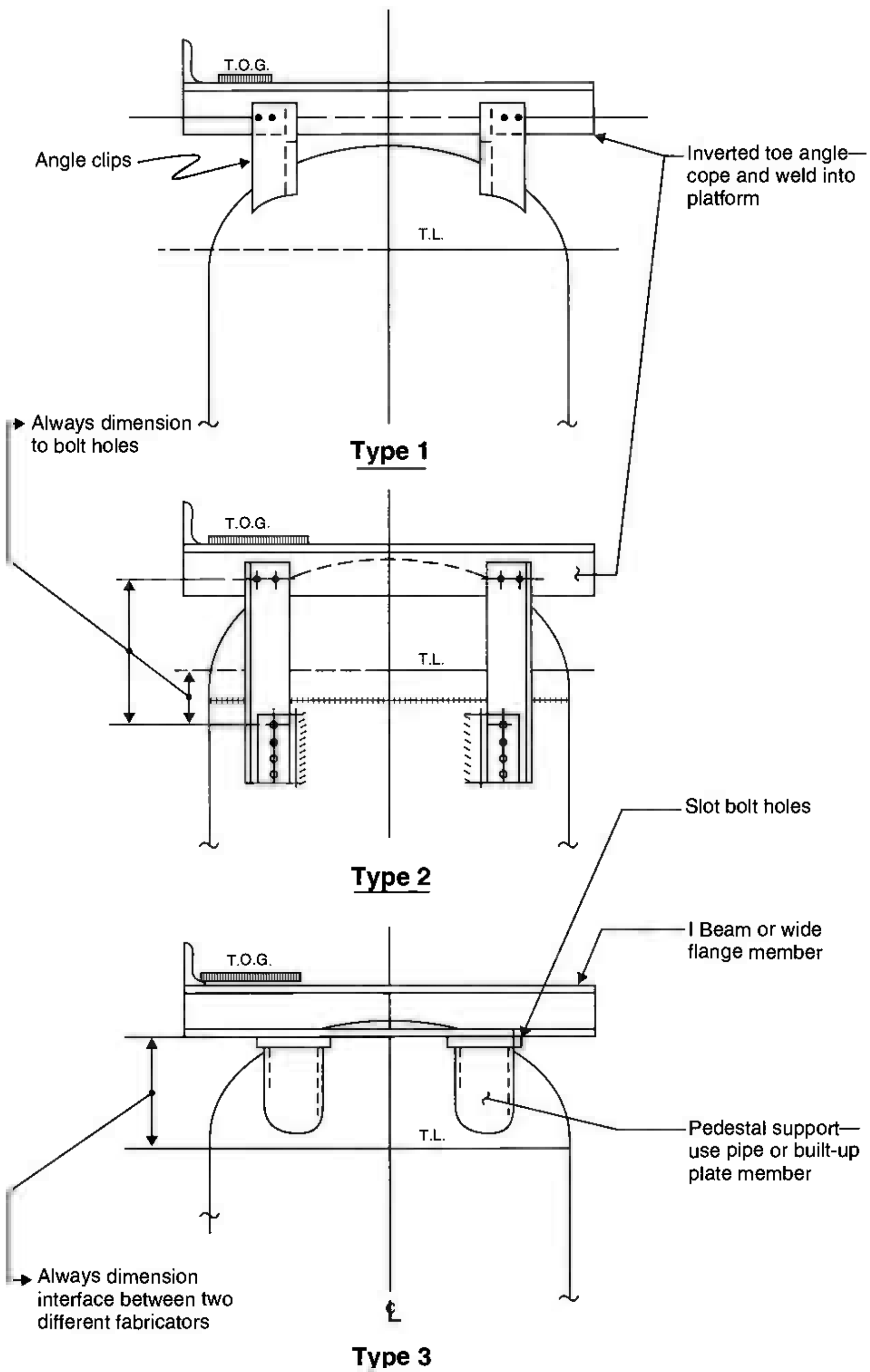
Allowable Capacity per Clip Based on
Allowable Shear per Bolt for A-325 Bolts
Based on AISC, 9th Edition. Allowable shear for ¾-in.-diameter bolts=9.3 kips and ⅞-in.-diameter bolts 12.6 kips.
6-in
Type 1
9-in. Clip
f_x=0.16M
3×3=9"
Type 2
12-in. Clip
f_x=0.1M
3"
6"
3"
Type 3
15-in. Clip
f_x=0.066M
3"
9"
3"
Type 4
18-in. Clip
f_x=0.049M
□ ⅞-in. ϕ bolt
△ ¾-in. ϕ bolt
Load (kips)
40
35
30
25
20
15
10
9
8
7
6
5
4
3
2.5
2
1.5
1
Type 4 △
Type 3 △
Type 2 △
Type 1 △
Type 4 □
Type 3 □
Type 2 □
Type 1 □
5
6
7
8
9
10
15
20
25
30
40
50
60
70
80
90
100
Distance from Centerline of Bolts
to Centerline of Load, Inches

Notes

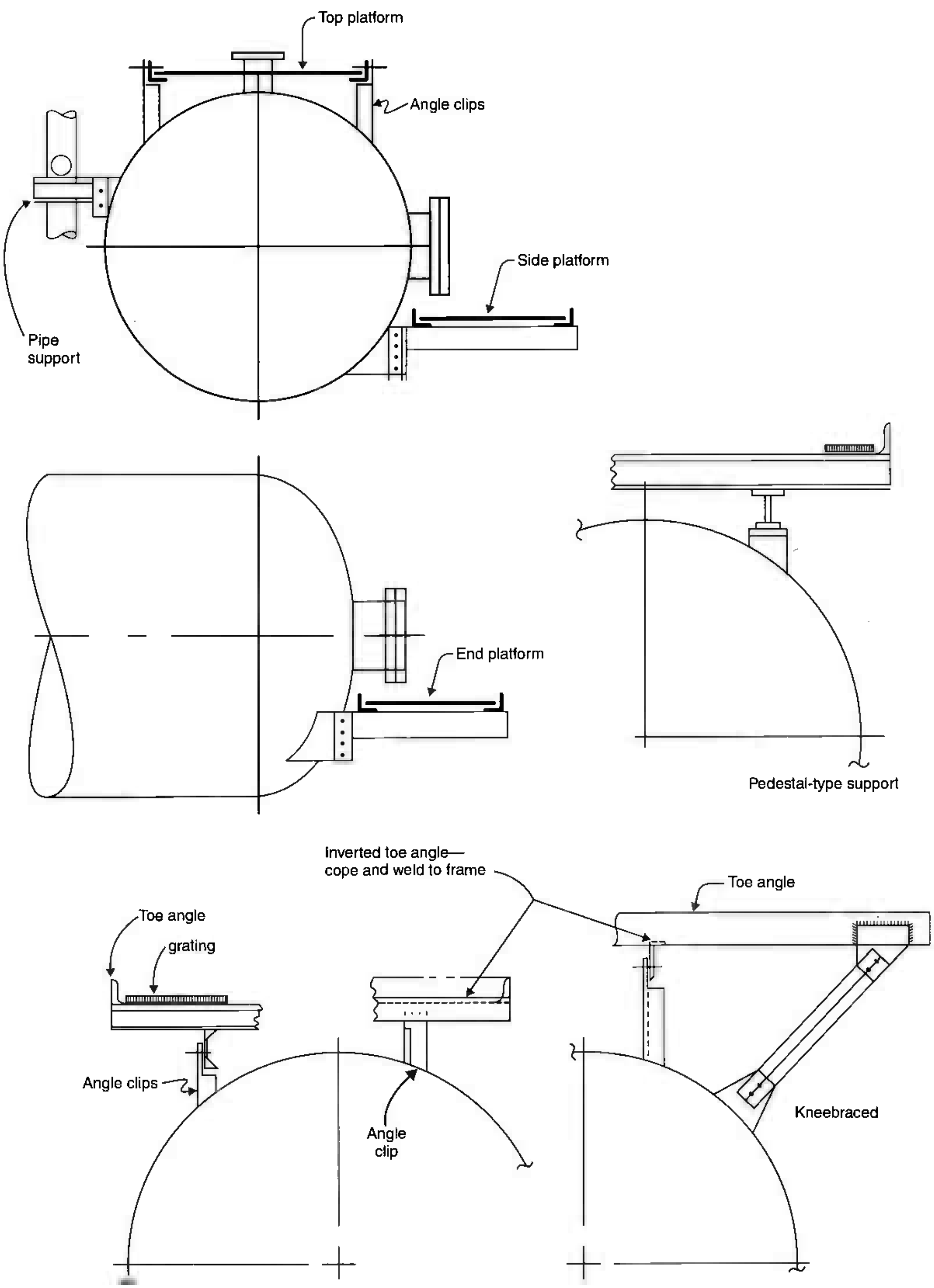
1. Dead loads: 30psf. Platform steel weight. This includes grating or floor plate, structural framing, supports, toe angle or plate, and handrailing. To find weight of steel, multiply area of platforms by 30 psf.
2. Live loads:
   - *Operating*: Approximately 25–30 psf. Live load is small because it is assumed there are not a lot of people or equipment on the platform while vessel is operating. Combine effects with shell stress due to design pressure.
   - *Maintenance/construction*: 50–75 psf. Live load is large because there could be numerous persons, tools, and equipment on platforms; however, there would be no internal pressure on vessel.
3. Assume each bracket shares one-half of the area between each of the adjoining brackets. Limit bracket spacing to 6 ft-0 in. arc distance and overhangs to 2 ft-0 in. For stability, bracket spacing should not exceed 60°.
4. Kneebraces should be 45° wherever possible. Always dimension to bolt holes, not to edge of brackets or top of clips.
5. Bracket spacing is governed by one of the following conditions:
   - *Shell stress*: Based on dead-load and live-load induced stress from platform support brackets. Shell stresses may be reduced by using a longer clip or reducing the angle between brackets.
   - *Bolt shear stress*: A-307 or A-325 in single or double shear. Bolt shear stresses may be reduced by increasing the size or number of bolts or increasing the distance between bolts.
   - *Maximum arc distance*: Measured at the outside of the platform. Based on the ability of the toe angle to transmit loads to brackets. Affects "stability" of platform.
   - *Stress/deflection of floor plate or grating*: Allowable live load affects "springiness of platforms." Use Tables 9-3 and 9-4 and assume "allowable live load" of 150–200 psf.
6. Shell stresses should be checked by an appropriate "local load" procedure.

Procedure 9-3: Design of Square and Rectangular Platforms

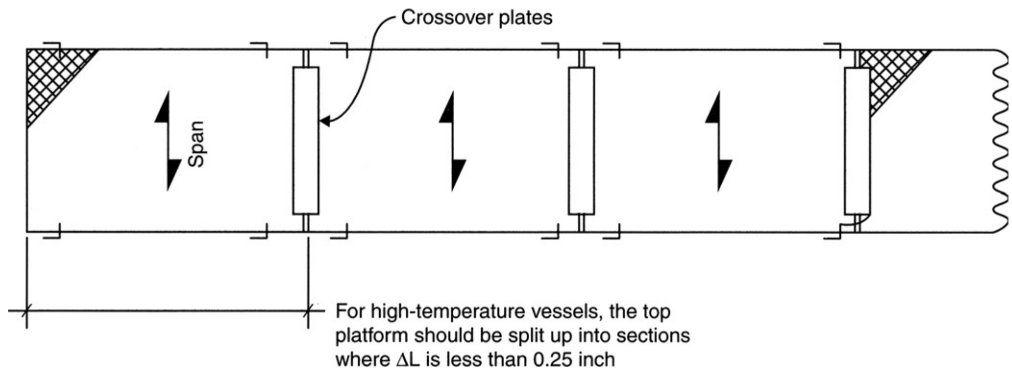
Top Platforms for Vertical Vessels



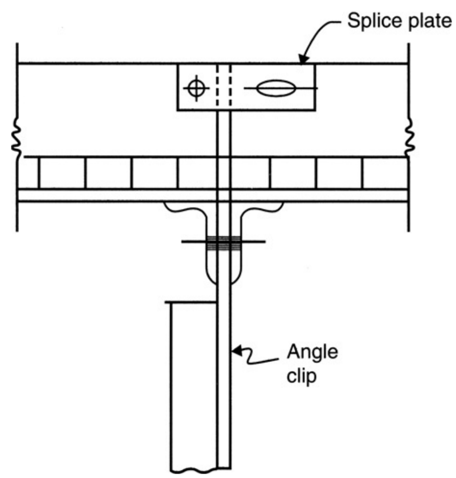
Platforms for Horizontal Vessels



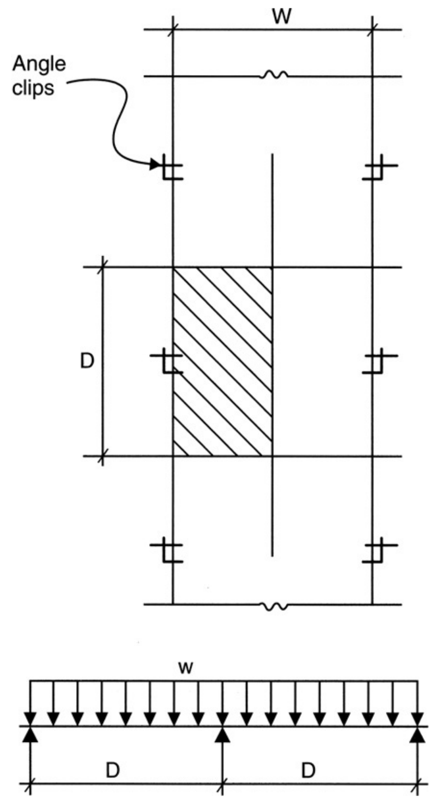
Long Walkways or Continuous Platform on Horizontal Vessel



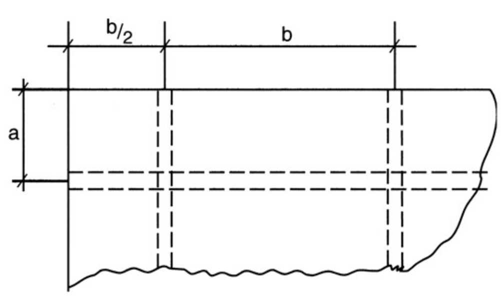
Horizontal Platform Splice (Not for Thermal Expansion)



Check of Toe Angle Frame



Maximum Length of Unsupported Toe Angle (Based on 105-psf Load and L6 × 3½ × 5/16)



a (ft)	b (ft)	
	Grating	Check plate
<1	15	∞
1½	10	12
2	8	9
2½	6	6

Check clip spacing:

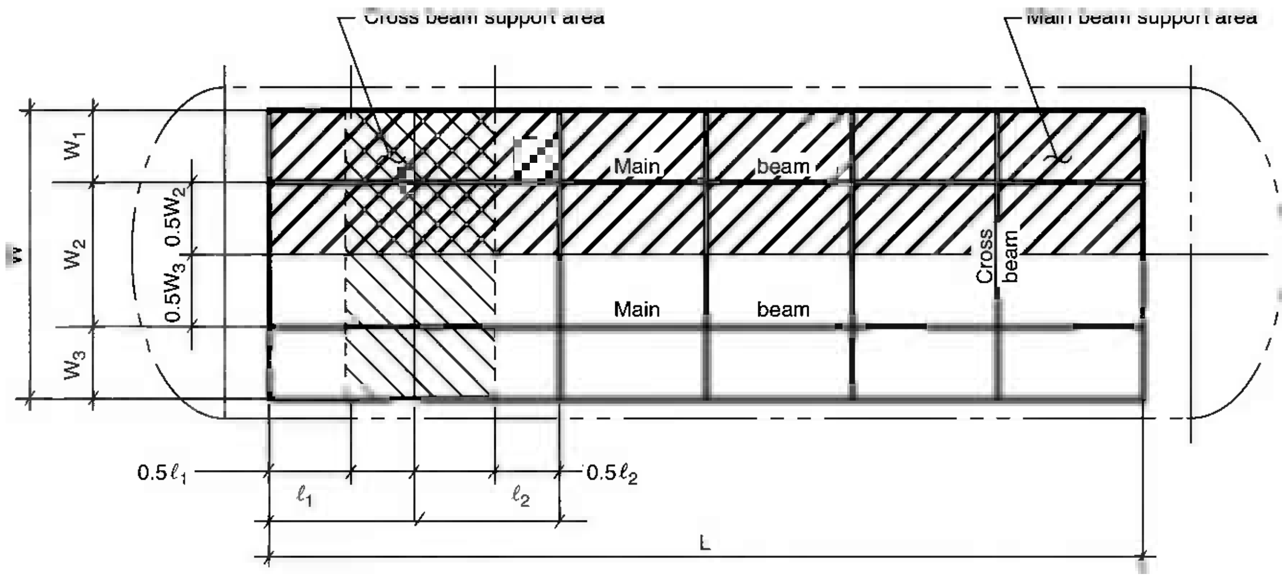
$$P = D.L + L.L. = \text{psf}$$

$$w = \frac{WP}{2}\ \frac{\text{lb}}{\text{ft}}$$

$$M = \frac{wD^2}{8}\ \text{ft-lbs}$$

$$\sigma = \frac{12M}{Z} < 21.6\ \text{ksi}$$

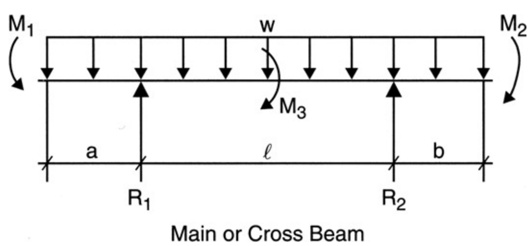
$Z =$ Section modulus of toe angle, in^3



Notation

A = area, sq ft
p = unit load, psf
P = total load, lb
w = unit load on beam, lb/linear foot
R = reaction, lb
M = moment, in.-lb
d = deflection, in.
K = end connection coefficient, use 1.0
r = radius of gyration of column, in.
f_a = axial stress, psi
F_a = allowable compressive stress, psi

Calculations



Main or Cross Beam

- *Area, A.*

$$A = (0.5w_2 + w_1)L$$

- *Load, P.*

$$P = A_p$$

- *Unit load on beam, w.*

$$w = \frac{P}{L}$$

$$R_1 = \frac{w\left[(a+\ell)^2 - b^2\right]}{2l}$$

$$M_1 = \frac{wa^2}{2}$$

$$M_2 = \frac{wb^2}{2}$$

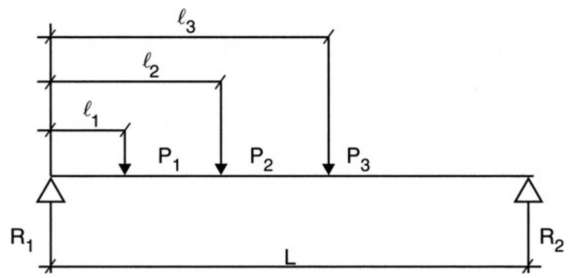
$$M_3 = R_1\left(\frac{R_1}{2w} - a\right)$$

$$\delta_{end} = \frac{wa}{24EI}\left(\ell^3 - 6a^2\ell - 3a^3\right)$$

$$\delta_{center} = \frac{w\ell^2}{384EI}\left(5\ell^2 - 24a^2\right)$$

Notes

1. Maximum distance between cross beams is governed by one of two conditions:
   a. Maximum span of grating or checkered plate.
   b. Deflection/stress of toe angle. Ability of toe angle to carry the load.
2. Each beam supports the load from one-half the area between the adjacent beams.



Beams—Multiple Loads

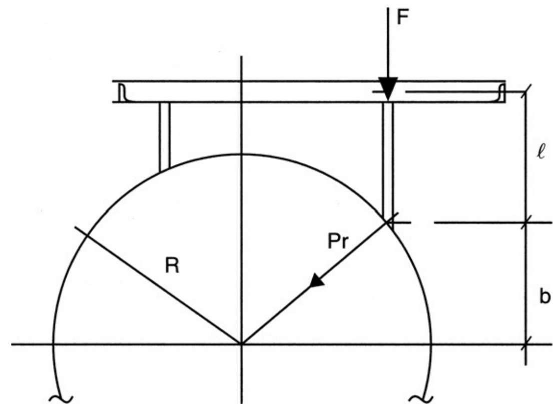
$$R_2 = \frac{l_1P_1 + l_2P_2 \cdots + \cdots l_nP_n}{L}$$

$$R_1 = \sum P_n - R_2$$

To find maximum moment:

1. Select maximum reaction.
2. Total all downward loads, starting from the reaction, until the value of the reaction is exceeded. This is the point where the maximum moment will occur.
3. The moments are equal to the right or left of that point. Sum the moments in either direction.

Design of Vessel Clips



- *Slenderness ratio.*

$$\frac{k\ell}{r}$$

Use $k = 1.0$.

- *F = reaction from main beam columns.*
- *Allowable compressive stress, F_a, based on slenderness ratio.*
- *Axial stress, f_a.*

$$f_a = \frac{F}{A}$$

- *Check stress ratio.*

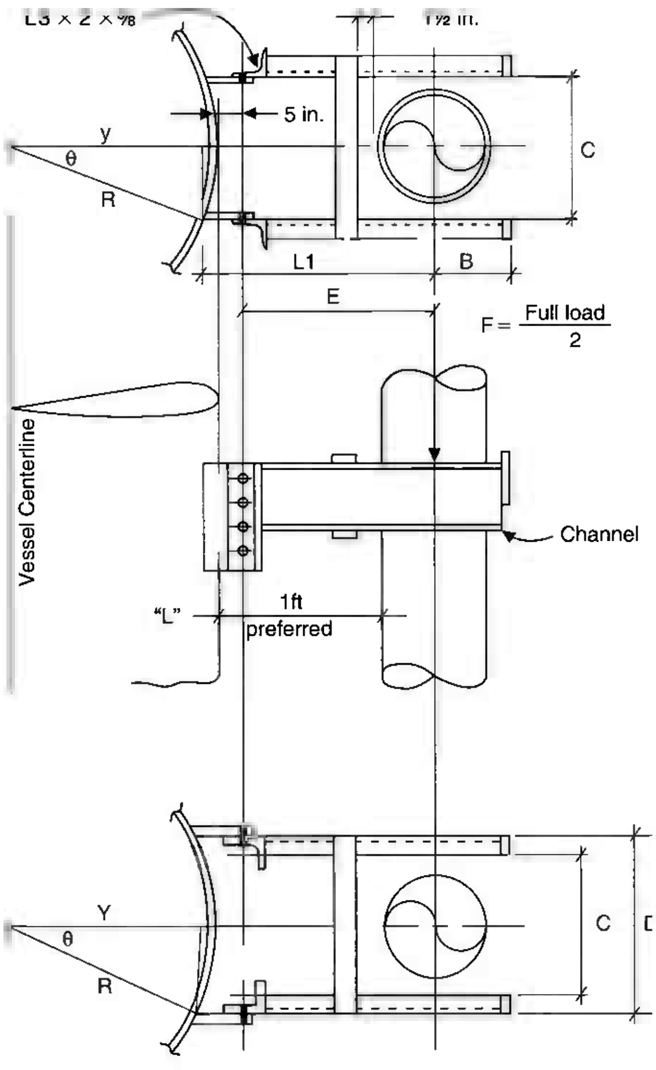
$$\frac{f_a}{F_a} < 0.15$$

- *Radial load in shell, P_r.*

$$P_r = \frac{bF}{R}$$

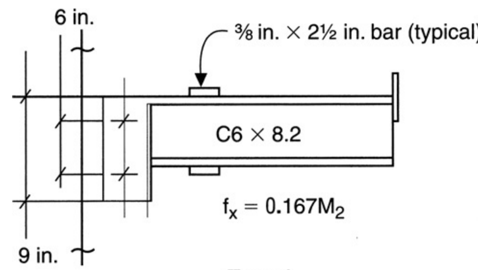
Procedure 9-4: Design of Pipe Supports

Unbraced Pipe Supports

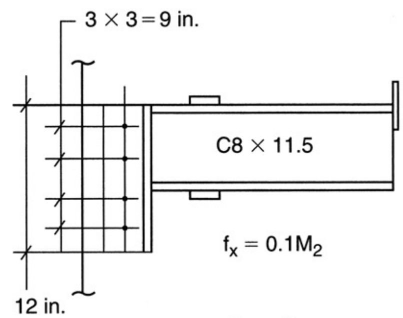


Alternate Case: Legs Turned In

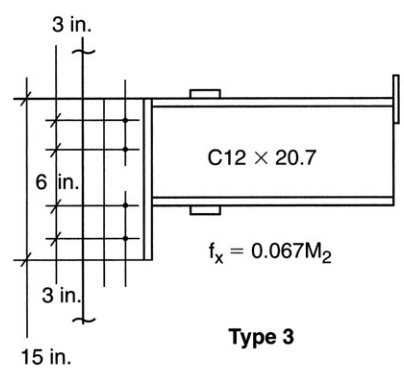
Types of Brackets



Type 1



Type 2



Type 3

Table 9-5
Pipe support dimensions

Dimension	Pipe Size											
	2 in.	3 in.	4 in.	6 in.	8 in.	10 in.	12 in.	14 in.	16 in.	18 in.	20 in.	24 in.
B	2.75	3.5	4.25	6	7.5	9	10.5	11.25	12.75	14	16	18
C	4	5	6.5	9	12	14	16	17	19	21	23	27
D—Type 1	7.75	8.75	10.25	12.75	15.75	17.75	19.75	20.75				
D—Type 2				13.5	16.5	18.5	20.5	21.5	23.5			
D—Type 3					18	20	22	23	25	27	29	33

Table 9-6
Weight of pipe supports, lb (without clips)

L Dimension	Support Type	Pipe Size									
		2 in.	3 in.	4 in.	6 in.	8 in.	10 in.	12 in.	14 in.	16 in.	18 in.
12 in.	1	12	13	15	18	22	25				
	2				22	27	31	35	37	41	45
	3					44	51	57	61	67	72
14 in.	1	15	16	18	21	25	28				
	2				26	31	35	39	41	45	49
	3					52	58	64	68	74	79
16 in.	1	18	19	20	24	28	31				
	2				30	35	39	43	45	49	51
	3					59	65	71	74	81	86
18 in.	1	20	21	23	26	31	34				
	2				34	39	43	46	49	53	56
	3					65	72	77	81	88	93
20 in.	1	23	24	26	29	33	36				
	2				38	42	46	50	53	57	60
	3					72	79	84	88	95	100

Kneebraced Pipe Supports

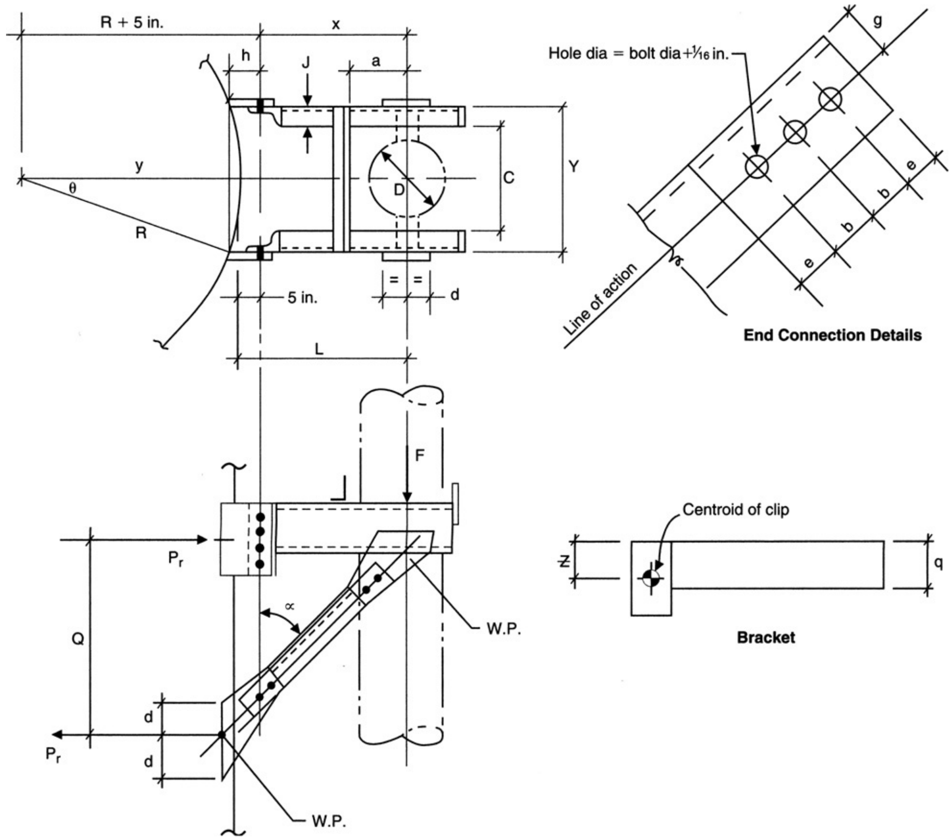
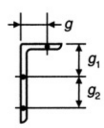


Table 9-7
Usual gauges for angles, in.



Leg	8	7	6	5	4	3½	3	2½	2	1¾	1½	1⅜	1¼	1
g	4½	4	3½	3	2½	2	1¾	1⅜	1⅛	1	⅞	⅞	¾	⅝
g_1	3	2½	2¼	2										
g_2	3	3	2½	1¾										

Dimensions

$$a = \frac{D}{2} + 3 \text{ in.}$$

$$Y = C + 2j + 1 \text{ in.}$$

$$x = L - 5 \text{ in.}$$

$$\theta = \text{arc sin}\left(\frac{Y}{2R}\right)$$

$$y = R \operatorname{Cos} \theta$$

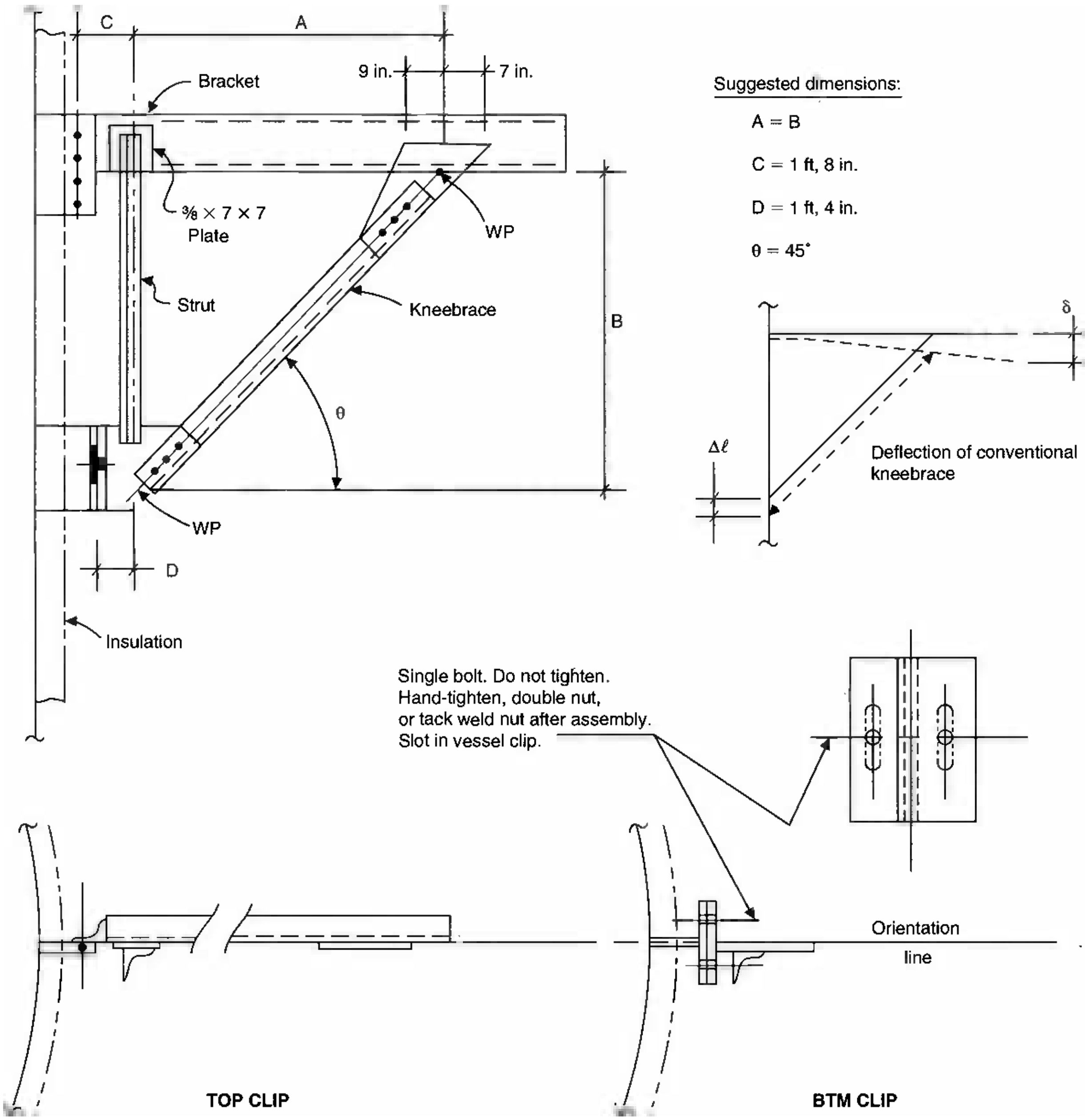
$$h = (R + 5 \text{ in.}) - y$$

$$Q = (x + h) + (q - z)$$

Table 9-8
Kneebraced pipe support dimensions

Allowable Load (kips)	Bracket Type	"L" Max	Angle Size	Bolt Qty & Size	b	e	d	j
12.5	1	36	2 ½ × 2 × ⅜	(2) ¾	2.5	1.25	2.5	1.92
17.5	1	36	3 × 2 × ⅜	(2) ⅞	2.75	1.5	3	1.92
24	1	36	3 × 2 × ⅜	(3) ⅞	2.75	1.5	3	1.92
21.5	1	54	3½ × 3 × ⅜	(2) 1	3	1.75	3.25	1.92
24	2	54	3½ × 3 × ⅜	(2) 1⅛	3.25	2	4	2.26
26.5	2	54	4 × 3 × ⅜	(2) 1¼	3.5	2.25	4.5	2.26
30	2	54	4 × 3 × ⅜	(3) 1	3	1.75	3.25	2.26
33.5	2	54	5 × 3 × ⅜	(3) 1⅛	3.25	2	4	2.26
37.5	2	54	5 × 3 × ⅜	(3) 1¼	3.5	2.25	4.5	2.26
26.5	3	66	6 × 3½ × ⅜	(2) 1¼	3.5	2.25	5.25	2.942
37.5	3	66	6 × 3½ × ⅜	(3) 1¼	3.5	2.25	5.25	2.942
26.5	3	75	6 × 4 × ⅜	(2) 1¼	3.5	2.25	5.25	2.942
37.5	3	75	6 × 4 × ⅜	(3) 1¼	3.5	2.25	5.25	2.942
50	3	75	6 × 4 × ⅜	(4)1¼	3.5	2.25	5.25	2.942

High-Temperature Brackets

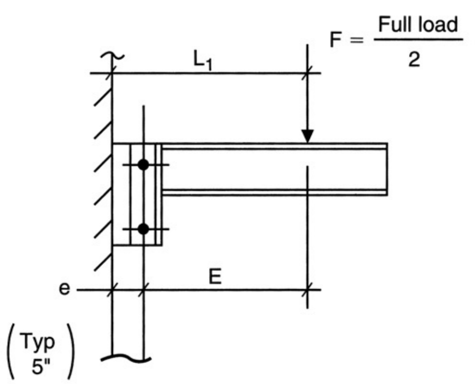


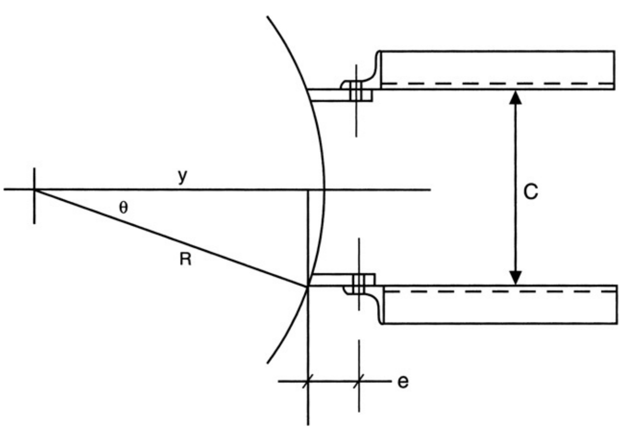
Design of Supports

Notation

A = cross-sectional area of kneebrace, $in.^2$
F = ½ of the total load on the support, lb
R_n = reaction, lb
P = compression load in kneebrace, lb
P_r = radial load in shell, lb
M_1 = moment at shell, in.-lb
M_2 = moment at line of bolts, in.-lb
r = radius of gyration, in.
N = number of bolts in clip
τ = shear load, lbs
E = modulus of elasticity, psi
I = moment of inertia, $in.^4$
Z = section modulus, $in.^3$
K = end connection coefficient
δ = deflection, in.
f_a = axial stress, psi
f_b = bending stress, psi
F_a = allowable axial stress, psi
F_b = allowable bending stress, psi

Cantilever-Type Brackets





- *Dimensions.*

$$\sin\theta = \frac{C}{2R}$$

$$\therefore\ \theta =$$

$$y = R\cos\theta$$

$$L_1 = (R - y) + L + \frac{1}{2}\ \text{pipe dia}$$

$$E = L_1 - [(R - y) + e]$$

- *Loads.*

$$M_1 = FL_1$$
$$M_2 = FE$$

- *Bracket check.*

$$f_b = \frac{M_1}{Z} < F_b$$

$$\delta = \frac{FL_1^3}{3EI}$$

- *Bolting check.*

Type	f_x
1	$0.167\ M_2$
2	$0.1\ M_2$
3	$0.067\ M_2$

$$f_y = \frac{F}{N}$$

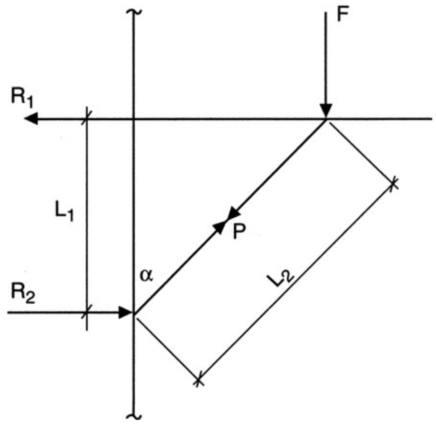
$$f_r = \sqrt{f_x^2 + f_y^2}$$

Compare with allowable shear.

- *Check shell for longitudinal moment, M_2.*

Design of Kneebraced Supports

Case 1



$$R_1 = R_2 = F \tan \alpha$$

$$P = \frac{F}{\cos \alpha} \quad L_2 = \frac{L_1}{\cos \alpha}$$

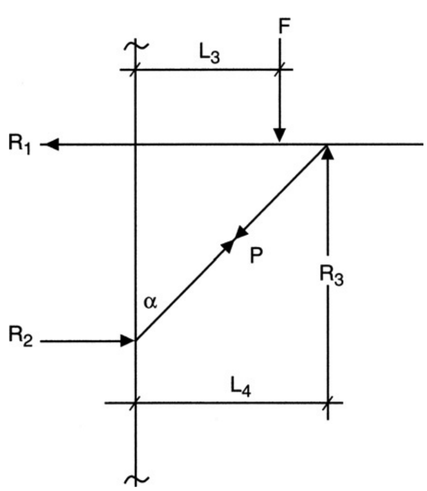
$$f_a = \frac{P}{A} < F_a$$

$$\frac{KL_2}{r} \quad F_a$$

$$P_r = R_1 \cos \theta$$

$$\tau = \frac{P}{N} \text{ or } \frac{R_1}{N}$$

Case 2



$$R_3 = \frac{L_3 F}{L_4}$$

$$R_1 = R_2 = R_3 \tan \alpha$$

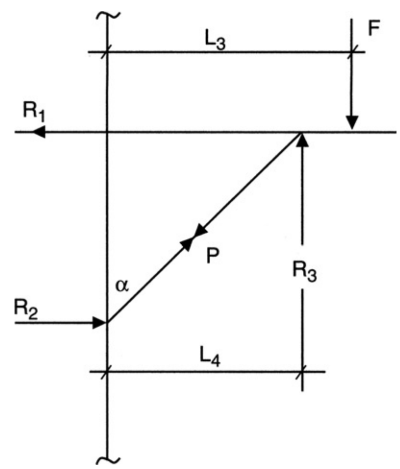
$$P = \frac{R_3}{\cos \alpha}$$

$$f_a = \frac{P}{A} < F_a \qquad f_b = \frac{L_4 - L_3}{Z}$$

$$\frac{KL_2}{r} \quad F_a$$

$$\tau = \frac{P}{N} \text{ or } \frac{R_1}{N}$$

Case 3



$$R_3 = \frac{L_3 F}{L_4}$$

$$R_1 = R_2 = R_3 \tan \alpha$$

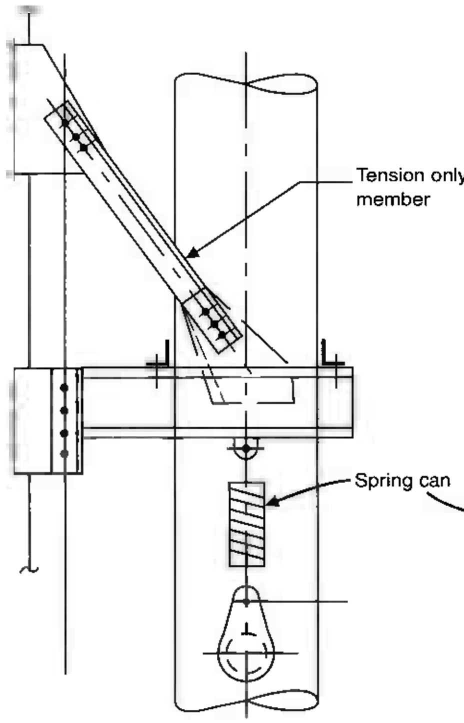
$$P = \frac{R_3}{\cos \alpha}$$

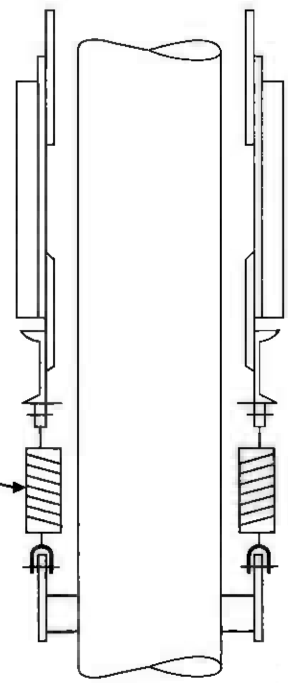
$$f_a = \frac{P}{A} < F_a \qquad f_b = \frac{L_3 - L_4}{Z}$$

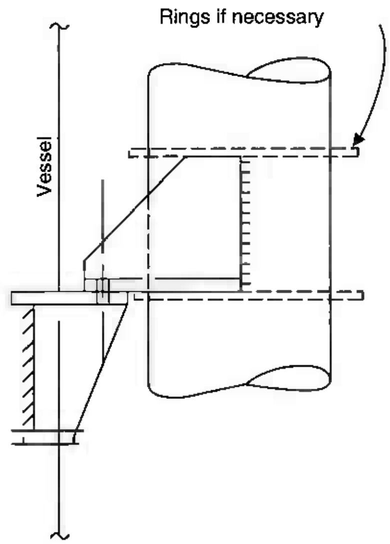
$$\frac{KL_2}{r} \quad F_a$$

$$\tau = \frac{P}{N} \text{ or } \frac{R_1}{N}$$

Alternate-Type Supports

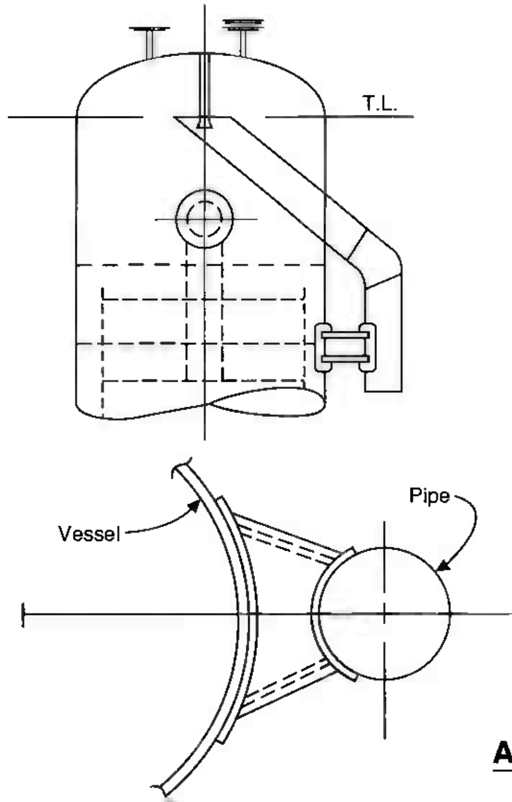


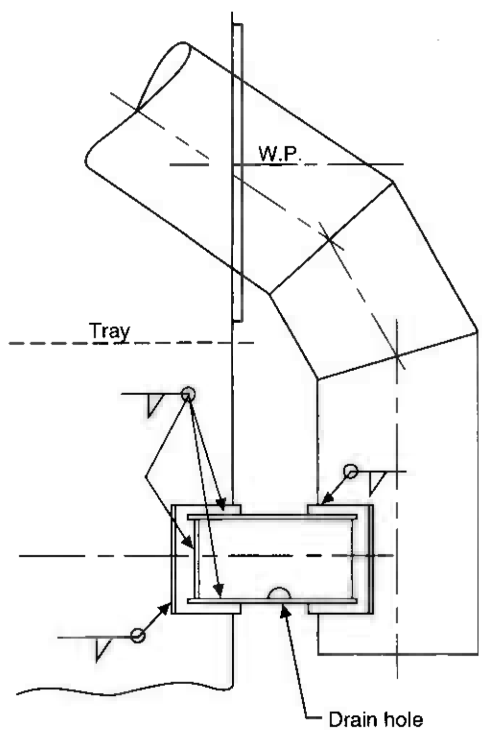




Pipe and Vessel Share Load to Reduce Load on Vessel Shell

Inverted Support, Large Lines with Spring Hangers





All Welded Integral Construction for Overhead Vapor Lines

Notes

1. Allowable deflection brackets should be limited to L/360.
2. Kneebracing should be used only if absolutely necessary.
3. Pipe support should be placed as close as possible to the nozzle to which it attaches. This limits the effect of differential temperature between the pipe and the vessel. If the line is colder than the vessel, the nozzle will tend to pick up the line. For the reverse situation (pipe hotter than vessel), the line tends to go into compression and adds load to the support.
4. The nozzle and the pipe support will share support of the overall line weight. Each will share the load in proportion to its respective stiffness. The procedure is to design the pipe support for the entire load, which is conservative. However, be aware that as the pipe support deflects, more of the load is transferred to the nozzle.
5. The pipe is normally supported by trunnions welded to the pipe. The trunnions can be shimmed to accommodate differences in elevation between the trunnions and the supports.
6. Design/selection of pipe supports:
   - Make preliminary selection of support type based on the sizing in the table.
   - Check allowable bolt loads per chart.
   - Check shell stresses via the applicable local load procedure.
7. The order of preference for overstressed supports, shells, or bolts is as follows:
   - Go to the next largest type of support.
   - If the loads in the bolts exceed that allowable, change the material or size of the bolts.
   - If the brackets are overstressed, increase the bracket size.
8. Use "high-temperature brackets" for kneebraced pipe supports or platform brackets when the design temperature of the vessel exceeds 650°F. This sliding support is utilized for hot, insulated vessels where the support steel is cold. This sliding support prevents the support from dipping as the vessel clips grow apart due to linear thermal expansion of the vessel while the kneebrace remains cold. This condition becomes more pronounced as the vessel becomes hotter and the distance between clips becomes greater.
9. Keep bolts outside of the insulation.
10. Vessel clip thickness should be ⅜ in. for standard clips up to 650°F. Above 650°F, clips should be ½ in. thick.
11. Bolt holes for Type 1, 2, or 3 supports should be $^{13}/_{16}$ -in.-diameter holes for ¾-in.-diameter bolts.

Procedure 9-5: Shear Loads in Bolted Connections

Table 9-9
Allowable loads, in kips

Material	Size		⅝ in.	¾ in.	⅞ in.	1 in.	1 ⅛ in.	1 ¼ in.	1 ⅜ in.	1 ½ in.
A-307	Single		3.68	5.30	7.21	9.42	11.9	14.7	17.8	21.2
	Double		7.36	10.6	14.4	18.8	23.8	29.4	35.6	42.4
A-325	Single		7.36	10.6	14.4	18.8	23.8	29.4	35.6	42.4
	Double		14.7	21.2	28.8	37.6	47.7	58.9	71.2	84.8

Values from AISC.

Cases of Bolted Connections

Case 1

n = no. of fasteners in a vertical row

m = no. of fasteners in a horizontal row = 2

I_p = polar moment of inertia about c.g. of fastener group: $I_x + I_y$

$$I_x = 2\left[\frac{nb^2(n^2-1)}{12}\right]$$

$$I_y = n\left[\frac{mD^2(m^2-1)}{12}\right]$$

$$f_x = \frac{(F\ell)(n-1)b}{2I_p}$$

$$f_y = \frac{F}{mn} + \frac{F\ell D}{2I_p}$$

$$f_r = \sqrt{f_x^2 + f_y^2}$$

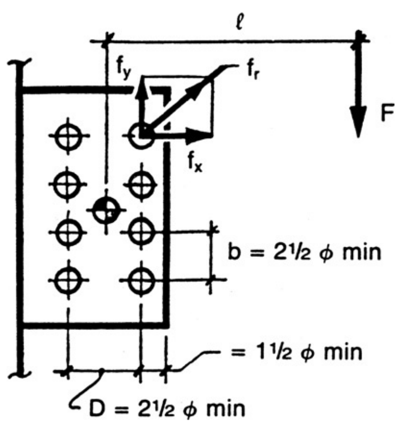


Figure 9-16. Longitudinal clip with double row of n bolts.

Case 2

$$f_x = \frac{F\ell}{e}$$

$$f_y = \frac{F}{2}$$

$$f_r = \sqrt{f_x^2 + f_y^2}$$

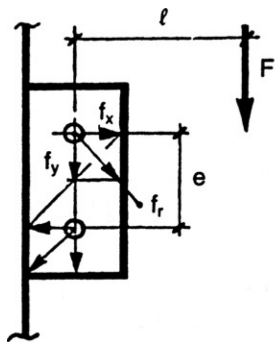


Figure 9-17. Longitudinal clip with two bolts.

Case 3

$$f = \frac{F\ell}{e}$$

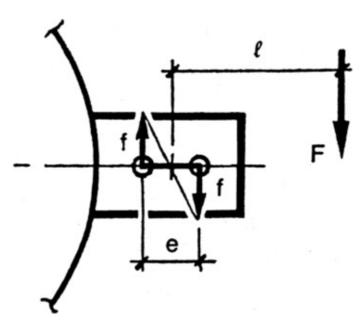


Figure 9-18. Circumferential clip with two bolts.

Case 4

$$f_x = x_n\left[\frac{Fl}{2\left(x_1^2 + x_2^2 + \ldots + x_n^2\right)}\right]$$

$$f_y = \frac{F}{n}$$

$$f_r = \sqrt{f_x^2 + f_y^2}$$

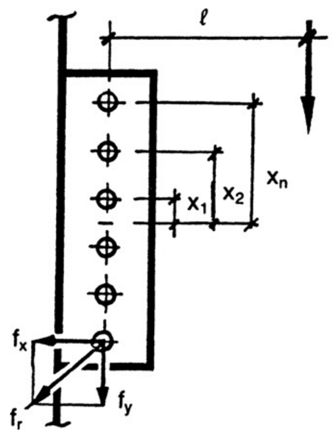


Figure 9-19. Longitudinal clip with single row of n bolts.

Case 5

$$f_x = \frac{Fl}{2b}$$

$$f_y = \frac{Fld}{2(b^2 + d^2)}$$

$$f_r = \sqrt{f_x^2 + f_y^2}$$

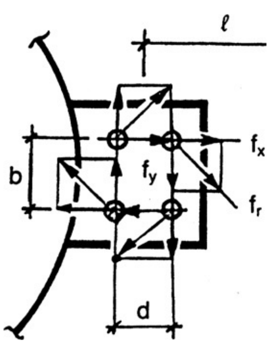


Figure 9-20. Circumferential clip with four bolts.

Shear Loads in Bolted Connections

Shear loads in bolted connections are classified as either "Friction Type" or "Bearing Type". A brief definition is as follows;

Friction Type Connection

A friction type connection is one in which the sole purpose of the bolts is to provide adequate tension such that the two plates being joined will not slip. In this manner the bolts are not technically in shear, but solely in tension. High strength bolts must be used for friction type connections. The following factors are critical to the joint functioning as designed;

a. The condition of the surface finish of the plates being joined on the contacting surfaces.
   1. Uncoated (Class A, B or C)
   2. Hot dipped galvanized and roughened (Class D)
   3. Blast cleaned, zinc rich paint (Class E or F)
   4. Blast cleaned, metalized zinc or alum (Class G or H)
   5. Contact surfaces are coated (Class I)
b. The tightness of the joint based on;
   1. Snug Tightened
   2. Pre-tensioned
   3. Slip Critical
c. Size of hole relative to bolt size. Dimensions given in Table J3.3.
   1. STD: Standard round holes
   2. OVS: Oversize round holes
   3. SSL: Short slotted holes
   4. LSL: Long Slotted holes

Bearing Type Connection

A bearing type connection is one in which the bolts are in shear because there is not significant enough friction in the joint to prevent slip. There are two major classifications of bearing type connections;

a. Connections with threads in shear plane (Type N)
b. Connections without threads in shear plane (Type X)

The following bolt hole/slot types may be used with a bearing type connection;

a. STD: Standard round holes, d + 1/16"
b. NSL: Long Slotted holes

Paint is acceptable for all types of bearing connections.

The cost to install friction type connections is variable depending on the degree of labor required to produce the joint. In general, the relative costs of each type of joint is as follows (from cheapest to most expensive):

a. Snug tightened (X): 1.0
b. Pre-tensioned (X): 1.2
c. Snug tightened (N): 1.3
d. Pre-tensioned (N): 1.6
e. Slip Critical (N or X): 3.1

Procedure 9-6: Design of Bins and Elevated Tanks [3–9]

The definition of a "bulk storage container" can be quite subjective. The terms "bunkers," "hoppers," and "bins" are commonly used. This procedure is written specifically for cylindrical containers of liquid or bulk material with or without small internal pressures.

There is no set of standards that primarily applies to bins and since they are rarely designed for pressures greater than 15 psi, they do not require code stamps. They can, however, be designed, constructed, and inspected in accordance with certain sections of the ASME Code or combinations of codes.

When determining the structural requirements for bins, the horizontal and vertical force components on the bin walls must be computed. A simple but generally incorrect design method is to assume that the bin is filled with a fluid of the same density as the actual contents and then calculate the "equivalent" hydrostatic pressures. While this is correct for liquids, it is wrong for solid materials. All solid materials tend to bridge or arch, and this arch creates two force components on the bin walls.

The vertical component on the bin wall reduces the weight load on the material below, and pressures do not build up with the depth as much as in the case of liquids. Consequently, the hoop stresses caused by granular or powdered solids are much lower than for liquids of the same density. However, friction between the shell wall and the granular material can cause high longitudinal loads and even longitudinal buckling. These loads must be carefully considered in the case of a "deep bin."

In a "shallow bin," the contents will be entirely supported by the bin bottom. In a "deep bin" or "silo," the support will be shared, partly by the bottom and partly by the bin walls due to friction and arching of material.

Notation

A = cross-sectional area of bin, ft^2
A_r = area of reinforcement required, in.^{2}
A_a = area of reinforcement available, in.^{2}
A_s = cross-sectional area of strut, in.^{2}
e = common log 2.7183
C.A. = corrosion allowance, in.
E = joint efficiency, 0.35–1.0
F = summation of all vertical downward forces, lb
F_a = allowable compressive stress, psi
f = vertical reactions at support points, lb
h_i = depth of contents to point of evaluation, ft
K_1, K_2 = Rankine factors, ratio of lateral to vertical pressure
M = overturning moment, ft-lb
N = number of supports
P = internal pressure, psi
p_n = pressure normal to surface of cone, psf
p_v = vertical pressure of contents, psf
p_h = horizontal pressure on bin walls, psf
Q = total circumferential force, lb
R_h = hydraulic radius of bin, ft
S = allowable tension stress, psi
T_1, T_{1s} = longitudinal force, lb/ft
T_2, T_{2s} = circumferential force, lb/ft
G = specific gravity of contents
θ = angle of repose of contents, degrees
ϕ = angle of filling, angle of surcharge, friction angle. Equal to θ for free filling or 0 if filled flush, degrees
β = angle of rupture, degrees
μ = friction coefficient, material on material
μ' = friction coefficient, material on bin wall
Δh = height of filling peak, depth of emptying crater, ft
C_s = a function of the area of shell that acts with strut to A_s

Weights

W = total weight of bin contents, lb
w = density of contents, lb/cu ft
W_T = total weight of bin and contents, lb
W_c = weight of cone and lining below elevation under consideration, lb
W_R = D.L. + L.L. of roof plus applied loads, lb (include weight of any installed plant equipment)
W_s = weight of shell and lining (cylindrical portion only), lb
W_1 = $W + W_c$
W_2 = weight of contents in cylindrical portion of bin, lb, $= \pi R^2 H w$
W_3 = load caused by vertical pressure of contents, lb, $= p_v \pi R^2$

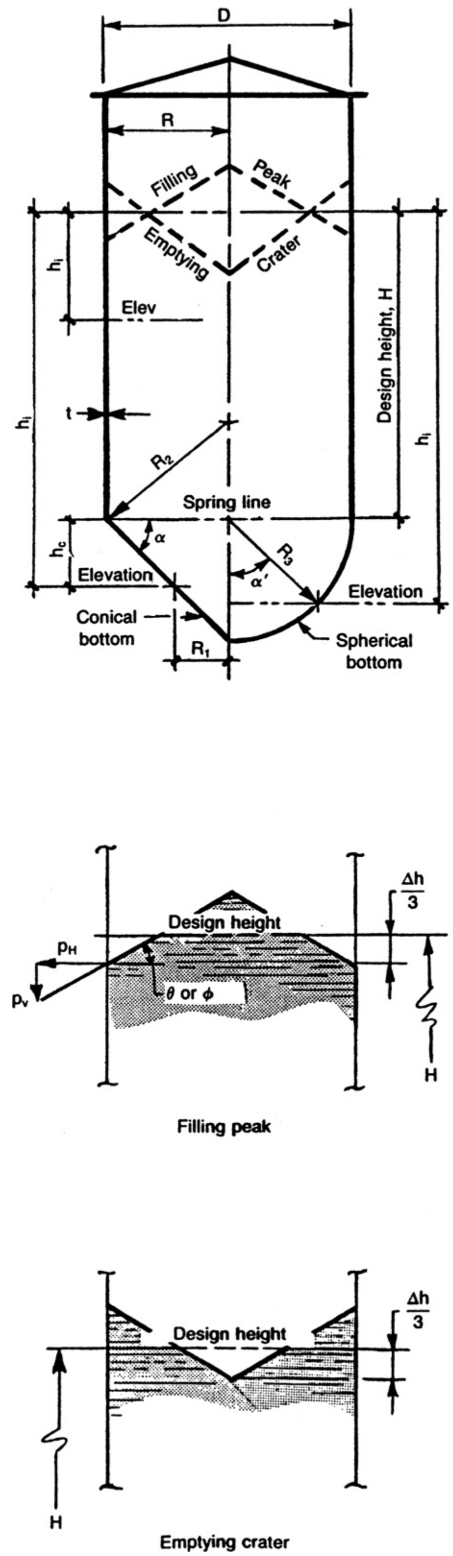


Figure 9-21. Dimensional data and forces of bin or elevated tank.

W_4 = portion of bin contents carried by bin walls due to friction, lb, $= W_2 - W_3$
$W_5 = W_R + W_4 + W_s$
$W_6 = W_T - W_c - W_{c1}$
W_7 = weight of bin below point of supports plus total weight of contents, lb
W_{c1} = weight of contents in bottom, lb

Bins

1. *Determine if bin is "deep" or "shallow."* The distinction between deep and shallow bins is as follows:
   - In a shallow bin the plane of rupture emerges from the top of the bin.
   - In a deep bin the plane of rupture intersects the opposite bin wall below both the top of the bin and/or the maximum depth of contents.

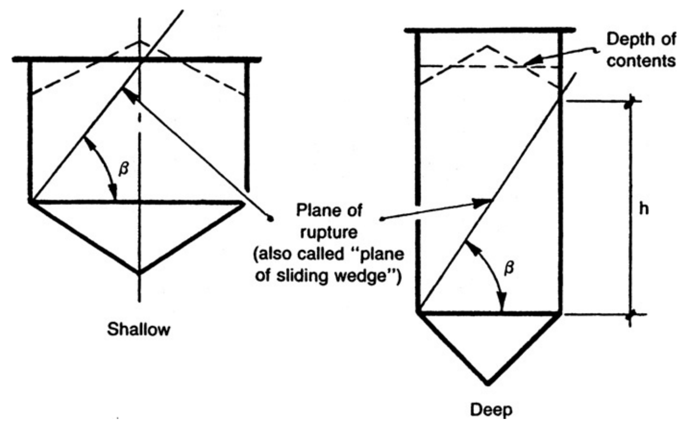


Figure 9-22. Examples illustrating the shallow vs. deep bin.

2. *Determine angle β.*

$$\tan\beta = \mu + \sqrt{\mu + \frac{1+\mu^2}{\mu+\mu'}}$$

If μ and μ' are unknown, compute β as follows:

$$\beta = \frac{90+\theta}{2}$$

and $h = D \tan\beta$.

If h is smaller than the straight side of the bin and below the design depth of the contents, the bin is assumed to be "deep" and the silo theory applies. If h is larger than the straight side of the bin or greater than the design depth of the contents, then the bin should be designed as "shallow." This design procedure is also known as the "sliding wedge" method.

Liquid-Filled Elevated Tanks

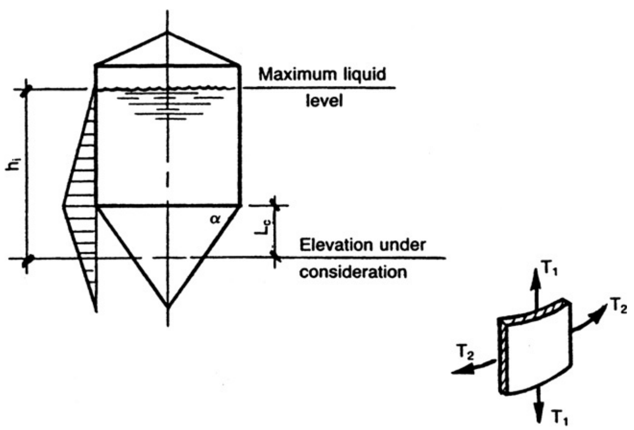


Figure 9-23. Dimensions and loads for a liquid-filled elevated tank.

- *Shell (API 650 & AWWA D100).*

$$t = \frac{2.6DHG}{SE} + C.A.$$

For A-36 material:
API 650: $S = 21{,}000$ psi
AWWA D100: $S = 15{,}000$ psi

- *Conical bottom (Wozniak).*
  At spring line,

$$T_1 = \frac{wR}{2\sin\alpha}\left(H + \frac{R\tan\alpha}{3}\right)$$

$$T_2 = \frac{wRH}{\sin\alpha}$$

At any elevation below spring line,

$$T_1 = \frac{w}{2\sin\alpha}\left(R - \frac{h_c}{\tan\alpha}\right)\left(H + \frac{2h_c}{3} + \frac{R\tan\alpha}{3}\right)$$

$$T_2 = \frac{wh_i}{\sin\alpha}\left(R - \frac{h_c}{\tan\alpha}\right)$$

$$t_c = \frac{(T_1 \text{ or } T_2)}{12SE\sin\alpha} + C.A.$$

- *Spherical bottom (Wozniak).*
  At spring line,

$$T_1 = wR_3\left[\frac{H}{2} + \frac{R_3}{3}\right]$$

$$T_2 = wR_3\left[\frac{H}{2} - \frac{R_3}{3}\right]$$

At bottom (max. stress),

$$T_1 = T_2 = \frac{wh_iR_3}{2}$$

$$t_s = \frac{(T_1 \text{ or } T_2)}{12SE} + C.A.$$

- *Ring compression at junction (Wozniak).*

$$Q = \frac{R^2w}{2\tan\alpha}\left(H + \frac{R\tan\alpha}{3}\right)$$

Shallow, Granular- or Powder-Filled Bin

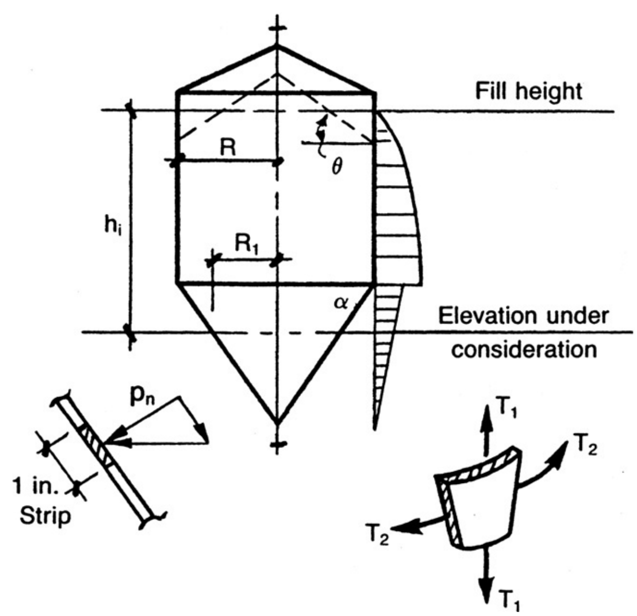


Figure 9-24. Dimensions and forces for a shallow bin.

- *Cylindrical Shell (Lambert).*

$$p_v = wh_i = \text{maximum at depth H}$$

$$K = K_1 \text{ or } K_2$$

$$P_h = p_vK\cos\phi$$

T_1 = compression only—from weight of shell, roof, and wind loads

Hoop tension, T_2, will govern design of shell for shallow bins

$$T_2 = p_h R$$

$$t = \frac{T_2}{12SE} + C.A.$$

- *Conical bottom (Ketchum).*

$$p_v = wh_i$$

Maximum at depth H =

$$p_n = \frac{p_v \sin^2(\alpha + \theta)}{\sin^3\alpha \left[1 + \dfrac{\sin\theta}{\sin\alpha}\right]^2}$$

$$W_1 = W + W_c$$

$$T_1 = \frac{W_1}{2\pi R_1 \sin\alpha}$$

$$T_2 = \frac{p_h R_1}{\sin\alpha}$$

$$t_c = \frac{T_1 \text{ or } T_2}{12SE} + C.A.$$

- *Spherical bottom (Ketchum).*

$$T_1 = T_2 = \frac{W_1}{2\pi R_3 \sin^2\alpha'}$$

Note: At $\alpha' = 90°$, $\sin^2\alpha' = 1$

$$t_s = \frac{T_1}{12SE} + C.A.$$

- *Ring compression (Wozniak).*

$$Q = T_1 R \cos\alpha$$

Deep Bins (Silo)—Granular/Powder Filled

- *Shell (Lambert).*

Hydraulic radius

$$R_h = \frac{R}{2}$$

- *Pressures on bin walls, p_v and p_h.*

$$K = K_1 \text{ or } K_2$$

$$e^{\left(\frac{-K\mu' h_i}{R_h}\right)}$$

e = common log 2.7183

$$p_v = \frac{wR_h}{\mu' K}\left[1 - e^{\left(\frac{-K\mu' h_i}{R_h}\right)}\right]$$

$$ph = p_v K$$

- *Weights.*

$$W_2 = \pi R^2 Hw$$
$$W_3 = p_v \pi R^2$$
$$W_4 = W_2 - W_3$$
$$W_5 = W_4 + W_R + W_s$$
$$W_R =$$
$$W_s =$$

- *Forces.*

$$T_1 = \frac{-W_5}{\pi D} - \frac{48M}{\pi D}$$

$$T_2 = p_h R$$

Note: For thin, circular steel bins, longitudinal compression will govern. The shell will fail by buckling from vertical drag rather than bursting due to hoop tension.

- *Maximum allowable compressive stress (Boardman formula).*

$$F_a = 2 \times 10^6 \left(\frac{t}{R}\right)\left(1 - \frac{100t}{3R}\right)$$

$$F_a = 10{,}000 \text{ psi maximum}$$

- *Thickness required shell, t.*

$$t = \frac{T}{12F_a}$$

- *Conical bottom (Ketchum).*

Note: Design bottoms to support full load of contents. Vibration will cause lack of side-wall friction.

At spring line,

$$p_v = wH$$

$$p_n = \frac{p_v \sin^2(\alpha + \theta)}{\sin^3\alpha \left[1 + \frac{\sin\theta}{\sin\alpha}\right]^2}$$

$$W_1 = W + W_c$$

$$T_1 = \frac{W_1}{2\pi R \sin\alpha}$$

$$T_2 = \frac{p_n R}{\sin\alpha}$$

$$t = \frac{(T_1 \text{ or } T_2)}{12SE} + \text{C.A.}$$

- *Spherical bottom (Ketchum).*

At spring line,

$$T_1 = T_2 = \frac{W_1}{2\pi R_3}$$

$$t = \frac{T_1}{12SE} + \text{C.A.}$$

- *Ring compression (Wozniak).*

$$Q = T_1 R \cos\alpha$$

Bins and Tanks with Small Internal Pressures

- *Pressures.*

P_1 = pressure due to gas pressure

P_2 = pressure due to static head of liquid

$$P_2 = \frac{wH}{144}$$

P_3 = pressure due to solid material

$$P_3 = \frac{wHK\cos\phi}{144}$$

P = total pressure

$$P = P_1 + P_2$$

or

$$P_1 + P_3 =$$

- *Shell (API 620).*

$$F = W_T$$

$$W_6 = W_T - W_c - W_{c1}$$

$$A = \pi R^2$$

$$T_{1s} = \frac{R}{2}\left(P + \frac{-W_6 + F}{A}\right)$$

$$T_{2s} = PR$$

$$t = \frac{(T_{1s} \text{ or } T_{2s})}{SE} + \text{C.A.}$$

- *Conical bottom (API 620).*

$$T_1 = \frac{R}{2\cos\alpha}\left(P + \frac{-W_6 + F}{A}\right)$$

$$T_2 = \frac{PR}{\sin\alpha}$$

$$t_c = \frac{(T_1 \text{ or } T_2)}{SE} + \text{C.A.}$$

- *Ring compression at spring line, Q (API 620).*

$$W_h = 0.6\sqrt{R_2(t_c - \text{C.A.})}$$

$$W_c = 0.6\sqrt{R(t - \text{C.A.})}$$

$$Q = T_2 W_h + T_{2s} W_c - T_1 R_2 \cos\alpha$$

Design of Compression Ring

Per API 620 the horizontal projection of the compression ring juncture shall have a width in a radial direction not less than 0.015 R. The compression ring may be used as a balcony girder (walkway) providing it is at least 3 ft-0 in. wide.

$$R_2 = \frac{R}{\sin\alpha}$$

$$W_h = 0.6\sqrt{R_2(t_c - \text{C.A.})}$$

$$W_c = 0.6\sqrt{R(t - \text{C.A.})}$$

Q = from applicable case =

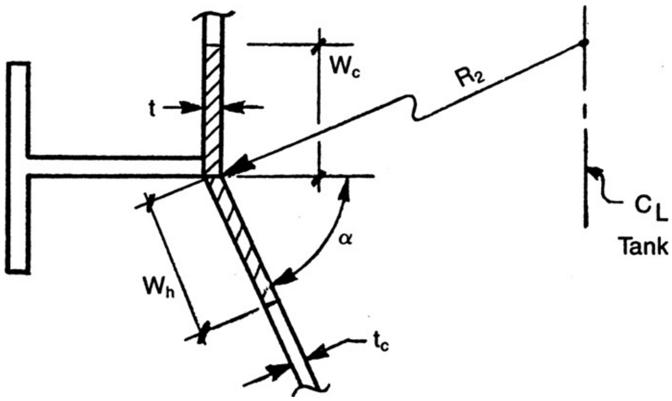


Figure 9-25. Dimensions at junction of cone and cylinder.

$$A_r = \frac{Q}{S}$$

$$A_a = W_c t + W_h t_c$$

- *Additional area required.*

$$A_r - A_a =$$

Struts

Strut are utilized to offset unfavorable high local stresses in the shell immediately above lugs when either lugs or rings are used to support the bin. These high localized stresses may cause local buckling or deformation if struts are not used.

- *Height of struts required, q.*

$$q = \frac{\pi R}{N}$$

- *Strut cross-sectional area required, A_s.*

$$A_s = \frac{fC_s}{S}$$

$$\text{where } f = \frac{W_7 R_f + 2M}{NR_f}$$

W_7 = weight of bin below point of supports plus total weight of contents, lb

The total cross-sectional area of single or double struts may be computed by this procedure. To determine C_s assume a value of A_s and a corresponding value of C_s from Figure 9-27. Substitute this value of C_s into the area equation and compute the area required. Repeat this procedure until the proposed A_s and calculated A_s are in agreement.

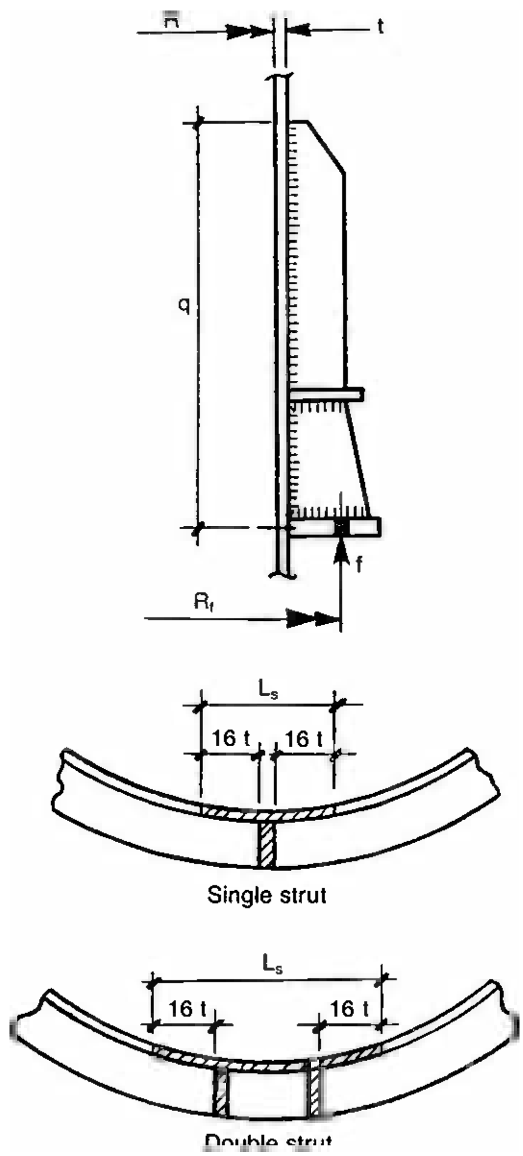


Figure 9-26. Dimensions and arrangement of single and double struts.

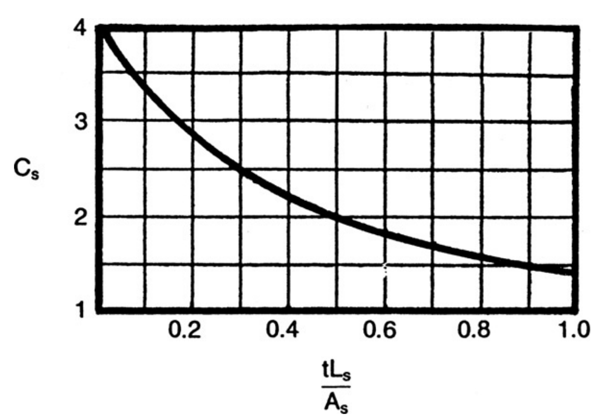


Figure 9-27. Graph of function C_s.

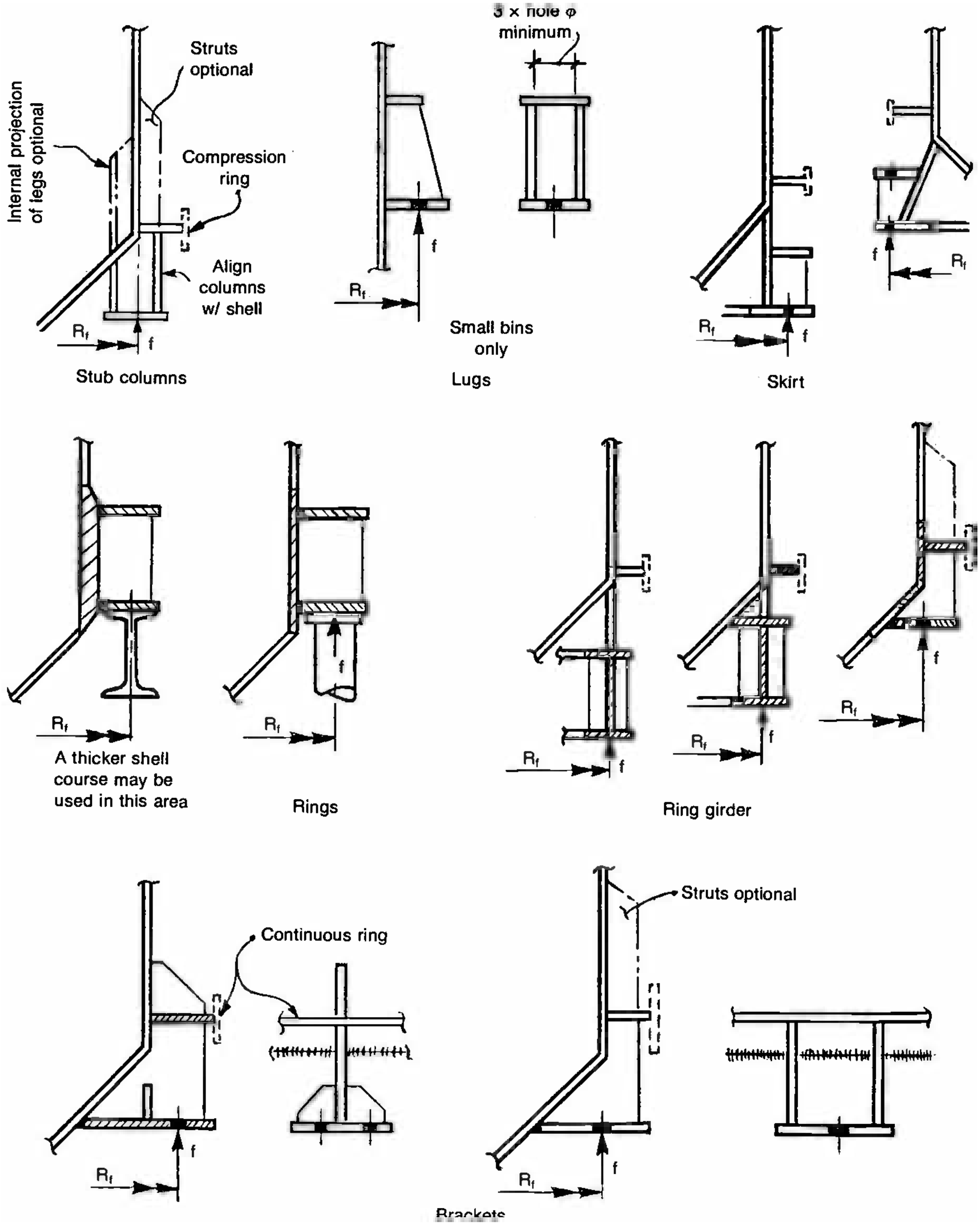


Figure 9-28. Typical support arrangements for bins and elevated tanks.

Supports

Bins may be supported in a variety of ways. Since the bottom cone-cylinder intersection normally requires a compression ring, it is common practice to combine the supports with this ring. This will take advantage of the local stiffness and is convenient for the support design.

Table 9-10
Material properties

Coefficients of Friction							
				μ' − Contents on Wall			
				On Steel		On Concrete	
Material	Density w	Angle of Repose θ	Contents on Contents μ	μ'	ϕ	μ'	ϕ
Portland cement	90	39°	0.32	0.93		0.54	
Coal (bituminous)	45–55	35°	0.70	0.59	25	0.70	35
Coal (anthracite)	52	27°	0.51	0.45	22	0.51	27
Coke (dry)	28	30°	0.58	0.55	20	0.84	20
Sand	90–110	30°–35°	0.67	0.60	20	0.58	30
Wheat	50–53	25°–28°	0.47	0.41		0.44	
Ash	45	40°	0.84	0.70	25	0.70	35
Clay—dry, fine	100–120	35°	0.70	0.70			
Stone, crushed	100–110	32°–39°	0.70	0.60			
Bauxite ore	85	35°	0.70	0.70			
Corn	44	27.5°	0.52	0.37		0.42	
Peas	50	25°	0.47	0.37		0.44	

If μ' is unknown it may be estimated as follows:

- Mean particle diameter $<$0.002 in., $\tan^{-1}\mu' = \theta$.
- Mean particle diameter $>$0.008 in., $\tan^{-1}\mu' = 0.75\,\theta$.

Table 9-11
Rankine factors K_1 and K_2

		Values of K_2 for angles ϕ							
θ	K1	10°	15°	20°	25°	30°	35°	40°	45°
10°	0.7041	1.0000							
12°	0.6558	0.7919							
15°	0.5888	0.6738	1.0000						
17°	0.5475	0.6144	0.7532						
20°	0.4903	0.5394	0.6241	1.0000					
22°	0.4549	0.4958	0.5620	0.7203					
25°	0.4059	0.4376	0.4860	0.5820	1.0000				
27°	0.3755	0.4195	0.4428	0.5178	0.6906				
30°	0.3333	0.3549	0.3743	0.4408	0.5446	1.0000			
35°	0.2709	0.2861	0.3073	0.3423	0.4007	0.5099	1.0000		
40°	0.2174	0.2282	0.2429	0.2665	0.3034	0.3638	0.4549	1.0000	
45°	0.1716	0.1792	0.1896	0.2058	0.2304	0.2679	0.3291	0.4444	1.0000

K_1, no surcharge *K_2, with surcharge*

$$K_1 = \frac{p_h}{p_v} = \frac{1-\sin\theta}{1+\sin\theta} = \quad K_2 = \frac{\cos\phi - \sqrt{\cos^2\phi - \cos^2\theta}}{\cos\phi + \sqrt{\cos^2\phi - \cos^2\theta}} =$$

Notes

1. Rankine factors K_1 and K_2 are ratios of horizontal to vertical pressures. These factors take into account the distribution of forces based on the filling and emptying properties of the material. If the filling angle is different from the angle of repose, then K_2 is used. Remember, even if the material is not heaped to begin with, a crater will develop when emptying. The heaping, filling peak, and emptying crater all affect the distribution of forces.
2. Supports for bins should be designed by an appropriate design procedure. See Chapter 4.
3. In order to assist in the flow of material, the cone angle should be as steep as possible. An angle of 45° can be considered as minimum, 50°–60° preferred.
4. While roofs are not addressed in this procedure, their design loads must be considered since they are translated to the shell and supports. As a minimum, allow 25 psf dead load and 50–75 psf live load plus the weight of any installed plant equipment (mixers, conveyors, etc.).
5. Purging, fluidizing techniques, and general vibration can cause loss of friction between the bin wall and the contents. Therefore its effect must be considered or ignored in accordance with the worst situation: in general, added to longitudinal loads and ignored for circumferential loads.
6. Surcharge: Most bunkers will be surcharged as a result of the normal filling process. If the surcharge is taken into account, the horizontal pressures will be overestimated for average bins. It is therefore more economical to assume the material to be flat and level at the mean height of the surcharge and to design accordingly. Where the bin is very wide in relation to the depth of contents the effects of surcharging need to be considered.

Procedure 9-7: Field-Fabricated Spheres

A sphere is the most efficient pressure vessel because it offers the maximum volume for the least surface area and the required thickness of a sphere is one-half the thickness of a cylinder of the same diameter. The stresses in a sphere are equal in each of the major axes, ignoring the effects of supports. In terms of weight, the proportions are similar. When compared with a cylindrical vessel, for a given volume, a sphere would weigh approximately only half as much. However, spheres are more expensive to fabricate, so they aren't used extensively until larger sizes. In the larger sizes, the higher costs of fabrication are balanced out by larger volumes.

Spheres are typically utilized as "storage" vessels rather than "process" vessels. Spheres are economical for the storage of volatile liquids and gases under pressure, the design pressure being based on some marginal allowance above the vapor pressure of the contents. Spheres are also used for cryogenic applications for the storage of liquified gases.

Products Stored

- Volatile liquids and gases: propane, butane, and natural gas.
- Cryogenic: oxygen, nitrogen, hydrogen, ethylene, helium, and argon.

Codes of Construction

Spheres are built according to ASME, Section VIII, Division 1 or 2, API 620 or BS 5500. In the United States, ASME, Section VIII, Division 1 is the most commonly used code of construction. Internationally spheres are often designed to a higher stress basis upon agreement between the user and the jurisdictional authorities. Spheres below 15 psig design pressure are designed and built to API 620.

The allowable stresses for the design of the supports is based on either AWWA D100 or AISC.

Materials of Construction (MOC)

Typical materials are carbon steel, usually SA-516-70. High-strength steels are commonly used as well (SA-537, Class 1 and 2, and SA-738, Grade B). SA-516-60 may be used to eliminate the need for PWHT in wet H_2S service. For cryogenic applications, the full range of materials has been utilized, from the low-nickel steels, stainless steels, and higher alloys. Spheres of aluminum have also been fabricated.

Liquified gases such as ethylene, oxygen, nitrogen, and hydrogen are typically stored in double-wall spheres, where the inner tank is suspended from the outer tank by straps or cables and the annular space between the tanks is filled with insulation. The outer tank is not subjected to the freezing temperatures and is thus designed as a standard carbon steel sphere.

Size, Thickness, and Capacity Range

Standard sizes range from 1000 barrels to 50,000 barrels in capacity. This relates in size from about 20 feet to 82 feet in diameter. Larger spheres have been built but are considered special designs. In general, thicknesses are limited to 1.5 in. to preclude the requirement for PWHT, however PWHT can be accomplished, even on very large spheres.

Supports

Above approximately 20 feet in diameter, spheres are generally supported on legs or columns evenly spaced around the circumference. The legs are attached at or near the equator. The plates in this zone of leg attachment may be required to be thicker, to compensate for the additional loads imposed on the shell by the supports. An internal stiffening ring or ring girder is often used at the junction of the centerline of columns and the shell to take up the loads imposed by the legs.

The quantity of legs will vary. For gas-filled spheres, assume one leg every third plate, assuming 10-feet-wide plates. For liquid-filled spheres, assume one leg every other plate.

Legs can be either cross-braced or sway-braced. Of the two bracing methods, sway-bracing is the more common. Sway-bracing is for tension-only members. Cross-bracing is used for tension and compression members. When used, cross-bracing is usually pinned at the center to reduce the sizes of the members in compression.

Smaller spheres, less than 20 feet in diameter, can be supported on a skirt. The diameter of the supporting skirt should be $0.7 \times$ the sphere diameter.

Heat Treatment

Carbon steel spheres above 1.5-in. thickness must be PWHT per ASME Code. Other alloys should be checked for thickness requirements. Spheres are often stress relieved for process reasons. Spheres made of high-strength carbon steel in wet H_2S service should be stress relieved regardless of thickness. When PWHT is required, the following precautions should be taken:

a. Loosen cross-bracing to allow for expansion.
b. Jack out columns to keep them level during heating and cooling.
c. Scaffold the entire vessel.
d. Weld thermocouple wires to shell external surface to monitor and record temperature.
e. Typically, internally fire it.
f. Monitor heat/cooling rate and differential temperature.

Accessories

Accessories should include a spiral stairway and a top platform to access instruments, relief valves, and vents. Manways should be used on both the top and bottom of the sphere. Nozzles should be kept as close as practical to the center of the sphere to minimize platforming requirements.

Methods of Fabrication

Field-fabricated spheres are made in one of two methods. Smaller spheres can be made by the expanded cube, soccer ball method, while larger ones are made by the orange peel method. The orange peel method consists of petals and cap plates top and bottom.

Typically all shell pieces are pressed and trimmed in the shop and assembled to the maximum shipping sizes allowable. Often, the top portion of the posts are fit up and welded in the shop to their respective petals.

Field Hydrotests

Typically the bracing on the support columns is not tightened fully until the hydrotest. While the sphere is full of water and the legs are at their maximum compression, the bracing is tightened so that once the sphere is emptied, all of the bracing goes into tension and there is the assurance that they remain in tension during service.

Settlement between the legs must be monitored during hydrotest to detect any uneven settlement between the posts. Any uneven settlement of over ½ in. between any pair of adjacent legs can cause distortion and damage to the sphere. Foundation requirements should take this requirement into consideration.

Notes

1. Spheres that operate either hot or cold will expand or contract differentially with respect to the support

columns or posts. The moment and shear forces resulting from this differential expansion must be accounted for in the design of the legs.

2. The minimum clearance between the bottom of the vessel and grade is 2ft 6in.
3. The weights shown in the tables include the weight of the sphere with an allowance for thinning ($^1/_{16}$ in.) and corrosion (1/8 in.) plus plate overtolerance. A clearance of 3 ft was assumed between the bottom of the sphere and the bottom of the base plate. The weights include columns, base plates, and bracing, plus a spiral stairway and top platform. Column weights were estimated from the quantities and sizes listed in the table.
4. For estimating purposes, the following percentages of the sphere shell weight should be added for the various categories:
   - Columns and base plates: 6–14%. For thicker, heavier spheres, the lower percentage should be used. For larger, thinner spheres, the higher percentage should be used.
   - Sway rods/bracing: 1–9%. Use the lower value for wind only and higher values where seismic governs. The highest value should be used for the highest seismic area.
   - Stairway, platform, and nozzles: 2–5%. Apply the lower value for minimal requirements and the higher where the requirements are more stringent.

Notation

A = surface area, sq ft
d = OD of column legs, in.
D = diameter, ft
D_m = mean vessel diameter, ft
E = joint efficiency
E_m = modulus of elasticity, psi
N = number of support columns
n = number of equal volumes
P = internal pressure, psig
P_a = maximum allowable external pressure, psi
P_m = MAWP, psig
R = radius, ft
R_c = radius, corroded, in
S = allowable stress, psi
t = thickness, new, in.
t_c = thickness, corroded, in.
t_p = thickness of pipe leg, in.
t_{rv} = thickness required for full vacuum, in.
V = volume, cu ft
W = weight, lb
w = unit weight of plate, psf

Conversion Factors

7.481 gallons/cu ft
0.1781 barrels/cu ft
5.614 cu ft/barrel
35.31 cu ft/cu meter
6.29 barrels/cu meter
42 gallons/barrel

Formulas

$$V = \frac{\pi D^3}{6} \quad \text{or} \quad V = \frac{4\pi R^3}{3}$$

$$V_n = \frac{\pi D^3}{6n} \quad \text{or} \quad V_n = \frac{4\pi R^3}{3n}$$

$$V_1 = \frac{\pi\, h_1^2}{3}(3R - h_1)$$

$$V_2 = \frac{\pi h_1}{6}\left(3r_1^2 + 3r_2^2 + h_2^2\right)$$

$$D = \sqrt[3]{\frac{6V}{\pi}}$$

$$A = \pi D^2 \quad \text{or} \quad A = 4\pi R^2$$

$$A_n = \pi D h_n \quad \text{or} \quad A_n = 2\pi R h_n$$

$$r_1 = \sqrt{2Rh_1 - h_1^2}$$

$$r_2 = \sqrt{R^2 - h_3^2}$$

$$\sin\alpha = \frac{r_1}{R} \quad \alpha$$

$$W = \pi D_m^2 w$$

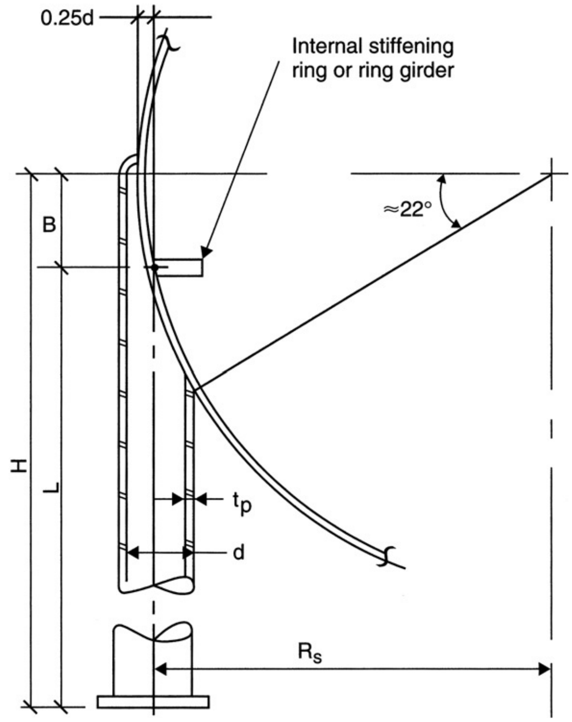
$$P_m = \frac{2SEt_c}{R_i + 0.2t_c}$$

$$P_a = \frac{0.0625E_m}{\left(\frac{R_o}{t_c}\right)^2}$$

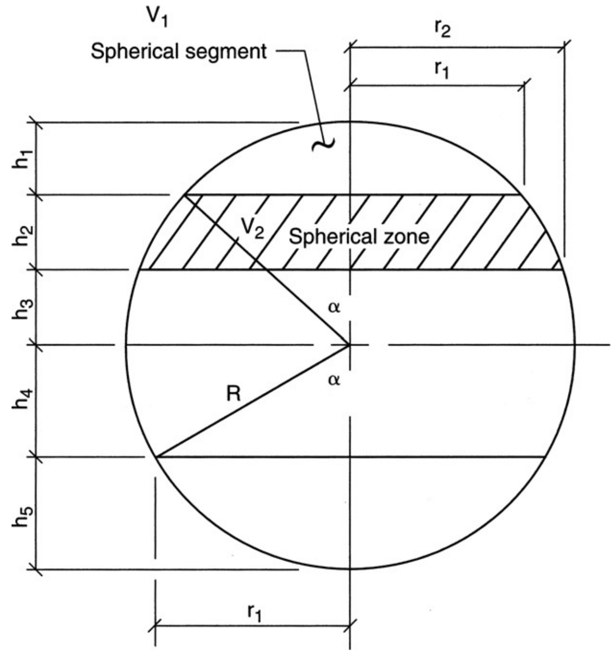
$$t_r = \frac{PR_c}{2SE - 0.2P} \text{ (Division 1)}$$

$$t_r = R_c\left(e^{0.5P/SE} - 1\right) \text{ (Division 2)}$$

Typical Leg Attachment



Dimensional Data



Liquid Level in a Sphere

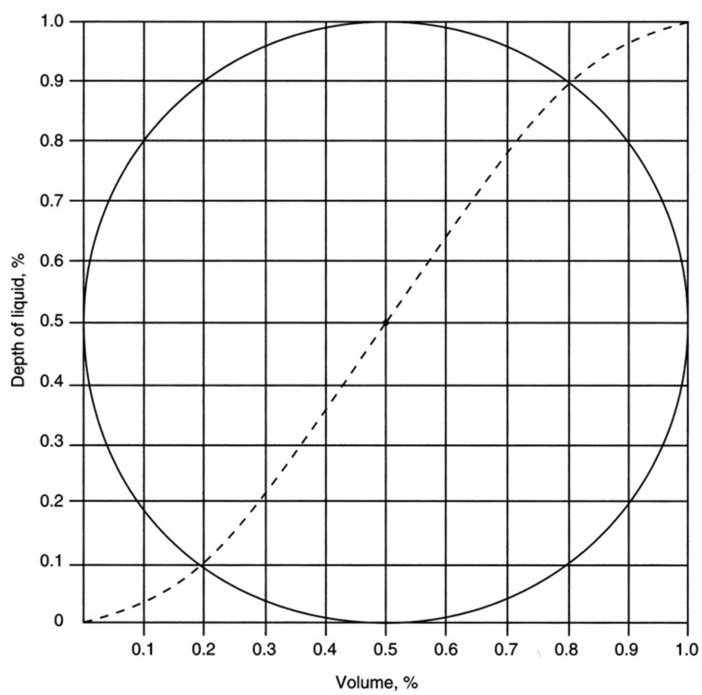


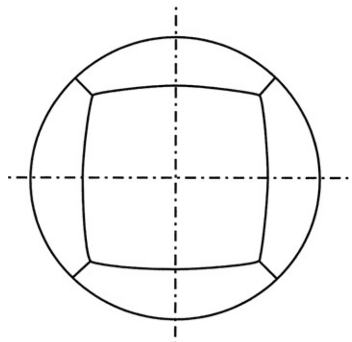
Table 9-12
Dimensions for "n" quantity of equal volumes

Figure	n	V_n	r_1	r_2	h_1	h_2	h_3
	3	$\frac{\pi D^3}{18}$	0.487D	—	0.387D	0.226D	—
	4	$\frac{\pi D^3}{24}$	0.469D	—	0.326D	0.174D	—
	5	$\frac{\pi D^3}{30}$	0.453D	0.496D	0.287D	0.146D	0.067D
	6	$\frac{\pi D^3}{36}$	0.436D	0.487D	0.254D	0.133D	0.113D

Table 9-13
Volumes and surface areas for various depths of liquid

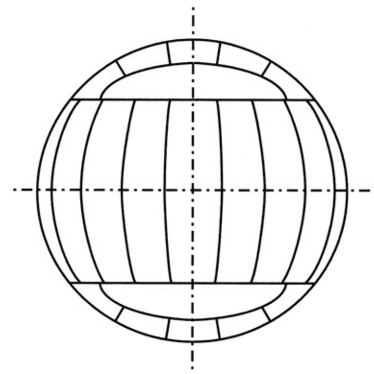
h_4	h_5	α	r_1	V_5	V_4	A_5	A_4
0.05D	0.45D	25.84	0.218D	$0.0038D^3$	$0.2580D^3$	$0.1571D^2$	$1.4137D^2$
0.10D	0.40D	36.87	0.300D	$0.0147D^3$	$0.2471D^3$	$0.3142D^2$	$1.2567D^2$
0.15D	0.35D	45.57	0.357D	$0.0318D^3$	$0.2300D^3$	$0.4712D^2$	$1.1000D^2$
0.20D	0.30D	53.13	0.400D	$0.0545D^3$	$0.2073D^3$	$0.6283D^2$	$0.9425D^2$
0.25D	0.25D	60.0	0.433D	$0.0818D^3$	$0.1800D^3$	$0.7854D^2$	$0.7854D^2$
0.30D	0.20D	66.42	0.458D	$0.1131D^3$	$0.1487D^3$	$0.9425D^2$	$0.6283D^2$
0.35D	0.15D	72.54	0.477D	$0.1475D^3$	$0.1143D^3$	$1.1000D^2$	$0.4712D^2$
0.40D	0.10D	78.46	0.490D	$0.1843D^3$	$0.0775D^3$	$1.2567D^2$	$0.3141D^2$
0.45D	0.05D	84.26	0.498D	$0.2227D^3$	$0.0391D^3$	$1.4137D^2$	$0.1571D^2$
0.50D	0D	90.0	0.500D	$0.2618D^3$	$0D^3$	$1.5708D^2$	$0D^2$

Types of Spheres



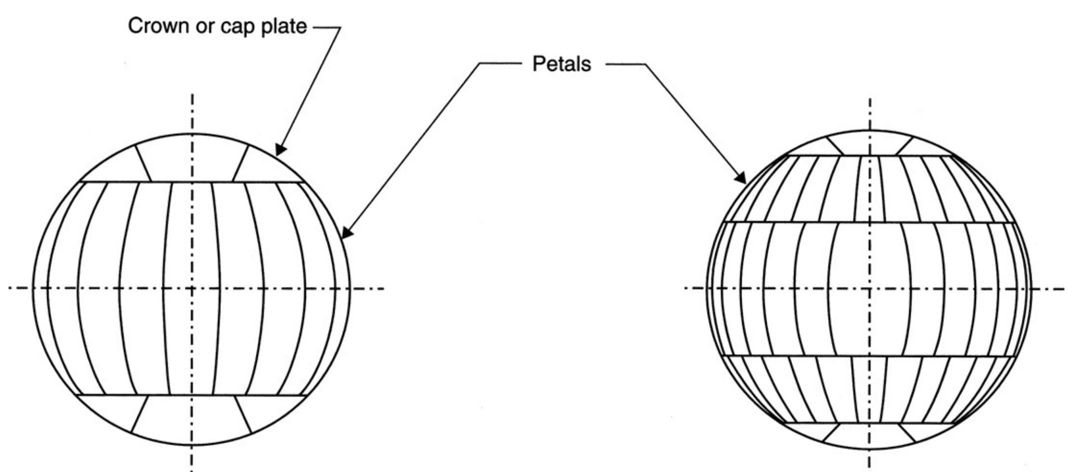
Expanded Cube, Square Segment, or Soccer Ball Type

- Small spheres only
- Sizes less than about 20 feet in diameter
- Volumes less than 750 bbls



Partial Soccer Ball Type

- Combines orange peel and soccer ball types
- Sizes 30 to 62 feet in diameter
- Volumes 2200 to 22,000 bbls



Meridian, Orange Peel, or Watermelon Type (3-Course Version)

- Consists of crown plates and petal plates
- Sizes 20 to 32 feet in diameter
- Volumes 750 to 3000 bbls

Meridian, Orange Peel, or Watermelon Type (5-Course Version)

- Consists of crown plates and petal plates
- Sizes up to 62 feet in diameter
- Volumes to 22,000 bbls

Table 9-14
Data for 50-psig sphere

D	t	Volume bbl—nom	bbls	ft^3	A	W	N	d	t_p	P_q	t_{rv}
20 ft-0 in.	0.3125	750	746	4188	1256	23.5	4	16	0.25	4.4	0.5
22 ft-3 in.	0.375	1000	1027	5767	1555	32.8	4	16	0.25	6.32	0.5625
25 ft-0 in.	0.375	1500	1457	8181	1963	41	4	16	0.25	5.01	0.5625
25 ft-6 in.	0.375	1500	1546	8682	2043	42.7	4	16	0.25	4.82	0.5625
28 ft-0 in.	0.375	2000	2047	11,494	2463	52.2	5	16	0.25	4	0.625
30 ft-3 in.	0.4375	2500	2581	14,494	2875	68.8	5	16	0.25	5.35	0.6875
32 ft-0 in.	0.4375	3000	3055	17,157	3217	78	6	18	0.25	4.78	0.6875
35 ft-0 in.	0.4375	3000	3998	22,449	3848	93.4	6	18	0.25	4	0.75
35 ft-3 in.	0.4375	4000	4084	22,934	3904	94.7	6	20	0.25	2.52	0.75
38 ft-0 in.	0.5	5000	5116	28,731	4536	123	6	22	0.25	4.88	0.8125
40 ft-0 in.	0.5	6000	5968	33,510	5027	138	6	22	0.25	4.41	0.8125
40 ft-6 in.	0.5	6000	6195	34,783	5153	142.3	7	24	0.25	4.3	0.875
43 ft-6 in.	0.5625	7500	7676	43,099	5945	181	7	24	0.29	5.07	0.875
45 ft-0 in.	0.5625	8500	8497	47,713	6362	193.6	7	24	0.29	4.74	0.9375
48 ft-0 in.	0.5625	10,000	10,313	57,906	7238	222.2	8	28	0.3	4.17	1
50 ft-0 in.	0.625	11,500	11,656	65,450	7854	269.4	8	28	0.3	5.01	1
51 ft-0 in.	0.625	12,500	12,370	69,456	8171	280.2	9	30	0.29	4.82	1
54 ft-9 in.	0.625	15,000	15,304	85,931	9417	326.8	9	32	0.344	4.18	1.0625
55 ft-0 in.	0.625	15,000	15,515	87,114	9503	330.6	9	32	0.344	4.15	1.125
60 ft-0 in.	0.6875	20,000	20,142	113,097	11,310	430.5	9	32	0.344	4.41	1.1875
60 ft-6 in.	0.6875	20,000	20,650	115,948	11,500	438.2	10	34	0.38	4.34	1.1875
62 ft-0 in.	0.6875	22,000	22,225	124,788	12,076	458.8	10	34	0.38	4.13	1.25
65 ft-0 in.	0.75	25,000	25,610	143,793	13,273	551.5	11	36	0.406	4.64	1.25
69 ft-0 in.	0.75	30,000	30,634	172,007	14,957	629.2	11	40	0.438	4.12	1.375
76 ft-0 in.	0.8125	40,000	40,936	229,847	18,146	874.1	12	42	0.503	4.11	1.5
81 ft-10 in.	0.875	50,000	51,104	286,939	21,038	1105	13	42	0.594	3.54	1.625
87 ft-0 in.	0.9375	60,000	61,407	344,791	23,779	1460	14	48	0.75	4.38	1.75

Note: Values are based on the following:

1. Material SA-516-70, S = 20,000 psi.
2. Joint efficiency, E = 0.85.
3. Corrosion allowance, c.a. = 0.125.

Table 9-15
Weights of spheres, kips

Dia. (ft)	Thickness (in.)										
	0.375	**0.4375**	**0.5**	**0.5625**	**0.625**	**0.6875**	**0.75**	**0.8125**	**0.875**	**0.9375**	**1**	**1.125**
20 ft-0 in.	26.8	30	[33.3]	36.5	39.8	43	46.3	49.5	52.7	55.9		
22 ft-6 in.	32.8	36.8	40.9	[45]	49	53.1	57.2	61.2	65.3	69.3		
25 ft-0 in.	41	46	51	[56]	61	66.1	71.1	76.1	81.1	86		
27 ft-6 in.	***48***	54.1	60.1	66.2	[72.3]	78.3	84.4	90.4	96.5	103		
30 ft-0 in.	60	66	73.2	80.4	87.6	[94.8]	102	109	117	124	131	
32 ft-6 in.	71.5	80	88.5	97	105	[114]	122	131	139	148	156	
35 ft-0 in.	81.1	***93.4***	103	113	123	133	[143]	152	162	172	182	202
37 ft-6 in.	98.3	110	121	132	143	155	166	[177]	189	200	211	234
40 ft-0 in.	105	122	***138***	151	164	177	189	[202]	215	228	241	266
42 ft-6 in.	129	143	158	172	187	201	216	230	[245]	259	274	303
45 ft-0 in.	145	161	177	194	210	226	242	259	275	[291]	307	340
47 ft-6 in.	161	179	197	***215***	233	251	269	287	305	324	[342]	378
50 ft-0 in.		209	229	249	269	289	309	330	350	370	[390]	430
52 ft-6 in.		234	256	278	300	322	344	366	388	411	433	[477]
55 ft-0 in.			282	306	***331***	355	379	403	428	452	476	[525]
57 ft-6 in.			313	340	366	393	419	446	472	499	525	578
60 ft-0 in.				373	402	431	459	488	517	546	575	633
62 ft-6 in.				399	431	***462***	493	525	556	587	619	650
65 ft-0 in.					484	518	552	585	619	653	687	755
69 ft-0 in.					553	591	***629***	667	706	744	782	858
76 ft-0 in.						782	828	***874***	920	967	1013	1106
81 ft-10 in.						944	998	1051	***1105***	1159	1212	1320
87 ft-0 in.							1278	1339	1400	***1460***	1521	1642

Notes:

1. Values that are underlined indicate 50-psig internal pressure design.
2. Values in brackets [ ] indicate full vacuum design.

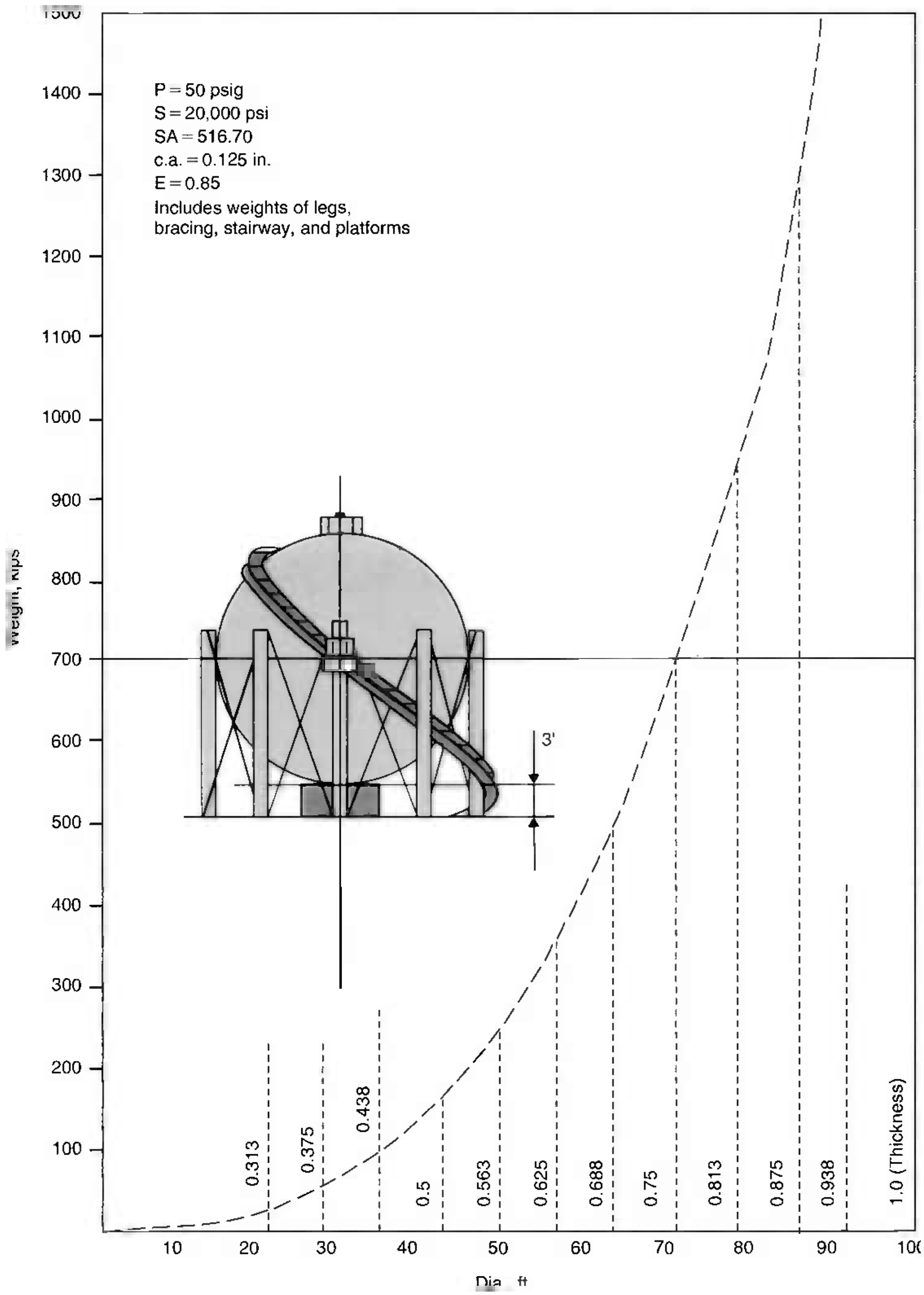
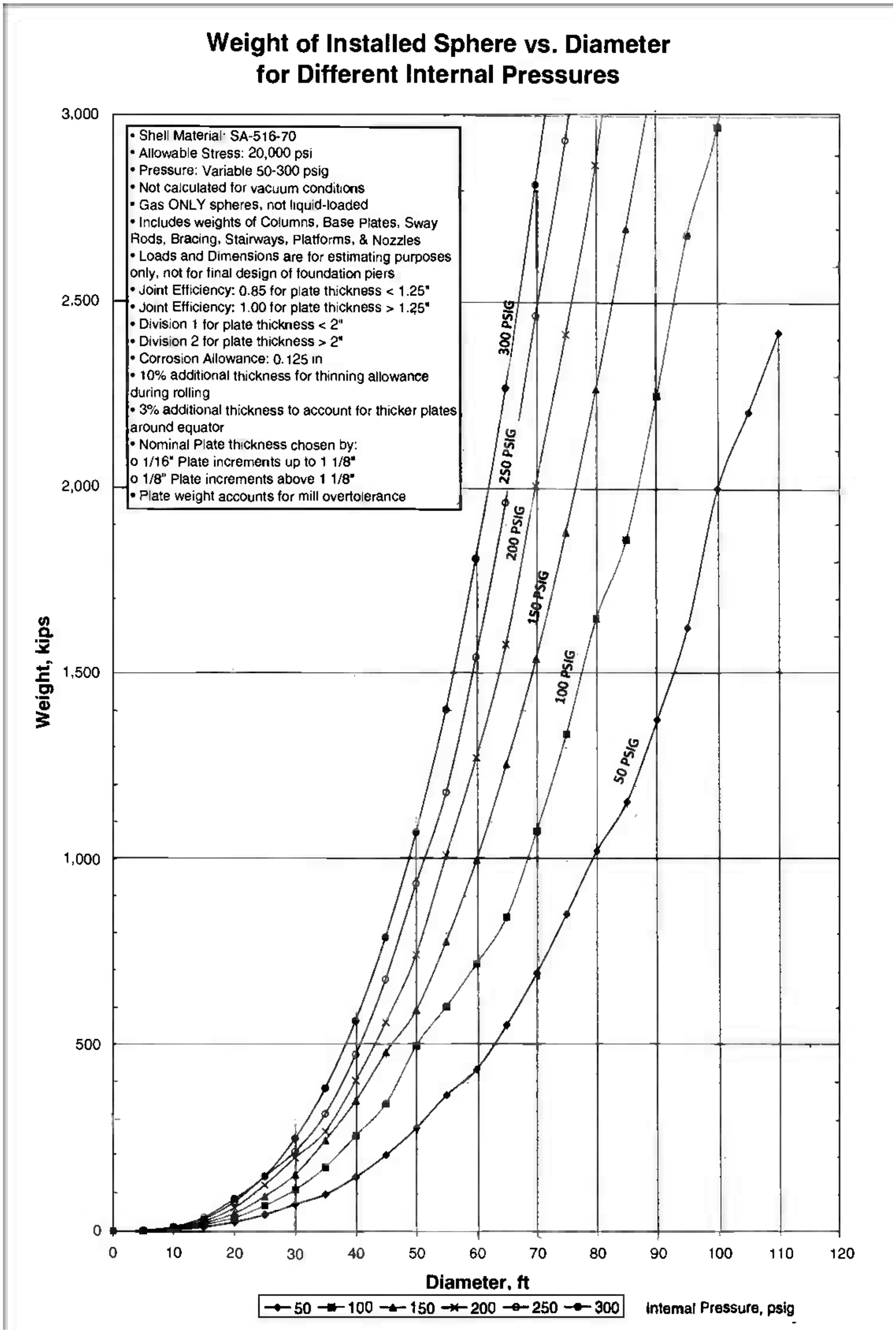


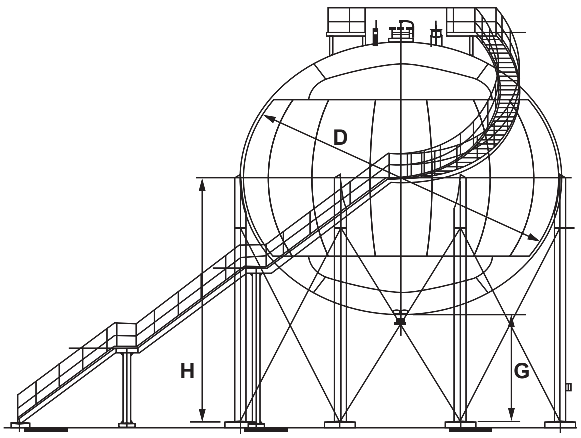
Figure 9-29. Weight of sphere.


Weight of Installed Sphere vs. Diameter
for Different Internal Pressures
• Shell Material: SA-516-70
• Allowable Stress: 20,000 psi
• Pressure: Variable 50-300 psig
• Not calculated for vacuum conditions
• Gas ONLY spheres, not liquid-loaded
• Includes weights of Columns, Base Plates, Sway Rods, Bracing, Stairways, Platforms, & Nozzles
• Loads and Dimensions are for estimating purposes only, not for final design of foundation piers
• Joint Efficiency: 0.85 for plate thickness < 1.25"
• Joint Efficiency: 1.00 for plate thickness > 1.25"
• Division 1 for plate thickness < 2"
• Division 2 for plate thickness > 2"
• Corrosion Allowance: 0.125 in
• 10% additional thickness for thinning allowance during rolling
• 3% additional thickness to account for thicker plates around equator
• Nominal Plate thickness chosen by:
o 1/16" Plate increments up to 1 1/8"
o 1/8" Plate increments above 1 1/8"
• Plate weight accounts for mill overtolerance
300 PSIG
250 PSIG
200 PSIG
150 PSIG
100 PSIG
50 PSIG
Weight, kips
3,000
2,500
2,000
1,500
1,000
500
0
0 10 20 30 40 50 60 70 80 90 100 110 120
Diameter, ft
50 100 150 200 250 300
Internal Pressure, psig

SPHERE DATA SHEET

EQUIPMENT ITEM NO. :

EQUIPMENT NAME :



DIMENSIONS	
D	
H	
G	

WEIGHTS	
FABRICATED	
EMPTY	
OPERATING	
TEST	
SHELL	
COLUMNS	
BRACING	
CONTENTS	
WATER	
FIREPROOFING	

DESIGN DATA		MATERIALS	
Design Pressure - Internal		Shell	
Design Pressure - External		Columns	
Design Temperature - Internal		Column Bracing	
Design Temperature - External		Flanges	
MDMT		Nozzle Necks- Pipe	
Specific Gravity		Bolting	
Capacity		Gaskets	
Corrosion Allowance			
Joint Efficiency		Base Plates	
PWHT		**CODES**	
Contents		Design	
Insulation/Thk		Seismic	
Fireproofing - Legs		Wind	
Service (Sour, Lethal, Cyclic, etc.)		Structural	

Design of Spheres

Nomenclature

A_b = Area, brace, in^2
A_c = Area, column, in^2
A_{cr} = Area required, column
A_s = Area, shell, in^2
A_g = Area, girder, in^2
A_T = Area, total, in^2
A_{br} = Area, brace, required, in^2
A_{sn} = Surface area of shell section, Ft^2
A_{cn} = Cross sectional area, Ft^2
C_a = Corrosion allowance, in
D_c = Centerline diameter of columns, Ft
d_c = Inside diameter of column, in
E = Joint efficiency
E_m = Modulus of elasticity, PSI
f = Maximum force in brace, Lbs
f_a = Axial stress, compression, PSI
f_T = Tension stress, PSI
F_a = Allowable axial stress, PSI
F_b = Allowable stress, bending, PSI
F_c = Allowable stress, compression, PSI
F_D = Axial load on column due to dead weight, Lbs
F_L = Axial load on column due to live load, ie. wind or seismic, lbs
F_T = Alowable stress, tension, psi
F_y = Yield strength of material at temperature, PSI
g = Acceleration due to gravity, 386 in/sec^2
I_b = Moment of inertia, bracing, in^4
I_r = Required moment of inertia, in^4
I_g = Moment of inertia, girder, in^4
I_s = Moment of inertia, shell, in^4
I_T = Moment of inertia, combined shell and girder total, in^4
I_c = Moment of inertia, column, in^4
k = End connection coefficient, columns
L' = Theoretical length of shell resisting loads, in
M_o = Overturning moment, Ft-Lbs
M_B = Internal bending moment in girder section between columns due to horizontal force, in-Lbs
M_C = Internal bending moment in girder section between columns due to vertical force, in-Lbs
M_P = Internal bending moment in post plate at column due to vertical force, in-Lbs
M_S = Internal bending moment in post plate between columns due to vertical force, in-Lbs
N = Number of columns
n = Number of active rods per panel use 1 for sway bracing; 2 for cross bracing
n' = Factor for cross bracing, use 1 for unpinned, 2 for pinned at center
P = Internal pressure, PSIG
P_x = External pressure, PSIG
P_h = Pressure due to static head of liquid, PSI
P_n = Total pressure at a given elevation, n, PSI
Q = Maximum axial force in column, Lbs
R = Inside radius, corroded, Ft
r = Inside radius, corroded, in
r_c = Radius of gyration, column, in
r_b = Radius of gyration, brace, in
r_o = Outside radius of sphere, in
r_g = Radius of gyration, girder, in
S = Allowable stress, shell, PSI
S_C = Combined stress, compression, PSI
S_g = Specific gravity, contents
S_r = Slenderness ratio
S_T = Combined stress, tension, PSI
T = Period of vibration, Sec's
T' = Greater of T_1 or T_2, Lbs / in
T_1 = Meridional load, Lb/in
T_2 = Latitudinal load, Lb/in
t = Thickness, sphere, in
t_c = Thickness, column, in
t_r = Thickness required, shell, in
V = Base shear, Lbs
V_S = Volume, Ft^3
V_{sn} = Partial volumes of section, Ft^3
V_n = Horizontal force per column, Lbs
W_o = Weight, operating, Lbs
W_w = Weight, water, Lbs
W_p = Weight, product, Lbs
W_s = Weight, steel, Lbs
w = Unit weight of liquid, PCF
Z_g = Section modulus, girder, in^3
Z_s = Section modulus, shell, in^3
Z_T = Section modulus, combined shell and girder, in^3
ΔL = Change in length of brace, in
δ = Lateral deflection of sphere, in

Table 9-16
Summary of loads at support locations

QTY of Columns	LEG No.	Case 1: At Columns Horiz (V_n)	Case 1: At Columns Vertical (Q)	Case 2: Between Columns Horiz (V_n)	Case 2: Between Columns Vertical (Q)
6	1	.0833 V	$F_D + F_L$	.125 V	$F_D + .866\ F_L$
	2	.2083 V	$F_D + .5\ F_L$	.25 V	F_D
	3	.2083 V	$F_D - .5\ F_L$	.125 V	$F_D - .866\ F_L$
	4	.0833 V	$F_D - F_L$	.125 V	$F_D - .866\ F_L$
	5	.2083 V	$F_D - .5F_L$	.25 V	F_D
	6	.2083 V	$F_D + .5\ F_L$	.125 V	$F_D + .866\ F_L$
8	1	.0366 V	$F_D + F_L$	.0625 V	$F_D + .9239\ F_L$
	2	.125 V	$F_D + .707\ F_L$	.1875 V	$F_D + .3827\ F_L$
	3	.2134 V	F_D	.1875 V	$F_D - .3827\ F_L$
	4	.125 V	$F_D - .707\ F_L$	.0625 V	$F_D - .9239\ F_L$
	5	.0366 V	$F_D - F_L$	.0625 V	$F_D - .9239\ F_L$
	6	.125 V	$F_D - .707\ F_L$	.1875 V	$F_D - .3827\ F_L$
	7	.2134 V	F_D	.1875 V	$F_D + .3827\ F_L$
	8	.125 V	$F_D + .707\ F_L$	.0625 V	$F_D + .9239\ F_L$
10	1	.0191 V	$F_D + F_L$	.0346 V	$F_D + .9511\ F_L$
	2	.0750 V	$F_D + .809\ F_L$	.125 V	$F_D + .5878\ F_L$
	3	.1655 V	$F_D + .309\ F_L$	.1809 V	F_D
	4	.1655 V	$F_D - .309\ F_L$	.125 V	$F_D - .5878\ F_L$
	5	.0750 V	$F_D - .809\ F_L$	.0346 V	$F_D - .9511\ F_L$
	6	.0191 V	$F_D - F_L$	.0346 V	$F_D - .9511\ F_L$
	7	.0750 V	$F_D - .809\ F_L$	.125 V	$F_D - .5878\ F_L$
	8	.1655 V	$F_D - .309\ F_L$	.1809 V	F_D
	9	.1655 V	$F_D + .309\ F_L$	.125 V	$F_D + .5878\ F_L$
	10	.0750V	$F_D + .809\ F_L$	.0346 V	$F_D + .9511\ F_L$
12	1	.0112 V	$F_D + F_L$	.0209 V	$F_D + .9659\ F_L$
	2	.0472 V	$F_D + .866\ F_L$	.0834 V	$F_D + .7071\ F_L$
	3	.1194 V	$F_D + .5\ F_L$	.1458 V	$F_D + .2588\ F_L$
	4	.1555 V	F_D	.1458 V	$F_D - .2588\ F_L$
	5	.1194 V	$F_D - .5\ F_L$	.0834 V	$F_D - .7071\ F_L$
	6	.0472 V	$F_D - .866\ F_L$	.0209 V	$F_D - .9659\ F_L$
	7	.0112 V	$F_D - F_L$	.0209 V	$F_D - .9659\ F_L$
	8	.0472 V	$F_D - .866\ F_L$	.0834 V	$F_D - .7071\ F_L$
	9	.1194 V	$F_D - .5\ F_L$	.1458 V	$F_D - .2588\ F_L$
	10	.1555 V	F_D	.1458 V	$F_D - .2588\ F_L$
	11	.1194 V	$F_D + .5\ F_L$	.0834 V	$F_D - .7071\ F_L$
	12	.0472 V	$F_D + .866\ F_L$	.0209 V	$F_D + .9659\ F_L$
16	1	.0048 V	$F_D + F_L$	.0091 V	$F_D + .9808\ F_L$
	2	.0217 V	$F_D + .9239\ F_L$	.0404 V	$F_D + .8315\ F_L$
	3	.0625 V	$F_D + .7071\ F_L$	.0846 V	$F_D + .5556\ F_L$
	4	.1034 V	$F_D + .3827\ F_L$	.1158 V	$F_D + .1951\ F_L$
	5	.1202 V	F_D	.1158 V	$F_D - .1951\ F_L$
	6	.1034 V	$F_D - .3827\ F_L$	.0846 V	$F_D - .5556\ F_L$
	7	.0625 V	$F_D - .7071\ F_L$	.0404 V	$F_D - .8315\ F_L$
	8	.0217 V	$F_D - .9239\ F_L$	.0091 V	$F_D - .9808\ F_L$
	9	.0048 V	$F_D - F_L$	.0091 V	$F_D - .9808\ F_L$
	10	.0217 V	$F_D - .9239\ F_L$	.0404 V	$F_D - .8315\ F_L$
	11	.0625 V	$F_D - .7071\ F_L$	.0846 V	$F_D - .5556\ F_L$
	12	.1034 V	$F_D - .3827\ F_L$	.1158 V	$F_D - .1951\ F_L$
	13	.1202 V	F_D	.1158 V	$F_D + .1951\ F_L$
	14	.1034 V	$F_D + .3827\ F_L$	.0846 V	$F_D + .5556\ F_L$
	15	.0625 V	$F_D + .7071\ F_L$	.0404 V	$F_D + .8315\ F_L$
	16	.0217 V	$F_D + .9239\ F_L$	.0091 V	$F_D + .9808\ F_L$

Table 9-17
Internal bending moments in girder section

No. of Columns	Due to Vertical Force, Q (Note 1)		Due to Horizontal Force, V (Note 2)	
	M_s	M_c	M_P	M_B
4	(-) .1366 QR_C	+ .0705 QR_C	+ .0683 VR_n	(-) .049 VR_n
6	(-) .0889 QR_C	+ .0451 QR_C	+ .0164 VR_n	(-) .013 VR_n
8	(-) .0662 QR_C	+ .0333 QR_C	+ .0061 VR_n	(-) .0058 VR_n
10	(-) .0527 QR_C	+ .0265 QR_C	+ .0030 VR_n	(-) .0029 VR_n
12	(-) .0438 QR_C	+ .0228 QR_C	+ .0016 VR_n	(-) .0016 VR_n
16	(-) .0328 QR_C	+ .0165 QR_C	+ .0007 VR_n	(-) .0007 VR_n

Notes:

1. R_C is in inches in order to get moment in in-Lbs
2. $R_n = R_C$ if no interal girder is used. $R_n = R_1$ if a girder is used.

Table 9-18
Angles

QTY OF COLUMNS	α	ϕ
4	45	90
6	60	60
8	67.5	45
10	72	36
12	75	30
16	77.5	25

Sphere-Dimensions & Data

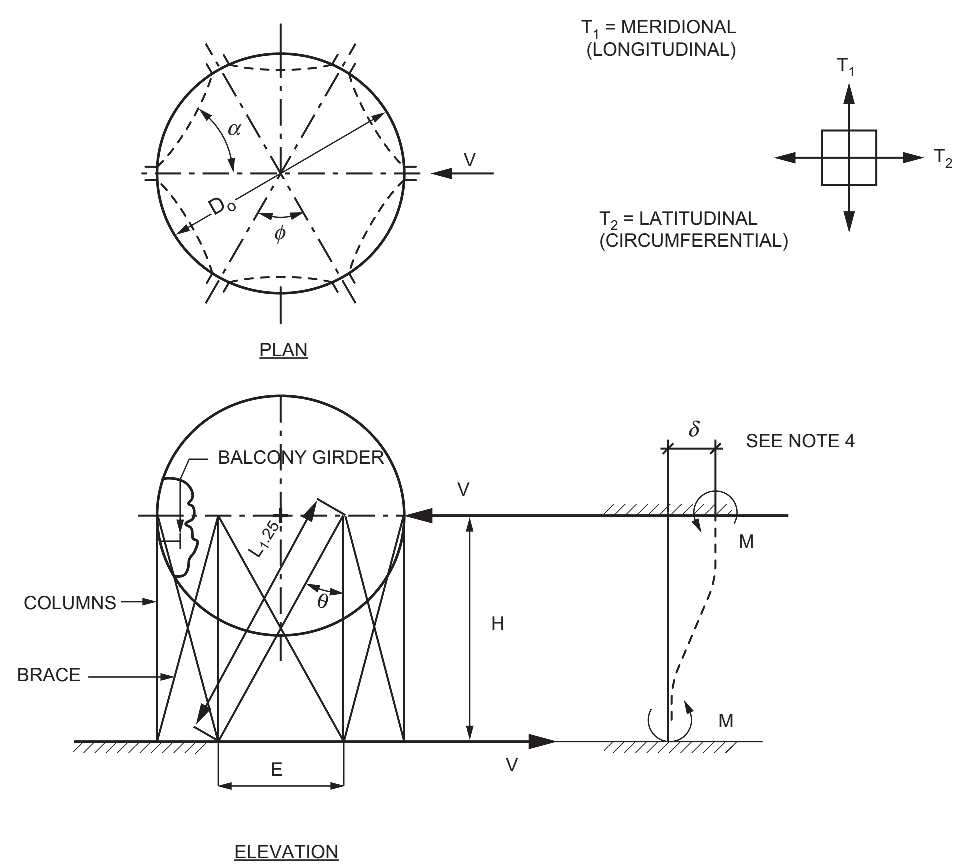


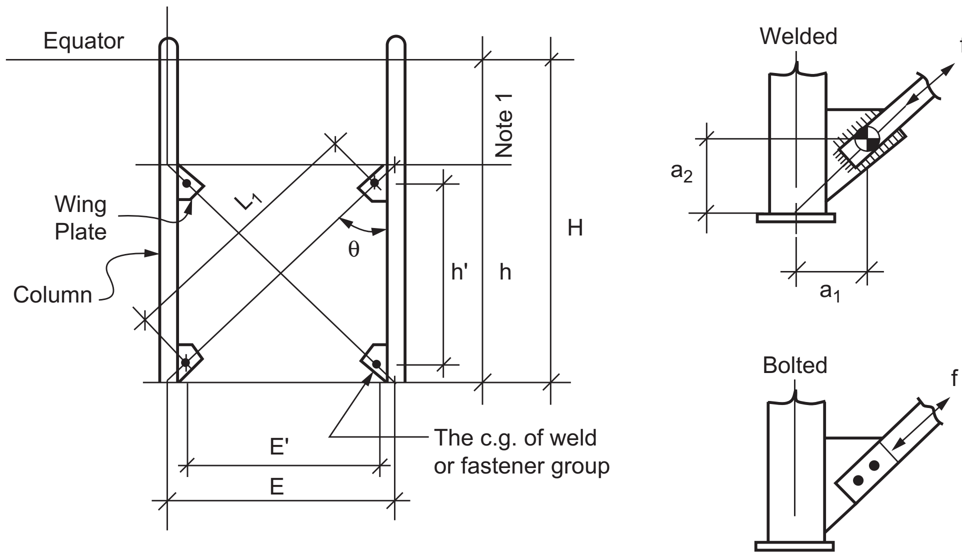
Table 9.19. Quantity and size of Support Columns

Capacity (BBLS)	Inside Dia (Ft-in)	Qty Columns	Column Size (OD X Thk (in)		Capacity (BBLS)	Inside Dia (Ft-in)	Qty Columns	Column Size (OD X Thk (in)
1,000	22-3	4	16 X .25		10,000	48-0	8	28 X .30
1,500	25-6	4	16 X .25		12,500	51-0	9	30 X .29
2,000	28-0	5	16 X .25		15,000	54-9	9	32 X .344
2,500	30-3	5	16 X .25		20,000	60-6	10	34 X .38
3,000	32-0	6	18 X .25		25,000	65-0	11	36 X .406
4,000	35-3	6	20 X .25		30,000	69-0	11	40 X .438
5,000	38-0	6	22 X .25		40,000	76-0	12	42 X .503
6,000	40-6	7	24 X .25		50,000	81-10	13	42 X .594
7,500	43-6	7	24 X .29					

Notes:

1. The column sizes shown are based on a sphere filled with water and normal wind/seismic loading. Higher pressure or significantly higher loadings will result in larger diameter columns or thicker columns.
2. The quantity of columns shown are based on 10 feet wide plates and a liquid sphere where columns are assumed for every other plate. For gas filled spheres, assume one column for every third plate.

Establish Leg & Brace Dimensions



Note 1: This dimesion shall be set by the designer/manufacturer. The clips are typically installed below dimension "B" as a minimum.

DIMENSIONS:	
ϕ =	h =
R_c =	$h' = h - 2a_2$ =
$E = 2[\sin .5\phi(R_c)]$ =	Find Angle θ ;
E =	$\tan\theta = E' / h'$ =
a_1 =	θ =
a_2 =	Find Length of Brace, L_1
$E' = E - 2a_1$ =	$L_1 = E' / \sin\theta$ =
H =	L_1 =

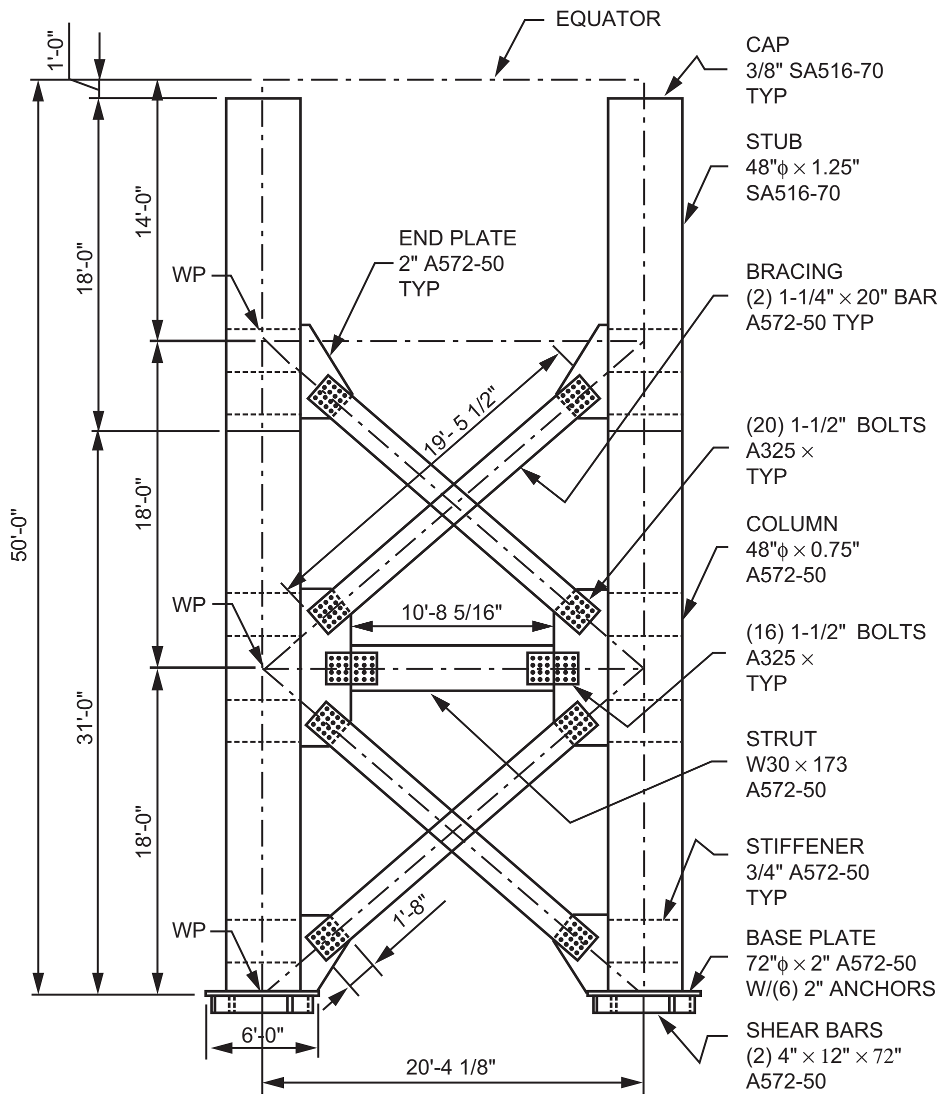
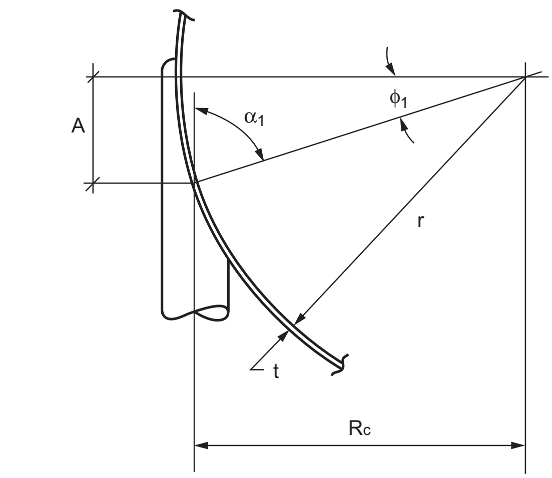
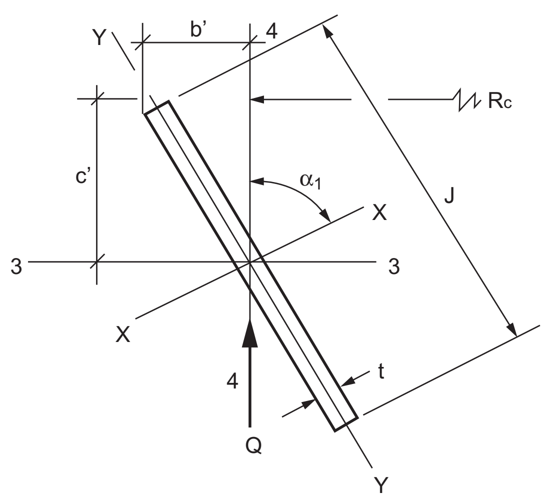


Figure 9-30. Two sets of cross bracing with horizontal struts for an 87' diameter sphere.

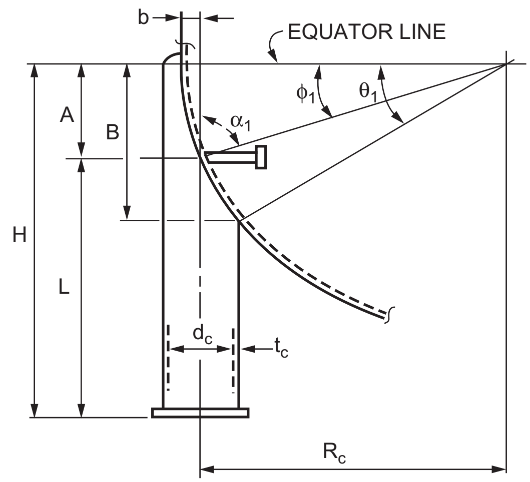
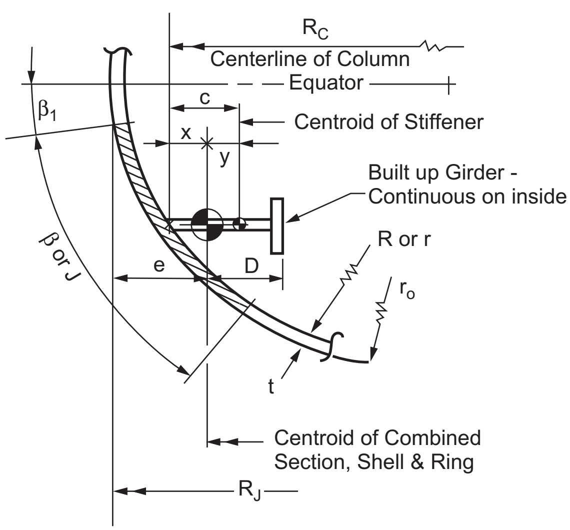
Properties of Shell Section (Without Girder) - Resisting Horizontal Loads

<table>
<tr><th>FIGURES</th><th>EQUATIONS/DIMENSIONS</th></tr>
<tr><td rowspan="12">

GENERAL ELEVATION</td><td>r =</td></tr>
<tr><td>t =</td></tr>
<tr><td>R_c =</td></tr>
<tr><td>$\text{Cos } \varphi_1 = (R_c / r) =$</td></tr>
<tr><td>$\varphi_1 =$</td></tr>
<tr><td>$\alpha_1 = 90 - \varphi_1 =$</td></tr>
<tr><td>$A = r \text{ Sin } \varphi_1 =$</td></tr>
<tr><td>J = Greater of following….</td></tr>
<tr><td>1. $2 (r\,t)^{1/2} =$</td></tr>
<tr><td>2. $32\,t =$</td></tr>
<tr><td>USE, J =</td></tr>
<tr><td>$b' = .5 L' \text{ Cos } \alpha_1 + .5 t \text{ Sin } \alpha_1 =$</td></tr>
<tr><td rowspan="13">

IDEALIZED SHELL SECTION
ACTING AS A GIRDER</td><td>$c' = .5 J \text{ Sin } \alpha_1 + .5 t \text{ Cos } \alpha_1 =$</td></tr>
<tr><td>$A_s = J\,t =$</td></tr>
<tr><td>$I_X = t J^3 / 12 =$</td></tr>
<tr><td>$I_Y = t^3 J / 12 =$</td></tr>
<tr><td>$I_3 = I_X \text{ Sin}^2 \alpha_1 + I_Y \text{ Cos}^2 \alpha_1 =$</td></tr>
<tr><td>$I_4 = I_X \text{ Cos}^2 \alpha_1 + I_Y \text{ Sin}^2 \alpha_1 =$</td></tr>
<tr><td>$Z_3 = I_3 / c' =$</td></tr>
<tr><td>$Z_4 = I_4 / b' =$</td></tr>
<tr><td></td></tr>
<tr><td></td></tr>
<tr><td></td></tr>
<tr><td></td></tr>
</table>

Dimensional Data for Combined Girder-Shell Section

FIGURES	EQUATIONS/DIMENSIONS
	R =
	r =
	t =
	r_o = r + t =
	R_c =
	d_c =
	t_c =
	H =
	Cos θ_1 = (R_c - .5d_c - t_c) / r =
	θ_1 =
	B = r Sin θ_1 =
	Cos ϕ_1 = (R_c / r) =
	ϕ_1 =
	α_1 = 90 - ϕ_1 =
	A = r Sin ϕ_1 =
	J = Greater of following….
	1. 2 (r t) $^{1/2}$ =
	2. 32 t =
	USE, J =
	x = A_g C / A_t
	y = C - x
	e = R_J - R
	R_1 = R_c - x =
	β = (180 J) / (π r_o)
	β_1 = φ_1 - β
	R_J = r_o Cos β_1

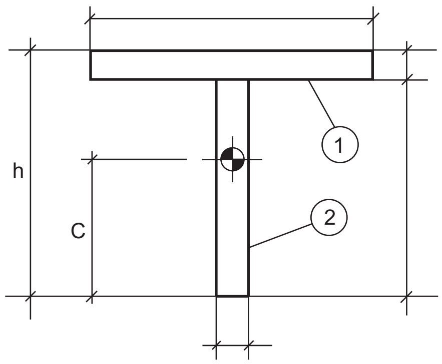
Sphere - Seismic Design			
	Method 1	**Method 2**	**Method 3**
V_n = Horizontal shear per leg	Worst case from table dependent on number of legs and direction of seisnic force (between legs or through legs).	NA	$V_n = V/N$
f = Max force in brace	$f = V_n / n \sin\theta$	$f = 2 W_o / 2 N \sin\theta$	$f = V_n / \sin\theta$
ΔL = Change in length of brace	$\Delta L = (f L_1) / (E_m A_b)$	$\Delta L = (f L_1)/(E_m A_b)$	$\Delta L = (2 W_o L_1)/(2 N E_m A_b \sin\theta)$
δ = Lateral deflection of sphere	$\delta = \Delta L / \sin\theta$	$\delta = \Delta L / \sin\theta$	$\delta = \Delta L / \sin\theta$
T = Period of vibration	$T = 2\pi(\delta / g)^{1/2}$	$T = 2\pi(\delta / g)^{1/2}$	$T = 2\pi(\delta / g)^{1/2}$

Notes:

1. Approx POV per ASCE 7–05; $T_a = C_t h_n^x$

Design of Girder

Properties of Stiffener Alone



DIMENSIONS OF INTERNAL GIRDER

Part	A	y	A y	A y^2	I
1					
2					
Σ					

$$C = \Sigma A\ y/\Sigma A$$

$$I_g = \Sigma A\ y^2 + \Sigma I - C\Sigma A\ y$$

Properties of Combined Section

- Area of Shell, A_s

  $A_s = J\ t$

- Area of combined section, A_t

  $A_t = A_s + A_g$

- Misc dimensions…

  $X = A_g\ C/A_t$

  $y = C - X$

  $D = h - X$

- Moment of inertia of combined section, I_T

  $I_T = I_g + I_g + A_S\ X^2 + A_g\ y^2$

- Section modulus, Z_T

  $Z_T = I_T/e$

- Radius of gyration of girder, r_g

  $r_g = (I_T/A_T)^{1/2}$

- Length of girder section exposed to load, L_g

  $L_g = (2\ \pi\ R_1)/N\ r_g$

- Slenderness ratio, S_r

  $S_r = L_g/r_g$

- Allowable stress;

  Tension = lesser of…
  1.2 (.6) F_y or 1.2 S

  Compression = Lesser of …
  1. Factor B X 1.2
  2. 1.2 (1.8) (10^6) (t / r)
  3. AISC allowable buckling stress based on slenderness ratio

- Factor A

  $A = (.125\ t)\ r_o$

Design of Columns

- Base Shear, V
  Use worst case of wind or seismic
  V = ___________

- Overturning Moment, M_o

  $M_o = H\ V$

- Maximum Dead load, F_D

  $F_D = (-)\ W_o/N$

- Maximum Live Load, F_L

  $F_L = +/(-)\ 48\ M_o/N\ D_c$

- Maximum Column Load, Q
  Select worst case from Table or use;

  $Q = F_D +/- F_L$

  Q max compression $= Q_C =$

  Q max tension $= Q_T =$

Note: If there is no uplift then there is no tension force.

- Leg selection;

A preliminary selection can be made and then checked.

Selection: __________
$A_c =$ ______________
$r_c =$ _______________

Compression Case

- Axial Stress, f_a

  $f_a = Q_C/A_C < F_a$

- Slenderness ratio, S_r

  $S_r = K\,h/r_c =$

- Allowable axial stress, F_a

  $F_a =$

Tension Case

- Tension stress, f_T

  $f_T = Q_T/A_C < F_T$

- Allowable tension stress, F_T

  $F_T = 1.2\,(.6)\,F_y$

Cross Bracing

Note: Loads in cross bracing are tension and compression.

Compression Case

- Required moment of inertia, i_r

  Case 1; Pinned at center …

  $$i_r = \left(f\,L_1^2\right)/\left(4\,\pi^2\,E_m\right)$$

  Case 2: Not pinned at center …

  $$i_r = \left(f\,L_1^2\right)/\left(\pi^2\,E_m\right)$$

- Use: ____________

  $I_b =$ ___________
  $r_b =$ ___________
  $A_b =$ ___________

- Slenderness ratio, S_r

  $S_r = K\,L_1/n'\,r_b$

  $n' = 1$ for not pinned; 2 for pinned

- Allowable axial stress, F_a

  Based on applicable slenderness ratio;

  $F_a =$ ___________

- Axial stress, f_a

  $f_a = f/A_b < F_a$

Tension Case

- Tension stress, f_T

  $f_T = f/A_b < F_T$

Sway Bracing

Note: Loads in sway bracing are tension only!

- Area of bracing required, A_{br}
  $A_{br} = f/F_T$
- Allowable tensile stress, F_T
  $F_T = 1.2\,(.6)\,F_y =$
- Use: ___________

Post Connection Plate

There are two major steps required in the design of the post connection plates. The steps are as follows;

Step 1: Determine if an internal girder is required.

Step 2: If a ring girder is required, design the girder to resist all load conditions.

Note: A girder is only required to resist horizontal loads. If stresses from vertical loads are excessive, than the shell thickness must be increased for the post connection plates.

Step 1:

a. Determine the properties of the shell in the girder section (Use worksheet)
b. Determine all the moments in the shell due to external loads, M_S, M_C, M_P and M_B, (Use worksheet)
c. Determine the stresses in the shell due to external loads (Use worksheet)
d. If the stresses in the shell due to horizontal loading are exceeded, either add a ring girder or increase the shell thickness. If a ring girder is added, proceed to Step 2.
e. If the stresses in the shell due to vertical loading are acceptable, than the design is acceptable as is. If the shell stresses are exceeded, the shell thickness must be increased until the stresses are below allowable.

Step 2: The shell stresses due to horizontal loading are exceeded and a ring girder must be added.

a. Approximate the size of the ring girder required from the following equation:

   $$Z_r = M_P/F_T$$

   Where M_P and F_T are the same as those calculated in Step 1.
b. Based on the section modulus required, select a member size that will approximate the value required.
c. Complete the worksheet with girder to determine the exact properties.
d. Check the stresses in the combined shell-girder section as follows:

   $$f_T = M_P/Z_T$$

Combined Stresses

The stress at any point in the sphere depends on whether the plane under consideration is above or below the LOS (Line of Support).

In the following equations, T' is the greater of T_1 or T_2

Tension Other than support region:

$$S_T = T'/t\,E$$

At support location:

$$S_T = (T'/t\,E) + f_T$$

Compression Other than support region:

$$S_C = T'/t$$

At support location:

$$S_C = (T'/t) - f_C$$

Spheres; Shell Thickness Procedure

The thickness required for a sphere varies at every elevation. The required thickness is a function of the pressure, liquid level, loadings, cross sectional area and weights of shell and liquid at any particular elevation. Since the weights and pressures are varying at every elevation, the required thickness is also changing.

However, the thickness will only be changed in step increments to accommodate fabrication limitations and actual sizes of plates selected. Adding extra seams to accommodate potential step breaks in plate thickness may not yield the most economical design since the welding costs related to these extra seams may outweigh the savings in plate thickness. These are shop decisions and the elevations where thickness breaks occur should not be arbitrarily established by the designer.

Typically, the designer will check the required shell thickness at elevations where a seam is required. Actual plate thicknesses used will include a thinning allowance to accommodate thinning which occurs in the pressing of the plates. The plates do not get lighter. The actual plate thickness utilized is dependent upon the sizes of the plate available from the shop, mill or warehouse and shop limitation regarding the pressing of plates.

The weight estimates should include a thinning allowance to account for the thicker plates utilized. Thinning allowance can be estimated as follows:

Thickness	Thinning Allowance
Up to 1”	.188”
1” to 2”	.375”
2” to 3”	.625”
3” to 4”	.75”
4” to 5”	1”
Over 5”	1.5”

In addition, the “post plates” may be made thicker to accommodate local loads and horizontal loads due to wind or seismic. Since the columns are not attached to every plate in the support band, the plates between the post plates may also be increased in thickness, but not as much as the post plates themselves. Once again, this is a shop decision based on the overall capabilities of the shop, as well as economic decisions.

So, a sphere can be made of a variety of plate thicknesses, both from top to bottom, but also around the equator where the support columns are attached. Occasionally, the post plates are the only plates in the sphere thick enough to require PWHT. In such a case, the PWHT will be carried out in the shop. Field PWHT will not be required. If the spheres are large enough, or multiple spheres are required, then mill runs for the exact thickness of material required are selected.

Stress in Post Connection Plates Due to External Loads				
	At Support Locations		Between Support Locations	
	Moment	Stress	Moment	Stress
Due to Vertical Force, Q	M_S = (Compression)	f_C =	M_C = (Tension)	f_T =
Due to Horizontal Force, V	M_P = (Tension)	f_T =	M_B = (Compression)	f_C =

Notes:

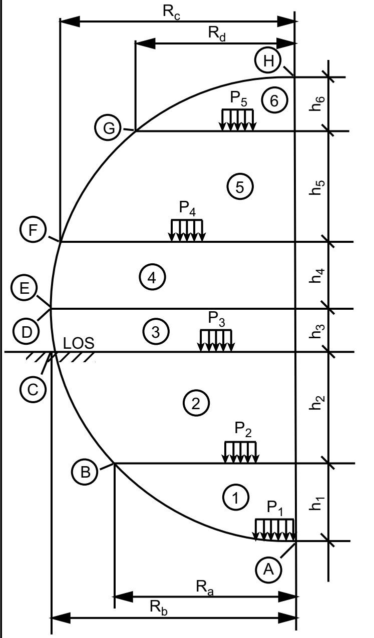
1. The post connection plates are the shell sections to which the "posts" or columns are attached. The post connection plates form a circumferential band all around the circumference at the location where the posts are attached. These plates take the horizontal and vertical loads imposed by dead and live loads. The post plates are typically thicker than all the other plates of sphere, whether a girder is provided or not. However a girder is usually the most economical way of distributing the loads encountered at this location.
2. The moments above are from the Table and are dependent on the number of legs.
3. Allowable stresses:

   Tension; $F_T = 1.2\,(.6)\,Fy =$
   Compression, $F_C = 1.2\,(\,1.8)\,(10^6)\,(t/r\,) =$

4. DATA REQUIRED:

N =	R_C =	Z_3 =	V =	F_y =
Q =	t =	Z_4 =	r =	

SPHERE - WEIGHTS, VOLUMES, AREAS, PRESSURES & LOADS



FIVE COURSE SPHERE SHOWN FOR EXAMPLE

h_n	R_n	t_n'	V_n	A_{sn}	W_L	W_S	W_T	Σh_n	P_n	A_{cn}
Σ										

EQUATIONS

$A_{sn} = 2\pi R_n h_n$	$R_a = [2 R h_1 - h_1^2]^{1/2}$	$V_1 = 1.047 h_1^2 (3R - h_1)$	GIVEN:
$A_{cn} = \pi R_n^2$	$R_b = [R^2 - h_3^2]^{1/2}$	$V_2 = .523 h_2 (3 R_a^2 + 3 R_b^2 + h_2^2)$	$R =$
$A_{S1} = 2\pi R_a h_1$	$R_c = [R^2 - h_4^2]^{1/2}$	$V_3 = .523 h_3 (3 R_b^2 + 3 R^2 + h_3^2)$	$r =$
$A_{S2} = 2\pi R_b h_2$	$R_d = [2 R h_6 - h_6^2]^{1/2}$	$V_4 = .523 h_4 (3 R^2 + 3 R_c^2 + h_4^2)$	$S_g =$
$A_{S3} = 2\pi R h_3$	$P_n = P + P_h$ or $P_h - P_X$	$V_5 = .523 h_5 (3 R_c^2 + 3 R_d^2 + h_5^2)$	$w = 62.4 S_g =$
$A_{S4} = 2\pi R h_4$	$P_h = .433 \Sigma h_n S_g$	$V_6 = 1.047 h_6^2 (3R - h_6)$	$P =$
$A_{S5} = 2\pi R_c h_5$	$W_L = V_n w$		$P_x =$
$A_{S6} = 2\pi R_d h_6$	$W_S = 144 (.2833) t_n A_{sn}$	$W_T = W_L + W_S$	

CALCULATION OF THICKNESS

DESIGN POINT OR LEVEL	P_n	W_{Tn}	A_{Cn}	W_{Tn} / A_{cn}	T_1	T_2	t_n	EQUATIONS
H								$T_1 = T_2 = P_1 r / 2$ = (Note 1)
G-G								Any elevation above the LOS $T_1 = .5\, r\ [P_n - W_{Tn} / A_{cn}]$
F-F								$T_2 = .5\, r\ [P_n + W_{Tn} / A_{cn}]$ t_r = Greater of ....
E-E								(Greater of T_1 or T_2) / SE (Greater of T_1 or T_2) / F_c
D-D								
C-C (LOS)								Any elevation at or below the LOS $T_1 = .5\, r\ [P_n + W_{Tn} / A_{cn}]$ $T_2 = .5\, r\ [P_n - W_{Tn} / A_{cn}]$
B-B								t_r = Greater of .... (Greater of T_1 or T_2) / SE (Greater of T_1 or T_2) / F_c
A								$T_1 = T_2 = P_1 r / 2$ = (Note 1)

NOTES:

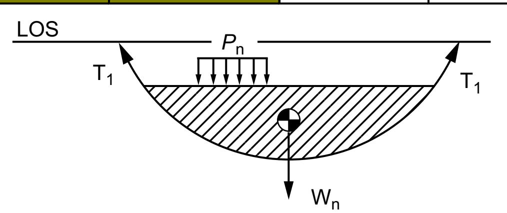
1) Formulas shown are for API 620 Sphere;

For ASME VIII-1, Use...

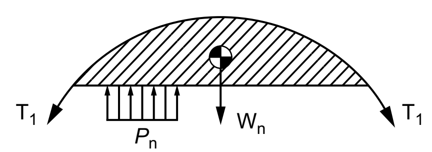
$t_r = (P_n r) / (2\, S\, E - .2\, P_n) =$

For ASME VIII - 2, Use...

$t_r = r\ (e^{.5P/S} - 1) =$

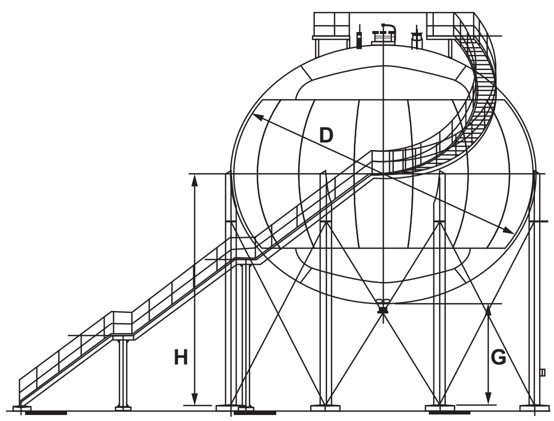


CASE 1: PLANE BELOW LINE OF SUPPORT (LOS)



CASE 2: PLANE ABOVE LINE OF SUPPORT (LOS)

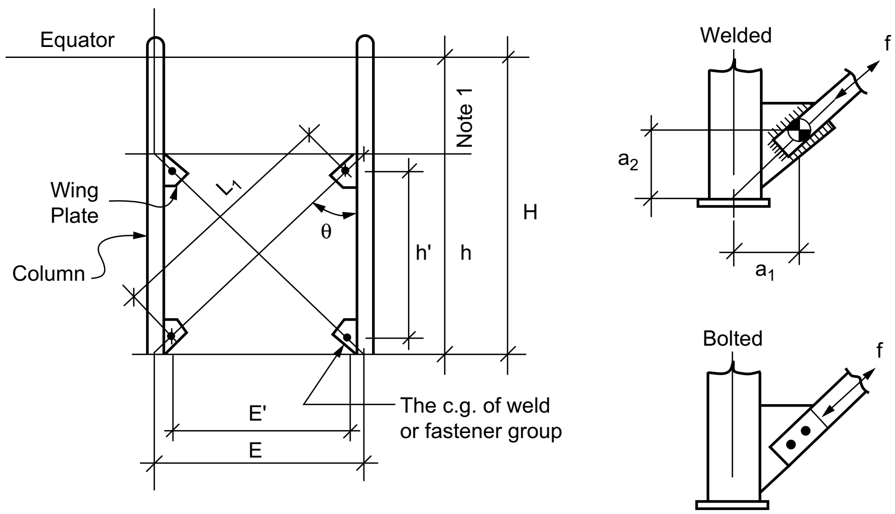
SPHERE DATA SHEET (EXAMPLE)			
EQUIPMENT ITEM NO. : 745-V-101 A/B			
EQUIPMENT NAME : NGL SPHERE			



DIMENSIONS	
D	39.37 FT ID
H	35.1 FT
G	15.1 FT
WEIGHTS	
FABRICATED	N.A.
EMPTY	818 KIPS
OPERATING	1,817 KIPS
TEST	2,807 KIPS
SHELL	704 KIPS
COLUMNS	23 KIPS
BRACING	59 KIPS
CONTENTS	1,018 KIPS
WATER	1,994 KIPS
FIREPROOFING	32 KIPS

DESIGN DATA		MATERIALS	
Design Pressure - Internal	580 PSIG	Shell	SA-516-65
Design Pressure - External	7.5 PSIG	Columns	SA-333-6
Design Temperature - Internal	212 ^{O}F	Column Bracing	SA-53-B
Design Temperature - External	150 ^{O}F	Flanges	SA-350-LF2
MDMT	(-) 29 ^{O}F	Nozzle Necks- Pipe	SA-333-6
Specific Gravity	0.517	Bolting	SA-193-B7
Capacity	5690 BBLS	Gaskets	SPIRAL WOUND- STYLE CG
Corrosion Allowance	0.125 IN	1/8" THK - 316 SST W/ GRAPHITE	
Joint Efficiency	1	Base Plates	SA-36
PWHT	YES	**CODES**	
Contents	NGL	Design	ASME VIII-2
Insulation/Thk	NO	Seismic	ASCE 7 -05
Fireproofing - Legs	YES, 2 IN	Wind	ASCE 7 -05
Service (Sour, Lethal, Cyclic, etc.)	N.A.	Structural	AISC

Establish Leg & Brace Dimensions

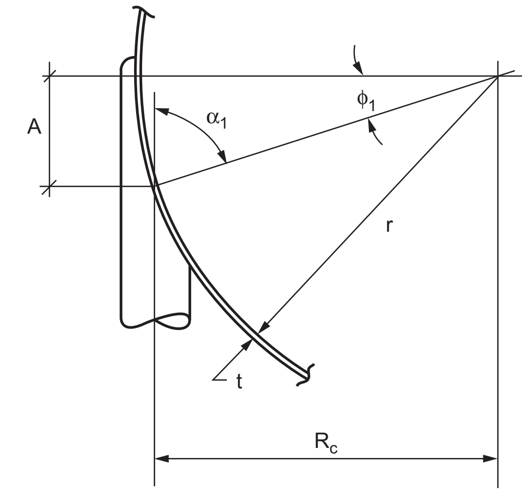


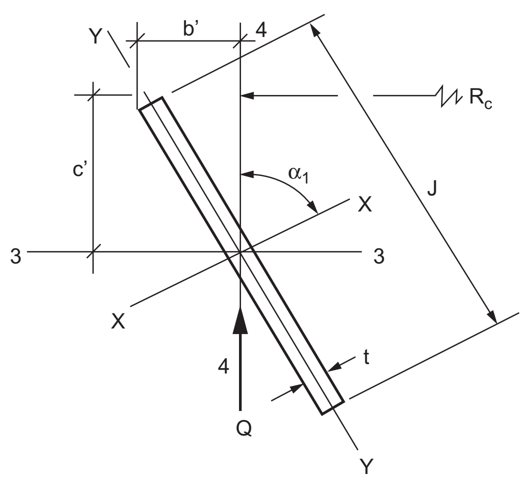
Note 1: This dimension shall be by the designer/manufacturer. The clips are typically installed below dimension "B" as a minimum.

DIMENSIONS:

$\phi = 45^\circ$	$h = 27.915$ ft
$R_c = 19.65$ ft	$h' = h - 2\,a_2 = 23.315$ ft
$E = 2\,[\,\sin .5\,\phi\,(R_c)\,] =$	***Find Angle*** ϑ***;***
$E = 15.04$ ft	$\tan\theta = E' / h' = 12.38/23.315 = .531$
$a_1 = 1.33$ ft	$\theta = 27.97^\circ$
$a_2 = 2.3$ ft	***Find Length of Brace,*** L_1
$E' = E - 2\,a_1 = 12.38$ ft	$L_1 = E' / \sin\theta = 12.38/\sin 27.97 =$
$H = 35.1$ ft	$L_1 = 26.4$ ft

Properties of Shell (Without Girder)- Resisting Horizontal Loads

<table>
<tr><th>FIGURES</th><th>EQUATIONS/DIMENSIONS</th></tr>
<tr><td rowspan="12">

GENERAL ELEVATION</td><td>$r = 236.33$ in (Corr)</td></tr>
<tr><td>$t = 3.375$ in (Corr)</td></tr>
<tr><td>$R_c = 235.83$ in</td></tr>
<tr><td>$\text{Cos}\ \phi_1 = (R_c / r) = .9979$</td></tr>
<tr><td>$\phi_1 = 3.72^{\circ}$</td></tr>
<tr><td>$\alpha_1 = 90 - \phi_1 = 86.27^{\circ}$</td></tr>
<tr><td>$A = r\ \text{Sin}\ \phi_1 = 15.33$ in</td></tr>
<tr><td>J = Greater of following....</td></tr>
<tr><td>1. $2\ (r\ t)^{1/2} = 56.48$ in</td></tr>
<tr><td>2. $32\ t = 108$ in</td></tr>
<tr><td>USE, $J = 108$ in</td></tr>
<tr><td>$b' = .5\ L'\ \text{Cos}\ \alpha_1 + .5\ t\ \text{Sin}\ \alpha_1 = 5.19$ in</td></tr>
<tr><td rowspan="12">

IDEALIZED SHELL SECTION
ACTING AS A GIRDER</td><td>$c' = .5\ J\ \text{Sin}\ \alpha_1 + .5\ t\ \text{Cos}\ \alpha_1 = 54$ in</td></tr>
<tr><td>$A_s = J\ t = 364.5\ \text{in}^2$</td></tr>
<tr><td>$I_X = t\ J^3 / 12 = 354,294\ \text{in}^4$</td></tr>
<tr><td>$I_Y = t^3\ J / 12 = 346\ \text{in}^4$</td></tr>
<tr><td>$I_3 = I_X\ \text{Sin}^2\ \alpha_1 + I_Y\ \text{Cos}^2\ \alpha_1 = 352,797\ \text{in}^4$</td></tr>
<tr><td>$I_4 = I_X\ \text{Cos}^2\ \alpha_1 + I_Y\ \text{Sin}^2\ \alpha_1 = 1844\ \text{in}^4$</td></tr>
<tr><td>$Z_3 = I_3 / c' = 6533\ \text{in}^3$</td></tr>
<tr><td>$Z_4 = I_4 / b' = 355\ \text{in}^3$</td></tr>
<tr><td></td></tr>
<tr><td></td></tr>
<tr><td></td></tr>
<tr><td></td></tr>
</table>

Data

$P = 580$ PSIG
$P_X = 7.5$ PSIG
D.L.L. = Full
$S_g = .517$
$w = 32.26$ PCF
$D = 39.37$ ft
$R = 19.685$ ft
$r = 236.22$ in
$D_c = 39.3$ ft
$R_c = 19.65$ ft
$H = 35.1$ ft
$g = 386$ in/sec^2
$E = 28.8 \times 10^6$ PSI
S @ 212 °F = 21,300 PSI
$F_y = 31.9$ KSI
$W_o = 1817$ kips

Properties of Columns

Size: 16 in Sch 40
$N = 8$
$d_c = 15$ in ID
$t_c = 0.5$ in
$A_c = 24.35$ in^2
$I_c = 732$ in^4
$r_c = 5.48$ in
$\varphi = 45^o$
$\alpha = 67.5^o$

Properties of Bracing

Size: 4 in Sch 40
$A_b = 3.17$ in^2
$I_b = 7.23$ in
$r_b = 1.51$ in
$n = 2$ (Cross Bracing)
$L_1 = 26.4$ ft

Wind/Seismic

Code: ASCE 7-05

- Base Shear, Wind; V

$$V = q_z\, G\, C_f\, A_f$$

$$= 36.24\ (.85)\ .7\ (1630) = 35,151\ \text{Lbs}$$

- Base Shear, Seismic: V

$$V = C_h\, W_o\, I$$
$$= .0694\ (1817\ \text{kips})\ 1.5 = 126.1\ \text{kips}$$
$$= 126.1\ \text{kips}$$

Therefore Seismic Governs;

- Overturning Moment, M_o

$$M_o = H\, V = 35.1\ (126.1\ \text{kips})$$
$$= 4426\ \text{Ft} - \text{kips}$$

Design of Columns

- Maximum Dead load, F_D

$$F_D = (-)\ W/N = 1817/8 = (-)\ 227.13\ \text{kips}$$

- Maximum Live Load, F_L

$$F_L = +/(-)\ 48\ M_o/N\, D_c =$$
$$= 48\ (4226)/8\ (39.3 \times 12) = +/(-)\ 6.71\ \text{kips}$$

- Maximum Column Load, Q
(Equation from Table)

$$Q = F_D - F_L = (-)\ 227.13 - 6.71 = 233.84\ \text{kips}$$

Slenderness ratio, S_r

$$S_r = K\, h/r_c = 1.0\ (12 \times 27.915)/5.48$$
$$= 61.12$$

- Allowable Compressive Stress, F_c

$$F_c = 17.33\ \text{KSI}$$

- Axial Stress, f_a

$$f_a = Q/A_c = 233.84^K/24.35 = 9.6\ \text{KSI}$$

- There is no tension in columns to consider!

Design of Bracing

- Max Horizontal Force per Column, V_n
(Equation from Table)

$$V_n = .2134\ V = .2134\ (126.1\ \text{kips}) = 26.9\ \text{kips}$$

- Maximum Force in Brace, f

$$f = V_n/n \sin\theta = 26.9/2\ (\sin 27.96)$$
$$= 28.68\ \text{kips}$$

- Required moment of Inertia, I_r

$$I_r = \left(f\ L_1^2\right)/\left(4\ \pi^2\ E_m\right)$$
$$= 28,680\ (26.4 \times 12)^2\ /\ 4\ \pi^2\ (28.8\ (10^6)$$
$$= 2.53\ \text{in}^4 < 7.27\ \text{in}^4$$

- Slenderness Ratio, S_r

$$S_r = K\ L_1/2\ r_b = 1\ (26.4 \times 12)/2 \times 1.51$$
$$= 105$$

- Allowable Axial Stress, F_a

$$F_a = 12.33\ \text{KSI}$$

- Axial Stress, f_a

$$f_a = f/A_b = 28.68/3.17 = 8.42\ \text{KSI}$$

Allowable Stresses

Girder / Post Connection Plates

- Tension: F_T, Lesser of following;

$$F_T = 1.2\ (.6)\ F_y = 1.2\ (.6)\ 35\ \text{KSI} = 25.2\ \text{KSI}$$
$$\text{Or } 1.2\ S = 1.2\ (21.3\ \text{KSI}) = 25.56\ \text{KSI}$$

- Girder properties:

$$I_T = 1844\ \text{in}^4$$
$$A_T = 364.5\ \text{in}^2$$

- Radius of Gyration, r_g

$$r_g = (I_T/A_T)^{1/2} = (1844/364.5)^{1/2} = 2.245\ \text{in}$$

- Length of Girder, L_g

$$L_g = 2\ \pi\ \left(R_1\right) = 2\ \pi\ \left(12 \times 19.65\ \text{ft}\right) = 1481\ \text{in}$$

- Slenderness Ratio, S_r

$$S_r = K\ L_g/\left(N\ r_g\right) = 1\ \left(1481\right)/\left(8 \times 2.245\ \text{in}\right)$$
$$= 82.46$$

- Allowable Compressive Stress, F_c

$$F_c = 15.13\ \text{KSI}$$

- $F_T = \text{Tension} = 1.2\ S = 1.2\ (21.3) =$
  $= 25.56\ \text{KSI}$

- Allowable Compressive Stress, F_c

F_C = Lesser of following....

1. Factor 'B' from ASME Code;

$$A = .125\ t/r_o = .125\ (3.375)/239.595$$
$$= .00176$$
$$B = 14,000\ \text{PSI}$$

2. $F_C = 1.8\ (10^6)\ (t/r_o)$
   $= 1.8\ (10^6)\ .01409$
   $= 25,355\ \text{PSI}$

Therefore F_C = 14,000 PSI

Combined Stress

Tension;
Worst case Plane B-B;

$$S_T = T'/t = 69572/3.375 = 20,613\ \text{PSI}$$

At support location (Plane C-C);

$$S_T = T'/t + f_T = 69495/3.375 + 281$$
$$= 20,832\ \text{PSI}$$
$$S = 21,300\ \text{PSI OK}$$

Sphere - Seismic Design (Sample Problem)			
DATA: $V = 126.1$ $N = 8$ $n = 2$ (Cross braced legs) $D = 39.37$ ft $\theta = 27.97^o$ $E_m = 28.8 \times 10^6$ PSI $g = 386$ in/sec^2 $W_o = 1817$ kips $A_b = 3.17$ in^2			
	Method 1	**Method 2**	**Method 3**
V_n = Horizontal shear per leg	$V_n = .2134\ V = .2134\ (126.1\ \text{kips}) = 26.9$ kips	NA	$V_n = V/N = 126.1 / 8 = 15.76$ kips
f = Max force in brace	$f = V_n / n \sin\theta = 26.9/2\ (\sin 27.97^o) = 28.67$ kips	$f = 2\ W_o / 2\ N \sin\theta = 2\ (1817\ \text{kips}) / 2\ (8) \sin 27.97^o = 484.26$ kips	$f = V_n / \sin\theta = 15.76\ \text{kips}/ \sin 27.97^o = 33.6$ kips
ΔL = Change in length of brace	$\Delta L = (f\ L_1) / (E_m\ A_b) = 28670\ (12)\ 26.4 / 28.8\ (10^6)\ 3.17 = 0.1$ in	$\Delta L = (f\ L_1) / (E_m\ A_b) = 484{,}260\ (12)\ 26.4 / 28.8\ (10^6)\ 3.17 = 1.68$ in	$\Delta L = (2\ W_o\ L_1) / (2\ N\ E_m\ A_b \sin\theta) = 2\ (1{,}817{,}000)\ 12\ (26.4) / 2\ (8)\ 28.8\ (10^6) \sin 27.97 = 1.68$ in
δ = Lateral deflection of sphere	$\delta = \Delta L / \sin\theta = .1 / \sin 27.97 = 0.213$ in	$\delta = \Delta L / \sin\theta = 1.68 / \sin 27.97 = 3.58$ in	$\delta = \Delta L / \sin\theta = 1.68 / \sin 27.97 = 3.58$ in
T = Period of vibration	$T = 2\ \pi\ (\delta / g)^{1/2} = 2\ (\pi)\ [.213 / 386]^{1/2} = .147$ Sec's	$T = 2\ \pi\ (\delta / g)^{1/2} = 2\ (\pi)\ [3.58 / 386]^{1/2} = .605$ Sec's	$T = 2\ \pi\ (\delta / g)^{1/2} = 2\ (\pi)\ [3.58 / 386]^{1/2} = .605$ Sec's

Notes:

1. Approx POV per ASCE7-05; $T_a = C_t\ h_n^x = .02\ (55)^{.75} = .404$ Sec's

Stress in Post Connection Plates Due to External Loads (Example)

	At Support Locations		Between Support Locations	
	Moment	**Stress**	**Moment**	**Stress**
Due to Vertical Force, Q	M_S = (Compression) = (-) .0662 Q R_C = (-) 3,650,697 in-Lbs	$f_C = M_S / Z_3$ = 558 PSI	M_C = (Tension) = + .0333 Q R_C = + 1,836,378 in-Lbs	$f_T = M_C / Z_3$ = 281 PSI
Due to Horizontal Force, V	M_P = (Tension) = + .0061 V R_C = + 181,403 in-Lbs	$f_T = M_P / Z_4$ = 510 PSI	M_B = (Compression) = (-) .0058 V R_C = (-) 172,481 in-Lbs	$f_C = M_B / Z_4$ = 485 PSI

Notes:

1. The post connection plates are the shell sections to which the "posts" or columns are attached. The post connection plates form a circumferential band all around the circumference at the location where the posts are attached. These plates take the horizontal and vertical loads imposed by dead and live loads. The post plates are typically thicker than all the other plates of sphere, whether a girder is provided or not. However a girder is usually the most economical way of distributing the loads encountered at this location.

2. The moments above are from the Table and are dependent on the number of legs.

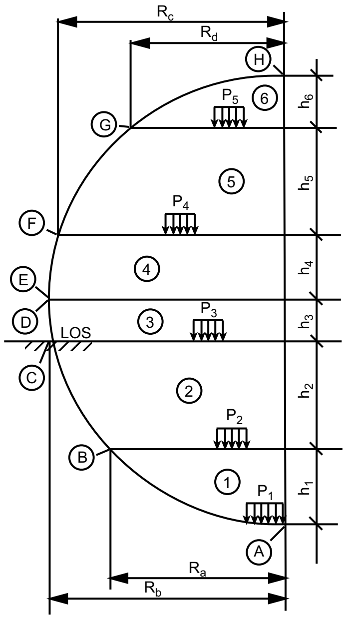
3. Allowable stresses;

   Tension; F_T = 1.2 (.6) F_y = 1.2 (.6) 31.9 KSI = 23 KSI
   Compression, F_C = 1.2 (1.8) (10^6) (t / r) = 1.2 (1.8) (10^6) (3.375 / 236.33) = 30,846 PSI

4. DATA REQUIRED:

N = 8	R_C = 235.83 in	Z_3 = 6533 in^3	V = 126,100 Lbs	F_y = 31.9 KSI
Q = 233,840 Lbs	t = 3.375 in	Z_4 = 355 in^3	r = 236.33 in	

Sphere - Weights, Volumes, Areas, Pressures & Loads



FIVE COURSE SPHERE SHOWN FOR EXAMPLE

h_n	R_n	t_n'	V_n	A_{sn}	W_L	W_S	W_T	Σh_n	P_n	A_{cn}
									580	
5	157.2	3.5	1415	412	45,650	58,834	105,000	5	581.1	77,635
9	226.1	3.5	7816	1065	252,144	152,082	404,300	14	583.1	160,600
5.69	236.1	3.5	6718	703	216,722	100,388	317,100	19.69	584.4	175,270
1.28	236.2	3.5	1551	158	50,035	22,562	72,600	21	584.7	175,270
13.41	235.7	3.5	12984	1587	418,863	226,624	645,500	34.37	587.7	174,530
5	157.2	3.5	1415	412	45,650	58,834	105,000	39.37	588.8	77,635
Σ			31,900	4337	1030 K	620 K	1650 K			

EQUATIONS

$A_{sn} = 2\pi R_n h_n$	$R_a = [2 R h_1 - h_1^2]^{1/2}$	$V_1 = 1.047 h_1^2 (3R - h_1)$	GIVEN:
$A_{cn} = \pi R_n^2$	$R_b = [R^2 - h_3^2]^{1/2}$	$V_2 = .523 h_2 (3R_a^2 + 3R_b^2 + h_2^2)$	$R = 19.685$ ft
$A_{S1} = 2\pi R_a h_1$	$R_c = [R^2 - h_4^2]^{1/2}$	$V_3 = .523 h_3 (3R_b^2 + 3R^2 + h_3^2)$	$r = 236.22$ in
$A_{S2} = 2\pi R_b h_2$	$R_d = [2 R h_6 - h_6^2]^{1/2}$	$V_4 = .523 h_4 (3R^2 + 3R_c^2 + h_4^2)$	$S_g = .517$
$A_{S3} = 2\pi R h_3$	$P_n = P + P_h$ or $P_h - P_X$	$V_5 = .523 h_5 (3R_c^2 + 3R_d^2 + h_5^2)$	$w = 62.4 S_g = 32.26$ PCF
$A_{S4} = 2\pi R h_4$	$P_h = .433 \Sigma h_n S_g$	$V_6 = 1.047 h_6^2 (3R - h_6)$	$P = 580$ PSIG
$A_{S5} = 2\pi R_c h_5$	$W_L = V_n w$		$P_x = 7.5$ PSIG
$A_{S6} = 2\pi R_d h_6$	$W_S = 144 (.2833) t_n A_{sn}$	$W_T = W_L + W_S$	

Calculation of Thickness

DESIGN POINT OR LEVEL	P_n	W_{Tn}	A_{Cn}	W_{Tn} / A_{cn}	T_1	T_2	t_r
H	580				68503.8	68503.8	3.22
G-G	581.1	105,000	77,635	1.35	68474.3	68793.2	3.23
F-F	583.1	404,300	160,600	2.52	68572.3	69167.6	3.25
E-E	584.4	317,000	175,270	1.81	68809.7	69237.3	3.25
D-D	584.4	72,600	175,270	0.41	68975.1	69071.9	3.24
C-C (LOS)	584.7	645,500	174,530	3.7	69495.9	68621.9	3.26
B-B	587.7	105,000	77,635	1.35	69572.7	69253.8	3.27
A	588.1				69460.5	69460.5	3.26

EQUATIONS

$T_1 = T_2 = P_1\, r / 2$ = (Note 1)

Any elevation above the LOS

$T_1 = .5\, r\, [\, P_n - W_{Tn} / A_{cn}\,]$

$T_2 = .5\, r\, [\, P_n + W_{Tn} / A_{cn}\,]$

t_r = Greater of ....

(Greater of T_1 or T_2) / SE

(Greater of T_1 or T_2) / F_c

Any elevation at or below the LOS

$T_1 = .5\, r\, [\, P_n + W_{Tn} / A_{cn}\,]$

$T_2 = .5\, r\, [\, P_n - W_{Tn} / A_{cn}\,]$

t_r = Greater of ....

(Greater of T_1 or T_2) / SE

(Greater of T_1 or T_2) / F_c

$T_1 = T_2 = P_1\, r / 2$ = (Note 1)

NOTES:

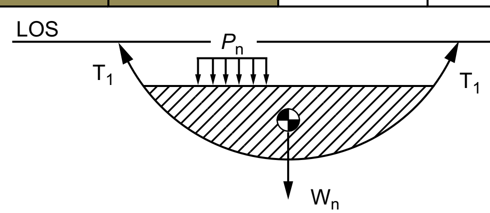
1) Formulas shown are for API 620 Sphere;

For ASME VIII-1, Use...

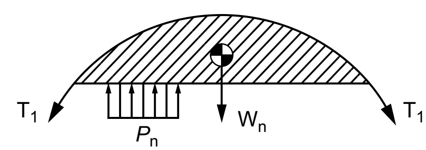
$t_r = (P_n\, r) / (2\, S\, E - .2\, P_n) =$

For ASME VIII - 2, Use...

$t_r = r\, (e^{.5P/S} - 1) =$



CASE 1: PLANE BELOW LINE OF SUPPORT (LOS)



CASE 2: PLANE ABOVE LINE OF SUPPORT (LOS)

Notes

1. A sphere supported at the equator will have maximum ring compression when the liquid level is at the equator and no gas pressure.
2. Horizontal components of thrust will cancel out horizontal components of shear when column acts through the centroid of the circular girder, and the columns are vertical.
3. Worst case for external pressure is when the sphere is half full (maximum ring compression).
4. Cold or hot insulated spheres will expand or contract differentially with respect to the base plate elevation. If the column base is bolted and grouted to its foundation pier, the following moment and shear force will occur;

   $$V = 2\,M/L \text{ and } M = \left(6\,E_m\,I\,\delta\right)/L^2$$

   This moment and shear will add compressive stresses to those already produced by the weight of steel, product and lateral loads.
5. Spheres with long columns may require two or more sets of tie rods or cross bracing. When more than one level of bracing is required, a compression strut at the common wing gusset is required.
6. Columns for spheres that are high pressure, liquid loaded, heavy wall or in high seismic areas are best with cross bracing. Conversely, columns for spheres in low seismic areas, gas filled or light wall are best with sway bracing.

References

[1] Magnusson, I. "Design of Davits, " Fluor Engineers, Inc., Irvine, Ca.
[2] Roark RJ. In: Formulas for Stress and Strain. 3rd edition. McGraw-Hill Book Co; 1954, Article 44, p. 146.
[3] Naberhaus E. Paul. "Structural Design of Bins". Chemical Engineering February 15, 1965:183–6.
[4] Lambert, F.W. "The Theory and Practical Design of Bunkers, " British Constructional Steelwork Association, Ltd., London.
[5] API-620, Recommended Rules for Design and Construction of Large, Welded, Low-Pressure Storage Tanks, 9th Edition, September 1996.
[6] AWWA D100-84, Welded Steel Tanks for Water Storage.
[7] API 650, Welded Steel Tanks for Oil Storage, 9th Edition, November 1993.
[8] Gaylord E.H., Gaylord C.N. In: "Steel Tanks, " from Structural Engineering Handbook. McGraw-Hill, Inc; 1968. section 23.
[9] Ketchum MS. In: Walls, Bins, and Grain Elevators. 3rd Edition. McGraw-Hill Book Co; 1929.

10
Transportation and Erection of Pressure Vessels

Procedure 10-1: Transportation of Pressure Vessels

The transportation of a pressure vessel by ship, barge, road, or rail will subject the vessel to one-time-only stresses that can bend or permanently deform the vessel if it is not adequately supported or tied down in the right locations. The shipping forces must be accounted for to ensure that the vessel arrives at its destination without damage.

It is very frustrating for all the parties involved to have a load damaged in transit and to have to return it to the factory for repairs. The cost and schedule impacts can be devastating if a vessel is damaged in transit. Certain minimal precautions can avoid the costly mistakes that often lead to problems. Even when all precautions are made, however, there is still the potential for damage due to unforseen circumstances involved in the shipping and handling process.

Care should be taken to ensure that the size and location of the shipping saddles, tie-downs, or lashing are adequate to hold the vessel but not deform the vessel. Long, thin-walled vessels, such as trayed columns, are especially vulnerable to these shipping forces. The important thing to remember is that someone must take the responsibility. The barge and rail people have their own concerns with regard to loading and lashing. These may or may not coincide with the concerns of the vessel designer.

The shipping forces for ships, barges, trucks, and rail are contained in this procedure. Each method of transportation has its own unique load schemes and resulting forces. Barge shipping forces will differ from rail due to the rocking motion of the seas. Rail shipments, however, go around corners at high speed. In addition, rail forces must allow for the “humping” of rail cars when they are joined with the rest of the train. Ocean shipments have to resist storms and waves without breaking free of their lashings.

Whereas horizontal vessels on saddles are designed for some degree of loading in that position, vertical vessels are not. The forces and moments that are used for the design of a vertical vessel assume the vessel is in its operating position. Vertical vessels should generally be designed to be put on two saddles, in a horizontal position, and transported by various means. That is the purpose of this procedure. Too often the details of transportation and erection are left in the hands of people who, though well versed in their particular field, are not pressure vessel specialists.

Often vessels are transported by multiple means. Thus there will be handling operations between each successive mode of transportation. Often a vessel must be moved by road to the harbor and then transferred to a barge or ship. Once it reaches its destination, it must be reloaded onto road or rail transport to the job site. There it will be off-loaded and either stored or immediately erected. A final transport may be necessary to move the vessel to the location where it will be finally erected. At each handling and transport phase there are different sets of forces exerted on the vessel that must be accounted for.

Shipping Saddles

The primary concern of the vessel designer is the location and construction of the shipping saddles to take these forces without overstressing or damaging the vessel. If saddles are to be relocated by the transporter, it is important that the new locations be reviewed. Generally only two shipping saddles should be used. However, this may not always be possible. Remember that the reason for using two saddles is that more than two saddles creates a statically indeterminate structure. You are never assured that any given saddle is going to take more than its apportioned load.

Here are some circumstances that would allow for more than two saddles to be used or for a special location of two saddles:

- Transporter objects due to load on tires.
- Transporter objects due to load on barge or ship.
- Very thin, long vessel.
- Heavy-walled vessels for spreading load on ship or transporters.

Shipping saddles can be constructed of wood or steel or combinations. The saddles should be attached to the vessel with straps or bolts so that the vessel can be moved without having to reattach the saddle. Horizontal vessels may be moved on their permanent saddles but should be checked for the loadings due to shipping forces and clearances for boots and nozzles. Shipping saddles should have a minimum contact angle of 120°, just like permanent saddles. Provisions for jacking can be incorporated into the design of the saddles to allow loading and handling operations without a crane(s).

Shipping saddles should be designed with the vessel and not left up to the transport company. In general, transportation and erection contractors do not have the capability to design shipping saddles or to check the corresponding vessel stresses for the various load cases.

Whenever possible, shipping saddles should be located adjacent to some major stiffening element. Some common stiffening elements include stiffening rings, heads (both internal and external), or cones. If necessary, temporary internal spiders can be used and removed after shipment is complete.

Key factors for shipping saddles to consider:

- Included angle.
- Saddle width.
- Type of construction.
- Lashing lugs.
- Jacking pockets.
- Method of attachment to the vessel.
- Overall shipping height allowable—check with shipper.

Recommended contact angle and saddle width:

Vessel Diameter	Contact Angle	Minimum Saddle Width
D < 13 ft-0 in.	120°	11 in.
13 ft-0 in. < D < 24 ft-0 in.	140°	17 in.
D > 24 ft-0 in.	160°	23 in.

Vessel Stresses

The stresses in the vessel shell should be determined by standard Zick's analysis. The location of shipping saddles should be determined such that the bending at the midspan and saddles is not excessive. Also, the stresses due to bending at the horn of the saddle is critical. If this stress is exceeded, the saddle angle and width of saddle should be increased. Also, move the saddle closer to the head or a major stiffening element.

Lashing

Vessels are lashed to the deck of ships and barges. In like manner they must be temporarily fixed to railcars, trailers, and transporters. Lashing should be restricted to the area of the saddle locations. Vessels are held in place with longitudinal and transverse lashings. Lashings should never be attached to small nozzles or ladder or platform clips. In some cases, lashing may be attached to lifting lugs and base rings. Lashings should not exceed 45° from the horizontal plane.

Other Key Factors to Consider

- Shipping clearances.
- Shipping orientation—pay close attention to lift lugs and nozzles.
- Shipping route.
- Lifting orientation.
- Type of transport.
- Watertight shipment for all water transportation.
- Escorts and permits.
- Abnormal loads—size and weight restrictions.
- Vessels shipped with a nitrogen purge.
- Shipping/handling plan.

Organizations That Have a Part in the Transportation and Handling of Pressure Vessels

- Vessel fabricator.
- Transport company.
- Engineering contractor.
- Railway authorities.
- Port authorities.
- Erection/construction company.
- Trailer/transporter manufacturer.
- Ship or barge captain.
- Crane company/operator.

Special Considerations for Rail Shipments

1. Any shipment may be subject to advance railroad approval.
2. Any shipment over 10 ft-6 in. wide must have railroad approval.
3. A shipping arrangement drawing is required for the following:
   a. All multiple carloads (pivot bolster required).
   b. All single carloads over 10 ft-6 in. wide.
   c. All single carloads over 15 ft-0 in. ATR (above top of rail).
   d. All single carloads that overhang the end(s) of the car and are over 8 ft-0 in. ATR.

4. Clearances must be checked for the following:
   a. Vessels greater than 9 feet in width.
   b. Vessels greater than 40 feet overall length.
   c. Vessels greater than 50 tons.
5. The railroad will need the following specific data as a minimum:
   a. Weight.
   b. Overall length.
   c. Method of loading.
   d. Loadpoint locations.
   e. Overhang lengths.
   f. Width.
   g. Height.
   h. Routing/route surveys.
   i. Center of gravity.
6. A swivel (pivot) bolster is required whenever the following conditions exist:
   a. Two or more cars are required.
   b. The capacity for a single car is exceeded.
   c. The overhang of a single car exceeds 15 feet.
7. Rated capacities of railcars are based on a uniformly distributed load over the entire length of the car. The capacity of a car for a concentrated load will only be a percentage of the rated capacity.
8. Rules for loads, loading, and capacities vary by carrier. Other variables include the types of cars the carrier runs, the availability, and the ultimate destination. Verify all information with the specific carrier before proceeding with the design of shipping saddles or locations.
9. For vessels that require pivot bolsters, the shipping saddles shall be adequately braced by diagonal tension/compression rods between the vessel and the saddle. The rods and clips attached to the vessel shell should be designed by the vessel fabricator to suit the specific requirements of the carrier.
10. If requested, rail bolsters can be returned to the manufacturer.
11. Loading arrangement and tie-downs will have to pass inspection by a representative of the railways and sometimes by an insurance underwriter prior to shipment.
12. Accelerometers can be installed on the vessel to monitor shipping forces during transit.
13. A rail expediter who accompanies the load should be considered for critical shipments.
14. The railroad will allow a fixed time for the cars to be offloaded, cleaned, and returned. Demurrage charges for late return can be substantial.

Outline of Methods of Vessel Shipping and Transportation

1. Road.
   a. Truck/tractor and trailer.
   b. Transporters—single or multiple, self-propelled or towed.
   c. Special—bulldozer.
   d. Frame adapters.
   e. Beams to span trailers or transporters.
   f. Rollers.
   g. Special.
2. Rail.
   a. Single car.
   b. Multiple cars.
   c. Special cars.
   d. Types of cars.
      - Flatcar.
      - Fishbelly flatcar.
      - Well car.
      - Heavy-duty car.
      - Gondola car.
3. Barge.
   a. River barge.
   b. Ocean-going barge.
   c. Lakes and canals.
4. Ships.
   a. Roll-on, roll-off type.
   b. Loading and off-loading capabilities.
   c. In-hull or on-deck.
   d. Floating cranes.
5. Other.
   a. Plane.
   b. Helicopter.
   c. Bulldozer.

Table 10-1
Overland shipping limits in the US

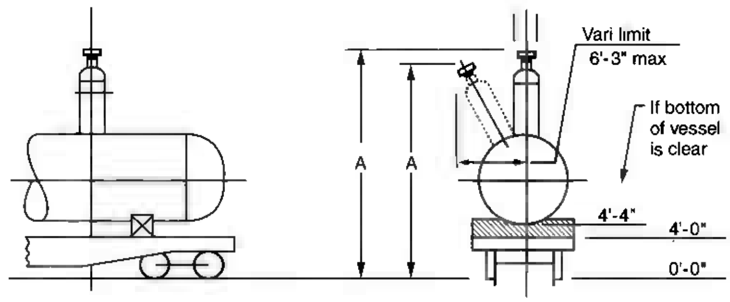
State	Length, ft	Width, ft	Height, ft	Gross Weight, lb
Alabama	150	16	16	180,000
Arizona	120	14	16	250,000
Arkansas	100	14	14	120,000
California	120	14	16	220,000
Colorado	130	17	16	228,000
Connecticut	120	15	16	250,000
Delaware	120	15	15	250,000
Florida	150	16	16	250,000
Georgia	100	14	14	120,000
Idaho	110	16	15’6"	200,000
Illinois	145	14’6"	15	250,000
Indiana	110	16	15	250,000
Iowa	120	18	16	250,000
Kansas	126	16’6"	16	250,000
Kentucky	110	16	15	250,000
Louisiana	125	18	16	250,000
Maine	125	16	16	250,000
Maryland	100	15’11	15’11	220,000
Massachusetts	115	14	14	240,000
Michigan	150	16	15	230,000
Minnesota	95	14’6"	14	250,000
Mississippi	100	14	14	120,000
Missouri	100	14	14	120,000
Montana	110	18	17	240,000
Nebraska	120	14	15’6"	212,000
Nevada	105	17	16	240,000
New Hampshire	120	15	16	250,000
New Jersey	120	18	16	220,000
New Mexico	120	14	16	250,000
New York	120	14	14	160,000
North Carolina	100	14	14	120,000
North Dakota	120	14’6"	15’6"	150,000
Ohio	100	14	14’10"	120,000
Oklahoma	100	16	16	212,000
Oregon	105	14	16	220,000
Pennsylvania	120	16	15’6"	201,000
Rhode Island	90	14	13’6"	120,000
South Carolina	125	16	14	250,000
South Dakota	120	14’6"	15’6"	150,000
Tennessee	120	16	15	250,000
Texas	125	20	18’11"	252,000
Utah	125	15	16’6"	250,000
Vermont	100	15	14	150,000
Virginia	150	14	15	150,000
Washington	150	14	16	200,000
West Virginia	150	16	15	212,000
Wisconsin	150	16	16	250,000
Wyoming	110	18	17	252,000

Note: This information in this Table is for general, reference information only and should not be relied upon for any given application. These values change regularly and local, state and national regulations should be checked for any given haul application.

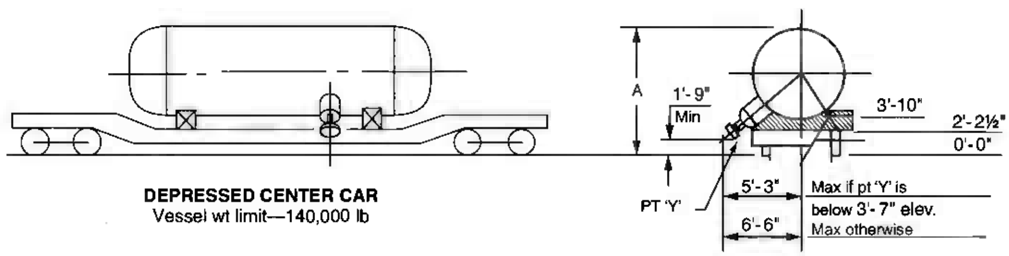
Rail—Types of Cars

Notes:

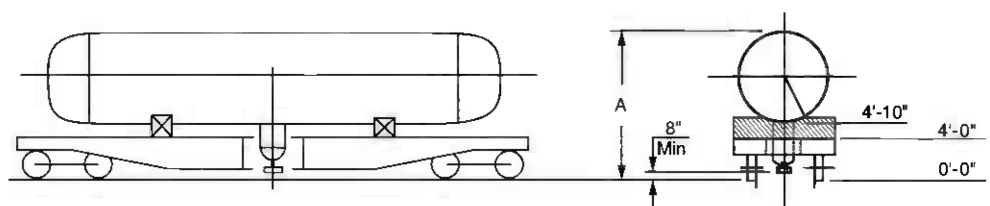
1. Allowable vessel weight ranges and limits are subject to reductions under certain conditions and as noted herein.
2. Dimension A = ATR, above top of rail.



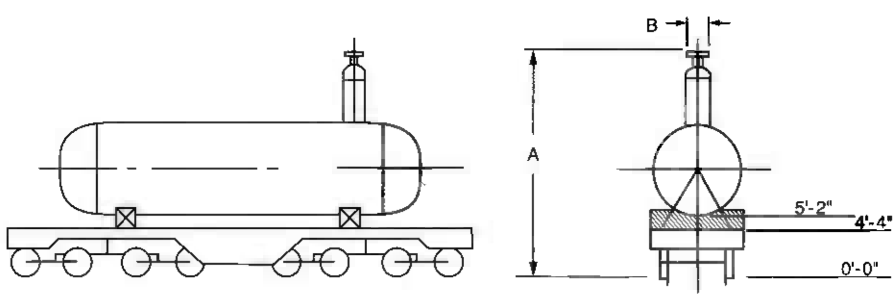
FISHBELLY FLATCAR
Vessel wt limit—140,000 lb



DEPRESSED CENTER CAR
Vessel wt limit—140,000 lb

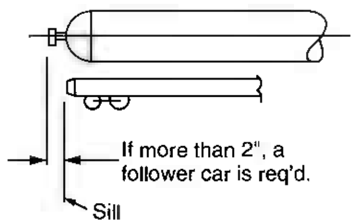


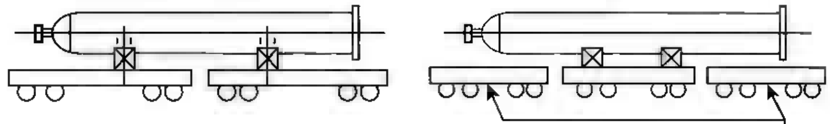
WELL-CAR
Vessel wt range (140,000 – 250,000 lb)



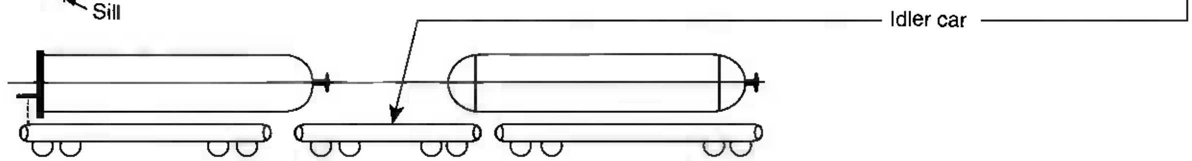
HEAVY-DUTY CAR
Vessel wt range (140,000 – 400,000 lb)

Rail—Multiple Car Loading Details

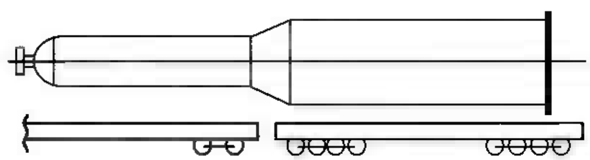




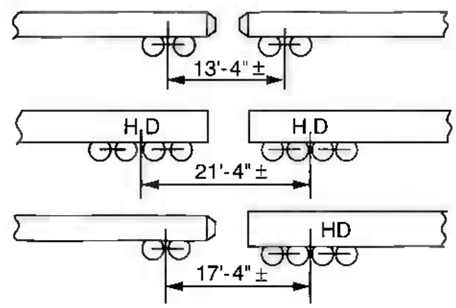
Double or Single Car—Borderline Cases



Bolster load with overhang over fourth car is acceptable. A brakewheel must be used in at least 1 of every 3 cars.

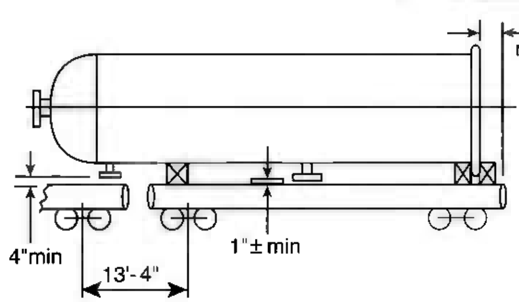


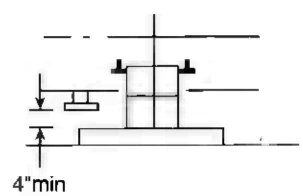
Avoid combination of dissimilar cars.

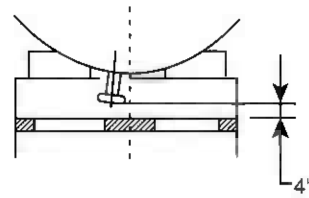


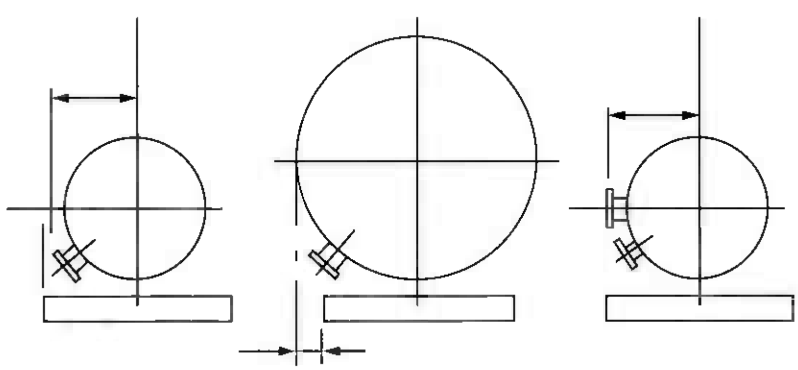
Truck Centers for Bolster Loads

Rail—Clearances



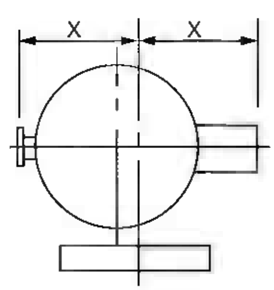






Clearance of Projections

Note: Minimum clearance to any moving part. This includes nozzles, shipping covers, or clips.



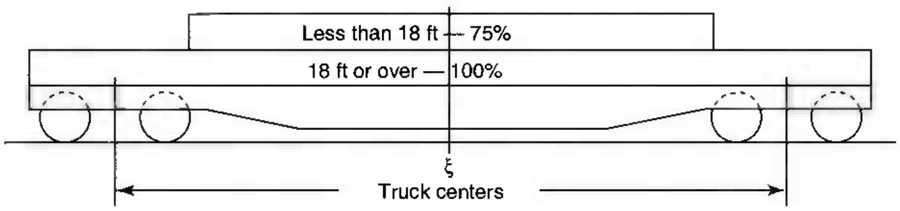
Offset Loads

Ballast may be required to offset heavy loads. "Depressed center cars" are favored for these applications.

Rail—Capacity Ratios for Concentrated Loads

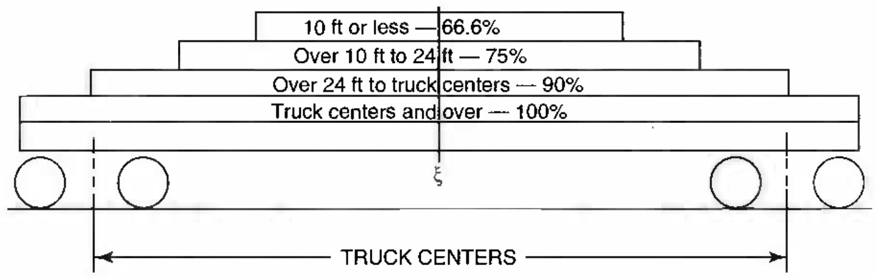
1. Flatcars with both fish belly center and fishbelly side sills and all flatcars built after January 1, 1965.

Less than 18 ft	75%
18 ft or over	100%



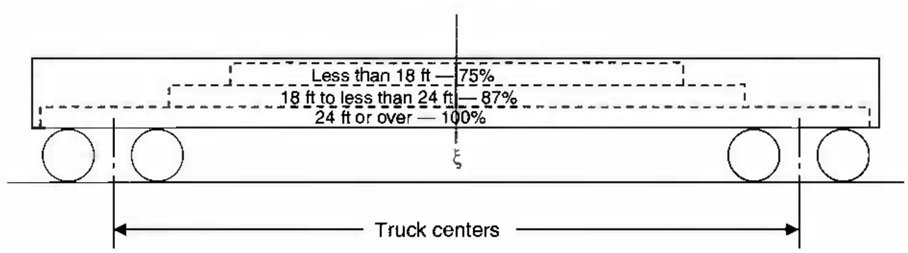
2. Flatcars not equipped with both fishbelly center and fishbelly side sills built prior to January 1,1965.

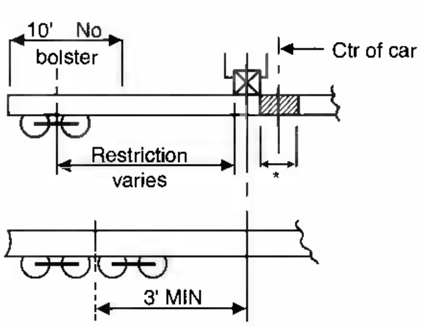
10 ft or less	66.6%
Over 10 ft to 21 ft	75%
Over 21 ft to truck centers	90%
Truck centers and over	100%



3. Gondola cars.

Less than 18 ft	75%
18 ft or less than 24 ft	87%
24 ft or over	100%

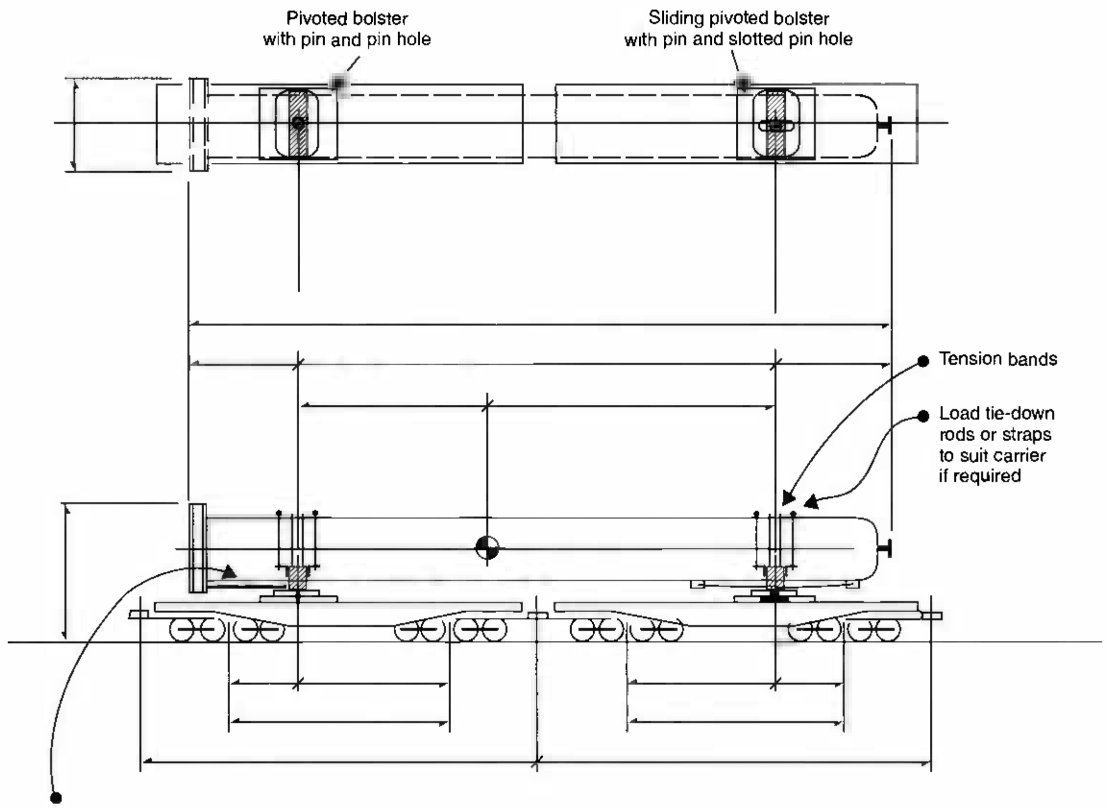




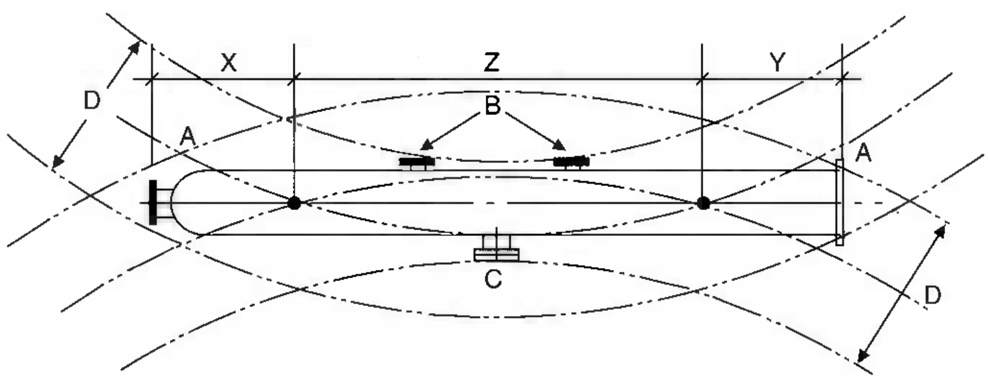
*3' restricted @ center of car to 50,000 lb except for heavy-duty cars

Bolster Locations

Rail—Details of Pivoted Bolster Loads



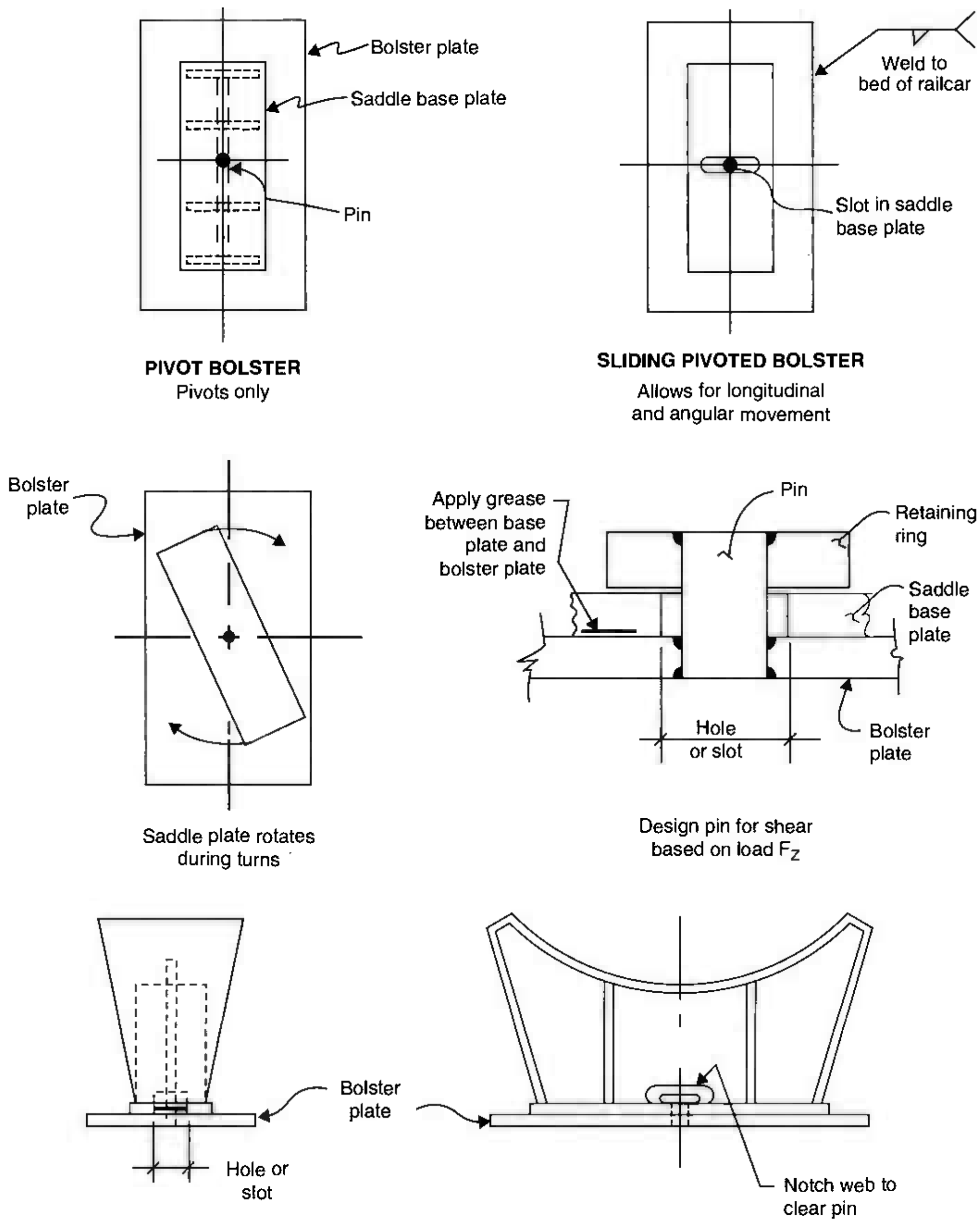
Longitudinal tie-downs are required at each saddle to suit the individual carrier. Tie-downs may consist of two brace rods, steel cables, and turnbuckles or a brace frame against the vessel base plate to take the longitudinal loads. The vessel fabricator should provide adequate clips or like attachment to the vessel for securing this bracing to the vessel shell. It is imperative that any welding to the vessel be done in the shop!



BOLSTER SETTING & CLEARANCES

1. Set X, Y, and Z so that clearance at points A, B, and C are adequate.
2. Watch relationship between bolsters and car trucks and car ends.
3. Add a minimum of 1 in. to all lateral dimensions to allow for shipping covers and small projections.
4. Dimension "D" shall be a maximum of 15 ft-5 in. of occupied space based on a 10° curve.

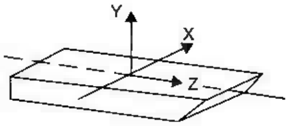
Details of Bolsters



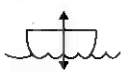
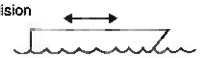
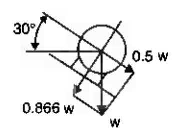
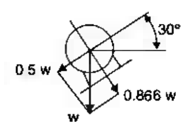
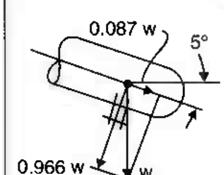
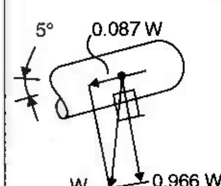
Notes:

1. Pivoting bolsters must be used for all rail shipments.
2. Pivoting bolsters must be utilized for all vessels spanning two or more railcars.
3. Design pin for shear based on full load of F_z.
4. Do not anchor the saddle plate to the bolster plate or the railway bed. The saddle plate must be free to rotate on the bolster plate. Only the bolster plate is anchored to the railway bed. The most common means of anchoring the bolster plate to the railway bed is welding. Design anchorage for a load of $\frac{1}{2}F_z$.
5. Apply grease generously between saddle base plate and bolster plate.
6. In general all clips or welds on the railcar will have to be removed, ground, and cleaned to the satisfaction of the railways prior to return.

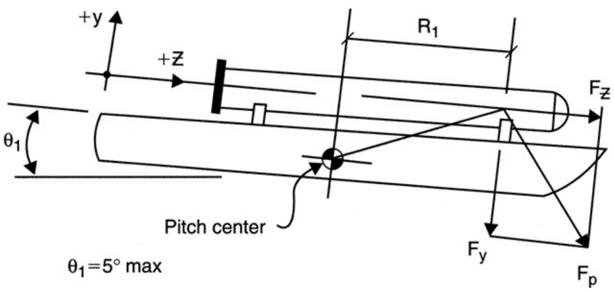
Table 10-2
Barge shipping forces



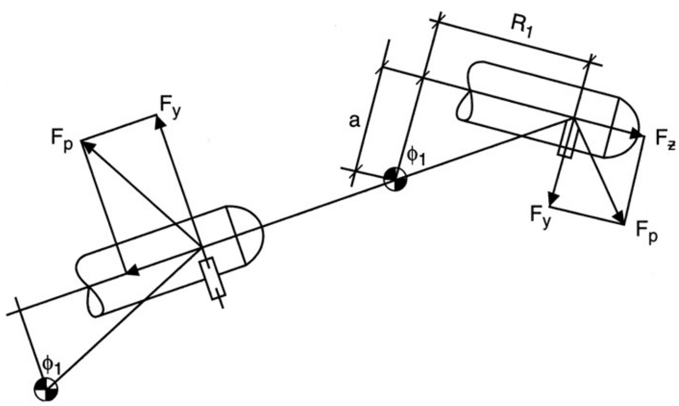
F = force due to barge motion, lb
W = shipping weight, lb
T = period of vibration of barge, secs

Case	Condition	F_x	F_y	F_z	Diagram
1	Gravity	—	−1.0w	—	—
2a	Roll 	±0.45w	−0.4w	—	—
2b		±0.45w	+0.4w	—	—
3a	Pitch 	—	−1.5w	1.0w	—
3b		—	+1.5w	1.0w	—
4	Heave 	—	1.2w	—	—
5	Collision 	—	—	±1.5w	—
6a	Roll + Gravity	+0.95w +0.05w	−1.266w	—	
6b		−0.95w −0.05w	−0.466w	—	
7a	Pitch + Gravity	—	−2.5w	±0.5w	
7b		—	+0.5w	±0.5w	

Pitch

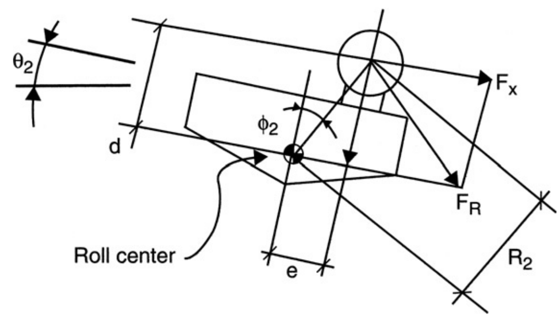


Cases 3a and 3b

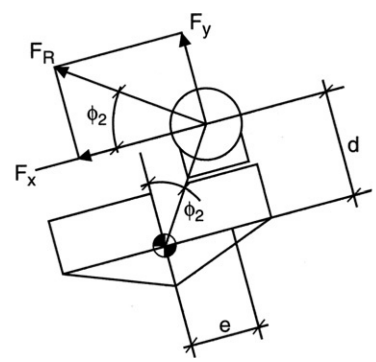


Roll

Case 2a: $\theta_2 = 30°$ max



Case 2b



Forces in Vessel Due to Pitch

General:

$$F = ma = \left(\frac{W}{g}\right)\left(\frac{2\pi}{T}\right)^2\left(\frac{R\theta\pi}{180}\right)$$

$$F = 0.0214\frac{WR\theta}{T^2}$$

$$\phi_1 = \tan^{-1}\left(\frac{a}{R_1}\right)$$

$$F_p = \frac{0.0214WR_1\theta_1}{T_1^2}$$

Case 3a: $F_y = -F_p \sin\phi_1$

$F_z = F_p \cos\phi_1$

Case 3b: $F_y = F_p \sin\phi_1$

$F_z = -F_p \cos\phi_1$

Forces in Vessel Due to Roll

$$\phi_2 = \tan^{-1}\left(\frac{e}{d}\right)$$

$$R_2 = \frac{e}{\sin\phi_2}$$

$$F_R = \frac{0.0214WR_2\theta_2}{T_2^2}$$

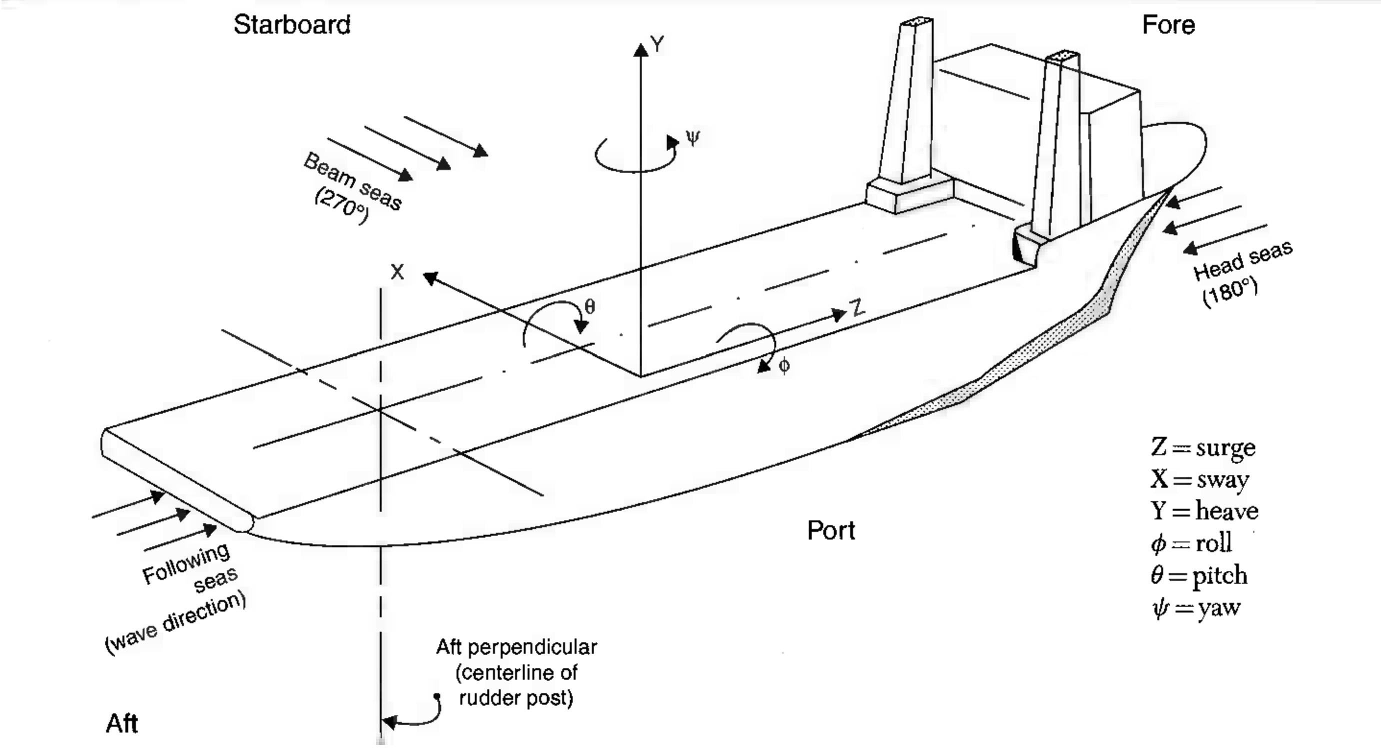
Case 2a: $F_y = -F_R \sin\phi_2$

$F_x = F_R \cos\phi_2$

Case 2b: $F_y = F_R \sin\phi_2$

$F_x = -F_R \cos\phi_2$

Directions of Ship Motions



The job of the designer is to translate the loads resulting from the movement of the ship into loads applied to the pressure vessel that is stored either at or below decks. The ship itself will rotate about its own center of buoyancy (CB) depending on the direction of the sea and the ship's orientation to that direction of sea. The vessel strapped to its deck is in turn affected by its location in relation to the CB of the ship. For example, if the CG of the vessel is located near the CB of the ship, the forces are minimized. The farther apart the two are in relation to each other, the more pronounced the effect on the vessel.

The ship's movement translates into loads on the three principal axes of the vessel. Saddles and lashings must be strong enough to resist these external forces without exceeding some allowable stress point in the vessel. The point of application of the load is at the CG of the vessel. These loads affect the vessel in the same manner as seismic forces do. In fact, the best way to think of these loads is as vertical and horizontal seismic forces. Vertical seismic forces either add or subtract to the weight of the vessel. Horizontal seismic forces are either transverse or longitudinal.

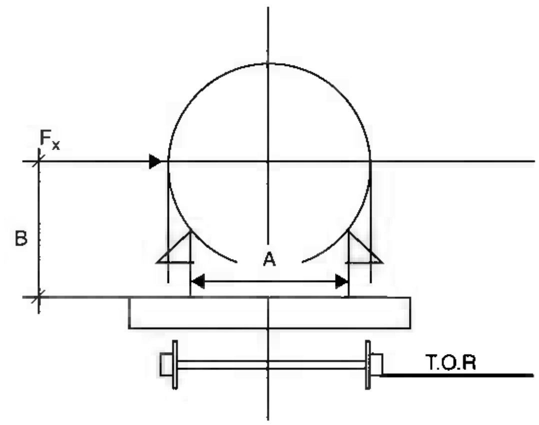
The X, Y, and Z axes translate into and are equivalent to the following loadings in the vessel:

X axis: horizontal transverse.
Y axis: corresponds to vertical loads by either adding or subtracting from the weight of the vessel.
Z axis: longitudinal axis of the vessel. All Z axis loads are longitudinal loadings.

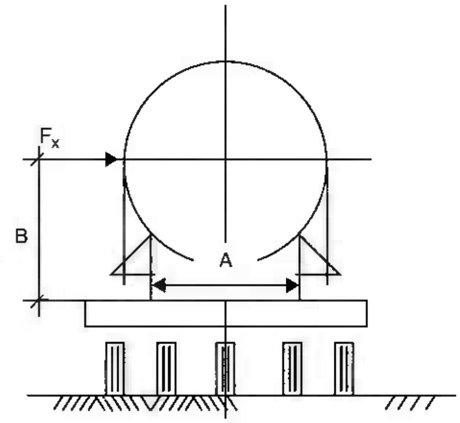
Load Combinations for Sea Forces

1. dead load + sway + heave + wind
2. dead load + surge + heave + wind

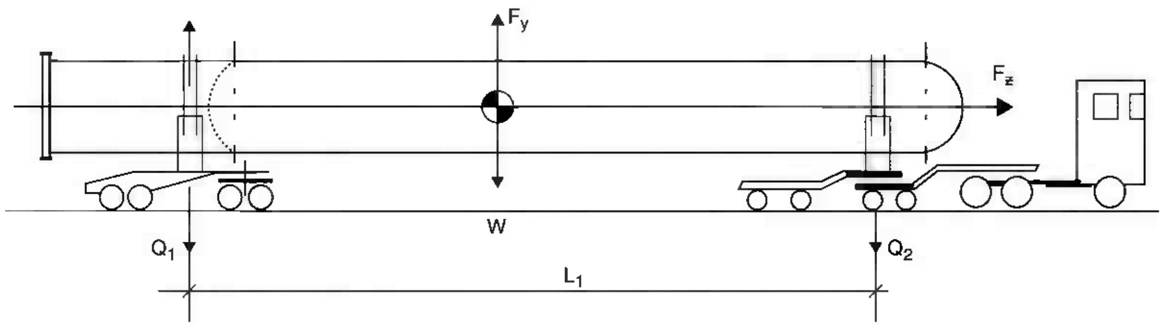
Forces on Truck and Rail Shipments



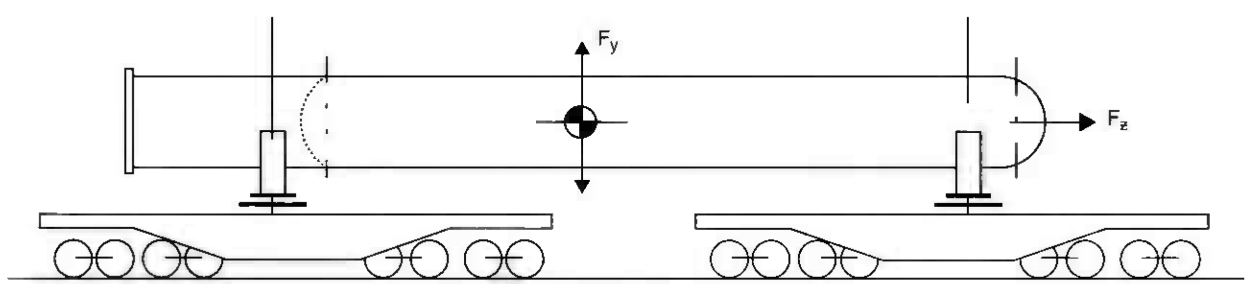
RAIL



TRUCK / TRAILER



TRUCK / TRAILER



RAIL

Axle Loads

The number of axles that must be placed under a load is determined by analyzing the weight restrictions and allowable bearing load from local, state or national regulations. The transportation contractor is responsible for determining the axle loads based on the equipment used and the weight and distribution of the loads. The authorities that permit the load will require an analysis of the axle loads to ensure that the roadbed is not overloaded. Axle loads include the weight of the vessel, transport saddles, beams, hauler (tractor), dollies, etc.

There are three different methods used to distribute the loads to the axles;

1. Flatbed: This method uses conventional tractor-trailer assembly with various numbers of axles and wheels under the trailer bed to distribute the weight to the road surface.
2. Bolstered: This method is used for abnormally long loads in which the sets of axles are attached directly to the load via the transport saddle. Both sets of axles will have steering capabilities.
3. Bolstered loads using equalizing transporting beams: This method is much the same as the bolstered, long vessel load. In this case the load is too heavy for a flatbed, yet too short for bolstered axles. The solution is to utilize beams between bolsters to suspend the load.

Transporter Stability

There are two types of stability checks that should be performed on each load. The first has to do with the tipping point of the load relative to the roadway as the load shifts due to the camber of the road. The second has to do with the turning radius of bolstered loads. As the load goes around curves, the C.G. shifts from being in line with the dollies, to an eccentric condition. In tight curves the eccentricity of the load can overload the outer set of wheels to the point of rollover. The two cases are;

1. Rollover stability due to road camber
2. Turning stability due to turning radius

Case 1:

Due to the camber in roads, the load will be subject to various angles, θ, that will change the location of the center of gravity of the load. On a flat surface, the center of gravity is in line with the centerline of the trailer. As the road camber increases, the C.G. is steadily moved toward the outer set of wheels. At some point the wheels are overloaded on one side and the entire assembly reaches a tipping point. Beyond this, the trailer turns over and the load is lost. This condition has resulted in numerous rollovers.

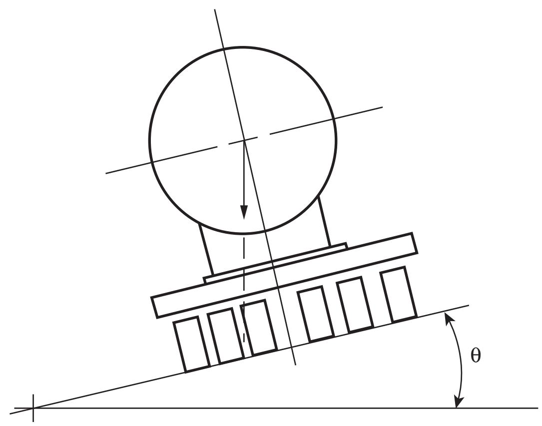


Figure 10.1. Rollover stability.

Case 2:

For bolstered loads, the vessel must swivel on the deck of the trailer in order to accommodate curves in the road and corners. As the curve or corner is negotiated, the actual CG gets further away from the projected load point. This is true whether you have a single pivoting bolster or two, however the situation is more pronounced with the double pivoting case. There have been a number of rollovers as a result of the eccentricity, ie, shifting the load to the outer row of wheels until the load becomes unstable.

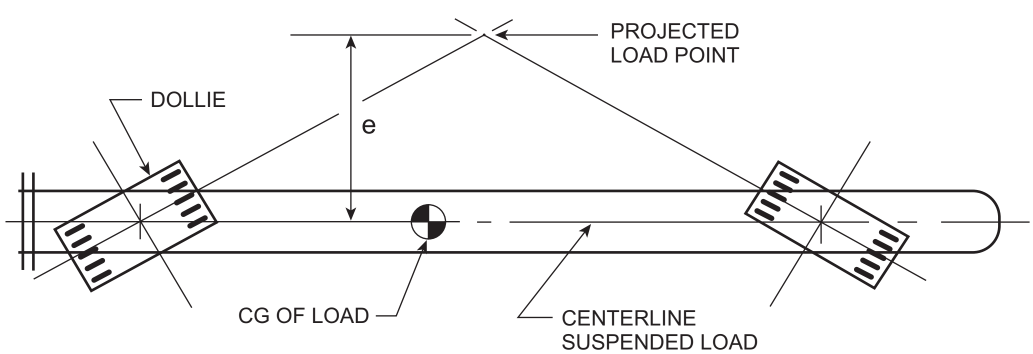
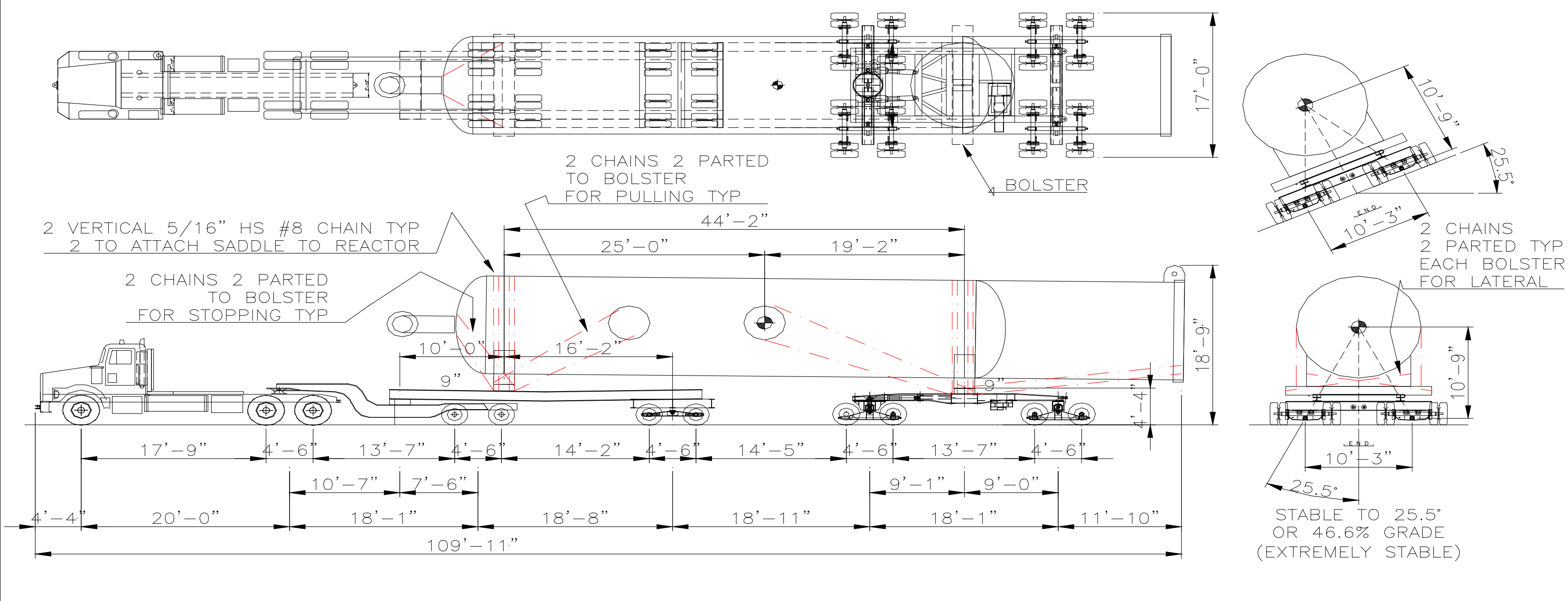


Figure 10.2. Turning stability.

AXLE LOADS - TYPICAL EXAMPLE - 11 AXLE, DUAL LANE TRANSPORTER



AXLE LOADS

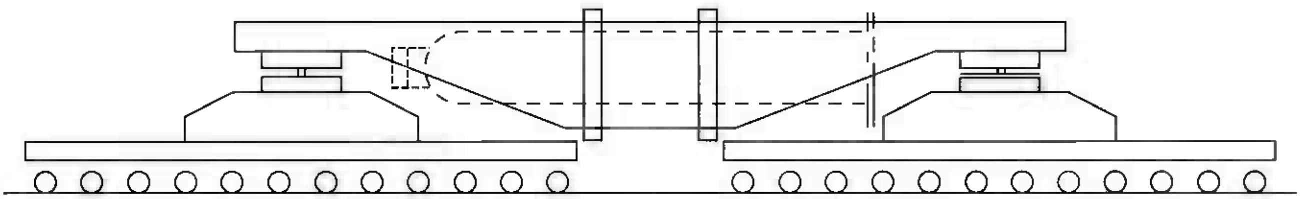
AXLE NO.		1	2-3	4-5	6-7		8	9	10	11	TOTALS
WIDTH (ft)		8	8	10	10		17		17		
TARE WEIGHT(lbs)		12200	15360	9640	12000		5450	5450	5450	5450	71000
MAX PAYLOAD(lbs)		1510	28610	46510	43680		39498	39498	39498	39498	278300
MAX LOAD (lbs)		13710	43970	56150	55680		44948	44948	44948	44948	349300
NO. OF TIRES		2	8	16	16		8	8	8	8	74
TIRE LOADING		6855	5496	3509	3480		5618	5618	5618	5618	
ALLOW LOAD		14000	45000	58000	58000		45000	45000	45000	45000	

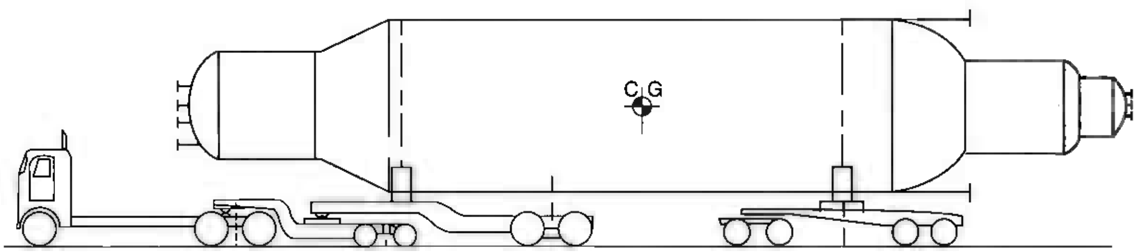
EQUIPMENT WEIGHT	
TRACTOR	22000
JEEP	9200
BEAMS	12000
DOLLY	6000
DBL DOLLY	21800
TOTAL	71000

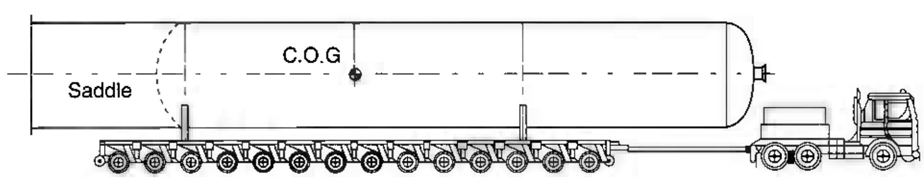
DESCRIPTION	
ITEM	REACTOR
FOR	
WEIGHT	278,300
SIZE	76'-6"LX15'-9"W X 14'-9"H
ORIGIN	
DESTINATION	

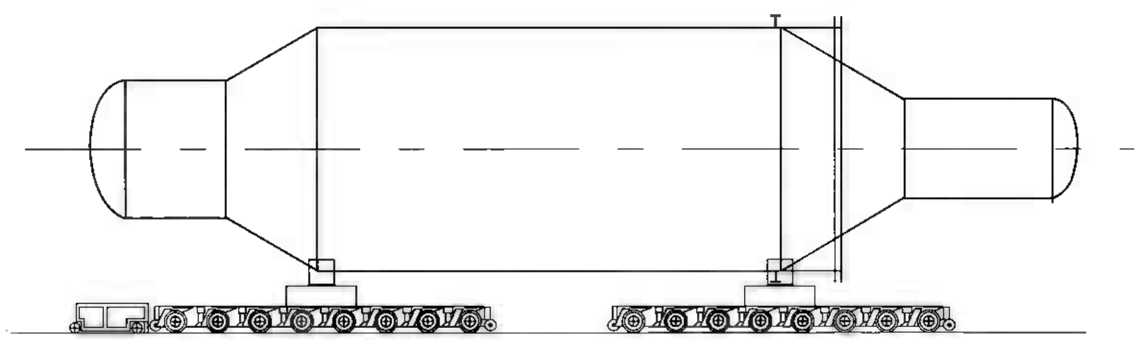
Examples of Road Transport

If a vessel is too heavy for one trailer and too short to span two trailers, then a pair of outrigger beams can be used to span the trailers and still distribute the load to the trailers. A wide variety of trailers, self-propelled transporters, and beam configurations have been utilized for these applications. Short, squat, heavy vessels are the most common.

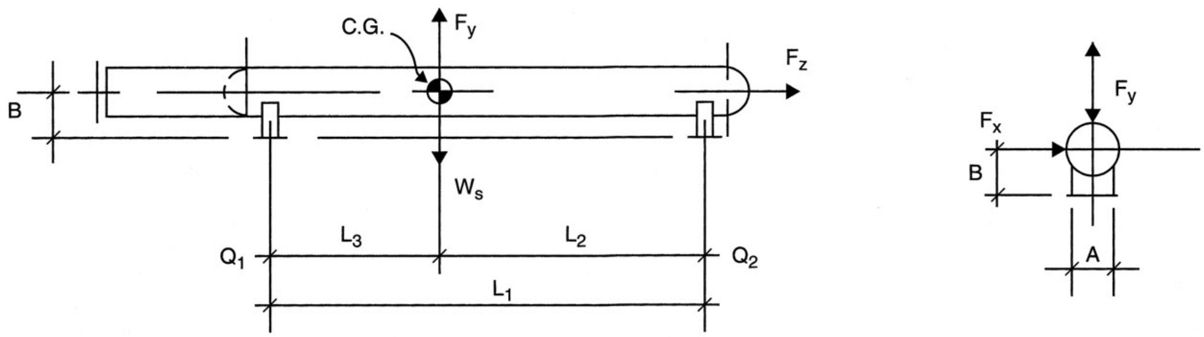








Summary of Loads/Forces on Vessels During Transportation



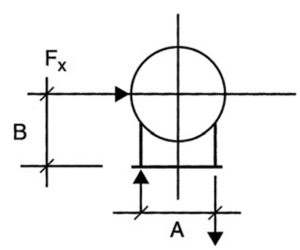
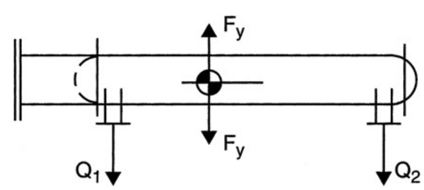
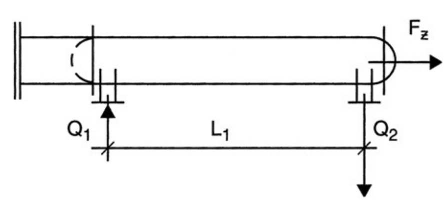
Loads F_x, F_y, $F_z = KW_s$

Verify coefficients with transport contractor/shipper.

Table 10-3
Transportation load coefficients, K

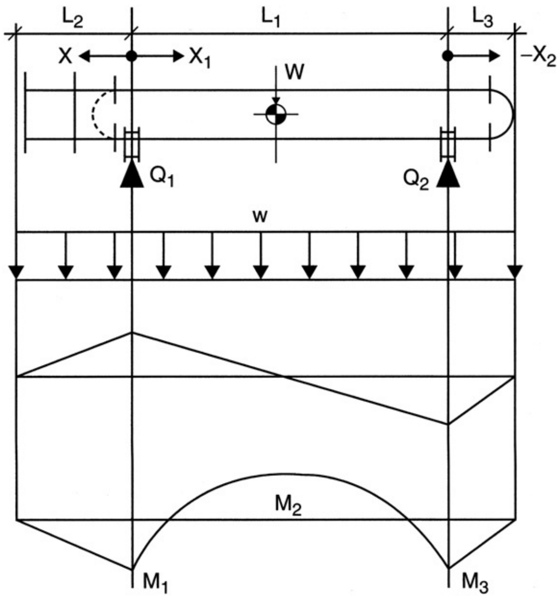
Forces	Road	Rail	Barge	Ocean
F_x	0.5	1.0	0.95	1.0
F_y	1.5	2.0	1.3	1.5
F_z	1.0	1.5	1.5	1.5

Table 10-4
Load per saddle due to transport forces

Due to . . .	Load per Saddle	Diagram
F_x	$Q_1 = \frac{W_s L_2}{L_1} + \frac{F_x B}{2A}$ $Q_2 = \frac{W_s L_3}{L_1} + \frac{F_x B}{2A}$	
F_y	$Q_1 = \frac{(W_s + F_y) L_2}{L_1}$ $Q_2 = \frac{(W_s + F_y) L_3}{L_1}$	
F_z	$Q_1 = \frac{W_s L_2}{L_1} + \frac{F_z B}{L_1}$ $Q_2 = \frac{W_s L_3}{L_1} + \frac{F_z B}{L_1}$	

Load Diagrams for Moments and Forces

Case 1



Note: W = weight of vessel plus any impact factors.

$$\mathrm{OAL} = L_1 + L_2 + L_3 \quad w = \frac{W}{\mathrm{OAL}}$$

$$Q_1 = \frac{w\left[(L_1 + L_2)^2 - L_3^2\right]}{2L_1}$$

$$Q_2 = W - Q_1$$

$$M_1 = \frac{wL_2^2}{2}$$

$$M_2 = Q_1\left(\frac{Q_1}{2w} - L_2\right)$$

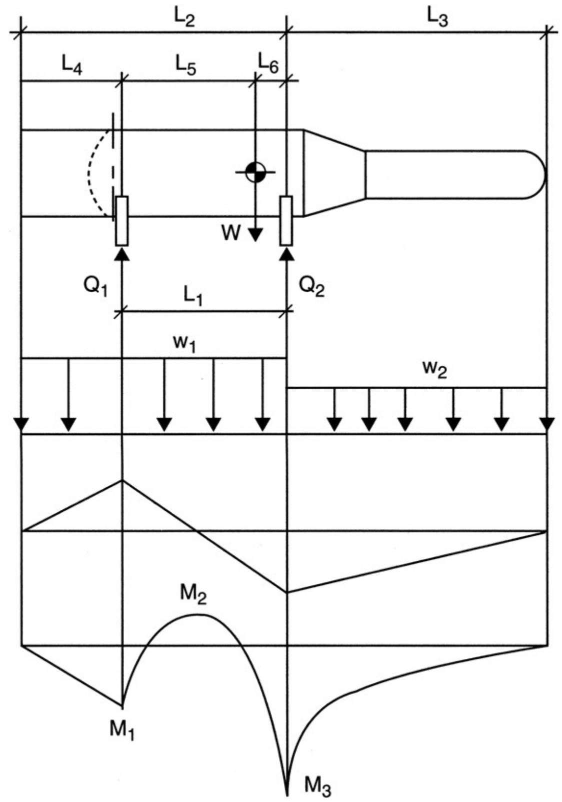
$$M_3 = \frac{wL_3^2}{2}$$

$$M_x = \frac{w(L_2 - X)^2}{2}$$

$$M_{x1} = \frac{w(L_2 + X_1)^2}{2} - Q_1X_1$$

$$M_{x2} = \frac{w(L_3 - X_2)^2}{2}$$

Case 2



$$w_1 = \frac{W_1}{L_2}$$

$$w_2 = \frac{W_2}{L_3}$$

$$Q_1 = \frac{WL_6}{L_1}$$

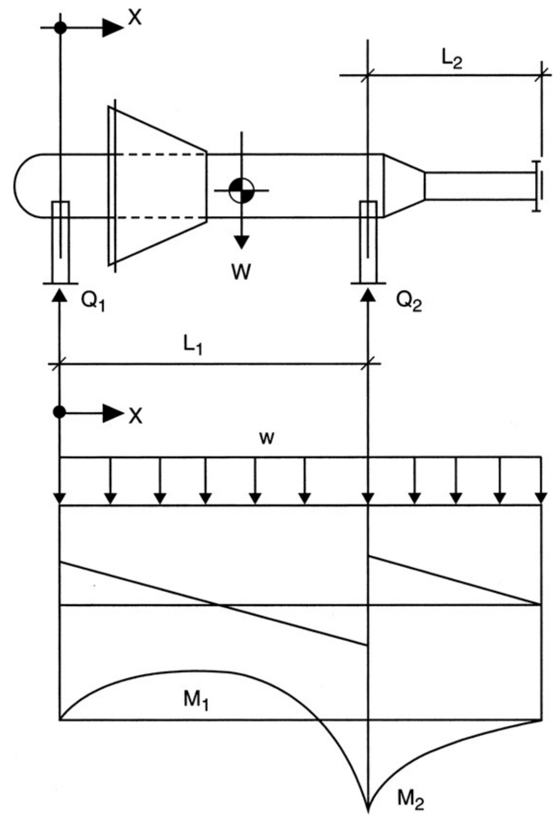
$$Q_2 = W - Q_1$$

$$M_1 = \frac{w_1L_4^2}{2}$$

$$M_2 = \frac{M_1 + M_3}{2} - \frac{w_1L_1^2}{8}$$

$$M_3 = \frac{w_2L_3^2}{2}$$

Case 3



$$Q_1 = \frac{WL_1}{2(L_1+L_2)} - \frac{WL_2^2}{2L_1(L_1+L_2)}$$

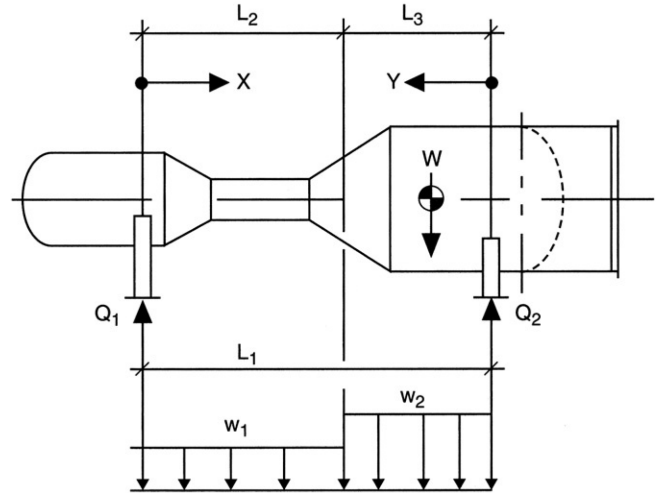
$$Q_2 = \frac{WL_1}{2(L_1+L_2)} + \frac{WL_2}{L_1+L_2} + \frac{WL_2^2}{2L_1(L_1+L_2)}$$

$$M_1 = \frac{Q_1^2(L_1+L_2)}{2W}$$

$$M_2 = Q_1L_1 - \frac{WL_1^2}{2(L_1+L_2)}$$

$$M_x = Q_1X - \frac{(WX^2)}{2(L_1+L_2)}$$

Case 4



$$w_1 = \frac{W_1}{L_2}$$

$$w_2 = \frac{W_2}{L_3}$$

$$Q_1 = \frac{w_1L_2(2L_1-L_2)+w_2L_3^2}{2L_1}$$

$$Q_2 = \frac{w_2L_3(2L_1-L_3)+w_1L_2^2}{2L_1}$$

Moment at any point X from Q_1:

$$M_x = Q_1X - \frac{w_1X^2}{2}$$

Moment at any point Y from Q_2:

$$M_y = Q_2(L_1-Y) - \frac{w_2(L_1-Y)^2}{2}$$

Transportation-Vertical Vessel on Two Saddles, Uniform Load Case, With Incorporation of Shipping Factors

Notation

F_2 = Additional load on Q_2, Lbs
F_Z = Longitudinal loading due to shipping forces, Lbs
F_Y = Vertical Loading due to shipping forces, Lbs
K_Z = Longitudinal impact factor
K_Y = Vertical impact factor
Q_1, Q_2 = Saddle loads without impact factors, Lbs
Q_1', Q_2' = Saddle loads with impact factors, Lbs
W = Shipping weight of vessel without impact factors, Lbs
W_T = Shipping weight with impact factors, Lbs
w_1 = Uniform load, without F_Y, Lbs/Ft
w_2 = Uniform load including F_Y, Lbs/Ft

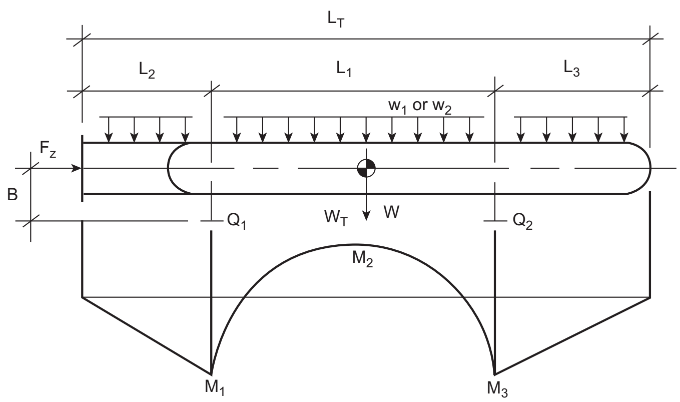


Figure 10.3. Data for uniform load case

NOTE: Assume that F_Z & F_Y do not occur at the same time

Case 1: Adding Load for F_z

- Longitudinal load, F_Z

$$F_Z = K_Z\, W$$

- Uniform load, w_1

$$w_1 = W/L_T$$

- Additional load on saddle, F_2

$$F_2 = (F_Z\, B)/L_1$$

- Saddle loads, Q_1, Q_2, Q_1', Q_2'

$$Q_1 = w_1\left[(L_1 + L_2)^2 - L_3^2\right]\Big/2\, L_1$$
$$Q_1' = Q_1 - F_2$$
$$Q_2 = W - Q_1$$
$$Q_2' = Q_2 + F_2$$

Case 2: Adding Loads for F_y

- Longitudinal load, F_Y

$$F_Y = K_Y\, W$$

- Vertical load, F_Y

$$W_T = W + F_Y$$

- Uniform load, w_2

$$w_2 = W_T/L_T$$

- Saddle loads, Q_1' & Q_2'

$$Q_1' = w_2\left[(L_1 + L_2)^2 - L_3^2\right]/2L_1$$
$$Q_2' = W_T - Q_1'$$

Select worst case and calculate moments;

$$M_1 = w_n\, L_2^2/2$$
$$M_2 = Q_1'\left(Q_1'/\left(2\, w_n\right) - L_2\right)$$
$$M_3 = w_n\, L_3^2/2$$

Sample Problem

Given;

B = 15.75 ft
L_1 = 124 ft
L_2 = 24 ft
L_3 = 21 ft
L_T = 169 ft

$W = 741$ kips
$K_Y = .5$
$K_Z = .6$

Calculation

$$F_Z = K_Z\ W = .6\ (741) = 444.6 \text{ kips}$$

$$F_Y = K_Y\ W = .5\ (741) = 370.5 \text{ kips}$$

$$W_T = W + F_Y$$

$$= 741 + 370.5 = 1,111.5 \text{ kips}$$

$$w_1 = W/L_T = 741/169 = 4.38 \text{ kips/ft}$$

$$w_2 = W_T/L_T = 1111.5/169 = 6.58 \text{ kips/ft}$$

$$F_2 = (F_Z\ B)/L_1 = [444.6\ (15.75)]/124$$

$$= \pm\ 56.47 \text{ kips}$$

Case 1: Adding Load for F_Z

- Saddle loads, Q_1, Q_2, Q_1', Q_2'

$$Q_1 = w_1 \left[(L_1 + L_2)^2 - L_3^2\right] \Big/ 2\ L_1$$

$$= 4.38 \left[(124 + 24)^2 - 21^2\right] \Big/ 2(124)$$

$$= 379 \text{ kips}$$

$$Q_1' = Q_1 - F_2 = 379 - 56.5 = 322.5 \text{ kips}$$

$$Q_2 = W - Q_1 = 741 - 379 = 362 \text{ kips}$$

$$Q_2' = Q_2 + F_2 = 362 + 56.5 = 418.5 \text{ kips}$$

Case 2: Adding Load for F_Y

- Saddle loads, Q_1' & Q_2'

$$Q_1' = w_2 \left[(L_1 + L_2)^2 - L_3^2\right] \Big/ 2\ L_1$$

$$= 6.58 \left[(124 + 24)^2 - 21^2\right] \Big/ 2\ (124)$$

$$= 569 \text{ kips}$$

$$Q_1' = W_T - Q_1' = 1111.5 - 569$$

$$= 542.5 \text{ kips}$$

Worst case is Case 2;
Determine moments…

$$M_1 = \left(w_2\ L_2^2\right)/2 = \left(6.58\left(24^2\right)\right)/2$$

$$= 1,895 \text{ ft} - \text{kips}$$

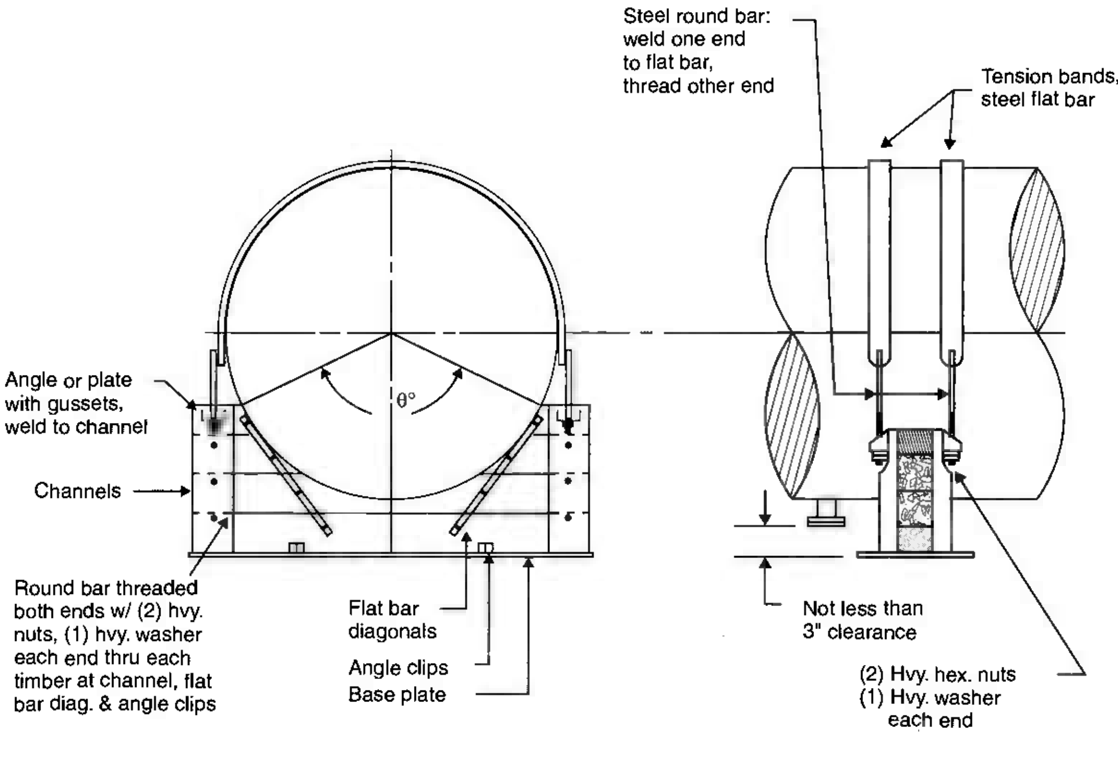
$$M_2 = Q_1'\left(Q_1'/\left(2\ w_2\right) - L_2\right)$$

$$= 569\ (569/(2 \cdot 6.58) - 24) = 10,946 \text{ ft} - \text{kips}$$

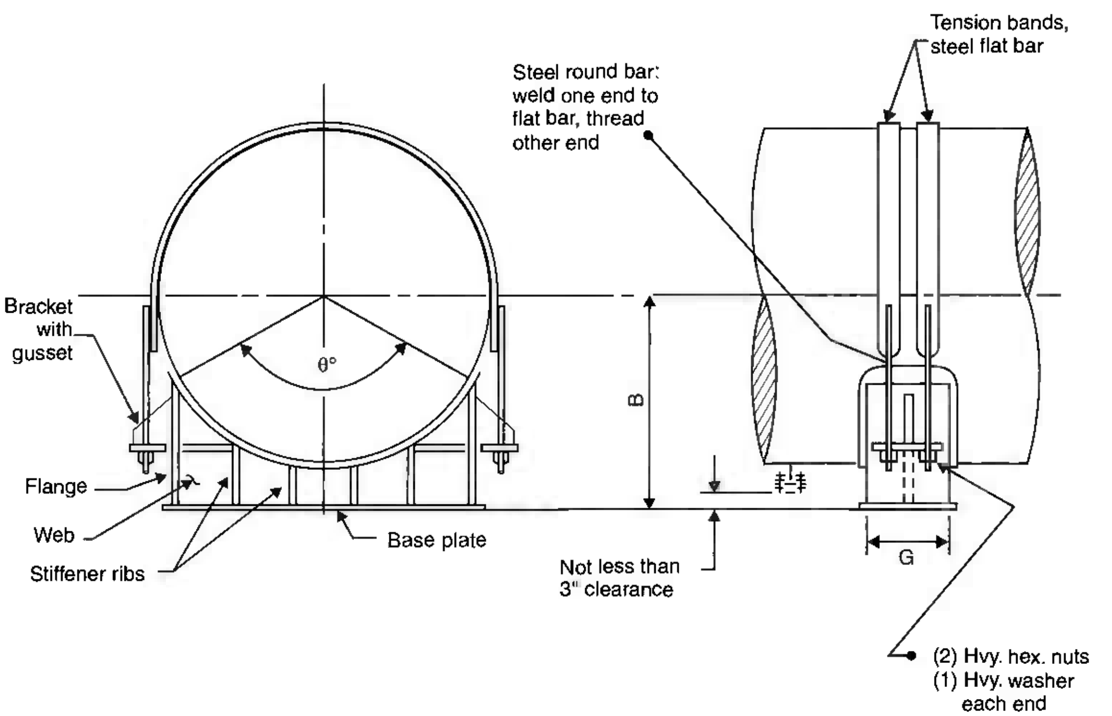
$$M_3 = \left(w_2\ L_3^2\right)/2 = \left(6.58\left(21^2\right)\right)/2 = 1,450 \text{ ft} - \text{kips}$$

Use these moments and loads to determine stresses in shell.

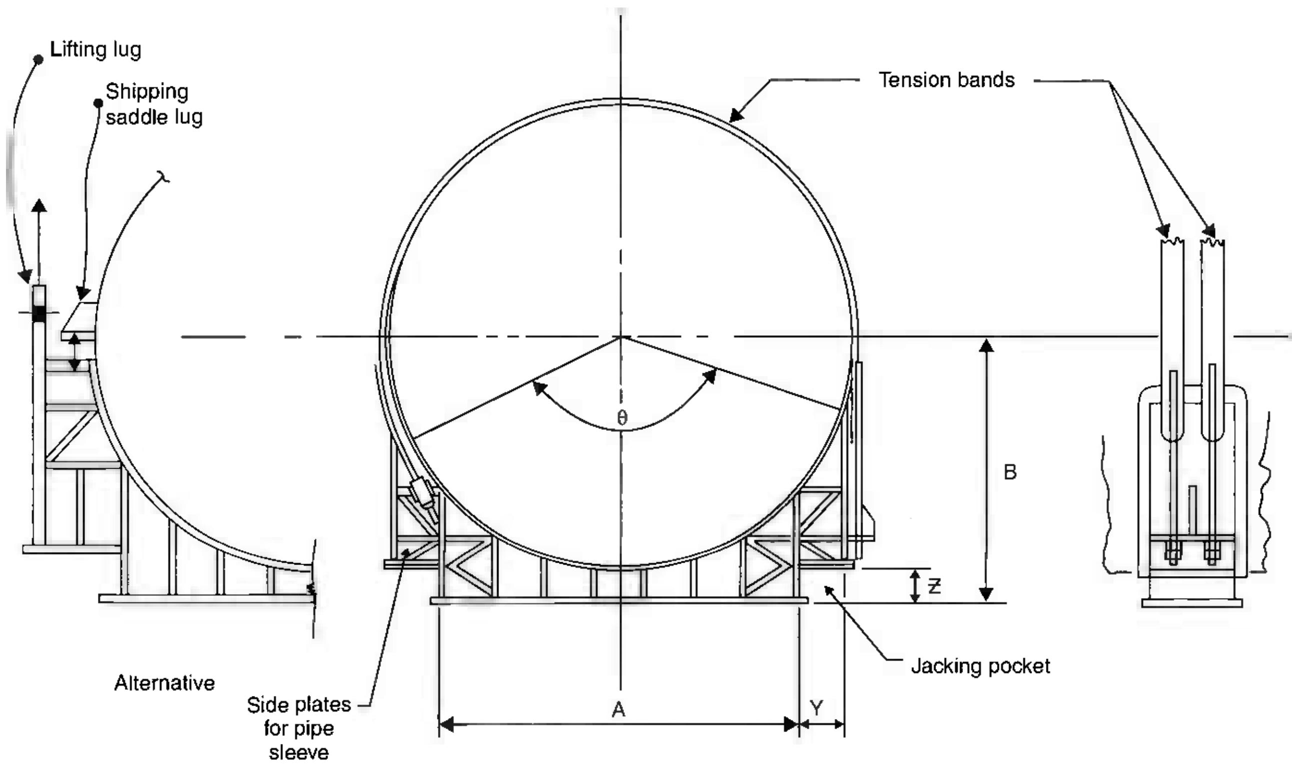
Shipping Saddles



TIMBER CONSTRUCTION

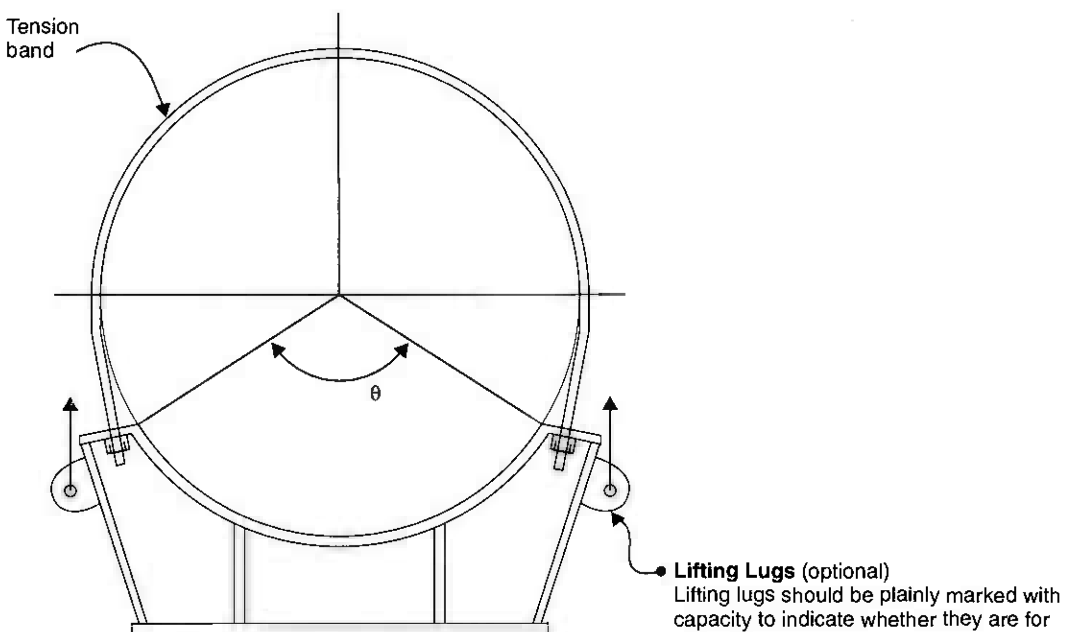


STEEL CONSTRUCTION



Shipping Saddle

Steel Construction with Jacking Pocket

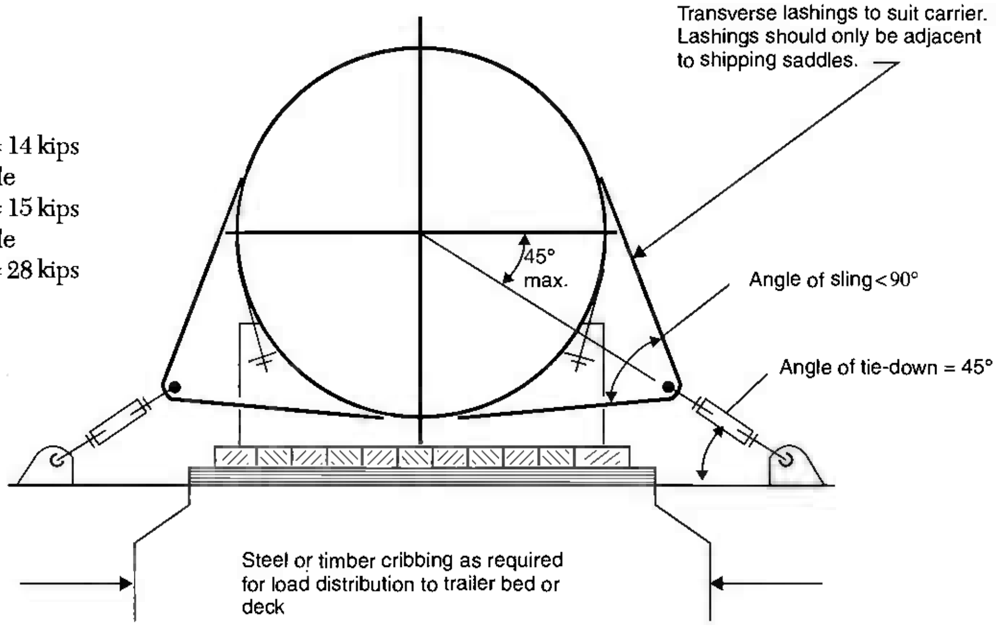


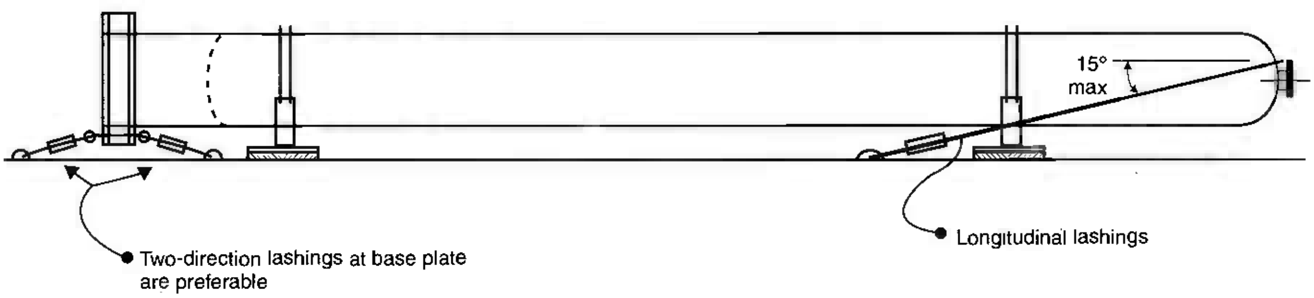
Alternate Construction

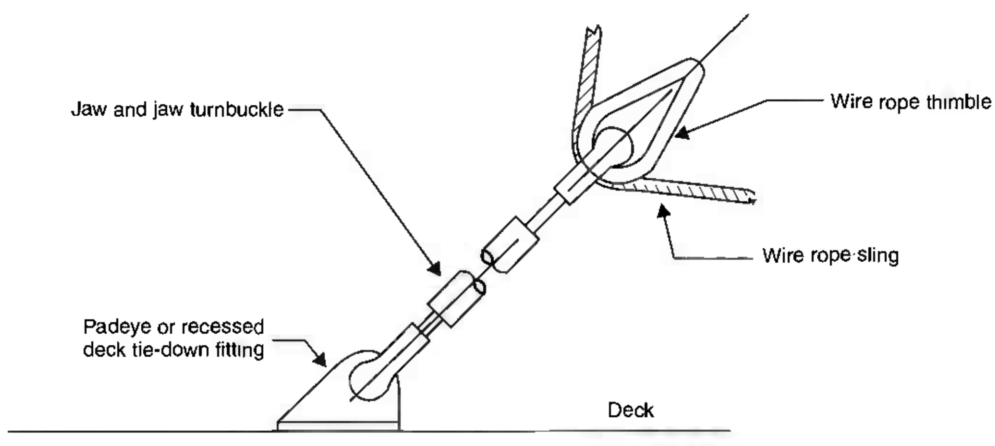
Lashing

Reference

- 1-in. wire rope
  Allowable load = 14 kips
- $1\frac{1}{4}$-in. turnbuckle
  Allowable load = 15 kips
- $1\frac{3}{4}$-in. turnbuckle
  Allowable load = 28 kips
- 1-in. Shackle
  SWL = 17 kips
- 1½-in. Shackle
  SWL = 34 kips





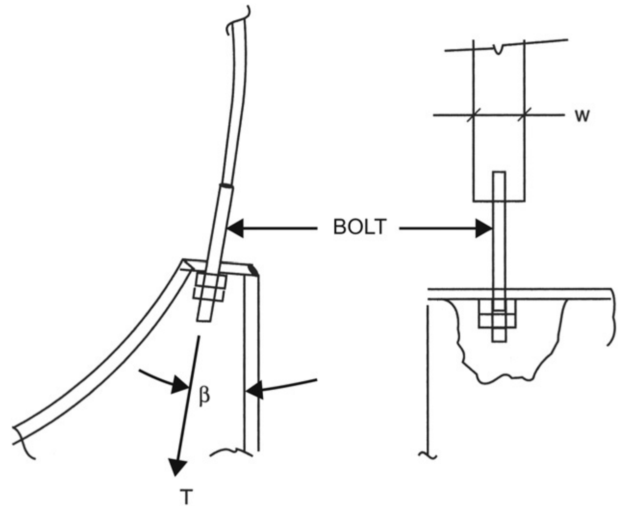


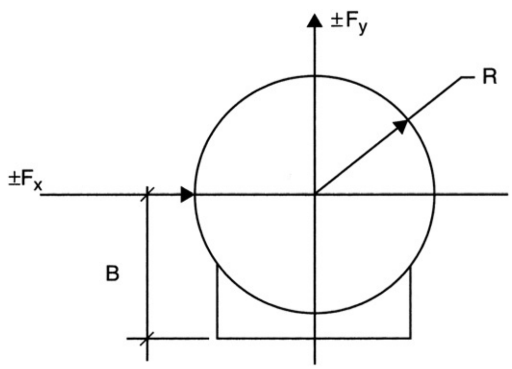
Detail of Lashing to Deck

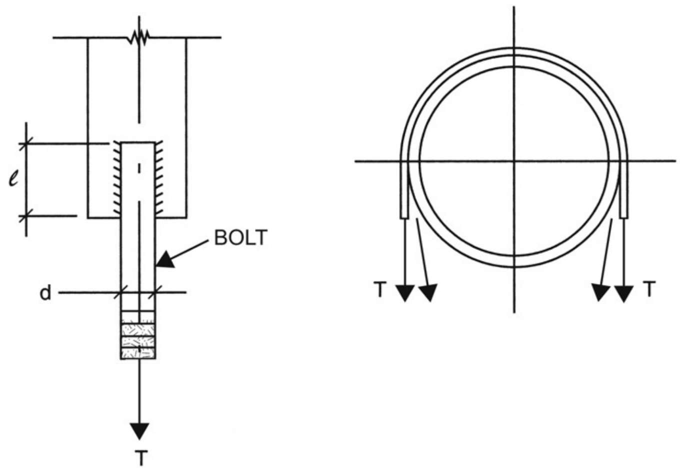
Tension Bands on Saddles

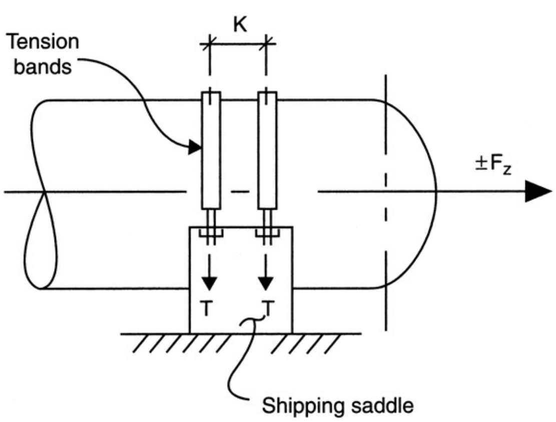
Notation

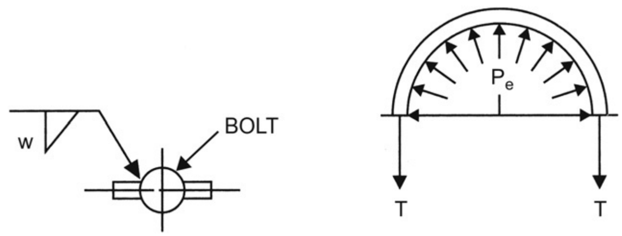
A_r = area required, in.^{2}
A_s = area of bolt, in.^{2}
A_b = area of band required, in.^{2}
A_w = allowable load on weld, lb/in.
B = saddle height, in.
d = bolt diameter, in.
f = load on weld, kips/in.
F_t = allowable stress, tension, psi
F_x, F_y, F_z = shipping, external forces, lb
K = maximum band spacing, in.
N = number of bands on one saddle
P_e = equivalent external pressure, psi
R = outside vessel radius, in.
T = tension load in band, lb
$T_{1,2,3}$ = load cases in bolt and band, lb
T_b = tension load in bolt, lb
W_s = weight of one saddle, lb
β = angle of tension bands, degrees
σ_a = stress in bolt, psi
σ_b = stress in band, psi











- *Find tension in band, T_1, due to shipping forces on saddle, F_x and F_y.*

$$T_1 = \cos\beta\left(\frac{F_xB}{4RN} + \frac{F_y - W_s}{4N}\right)$$

- *Area required for bolt.*

$$A_r = \frac{T_1}{F_t}$$

- *Find bolt diameter, d.*

$$d = \sqrt{\frac{4A_r}{\pi}}$$

Select nominal bolt diameter:

$A_s =$

- *Find maximum stress in bolt due to manual wrenching, σ_a.*

$$\sigma_a = \frac{45,000}{\sqrt{d}}$$

Table 10-5
Allowable load, weld

Weld Size, w	E60XX*	E70XX*
3/16 in.	2.39	2.78
1/4 in.	3.18	3.71
5/16 in.	3.98	4.64
3/8 in.	4.77	5.57
7/16 in.	5.56	6.50

Kips/in. of weld.

- *Maximum tension load in bolts, T_2.*

$$T_2 = \sigma_a A_s$$

- *Load due to saddle weight, T_3.*

$$T_3 = \frac{W_s}{2N}$$

Note: Include impact factor in weight of saddle.

- *Find maximum load, T.*

$T =$ greater of T_1, T_2, or T_3.

- *Load on weld, f.*

$$f = \frac{T}{4\ell}$$

- Determine size of weld from table based on load, f.

Use w =

- *Maximum band spacing, K.*

$$K = \frac{4\sqrt{Rt}}{1.285}$$

- *Find area required for tension band, A_r.*

$$A_r = \frac{T}{F_t}$$

Use:

- *Check shell stresses due to force T, P_e.*

$$P_e = \frac{4T}{\pi RK} < \text{ASME factor "B"}$$

Alternate Procedure

Tension Band Notation

N = Number of bands on one saddle
Q = Total load on one saddle, Lbs
R = Outside vessel radius, in
T = Tension load in band, Lbs
β = Angle of tension bands, degrees
K_1 = Transverse shipping coefficient
K_2 = Vertical shipping coefficient
X = Horizontal distance to centroid of saddle reaction, in
Y = Vertical distance to centroid of saddle, in

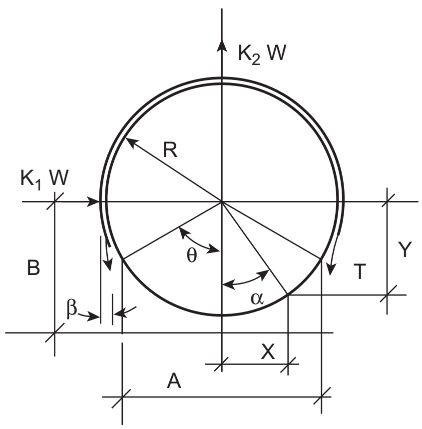


Figure 10-4. Dimensions of shipping saddle for alternate case.

Calculation

- Vertical distance to centroid of saddle, Y

  $Y = R \sin \theta / \theta$

- Find angle, α

  $\alpha = \cos^{-1}(Y/R)$

- Horizontal distance to centroid of saddle, X

  $X = R \sin \alpha$

- Tension load in band, T

  $T = [Q\,[(K_1\ Y/X) + K_2]]/[2N \cos \beta]$

Notes

1. Vertical reaction can be a result of longitudinal load. Use largest value
2. Use $K_2 = 0$ for transverse case
3. Use $K_1 = 0$ for longitudinal case
4. Use worst case of T_1 or T_2 and design the balance of components per previous method

Example

K_1 = .25
K_2 = .5
Q = 500 kips
R = 92.5 in
θ = 75° = 1.308 rad
β = 7.5°
N = 2

$Y = R \sin \theta/\theta$
$Y = 92.5 \sin 75 / 1.308 = 68.3$ in

$\alpha = \cos^{-1} (Y / R)$
$\alpha = \cos^{-1} (68.3 / 92.5) = 42.4^{o}$

$X = R \sin \alpha$
$X = 92.5 (\sin 42.4) = 62.4$ in

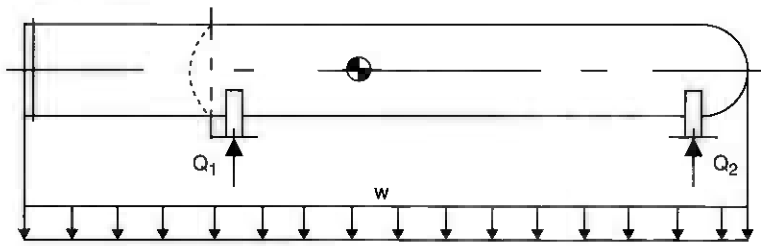
Transverse ($K_2 = 0$)

$T_1 = [Q\,[(K_1\ Y / X) + K_2]] / [2N \cos \beta]$
$T_1 = [500\,[(0.25 \cdot 68.3/62.4) + 0]] /[2 \cdot 2 \cos 7.5] = 34.35$ kips

Longitudinal ($K_1 = 0$)

$T_2 = [Q\,[(K_1\ Y / X) + K_2]] / [2N \cos \beta]$
$T_2 = [500\,[0 + 0.5]] / [2 \cdot 2 \cos 7.5] = 63.04$ kips

Check Vessel Shell Stresses



LOAD DIAGRAM

Notation

$Z = \pi R^2 t$
r = radius of vessel, in.
R = radius of vessel, ft.
b = width of saddle, in.
$d = b + 1.56\sqrt{rt}$

Stress Type	General	At Saddle 1	At Saddle 2
Longitudinal bending at saddles	— —	$S_1 = \frac{M}{K_1 r^2 t}$ $S_2 = -\left(\frac{M}{K_7 r^2 t}\right)$	$S_1 = \frac{M}{K_1 r^2 t}$ $S_2 = -\left(\frac{M}{K_7 r^2 t}\right)$
Longitudinal bending at midspan	$S_3 = \frac{M}{Z}$		
Tangential shear	—	$S_7 = \frac{K_3 Q_1}{rt}$	$S_7 = \frac{K_3 Q_2}{rt}$
Circumferential stress at horn of saddle $L_1 > 8R$ $L_1 < 8R$	— —	$S_9 = -\left(\frac{Q_1}{4td}\right) - \frac{3K_6 Q_1}{2t^2}$ $S_{10} = -\left(\frac{Q_1}{4td}\right) - \frac{12K_6 Q_1 R}{L_1 t^2}$	$S_9 = -\left(\frac{Q_2}{4td}\right) - \frac{3K_6 Q_2}{2t^2}$ $S_{10} = -\left(\frac{Q_2}{4td}\right) - \frac{3K_6 Q_2 R}{L_1 t^2}$
Circumferential compression	—	$S_{12} = -\left(\frac{K_5 Q_1}{td}\right)$	$S_{12} = -\left(\frac{K_5 Q_2}{td}\right)$

Notes:
1. Also check shell stresses at each change of thickness and diameter.
2. See procedure for the design of saddles for horizontal vessels for a detailed description of shell stresses and for values of coefficients K_1 through K_7.
3. Values of M and Q should be determined from the previous pages at the applicable location.
4. Allowable stresses:

Tension: $0.9F_y$
Compression: 1.2 x Factor "B" from ASME Code

Procedure 10-2: Erection of Pressure Vessels

The designer of pressure vessels and similar equipment will ultimately become involved in the movement, transportation, and erection of that equipment. The degree of that involvement will vary due to the separation of duties and responsibilities of the parties concerned. It is prudent, however, for the designer to plan for the eventuality of these events and to integrate these activities into the original design. If this planning is done properly, there is seldom a problem when the equipment gets to its final destination. Conversely, there have been numerous problems encountered when proper planning has not been done.

There is also an economic benefit in including the lifting attachments in the base vessel bid and design. These lifting attachments are relatively inexpensive in comparison to the overall cost of the vessel and minuscule compared to the cost of the erection of the equipment. The erection alone for a major vessel can run into millions of dollars. If these attachments are added after the purchase order is awarded, they can become expensive extras.

There are also the consequences to life, property, and schedules if this activity is not carried out to a successful conclusion. Compared to the fabricated cost of the lifting attachments, the consequences to life, property, and schedule are too important to leave the design of these components and their effect on the vessel to those not fully versed in the design and analysis of pressure vessels.

In addition, it is important that the designer of the lifting attachments be in contact with the construction organization that will be executing the lift. This ensures that all lifting attachments meet the requirements imposed by the lifting equipment. There are so many different methods and techniques for the erection of vessels and the related costs of each that a coordinated effort between the designer and erector is mandatory. To avoid surprises, neither the designer nor the erector can afford to work in a vacuum. To this end, it is not advisable for the vessel fabricator to be responsible for the design if the fabricator is not the chief coordinator of the transport and erection of the vessel.

Vessels and related equipment can be erected in a variety of ways. Vessels are erected by means of single cranes, multiple cranes, gin poles, jacking towers, and other means. The designer of the lifting attachments should not attempt to dictate the erection method by the types of attachments that are designed for the vessel. The selection of one type of attachment versus another could very well do just that.

Not every vessel needs to be designed for erection or have lifting attachments. Obviously the larger the vessel, the more complex the vessel, the more expensive the vessel, the more care and concern that should be taken into account when designing the attachments and coordinating the lift. The following listing will provide some guidelines for the provision of special lifting attachments and a lifting analysis to be done. In general, provide lifting attachments for the following cases:

- Vessels over 50,000 lb (25 tons).
- Vessels with L/D ratios greater than 5.
- Vertical vessels greater than 8 ft in diameter or 50 ft in length.
- Vessels located in a structure or supported by a structure.
- High-alloy or heat-treated vessels (since it would not be advisable for the field to be doing welding on these vessels after they arrive on site, and wire rope slings could contaminate the vessel material).
- Flare stacks.
- Vessels with special transportation requirements.

At the initial pick point, when the vessel is still horizontal, the load is shared between the lifting lugs and the tail beam or lug, based on their respective distances to the vessel center of gravity. As the lift proceeds, a greater percentage of the load is shifted to the top lugs or trunnions until the vessel is vertical and all of the load is then on the top lugs. At this point the tail beam or shackle can be removed.

During each degree of rotation, the load on the lugs, trunnions, tailing device, base ring, and vessel shell are continually varying. The loads on the welds attaching these devices will also change. The designer should evaluate these loadings at the various lift angles to determine the worst coincident case.

The worst case is dependent on the type of vessel and the type of attachments. For example, there are three types of trunnions described in this procedure. There is the bare trunnion (Type 3), where the wire rope slides around the trunnion itself. While the vessel is in the horizontal position (initial pick point), the load produces a circumferential moment on the shell. Once the vessel is in the upright position, the same load produces a longitudinal moment in the shell. At all the intermediate angles of lift there is a combination of circumferential and longitudinal moments. The designer should check the two worst cases at 0° and 90° and several combinations in between.

The same trunnion could have a lifting lug welded to the end of the trunnion (Type 1). This lug also produces circumferential and longitudinal moments in the shell. However, in addition this type of lug will produce a torsional moment on the shell that is maximum at 0° and zero at 90° of angular rotation. The rotating lug (Type 2) eliminates any torsional moment.

There is one single lift angle that will produce the maximum stress in the vessel shell but no lift angle that is the worst for all vessels. The worst case is dependent on the type of lift attachments, distances, weights, and position relative to the center of gravity.

The minimum lift location is the lowest pick point that does not overstress the overhanging portion of the vessel. The maximum lift location is the highest pick point that does not overstress the vessel between the tail and pick points. These points become significant when locating the lift points to balance the stress at the top lug, the overhang, and the midspan stress.

The use of side lugs can sometimes provide an advantage by reducing the buckling stress at midspan and the required lift height. Side lugs allow for shorter boom lengths on a two-crane lift or gin poles. A shorter boom length, in turn, allows a higher lift capacity for the cranes. The lower the lug location on the shell, the shorter the lift and the higher the allowable crane capacity. This can translate into dollars as crane capacity is affected. The challenge from the vessel side is the longitudinal bending due to the overhang and increased local shell stresses. All of these factors must be balanced to determine the lowest overall cost of an erected vessel.

Requirement for Erection and Setting of Vertical Vessels

The following is a brief synopsis of general recommendations regarding the setting, leveling and shimming of vertical vessels. The following should be considered as guidelines only. There are no codes or standards that are applied. In general, company specifications contain contract requirements for the contractors scope of supply or duties. The following lists help to clarify general construction practices with regard to the setting of vertical vessels and towers.

Contractor Duties

1. Prepare tops of foundations (bush hammer if required)
2. Perform surveying as required to establish centerlines, sole plate or shim elevations at bottom of base of equipment
3. Shimming
4. Erect equipment
5. Level/plumbing
6. Final alignment
7. Grouting
8. Bolting

Tolerances

Out of vertical tolerance for vertical vessels, unless specified otherwise, shall be 0.1% of the vessel height, or about ¼ inch for every 20 feet to a maximum of ¾ inches.

Soleplates (also called bearing pads, leveling plates or embedments)

Soleplates are stainless steel plates, 0.5 inches to 0.75 inches thick, set in grout, on top of the foundation at the exact height of the underside of the base plate. As a rule, two soleplates should be installed per anchor bolt, one on each side of the bolt. Depending on the tower diameter, and the distance between the anchor bolts, another soleplate may be installed between adjacent anchor bolts. The dimensions of the soleplates will vary according to the width of the vessel base ring and vessel weight. Soleplates are supported in place by a mixture of Portland cement and sand in proportions 1:3. The vessel should not be erected until the soleplates have been in place for 28 days to allow for concrete curing. Shims and soleplates will remain in place after the grouting operation.

Shims

Shims are used to provide precise leveling of the vessel. Shim packs may be grouted into the foundation in lieu of sole plates but this practice is unusual. Typically, shims are used on top of the sole plates for the leveling operation. Special shims may be required for unique applications such as a large vessel supported on a braced frame structure with minimal contact/bearing at each support point. The following are some guidelines for the use of shims.

1. Shims. If left in place, shall be stainless steel
2. Shims must have rounded corners
3. Shims will be fixed in place
4. Shims shall be deburred
5. Shims shall be full bearing
6. Shims may be horseshoe type
7. Shims thinner than 0.001 inches are not allowed
8. Shims with holes are not allowed
9. Shims should be the full width of the base plate

Leveling/Straightness/Plumbness

After the vessel has been placed on its foundation it must be checked to be certain it is vertical and plumb. Leveling is normally checked by use of two theodilites, 90 degrees apart. The theodilites shall be spaced an adequate distance from the vessel to allow visual field of the entire height of the vessel. Adjustments can be made to the vessel alignment by means of wedges, either powered or not, and then shimmed. The wedges should not be left in place after shimming.

The vessel may be heated by the sun to a higher temperature on one side than the other. This can create a slight “banana” effect which should be taken into account when checking levelness. The equation for calculating the deflection from this effect is as follows:

$$\varsigma = \left[\pi\ D^2\ t\ H^2\ \alpha \Delta T\right] / 8\ I$$

where;

ς = Deflection, in
D = Diameter, ft
T = Thickness, in
H = Height, ft
α = Coefficient of thermal expansion, in/in/°F
ΔT = Temperature difference from one side of the column to the other, °F
I = Moment of inertia of vessel cross section, ft^4

Bolting

After the vessel is aligned and shimmed, the nuts on the anchor bolts must be tightened. The vessel should not be left standing without the crane attached unless all anchor bolts have been tightened.

The anchor bolts should not be tightened to their maximum load until the drypacking under the base plate is complete. At this stage, the base plate is suspended between the soleplates until the drypack is installed. Since the soleplates straddle the anchor bolts, there is a chance of deforming the baseplate prior to the installation of the drypack, if the anchor bolts are over tightened.

After drypacking, the anchor bolts should be tightened to the correct torque to produce the maximum allowable bolt stress. The anchor bolts should not be tightened beyond the point of maximum allowable bolt stress.

Note that the initial anchor bolt tension does not increase the maximum bolt tension caused by wind or earthquake. This initial tension will only clamp the base ring to the concrete. Both are in equal compression until the external load is applied. The external load reduces the compression in the concrete before additional load is applied to the bolts. After the external load overcomes all the compression in the concrete, the stress in the bolt will increase to the value it would have been, had there been no initial tension.

Grouting

Grout under base plates shall provide full uniform load transfer between the bottom of the base plate and the top of the foundation. Load transfer to the foundation must be through the grout, not through the shims or soleplate.

Prior to setting of the vessel, the top of the foundation should be bush hammered and cleaned. This ensures that the grout will adhere to the surface of the foundation. Bush hammering may be done strictly under the base plate or across the entire top of the foundation.

Once the vessel is leveled, shimmed and bolted it is ready to be grouted. Grouting shall consist of filling the void area between the top of foundation and the underside of base plate with cementatious grout. The grout shall be installed in accordance with the manufacturers recommendations and any applicable contract specifications.

Depending on the type of grout to be used, grout dams may be used.

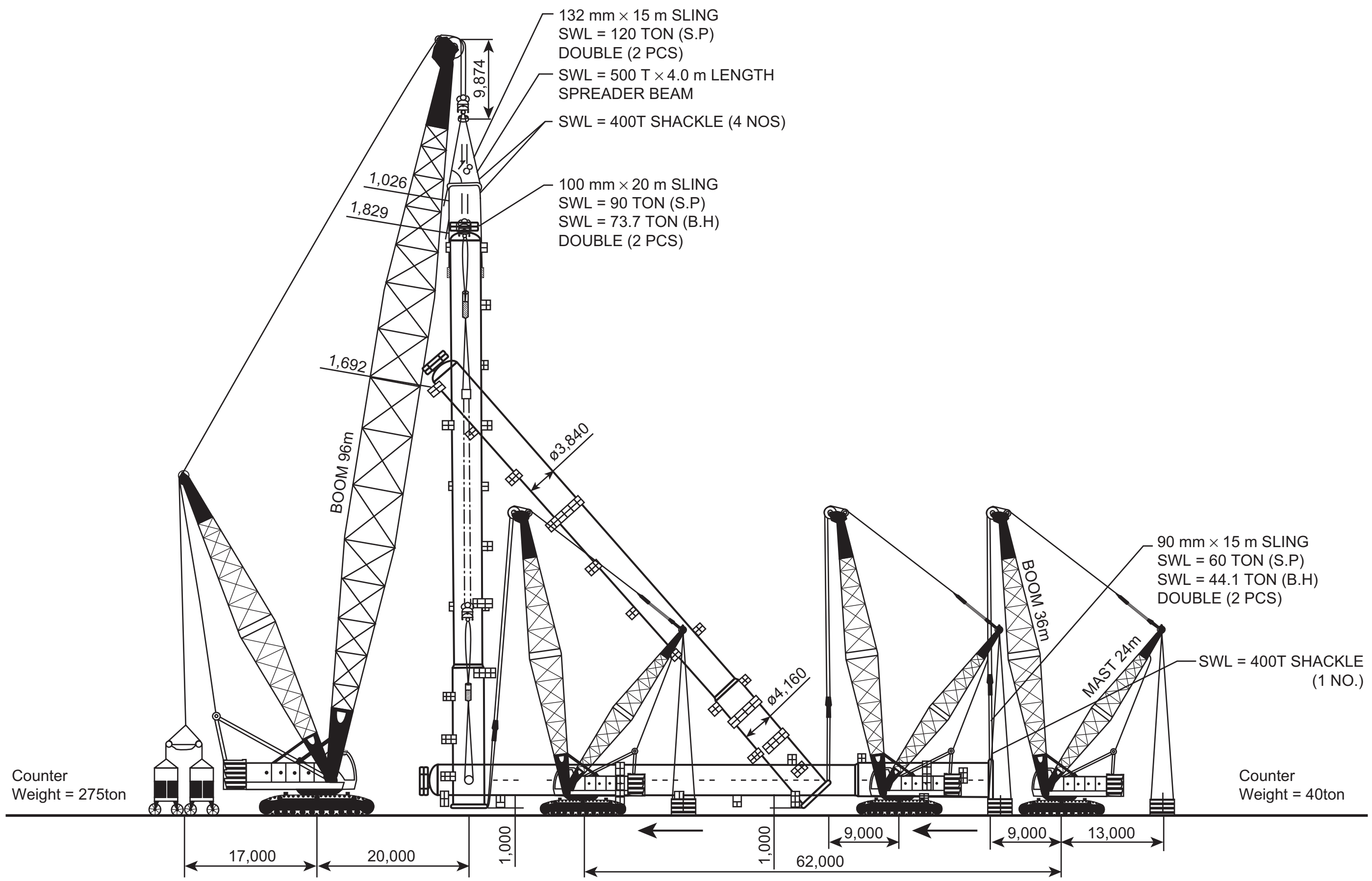


Figure 10-5. Typical example of erection study.

Steps in Design

Given the overall weight and geometry of the vessel and the location of the center of gravity based on the erected weight, apply the following steps to either complete the design or analyze the design.

Step 1: Select the type of lifting attachments as an initial starting point:

Lift end (also referred to as the "pick end"):

a. *Head lug:* Usually the simplest and most economical, and produces the least stress.
b. *Cone lug:* Similar to a head lug but located at a conical transition section of the vessel.
c. *Side lug:* Complex and expensive.
d. *Top flange lug:* The choice for high-pressure vessels where the top center flange and head are very rigid. This method is uneconomical for average applications.
e. *Side flange lug:* Rarely used because it requires a very heavy nozzle and shell reinforcement.
f. *Trunnions:* Simple and economical. Used on a wide variety of vessels.
g. Other.

Tail end:

a. Tail beam.
b. Tail lug.
c. Choker (cinch); see later commentary.

Tailing a column during erection with a wire rope choker on the skirt above the base ring is a fairly common procedure. Most experienced erectors are qualified to perform this procedure safely. There are several advantages to using a tailing choker:

- Saves material, design, detailing, and fabrication.
- Simplifies concerns with lug and shipping orientations.
- May reduce overall height during transportation.

There are situations and conditions that could make the use of a tailing choker impractical, costly, and possibly unsafe. Provide tailing lugs or a tailing beam if:

- The column is more than about 10 feet in diameter. The larger the diameter, the more difficult it is for the wire rope to cinch down and form a good choke on the column.
- The tail load is so great that it requires the use of slings greater than about 1½ inches in diameter. The larger the diameter of the rope, the less flexible it is and the more likely that it could slip up unexpectedly during erection.

Step 2: Determine the forces T and P for all angles of erection.

Step 3: Design/check the lifting attachments for the tailing force, T, and pick force, P.

Step 4: Design/check the base ring assembly for stresses due to tailing force, T.

Step 5: Determine the base ring stiffening configuration, if required, and design struts.

Step 6: Check shell stresses due to bending during lift. This would include midspan as well as any overhang.

Step 7: Analyze local loads in vessel shell and skirt due to loads from attachments.

Allowable Stresses

Per AISC:

Tension

$F_t = 0.6F_y$ on gross area

$= 0.5F_u$ on effective net area

$= 0.45F_y$ for pin-connected members

Compression
(for short members only)

$F_c =$ Use buckling value.

$=$ for vessel shell: $1.33 \times$ ASME Factor "B"

Shear

$F_s =$ Net area of pin hole: $0.45F_y$

$=$ other than pin-connected members: $0.4F_y$

$=$ fillet welds in shear:

E60XX: 9600 lb/in. or 13,600 psi

E70XX: 11,200 lb/in. or 15,800 psi

Bending

$F_b = 0.6F_y$ to $0.75F_y$, depending on the shape of the member

Bearing

$F_p = 0.9F_y$

Combined

Shear and tension:

$$\frac{\sigma_a}{F_a} + \frac{\tau}{F_s} \leq 1$$

Tension, compression and bending:

$$\frac{\sigma_a}{F_a} + \frac{\sigma_b}{F_b} \leq 1 \text{ or } \frac{\sigma_T}{F_T} + \frac{\sigma_b}{F_b} \leq 1$$

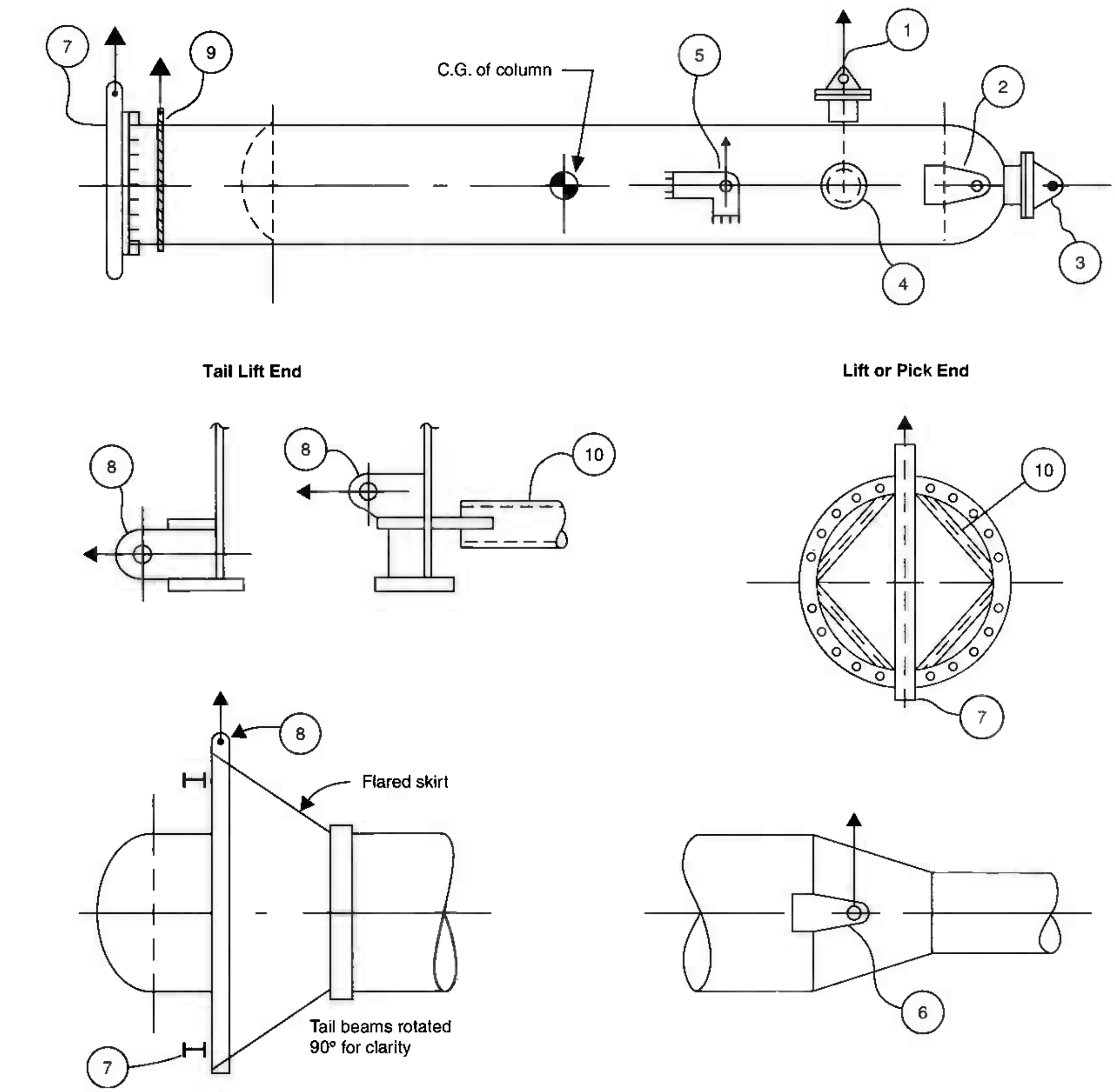
Note: Custom-designed lifting devices that support lifted loads are generally governed by ASME B30.20 "Below the hook lifting devices." Under this specification, design stresses are limited to $F_y/3$. The use of AISC allowables with a load factor of 1.8 or greater will generally meet this requirement.

Notation

A = area, in.^{2}
A_a = area, available, in.^{2}
A_b = area, bolt, in.^{2}
A_n = net cross-sectional area of lug, in.^{2}
A_p = area, pin hole, in.^{2}
A_r = area, required, in.^{2}
A_s = area, strut, in.^{2} or shear area of bolts
C = lug dimension, see sketch
D_o = diameter, vessel OD, in.
D_1 = diameter, lift hole, in.
D_2 = diameter, pin, in.
D_3 = diameter, pad eye, in.
D_{sk} = diameter, skirt, in.
D_m = mean vessel diameter, in.
E = modulus of elasticity, psi
f_r = tail end radial force, lb
f_L = tail end longitudinal force, lb
f_s = shear load, lb or lb/in.
F_a = allowable stress, combined loading, psi
F_b = allowable stress, bending, psi
F_c = allowable stress, compression, psi
F_p = allowable stress, bearing pressure, psi
F_s = allowable stress, shear, psi
F_t = allowable stress, tension, psi
F_y = minimum specified yield stress, psi
I = moment of inertia, in.^{4}
J_w = polar moment of inertia of weld, in.^{4}
K = end connection coefficient
K_L = overall load factor combining impact and safety factors, 1.5–2.0
K_i = impact factor, 0.25–0.5
K_r = internal moment coefficient in circular ring due to radial load
K_s = safety factor
K_T = internal tension/compression coefficient in circular ring due to radial load
L_s = length of skirt/base stiffener/strut, in.
M = moment, in.-lb
M_b = bending moment, in.-lb
M_C = circumferential moment, in.-lb
M_L = longitudinal moment, in.-lb
M_T = torsional moment, in.-lb
N_b = number of bolts used in tail beam or flange lug
N = width of flange of tail beam with a web stiffener ($N = 1.0$ without web stiffener)
n_L = number of head or side lugs
P = pick end load, lb
P_e = equivalent load, lb
P_L = longitudinal load per lug, lb
P_r = radial load, lb
P_T = transverse load per lug, lb
R_b = radius of base ring to neutral axis, in.
r = radius of gyration of strut, in.
R_c = radius of bolt circle of flange, in.
S_u = minimum specified tensile stress of bolts, psi
t_b = thickness of base plate, in.
t_g = thickness of gusset, in.
t_L = thickness of lug, in.
t_P = thickness of pad eye, in.
t_s = thickness of shell, in.
T = tail end load, lb
T_b = bolt pretension load, lbs
T_t = tangential force, lb
w_1 = fillet weld size, shell to re-pad
w_2 = fillet weld size, re-pad to shell
w_3 = fillet weld size, pad eye to lug
w_4 = fillet weld size, base plate to skirt
w_5 = uniform load on vessel, lb/in.
W_E = design erection weight, lb
W_L = erection weight, lb
Z = section modulus, in.^{3}
α = angular position for moment coefficients in base ring, clockwise from 0°
β = angle between parallel beams, degrees
σ = stress, combined, psi
σ_b = stress, bending, psi
σ_p = stress, bearing, psi
σ_c = stress, compression, psi
σ_{cr} = critical buckling stress, psi
σ_T = stress, tension, psi
τ = shear stress, psi
τ_T = torsional shear stress, psi
θ = lift angle, degrees
θ_B = minimum bearing contact angle, degrees
θ_H = sling angle to lift line, horizontal, degrees
θ_v = sling angle to lift line, vertical, degrees

Procedure 10-3: Lifting Attachments and Terminology

Types of Lifting Attachments



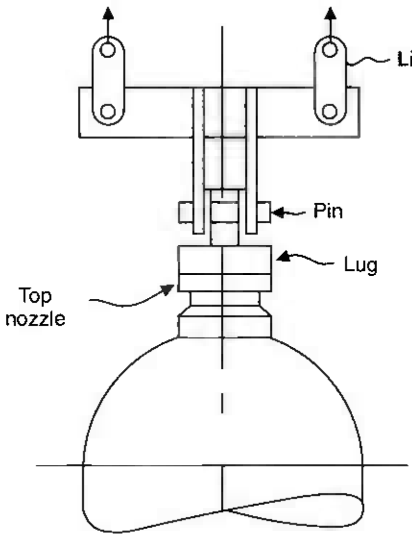
LIFT END OPTIONS

1. Shell flange lug
2. Top head lug
3. Top flange lug
4. Trunnion
5. Side lug
6. Cone lug

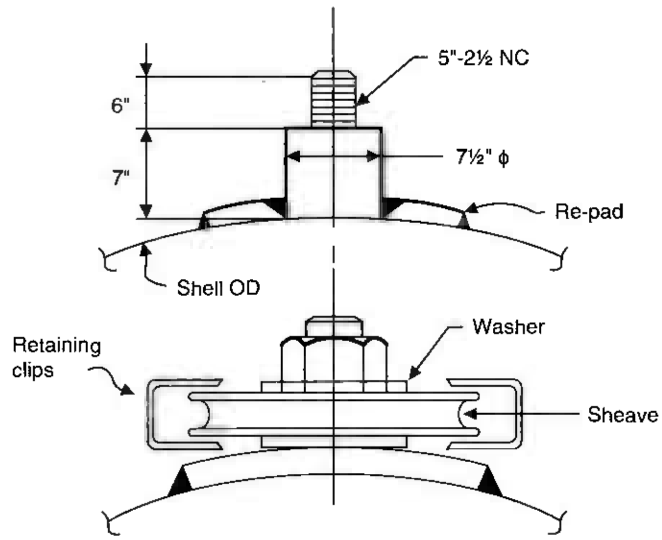
TAIL LIFT OPTIONS

7. Tail beam
8. Tail lug
9. Choker (sling)
10. Base ring stiffener

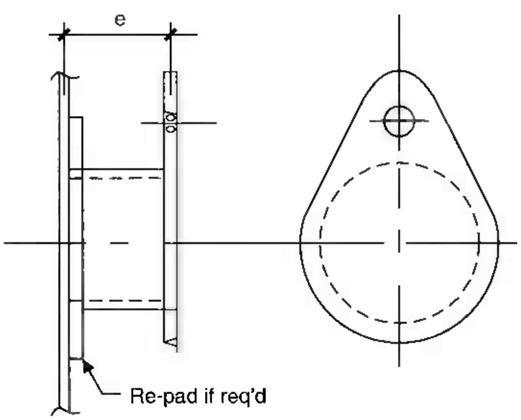
Miscellaneous Lifting Attachments



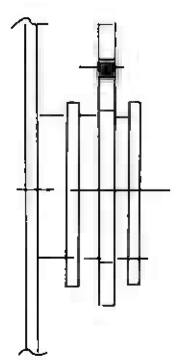
TOP FLANGE LUG WITH SPREADER



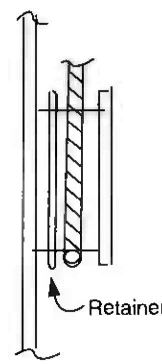
SHEAVE ASSEMBLY DIRECT MOUNT—200 TON



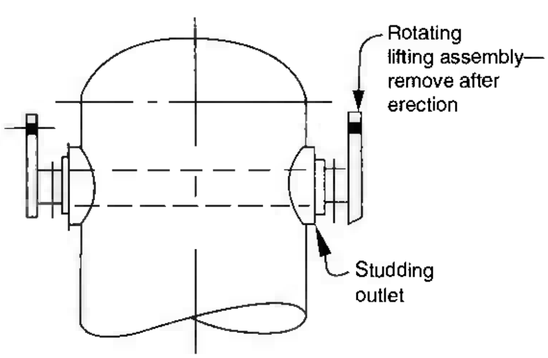
TRUNNION WITH FIXED LUG TYPE 1



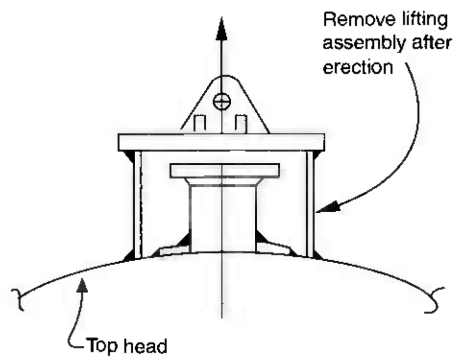
TRUNNION WITH ROTATING LUG TYPE 2



TRUNNION WITHOUT LUG TYPE 3



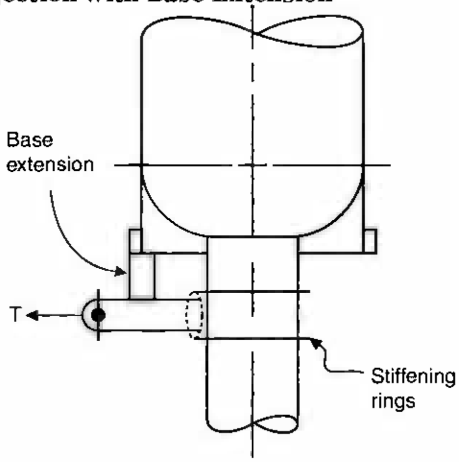
STIFFENER PIPE THROUGH STUDDING OUTLETS—BLIND AFTER ERECTION



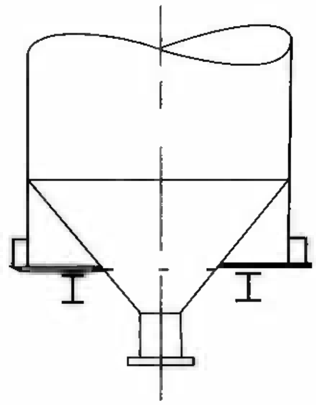
TEMPORARY TOP HEAD FLANGE LUG OVER TOP NOZZLE

Tailing Devices for Vessels with Chambers Projecting through Skirt

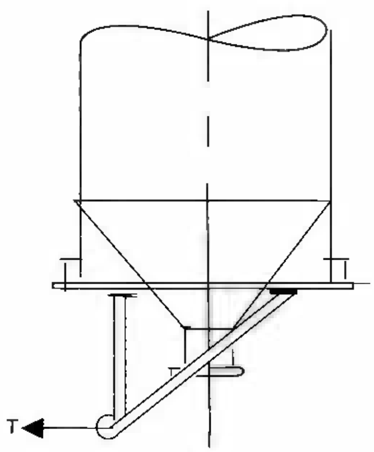
Utilize Projection with Base Extension



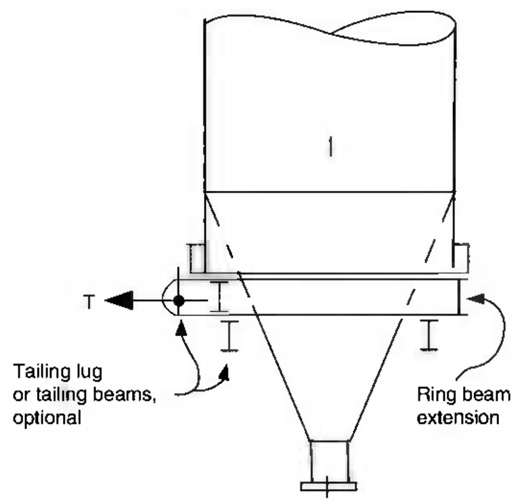
Parallel Tailing Beams Without Skirt Stiffeners



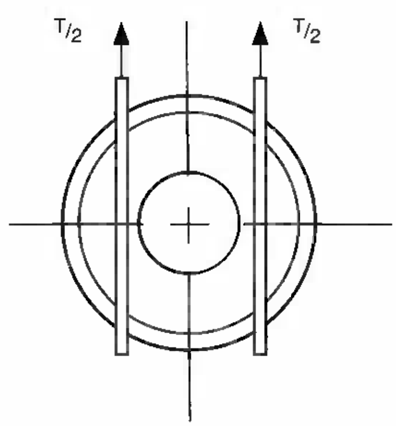
Frame-Type Tailing Device



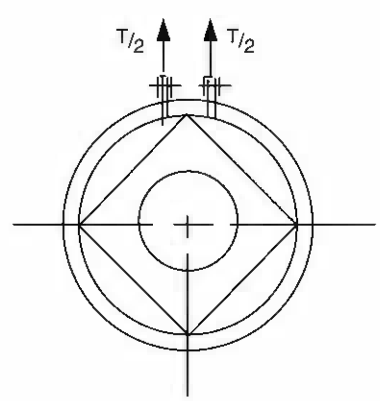
Bolted on Ring Beam Extension

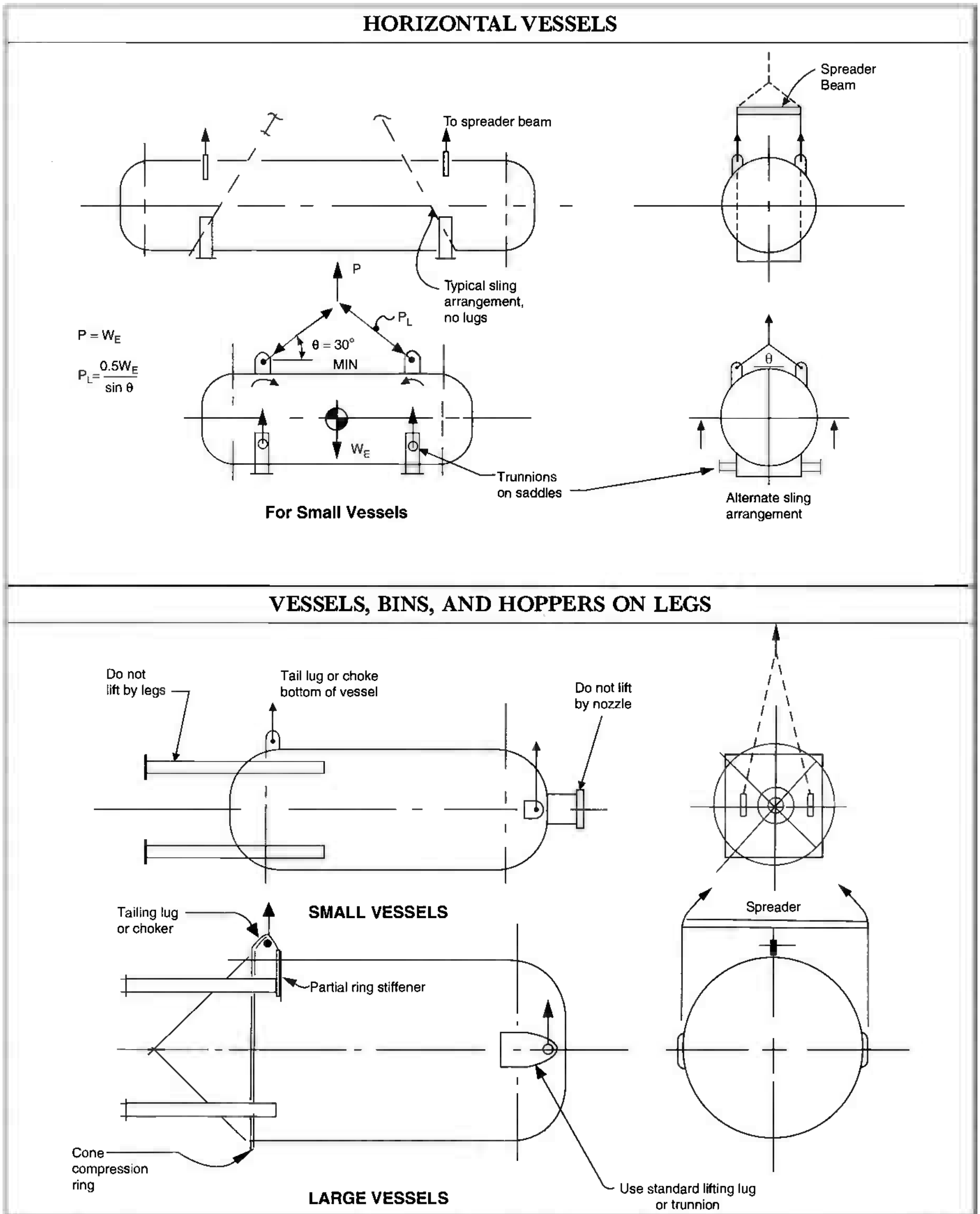


Base with Parallel Tailing Beams

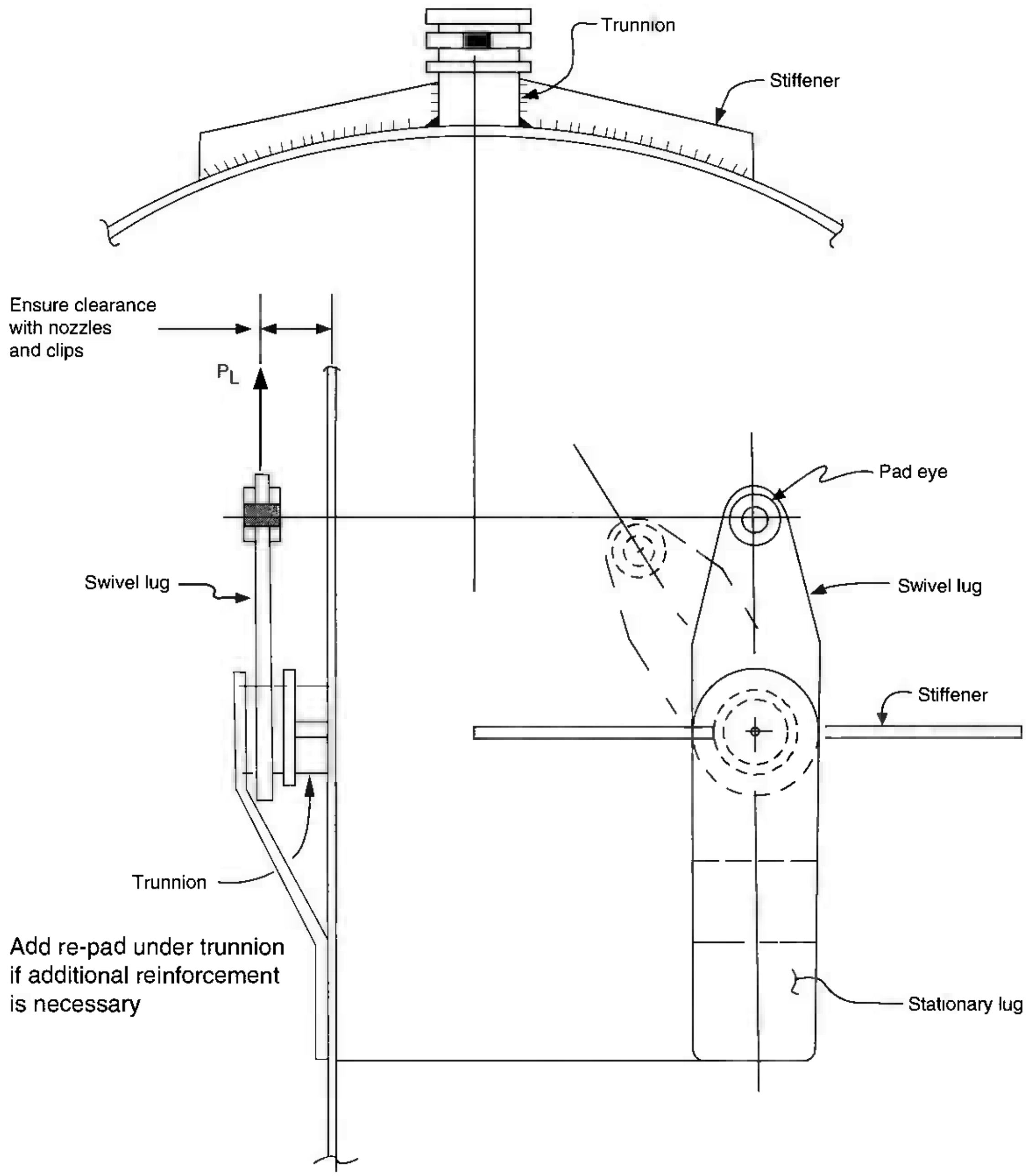


Base with Internal Base Stiffeners and Dual Tailing Lugs

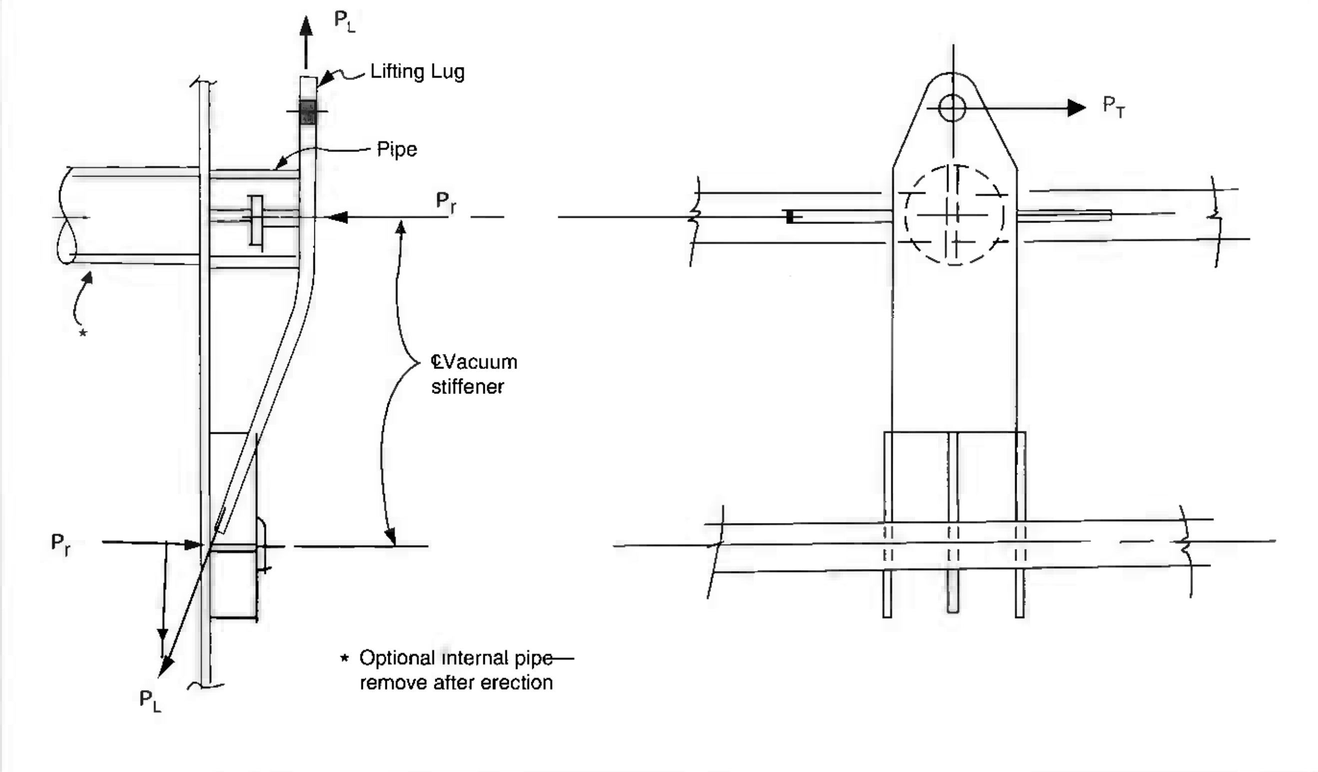



HORIZONTAL VESSELS
Spreader Beam
To spreader beam
Typical sling arrangement, no lugs
P
P_L
$P = W_E$
$P_L = \frac{0.5W_E}{\sin\theta}$
$\theta = 30°$ MIN
θ
W_E
Trunnions on saddles
Alternate sling arrangement
For Small Vessels
VESSELS, BINS, AND HOPPERS ON LEGS
Do not lift by legs
Tail lug or choke bottom of vessel
Do not lift by nozzle
SMALL VESSELS
Tailing lug or choker
Partial ring stiffener
Spreader
Cone compression ring
LARGE VESSELS
Use standard lifting lug or trunnion

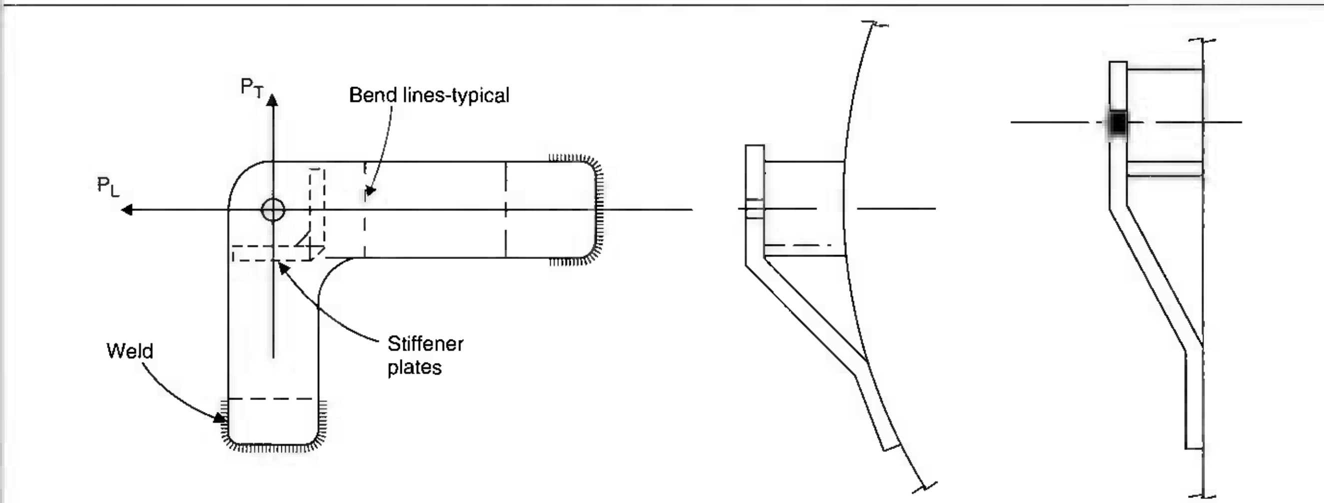
SIDE LUG WITH SWIVEL LUG



SIDE LUG COMBINATION WITH STIFFENERS

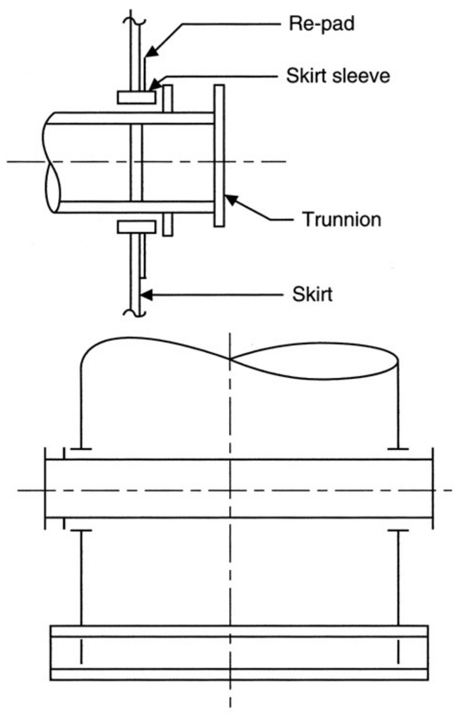


TYPICAL SIDE LUG

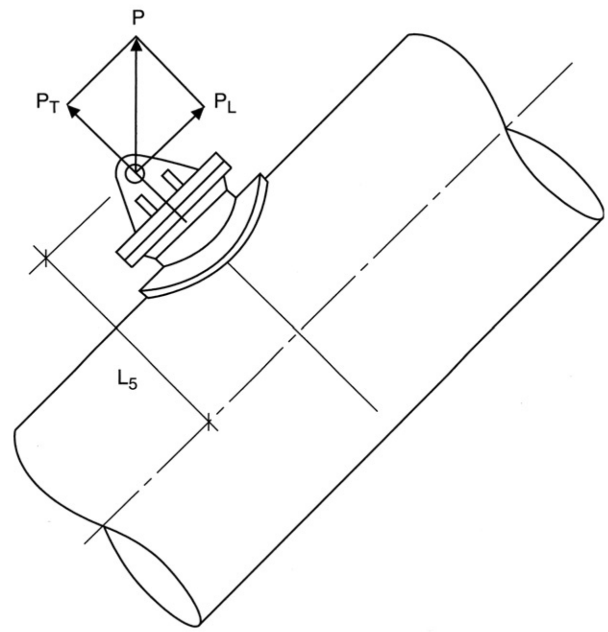


Tailing Trunnion

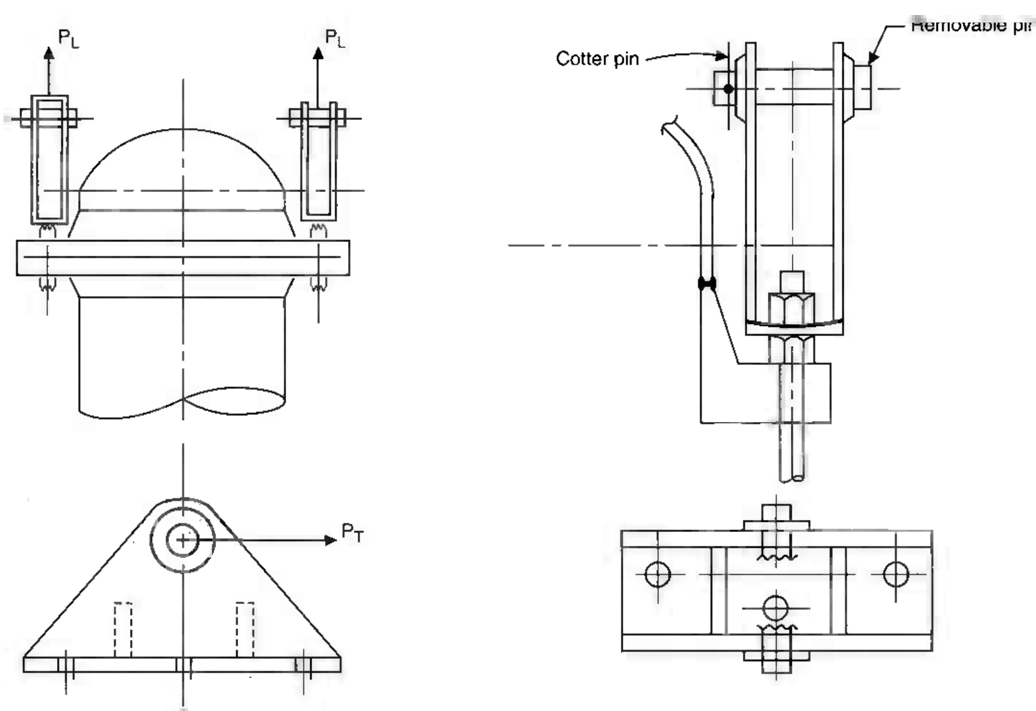
Utilizes reinforced openings in skirt with through pipe. Pipe is removed after erection and the openings used as skirt manways.



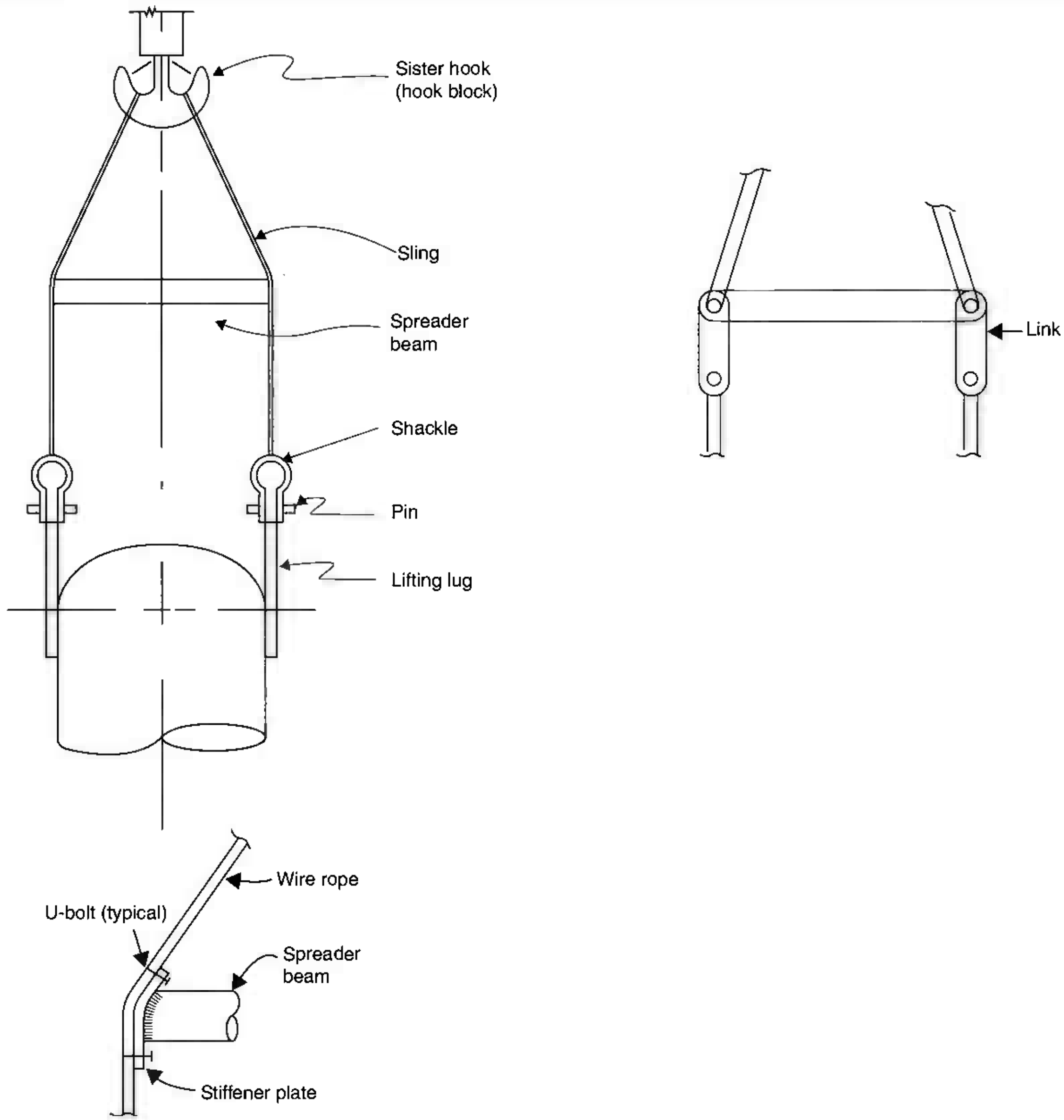
Shell Flange Lug



Lifting Device Utilizing Top Body Flanges



Rigging Terminology



1. Boom
2. Mast
3. Gin pole
4. Crane mats
5. Dead men
6. Outriggers
7. Load block
8. Whip line
9. Cranes
10. Derricks
11. Hoist
12. Hooks
13. Jacks
14. Slings
15. Pins
16. Spreader beams
17. Equalizer beams
18. Links
19. Shackles
20. Wire rope
21. Counterweight
22. Trailing counterweight
23. Struts
24. Lashings
25. Guy streamers
26. Bail
27. Tensioning blocks
28. Hitch plate
29. Pin extractor
30. Choker
31. Tail crane
32. Tail sled

Miscellaneous Lugs, $W_L < 60$ kips

Table 10-6
Lug dimensions

W_L kips	A	D_1	B	C	t_L	w_1	W_L kips	A	D_1	B	C	t_L	w_1
4	3	0.88	1.5	2	0.5	0.25	20	7	1.75	3	3	1	0.38
6	3.5	1	1.63	2	0.63	0.25	25	7	2.38	4	4	1	0.44
8	4	1.13	1.75	2	0.63	0.25	35	8	2.38	4	4	1.125	0.5
10	4.5	1.25	2	2	0.75	0.25	40	8	2.38	4	4	1.125	0.63
12	5	1.38	2.13	3	0.88	0.25	45	8	2.88	4	4	1.125	0.63
14	5.5	1.5	2.38	3	1	0.38	50	10	2.88	4	4	1.25	0.75
16	6.5	1.63	2.5	3	1	0.38	55	10	2.88	4	4	1.25	0.75
18	7	1.75	2.75	3	1	0.38	60	10	2.88	4	4	1.25	0.88

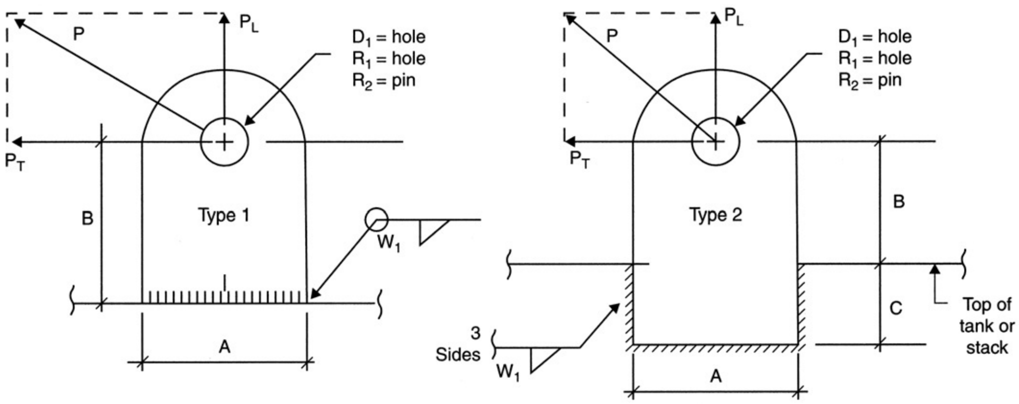


Figure 10-6. Dimensions and forces.

Calculations

Due to bending:

$$t_L = \frac{6P_TB}{A^2F_b}$$

Due to shear:

$$t_L = \frac{P_T}{(A - D_1)F_s}$$

Due to tension:

$$t_L = \frac{P_L}{(A - D_1)F_t}$$

Notes

1. Table 10-6 is based on an allowable stress of 13.7 ksi.
2. Design each lug for a 2:1 safety factor.
3. Design each lug for a minimum 10% side force.

Hertzian Stress, Bearing

$$\sigma_p = 0.418\sqrt{\frac{\left[E\left(\frac{P}{t_L}\right)(R_1 - R_2)\right]}{R_1R_2}} < 2F_y$$

Shear Load in Weld

Type 1: greater of following:

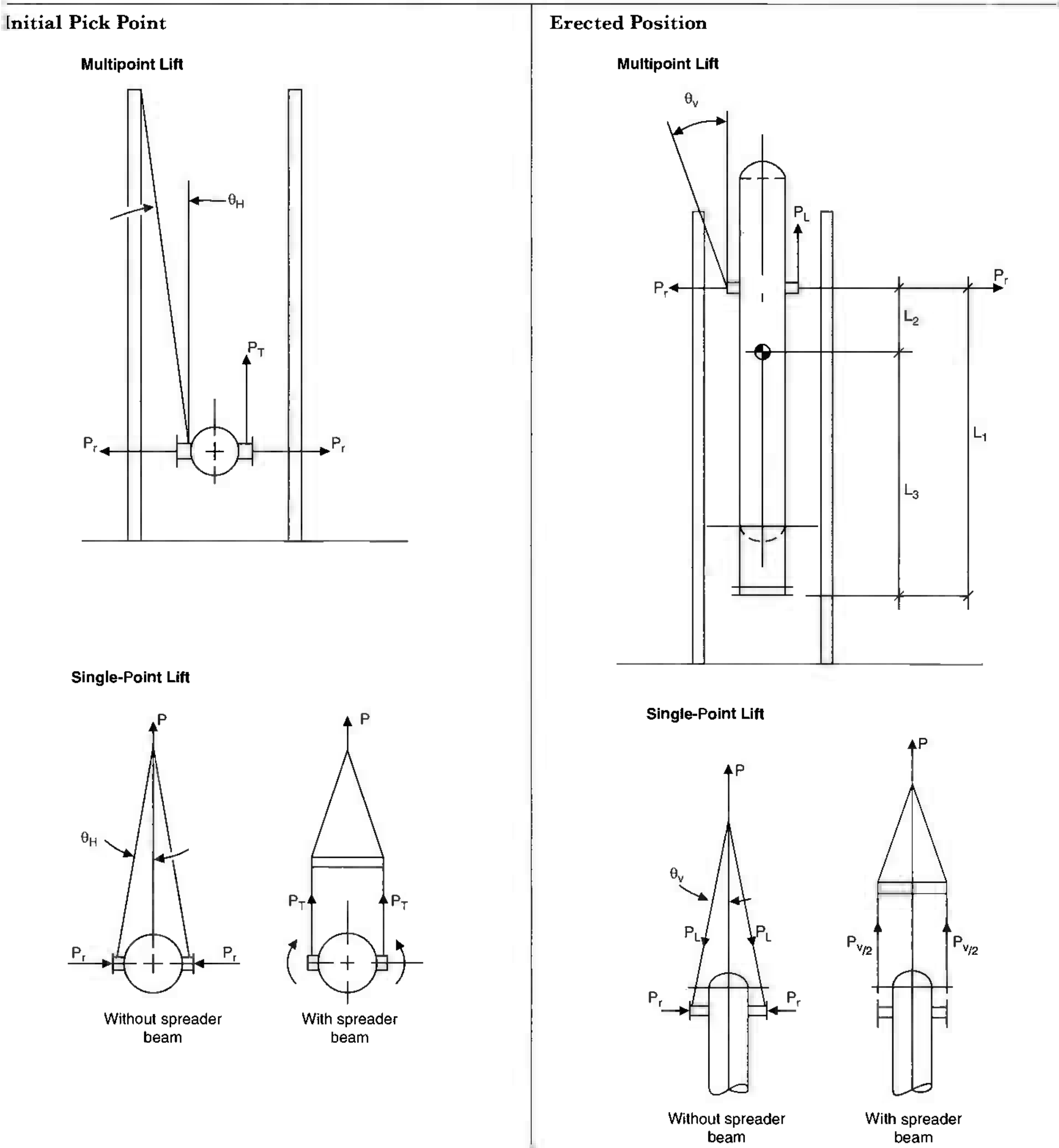
$$\tau_w = \frac{6P_TB}{2A^2}$$

$$\tau_w = \frac{P_L}{2A}$$

Type 2: Use design for top head lug.

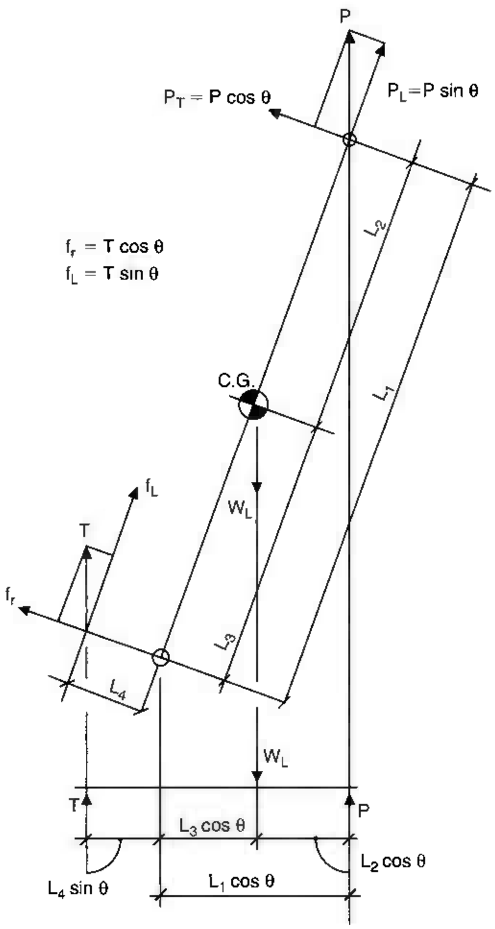
Procedure 10-4: Lifting Loads and Forces

Effect of Lift Line Orientation to Lug

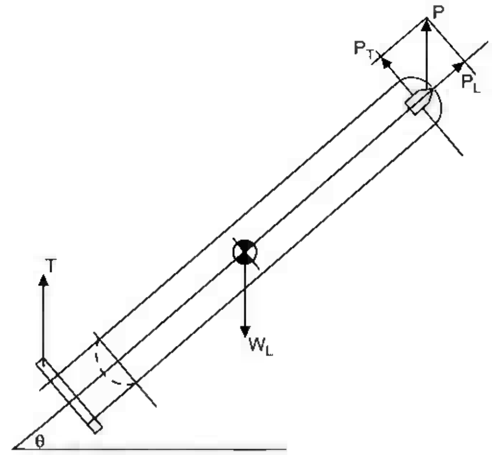


Force and Loading Diagrams

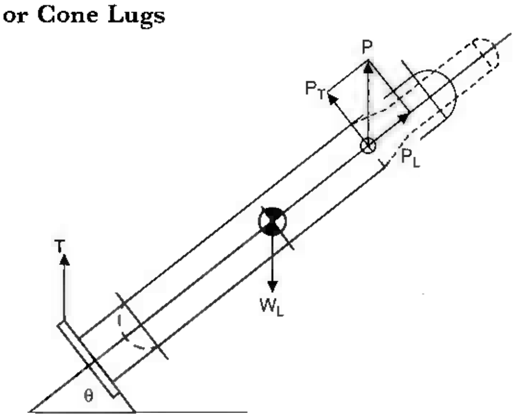
Free-Body Diagram



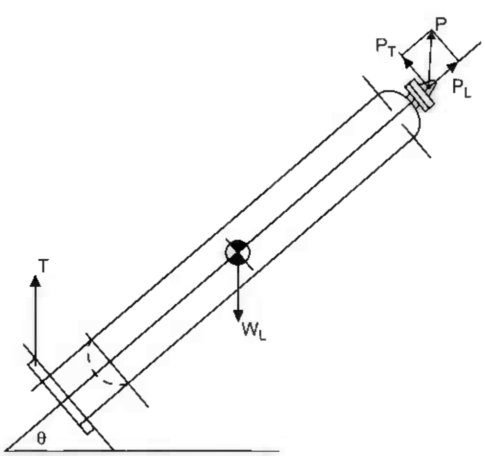
Top Head Lug



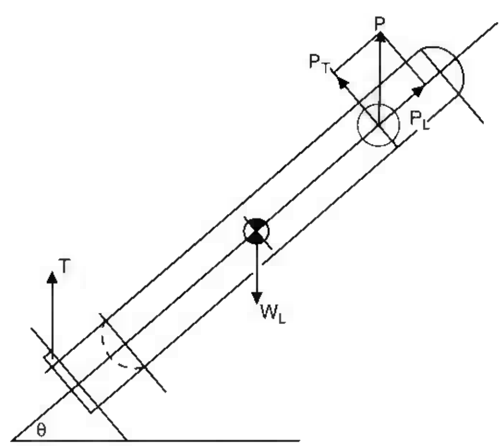
Side or Cone Lugs



Top Flange Lug



Trunnions



Loads

- *Overall load factor, K_L.*
  $K_L = K_i + K_s$
- *Design lift weight, W_L.*
  $W_L = K_L W_E$
- *Tailing load, T.*

$$T = \frac{W_L \cos\theta \ L_2}{\cos\theta \ L_1 + \sin\theta \ L_4}$$

At $\theta = 0$, initial pick point, vessel horizontal:

$$T = \frac{W_L L_2}{L_1} \quad \text{and} \quad P = \frac{W_L L_3}{L_1} \quad \text{or} \quad P = W_L - T$$

At $\theta = 90°$, vessel vertical:

$T = 0$ and $P = W_L$

- *Calculate the loads for various lift angles, θ.*

Loads T and P		
θ	T	P
0		
10		
20		
30		
40		
50		
60		
70		
80		
90		

Lift angles shown are suggested only to help find the worst case for loads T and P.

- *Maximum transverse load per lug, P_T.*

$$P_T = \frac{P\cos\theta}{n_L}$$

- *Maximum longitudinal load per lug, P_L.*

$$P_L = \frac{P\sin\theta}{n_L}$$

- *Radial loads in shell due to sling angles, θ_v or θ_H.*

$P_r = P_T \tan\theta_H$ Vessel in horizontal

$P_r = P_L \tan\theta_v$ Vessel in vertical

- *Tailing loads, f_L and f_r.*

$f_L = T\cos\theta$

$f_r = T\sin\theta$

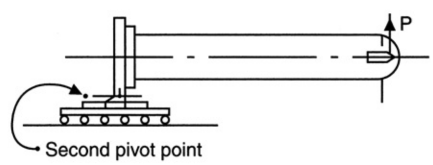
- *Longitudinal bending stress in vessel shell, σ_b.*

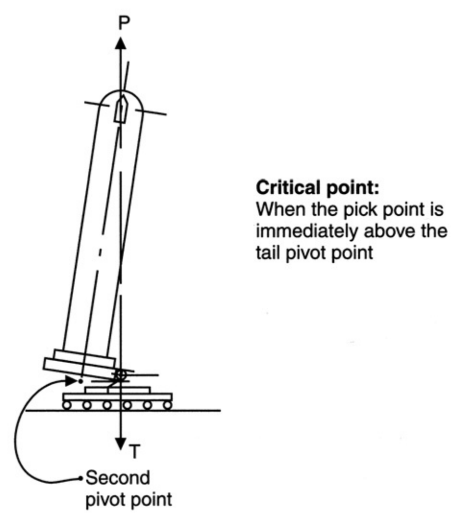
$$\sigma_b = \frac{4M}{\pi D_m^2 t}$$

Maximum moment occurs at initial pick, when $\theta = 0$. See cases 1 through 4 for maximum moment, M.

Note

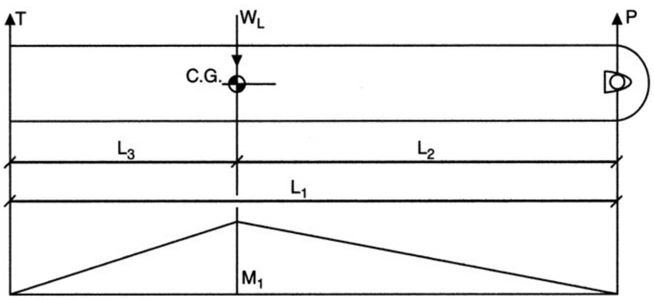
If the tailing point is below the CG as is the case when a tailing frame or sled is used, the tail support could see the entire weight of the vessel as erection approaches 90°.





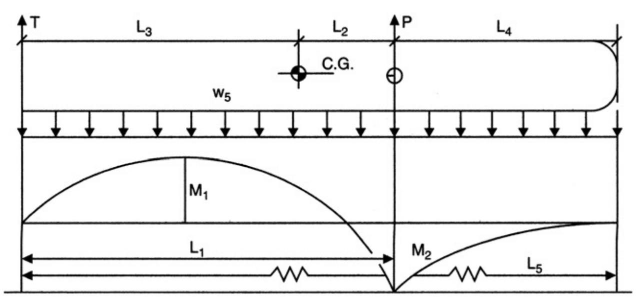
Dimensions and Moments for Various Vessel Configurations

Case 1: Top Head Lug, Top Head Trunnion, or Top Head Flange



$$M_1 = \frac{W_L L_3 L_2}{L_1}$$

Case 2: Side Lug or Side Trunnion

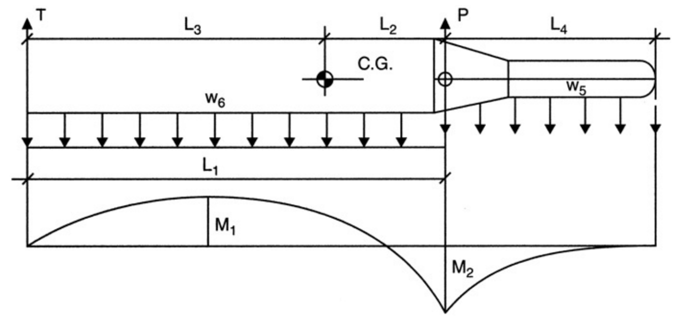


$$w_5 = \frac{W_L}{L_5}$$

$$M_1 = \frac{w_5}{8L_1^2}(L_1 + L_4)^2(L_1 - L_4)^2$$

$$M_2 = \frac{w_5 L_4^2}{2}$$

Case 3: Cone Lug or Trunnion

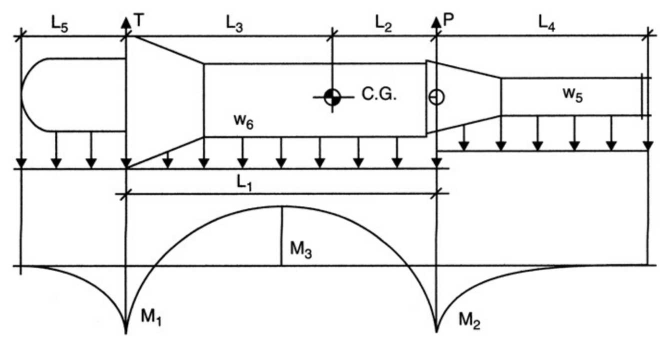


$$w_5 = \frac{W_{L_1}}{L_4} \quad w_6 = \frac{W_{L_2}}{L_1}$$

$$M_1 = \frac{w_6}{8L_1^2}(L_1 + L_4)^2(L_1 - L_4)^2$$

$$M_2 = \frac{w_5 L_4^2}{2}$$

Case 4: Cone Lug or Trunnion with Intermediate Skirt Tail



$$w_5 = \frac{W_{L_1}}{L_4} \quad w_6 = \frac{W_{L_2}}{L_1 + L_5}$$

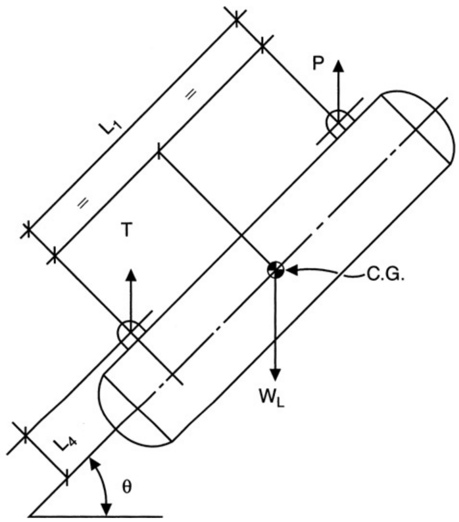
$$M_1 = \frac{w_6 L_5^2}{2}$$

$$M_2 = \frac{w_5 L_4^2}{2}$$

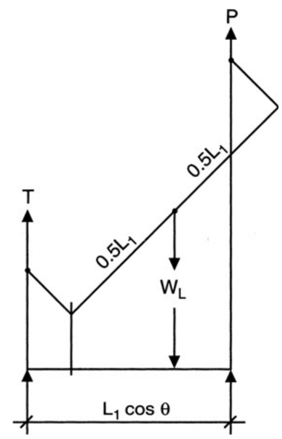
$$M_3 = \left(\frac{M_1 + M_3}{2}\right) - \frac{w_6 L_1^2}{8}$$

Find Lifting Loads at Any Lift Angle for a Symmetrical Horizontal Drum

Dimensions and Forces



Free-Body Diagram



Example

Steam drum:

$$W_L = 600\,\text{kips}$$

$$L_1 = 80\,\text{ft}$$

$$L4 = 5\,\text{ft}$$

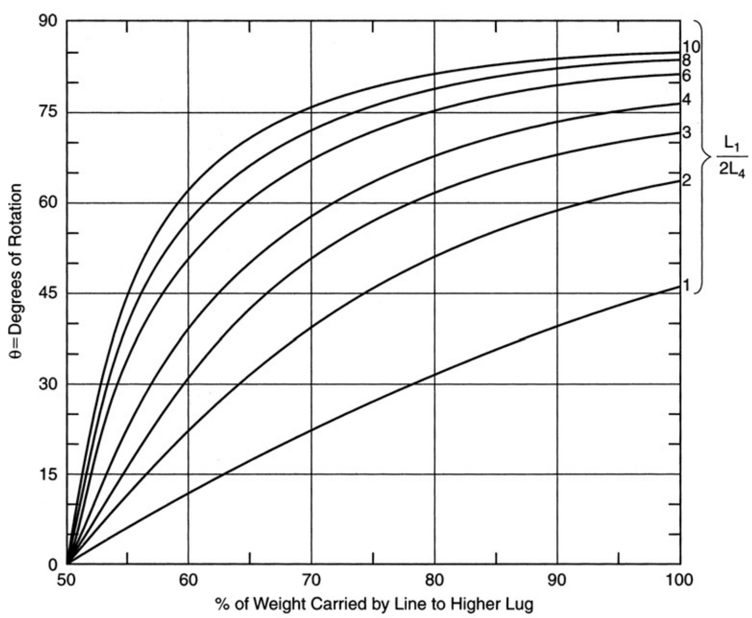
$$\frac{L_1}{2L_4} = \frac{80}{10} = 8$$

Curve is based on the following equation:

$$\frac{P}{W_L} = \frac{L_4}{L_1}(\tan\theta) + 0.5$$

Results from curve

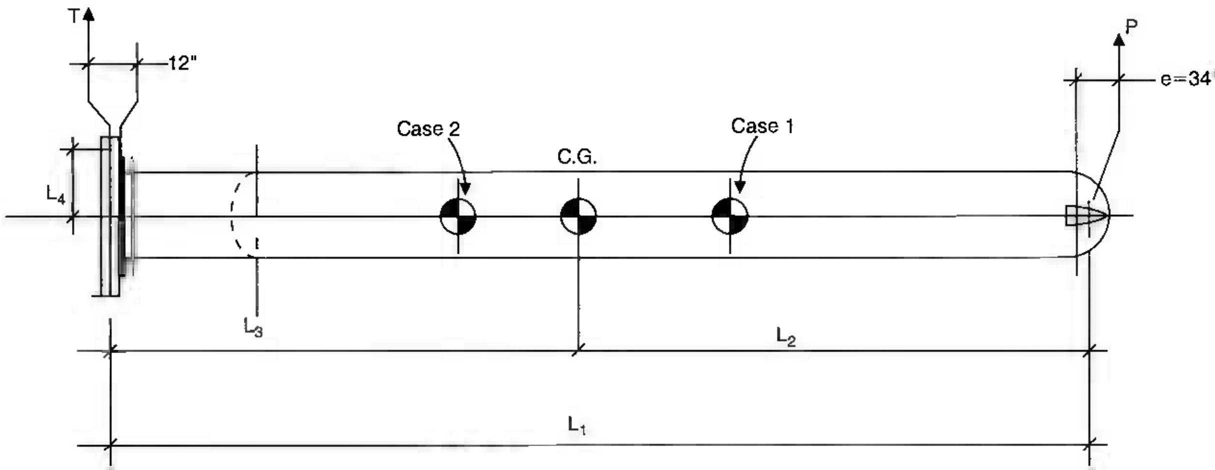
@ $\theta = 15^\circ = 51.6\%$
@ $\theta = 30^\circ = 53.6\%$
@ $\theta = 45^\circ = 56.3\%$
@ $\theta = 60^\circ = 60.8\%$
@ $\theta = 75^\circ = 73.3\%$



Reprinted by permission of The Babcock and Wicox Company, a McDermott Company.

Sample Problem

Distillation column:
18 ft in diameter x 280 ft OAL
260 ft tangent-to-tangent
W_L = 200 tons (400 kips)



Case 1: $L_3 > L_2$

$L_1 = 280 + 2.833 + 1 = 283.83$ ft
$L_2 = 283.83 - 162 = 121.83$ ft
$L_3 = 161 + 1 = 162$ ft
$L_4 = 10$ ft

Loads T and P		
θ	T	P
0	171.7	228.3
10	170.6	229.4
20	169.6	230.4
30	168.3	231.7
40	166.8	233.2
50	164.8	235.2
60	161.9	238.1
70	156.6	243.4
80	143.2	256.8
90	0	400

Case 2: $L_3 < L_2$

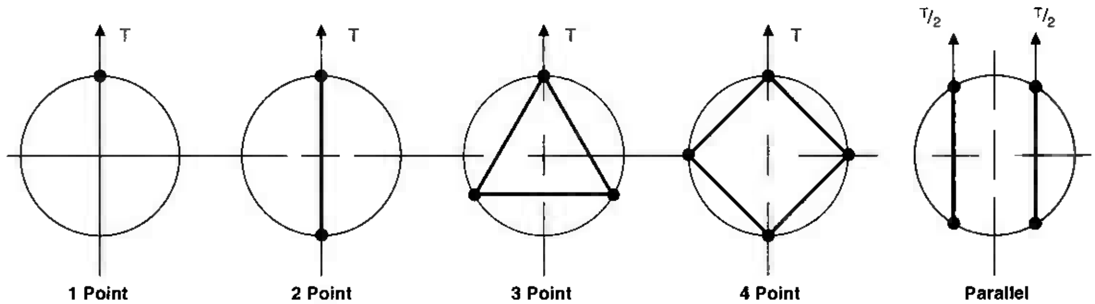
$L_1 = 283.83$ ft
$L_2 = 162$ ft
$L_3 = 121.83$ ft
$L_4 = 10$ ft

Loads T and P		
θ	T	P
0	228.3	171.7
10	226.9	173.1
20	225.4	174.6
30	223.7	176.3
40	221.7	178.3
50	219.1	180.9
60	215.1	184.9
70	208.1	191.9
80	190.1	209.8
90	0	400

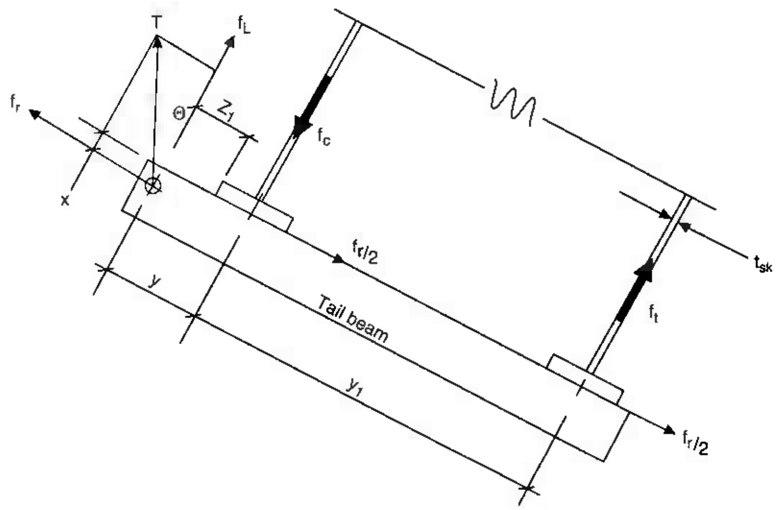
Procedure 10-5: Design of Tail Beams, Lugs, and Base Ring Details

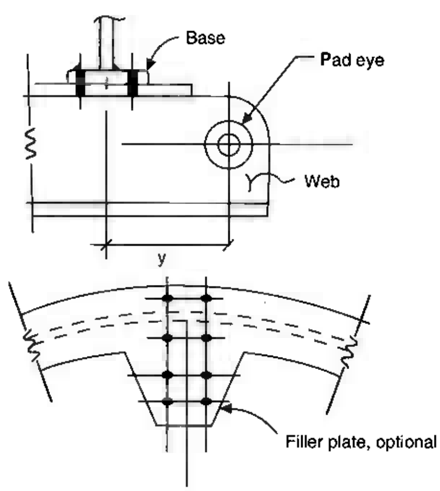
Design of Base Plate, Skirt, and Tail Beam

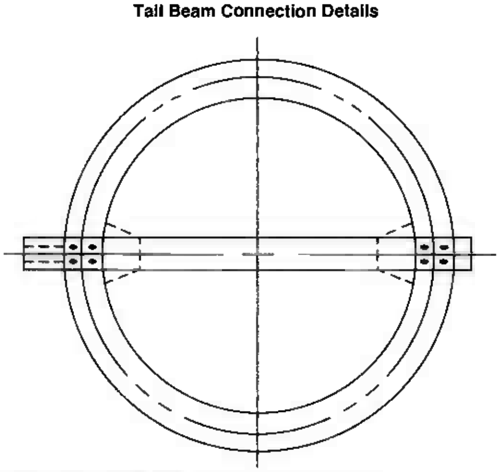
Base Ring-Stiffening Configuration



Loadings in Skirt and Base

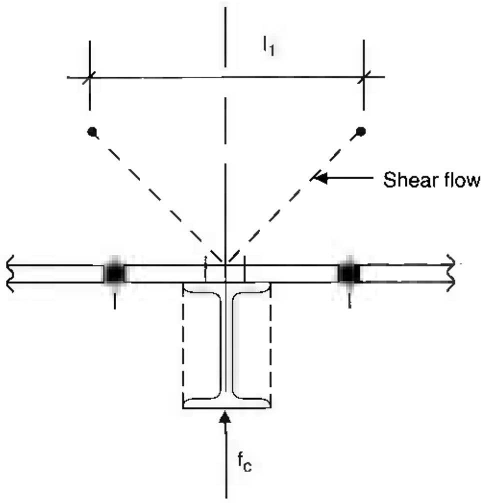






Skirt Crippling Criteria with Tailing Beam

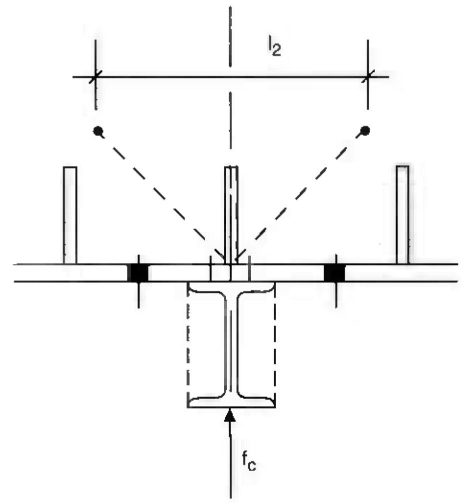
Base Type 1: Base Ring Only



$l_1 = N + 2l_r$

$l_r = 16t_{sk}$

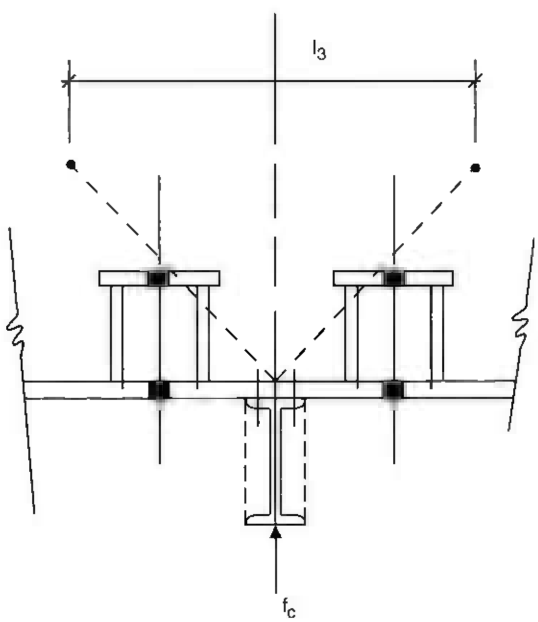
Base Type 2: Gussets Only



$l_2 = N + 2l_r$

$l_r = 16t_{sk}$

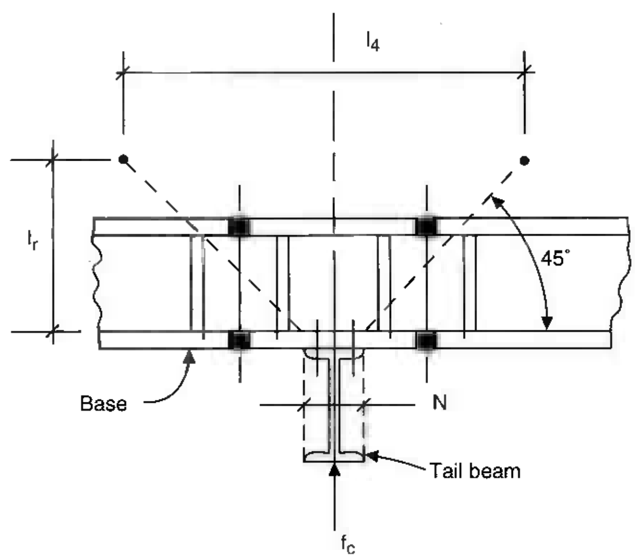
Base Type 3: w/Anchor Chairs



$l_3 = N + 2l_r$

$l_r = 0.55\sqrt{D_{sk}t_{sk}}$

Base Type 4: w/Continuous Top Ring

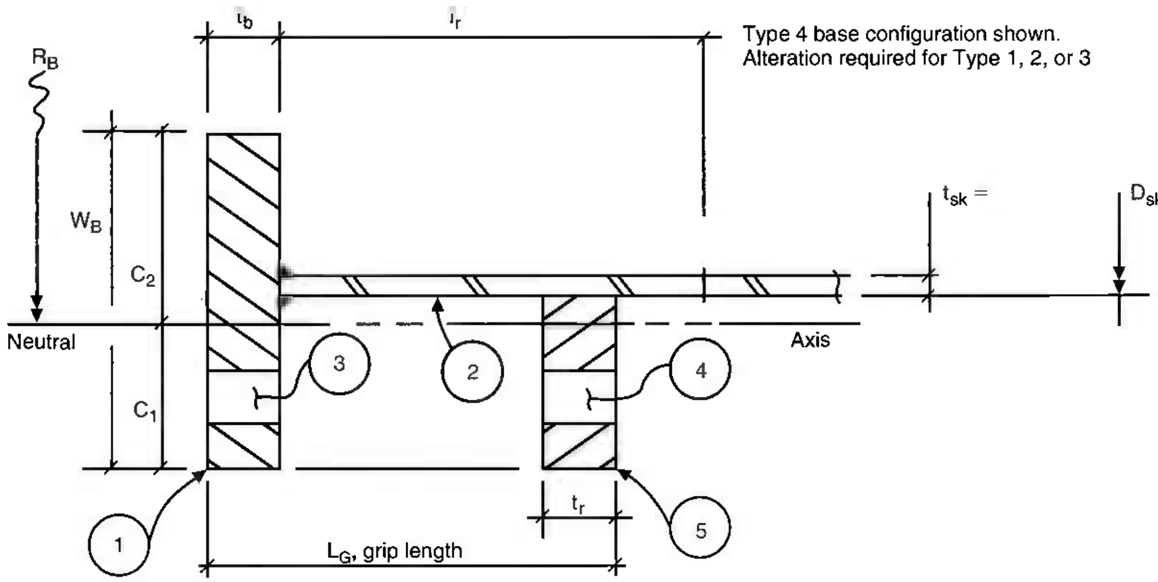


Note: N = 1 in. if a web stiffener is not used.

$l_4 = N + 2l_r$

$l_r = (L_G - t_b) + 0.55\sqrt{D_{sk}t_{sk}}$

Base Ring Design Check



Item	A	Y	Y^2	AY	AY^2	I_o
1						
2						
3(−)						
4(−)						
5						
Σ						

$$C_1 = \frac{\Sigma AY}{\Sigma A}$$

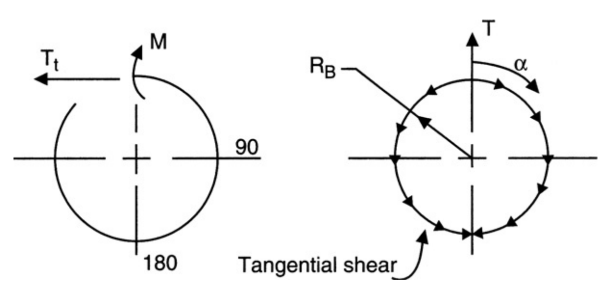
$$C_2 = W_B - C_1$$

$$I = \Sigma AY^2 + \Sigma I_o - C_1 \Sigma AY$$

$$R_B = \text{inside radius of base plate} + C_2$$

Internal Forces and Moments in the Skirt Base During Lifting

To determine the stresses in the base ring as a result of the tailing load, the designer must find the coefficients K_r and K_T based on angle α as shown and the type of stiffening in the skirt/base ring configuration.



$$M = K_r T R_B$$

$$T_t = K_T T$$

Skirt/Tail Beam Calculations

Tail Beam

- *Tailing loads, f_L and f_r.*

$$f_L = T\cos\theta$$

$$f_r = T\sin\theta$$

- *Maximum bending moment, M_b.*

$$M_b = xf_r + yf_L$$

- *Maximum bending stress, σ_b.*

$$\sigma_b = \frac{M_b}{Z}$$

Tail Beam Bolts

- *Shear load, f_s.*

$$f_s = \frac{0.5f_r}{n}$$

- *Shear stress, τ.*

$$\tau = \frac{f_s}{A_b}$$

- *Tension force, f_t.*

Note: y_1 = mean skirt diameter or centerline of bolt group if a filler plate is used.

$$f_t = \frac{M_b}{y_1}$$

Skirt

- *Tension stress in bolts, σ_T.*

$$\sigma_T = \frac{f_T}{N_bA_b}$$

- *Compressive force in skirt, f_c.*

$$f_c = f_L + f_t$$

- *Skirt crippling is dependent on the base configuration and lengths l_1 through l_4.*

N = 1 in. if web stiffeners are not used
N = width of top flange of tail beam if web stiffeners are used

- *Compressive stress in skirt, σ_c.*

$$\sigma_c = \frac{f_c}{t_{sk}l_n}$$

- *Check shear stress, τ, in base to skirt weld.*

$$\tau = \frac{f_r}{\pi D_{sk} \cdot 0.707w_4}$$

Base Plate

- *Bending moment in base plate, M_b.*

$$M_b = K_rTR_B$$

- *Find tangential force, T_t.*

$$T_t = K_TT$$

- *Total combined stress, σ.*

$$\sigma_T = \frac{M_bC_1}{I} + \frac{T_t}{A} \quad \text{(tension)}$$

$$\sigma_C = \frac{-M_bC_2}{I} - \frac{T_t}{A} \quad \text{(compression)}$$

Size Base Ring Stiffeners

F_1 = force in strut or tailing beam, lb
F_1 is (+) for tension and (–) for compression

- *Tension stress*, σ_T.

$$\sigma_T = \frac{F_n}{A_s}$$

- *Critical buckling stress per AISC*, σ_{cr}.

$$C_c = \sqrt{\frac{2\pi^2}{F_y}}$$

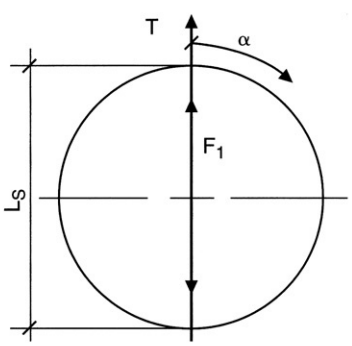
$$\sigma_{cr} = \frac{\left[\left(1 - \left(KL_s^2/r\right)/2C_c^2\right)\right]F_y}{(5/3) + ((3KL_s/r)/8C_c) - \left((KL_s/r)^3/8C_c^3\right)}$$

- *Actual compressive stress*, σ_c.

$$\sigma_c = \frac{F_n}{A_s}$$

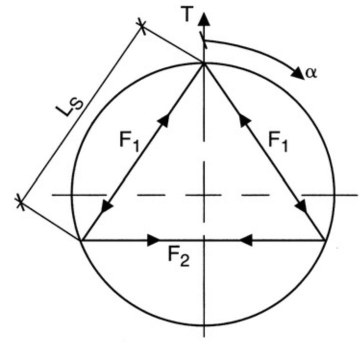
Note: Evaluate all struts as tension and compression members regardless of sign, because when the vessel is sitting on the ground, the loads are the reverse of the signs shown.

Two Point



$F_1 = (+)0.5T$

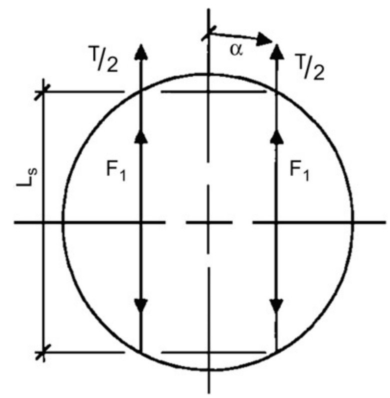
Three Point



$F_1 = (+)0.453T$

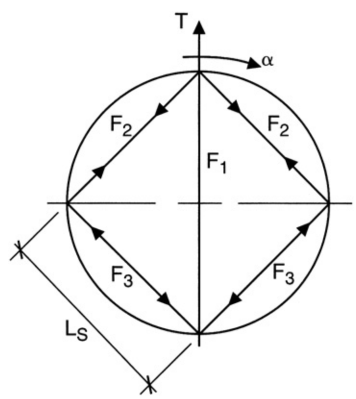
$F_2 = (-)0.329T$

Parallel Beams/Struts



$F_1 = (+)0.25T$

Four Point



$F_1 = (+)0.5T$

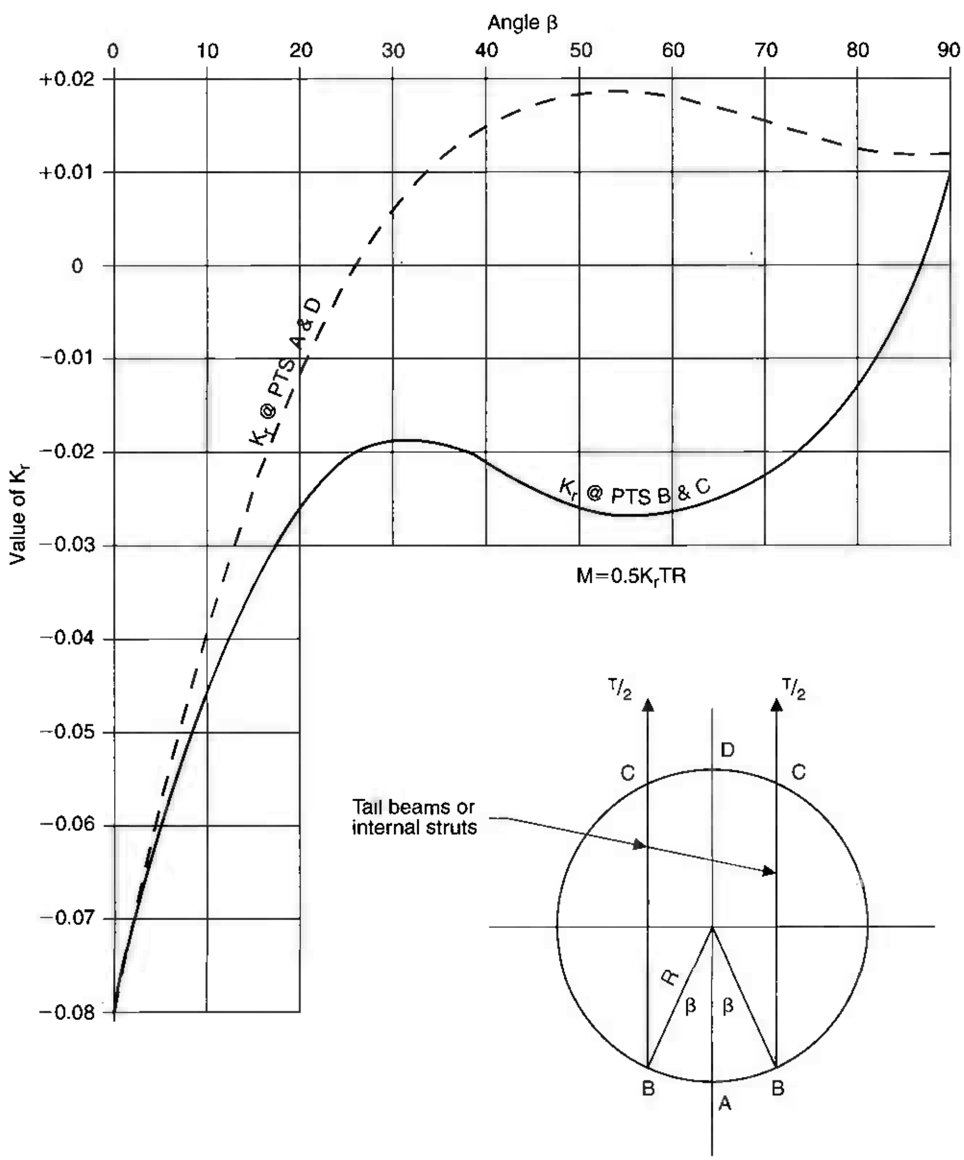
$F_2 = (-)0.273T$

$F_3 = (+)0.273T$

Table 10-7
Internal moment coefficients for base ring

	One Point		Two Point		Three Point		Four Point	
Angle α	K_r	K_T	K_r	K_T	K_r	K_T	K_r	K_T
0	0.2387	−0.2387	0.0795	−0.2387	−0.0229	0.1651	0.0093	−0.1156
5	0.1961	−0.2802	0.0587	−0.2584	−0.0148	0.1708	0.0048	−0.1188
10	0.1555	−0.3171	0.0398	−0.2736	−0.0067	0.1764	0.0012	−0.1188
15	0.1174	−0.3492	0.0229	−0.2845	−0.0055	0.1747	−0.0015	−0.1155
20	0.0819	−0.3763	0.0043	−0.2908	−0.0042	0.1729	−0.0033	−0.1089
25	0.0493	−0.3983	−0.0042	−0.2926	0.0028	0.1640	−0.0043	−0.0993
30	0.0197	−0.4151	−0.0145	−0.2900	0.0098	0.1551	−0.0045	−0.0867
35	−0.0067	−0.4266	−0.0225	−0.2831	0.0103	0.1397	−0.0041	−0.0713
40	−0.0299	−0.4328	−0.0284	−0.2721	0.0107	0.1242	−0.0031	−0.0534
45	−0.0497	−0.4340	−0.0321	−0.2571	0.0093	0.1032	−0.0017	−0.0333
50	−0.0663	−0.4301	−0.0335	−0.2385	0.0078	0.0821	−0.0001	−0.0112
55	−0.0796	−0.4214	−0.0340	−0.2165	0.0052	0.0567	0.0017	0.0126
60	−0.0897	−0.4080	−0.0324	−0.1915	0.0025	0.0313	0.0033	0.0376
65	−0.0967	−0.3904	−0.0293	−0.1638	0.0031	0.0031	0.0046	0.0636
70	−0.1008	−0.3688	−0.0250	−0.1338	0.0037	−0.0252	0.0055	0.0901
75	−0.1020	−0.3435	−0.0197	−0.1020	−0.0028	−0.0548	0.0056	0.1167
80	−0.1006	−0.3150	−0.0136	−0.0688	−0.0092	−0.0843	0.0049	0.1431
85	−0.0968	−0.2837	−0.0069	−0.0346	−0.0107	−0.1134	0.0031	0.1688
90	−0.0908	−0.2500	0	0	−0.0121	−0.1425	0	0.1935
95	−0.0830	−0.2144	0.0069	0.0416	−0.0114	−0.1694	−0.0031	−0.1688
100	−0.0735	−0.1774	0.0135	0.0688	−0.0107	−0.1963	−0.0049	−0.1431
105	−0.0627	−0.1394	0.0198	0.1020	−0.0074	−0.2194	−0.0057	−0.1167
110	−0.0508	−0.1011	0.0250	0.1338	−0.0033	−0.2425	−0.0055	−0.0901
115	−0.0381	−0.0627	0.0293	0.1638	0.0041	−0.2603	−0.0046	−0.0636
120	−0.0250	−0.0250	0.0324	0.1915	0.0114	−0.2781	−0.0033	−0.0376
125	−0.0016	0.0118	0.0340	0.2165	0.0107	−0.1060	−0.0017	−0.0126
130	0.0116	0.0471	0.0335	0.2385	0.0100	0.0661	0.0001	0.0112
135	0.0145	0.0804	0.0321	0.2571	0.0083	0.0448	0.0017	0.0333
140	0.0268	0.1115	0.0284	0.2721	0.0066	0.0234	0.0031	0.0534
145	0.0382	0.1398	0.0225	0.2831	0.0045	0.0104	0.0041	0.0713
150	0.0486	0.1551	0.0145	0.2900	0.0024	−0.0026	0.0045	0.0867
155	0.0577	0.1870	0.0042	0.2926	−0.0005	−0.0213	0.0043	0.0993
160	0.0654	0.2053	−0.0083	0.2908	−0.0015	−0.0399	0.0033	0.1089
165	0.0715	0.2198	−0.0225	0.2845	−0.0028	−0.0484	0.0015	0.1155
170	0.0760	0.2301	−0.0398	0.2736	−0.0041	−0.0569	−0.0012	0.1188
175	0.0787	0.2366	−0.0587	0.2584	−0.0046	−0.0597	−0.0048	0.1188
180	0.0796	0.2387	−0.0795	0.2387	−0.0051	−0.0626	−0.0093	0.1156

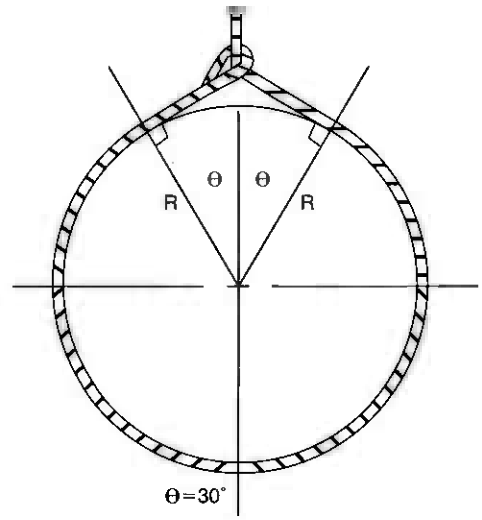
Values of Moment Coefficient, K_r, for Base Ring With Two Parallel Tail Beams or Internal Struts

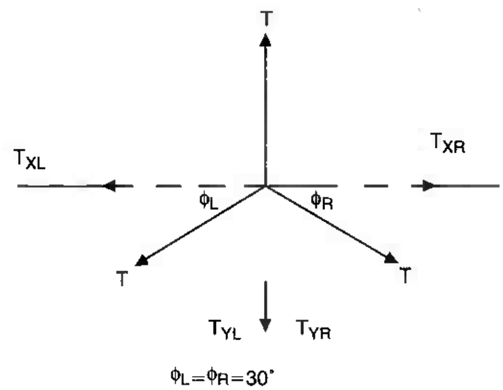


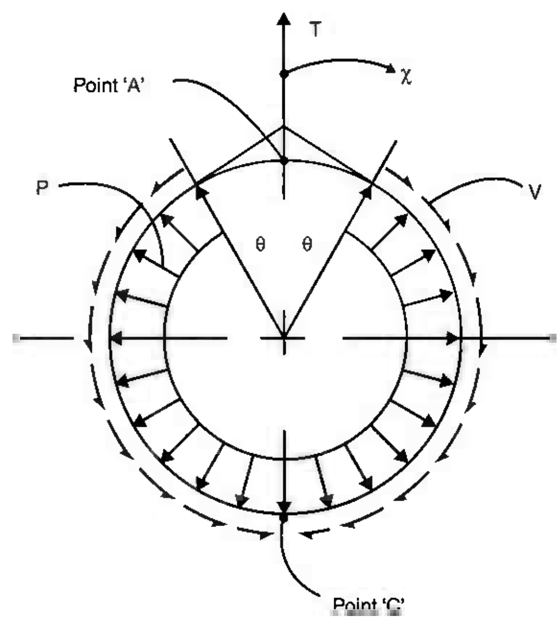
Notes:

1. Based on R. J. Roark, *Formulas for Stress and Strain*, 3rd Edition, Case 25.
2. The curve shows moment coefficients at points C and D. The moment coefficients at point A and B are equal and opposite.
3. Positive moments put the inside of the vessel in circumferential tension.
4. The signs of coefficients are for hanging loads. For point support loads underneath the vessel, the signs of the coefficients should be reversed.

Design of Vessel for Choker (Cinch) Lift at Base







- *Uniform load, p.*

$$p = \frac{T}{R}$$

- *Moments in ring at points A and C.*

$$M_A = -0.1271TR$$
$$M_C = -0.0723TR$$

- *Tension/compression forces in ring at points A and C.*

$$T_A = -0.6421T$$
$$T_c = -1.2232T$$

- *Combined stress at point A, inside of ring.*

$$\sigma_A = \frac{T_A}{A} + \frac{M_A}{Z_{in}}$$

- *Combined stress at point A, outside of ring.*

$$\sigma_A = \frac{T_A}{A} - \frac{M_A}{Z_{out}}$$

- *Combined stress at point C, inside of ring.*

$$\sigma_c = \frac{T_c}{A} + \frac{M_c}{Z_{in}}$$

- *Combined stress at point C, outside of ring.*

$$\sigma_c = \frac{T_c}{A} - \frac{M_c}{Z_{out}}$$

Note: Assume that the choker is attached immediately at the base ring even though this may be impossible to achieve. Then use the properties of the base ring for A and Z.

From R. J. Roark, *Formulas for Stress and Strain,* 5th Edition, McGraw-Hill Book Co., Table 17, Cases 12 and 18 combined.

Design of Tailing Lugs

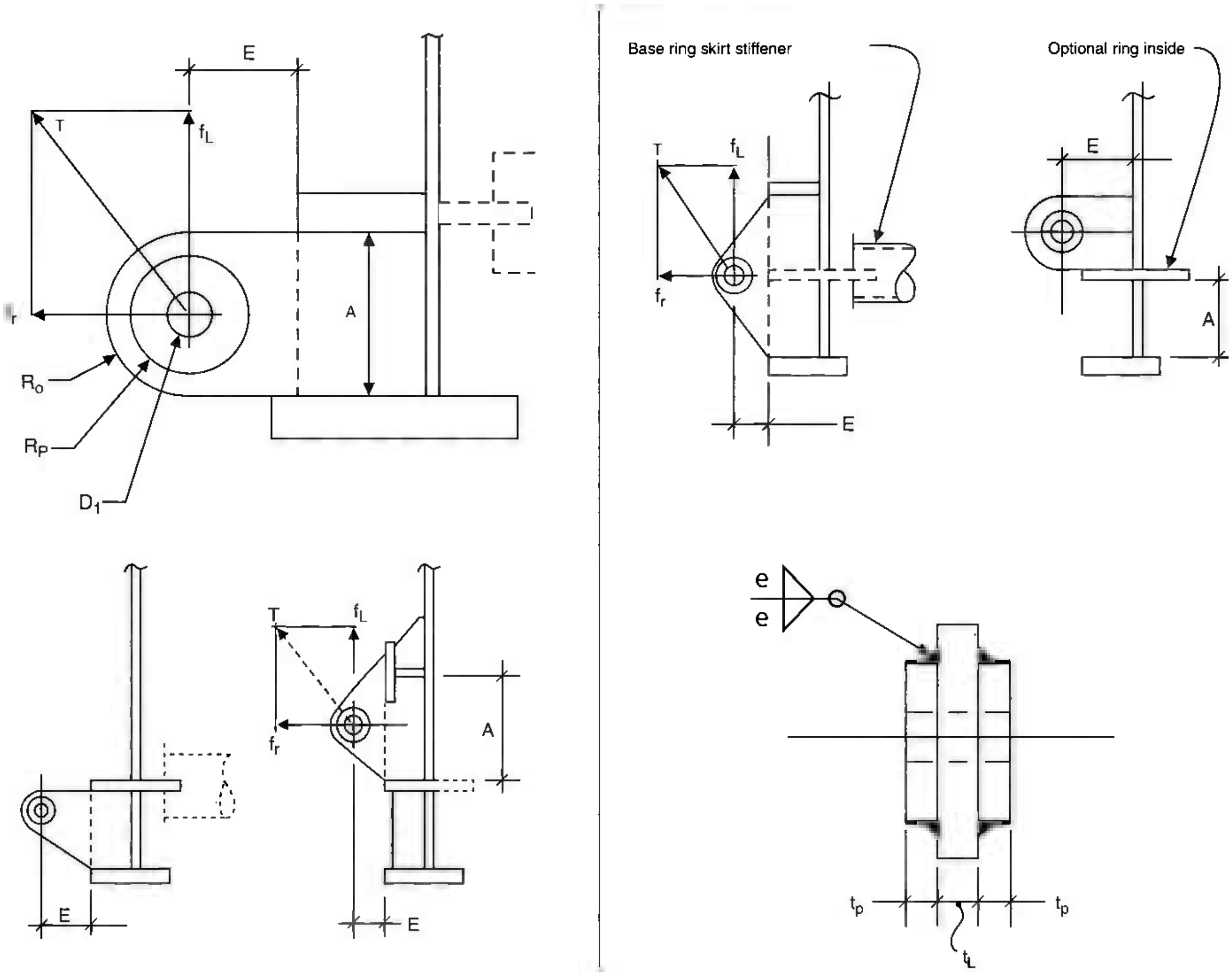


Table 10-8
Dimensions for tailing lugs

Tail Load (kips)	t_L	t_P	E	R_o	R_P	D_1	e
<10			None required				
10 to 20	0.75	NR	3	4	NR	2.375	NR
21 to 40	0.75	0.375	3	4	3.5	2.375	0.3125
41 to 70	1	0.5	3	4	3.5	2.375	0.3125
71 to 100	1	0.5	4	5	4.5	3.4375	0.3125
101 to 130	1.5	0.5	4	5	4.5	3.4375	0.3125
131 to 170	1.625	0.75	4	5	4.5	3.4375	0.375
171 to 210	1.625	0.75	5	6	5.5	4.5	0.375
211 to 250	2	0.75	5	6	5.5	4.5	0.4375
251 to 300	2.25	1	5	6	5.5	4.5	0.5
>300			Special design required				

Formulas

The tailing lug is designed like all other lugs. The forces are determined from the tailing load, T, calculated per this procedure. The ideal position for the tailing lug is to be as close as possible to the base plate for stiffness and transmitting these loads through the base to the skirt. The option of using a tailing lug versus a tailing beam is the designer's choice. Either can accommodate internal skirt rings, stiffeners, and struts.

Design as follows:

- *Area required at pin hole, A_r.*

$$A_r = \frac{T}{F_s}$$

- *Area available at pin hole, A_a.*

$$A_a = (At_L)\text{-}(D_1 t_L)$$

- *Bending moment in lug, M_b.*

$$M_b = f_L E$$

- *Section modulus of lug,* Z.

$$Z = \frac{t_L A^2}{6}$$

- *Bending stress in lug*, σ_b.

$$\sigma_b = \frac{M_b}{Z}$$

- *Area required at pin hole for bearing, A_r.*

$$A_r = \frac{T}{F_p}$$

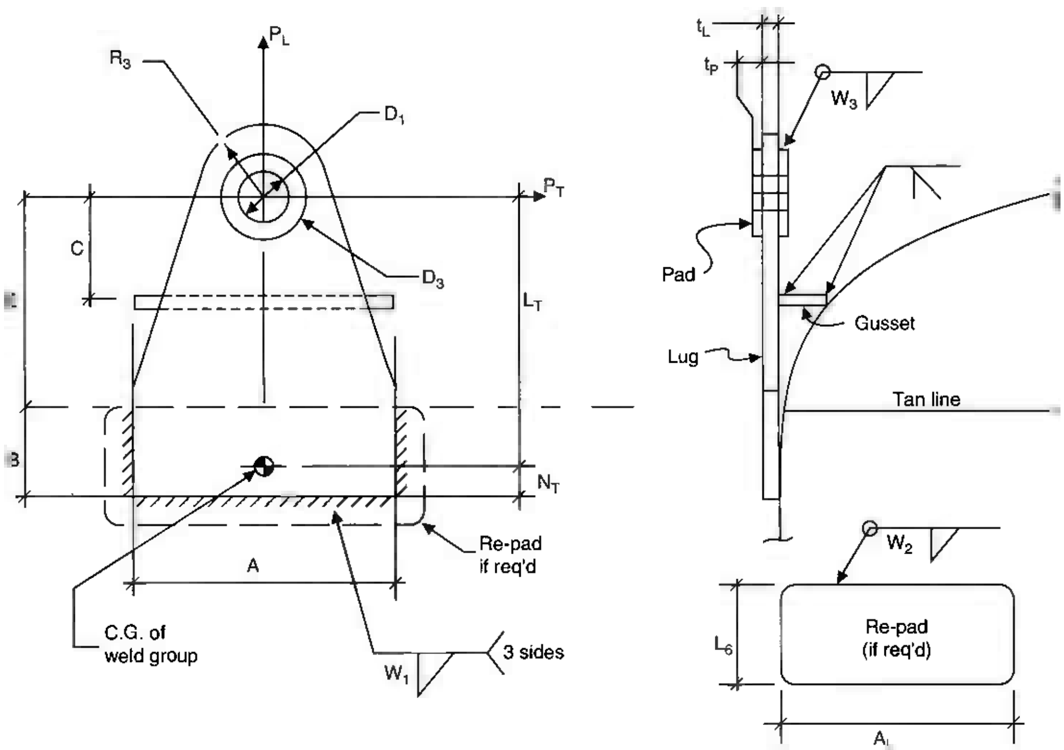
- *Area available at pin hole for bearing, A_a.*

$$A_a = D_2 t_L$$

Note: Substitute $t_L + 2t_p$ for t_L in the preceding equations if pad eyes are used.

Procedure 10-6: Design of Top Head and Cone Lifting Lugs

Design of Top Head/Cone Lug



Dimensions

$$N_T = \frac{B^2}{A + 2B}$$

$$L_T = E + B - N_T$$

$$\theta_1 = \arctan \frac{2L_1}{A}$$

$$L_2 = \frac{L_1}{\sin \theta_1}$$

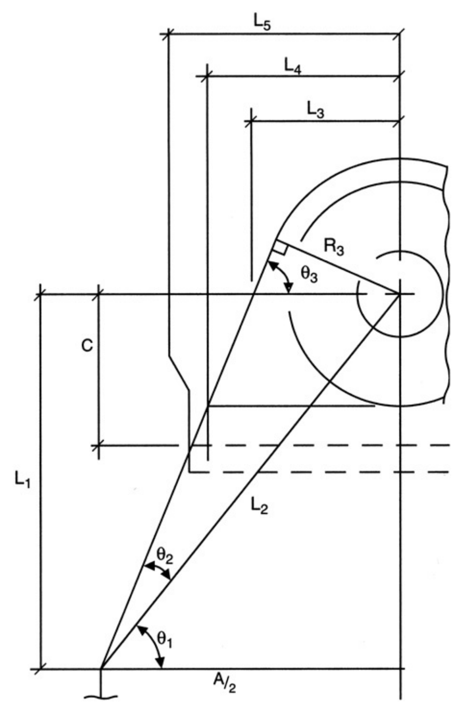
$$\theta_2 = \arcsin \frac{R_3}{L_2}$$

$$\theta_3 = \theta_1 + \theta_2$$

$$L_3 = \frac{R_3}{\sin \theta_3}$$

$$L_4 = 0.5A - \frac{L_1 - 0.5D_3}{\tan \theta_3}$$

$$L_5 = 0.5A - \frac{L_1 - C}{\tan \theta_3}$$



Lug

- *Maximum bending moment in lug, M_L.*

$$M_L = PE$$

- *Section modulus, lug, Z.*

$$Z = \frac{A^2 t_L}{6}$$

- *Bending stress, lug, σ_b.*

$$\sigma_b = \frac{M_L}{Z}$$

- *Thickness of lug required, t_L.*

$$t_L = -\frac{6M_L}{A^2 F_b}$$

- *Tension at edge of pad, σ_T.*

$$\sigma_T = \frac{P_L}{2L_4 t_L}$$

- *Net section at pin hole, Ap.*

$$A_p = 2L_3 t_L + 2t_p(D_3 - D_1)$$

- *Shear stress at pin hole, τ.*

$$\tau = \frac{P_L}{A_p}$$

- *Net section at top of lug, A_n.*

$$A_n = t_L\left(R_3 - \frac{D_1}{2}\right) + 2t_p\left(\frac{D_3 - D_1}{2}\right)$$

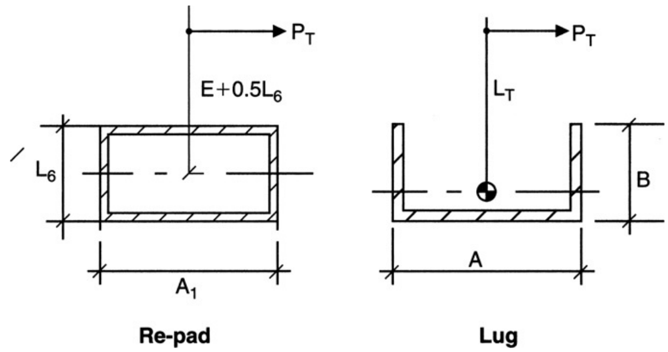
- *Shear stress at top of lug, τ.*

$$\tau = \frac{P_T}{A_n}$$

- *Pin bearing stress, σ_p.*

$$\sigma_p = \frac{P_T}{D_3(t_L + 2t_p)}$$

Check Welds



- *Polar moment of inertia, J_w.*

$$\text{Re-pad:}\quad J_w = \frac{(A_1 + L_6)^3}{6}$$

$$\text{Lug:}\quad J_w = \frac{(A + 2B)^3}{12} - \frac{B^2(A + B)^2}{(A + 2B)}$$

- *Moment, M_1.*

$$M_1 = L_T P_T$$

Lug Weld

- *Find loads on welds.*
- *Transverse shear due to P_T, f_1.*

$$f_1 = \frac{P_T}{A + 2B}$$

- *Transverse shear due to M_1, f_2.*

$$f_2 = \frac{M_1(B - N_T)}{J_w}$$

- *Longitudinal shear due to M_1, f_3.*

$$f_3 = \frac{M_1 B}{J_w}$$

- *Combined shear load, f_r.*

$$f_r = \sqrt{(f_1 + f_2)^2 + f_3^2}$$

- *Size of weld required, w_1.*

$$w_1 = \frac{f_r}{0.707F_s}$$

Note: If w_1 exceeds the shell plate thickness, then a re-pad must be used.

Re-pad Weld

- *Moment, M_2.*

$$M_2 = P_T(E + 0.5L_6)$$

- *Transverse shear due to P_T, f_1.*

$$f_1 = \frac{P_T}{2A_1 + 2L_6}$$

- *Transverse shear due to M_2, f_2.*

$$f_2 = \frac{0.5M_2L_6}{J_w}$$

- *Longitudinal shear due to M_2, f_3.*

$$f_3 = \frac{M_2L_6}{J_w}$$

- *Combined shear load, f_r.*

$$f_r = \sqrt{(f_1 + f_2)^2 + f_3^2}$$

- *Size of weld required, w_1.*

$$w_2 = \frac{f_r}{0.707F_s}$$

Pad Eye Weld

- *Unit shear load on pad, f_4.*

$$f_4 = \frac{P_T t_p \pi D_2}{2t_p + t_L}$$

- *Size of weld required, w_3.*

$$w_3 = \frac{f_4}{0.707F_s}$$

Top Head Lug for Large Loads

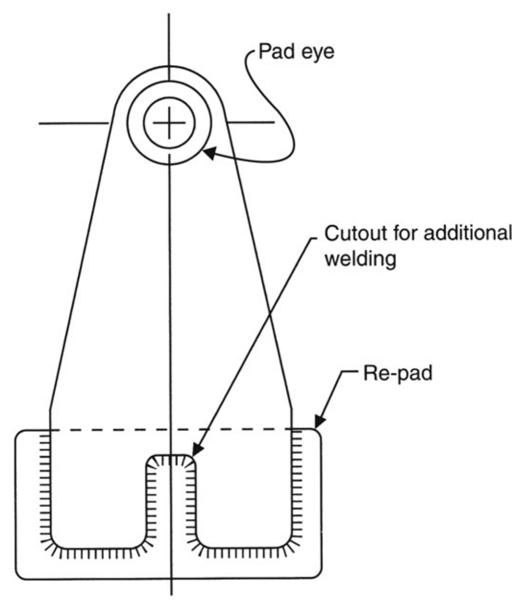
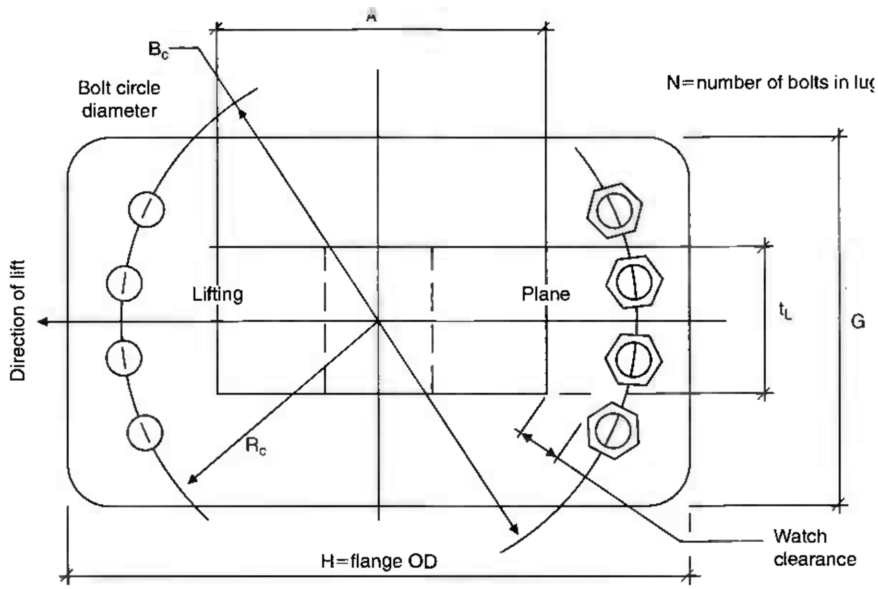


Table 10-9
Dimensions for top head or cone lugs

Type	Note	Total Erection Weight (tons)	Shackle Size (tons)	Lug Thickness t_L	A	B	C	E	R_3	W_1	Gusset Thickness t_q	Pads D_3	Pads t_p	Pads W_2	Lift Hole Dia D_1	Lug Matl. Min. Yield (psi)
36-in. to 48-in. Inside Diameter																
1-A		0–30	35	1	12	12	7	13	3	⅜	½				2½	30,000
1-B		31–65	50	1½	14	12	8	14	4	¾	½	7	⅜	¼	3	30,000
1-C	1	66–100	50	1¾	16	14	9	15	4½	1	¾	8	½	¼	3	30,000
54-in. to 72-in. Inside Diameter																
2-A		0–30	35	1	12	12	7	15	3	⅜	½				2½	30,000
2-B		31–65	50	1½	16	14	8	17	4	⅝	½	7	⅜	¼	3	30,000
2-C	1	66–100	50	1¾	18	14	9	18	4½	⅞	½	8	½	¼	3	30,000
2-D	2	101–150	75	2	20	16	11	20	5	1¼	¾	9	¾	⅜	3½	38,000
78-in. to 108-in. Inside Diameter																
3-A		0–30	35	1	14	10	6	18	3	½	½				2⅜	30,000
3-B		31–65	50	1	20	12	7	19	4	⅝	½	7	⅜	¼	2⅞	30,000
3-C	1	66–100	50	1¼	22	14	9	21	4½	¾	¾	8	¾	½	2⅞	30,000
3-D	2	101–150	75	1¾	22	16	10	23	5	1¼	1	9	1	⅜	3⅜	38,000
3-E	3	151–200	130	2	25	18	12	25	6½	1⅜	1	12	1	½	4⅜	38,000
114-in. to 144-in. Inside Diameter																
4-A		0–30	35	1	14	10	5	20	3	½	½				2⅜	30,000
4-B		31–65	50	1	22	14	7	22	4	½	½	7	⅜	¼	2⅞	30,000
4-C	1	66–100	50	1¼	26	14	9	25	4½	¾	¾	8	¾	¼	2⅞	30,000
4-D	2	101–150	75	1¾	26	16	12	27	5	1¼	1	9	1	⅜	3⅜	38,000
4-E	3	151–200	130	2	28	18	12	27	6½	1⅜	1	12	1	½	4⅜	38,000
150-in. to 180-in. Inside Diameter																
5-A		0–30	35	1	14	10	5	21	3	½	½				2⅜	30,000
5-B		31–65	50	1	22	14	6	23	4	⅝	½	7	⅜	¼	2⅞	30,000
5-C	1	66–100	50	1¼	26	14	10	28	4½	¾	¾	8	¾	¼	2⅞	30,000
5-D	2	101–150	75	1¾	26	16	12	30	5	1¼	1	9	1	⅜	3⅜	38,000
5-E	3	151–200	130	2	28	18	12	30	6½	1⅜	1⅜	12	1	½	4⅜	38,000
186-in. to 216-in. Inside Diameter																
6-A		0–30	35	1	16	10	4	24	3	½	½				2⅜	30,000
6-B		31–65	50	1	24	14	6	26	4	½	½	7	⅜	¼	2⅞	30,000
6-C	1	66–100	50	1¼	28	14	9	31	4½	¾	¾	8	¾	¼	2⅞	30,000
6-D	2	101–150	75	1¾	28	16	12	34	5	1¼	1	9	1	⅜	3⅜	38,000
6-E	3	151–200	130	2	30	18	12	34	6½	1⅜	1⅜	12	1	½	4⅜	38,000

Notes:
1. For 75-ton shackle, increase lift hole to 3.375
2. For 130-ton shackle, increase lift hole to 4.375
3. For 150-ton shackle, increase lift hole to 5.125

Procedure 10-7: Design of Flange Lugs



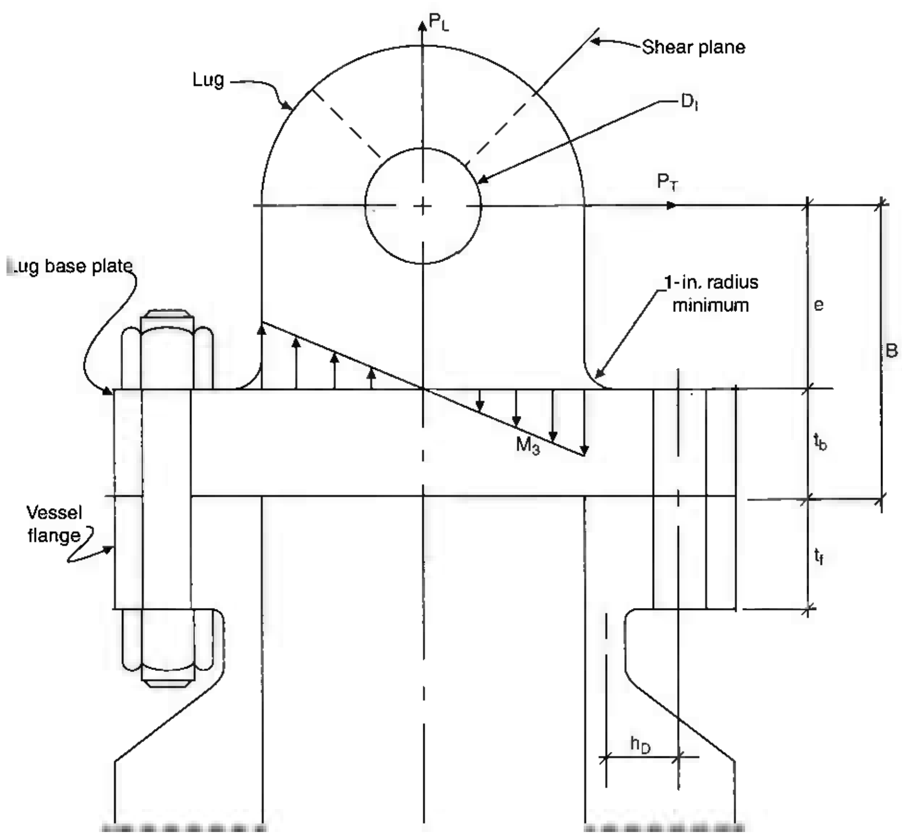


Table 10-10
Flange lug dimensions

Load Capacity (tons)	D_1	t_L	t_b	A	B	G	H	E
50	3.38	2	2	8	11	12	30	9
100	5	3	3	14	12	24	36	9
200	6	4	4	18	14	30	40	10
400	8	5	5	20	16	36	46	11
600	9	6	6	24	22	40	58	16
800	10	9	7	28	24	42	60	17

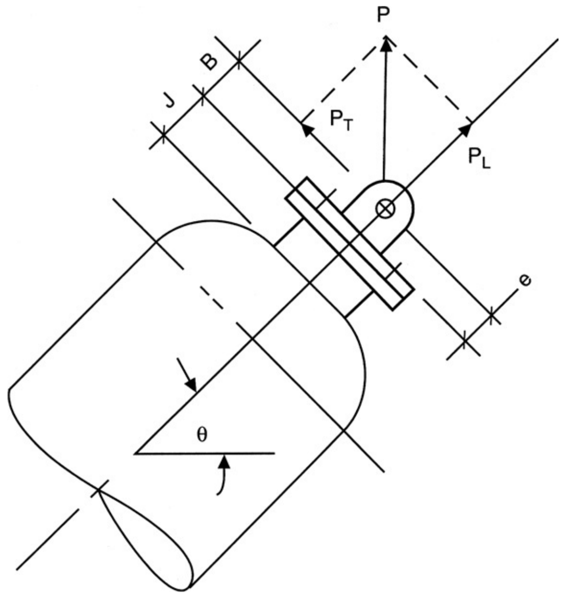
Table 10-11
Bolt properties

Bolt Size	A_b	A_s	T_b
0.5–13	0.196	0.112	12
0.625–11	0.307	0.199	19
0.75–10	0.442	0.309	28
0.875–9	0.601	0.446	39
1–8	0.785	0.605	51
1.125–8	0.994	0.79	56
1.25–8	1.227	1	71
1.375–8	1.485	1.233	85
1.5–8	1.767	1.492	103
1.75–8	2.405	2.082	182
2–8	3.142	2.771	243
2.25–8	3.976	3.557	311
2.5–8	4.909	4.442	389
2.75–8	5.94	5.43	418
3–8	7.069	6.506	501
3.25–8	8.3	7.686	592
3.5–8	9.62	8.96	690
3.75–8	11.04	10.34	796
4–8	12.57	11.81	910

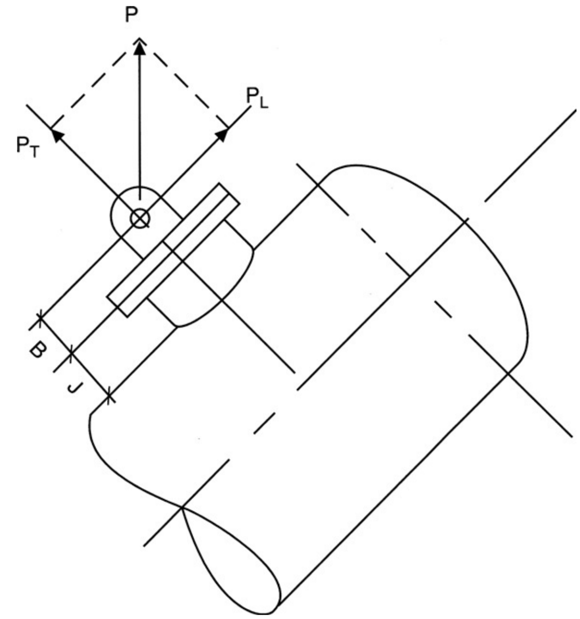
Table 10-12
Values of S_u

Bolt Dia, d_b	Material	S_u (ksi)
< 1	A-325	120
1.125–1.5	A-325	105
1.625–2.5	A-193-B7	125
2.625–4	A-193-B7	110

Top Flange Lug



Side Flange Lug



$$P_L = P \sin\theta$$

$$P_T = P \cos\theta$$

$$P_E = \frac{P_L}{A} + \frac{3P_T e}{A^2}$$

$$M_1 = P_T B$$

$$M_2 = P_T(B + J)$$

$$M_3 = P_T e$$

$$M_u = X_n \cos\alpha_n N_b$$

$$M_a = \frac{M_u M_1}{\sum M_u}$$

$$X_n = R_b \cos\alpha_n$$

$$y_n = R_b \sin\alpha_n$$

$$F_n = \frac{M_a}{X_n N_b}$$

$$f_s = \frac{P_T}{N}$$

$$\sigma_T = \frac{F_n}{A_s}$$

$$F_s = 15 \text{ ksi} \left(1 - \frac{\sigma_T A_b}{T_b}\right)$$

$$A_s = 0.7854\ (d - 0.1218)^2$$

$$\tau = \frac{f_s}{A_s} < F_s$$

$$T_b = 0.75_u A_s$$

$$0.6F_y < F_T < 40 \text{ ksi}$$

Design Process

1. Determine loads
2. Check of lug:
   a. Shear at pin hole.
   b. Bending of lug.
   c. Bearing at pin hole.
3. Check of base plate.
4. Check of nozzle flange.
5. Check of flange bolting.
6. Check of local load at nozzle to head or shell junction.

Step 1: Determine loads.

- Determine loads P_T and P_L for various lift angles, θ.
- Determine uniform loads w_1 and w_2 for various angles, θ.
- Using w_1 and w_2, solve for worst case of combined load, P_E.
- Determine worst-case bending moment in lug, M_3.

Step 2: Check of lug.

a. Shear at pin hole:
   - *Area required, A_r.*

$$A_r = \frac{P_E}{F_s}$$

   - *Area available at pin hole, A_a.*

$$A_a = (At_L) - (D_1 t_L)$$

b. Bending of lug due to M_3:
   - *Section modulus, Z.*

$$Z = \frac{t_L A^2}{6}$$

   - *Bending stress, lug, σ_b.*

$$\sigma_b = \frac{M_3}{Z}$$

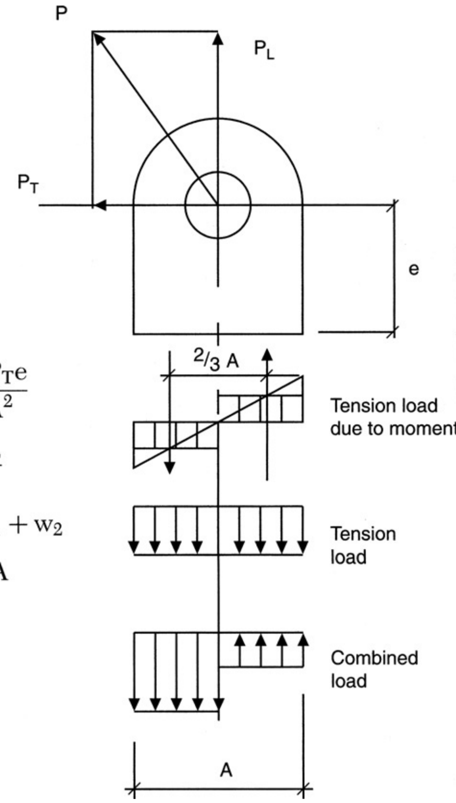
c. Bearing at pin hole:
   - *Bearing required at pin hole A_r.*

$$A_r = \frac{P_E}{F_p}$$

   - *Bearing available, A_a.*

$$A_a = D_2 t_L$$

Maximum Tension in Lug



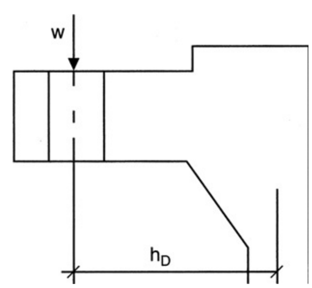
$$w_1 = \frac{3P_T e}{A^2}$$

$$w_2 = \frac{P_L}{A}$$

$$w = w_1 + w_2$$

$$P_E = wA$$

Check of Nozzle Flange



- *Unit load, w.*

$$w = \frac{P_E}{\pi B_c}$$

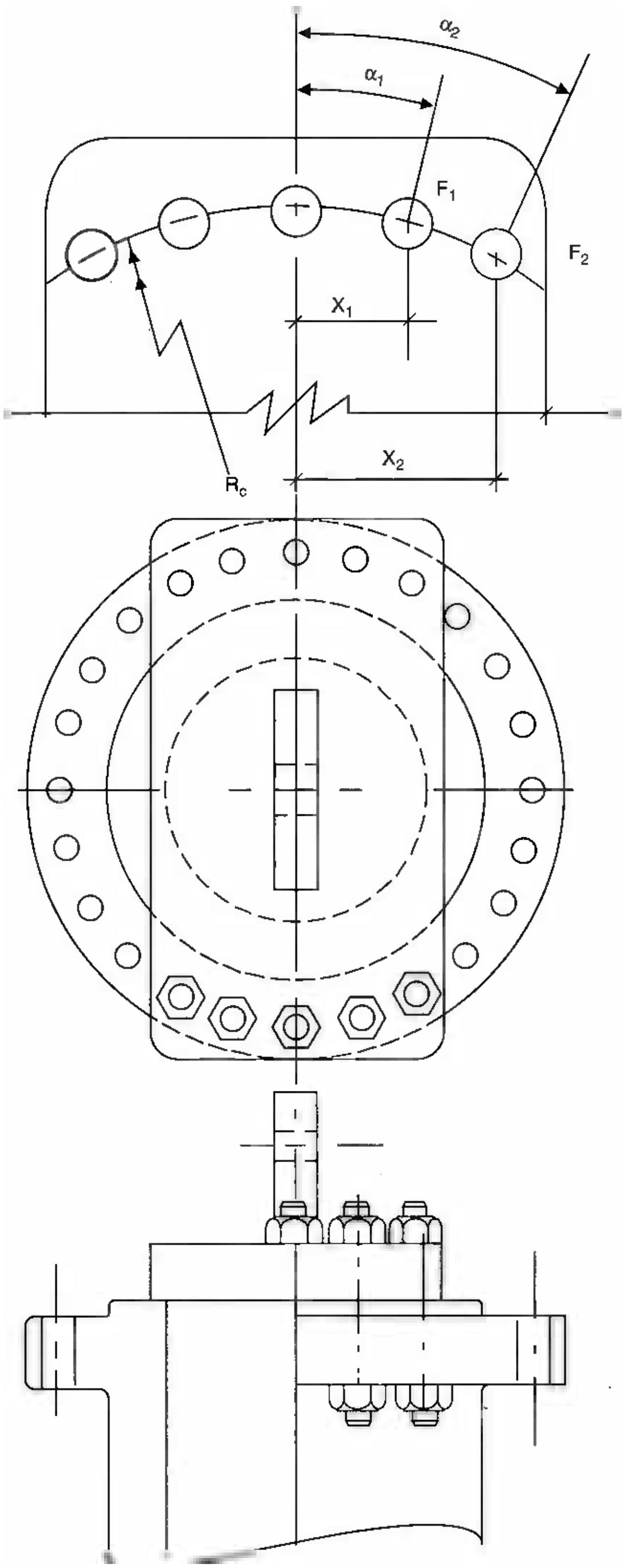
- *Bending moment, M.*

$$M = wh_D$$

- *Bending stress, σ_b.*

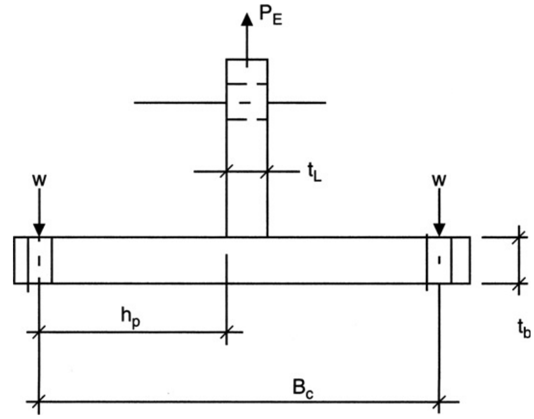
$$\sigma_b = \frac{6M}{t_f^2}$$

Bolt Loads for Rectangular Lugs



Design of Full Circular Base Plate for Lug

- If a full circular plate is used in lieu of a rectangular plate, the following evaluation may be used.



- *Unit load on bolt circle, w.*

$$w = \frac{P_E}{\pi B_c}$$

- *Edge distance from point of load, h_p.*

$$h_p = \frac{B_C - t_L}{2}$$

- *Bending moment, M.*

$$M = wh_p$$

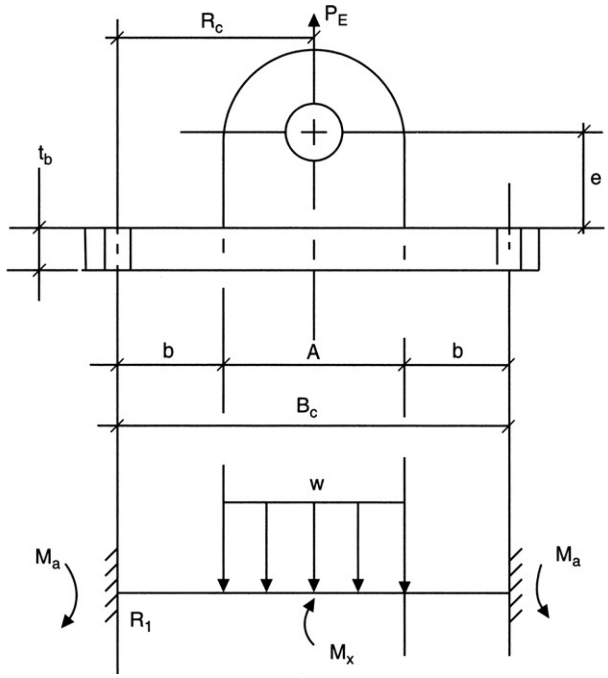
- *Bending stress, σ_b.*

$$\sigma_b = \frac{6M}{t_b^2}$$

- *Check bolting same as rectangular flange.*

Design of Lug Base Plate

(From R. J. Roark, *Formulas for Stress and Strain*, McGraw-Hill Book Co, 4th Edition, Table III, Case 34.)



- *Uniform load, w.*

$$w = \frac{P_E}{A}$$

- *End reaction, R_1.*

$$R_1 = \frac{wA}{2}$$

- *Edge moment, M_a.*

$$M_a = \frac{wA}{24B_c}\left[\frac{24R_c^3}{B_c} - \frac{6(b+A)A^2}{B_c} + \frac{3A^3}{B_c} + 4A^2 - 24R_c^2\right]$$

- *Moment at midspan, M_x.*

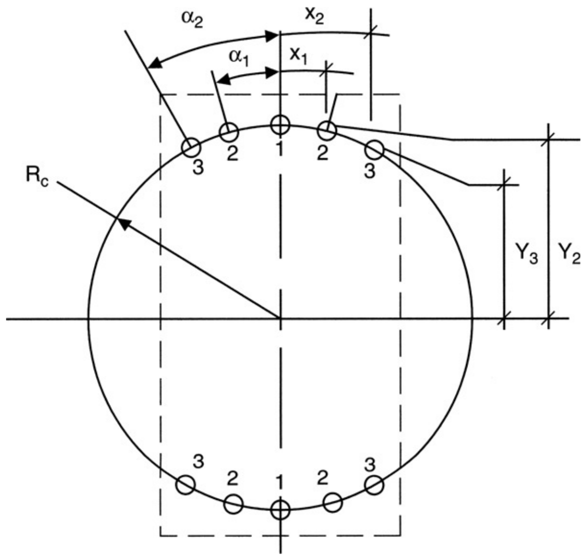
$$M_x = M_a + R_1R_c - \frac{wA}{2}\left[\frac{(R_c-b)^2}{A}\right]$$

- *Thickness required, t_b.*

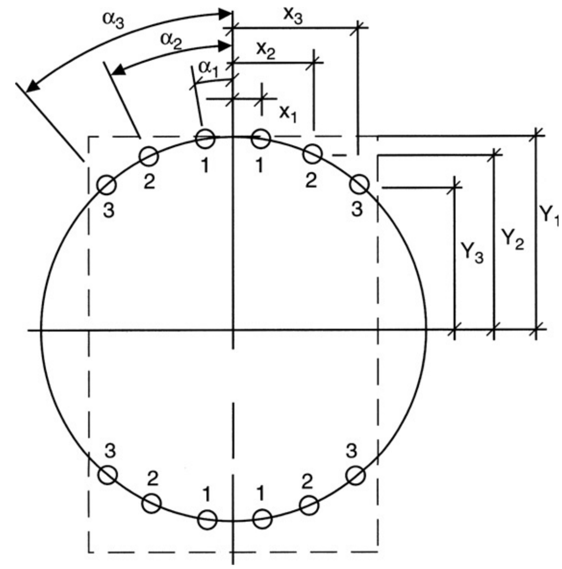
$$t_b = \sqrt{\frac{6M_x}{GF_b}}$$

Check of Bolts

Case 1: Bolts on Centerline

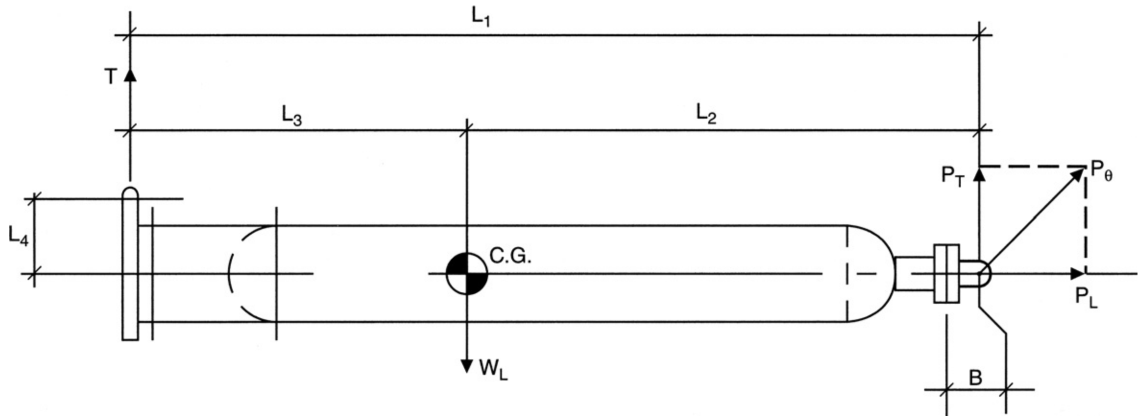


Case 2: Bolts Straddle Centerline



Bolt	1	2	3	4	5	Bolt	1	2	3	4	5
α_n						α_n					
x_n						x_n					
y_n						y_n					
N_b						N_b					
M_u						M_u					
M_a						M_a					
F_n						F_n					
σ_T						σ_T					
F_s						F_s					

Sample Problem: Top Flange Lug



Given

$L_1 = 90$ ft
$L_2 = 50$ ft
$L_3 = 40$ ft
$L_4 = 9.5$ ft
F_y bolting $= 75$ ksi
F_y lug $= 36$ ksi
F_y flange $= 36$ ksi
$F_s = 0.4(36) = 14.4$ ksi
$F_T = 0.6(36) = 21.6$ ksi
$F_b = 0.66(36) = 23.76$ ksi
$W_L = 1200$ kips
$B_c = 54$ in.
$R_c = 27$ in.
$B = 22$ in.
$t_b = 6$ in.
$t_L = 6$ in.
$t_f = 11$ in.
$D_1 = 9$ in.
$D_2 = 8$ in.
Bolt size $=$ 3-1/4-8 UNC
$A_b = 8.3$ in.^{2}
$A_s = 7.686$ in.^{2}
$T_b = 592$ kips
$S_u = 110$ ksi
$e = 16$ in.
$G = 40$ in.
$A = 24$ in.
$h_D = 9.5$ in.
$b = 0.5(B_c - A)$

Results

P_T max $= 537$ kips @ $\theta = 10°$
P_L max $=1200$ kips @ $\theta = 90°$
P_E max $=1277$ kips @ $\theta = 40°$
σ_T bolt, max $= 20.11$ ksi ≤ 40 ksi
τ bolt, max $= 6.98$ ksi ≤ 10.77 ksi

Step 1: Determine loads.

	Angle of Lift, Degrees									
θ	**0**	**10**	**20**	**30**	**40**	**50**	**60**	**70**	**80**	**90**
T_θ	666	654	642	629	613	592	564	517	417	0
P_θ	534	546	558	571	587	608	636	683	783	1200
P_T	534	537	525	494	450	391	318	234	136	0
P_L	0	95	191	286	377	465	551	642	771	1200
w_1	0	3.96	7.96	11.92	15.71	19.38	22.96	26.75	32.13	50
w_2	44.5	44.75	43.75	41.16	37.5	32.58	26.5	19.5	11.33	0
w	44.5	48.71	51.71	53.08	53.21	51.96	49.46	46.25	43.46	50
P_E	1068	1169	1241	1274	1277	1247	1187	1110	1043	1200
M_1	11,748	11,814	11,550	10,868	9900	8602	6996	5148	2992	0
f_s, bolts (10)	53.4	53.7	52.5	49.4	45	39.1	31.8	23.4	13.6	0
f_s, bolts (12)	44.5	44.75	43.75	41.16	37.5	32.6	26.5	19.5	11.33	0
T, bolts (10)	6.94	6.98	6.83	6.42	5.85	5.08	4.13	3.04	1.77	0
T, bolts (12)	5.79	5.82	5.69	5.35	4.88	4.24	3.44	2.53	1.47	0

Step 2: Check bolts for tension load.

Case 1: N = (10) Bolts						**Case 2: N = (12) Bolts**				
α_n	0	15	30			7.5	22.5	37.5		
$\cos \alpha_n$	1	0.965	0.866			0.991	0.923	0.793		
X_n	0	7	13.5			3.52	10.33	16.44		
N_b	2	4	4			4	4	4		
M_u	0	27.05	46.76	$\Sigma = 73.81$		13.95	38.13	52.15	$\Sigma = 104.22$	
M_a	0	4329	7484	$\Sigma = 11{,}814$		1581	4322	5911	$\Sigma = 11{,}814$	
F_n	0	154.6	138.6			112.3	104.6	89.9		
σ_T		20.11	18.03			14.61	13.61	11.7		
F_s		10.77	11.21			11.93	12.13	12.53		

1.0 Check Lug

a. Shear at pin hole:

- *Area required, A_r.*

$$A_r = \frac{P_E}{F_S} = \frac{1277}{14.4} = 88.68 \text{ in.}^2$$

- *Area available at pin hole, A_a.*

$$A_a = (At_L) - (D_1 t_L) = (24 \cdot 6) - (9 \cdot 6) = 90 \text{ in.}^2$$

b. Bending of lug due to M_3:

- *Maximum moment, M_3.*

$$M_3 = P_T e = 537(16) = 8592 \text{ in.} - \text{kips}$$

- *Section modulus, Z.*

$$Z = \frac{t_L A^2}{6} = \frac{(6 \cdot 24^2)}{6} = 576 \text{ in.}^3$$

- *Bending stress, lug, σ_b.*

$$\sigma_b = \frac{M_3}{Z} = \frac{8592}{576} = 14.91 \text{ ksi}$$

- *Thickness required, t_L.*

$$t_L = \frac{6M}{F_b A^2} = \frac{6 \cdot 8592}{23.76(24^2)} = 3.76 \text{ in.}$$

c. Bearing at pin hole:

- *Bearing required at pin hole, A_r.*

$$A_r = \frac{P_E}{F_P} = \frac{1277}{32.4} = 39.41 \text{ in.}^2$$

- *Bearing available, A_a.*

$$A_a = D_2 t_L = 8 \cdot 6 = 48 \text{ in.}^2$$

2.0 Check Lug Base Plate

- *Uniform load, w.*

$$w = \frac{P_E}{A} = \frac{1277}{24} = 53.2 \frac{\text{kips}}{\text{in.}}$$

- *End reaction, R_1.*

$$R_1 = \frac{P_E}{2} = \frac{1277}{2} = 638.5 \text{ kips}$$

- *Edge moment, M_a.*

$$M_a = \frac{wA}{24B_c}\left[\frac{24R_c^3}{B_c} - \frac{6(b+A)A^2}{B_c} + \frac{3A^3}{B_c} + 4A^2 - 24R_c^2\right]$$

$$M_a = 0.985(8748 - 2496 + 768 + 2304 - 17,496)$$
$$= -8049 \text{ in.} - \text{kips}$$

- *Moment at mid, M_x.*

$$M_x = M_a + R_1 R_c - \frac{wA}{2}\left[\frac{(R_c - b)^2}{A}\right]$$

$$M_x = -8049 + 17,240 - 3831$$
$$= 5360 \text{ in.} - \text{kips}$$

- *Section modulus, Z.*

$$Z = \frac{(t_b^2 G)}{6} = \frac{(6^2 \cdot 40)}{6} = 240 \text{ in.}^3$$

- *Bending stress, σ_b.*

$$\sigma_b = \frac{M_x}{Z} = \frac{5360}{240} = 22.33 \text{ ksi}$$

- *Allowable bending stress, F_b.*

$$F_b = 0.66F_y = 0.66(36) = 23.76 \text{ ksi}$$

3.0 Check of Vessel Flange

- *Unit load, w.*

$$w = \frac{P_E}{\pi B_c} = \frac{1277}{\pi 54} = 7.52 \frac{\text{kips}}{\text{in.}}$$

- *Bending moment, M_b.*

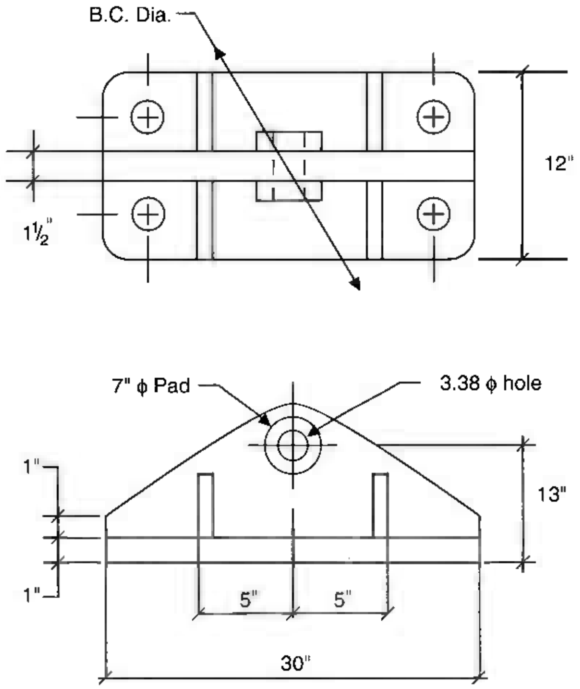
$$M_b = wh_D = 7.52(9.2) = 69.25 \text{ in.} - \text{kips}$$

- *Bending stress, σ_b.*

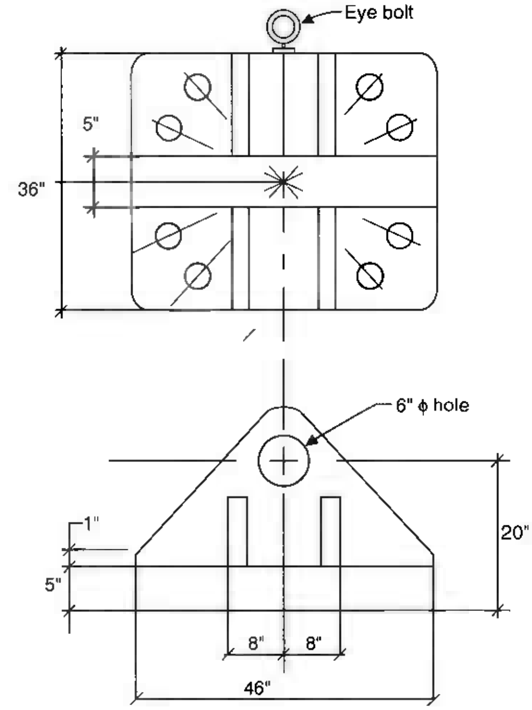
$$\sigma_b = \frac{6M_b}{t_f^2} = \frac{[6(69.25)]}{11.25^2} = 3.28 \text{ ksi}$$

Top Flange Lugs—Alternate Construction

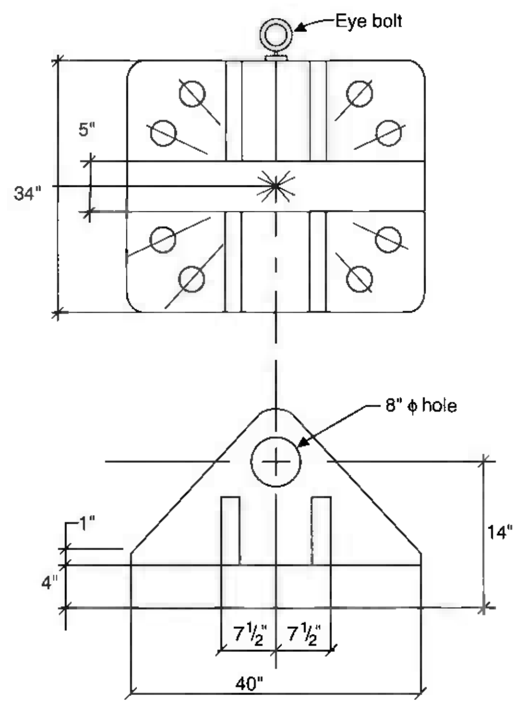
0-Ton Capacity



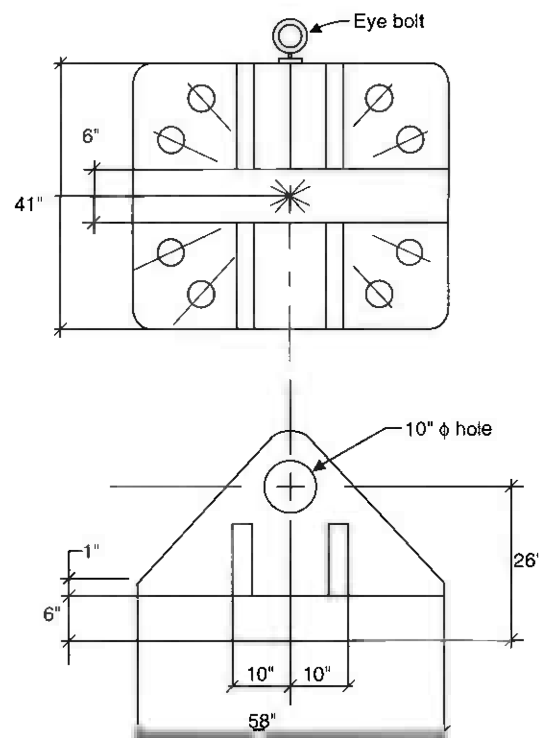
400-Ton Capacity



200-Ton Capacity

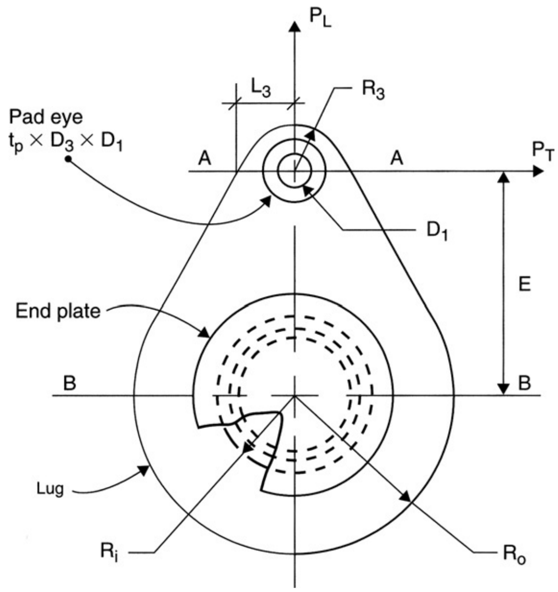


600-Ton Capacity

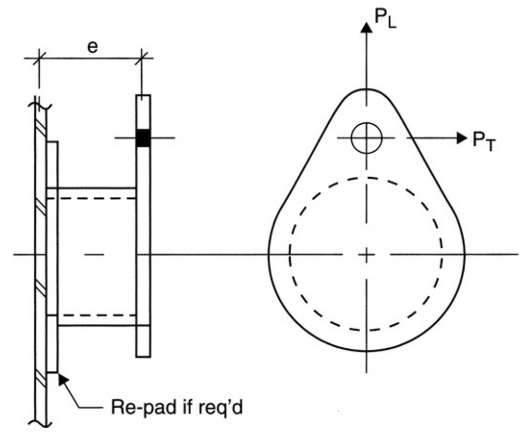


Procedure 10-8: Design of Trunnions

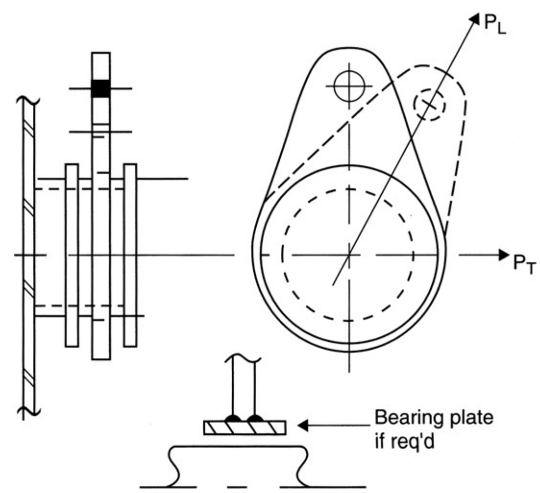
Lug Dimensions



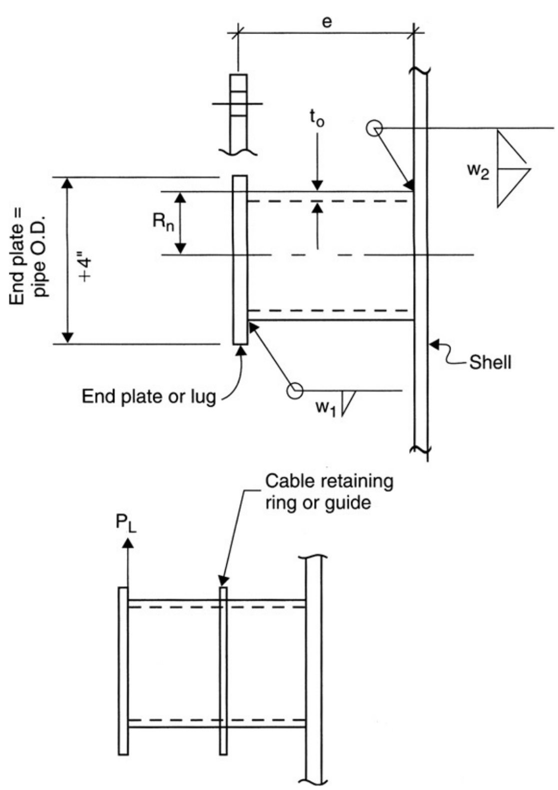
Type 1: Trunnion and Fixed Lug



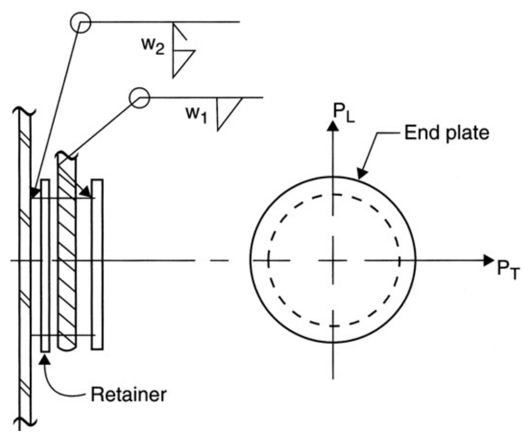
Type 2: Trunnion and Rotating Lug



Dimensions for Trunnion



Type 3: Trunnion Only



Type 1: Trunnion and Fixed Lug

There are four checks to be performed:

1. Check lug.
2. Check trunnion.
3. Check welds.
4. Check vessel shell.

Check Lug

Transverse (vessel horizontal)

$$M = P_T E \quad \text{and} \quad Z = \frac{4R_o^2 t_L}{6}$$

Therefore,

$$t_L = \frac{1.5 P_T E}{R_o^2 F_b}$$

Longitudinal (vessel vertical)

- *Cross-sectional area at pin hole, A_p.*

$$A_p = 21_3 t_L + 2t_p(D_3 - D_1)$$

- *Cross-sectional area at top of lug, A_n.*

$$A_n = t_L\left(R_T - \frac{D_1}{2}\right) + 2t_p\left(\frac{D_3 - D_1}{2}\right)$$

- *Shear stress, τ.*

$$\tau = \frac{P_L}{A_P} \quad \text{or} \quad \tau = \frac{P_L}{A_n}$$

- *Pin bearing stress, σ_p.*

$$\sigma_p = \frac{P_L}{D_2(t_L + 2t_p)}$$

Check Trunnion

- *Longitudinal moment, M_L (vessel vertical).*

$$M_L = P_L e$$

- *Torsional moment, M_T (vessel horizontal).*

$$M_T = P_T E$$

- *Bending stress, σ_b.*

$$\sigma_b = \frac{M_L}{Z}$$

- *Torsional shear stress, τ_T.*

$$\tau_T = \frac{M_T}{2\pi R_n t_o}$$

Check Welds

- *Section modulus of weld, S_w.*

$$S_w = \pi R_n^2$$

- *Polar moment of inertia, J_w.*

$$J_w = 2\pi R_n^3$$

- *Shear stress in weld due to bending moment, f_s.*

$$f_s = \frac{M_L}{S_w}$$

- *Torsional shear stress in weld, τ_T.*

$$\tau_T = \frac{M_T R_n}{J_w}$$

- *Size of welds required, w_1 and w_2.*

$w_1 >$ thickness of end plate
$w_2 =$ width of combined groove and fillet welds

$$w_2 = \frac{f_s}{F_s} > \frac{3}{8}\text{ in.}$$

Type 2: Trunnion and Rotating Lug

- *Net section at Section A-A, A_p.*

$$A_p = 21_3 t_L + 2t_p(D_3 - D_1)$$

- *Shear stress at pin hole*, τ.

$$\tau = \frac{P_L}{A_p}$$

- *Net section at* Section B-*B*, A_n.

$$A_n = 2t_L(R_o - R_i)$$

- *Shear stress at trunnion*, τ.

$$\tau = \frac{P_L}{A_n}$$

- *Minimum bearing contact angle for lug at trunnion*, θ_B.

$$\theta_B = \frac{(15.9P_L)}{R_n t_L F_p}$$

- *Pin hole bearing stress*, σ_p.

$$\sigma_p = \frac{P_L}{D_3(t_L + 2t_p)}$$

Check Welds

- *Longitudinal moment*, M_L (*vessel vertical*).

$$M_L = P_L e$$

- *Section modulus of weld*, S_w.

$$S_w = \pi R_n^2$$

- *Shear stress in weld due to bending moment*, f_s.

$$f_s = \frac{M_L}{S_w}$$

- *Size of welds required*, w_1 *and* w_2.

$w_1 >$ thickness of end plate
$w_2 =$ width of combined groove and fillet welds

$$w_2 = \frac{f_s}{F_s} > \frac{3}{8}\text{ in.}$$

Type 3: Trunnion Only

Vessel Vertical

- *Longitudinal moment*, M_L.

$$M_L = P_L e$$

- *Bending stress in trunnion*, σ_b.

$$\sigma_b = \frac{M_L}{Z}$$

Vessel Horizontal

- *Circumferential moment*, M_c.

$$M_c = P_T e$$

- *Bending stress in trunnion*, σ_b.

$$\sigma_b = \frac{M_c}{Z}$$

Check Welds

- *Longitudinal moment*, M_L (*vessel vertical*).

$$M_L = P_L e$$

- *Section modulus of weld*, S_w.

$$S_w = \pi R_n^2$$

- *Shear stress in weld due to bending moment*, f_s.

$$f_s = \frac{M_L}{S_w}$$

- *Size of welds required*, w_1 *and* w_2.

$w_1 >$ thickness of end plate
$w_2 =$ width of combined groove and fillet welds

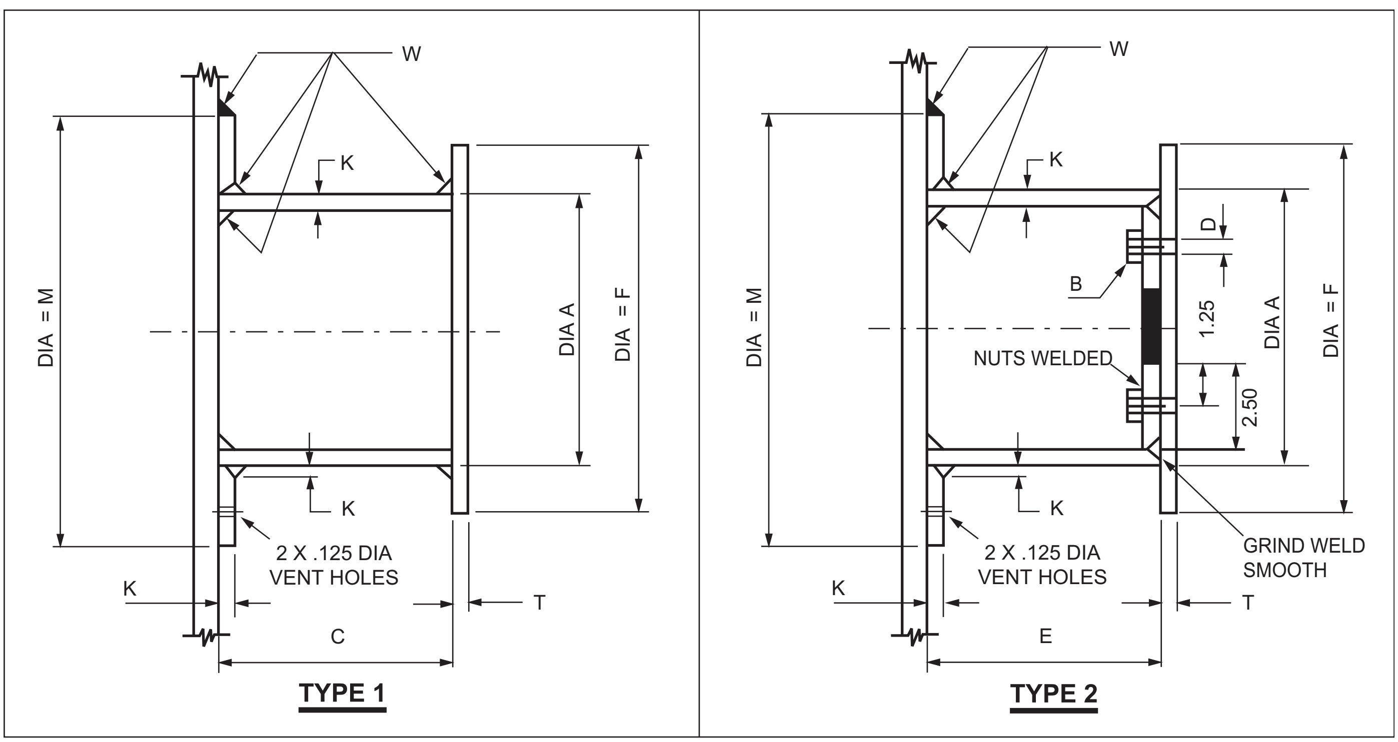
$$w_2 = \frac{f_s}{F_s} > \frac{3}{8}\text{ in.}$$

Table 10-13
Dimensions of trunnions

	Allowable Load, Tons	Pipe Size "A"	C	D	E	F	M	N	T	W	Weight, Lbs
1	0-5	4" Std	4	-	-	8.5	8	-	0.5	0.25	30
2	5-10	6" Std	5	-	-	10.5	12	-	0.5	0.25	55
3	10-15	8" Std	6	0.625	8	12.5	15	4	0.5	0.375	90
4	15-25	10" Std	6	0.625	8	15	18	4	0.75	0.375	150
5	25-35	12" Sch 80	6	0.75	8	17	21	4	0.75	0.375	240
6	35-45	12" Sch 80	6	0.75	8	17	21	4	0.75	0.5	240
7	45-60	14" Sch 80	6	0.75	8	18	24	8	1	0.5	375
8	60-75	14" Sch 80	7	0.875	8	18	24	8	1	0.625	400
9	75-100	16" Sch 80	7	0.875	8	25	27	8	1	0.625	625
10	100-125	16" Sch 80	7	0.875	8	25	27	8	1.125	0.875	660
11	125-150	18" Sch 80	7	0.875	8	27	30	12	1.125	0.875	850
12	150-200	18" Sch 80	8	0.875	8	27	30	12	1.125	1	875
13	200-250	20" Sch 80	8	0.875	10	30	36	16	1.25	1	1000
14	300	24" Sch 80	8	1	10	34	36	16	1.25	1.125	1440
15	400	24" Sch 80	8	1	10	34	40	20	1.375	1.125	1675
16	500	30" × 1.25"	10	1.25	12	42	48	20	1.375	1.375	2400
17	600	36" × 1.25"	10	1.25	12	48	60	24	1.5	1.375	3600

Notes:

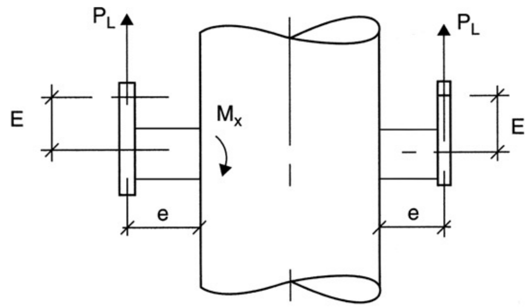
1. Do not use re-pads for cyclic service
2. B = D - .125
3. K = Pipe Wall Thk
4. Dimensions are given for reference only. All loadings and stress shall be checked prior to use.

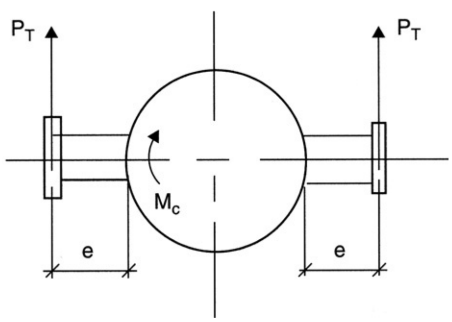


Procedure 10-9: Local Loads in Shell Due to Erection Forces

Trunnions

Fixed Lug Trunnion





- *Maximum longitudinal moment, M_x.*

$$M_x = P_L e$$

- *Maximum circumferential moment, M_c.*

$$M_c = P_T e$$

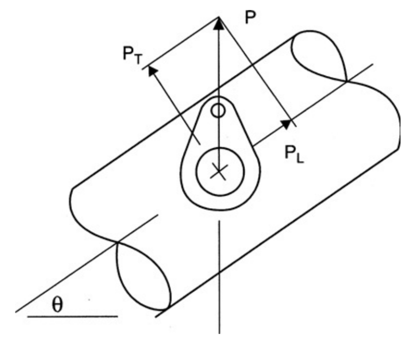
- *Maximum torsional moment, M_T.*

$$M_T = P_T E$$

- *Loads for any given lift angle, θ.*

$$P_L = 0.5P \sin\theta$$
$$P_T = 0.5P \cos\theta$$

Rotating Trunnion



- *Maximum longitudinal moment, M_x.*

$$M_x = P_L e$$

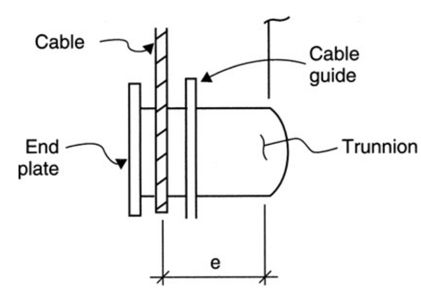
- *Maximum circumferential moment, M_c.*

$$M_c = P_T e$$

- *Loads for any given lift angle, θ.*

$$P_L = 0.5P \sin\theta$$
$$P_T = 0.5P \cos\theta$$

Trunnion—No Lug



- *Maximum longitudinal moment, M_x.*

$$M_x = P_L e$$

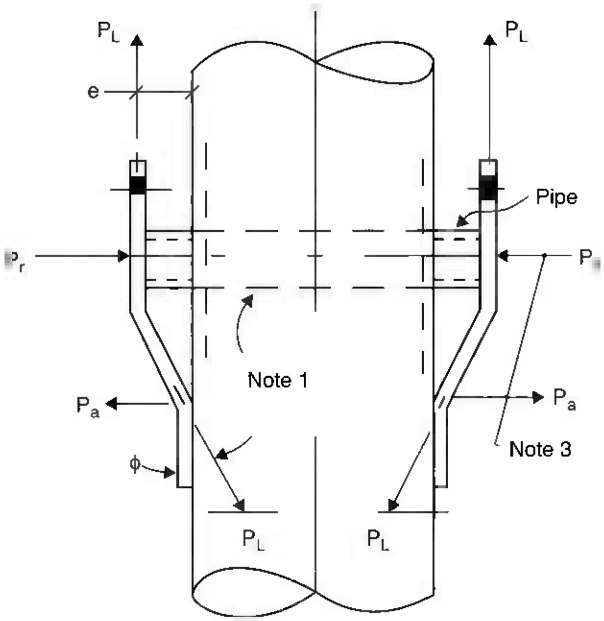
- *Maximum circumferential moment, M_c.*

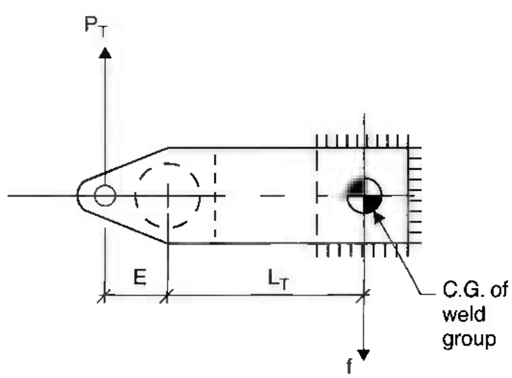
$$M_c = P_T e$$

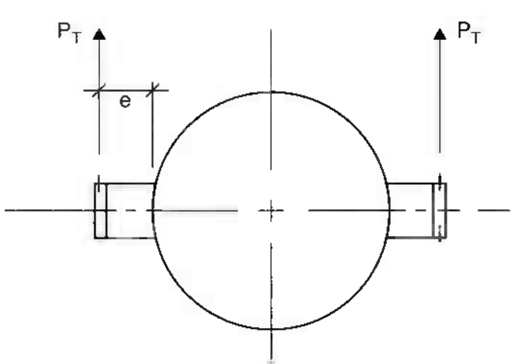
- *Loads for any given lift angle, θ.*

$$P_L = 0.5P \sin\theta$$
$$P_T = 0.5P \cos\theta$$

Side Lugs







Notes:

1. Optional internal pipe. Remove after erection.
2. Radial load, P_r, is the axial load in the internal pipe stiffener if used in lieu of radial load in shell.
3. Circumferential ring stiffeners are optional at these elevations.

- *Circumferential moment,* M_c.

$$M_c = P_T e$$

- *Longitudinal moment,* M_x.

$$M_x = P_L e$$

- *Load on weld group, f.*

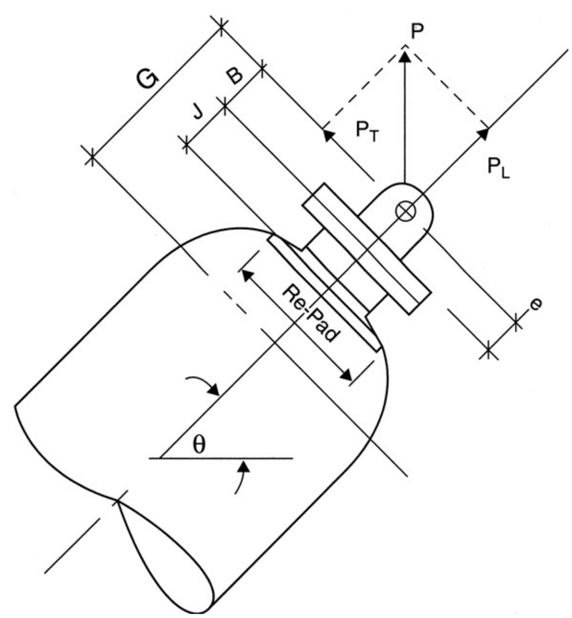
$$f = \frac{P_T E}{L_T}$$

- *Radial loads,* P_r *and* P_a.

$$P_r = P_L e$$

$$P_a = P_L \sin \phi$$

Top Flange Lug



- *Loads,* P_T *and* P_L.

$$P_T = P \cos \theta$$

$$P_L = P \sin \theta$$

- *Moment on flange, M.*

$M = P_T B$

- *Moment on head, M.*

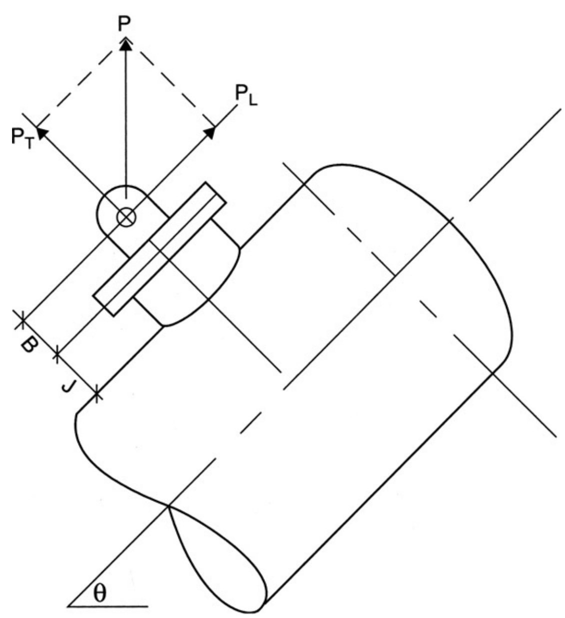
$M = P_T(B + J)$

- *Moment on vessel, M.*

$M = P_T G$

- *Radial load on head and nozzle* $= P_L$.

Side Flange Lug



- *Loads,* P_T *and* P_L.

$PL = P \cos \theta$

$PT = P \sin \theta$

- *Moment on flange, M.*

$M = P_L B$

- *Longitudinal moment on shell,* M_x.

$M = P_T(B + J)$

- *Radial load on shell and nozzle* $= P_T$.

Procedure 10-10: Miscellaneous

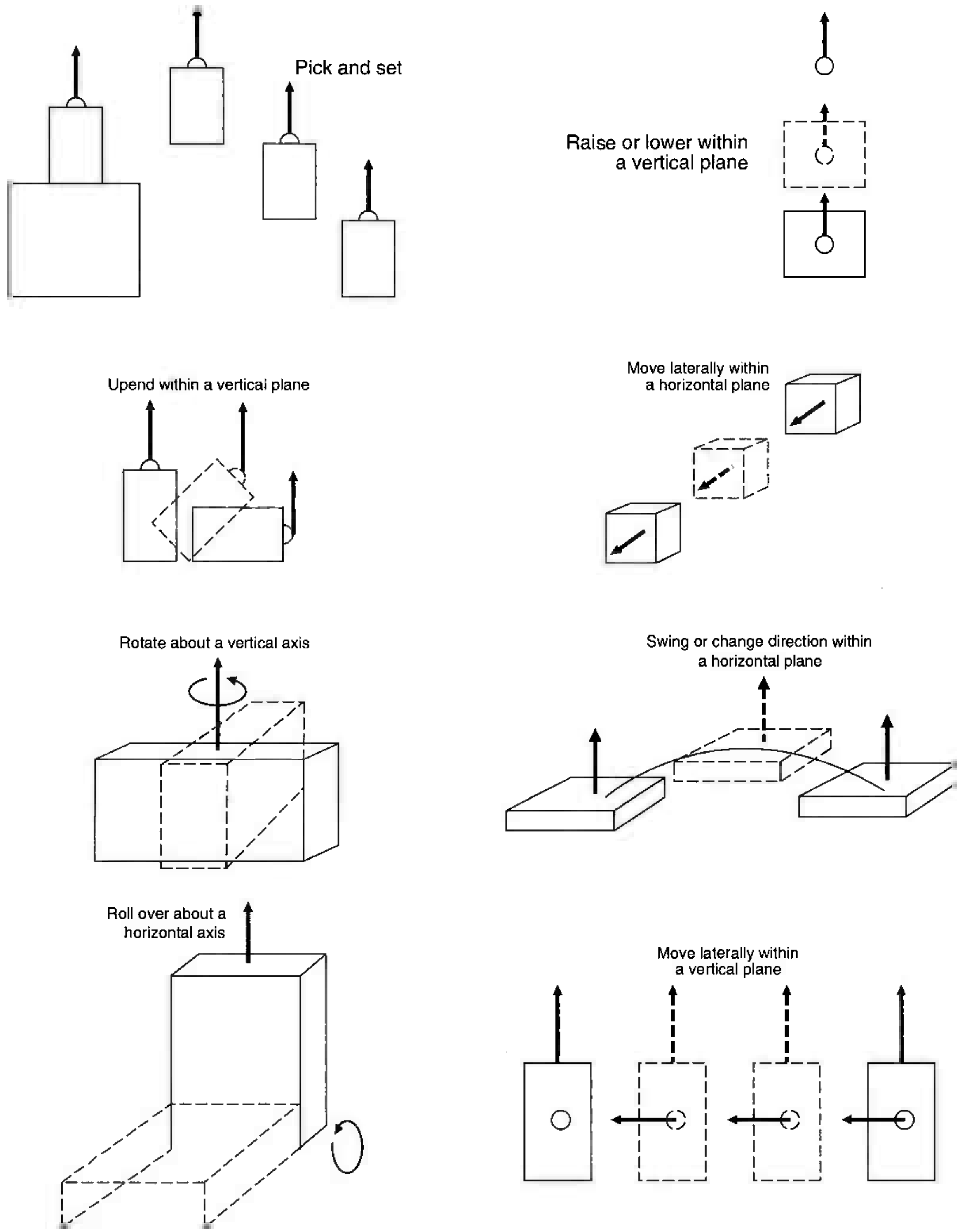


Figure 10-7. Fundamental handling operations. *Reprinted by permission of the Babcock and Wilcox Company, a McDermott Company.*

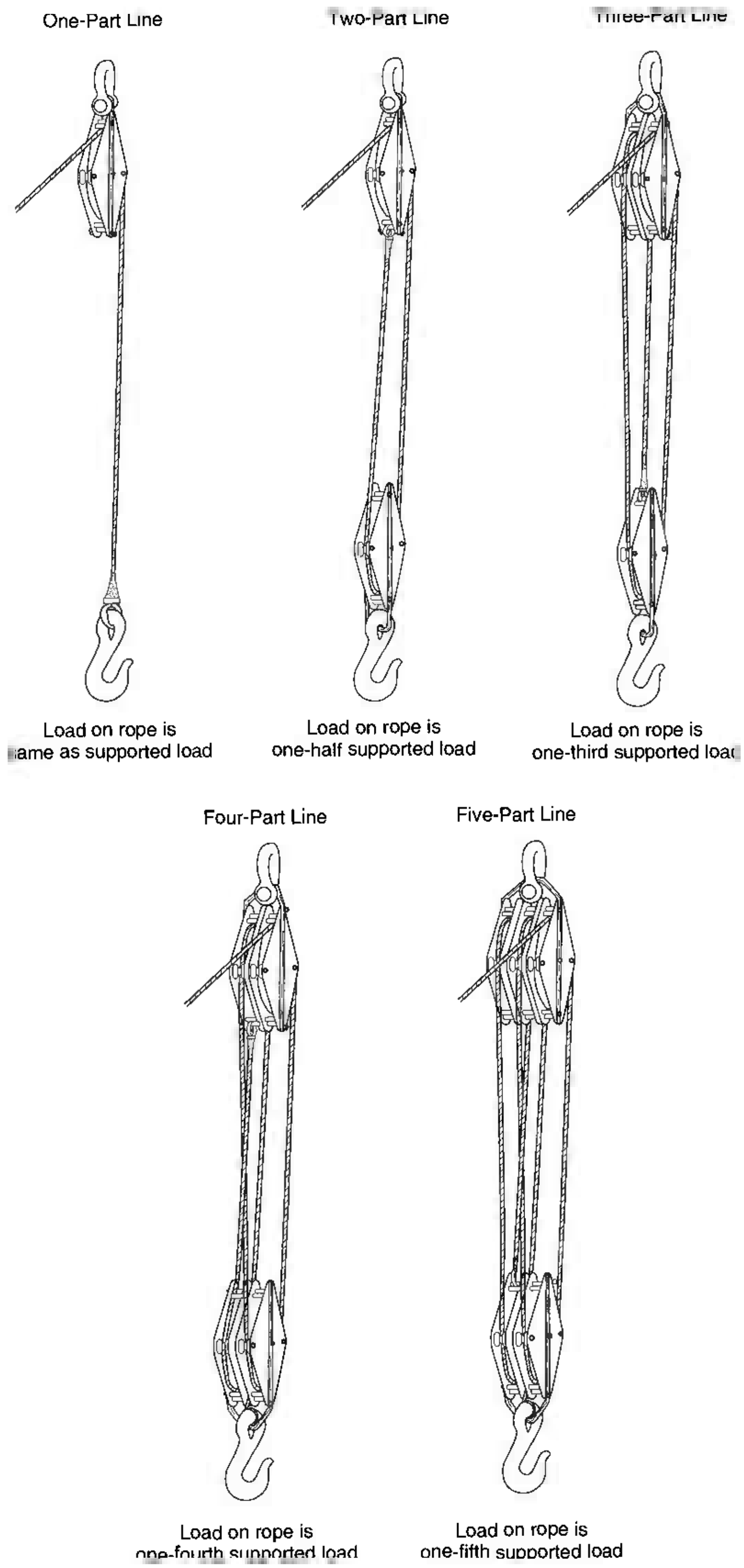
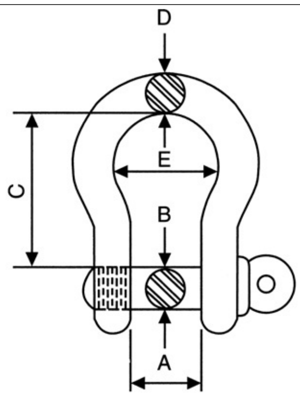
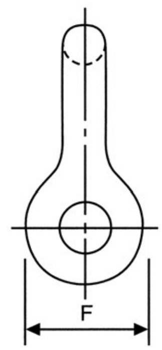


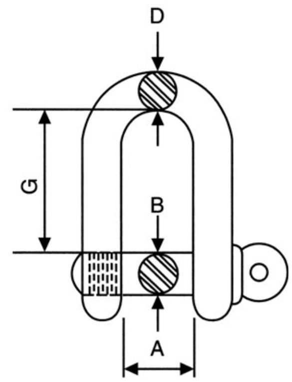
Figure 10-8. Loads on wire rope for various sheave configurations.

Table 10-14
Forged Steel Shackles



Anchor Shackle Screw Pin





Chain Shackle Screw Pin

Dimensions in Inches											
Size D (in.)	Safe Load (lb)	D (min)	A	Tolerance A Dim.	B	B (min)	C	G	Tolerance C and G Dim.	E	F
¼	475	7/32	15/32	± 1/16	5/16	9/32	1⅛	⅞	±1/16	¾	11/16
⅜	1,050	11/32	21/32	± 1/16	7/16	25/64	1 7/16	1¼	±⅛	1	31/32
7/16	1,450	25/64	23/32	± 1/16	½	29/64	1 11/16	1 7/16	±⅛	1⅛	1 1/16
½	1,900	29/64	13/16	± 1/16	⅝	9/16	1⅞	1⅝	±⅛	1⅜	1 5/16
⅝	2,950	9/16	1 1/16	± 1/16	¾	43/64	2 13/32	2	±⅛	1⅝	1 9/16
¾	4,250	43/64	1¼	± 1/16	⅞	25/32	2 27/32	2⅜	±¼	2	1⅞
⅞	5,750	25/32	1 7/16	± 1/16	1	57/64	3 5/16	2 13/16	±¼	2¼	2⅛
1	7,550	57/64	1 11/16	± 1/16	1⅛	1 1/32	3¾	3 3/16	±¼	2½	2⅜
1⅛	8,900	1 1/32	1 27/16	±⅛	1¼	1 7/64	4¼	3 9/16	±¼	2¾	2⅝
1¼	11,000	1 7/64	2 1/32	±⅛	1⅜	1 15/64	4 11/16	3 15/16	±¼	3⅛	3
1⅜	13,300	1 15/64	2¼	±⅛	1½	1 11/32	5¼	4 7/16	±¼	3½	3 5/16
1½	15,600	1 11/32	2⅜	±⅛	1⅝	1 29/64	5¾	4⅞	±¼	3¾	3⅝
1¾	21,500	1 35/64	2⅞	±⅛	2	1 25/32	7	5¾	±¼	4¼	4⅛
2	28,100	1 25/32	3¼	±⅛	2¼	2 1/64	7¾	6¾	±¼	5¼	5
2¼	36,000	2 1/64	3¾	±⅛	2½	2 15/64	9¼	7⅛	±¾	5½	5¼
2½	45,100	2 15/64	4⅛	±⅛	2¾	2 15/32	10½	8	±¾	6¼	6
3	64,700	2 11/16	5	±⅛	3¼	2 29/32	13	11½	±¾	6¾	6½

Notes:

For shackles with safe loads greater than the maximum shown, use Crosby–Laughlin (The Crosby Group, Div. of American Hoist & Derrick Co, Tulsa, OK 74101), Skookum (Skookum Co., Inc., Portland, OR 97203), or equal with an ultimate strength at least 5 times the safe working load.
Allowable loads are lower than OSHA requirements tabulated in Section 1926.251, Table H-19.

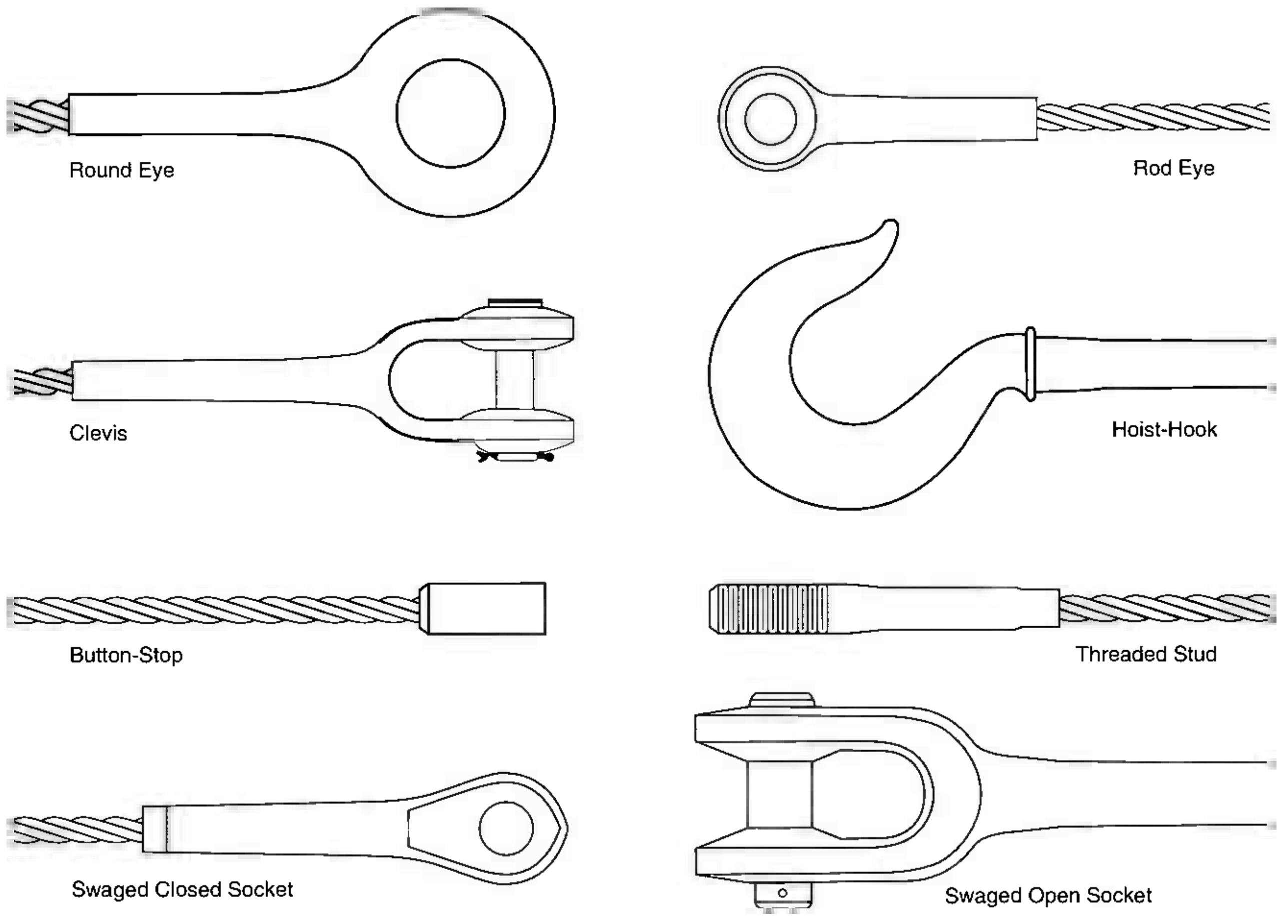


Figure 10-9. Wire rope end configurations.

Table 10-15
Material transportation and lifting

Material-Handling System	Description	Capacity t (t_m)
Site Transport: Flatbed trailers	Bed dimension 8 × 40 ft (*2.4 × 12.2m*)—deck height 60 in. (*1524 mm*) used to transport materials from storage to staging area.	20 (*18*)
Extendable trailers	Bed dimension up to 8 × 60 ft (*2.4 × 18.3m*)—deck height 60 in. (*1524 mm*) used to transport materials from storage to staging area.	15 (*14*)
Lowboy and dropdeck	Bed dimension up to 8 × 40 ft (*2.4 × 12.2m*)—deck height of 24 in. (*610 mm*) used to transport materials from storage to staging area.	60 (*54*)
Crawler transporter	Specially designed mechanism for handling heavy loads; Lampson crawler transporter, for an example of the Lampson design.	700 (*635*)
Straddle carrier	Mobile design to transport structural steel, piping, and other assorted items; straddle carrier, for an example of this design.	30 (*27*)
Rail	Track utilized to transport materials to installed location. Continuous track allows material in-stallation directly from delivery car.	as designed
Roller and track	Steel machinery rollers located relative to component center of gravity handle the load. Rollers traverse the web of a channel welded to top flange of structural member below.	2000 (*1814*)
Plate and slide	Sliding steel plates. Coefficient of friction—0.4 steel on steel, 0.09 greased steel on steel, 0.04 Teflon on steel. Sliding plate transport for movement of 1200 *t* (*1089 t_m*) vessel.	as designed
Air bearings or air pallets	Utilizes film of air between flexible diaphragm and flat horizontal surface. Air flow 3 to 200 ft^3/min (*0.001 to 0.09 m^3/s*). 1 lb (*4.5N*) lateral force per 1000 lb (*454 kg*) vertical load.	75 (*68*)
High line	Taut cable guideway anchored between two points and fitted with inverted sheave and hook.	5 (*4.5*)
Lifting: Chain hoist	Chain operated geared hoist for manual load handling capability. Standard lift heights 8 to 12 ft (*2.4 to 3.7m*).	25 (*23*)
Hydraulic rough terrain cranes	Telescopic boom mounted on rubber tired self-propelled carrier.	90 (*82*)
Hydraulic truck cranes	Telescopic boom mounted on rubber tired independent carrier.	450 (*408*)
Lattice boom truck cranes	Lattice boom mounted on rubber tired independent carrier.	800 (*726*)
Lattice boom crawler cranes	Lattice boom mounted on self-propelled crawlers.	2200 (*1996*)
Fixed position crawler cranes	Lattice boom mounted on self-propelled crawlers and equipped with specifically designed attachments and counterweights.	750 (*680*)
Tower gantry cranes	Tower mounted lattice boom gantry for operation above work site.	230 (*209*)
Guy derrick	Boom mounted to a mast supported by wire rope guys. Attached to existing building steel with load lines operated from independent hoist. Swing angle 360 deg (*6.28 rad*).	600 (*544*)
Chicago boom	Boom mounted to existing structure which acts as mast, and to which is attached boom topping lift and pivoting boom support bracket. Load lines operated from independent hoist. Swing angle from 180 to 270 deg (*3.14 to 4.71 rad*).	function of support structure
Stiff leg derrick	Boom attached to mast supported by two rigid diagonal legs and horizontal sills. Horizontal angle between each leg and sill combination ranges from 60 to 90 deg (*1.05 to 1.57 rad*); swing angle from 270 to 300 deg (*4.71 to 5.24 rad*).	700 (*635*)
Monorail	High capacity load blocks suspended from trolleys which traverse monorail beams suspended from boiler support steel. Provides capability to lift and move loads within boiler cavity.	400 (*363*)
Jacking systems	Custom designed hydraulic or mechanical system for high capacity special lifts.	as specified

Reprinted by permission of the Babcock and Wilcox Company, a McDermott Company.

Notes

1. This procedure is for the design of the vessel and the lifting attachments only. It is not intended to define rigging or crane requirements.
2. Lifting attachments may remain on the vessel after erection unless there is some process- or interference-related issue that would necessitate their removal.
3. Load and impact factors must be used for moving loads. It is recommended that a 25% impact factor and a minimum load factor of 1.5 be used. The combined load and impact factor should be 1.5–2.0.
4. Allowable stress compression should be $0.6F_y$ for structural attachments and ASME Factor "B" times 1.33 for the vessel shell.
5. Vessel shipping orientation should be established such that a line through the lifting lugs is parallel to grade if possible. This prevents the vessel from having to be "rolled" to the correct orientation for loading and offloading operations.
6. If a spreader beam is not used, the minimum sling angle shall be 30° from the horizontal position. At 30°, the tension in each sling is equal to the total design load. Thus a load factor of 2 is mandatory for these cases. This requires that each lug be designed for the full load.
7. Vessels should never be lifted by a nozzle or other small attachments unless specifically designed to do so.
8. All local loads in vessel shell or head resulting from loadings imposed during erection of the vessel shall be analyzed using a suitable local load procedure.
9. Tailing attachment shall be designed such that they may be unbolted without having to get under the load while it is suspended. As an alternative, the vessel must be set down at grade before a person can get under the base ring to unbolt the tailing beam. Be advised that the base and skirt may not be designed for point support if cribbing is used to build up the base for access.
10. A tailing lug, as opposed to a tailing beam, allows the load to be disconnected from the vessel without a person's getting under a suspended load to unhook.
11. This procedure assumes that the pin diameter is no less than $\frac{1}{16}$ in. less than the hole diameter. If the pin diameter is greater than $\frac{1}{16}$ in. smaller than the hole diameter, then the bearing stresses in the lug at the contact point are increased dramatically due to the stress concentration effect.
12. Internal struts in the skirt or base plate are required only if the base/skirt configuration is overstressed.
13. If bearing or shear stresses are exceeded in the lug, add pad eyes.
14. Trunnions may be used as tailing devices as long as the resulting local loads in the skirt are analyzed.
15. Do not use less than Schedule 40 pipe for trunnions.
16. Specific notes for trunnions:
    a. Type 1, fixed lug: Normal use but generally for small to medium vessels (less than 100 tons).
    b. Type 2, rotating lug: Best use is when multiple vessels are to be lifted with the same lug. The lug may be removed by removing the end plate and sliding the lug off. Then the lug is reinstalled on the next vessel. For heavier loads, an internal sleeve should be attached to the lug to increase the bearing area on the trunnion.
    c. Type 3, trunnion only: No size limitation or weight limitation. The cable and trunnions should be lubricated prior to lifting to prevent the cables from binding. The bend radius of the cables may govern the diameter of the trunnion. Check with erection contractor.

11
Materials

11.1. Types of Materials

Steel & Alloys

1.0 Ferrous >50% Fe
- A. Carbon Steel
  - Low <.3% C
  - Medium .35 – .55% C
  - High .6 – 1.5% C
- B. Low Alloy (Chrome and Cr-Mo)
  - <5% addition of alloy, i.e. Mn, Ni, Cr, etc.
  - Examples are Cr-Mo, C-1/2 Mo, etc.
- C. High Alloy
  - >10% addition of alloy compostion
  - Stainless Steels
    - a. Ferritic
    - b. Martensitic
    - c. Austenitic
      1. Stabilized
      2. Low Carbon
      3. High carbon
    - d. Duplex
- D. Cast Iron >1.5% C
  1. Ductile
  2. Malleable
  3. White
  4. Gray

2.0 Non-Ferrous <50% Fe
- Copper
- Nickel
- Aluminum
- Magnesium
- Titanium
- Brass
- Bronze

2.1 Common names for Non-Ferrous Alloys
- Inconel
- Incoloy
- Monel
- Carpenter 20
- Cupronickel
- Aluminum Bronze

Table 11-1
Material specifications

Matl	Plate	Pipe	Tube	Bar	Figs	Fittings
Nick 200	SB-162	SB-161	SB-163	SB-160	SB-160	B-366-WPN
Monel 400	SB-127	SB-165	SB-163	SB-164	SB-164	B-366-WPNC
Inco 600	SB-443	SB-444	SB-163	SB-446	SB-166	B-366-WPNCI
Incoloy 825	SB-424	SB-423	SB-163	SB-425	SB-408	–
Hast C-4	SB-575	SB-619	SB-622	SB-574	SB-622	–
Carp 20	SB-463	SB-464	SB-468	SB-473	SB-462	–
SST	SA-240	SA-312	SA-213	SA-276	SA-182	SA-403
			SA-269	SA-479		
CS	SA-516	SA-106-B	SA-179	SA-306	SA-105	SA-234-WPB
Titanium	SB-265	SB-337	SB-338	SB-348	SB-381	SB-363
Alum 6061	SB-209	SB-241	SB-210	SB-211	SB-247	
Chrome	SA-387	SA-335	SA-213	SA-739	SA-182	SA-234
T-405 12 Cr			Use SST Designations			
T-410 13 Cr			Use SST Designations			
T-430 17 Cr			Use SST Designations			
3½ Ni	SA-203-D	SA-333-3	SA-334-3		SA-350-LF3	SA-420-WPL3
Hast G-30	SB-582	SB-622	SB-622	SB-581	SB-581	
Nitronic 50 (UNS) 20910	SA-240-XM19	SA-312-XM19	SA-213-XM19	SA-479-XM19	SA-182-XM19	SA-403-XM19
Inco 800	SB-409	SB-407	SB-407	SB-408	SB-408	B-366

- Hastelloy
- Waspalloy
- E-brite

2.2 Clad materials: A combination of base material and some alloy cladding or weld overlay (WOL)

- Methods of manufacture;
  1. Roll bonded
  2. Explosion bonded
  3. Weld overlay (WOL)
- ASTM/ASME material specifications for clad plate are as follows;

  SA-263: Specification for stainless chromium steel - clad plate

  SA-264: Specification for stainless chromium Nickel steel-clad plate

  SA-265: Specification for nickel and nickel base alloy-clad steel plate

- Terms and definitions for cladding/weld overlay (WOL);

  Integral Bonded Cladding: A composite material produced by roll bonding or explosion welding (EXW).

  Weld Overlay (WOL): Produced by weld depositing a dissimilar material on a base metal surface.

  Back Cladding: A localized WOL operation between two integrally clad or weld overlayed sections. This process is also known as "clad restoration".

  Total Depth: Total thickness of WOL or back cladding

  Effective Depth: The thickness of WOL or back cladding having the specified chemical composition

  Sleeve Lining: The installation of a cylindrical sleeve of alloy material on the inside diameter of a connection.

Table 11-2
Properties of commonly used pressure vessel materials

	Material	Mechanical Properties			Chemical Properties %							
	Designation	UTS	YS	Elong	C max	Si	Mn	P max	S max	Ni	Cr	Mo
PLATE	SA-36	58–80	36	23	0.25			0.04	0.05			
	SA-285-C	55–75	30	27	0.28		0.9	0.035	0.04			
	SA-515-55	55–75	30	27	0.20	0.15–0.40	0.9	0.035	0.04			
	SA-515-60	60–80	32	25	0.24	0.15–0.40	0.9	0.035	0.04			
	SA-515-70	70–90	38	21	0.31	0.15–0.40	1.2	0.035	0.04			
	SA-516-55	55–75	30	27	0.18	0.15–0.40	0.6–0.9	0.035	0.04			
	SA-516-60	60–80	32	23	0.21	0.15–0.40	0.6–0 9	0.035	0.04			
	SA-516-70	70–90	38	21	0.27	0.15–0.40	0.85–1.2	0.035	0.04			
	SA-204-B	70–90	40	21	0.2	0.15–0.40	0.9	0.035	0.04			0.45–0.60
	SA-302-B	80–100	50	18	0 2	0.15–0.40	1.15–1.5	0.035	0.04			0.45–0.60
	SA-387-11-2	75–100	45	22	0.17	0.5–0.8	0.40–0 65	0 035	0.04			
	SA-203-A	65–85	37	19	0.17	0.15–0.40	0.7	0 035	0.04	2.1–2 5		
	SA-203-D	65–85	37	19	0.17	0.15–0.40	0.7	0.035	0.04	3.25–3.75		
	SA-240-304	75	30	40	0.08	1.0	2.0	0.045	0.03	8–10.5	18–20	
	SA-240-316	75	30	40	0.08	1.0	2.0	0.045	0.03	10–14	16–18	2–3
PIPE	SA-53	60	35		0 3		1.2	0.05	0.06			
	SA-106-B	60	35	30	0.3	0.1	0.29–1.06	0 048	0.058			
	SA-333-3	65	35	30	0.19	0.18–0.37	0.31–0.64	0.05	0.05	3.18–3 82		
	SA-333-6	60	35	30	0.3	0.1	0.29–1.06	0.048	0.058			
	SA-335-P1	55	30	30	0.1–0.2	0.1–0.5	0.3–0.8	0.045	0.045			0.44–0.65
	SA-335-P11	60	30	30	0.15	0.50–1.0	0.3–0.6	0.03	0.03		1–1.5	0.44–0.65
	SA-312-304	75	30	35	0.08	0.75	2	0.04	0.03	8–11	18–20	
	SA-312-316	75	30	35	0.08	0.75	2	0.04	0.03	11–14	16–18	2–3
FORGINGS	SA-105	70	36	22	0.35	0.35	0.60–1.05	0.04	0.05			
	SA-350-LF2	70–95	36	22	0 3	0.15–0.30	1 35	0.035	0.04			
	SA-350-LF3	70–95	37.5	22	0 2	0.20–0.35	0.9	0.035	0.04	3.25–3.75		
	SA-182-F1	70	40	25	0.28	0.15–0.35	0.8–0 9	0 045	0.045			0.44–0.65
	SA-182-F11	70	40	20	0.1–0.2	0.50–1.0	0.3–0.8	0.04	0.04		1.0–1.5	0 44–0.65
	SA-182-304	75	30	30	0.08	1.0	2	0.04	0.03	8–11	18–20	
	SA-182-316	75	30	30	0.08	1.0	2	0.04	0.03	10–14	16–18	
MISC.	SA-234-WPB	60	35		0.3	0.1	0.29–1.06	0.05	0.058			
	SA-193-B7	125	105	16	0.37–0.49	0.15–0.35	0.85–1.1	0.04	0.04		0.75–1.2	0.15–0.25
	SA-193-B16	125	105	18	0.36–0.44	0.15–0.35	0.45–0.70	0.04	0.04		0.80–1.15	0.50–0.65
	SA-320-L7	125	105	16	0.38–0.46	0.15–0.35	0 75–1.0	0 035	0.04		0.80–1.1	0.15–0.25

Table 11-3
Stainless steel

TYPE		CHEMICAL COMPOSITION				MECH PROPERTIES			REMARKS
		Cr	Ni	C (Max)	Other (Max)	Tensile KSI	Yield KSI	Elong (%)	
AUSTENITIC	201	16-18	3.5-5.5	0.15	Mn 5.5-7.5 N .25	115	55	55	Low Ni Version of 301
	202	17-19	4-6	0.15	Mn 7.5-10 N .25	105	55	55	Low Ni Version of 302
	203EZ	16-18	5-6	0.07	Mn 5.5-6.5 Cu 1.75-2.15 S.18-.35	90	40	50	Free machining grade
	216	19.75	6	0.08	Mn 8.25 Mo 2.5 N.37	100	55	45	Similar to 316 w/ better high strength properties
	301	16-18	6-8	0.15	Mn 2 Si 1	110	40	60	High work hardening, structural grade
	302	17-19	8-10	0.15	Mn 2 Si 1	90	40	55	General purpose SST
	303	17-19	8-10	0.15	Mn 2 Si 1 P .2 Mo .6 S .15 Min	90	35	50	Free machining version of 302
	303SE	17-19	8-10	0.15	Mn 2 Si 1 Se .15 Min P .2 S .06	90	35	50	Better surface finish than 303
	303PB	17-19	8-10	0.15	Mn 2 Si 1 Mo .6 Pb .12-.3	90	35	50	Leaded version of 303 for high volume machining
	304	18-20	8-10	0.08	Mn 2 Si 1	85	35	55	Low carbon variation of 302
	304L	18-20	8-10	0.03	Mn 2 Si 1	80	30	55	Extra low carbon 304
	305	17-19	10-13	0.12	Mn 2 Si 1	85	35	55	Low work hardening. Good spinning and deep drawing
	309	22-24	12-15	0.2	Mn 2 Si 1	95	40	45	High temperature applications
	310	24-26	19-22	0.25	Mn 2 Si 1.5	95	45	50	Excel corrosion resistance
	316	16-18	10-14	0.08	Mn 2 Si 1 Mo 2-3	85	35	60	Best corrosion resistance of standard SST's. High temperature strength
	316L	16-18	10-14	0.03	Mn 2 Si 1Mo 2-3	78	30	55	Extra low carbon 316
	317	18-20	11-15	0.08	Mn 2 Si 1Mo 3-4	90	40	50	316 w/ better creep resistance
	321	17-19	9-12	0.08	Mn 2 Si 1 P.04 S .03 Ti 5 x C Min	85	35	55	Stabilized w/ Ti
	347	17-19	9-12	0.08	Mn 2 Si 1Cb-Ta 10 x C Min	95	40	50	Stabilized w/ Cb
MARTENSITIC	403	11.5-13	0	0.15	Mn 1 Si .5	75	40	35	Turbine quality for highly stresed parts
	405	11.5-14.5	0	0.08	Mn 1 Si 1 Al .1-.3	70	40	30	Variation of 410 w/ limited hardenability
	410	11.5-12.5	0	0.15	Mn 1 Si 1	75	40	35	Low cost, general purpose, heat treatable SST
	414	11.5-13.5	1.25-2.5	0.15	Mn .6 P .04 S .03 Si .5 V .15 Al .015	110	85	18	Modified 410 w/ 2% Ni
	416	12-14	0	0.15	Mn 1.25 Si 1 S .15 Min	75	40	30	Free machining version of T-410
	440A	16-18	0	0.75	Mo .75 Mn 1 Si 1	105	60	20	High Carbon variety. Can be heat treated for high strength.
	440B	16-18	0	0.95	Mo .75 Mn 1 Si 1	107	62	18	
	440C	16-18	0	1.2	Mo .75 Mn 1 Si 1	110	65	14	
	501	4-6	0	0.15	Mo .45-.65 Mn .3-.6 Si .5	70	30	28	Economical. Cr & Mo added for mild corrosion resistance and elevated temperature service
	502	4-6	0	0.1	Mo .45-.65 Mn .3-.6 Si .5	70	30	30	
FERRITIC	409	10.5-11.75	0.5	0.08	Mn 1 P .045 S .045 Si 1 Ti 6xC Min	70	40	30	Economical. Easy to fabricate.
	430	14-18	0	0.12	Mn 1 Si 1	75	45	30	Most widely used non-hardenable type. Good heat resistance & mechanical properties
	430F	14-18	0	0.12	Mn 1.25 Si 1 P .06 S .15 Min	80	55	25	Free machining version of 430
	446	23-27	0	0.2	Mn 1.5 Si 1 N .25	80	50	25	Good oxidation resistance in sulfuric atmospheres
SPECIAL	A286	15	26	0.05	Mn 1.4 Si .4 Mo 1.25 Ti 2.15 V .3 Al .2 B ..03 Fe Bal	93	27	48	An austenitic alloy w/ exceptionally high strength at elevated temperatures
	AM350	16.5	4.3	0.1	Mn .8 Si .25 Mo 2.75 N .1 Fe Bal	145	60	40	These austenitic varieties are heat treatable, superior corrosion resistance. Provided in annealed condition and age hardened.
	AM355	15.5	4.3	0.13	Mn .95 Si .25 Mo 2.75 N .1 Fe Bal	186	55	29.5	
	PH13-8Mo	12.25-13.25	7.5-8.5	0.05	Mn .1 P .01 S .008 Si .1 Al 1.35 Mo 2.5 N .01	160	100	0.15	
	15-5PH	14-15.5	3.5-5.5	0.07	Mn 1 P .04 S .03 Si 1 Cu 4.5 Cb Ta .45	150	110	10	
	PH15-7Mo	14-16	6.5-7.75	0.09	Mn 1 Si 1 P .04 S .03 Al .75-1.5 Mo 2-3	130	55	35	
	17-4PH	15.5-17.5	3-5	0.07	Mn 1 Si 1 Cu 3-5 Fe Bal	150	110	10	
	17-7PH	16-18	6.5-7.75	0.09	Mn 1 Si 1 Al .75-1.5 Fe Bal	130	40	35	

TABLE 11-4
Nickel Alloys

Item	Common Name	% Ni	Other Major Ingredients
1	Nickel 200	99.5	None
2	Duranickel 301	94	4.5% Al, .5% Ti
3	Monel 400	66	32% Cu
4	Inconel 600	78	15% Cr, 7% Fe
5	Incoloy 800	32.5	46% Fe, 21% Cr
6	Carpenter 20	32-38	20% Cr, 3% Mo, 3% Cu
7	Hastelloy C-22	55	21% Cr, 13% Mo, 2.5% Co
8	Hastelloy G-30	36	30% Cr, 5% Mo, 5% Co, 15% Fe
9	Hastelloy C-276	55	15.5% Cr, 16% Mo, 2.5% Co, 4% W
10	E-Brite	73	26% Cr, 1% Mo
11	Waspalloy	55	14% Co, 19% Cr, 4.3% Mo, 2% Fe, 3% Ti
12	Nimonic 75	75	19% Cr
13	Nimonic 90	57	19.5% Cr, 16.5% Co
14	Cupronickel	30	70% Cu

11.2. Properties of Materials

1. Physical Properties
   a. Density
   b. Melting point
   c. Boiling point
   d. Specific heat
   e. Latent heat of fusion
   f. Thermal conductivity
   g. Anisotropy
   h. Thermal expansion
   i. Viscosity
   j. Solidus/liquidus
   k. Vapor pressure
   l. Specific gravity
2. Mechanical Properties
   a. Brittle fracture
   b. Fatigue
   c. Endurance limit
   d. Creep
   e. Hardness
   f. Abrasion resistance
   g. Impact resistance
   h. Strength/ductility/toughness
      1. Tensile
      2. Yield
      3. Elongation
      4. Proportional limit
      5. Modulus of elasticity
      6. Compression
      7. Shear
      8. Reduction of area
3. Fabrication Properties
   a. Weldability
   b. Machinability
   c. Formability
   d. Hardenability
4. Special Properties
   a. Coefficient of friction
   b. Water absorption
   c. Thixotropy
   d. Light transmittance, reflectivity, emissivity
5. Chemical Properties
   a. Corrosion properties
   b. Galvanic series
   c. Acid/base
   d. Graphitization
   e. Alloy chemistry
6. Electrical Properties
   a. Resistivity
   b. Conductivity
   c. Magnetic
   d. Piezoelectric
   e. Photoconductivity
   f. Thermoelectric
   g. Dielectric strength

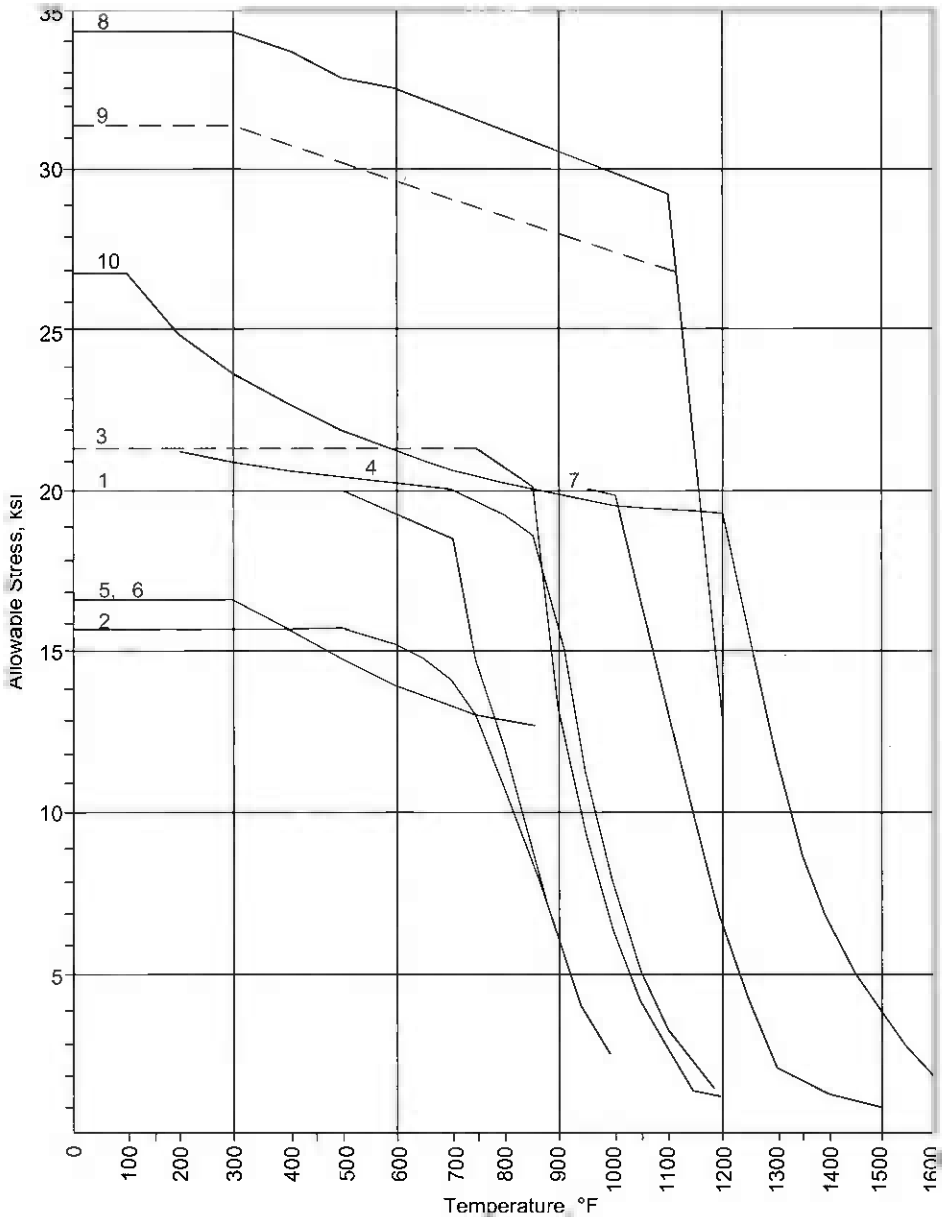


Figure 11-1. Allowable stresses per ASME Section II, Part D

Materials

1. SA-516-70, SA-515-70
2. SA-285-C
3. SA-387-11-2
4. SA-387-22-2
5. SA-240-316L, High Stress
6. SA-240-304L, High Stress
7. SB-409-800
8. SB-443-625-1, High Stress
9. SB-443-625-1, Low Stress
10. SB-443-625-2, High Stress

Table 11-5
Material properties

Material		Temperature, °F														
		70	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
	E	29.4	28.8	28.3	27.9	27.3	26.5	25.5	24.2	22.5	20.4	18.0				
Carbon steel C ≤ 0.3%	α	6.4	6.7	6.9	7.1	7.3	7.4	7.6	7.8	7.9	8.1	8.2	8.3	8.4		
	F_y	38.0	34.8	33.6	32.5	31.0	29.1	27.2	25.5	24.0	22.6					
	E	29.6	29.0	28.5	28.0	27.4	26.9	26.2	25.6	24.8	23.9	23.0	21.8	20.5	18.9	
Chrome moly 1/2 - 2Cr steels	α	6.4	6.7	6.9	7.1	7.3	7.4	7.6	7.8	7.9	8.1	8.2	8.3	8.4		
	F_y	45.0	41.5	39.5	37.9	36.5	35.3	34.0	32.5	30.6	28.2					
	E	30.6	29.9	29.4	28.8	28.3	27.7	27.0	26.3	25.6	24.7	23.7	22.5	21.1	19.4	
Chrome moly 2 1/4 - 3Cr steels	α	6.4	6.7	6.9	7.1	7.3	7.4	7.6	7.8	7.9	8.1	8.2	8.3	8.4		
	F_y	45.0	41.2	39.4	38.1	37.3	36.5	35.6	34.3	32.4	29.7					
	E	31.0	30.3	29.7	29.2	28.6	28.1	27.5	26.9	26.2	25.4	24.4	23.3	22.0	20.5	
Chrome moly 5 - 9Cr steels	α	6.4	6.7	6.9	7.0	7.1	7.2	7.2	7.3	7.4	7.5	7.6	7.6	7.7	7.8	
	F_y	45.0	40.7	39.2	38.7	38.4	37.8	36.7	34.6	31.6	27.7					
	E	29.2	28.4	27.9	27.3	26.8	26.2	25.5	24.5	23.2	21.5	19.2	16.5			
High chrome moly 12 - 17 steels	α	5.9	6.2	6.3	6.4	6.5	6.5	6.6	6.7	6.7	6.8	6.8	6.9	6.9	7.0	7.0
	F_y	25.0	23.0	22.2	21.8	21.5	21.0	20.2	18.9	16.9	14.4					
	E	28.3	27.5	27.0	26.4	25.9	25.3	24.8	24.1	23.5	22.8	22.0	21.2	20.3	19.2	18.1
Austenitic stainless steel	α	8.5	8.9	9.2	9.5	9.7	9.9	10.0	10.1	10.2	10.3	10.4	10.6	10.7	10.8	10.8
	F_y	30.0	25.0	22.4	20.7	19.4	18.4	17.6	16.9	16.2	15.5					
	E	31.0	30.3	29.9	29.4	29.0	28.6	28.1	27.6	27.1	26.5	25.9	25.3	24.6	23.9	23.1
Inconel 600	α	6.8	7.1	7.3	7.5	7.6	7.8	7.9	8.0	8.2	8.3	8.4	8.6	8.7	8.9	9.0
	F_y	35.0	32.0	31.2	30.7	30.3	29.9	29.4	28.7	27.3						
	E	28.5	27.9	27.5	27.1	26.7	26.2	25.8	25.4	24.9	24.4	23.8	23.2	22.6	21.9	21.2
Incoloy 800	α	7.9	8.4	8.6	8.8	8.9	9.0	9.1	9.2	9.3	9.4	9.5	9.6	9.7	9.8	10.0
	F_y	30.0	27.7	26.6	25.8	25.1	24.5	23.8	23.2	22.7	22.1					

Notes:

1. Nomenclature and units are as follows:
   - E = modulus of elasticity, 10^6 psi
   - α = mean coefficient of thermal expansion from 70°F, in/in/°F
   - F_y = minimum specified yield strength, ksi
2. α and F_y are for the following grades:
   - Carbon steel C ≤ 0.3% = SA-516-70
   - Chrome moly 1/2 - 2Cr steels = SA-387-11-2
   - Chrome moly 2 1/4 - 3Cr steels = SA-387-22-2
   - Chrome moly 5 - 9Cr steels = SA-387-5-2
   - High chrome moly 12 - 17 steels = SA-240-405
   - Austenitic stainless steel = SA-240-304
   - Inconel 600 = SB-168
   - Incoloy 800 = SB-409

Source: ASME Section II, Part D

Table 11-6
Values of yield strength, ksi

Material	Temp 100°	200°	300°	400°	500°	600°	700°	800°	900°	1000°
SA-285c, SA-516-55	30	27.4	26.6	25.7	24.3	22.2	21.6	20.0	19.1	16.7
SA-516-60	32	29.2	28.4	27.5	26	23.7	23.1	21.3	20.3	17.8
SA-516-65	35	31.9	31	30	28.3	25.9	25.2	23.3	22.2	19.5
SA-105	36	32.9	31.9	30.9	29.2	26.6	26	24.0	22.9	20.1
SA-516-70	38	34.8	33.6	33.5	31.0	29.1	27.2	25.5	24.0	22.6
SA-204-B (C – ½Mo)	40	37.6	36.1	34.8	33.8	32.7	31.5	30.0	27.9	25.2
SA-302-B (Mn – Mo)	50	47.2	45.3	44.5	43.2	42.0	40.6	38.8	34.9	28.4
SA-387-2-2 (½Cr – Mo)	–	–	–	–	–	–	–	–	–	–
SA-387-12-2 (1Cr – ½Mo)	40	36.9	35.1	33.7	32.5	31.4	30.2	28.8	27.2	25.0
SA-387-11-2 (1¼Cr – ½Mo)	45	41.5	39.5	37.9	36.5	35.3	34.0	32.4	30.6	28.2
SA-387-22-2 (2¼Cr – 1Mo)	45	41.2	39.4	38.1	37.3	36.5	35.6	34.3	32.4	29.7
T-405 (13Cr)	25	23.0	22.2	21.8	21.5	21.1	20.2	18.9	16.9	14.4
T-410/T-430 (13/17Cr)	30	27.6	26.6	26.1	25.8	25.3	24.2	22.7	20.3	17.2
T-304 SST	30	25.0	22.4	20.7	19.4	18.4	17.6	16.9	16.2	15.5
T-304L SST	25	21.4	19.2	17.5	16.4	15.5	15.0	14.5	14.0	13.3
T-316 SST	30	25.9	23.4	21.4	20.0	18.9	18.2	17.6	17.3	17.0
T-321 SST	30	27.0	24.8	23.0	21.5	20.3	19.4	18.8	18.4	18.8
T-347 SST	30	27.6	25.7	24.0	22.6	21.5	20.7	20.3	20.2	20.1
SA-203-B (2½ Ni)	40	–	–	–	–	–	–	–	–	–
SA-203-D (3½ Ni)	37	–	–	–	–	–	–	–	–	–
Nickel 200	15	15	15	15	15	15	–	–	–	–
Monel 400	28	24.7	22.4	22.2	22.2	22.2	22.2	21.4	–	–
Inconel 600	35	32.7	31.0	29.9	28.8	27.9	27	26.1	–	–
Incoloy 800	30	27.6	26.0	25.0	24.1	23.9	23.5	23.0	–	–

Source: ASME Section II, Part D

Table 11-7
Material selection guide

<table>
<tr><th colspan="2">Design Temperature, °F</th><th>Material</th><th>Plate</th><th>Pipe</th><th>Forgings</th><th>Fittings</th><th>Bolting</th></tr>
<tr><td rowspan="2">Cryogenic</td><td>−425 to −321</td><td>Stainless steel</td><td>SA-240-304, 304L, 347, 316, 316L</td><td>SA-312-304, 304L, 347, 316, 316L</td><td>SA-182-304, 304L, 347, 316, 316L</td><td>SA-403-304, 304L, 347, 316, 316L</td><td rowspan="2">SA-320-B8 with SA-194-8</td></tr>
<tr><td>−320 to −151</td><td>9 nickel</td><td>SA-353</td><td>SA-333-8</td><td>SA-522-1</td><td>SA-420-WPL8</td></tr>
<tr><td rowspan="5">Low temperature</td><td>−150 to −76</td><td>3½ nickel</td><td>SA-203-D</td><td rowspan="2">SA-333-3</td><td rowspan="2">SA-350-LF3</td><td rowspan="2">SA-420-WPL3</td><td rowspan="3">SA-320-L7 with SA-194-4</td></tr>
<tr><td>−75 to −51</td><td>2½ nickel</td><td>SA-203-A</td></tr>
<tr><td>−50 to −21</td><td rowspan="4">Carbon steel</td><td>SA-516-55, 60 to SA-20</td><td>SA-333-6</td><td rowspan="2">SA-350-LF2</td><td rowspan="2">SA-420-WPL6</td></tr>
<tr><td>−20 to 4</td><td>SA-516-All</td><td>SA-333-1 or 6</td><td rowspan="6">SA-193-B7 with SA-194-2H</td></tr>
<tr><td>5 to 32</td><td>SA-285-C</td><td rowspan="2">SA-53-B
SA-106-B</td><td rowspan="2">SA-105
SA-181-60,70</td><td rowspan="2">SA-234-WPB</td></tr>
<tr><td>Intermediate</td><td>33 to 60
61 to 775</td><td>SA-516-All
SA-515-All
SA-455-II</td></tr>
<tr><td rowspan="7">Elevated temperature</td><td>776 to 875</td><td>C-½Mo</td><td>SA-204-B</td><td>SA-335-P1</td><td>SA-182-F1</td><td>SA-234-WP1</td></tr>
<tr><td rowspan="2">876 to 1000</td><td>1Cr-½Mo</td><td>SA-387-12-1</td><td>SA-335-P12</td><td>SA-182-F12</td><td>SA-234-WP12</td></tr>
<tr><td>1-¼Cr-½Mo</td><td>SA-387-11-2</td><td>SA-335-P11</td><td>SA-182-F11</td><td>SA-234-WP11</td></tr>
<tr><td>1001 to 1100</td><td>2¼Cr-1Mo</td><td>SA-387-22-1</td><td>SA-335-P22</td><td>SA-182-F22</td><td>SA-234-WP22</td><td>with SA-193-B5 SA-194-3</td></tr>
<tr><td rowspan="2">1101 to 1500</td><td>Stainless steel</td><td>SA-240-347H</td><td>SA-312-347H</td><td>SA-182-347H</td><td>SA-403-347H</td><td rowspan="3">SA-193-BB with SA-194-B</td></tr>
<tr><td>Incoloy</td><td>SB-424</td><td>SB-423</td><td>SB-425</td><td>SB-366</td></tr>
<tr><td>Above 1500</td><td>Inconel</td><td>SB-443</td><td>SB-444</td><td>SB-446</td><td>SB-366</td></tr>
</table>

From Bednar, H H , *Pressure Vessel Design Handbook*, Van Nostrand Reinhold Co , 1981

Note:

1. Material specifications shown are for non-corrosive service only. Actual material selection must take corrosion rates into account.
2. This table should be used as a guideline for material selection based on temperature only. No consideration for various services has been made in the assembly of this table.

11.3. Bolting

Specifications

SA-193: Specification for Alloy steel and Stainless Steel Bolting Material for High-temperature Service
SA-320: Specification for Alloy steel Bolting Materials for Low Temperature Service
SA-540: Specification for Alloy Steel Bolting Materials for Special Applications
SB-637: Specification for Precipitation hardening Nickel Alloy bars, Forging and Forging stock for High Temperature Service

Some other Specifications utilized by the process industry:

SA-307: Carbon steel bolting, low strength
SA-325: Used mainly in structural applications
SA-490: Used mainly in structural applications

Some exotic applications;

1. SA-193-B408 to UNS NO8810	Inco 718
2. SA-193-B8S	Nitronic 60
3. SA-193-XM-19	Nitronic 50
4. SA-540-B22-Class 1:	
5. SB-637-NO7718	Inco 718

Heat treat condition (Class) of SA-193

Class 1: N&T or Q&T and solution treated
Class 1A: Solution treated after finishing
Class 1B: Solution treated for Nitrogen bearing SST
Class 1C: Solution treated for Nitrogen bearing SST
Class 2: Annealed and strain hardened

In general all SST bolting should have the threads formed after heat treatment. However certain grades of Classes 1A, 1B, and 1C are to be solution treated in the final condition.

Bolting

Notes

1. Bolt and thread dimensions shall be in accordance with ANSI B1.1.
2. Nut dimensions shall be a accordance with ANSI B18.2.2.
3. Washer dimensions shall be per ANSI B18.22.1.
4. Mechancial testing of bolting materials shall be conducted in accordance with ASTM A-370.
5. Where practical all threads shall be cut or formed after heat treatment. Heat treatment can be performed after threading if agreed by purchaser. Certain grades of SA-193 are required to be heat treated after threading. Grade B7M is required to be Q&T after threading.
6. Class of Fit;
   a. Class 1: Loose
   b. Class 2: Standard
   c. Class 3: Exceptionally high quality
   d. Class 4: Selective Fit
7. Thread Series:
   a. NC: National Course
   b. NF: National Fine
   c. UNEF: Extra Fine
   d. 8 thread series (typically used in flange bolting above 1 inch diameter)
   e. 12 thread series
   f. 16 thread series

Table 11-8
Bolting materials

Type of Material	Symbol	Material Specification: Bolts	Material Specification: Nuts
Aluminum alloy 2014-T6	AL	B211, TP-2014-T6	B211, TP-2014-T6
AISI T-501(5 Cr)	B5	SA-193-B5	SA-194-3
AISI T-410(12 Cr)	B6	SA-193-B6	SA-194-6
AISI T-4140, 4142, 4145	B7	SA-193-B7	SA-194-2H
304 SS	B8	SA-193-B8	SA-194-8
Cr-Mo-V	B16	SA-193-B16	SA-194-2H
Carbon steel	CS1	SA-307-B	SA-307-B
Carbon steel	CS2	SA-325	SA-325
Copper alloy, CDA 630	CU	CDA 630 to SB-150	CDA 630 to SB-150
Hastelloy C	HC	SB-336 annealed	SB-336 annealed
Hastelloy X	HX	SA-193 to B-435	SA-193 to B-435
AISI T-4140, 4142, 4145	L7	SA-320-L7	SA-194-4
Monel 400	M4	SA-193 to B-164	SA-193 to B-164
Inconel 600	N6	SA-193 to B-166	SA-193 to B-166
Incoloy 800	L8	SA-193 to B-408	SA-193 to B-408
19 Cr – 9 Ni	SS	SA-453 GR 651, CL A	SA-453 GR 651, CL A
321 SS	8T	SA-193-B8T	SA-194-8T
316 SS	8M	SA-193-B8M	SA-194-8M
Nitronic 60	8S	SA-193-B8S	SA-194-8S

Table 11-9
Bolting application

Service		Temperature Range, °F: −121 to −420	−51 to −120	−21 to −50	59 to −20	60 to 399	400 to 649	650 to 849	850 to 999	1000 to 1099	1100 to 1199	1200 to 1499	>1500
Low Temperature	SST	B8	L7	L7									
	ALUM	B8	AL	AL	AL	AL							
	9 Ni	B8	L7	L7	B7	B7							
	3-½ Ni		L7	L7	B7	B7							
	CS			L7	B7	B7							
	Copper			CU	CU	CU	CU						
Intermediate Temperature	C.I.				CS	CS							
	CS				B7	B7	B7						
	Low alloy					B7	B7	B7	B7				
Elevated Temperature	Low alloy					B7	B7	B7	B16	B16	B5		
	321 SS					8T	8T	8T	8T	8T			
	316 SS					8M	8M	8M	8M	8M	8M	8M	
	Corro-sion					M4	M4	M4	M4	N6	N6	L8	L8
	Corro-sion								HC				HX

Table 11-10
Allowable stress for bolts

Material	Spec	Class	Min. Spec. Tensile	Min. Spec. Yield	Size, in.	100	200	300	400	500	600	700	800	900	1000	1100	1200
Carbon steel	Sa-307–B		60			7.00	7.00	7.00	7.00								
Carbon steel	SA-325		105	81		20.20	20.20	20.20	20.20	20.20	20.20						
5Cr-½ Mo	SA-193-B5		100	80	<4	20.00	20.00	20.00	20.00	20.00	20.00	20.00	18.50	10.40	5.60	3.10	1.30
13Cr	SA-193-B6		110	85	<4	21.20	21.20	21.20	21.20	21.20	21.20	21.20	19.50	12.00			
1Cr-⅕Mo	SA-193-B7		125	105	<2.5	25.00	25.00	25.00	25.00	25.00	25.00	25.00	21.00	12.50	4.50		
			115	95	2.5–4	23.00	23.00	23.00	23.00	23.00	23.00	23.00	20.00	12.50	4.50		
			100	75	4–7	18.70	18.70	18.70	18.70	18.70	18.70	18.70	18.00				
18Cr-8Ni	SA-193-B8	2	125	100	<.75	25.00	25.00	25.00	25.00	25.00	25.00	25.00	25.00	25.00	24.10		
			115	80	.75–1	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00		
			105	65	1–1.25	16.30	16.30	16.30	16.30	16.30	16.30	16.30	16.30	16.30	16.30		
			100	50	1.25–1.5	12.50	12.50	12.50	12.50	12.50	12.50	12.50	12.50	12.50	12.50		
1Cr-½Mo-V	SA-193-B16		125	105	<2.5	25.00	25.00	25.00	25.00	25.00	25.00	25.00	25.00	20.50	11.00	2.80	
			110	95	2.5–4	23.00	23.00	23.00	23.00	23.00	23.00	23.00	22.00	18.50	11.00	2.80	
			100	85	4–7	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	16.70	11.00	2.80	
1Cr-½Mn-¼Mo	SA-540-B22	1	165	150	<1.5	33.00	33.00	33.00	33.00	33.00	33.00	33.00					
		2	155	140	<3	31.00	31.00	31.00	31.00	31.00	31.00	31.00					
		3	145	130	<4	29.00	29.00	29.00	29.00	29.00	29.00	29.00					
		4	135	120	<4	27.00	27.00	27.00	27.00	27.00	27.00	27.00					
		5	120	105	<2	24.00	24.00	24.00	24.00	24.00	24.00	24.00					
		5	115	100	2–4	23.00	23.00	23.00	23.00	23.00	23.00	23.00					
1Cr-⅕Mo	SA-320-L7		125	105	<2.5	25.00	25.00	25.00	25.00	25.00	25.00	25.00					
1C-¼ Mo	SA-320-L7A		125	105	<2.5	25.00	25.00	25.00	25.00	25.00	25.00						
1C-1/5 Mo	SA-320-L7M		100	80	<2.5	20.00	20.00	20.00	20.00	20.00	20.00	20.00	18.50	12.50	4.50		
1 ¾Ni-¾ Cr-¼Mo	SA-320-L43		125	105	<4	25.00	25.00	25.00	25.00	25.00	25.00	25.00					

Notes:

1. All values are in ksi.
2. Values per ASME, Section II, Part D.

Table 11-11
Material designation and strength

Material	Bolts SA-193-	Size (dia, in.)	UTS (ksi)	Min Spec Yield (ksi)	Nuts SA-194-
5Cr-½Mo	B5	<4	100	80	3
12Cr(T-410 SS)	B6	<4	110	85	6
1Cr-$^{1}/_{5}$ Mo	B7	<2.5	125	105	2H
1Cr-$^{1}/_{5}$ Mo	B7	2.5 to 4	115	95	2H
1Cr-$^{1}/_{5}$ Mo	B7	4 to 7	100	75	2H
1Cr-$^{1}/_{5}$ Mo	B7M	<2.5	100	80	2H
1Cr-½Mo-V	B16	<2.5	125	105	2H
1Cr-½Mo-V	B16	2.5 to 4	110	95	2H
1Cr-½Mo-V	B16	4 to 7	100	85	2H
304 SS	B8-2	<0.75	125	100	8
304 SS	B8-2	75 to 1	115	80	8
304 SS	B8-2	1 to 1.25	105	65	8
304 SS	B8-2	1.25 to 1.5	100	50	8
316 SS	B8M-2	<0.75	110	95	8M
316 SS	B8M-2	0.75 to 1	100	80	8M
316 SS	B8M-2	1 to 1.25	95	65	8M
316 SS	B8M-2	1.25 to 1.5	90	50	8M
321 SS	B8T-2	<0.75	125	100	8T
321 SS	B8T-2	0.75 to 1	115	80	8T
321 SS	B8T-2	1 to 1.25	105	65	8T
321 SS	B8T-2	1.25 to 1.5	100	50	8T
347 SS	B8C-2	<0.75	125	100	8C
347 SS	B8C-2	75 to 1	115	80	8C
347 SS	B8C-2	1 to 1.25	105	65	8C
347 SS	B8C-2	1.25 to 1.5	100	50	8C
Nitronic 60	B8S	—	95	50	8S
SA-320 (Low Temp)					
304 SS	B8A	—	75	30	8
316 SS	B8MA	—	75	30	8M
321 SS	B8TA	—	75	30	8T
347 SS	B8CA	—	75	30	8C

Table 11.12
Summary of requirements for 100% X-ray and PWHT*

P. No.	GRP. No.	Material Description	PWHT	Temperature °F	100% R.T.
1	1	Carbon steel: SA-36, SA-285-C, SA-515/-516 Grades 55, 60, 65	>1.5 in.	1100°	>1.25 in.
	2	Carbon steel: SA-515/-516 Grade 70, SA-455-I or II	>1.5 in.	1100°	>1.25 in.
3	1	Low alloy: C-½ Mo (SA-204-B)	>.625 in.	1100°	>.75 in.
	2	Low alloy: 1/2 Cr-½ Mo (SA-387-2-2)	>.625 in.	1100°	>.75 in.
	3	Low alloy: Mn-Mo (SA-302-B)	All	1100°	>.75 in.
4	1	Low alloy: 1Cr-½ Mo (SA-387-12-2) 1Cr-½ Mo (SA-387-11-2)	(1)	1100°	>.625 in.
5	1	Low alloy: 2Cr-1Mo (SA-387-22-2) 3Cr-1Mo (SA-387-21-2)	All	1250°	All
	2	Low alloy: 5,7,9Cr-½ Mo	All	1250°	All
6	1	13Cr (410) Martensitic SST	(2)	1250°	(2)
7	1	13Cr (405, 410S) Martensitic SST	(2)	1350°	(2)
	2	17Cr (430) Ferritic SST	All	1350°	(2)
8	1	(304,316,321,347) Austenitic SST	–	1950°	>1.5 in.
	2	(309,310) Austenitic SST	–	1950°	>1.5 in.
9A	1	Low alloy: 2½ Ni (SA-203-A,B)	>.625 in.	1100°	>.625
9B	1	Low alloy: 3½ Ni (SA-203-D,E)	>.625 in.	1100°	>.625
41	–	Nickel 200	–	–	>1.5 in.
42	–	Monel 400	–	–	>1.5 in.
43	–	Inconel 600, 625	–	–	>.375 in.
45	–	Incoloy 800, 825	–	–	>.375 in.

Notes:

1. See ASME Code, Section VIII, Div. 1 Table UCS-56, for concessions/restrictions.
2. PWHT or radiography depends upon carbon content, grade of material, type of welding, thickness, preheat and interpass temperatures, and types of electrodes. See ASME Code, Section VIII, Div. 1 Table UHA-32, and paragraphs UHA 32 and 33 for concessions/restrictions.
3. Radiography shall be performed after PWHT when required. 100% R.T. is required for all vessels in lethal service (ASME Code UW-2(a)). Materials requiring impact testing for low temperature service shall be PWHT (ASME Code, UCS-67(c)).
4. Radiography applies to category A and B, type 1 or 2 joints only. Thicknesses refer to thinner of two materials being joined.

**Per ASME Code, Section VIII, Div. 1 for commonly used materials.*

11.4. Testing & Examination

The terms examination and test are often used interchangeably. This is not correct in a purely technical sense. The two terms have different meanings and should not be confused.

Examination

An examination is a passive evaluation technique or procedure that evaluates the material or work piece without subjecting the work piece to potential stress or strain. Examination cannot cause the work piece to be deformed or to fail.

Nondestructive Examination (NDE) is defined as the development and application of technical methods to examine materials and/or components in ways that do not impair future usefulness and serviceability in order to detect, locate, measure, interpret and evaluate flaws.

Examples of NDE:

- UT (Ultrasonic)
- PT (Penetrant or sometimes Liquid Penetrant)
- MT (Magnetic Particle Testing)
- PMI (Positive Material Identification)
- RT (Radiographic Testing)
- Eddy Current

Test

A test is an invasive procedure that engages the equipment, work piece or material as part of the testing process. The material, equipment or coupon participates in the test. Testing may cause the work piece to fail or be deformed. Tests are divided into destructive tests and non-destructive tests.

Examples of Tests

Destructive Tests:

- Tensile Test
- Bend Test
- Charpy Impact Test
- HIC Testing
- Autoclave testing for disbonding

Non-Destructive Tests:

- Hydrotest
- Pneumatic Test
- Soap Bubble Test
- Helium Leak Test
- Performance Test
- Acoustic Emission Test
- Hardness Test
- Ferrite Check
- Copper Sulfate Test
- PMI

Defects and Failures

No material or fabrication is perfect. Every man-made material has defects, no matter how slight. Our quest as engineers is not to find or create the perfect material or part, but rather to establish the maximum size of defect that is acceptable and will allow the part to function.

Different codes and standards have different acceptance levels for the same defect. Thus a defect, or indication, may be rejectable under one code but acceptable under another.

Defect: One or more flaws whose aggregate size, shape, orientation, location or properties do not meet specified acceptance criteria and are rejectable. A flaw becomes a defect when it is rejectable under a specific code or standard. A defect can further be defined as a lack or absence of something essential for completeness or perfection. Defects can range in size from atom sized dislocations to major discontinuities.

Examples of defects are as follows;

1. Discontinuity
2. Indication
3. Flaw
4. Flaw characterization
5. Imperfection
6. Nonrelevant indication
7. Relevant indication

Failure/Rejection: Failure of a material or part is defined as the omission to perform or non-performance, often involving deterioration or decay. A part may not be suitable for its intended purpose for a variety of reasons such as tolerance, surface finish, etc, and this would be a condition for rejection.

Many apparent defects may not reduce the service life of the vessel or part. On the other hand, a small, unseen sub-surface defect, may actually lead to failure.

NDE or NDT can help to find defects, but it is the code, standard or specification that determines if the defect is acceptable or not.

Hydrostatic, Pneumatic, and Proof Testing of Pressure Vessels

Shop Hydrostatic Testing (UG-99)

1. ASME Section VIII, Division 1 requires that the minimum metal temperature during hydrostatic test shall be at least 30°F above the MDMT of the vessel that is stamped on the nameplate but not greater than 120°F.
2. Gauges used for hydrotests shall have been calibrated within one year of the test for not less than 1.5 times, nor more than 4 times the pressure of the test.
3. Minimum holding time for test shall be 1 hour minimum.
4. The vessel may be tested with painting and or lining on both the exterior and interior. Although not an ASME Code requirement it is recommended that the weld seams should be left unpainted approximately 1 inch on either side of the seam.
5. Testing may be performed after refractory lining. That is, the test may be performed through the lining. The ASME Code Inspector (AI) must give permission prior to testing.
6. There is no "maximum" hydrostatic test pressure specified by the Code. Good engineering practice limits the general primary membrane stress to .9 F_y for carbon steel components and F_y for stainless steel components.
7. Vessels may be tested with liquids other than water.

Field Hydrostatic Testing. Field hydrostatic testing is not an ASME Code requirement. Field testing is optional to test the tightness of piping joints attached to the vessel after installation. Often it is a contractor's policy to design vessels and vessel foundations, with rare exceptions, for a future field hydrotest in the corroded condition. This gives clients the option to test the vessel on its foundation in the future should alterations or repairs warrant such a test.

1. Vessels may be filled with water and tested as part of the piping field test. It is easier and preferable in most cases to "test through" the vessels as opposed to blanking off the connections to test the field piping. In addition this tests the tightness of joints attached to the vessel.
2. Horizontal vessels may be tested at the same pressure as the shop test.
3. Vertical vessels may be tested to a maximum of the shop test pressure minus the hydrostatic head of liquid.
4. Vessels may be hydrotested, pneumatic tested or a combination of both.
5. All test pressures should be gauged at the top of the vessel as a minimum. For very large field fabricated vessels, gauges may also be employed on the side or bottom of the vessel.
6. Ensure that the vessel has been designed for a field hydrostatic test.
7. Either remove SST internals or use water with less than 50 PPM chlorides. Water greater than 200 PPM chlorides should not be used.
8. Examples of vessels that are typically exempted from field testing are large refractory lined vessels such as FCC reactors and regenerators.

Pneumatic Testing (UG-100)

1. Pneumatic testing may be used in lieu of hydrostatic testing provided;
   a. Cannot be safely filled or supported with water
   b. Cannot be dried sufficiently after the test
   c. A hydrostatic test could result in damage to internals
2. A pneumatic test shall be 1.1 times the MAWP times the stress material ratio (SMR)
3. Pressure shall be applied gradually up to ½ the test pressure. After that the pressure shall be increased in increments of 1/10 the ultimate pressure. Then the test pressure shall be reduced to 4/5 of the test pressure and held for a suitable period of time to allow inspection.
4. Secondary containment closures of nuclear power plants are pneumatically tested to approximately 65 PSIG and are over 200 feet in diameter.
5. Vessels to be pneumatically tested may be painted or coated internally and externally prior to the test.
6. All nozzle and attachment welds shall be 100% PT or MT examined, dependent on material, prior to testing.

Proof Testing (UG-101). A proof test is used to establish the MAWP by testing rather than calculation. A proof test is required to determine the MAWP of a vessel only when no other suitable means can do so with sufficient accuracy. That is, the configuration or construction cannot be adequately analyzed to mathematically develop a MAWP. This is an elaborate and cumbersome procedure that is rarely used. Designs that cannot be demonstrated by normal analytical techniques are rare exceptions.

The author has only been involved with two proof tests. One had a very large round flanged opening in a 2:1 SE head, offset from the vessel centerline, that resulted in significant loss of structural integrity of the head. Normal Code reinforcement methods were inadequate to replace the loss of rigidity of the head.

The second case involved the use of a 24 inch by 24 inch tee as part of the vessel shell. The proof test was done at the insistence of the ASME Code inspector. Proof tests may be required by the AI regardless of the mathematical proofs and FEA performed, if in their opinion, the design is sufficiently complex as to defy mathematical modeling.

Proof tests may result in permanent vessel deformation.

There are two types of proof tests provided for by the ASME Code. They are as follows;

A. Tests based on the yielding of the part
B. Tests based on the bursting of the part.

In the second test, at least two vessels are built by identical means. One vessel is pressured to either failure, leakage or bursting. This "failure pressure" is then used to develop an MAWP for the other vessel. The surviving vessel is then given a standard hydrostatic or pneumatic test based on the MAWP established.

Hardness Testing

Hardness testing is a critical NDE tool utilized to ensure that the welding, heat treatment and fabrication methods have not altered the original material of construction in a deleterious way. Hardness testing is a quick method to determine that forming and fabrication techniques have not made the material too soft or too hard. Only ductile materials are allowed for pressure vessel construction, however, forming, welding and heat treatment can alter the original material properties. The altered properties can be a result of work hardening (strain hardening) or metallurgical change. Almost all metals work harden.

Although hardness testing is normally performed to detect if a material is too hard, it can also identify materials that are too soft. This would indicate a loss of tensile strength which may have weakened the material beyond its ability to satisfy the ASME Code design. In the past we have seen vessels that were subject to a prolonged sub-critical anneal in order to reduce excessive hardness readings. The prolonged heat treatment weakened the material and the vessel had to be derated for the reduced tensile strength.

Hardness Requirements

Hardness requirements vary by type of material and service. We have had contracts where the Brinell hardness for carbon steel in wet H2S service was 185, however this number would typically be 200 HB or below. API 934 requires, for vessels of 2-1/4 or 3 Cr material, in high temperature, high pressure hydrogen service, a hardness of 225 HB for conventional materials and 235 HB for advanced steels. Carbon steels in hydrogen service should be 200 HB maximum. For general service, the following can apply but refer to specific client specifications for details;

P-1 (Corrosive service)	200HB
P-1	225HB
P-3 & 4	225HB
P-5,6 & 7	241HB
P-10	225HB

Locations of Test

Weld procedures require hardness testing on a section of the weld. The tests are performed on the weld metal, the HAZ and the base metal. Production tests are on the weld metal only for each welding process, filler metal and technique used. The location for the testing of the production welds shall be at the discretion of the inspector.

When?

After PWHT when required.

Quantity of Tests?

One set of hardness tests shall be taken in the weld metal and one in the HAZ. Each set shall consist of three samples.

Methods

The property of “hardness” is based on the material’s ability to resist scratching, wear, penetration, machinability, or the ability to cut. There are over 30 methods used for measuring hardness. A representative listing is as follows;

1. Knoop
2. Rockwell
3. Vickers
4. Moh’s: Comparitive scale between talc and diamond
5. Shore Durometer: For rubber and plastics
6. Scleroscope: utilizes a steel ball bounced off the specimen
7. Brinell

Of these, only three are really important to the steel/refining industry. They are;

1. Rockwell
2. Vickers
3. Brinell

In our industry, the most common application is the Brinell test method but the following will summarize the three processes.

Brinell Testing (BHN)

Brinell was a Swedish engineer who developed this testing process to determine both hardness and tensile strength. It consists of indenting the surface of the metal by a hardened steel ball under a load and then measuring the indentation. The diameter of the ball is 10 mm and the load is 3000 Kg, 1000 Kg for copper and 500 Kg for aluminum. Other ball sizes can be used and the results are just ratioed off of the ball size.

The time of the loading is 15 seconds. The indentation is measured with either a special microscope fitted with a scale or a portable version.

The indentation falls into two categories:

1. Piling up: Indicates a low rate of hardening by deformation
2. Sinking: Indicates the ability to work harden.

The approximate tensile strength in PSI can be ascertained by multiplying the Brinell hardness number × 500. For metric applications the tensile strength in N/mm^2 can be obtained by multiplying the Brinell number by 3.54 for annealed steels, and 3.24 for quench and tempered steels.

The numerical Brinell hardness number is equal to the load divided by the spherical surface area of the indentation expressed in Kg/mm^2

$$BHN = \text{Brinell Hardness Number}$$
$$= \text{Load/Surface area of indentation}$$

The Brinell test has several limitations;

1. Cannot be used on soft materials
2. The test may not be valid for thin specimens. The minimum thickness is about 0.313 in.
3. The test is not valid for case hardened materials
4. The test should not be conducted too close to the edge of material
5. The indentation may be objectionable on the finished part
6. The edge of the indentation may be difficult to see on some materials

Errors arise when the Brinell test is performed on very hard materials, resulting in low values owing to: (a) the spherical shape of the indentor (b) flattening of the ball. The Brinell number is not reliable above 600.

Vickers Pyramid Hardness Testing (DPH)

The Vickers Hardness Test is similar to the Brinell method, with a square based pyramid used as the indentor. As in the Brinell test, the Vickers number is the ratio of the load to the surface area of the indentation in kilograms per square millimeter. An advantage of the Vickers test is the increased accuracy in determining the diagonal of a square as opposed to the diameter of a circle.

Although Vickers test method is different than Brinell, the scales are identical up to about a hardness of 300. The Vickers test is less prone to the errors produced by the Brinell system because a diamond square based pyramid is used, which does not deform as easily as a ball.

Since the impressions are small, the machine is very suitable for polished or hardened materials.

Rockwell Hardness Testing

The Rockwell hardness test has eight different scales labeled A thru H. Each scale supports a different material, brass, bronze and soft metals. For our industry we are only concerned with the B and C scales designated as R_B and R_C. The Rockwell C scale is used for determining the hardness of hard steels. A conical diamond indentor is employed, called a brale, under a 150 Kg load. The Rockwell B test applies a 100 Kg load under a $^1/_{16}$ inch steel ball. For comparison purposes, the hardest steel is R_C 65. For reference, a steel is considered unmachinable when the hardness exceeds R_C 35.

Charpy Impact Testing (CIT)

Charpy impact testing, also known as Charpy V notch testing, is performed on materials to determine toughness properties, usually at low temperature. CIT enables one to determine the transition temperature between brittle and ductile failure for any material or material specimen. CIT is also a good indication of a materials ability to absorb shock loads at low temperatures. CIT is used predominantly for carbon and low alloy steels. It is not used for stainless steels because stainless steels do not fail in a brittle manner until extremely low temperatures.

CIT is a destructive test that utilizes test specimens machined from actual production test plates or sample materials. The test specimens can be machined for impact testing of all base metal, all weld metal or the heat affected zone (HAZ). Weld samples can be taken from the root area or any other specific area of the weld metal.

For forgings and fittings, the Code allows specimens to be machined from a sample piece of material that has experienced the same forming, forging or gross reduction of area as the work piece it represents. As an alternative, two forgings, flanges or fittings may be produced and one set destructively tested.

Test specimens are 10 mm × 10 mm × 55 mm long with a 2 mm deep 45 degree groove or notch cut into the specimen. Provisions are made in the ASME Code for thinner materials where sub-size specimens are used. The specimens are cooled in liquid nitrogen and allowed to warm to the exact temperature required for testing. At this point the specimen is placed in the swinging hammer device for testing. The hammer is pre-loaded to a certain starting position and locked. Once the hammer is released the amount of absorbed energy is recorded by the furthest travel of the swing arm. The absorbed energy, in foot pounds or joules, is the difference between the starting position of the hammer and the finishing position.

The main factors determined by the test are;

1. Absorbed energy
2. Lateral expansion
3. Percent shear

Lateral expansion is the amount of growth experienced by the specimen at the notch. The percent shear, or percent of fibrous fracture, is determined by comparing the samples to fracture appearance charts in ASTM A 370.

The ASME Code, Section VIII, Paragraph UG-84 defines the testing process and gives minimum absorbed energy for the test based on the material, thickness and tensile strength. Materials should be tested in their final heat treated condition. Each test consists of three samples each. The average of all three specimens shall meet the criteria specified, however one specimen is allowed to be 2/3 of the average energy required for the three specimens. In the event of a failure a retest can be performed. The retest shall consist of three new specimens and must meet the original criteria.

Types of Magnetic Particle Inspection

Techniques;

1. Continuous Technique
2. Residual Technique

Methods;

1. Dry powder application – Best for deep subsurface defects and is most convenient for field inspection with portable equipment. Best for rough surfaces.
   a. sprinkling from a shaker
   b. spraying from a puff bottle
   c. spraying from blowers
2. Wet application – Best for detecting fine cracks on relatively smooth material.
   a. Spray can – particles suspended in a liquid vehicle
3. Fluorescent – Must be observed under ultraviolet light
   a. Wet – the finest detection method of MT methods. Used for detecting microfissures in base material and welds. Usually required for all interior surfaces for vessels in hydrogen or wet H2S service.
   b. Dry

Types of Ultrasonic Examination

Major Types

1. Angle Beam – Used predominantly for forgings
2. Straight Beam – Used for plates
3. Time of Flight Diffraction (TOFD) – Used to provide a permanent record.
4. Shear Wave – May be either straight or angle beam. Refers to the direction in which the pulse echo enters the material.
5. Phased Array (PAUT)- Multiple probe technique for finding very small, minute cracks.

ASME Specifications for UT Examination

SA-388: Specification for the UT examination of heavy steel forgings

SA-435: Specification for straight beam UT examination of steel plates (½ in and over)

SA-577: Specification for the UT angle beam examination of steel plates

SA-578: Specification for straight beam UT examination of plain and clad steel plates for special applications (used to examine clad plates made to SA-263, -264, and -265, ⅜ in and over)

SA-745: Practice for UT examination of stainless steel forgings (straight and angle beam examination)

SA-20:

Supplement 8: UT examination in accordance with SA-435

Supplement 11: UT examination in accordance with SA-577

Supplement 12: UT examination in accordance with SA-578

Optional extent of examination for plate: 4 in linear or 9 in grid

Forgings are 100% examined

Acceptance criteria shall be per ASME Section VIII, Division 1

Requirements and methods shall be per ASME Section V, Article 5

All plate greater than 4 inches thick shall be UT examined per SA-435

All forgings greater than 4 inches thick shall be 100% UT examined per SA-388

Code Case 2235: Allows for UT of welds in lieu of radiography where required.

Types of Radiographic Examination (RT)

1. X-Ray
2. Gamma Ray
3. Source (Isotopes)
   a. Iridium 192
   b. Cobalt 60
   c. Selenium 75 (Thin Wall)
4. Linear Accelerator
5. Digital RT
6. Computed RT

11.5. Heat Treatment

1. **Types of Heat Treatment**
   a. Spheroidizing
   b. Tempering
   c. Homogenization (soaking of cast metals)
   d. Stress Relieving (sub-critical anneal / PWHT)
      - Global
      - Local
      - Progressive
   e. Normalizing
   f. Case Hardening
      - Nitriding
      - Carburizing
      - Cyaniding (.010 to .020 deep)
   g. Annealing
      - Process
      - Full
   h. Quenching
      - Direct
      - Interrupted
   i. Solution Annealing
   j. Combination
      - Quench and Temper
2. **Purpose for Heat Treatment**
   a. To harden, strengthen or toughen metal
   b. To soften metal, improve ductility
   c. To improve machinability
   d. To alter electrical or magnetic properties
   e. To refine or coarsen grain structure
   f. To surface treat metal
   g. To produce a constitutional change
   h. Removal of contained gas from materials
   i. Improving creep ductility
   j. Improving resistance to SCC
   k. Improving fatigue strength
3. **Types of Hardening of Metals**
   a. Alloy Hardening
   b. Cold Working
      - Forming
      - Drawing
      - Shot peening
   l. Precipitation Hardening
      - 17-7 PH Stainless Steel
      - Maraging steel
      - Aluminum alloys
   m. Transformation Hardening
      - Martensite (heat treating)
      - Quench & tempering
      - Austempering
      - Martempering
   n. Surface Hardening (case hardening)
      - Nitriding
      - Carbuerizing
      - Cyaniding
      - Coatings (spray on type)

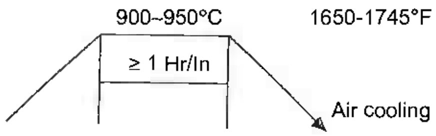
Heat Treatment Terms & Definitions

Air Hardening: A process used in steels that contain sufficient quantity of carbon or other alloys that will harden during air cooling.

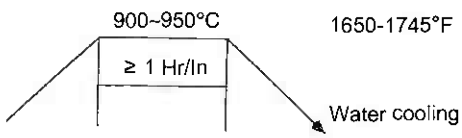
Annealing: The term annealing for carbon and alloy steels, implies slow cooling to soften or change the microstructure or crystalline structure. It is used to remove stresses, induce softness, alter ductility, toughness, improve machinability, dimensional stability, magnetic or other physical or mechanical properties. There are many specific types of annealing to include; black annealing,

TYPES OF HEAT TREATMENT

NORMALIZING



QUENCHING



TEMPERING

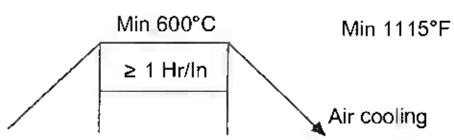
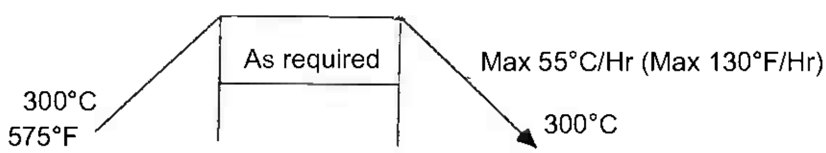


TABLE: PWHT TEMPERTURES

MATERIAL	TEMPERATURE °C	TEMPERATURE °F
Carbon Steel	595-650	1100-1200
Low Nickel	595-635	1100-1175
.3-.5 Mo Stl	595-660	1100-1220
1 Cr - 1/2 mo	680-720	1250-1325
1-1/4 Cr - 1/2 Mo	680-720	1250-1325
2-1/4 Cr - 1Mo	680-720	1250-1325
5 Cr- 1/2 Mo	720-760	1325-1400
9 Cr - 1 Mo	720-760	1325-1400
3 5 Ni	595-635	1100-1175
13 Cr	700-720	1250-1325
17 Cr	730-790	1350-1450

PWHT (See Table for Temperature Ranges)

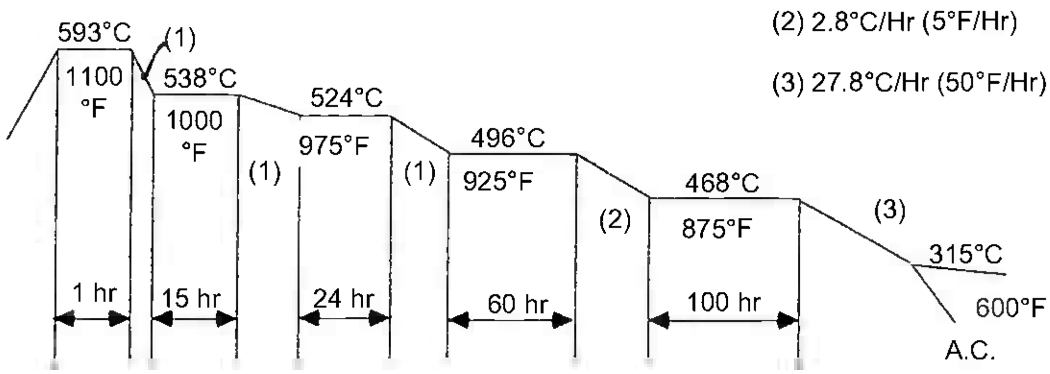


STEP COOLING FOR TEST BLOCK (Cr-Mo Only)

(1) 5.6°C/Hr (10°F/Hr)

(2) 2.8°C/Hr (5°F/Hr)

(3) 27.8°C/Hr (50°F/Hr)



blue annealing, box annealing, Bright annealing, cycle annealing, flame annealing, process annealing, full annealing, sub-critical annealing, quench annealing and isothermal annealing. When applied solely to the relief of stress, the process is more accurately called stress relieving.

Full Anneal: Austenitizing and then cooling at a rate such that the hardness of the part is minimum.

Heat Treatment: A combination of heating or cooling of a metal to obtain desired conditions or properties. Heating, solely for the purpose of hot working is excluded from this definition.

Homogenization (Soaking): A high temperature heat treatment intended to eliminate or decrease chemical segregation by diffusion.

Interrupted Quench: A quenching procedure where the initial quench is interrupted, followed by final quenching at a different rate or in a different medium to alter the quench depth or properties.

Normalizing: Heat treatment of an iron based alloy is heated to a temperature at least 100 degrees above the transformation range and then cooled in still air. This process produces a recrystallization and refinement of the grain in the material that results in uniform hardness and structure.

Process Anneal: A generic term to define a heat treatment that improve workability.

Quenching: Also known as quench hardening. Heating uniformly to a pre-determined temperature and cooling rapidly in air or liquid to produce a desired crystalline structure. Quenching can be done in water, brine, oil, polymer or even forced or still air. There are two types of quenching. The first to obtain mechanical properties. The second to retain uniformity of material.

Recrystallization (or anneal): Used for non-ferrous or work hardened metals, to soften and remove strain hardening.

Solution Annealing: A process in which certain alloys are heated to a suitable temperature to allow the constituents to enter into solid solution. The ingredients are held in this state until rapid cooling occurs. In stainless steel, the material is heated to 1950°F and quenched rapidly in liquid. The purpose is to freeze the constituents in the austenitic phase.

Spheroidization: Heating and cooling in a cycle designed to produce spheroidal or globular forms of carbide within the microstructure. It is used primarily in cast iron.

Stabilizing Treatment: A heat treatment to stabilize the dimensions of the part or work piece. In stainless steels it refers to heating the metal to below the solution heat treatment temperature to allow the precipitation of carbides to combine with certain alloy ingredients, specifically titanium or columbium (niobium).

Stress Relief: A heat treatment process to reduce internal residual stress by heating to a desired temperature and holding for a suitable period of time, also known as a sub-critical anneal. Residual stresses can be induced by forging, casting, forming, welding or cold working of metal.

Sub-Critical Anneal: A high temperature tempering process for steel that produces many of the benefits of annealing but does not require cooling at a controlled rate.

Tempering: Heating a quench hardened or normalized ferrous alloy to a temperature below the transformation range to produce desired changes in properties, predominantly to soften or toughen. It is used to remove brittleness from quench hardened steel. In chrome moly Q&T steel, the tempering processs is accomplished by the final PWHT.

Transformation Range: The temperatures at which austenite forms during heating or cooling.

Transformation Temperature: The temperature at which a change in phase occurs.

Autoclave Testing: This is a test for cladding/overlay disbondment for all reactors in hydrogen service. The autoclave testing proves that the WPS is good for the intended service. Typically results of previous tests for a particular WPS will be acceptable if the other parameters are in the range. The test consists of welding a sample and putting it in a hydrogen rich atmosphere in an autoclave for a specified time at the design conditions. Then sample is removed for disbanding tests. Minimum shear is 20,000 PSI.

DHT (Dehydrogenation Heat Treatment): This is a heat treatment procedure used during the fabrication cycle only when welding or preheat is interrupted or stopped. Done at 600 – 650°F (min 570°F). Very common and used in lieu of ISR. It is a bakeout to ensure that trapped hydrogen in the welds has the opportunity to escape to the atmosphere. Used for less restrained welds like main reactor seams.

ISR (Intermediate Stress Relieve): This is a heat treatment procedure used during the fabrication cycle prior to allowing the material temperature to cool below the preheat temperature. Done at 1150°F. Must be done in a furnace to achieve these temperatures. Fabricators usually elect to do DHT in lieu of ISR due to ease and convenience.

Step Cooling: This is a testing procedure only to evaluate the long term effects of temper embrittlement. The test was developed by GE originally for turbine blades and since adopted by the refining industry for hydroprocessing reactors. This test of temper embrittlement is used for 2-1/4 Cr materials only. The heat treatment takes about 12 days before testing of the coupon can occur.

J Factor: Since temper embrittlement is a function of alloy and tramp elements present in the steel. The J factor is based on a mathematical equation that combines the overall effects of the various ingredients that are to be controlled. For 2-1/4 Cr and 3 Cr the limit is typically 100. For 1-1/4 Cr it is about 180 maximum. It applies to all product forms except tubing. The equation is;

$$\begin{aligned}\text{J Factor} &= (\text{Si} + \text{Mn}) \times (\text{P} + \text{Si}) \\ &\quad \times 10^4 (\text{Si, Mn, P \& Si are wt \%})\end{aligned}$$

In addition the Cu content should be limited to .2% Maximum and Ni to .3% Maximum.

X Factor: Similar to J factor but only applies to weld consumables.

$$\begin{aligned}\text{X Factor} &= (10\text{P} + 5\text{Sb} + 4\text{Sn} + \text{As})/100 < \text{or} \\ &= \text{to } 15(\text{P, Sb, Sn, As are in PPM})\end{aligned}$$

Minimum PWHT: Consists of one heat treatment cycle.

Maximum PWHT: Consists of all heat treatment cycles specified (usually three) in order to allow for future PWHT cycles that may be required. The material and WPS are prequalified with multiple PWHT cycles to ensure properties.

Post Weld Heat Treatment (PWHT) Alternatives

Post weld heat treatment of a vessel that cannot be contained in a single furnace load:

Although it is preferable to post weld heat treat the completed vessel in an enclosed furnace, this is not always possible. If it is not possible to place the entire vessel inside a furnace for PWHT the alternatives are as follows:

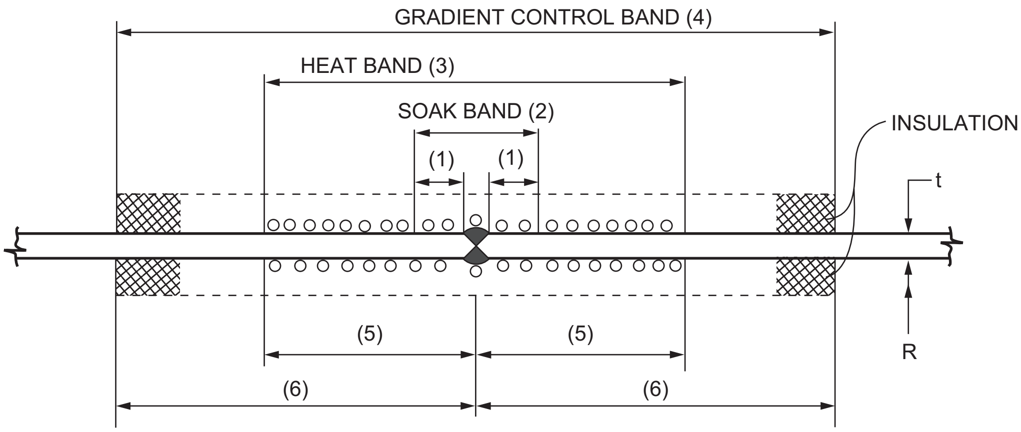
1. Heating the vessel portions in more than one heat: If the vessel is too long for the furnace then one portion of the completed vessel is post weld heat treated and then the vessel is turned around to do the remaining portion. This method requires a thermal gradient control band for that portion of the vessel extending outside the furnace. The minimum overlap of each portion is 5 feet.
2. Heating the vessel in two or more sections in a furnace, then perform a local PWHT on the final closure seams.
3. Heating vessel internally: Typically done with burners placed through nozzles or manways. The vessel shall be fully insulated on the outside and fully instrumented for temperature control.

Local Post Weld Heat Treatment:

This procedure is typically done following a repair or as the closure seam for a vessel that is post weld heat treated in multiple sections.

1. Heating shall be done with a full circumferential band, typically with electric heating coils.
2. There are three types of bands required for a local PWHT and are as follows:
   a. Soak band: The soak band is the area including the weld seam as well as an area on either side of the weld. The minimum band width shall be the greater of 2t or 2 inches on either side of the weld measured from the toe of the weld. The time and temperature of the soak band shall meet the PWHT criteria.
   b. Heating band: The heating band shall extend to each side of the soak band for a minimum distance of $2(Rt)^{1/2}$. The temperature shall be lower than the soak temperature and determined to prevent severe thermal gradients.
   c. Gradient control band (also known as the insulation band): The GCB shall extend either side of the heating band. The distance shall be determined to reduce thermal gradients. Heating coils may, or may not be required in this region.
3. An insulating bulkhead may be required to isolate the heat within a certain area of the vessel. The bulkhead is positioned to prevent heat loss to adjacent areas.

REQUIREMENTS FOR LOCAL PWHT

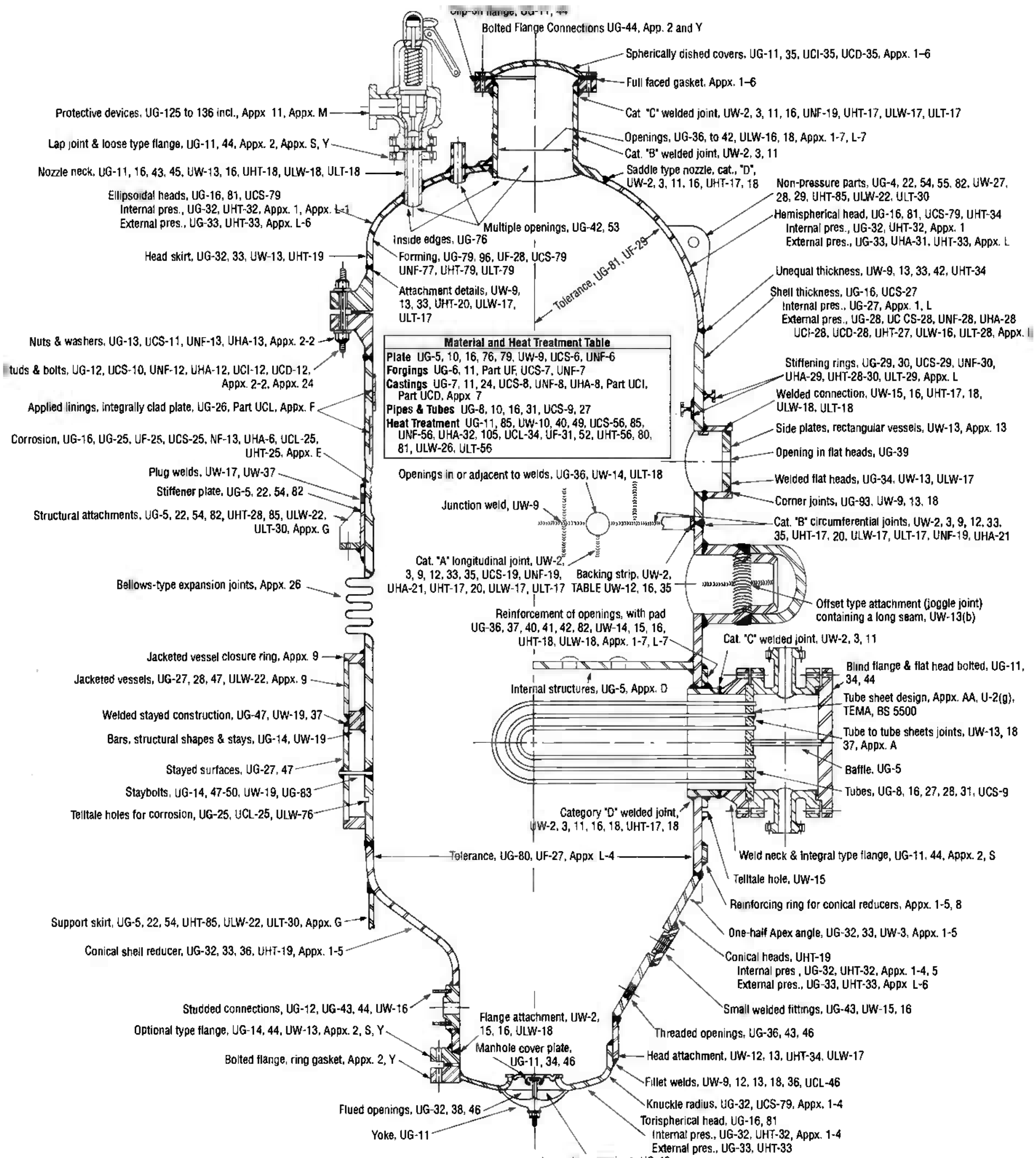


NOTES:

(1) Distance = Larger of 2t or 2"
(2) Soak Band: PWHT time & temperature
(3) Heat Band: ½ PWHT temperature
(4) Gradient control Band: No heat mandatory. Insulation only.
(5) Distance = 2 $(Rt)^{1/2}$
(6) Distance = 4 $(Rt)^{1/2}$
(7) All bands shall be full circumferential bands.
(8) Heating coils should be used inside and out, if possible

Appendices

Appendix A: Guide to ASME Section VIII, Division 1



Organization of Section VIII, Div. 1

Introduction—Scope and Applicability
Subsection A—part UG—General requirements for all construction and all materials.
Subsection B—Requirements for method of fabrication
- Part UW—Welding
- Part UF—Forging
- Part UB—Brazing

Subsection C—Requirements for classes of material.
- Part UCS—Carbon and low alloy steels
- Part UNF—Nonferrous materials
- Part UHA—High alloy steels
- Part UCI—Cast iron
- Part UCL—Clad plate and corrosion resistant liners
- Part UCD—Cast ductile iron
- Part UHT—Ferritic steels with tensile properties enhanced by heat treatment
- Part ULW—Layered construction
- Part ULT—Low Temperature Materials
- Mandatory appendices—1through 29
- Nonmandatory appendices—A through Y, AA, CC, DD, EE

Quality Control System	U-2, Appx. 10
Material—General	UG-4, 10, 11, 15, Appx. B
(a) Plate	UG-5
(b) Forgings	UG-6
(c) Castings	UG-7
(d) Pipe & Tubes	UG-8
(e) Welding	UG-9
(f) Bolts & Studs	UG-12
(g) Nuts &Washes	UG-13
(h) Rods & Bars	UG-14
(i) Standard Parts	UG-11, 44
Design Temperature	UG-20
Design Pressure	UG-21, UG-98
Loadings	UG-22, Appx. G
Stress—Max. Allowable	UG-23
Manufacturer's Responsibility	U-2, UG-90
Inspector's Responsibility	U-2, UG-90
User's Responsibility	U-2

General Notes

Pressure Tests	UG-99, 100, 101, UW-50, UCI-99, UCD-99
Low Temperature Service	UW-2, Part ULT
Quick Actuating Closures	U-1, UG-35, ULT-2
Service Restrictions	UW-2, UB-3, UCL-3, UCD-2
Nameplates. Stamping & Reports	UG-115 to 120 UHT-115, ULW-115, ULT-115, Appx. W
Non-Destructive Examination	
(a) Radiography	UW-51, 52
(b) Ultrasonic	Appx. 12
(c) Magnetic Particle	Appx. 6
(d) Liquid Penetrant	Appx. 8
Porosity Charts	Appx. 4
Code Jurisdiction Over Piping	U-1
Material Tolerances	UG-16
Material Identification, Marking and Certification	UG-77, 93, 94
Dimpled or Embossed Assemblies	Appx. 17

Courtesy of Hartford Steam Boiler Inspection and Insurance Company

Appendix B: Design Data Sheet for Vessels

1 Customer/Client				
2 Customer Order No.				
3 Shop Order No.				
4 Design Drawing				
5 Specifications				
6 Vessel Name				
7 Equipment/Item Number				
8 Design Code & Addenda				
9 Design Pressure & Temperature	**Internal**		**External**	
10 Operating Pressure & Temperature				
11 Vessel Diameter				
12 Volume				
13 Design Liquid Level				
14 Contents & Specific Gravity				
15 Service				
16 MAWP (Corrosion at Design Temperature)			**Limited by**	
17 MAP (N & C)				
18 Test pressures	**Shop**		**Field**	
19 Heat treatment				
20 Joint efficiencies	**Shell**			
	Heads			
21 Corrosion allowance	**Shell**		**22 Flange ratings**	
	Heads		**MAP:**	**psig at Ambient**
	Nozzles		**MAWP:**	**psig at D.T.**
	Boot		**Hydro:**	**psig**
23 Materials		**24 Allowable Stress**	**Ambient**	**D.T.**
Shell				
Heads				
Nozzles				
Flanges				
Bolting				
Supports				
25 Weights	**Fabricated**	**Operating**		
	Empty	**Test**		
26 Notes/remarks				

Appendix C: Joint Efficiencies (ASME Code) [3]

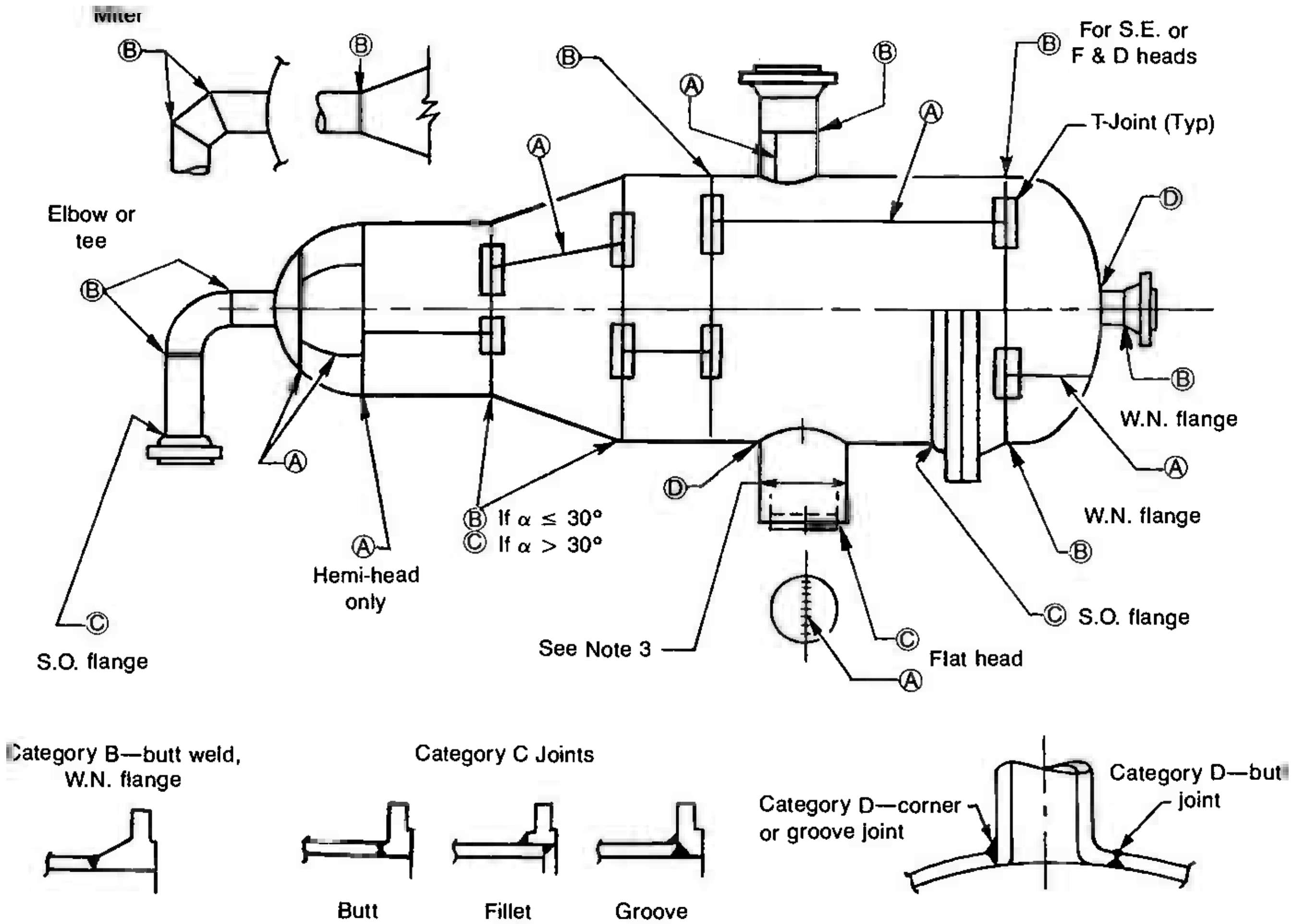
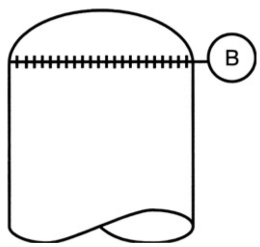
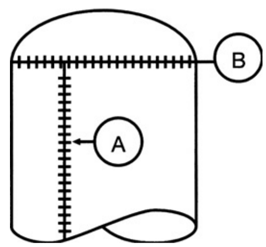
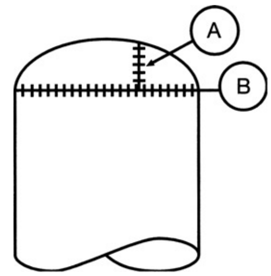
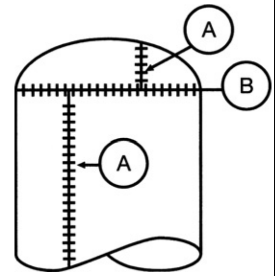


Figure C-1. Categories of welded joints in a pressure vessel.

Table C-1
Types of joints and joint efficiencies

Types of Joints		X-Ray Full	X-Ray Spot	None	Types of Joints		X-Ray Full	X-Ray Spot	None
1	Single- and double-butt joints	1.0	0.85	0.7	4	Double full fillet lap joint	—	—	0.55
2	Single-butt joint with backing strip	0.9	0.8	0.65	5	Single full fillet lap with plugs	—	—	0.5
3	Single butt joint without backing strip	—	—	0.6	6	Single full fillet lap joint	—	—	0.45

Table C-2
Application of joint efficiencies

Extent of Radiography		Case 1		Case 2		Case 3		Case 4	
		Seamless Head Seamless Shell		Seamless Head Welded Shell		Welded Head Seamless Shell		Welded Head Welded Shell	
									
		Head	**Shell**	**Head**	**Shell**	**Head**	**Shell**	**Head**	**Shell**
Full	Cat. A and B	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
Spot	Cat. A only	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85
	(4)	1.0	1.0	1.0	0.85	0.85	1.0	—	—
Part	(2)	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
None	Cat. A and B	0.85	0.85	0.85	0.7	0.7	0.85	0.7	0.7

Notes

1. In Table C-2 joint efficiencies and allowable stresses for shells are for longitudinal seams only and all joints are assumed as Type 1 only.
2. *"Part" radiography:* Applies to vessels *not* fully radiographed where the designer wishes to apply a joint efficiency of 1.0 per ASME Code, Table UW-12, for only a specific part of a vessel. Specifically for any part to meet this requirement, you must perform the following:
   - *(ASME Code, Section UW(5))*: Fully x-ray any Category A or D *butt* welds.
   - *(ASME Code, Section UW-11(5)(b))*: Spot x-ray any Category B or C *butt* welds attaching the part.
   - *(ASME Code, Section UW-11(5)(a))*: All butt joints must be Type 1 or 2.
3. Any Category B or C butt weld in a nozzle or communicating chamber of a vessel or vessel part which is to have a joint efficiency of 1.0 and exceeds either 10-in. nominal pipe size or 1⅛ in. in wall thickness shall be fully radiographed. See ASME Code, Sections UW-11(a)(4).
4. In order to have a joint efficiency of 1.0 for a seamless part, the Category B seam attaching the part must, as a minimum, be spot examined.

Appendix D: Properties of Heads

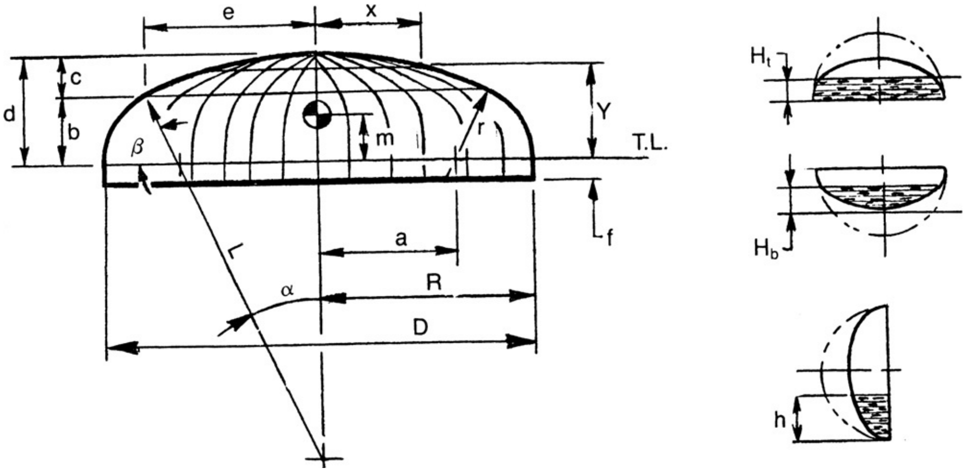


Figure D-1. Dimensions of heads.

Formulas

$$a = \frac{D - 2r}{2}$$
$$\alpha = \arcsin\left(\frac{a}{L - r}\right)$$
$$\beta = 90 - \alpha$$
$$b = \cos \alpha \, r$$
$$c = L - \cos \alpha \, L$$
$$e = \sin \alpha \, L$$
$$\phi = \frac{\beta}{2}$$

Volume

$$V_1 = (\text{frustum}) = 0.333b\pi\left(e^2 + ea + a^2\right)$$
$$V_2 = (\text{spherical segment}) = \pi c^2\left(L - \frac{c}{3}\right)$$
$$V_3 = (\text{solid of revolution})$$
$$= \frac{120r^3\pi \sin \phi \cos \phi + a\phi\pi^2 r^2}{90}$$

TOTAL VOLUME: $V = V_1 + V_2 + V_3$

Depth of Head

$$A = L - r$$
$$B = R - r$$
$$d = L - \sqrt{A^2 - B^2}$$

Table D-1
Partial volumes

Type	Volume to H_t	Volume to H_b	Volume to h
HEMI	$\frac{\pi D^2 H_1}{4}\left[1 - \frac{4H_t^2}{3D^2}\right]$	$\frac{\pi D H_b^2}{2}\left[1 - \frac{2H_b}{3D}\right]$	$\frac{\pi h^2(1.5D - h)}{6}$
2:1 S.E.	$\frac{\pi D^2 H_t}{4}\left[1 - \frac{16H_t^2}{3D^2}\right]$	$\pi D H_b^2\left[1 - \frac{4H_b}{3D}\right]$	$\frac{\pi h^2(1.5D - h)}{12}$
100%–6% F & D	$\frac{3VH_t}{2d}\left[1 - \frac{H_t^2}{3d^2}\right]$	$\frac{3VH_b^2}{2d^2}\left[1 - \frac{H_b}{3d}\right]$	$\frac{3Vh^2}{D^2}\left[1 - \frac{2h}{3D}\right]$

Table D-2
General data

			C.G.-m			Points on heads	
Type	Surf. Area	Volume	Empty	Full	Depth of head-d	X=	Y =
HEMI	$\pi D^2/2$	$\pi D^3/12$	0.2878D	0.375D	0.5D	$\sqrt{R^2 - Y^2}$	$\sqrt{R^2 - X^2}$
2:1 S.E.	$1.084D^2$	$\pi D^3/24$	0.1439D	0.1875D	0.25D	$0.5\sqrt{D^2 - 16Y^2}$	$0.25\sqrt{D^2 - 4X^2}$
100%–6% F&D	$0.9286D^2$	$0.0847D^3$	0.100D		0.162D		

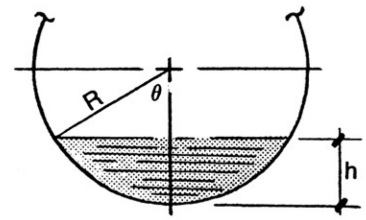
Appendix E: Volumes and Surface Areas of Vessel Sections

Notation

ℓ = height of cone, depth of head, or length of cylinder
α = one-half apex angle of cone
D = large diameter of cone, diameter of head or cylinder
R = radius
r = knuckle radius of F & D head
L = crown radius of F & D head
h = partial depth of horizontal cylinder
K, C = coefficients
d = small diameter of truncated cone
V = volume

$$K = \frac{L}{R} - \sqrt{\left(\frac{L}{R} - 1\right)\left(\frac{L}{R} + 1 - \frac{2r}{R}\right)}$$

$$e = \sqrt{1 - \frac{\ell^2}{R^2}}$$



$$\theta = \arccos\frac{R - h}{R}$$

$$V = R^2\ell\left[\left(\frac{\pi\theta^{o}}{180}\right) - \sin\theta\cos\theta\right]$$

or

$V = \pi R^2 \ell c$ (See Table E-3 for values of c.)

Figure E-1. Formulas for partial volumes of a horizontal cylinder.

Table E-1
Volumes and surface areas of vessel sections

Section	Volume		Surface area	
	Diameter	Radius	Diameter	Radius
Sphere	$\pi D^3 / 6$	$4 \pi R^3 / 3$	πD^2	$4 \pi R^2$
Hemi Head	$\pi D^3 / 12$	$2 \pi R^3 / 3$	$\pi D^2 / 2$	$2 \pi R^2$
2:1 S.E. Head	$\pi D^3 / 24$	$\pi R^3 / 3$	$1.084 D^2$	$4.336 R^2$
Ellipsoidal Head	$\pi D^2 h / 6$	$2 \pi R^2 h / 3$	$.5 [.5\pi D^2 + (\pi h^2/e) \text{Ln} (1 + e / 1 - e)]$	$.5 [2\pi R^2 + (\pi h^2/e) \text{Ln} (1 + e / 1 - e)]$
100% - 6% F&D Head	$.08467 D^3$	$.6774 R^3$	$.9286 D^2$	$3.7144 R^2$
F&D Head	$\pi D^3 K / 12$	$2 \pi R^3 K / 3$	$.25 \pi D^2[1 + 4h^2/D^2 (2 - 2h/D)]$	$\pi R^2[1 + h^2/R^2 (2 - h/R)]$
Cone	$\pi D^2 h / 12$	$\pi R^2 h / 3$	$\pi Dh / 2 \cos \alpha$	$\pi Rh / \cos \alpha$
Truncated Cone	$.0833[\pi h (D^2 + Dd + d^2)]$	$.33[\pi h (R^2 + Rr + r^2)]$	$\pi [.5(D + d)] [h^2 + .5 (D - d)^2]^{1/2}$	$\pi (R + r) [h^2 + (R - r)^2]^{1/2}$
30° Truncated Cone	$.227 (D^3 - d^3)$	$1.826 (R^3 - r^3)$	$.5 \pi (D^2 - d^2)$	$2 \pi (R^2 - r^2)$
Cylinder	$\pi D^2 h / 4$	$\pi R^2 h$	πDh	$2 \pi Rh$

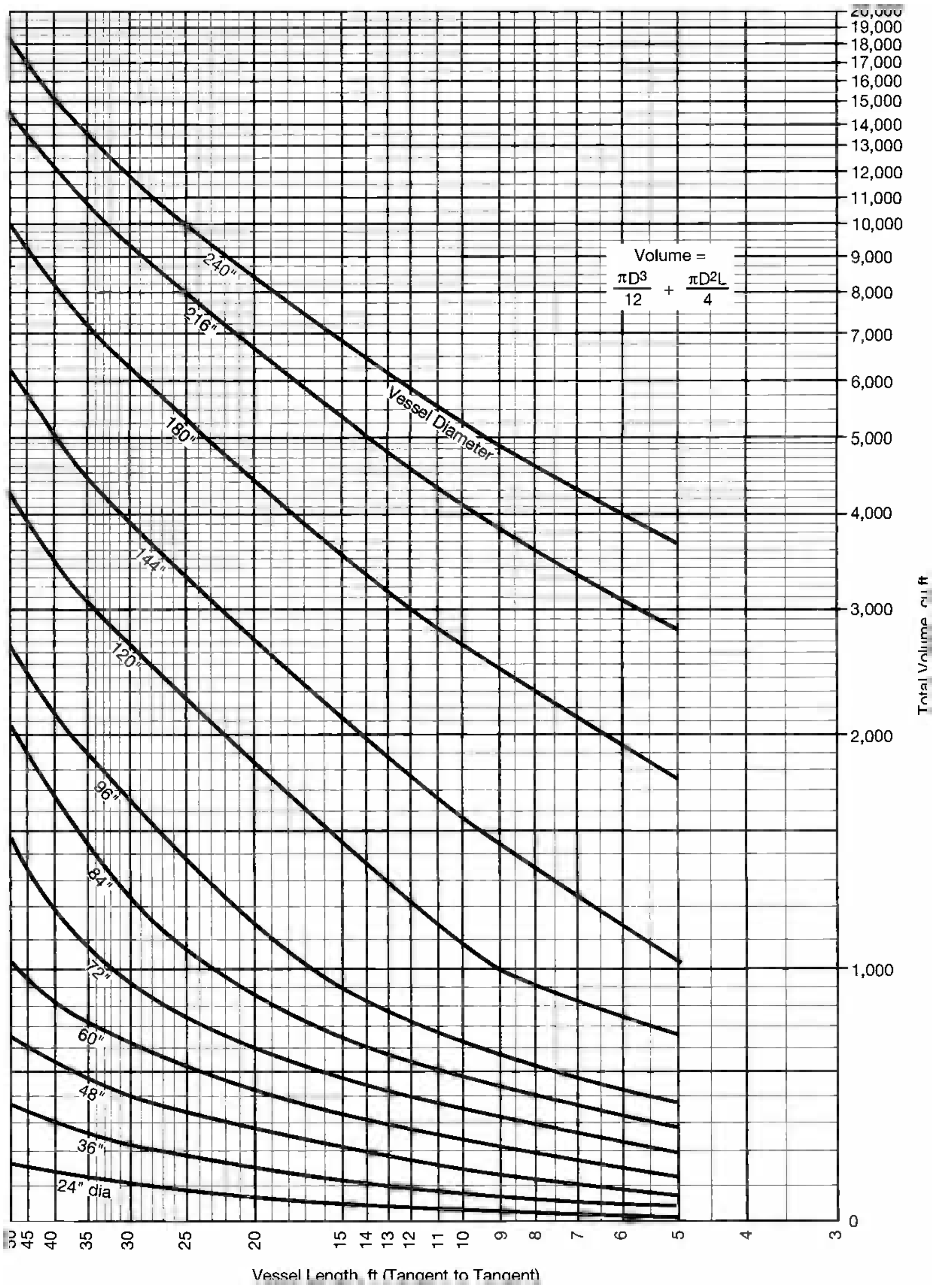


Figure E-2. Volume of vessels (includes shell plus (2) 2:1 S.E. Heads).

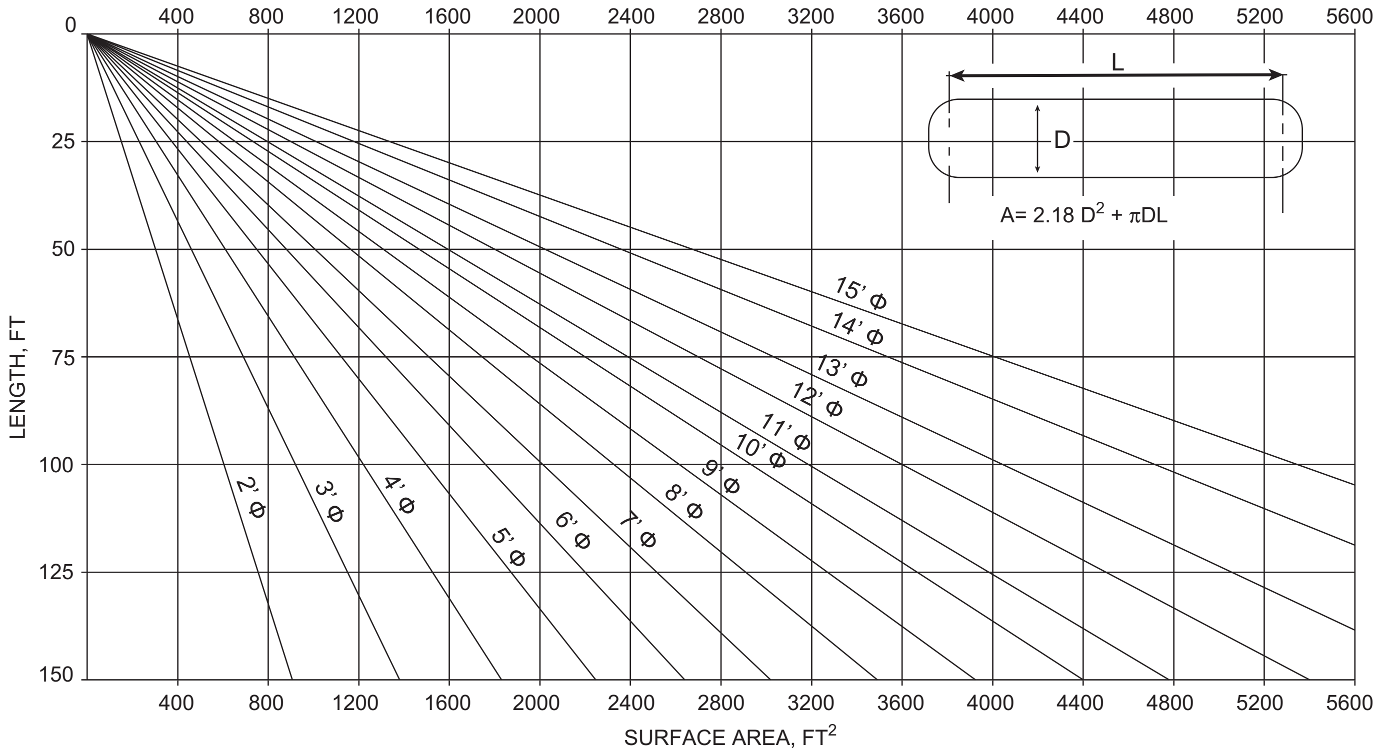


Figure E-3. Surface area of vessel with (2) 2:1 S.E. Heads.

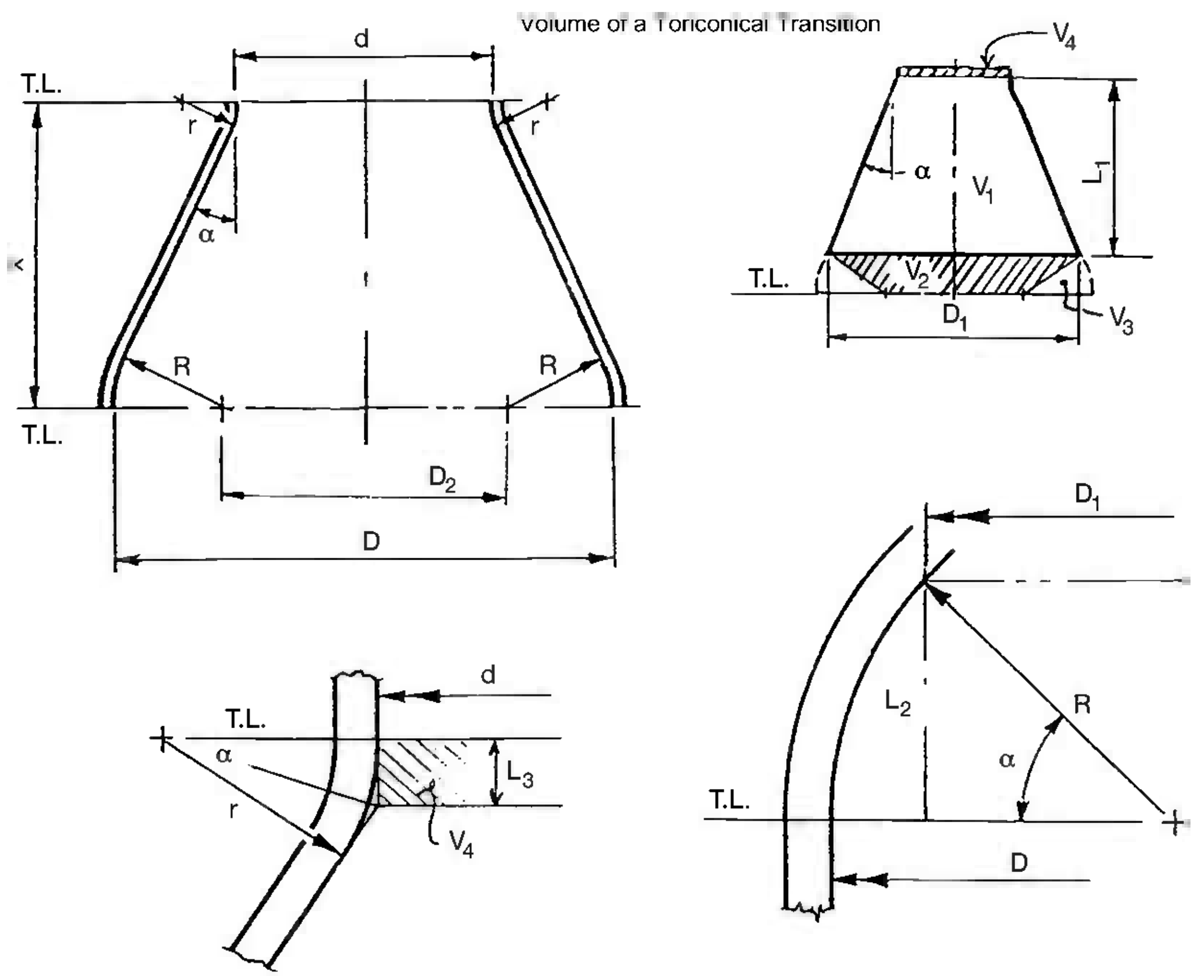


Figure E-4. Volume of a toriconical transition.

Dimensions

$D =$

$d =$

$R =$

$r =$

$x =$

$\alpha =$

$L_2 = \sin \alpha\, R =$

$L_3 = \tan \dfrac{\alpha}{2}(r) =$

$L_1 = x - L_2 - L_3 =$

$D_1 = D - 2(R - R \cos \alpha) =$

$D_2 = D - 2R =$

Volumes

$$V_1 = \frac{\pi L_1 \left(D_1^2 + D_1 d + d^2\right)}{12} =$$

$$V_2 = \frac{\pi L_2 \left(D_1^2 + D_1 D_2 + D_2^2\right)}{12} =$$

$$V_3 = \frac{120R^3 \pi \sin(\alpha/2)\cos(\alpha/2) + .25 D_2 R^2(\alpha/2)}{90} =$$

$$V_4 = \frac{\pi d^2 L_3}{4} =$$

$$\sum V = V_1 + V_2 + V_3 + V_4 =$$

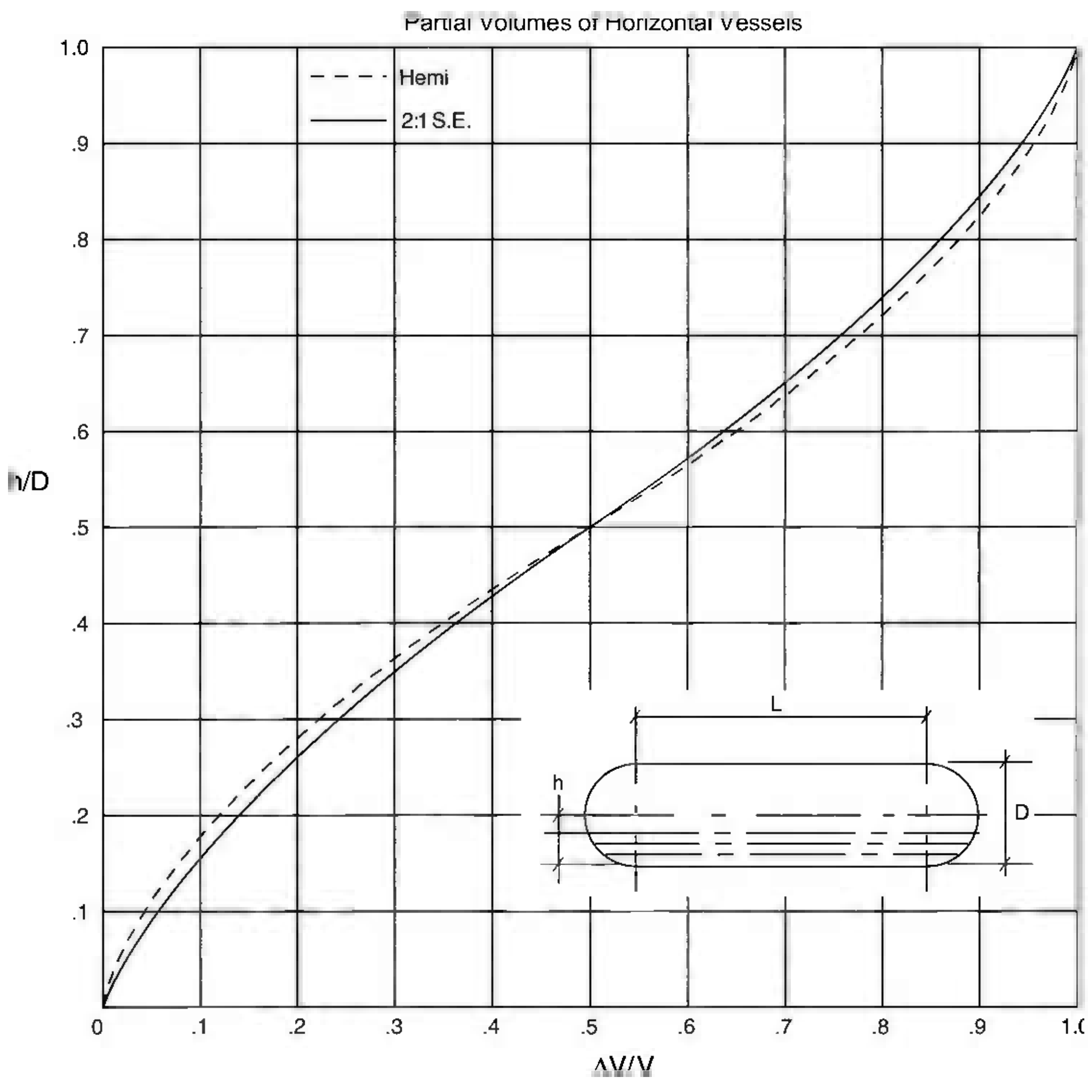
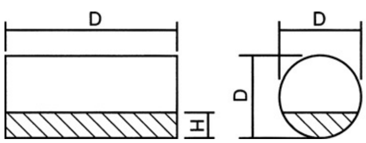


Figure E-5. Partial volumes of horizontal vessels.

Table E-2
Formulas for full and partial volumes

	Full Volume, V	Partial Volume, ΔV
Cylinder	$\frac{\pi D^2 L}{4}$	$\frac{\pi D^2 LC}{4}$
(2) Hemi-heads	$\frac{\pi D^3}{6}$	$\frac{\pi h^2(1.5D - h)}{6}$
(2) 2:1 S.E. Heads	$\frac{\pi D^3}{12}$	$\frac{\pi h^2(1.5D - h)}{3}$

Table E-3
Partial volumes in horizontal cylinders



Partial volume in height (H) = cylindrical coefficient for H/D × total volume

$$\text{Total volume} = \frac{\pi L D^2}{4}$$

Coefficients for Partial Volumes of Horizontal Cylinders, C

H/D	0	1	2	3	4	5	6	7	8	9
0.00	0.000000	0.000053	0.000151	0.000279	0.000429	0.000600	0.000788	0.000992	0.001212	0.001445
0.01	0.001692	0.001952	0.002223	0.002507	0.002800	0.003104	0.003419	0.003743	0.004077	0.004421
0.02	0.004773	0.005134	0.005503	0.005881	0.006267	0.006660	0.007061	0.007470	0.007886	0.008310
0.03	0.008742	0.009179	0.009625	0.010076	0.010534	0.010999	0.011470	0.011947	0.012432	0.012920
0.04	0.013417	0.013919	0.014427	0.014940	0.015459	0.015985	0.016515	0.017052	0.017593	0.018141
0.05	0.018692	0.019250	0.019813	0.020382	0.020955	0.021533	0.022115	0.022703	0.023296	0.023894
0.06	0.024496	0.025103	0.025715	0.026331	0.026952	0.027578	0.028208	0.028842	0.029481	0.030124
0.07	0.030772	0.031424	0.032081	0.032740	0.033405	0.034073	0.034747	0.035423	0.036104	0.036789
0.08	0.37478	0.038171	0.038867	0.039569	0.040273	0.040981	0.041694	0.042410	0.043129	0.043852
0.09	0.044579	0.045310	0.046043	0.046782	0.047523	0.048268	0.049017	0.049768	0.050524	0.051283
0.10	0.052044	0.052810	0.053579	0.054351	0.055126	0.055905	0.56688	0.057474	0.058262	0.059054
0.11	0.059850	0.060648	0.061449	0.062253	0.063062	0.063872	0.064687	0.065503	0.066323	0.067147
0.12	0.067972	0.068802	0.069633	0.070469	0.071307	0.072147	0.72991	0.073836	0.074686	0.075539
0.13	0.076393	0.077251	0.078112	0.078975	0.079841	0.080709	0.081581	0.082456	0.083332	0.084212
0.14	0.085094	0.085979	0.086866	0.087756	0.088650	0.089545	0.090443	0.091343	0.092246	0.093153
0.15	0.094061	0.094971	0.095884	0.096799	0.097717	0.098638	0.099560	0.100486	0.101414	0.102343
0.16	0.103275	0.104211	0.105147	0.106087	0.107029	0.107973	0.108920	0.109869	0.110820	0.111773
0.17	0.112728	0.113686	0.114646	0.115607	0.116572	0.117538	0.118506	0.119477	0.120450	0.121425
0.18	0.122403	0.123382	0.124364	0.125347	0.126333	0.127321	0.128310	0.129302	0.130296	0.131292
0.19	0.132290	0.133291	0.134292	0.135296	0.136302	0.137310	0.138320	0.139332	0.140345	0.141361
0.20	0.142378	0.143308	0.144419	0.145443	0.146468	0.147494	0.148524	0.149554	0.150587	0.151622
0.21	0.152659	0.153697	0.154737	0.155779	0.156822	0.157867	0.158915	0.159963	0.161013	0.162066
0.22	0.163120	0.164176	0.165233	0.166292	0.167353	0.168416	0.169480	0.170546	0.171613	0.172682
0.23	0.173753	0.174825	0.175900	0.176976	0.178053	0.179131	0.180212	0.181294	0.182378	0.183463
0.24	0.184550	0.185639	0.180729	0.187820	0.188912	0.190007	0.191102	0.192200	0.193299	0.194400
0.25	0.195501	0.196604	0.197709	0.198814	0.199922	0.201031	0.202141	0.203253	0.204368	0.205483
0.26	0.206600	0.207718	0.208837	0.209957	0.211079	0.212202	0.213326	0.214453	0.215580	0.216708
0.27	0.217839	0.218970	0.220102	0.221235	0.222371	0.223507	0.224645	0.225783	0.226924	0.228065
0.28	0.229209	0.230352	0.231408	0.232644	0.233791	0.234941	0.236091	0.237242	0.238395	0.239548

(*Continued*)

Table E-3
Partial volumes in horizontal cylinders—cont'd

0.29	0.240703	0.241859	0.243016	0.244173	0.245333	0.246494	0.247655	0.248819	0.249983	0.251148
0.30	0.252315	0.253483	0.254652	0.255822	0.256992	0.258165	0.259338	0.260512	0.261687	0.262863
0.31	0.264039	0.265218	0.266397	0.267578	0.268760	0.269942	0.271126	0.272310	0.273495	0.274682
0.32	0.275869	0.277058	0.278247	0.279437	0.280627	0.281820	0.283013	0.284207	0.285401	0.286598
0.33	0.287795	0.288992	0.200191	0.291300	0.292591	0.293793	0.294995	0.296198	0.297403	0.298605
0.34	0.299814	0.301021	0.302228	0.303438	0.304646	0.305857	0.307068	0.308280	0.309492	0.310705
0.35	0.311918	0.313134	0.314350	0.315566	0.316783	0.318001	0.319219	0.320439	0.321660	0.322881
0.36	0.324104	0.325326	0.326550	0.327774	0.328999	0.330225	0.331451	0.332678	0.333905	0.335134
0.37	0.336363	0.337593	0.338823	0.340054	0.341286	0.342519	0.343751	0.344985	0.346220	0.347455
0.38	0.348690	0.349920	0.351164	0.352402	0.353640	0.354879	0.356119	0.357359	0.358599	0.359840
0.39	0.361082	0.362325	0.363568	0.364811	0.366056	0.367300	0.368545	0.369790	0.371036	0.372282
0.40	0.373530	0.374778	0.376026	0.377275	0.378524	0.379774	0.381024	0.382274	0.383526	0.384778
0.41	0.386030	0.387283	0.388537	0.389790	0.391044	0.392298	0.393553	0.394808	0.396063	0.397320
0.42	0.398577	0.399834	0.401092	0.402350	0.403608	0.404866	0.406125	0.407384	0.408645	0.409904
0.43	0.411165	0.412426	0.413687	0.414949	0.416211	0.417473	0.418736	0.419998	0.421261	0.422525
0.44	0.423788	0.425052	0.426316	0.427582	0.428846	0.430112	0.431378	0.432645	0.433911	0.435178
0.45	0.436445	0.437712	0.438979	0.440246	0.441514	0.442782	0.444050	0.445318	0.446587	0.447857
0.46	0.449125	0.450394	0.451663	0.452932	0.454201	0.455472	0.456741	0.458012	0.459283	0.460554
0.47	0.461825	0.463096	0.464367	0.465638	0.466910	0.468182	0.469453	0.470725	0.471997	0.473269
0.48	0.474541	0.475814	0.477086	0.478358	0.479631	0.480903	0.482176	0.483449	0.484722	0.485995
0.49	0.487269	0.488542	0.489814	0.491087	0.492360	0.493633	0.494906	0.496179	0.497452	0.498726
0.50	0.500000	0.501274	0.502548	0.503821	0.505094	0.506367	0.507640	0.508913	0.510186	0.511458
0.51	0.512731	0.514005	0.515278	0.516551	0.517824	0.519097	0.520369	0.521642	0.522914	0.524186
0.52	0.525459	0.526731	0.528003	0.529275	0.530547	0.531818	0.533090	0.534362	0.535633	0.536904
0.53	0.538175	0.539446	0.540717	0.541988	0.543259	0.544528	0.545799	0.547008	0.548337	0.549606
0.54	0.550875	0.552143	0.553413	0.554682	0.555950	0.557218	0.558486	0.559754	0.561021	0.562288
0.55	0.563555	0.564822	0.566089	0.567355	0.568622	0.569888	0.571154	0.572418	0.573684	0.574948
0.56	0.576212	0.577475	0.578739	0.580002	0.581264	0.582527	0.583789	0.585051	0.586313	0.587574
0.57	0.588835	0.590096	0.591355	0.592616	0.593875	0.595134	0.596392	0.597650	0.598908	0.600166
0.58	0.601423	0.602680	0.603937	0.605192	0.606447	0.607702	0.608956	0.610210	0.611463	0.612717
0.59	0.613970	0.615222	0.616474	0.617726	0.618976	0.620226	0.621476	0.622725	0.623974	0.625222
0.60	0.626470	0.627718	0.628964	0.630210	0.631455	0.632700	0.633944	0.635189	0.636432	0.637675
0.61	0.638918	0.640160	0.641401	0.642641	0.643881	0.645121	0.646360	0.647598	0.648836	0.650074
0.62	0.651310	0.652545	0.653780	0.655015	0.656249	0.657481	0.658714	0.659946	0.661177	0.662407
0.63	0.663637	0.664866	0.666095	0.667322	0.668549	0.669775	0.671001	0.672226	0.673450	0.674674
0.64	0.675896	0.677119	0.678340	0.679561	0.680781	0.681999	0.683217	0.684434	0.685650	0.686866
0.65	0.688082	0.689295	0.690508	0.691720	0.692932	0.694143	0.695354	0.696562	0.697772	0.698979
0.66	0.700186	0.701392	0.702597	0.703802	0.705005	0.706207	0.707409	0.708610	0.709809	0.711008
0.67	0.712205	0.713402	0.714599	0.715793	0.716987	0.718180	0.719373	0.720563	0.721753	0.722942
0.68	0.724131	0.725318	0.726505	0.727690	0.728874	0.730058	0.731240	0.732422	0.733603	0.734782
0.69	0.735961	0.737137	0.738313	0.739488	0.740662	0.741835	0.743008	0.744178	0.745348	0.746517
0.70	0.747685	0.748852	0.750017	0.751181	0.752345	0.753506	0.754667	0.755827	0.756984	0.758141
0.71	0.759297	0.760452	0.761605	0.762758	0.763909	0.765059	0.766209	0.767356	0.768502	0.769648
0.72	0.770791	0.771935	0.773076	0.774217	0.775355	0.776493	0.777629	0.778765	0.779898	0.781030
0.73	0.782161	0.783292	0.784420	0.785547	0.786674	0.787798	0.788921	0.790043	0.791163	0.792282
0.74	0.793400	0.794517	0.795632	0.796747	0.797859	0.798969	0.800078	0.801186	0.802291	0.803396
0.75	0.804499	0.805600	0.806701	0.807800	0.808898	0.809993	0.811088	0.812180	0.813271	0.814361
0.76	0.815450	0.816537	0.817622	0.818706	0.819788	0.820869	0.821947	0.823024	0.824100	0.825175
0.77	0.826247	0.827318	0.828387	0.829454	0.830520	0.831584	0.832647	0.833708	0.834767	0.835824
0.78	0.836880	0.837934	0.838987	0.840037	0.841085	0.842133	0.843178	0.844221	0.845263	0.846303
0.79	0.847341	0.848378	0.849413	0.850446	0.851476	0.852506	0.853532	0.854557	0.855581	0.856602
0.80	0.857622	0.858639	0.859655	0.860668	0.861680	0.862690	0.863698	0.864704	0.865708	0.866709
0.81	0.867710	0.868708	0.869704	0.870698	0.871690	0.872679	0.873667	0.874653	0.875636	0.876618
0.82	0.877597	0.878575	0.879550	0.880523	0.881494	0.882462	0.883428	0.884393	0.885354	0.886314

Table E-3
Partial volumes in horizontal cylinders—cont'd

0.83	0.887272	0.888227	0.889180	0.890131	0.891080	0.892027	0.892971	0.893913	0.894853	0.895789
0.84	0.896725	0.897657	0.808586	0.899514	0.900440	0.901362	0.902283	0.903201	0.904116	0.905029
0.85	0.905939	0.906847	0.907754	0.908657	0.909557	0.910455	0.911350	0.912244	0.913134	0.914021
0.86	0.914906	0.915788	0.916668	0.917544	0.918410	0.919291	0.920159	0.921025	0.921888	0.922749
0.87	0.923607	0.924461	0.925314	0.926164	0.927000	0.927853	0.928693	0.929531	0.930367	0.931198
0.88	0.932028	0.932853	0.933677	0.934497	0.935313	0.936128	0.936938	0.937747	0.938551	0.939352
0.89	0.940150	0.940946	0.941738	0.942526	0.943312	0.044095	0.944874	0.945649	0.946421	0.947190
0.90	0.947956	0.948717	0.949476	0.950232	0.950983	0.951732	0.952477	0.953218	0.953957	0.954690
0.91	0.955421	0.956148	0.956871	0.957590	0.958306	0.959019	0.959757	0.960431	0.961133	0.961829
0.92	0.962522	0.963211	0.963896	0.964577	0.9665253	0.965927	0.966595	0.967260	0.967919	0.968579
0.93	0.969228	0.969876	0.970519	0.971158	0.971792	0.972422	0.973048	0.973669	0.974285	0.974897
0.94	0.975504	0.976106	0.976704	0.977297	0.977885	0.978467	0.979045	0.979618	0.980187	0.980750
0.95	0.981308	0.981859	0.982407	0.982948	0.983485	0.984015	0.984541	0.985060	0.985573	0.986081
0.96	0.986583	0.987080	0.987568	0.988053	0.988530	0.989001	0.989466	0.989924	0.990375	0.990821
0.97	0.991258	0.991690	0.992114	0.992530	0.992939	0.993340	0.993733	0.994119	0.994497	0.994866
0.98	0.995227	0.995579	0.995923	0.996257	0.996581	0.996896	0.997200	0.997493	0.997777	0.998048
0.99	0.998308	0.998555	0.998788	0.999008	0.999212	0.999400	0.999571	0.999721	0.999849	0.999047
1.00	1.000000									

Reprinted by permission of AISI, Committee of Steel Plate Producers and Steel Plate Fabricators Association, Inc. from Steel Plate Engineering Data, Volume 2

Appendix F: Vessel Nomenclature and Definitions

Types of Vessels

Shop-Fabricated Pressure Vessels

1. Process vessels
   a. Trayed columns
   b. Reactors
   c. Packed columns
2. Drums and miscellaneous vessels
   a. Horizontal
   b. Vertical
3. Storage vessels
   a. Bullets
   b. Spheres

Field-Fabricated Pressure Vessels

- Any of the above listed vessels can be field fabricated; however, normally only those vessels that are too large to transport in one piece are field fabricated.
- Although it is significantly more expensive to field fabricate a vessel, the total installed cost may be cheaper than a shop fabricated vessel that is erected in a single piece due to the cost of transportation and erection.
- There are always portions of field fabricated vessels that are shop fabricated. These can be as small as nozzle assemblies or as large as major vessel portions.

Classification of Vessels

Function: Type of vessel, i.e., reactor, accumulator, column, or drum

Material: Steel, cast iron, aluminum, etc.

Fabrication Method: Field/shop fabricated, welded, cast forged, multi-layered, etc.

Geometry: Cylindrical, spherical, conical, etc.

Pressure: Internal, external, atmospheric

Heating Method: Fired or unfired

Orientation: Vertical, horizontal, sloped

Installation: Fixed, portable, temporary

Wall Thickness: Thin/thick walled

Example: Vertical, unfired, cylindrical, stainless steel, heavy-walled, welded reactor for internal pressure

Vessel Parts

Vessel Heads (End Closures)

1. Types
   a. Hemi
   b. Elliptical
   c. Torispherical (flanged and dished)
   d. Conical, toriconical

   e. Flat (bolted or welded)
   f. Miscellaneous (flanged and flued)
   g. Spherically dished covers
   h. Closures (T-bolt, finger pin, quick opening)
2. Types of manufacture
   a. Pressed
   b. Spun
   c. Bumped
   d. Forged
   e. Hot or cold formed
3. Terminology
   a. Knuckle radius
   b. Crown radius
   c. Dished portion
   d. Straight flange

Vessel Supports

1. Types
   a. Skirt (straight or conical)
   b. Legs (braced or unbraced)
   c. Saddles (attached or loose)
   d. Rings
   e. Lugs
   f. Combination (lugs and legs, rings and legs, rings and skirt)

Nozzles

1. Types
   a. Integrally reinforced
   b. Built-up construction
   c. Pad type (studding outlet)
   d. Sight glasses
   e. Elliptical manways
2. Types of service
   a. Manways
   b. Inspection openings
   c. PSV
   d. Instrument connections
   e. Vents
   f. Drains
   g. Process connections

Flanges

1. Types
   a. Slip on
   b. Weld neck, long weld neck
   c. Lap joint
   d. Blind
   e. Screwed
   f. Plate flanges
   g. Studding outlets
   h. Reverse-type flange
   i. Reducing flange
   j. Graylock hub connector
   k. Socket weld
2. Flange Facing
   a. Flat face
   b. Raised face
   c. Finish (smooth, standard, serrated)
   d. Ring joint
   e. Tongue and groove
   f. Male and female

Gaskets

1. Types
   a. Ring, non-asbestos sheet
   b. Flat metal
   c. Spiral wound
   d. Metal jacketed
   e. Corrugated metal
   f. Rings (hexagonal or oval)
   g. Yielding metal gaskets (lens ring, delta ring, rectangular ring)
   h. Elastomeric (rubber, cork, etc.)

Internals

1. Types
   a. Trays, seal pans
   b. Piping distributors
   c. Baffles
   d. Demisters
   e. Packing
   f. Liquid distributors
   g. Vortex breakers
   h. Bed supports
   i. Coils

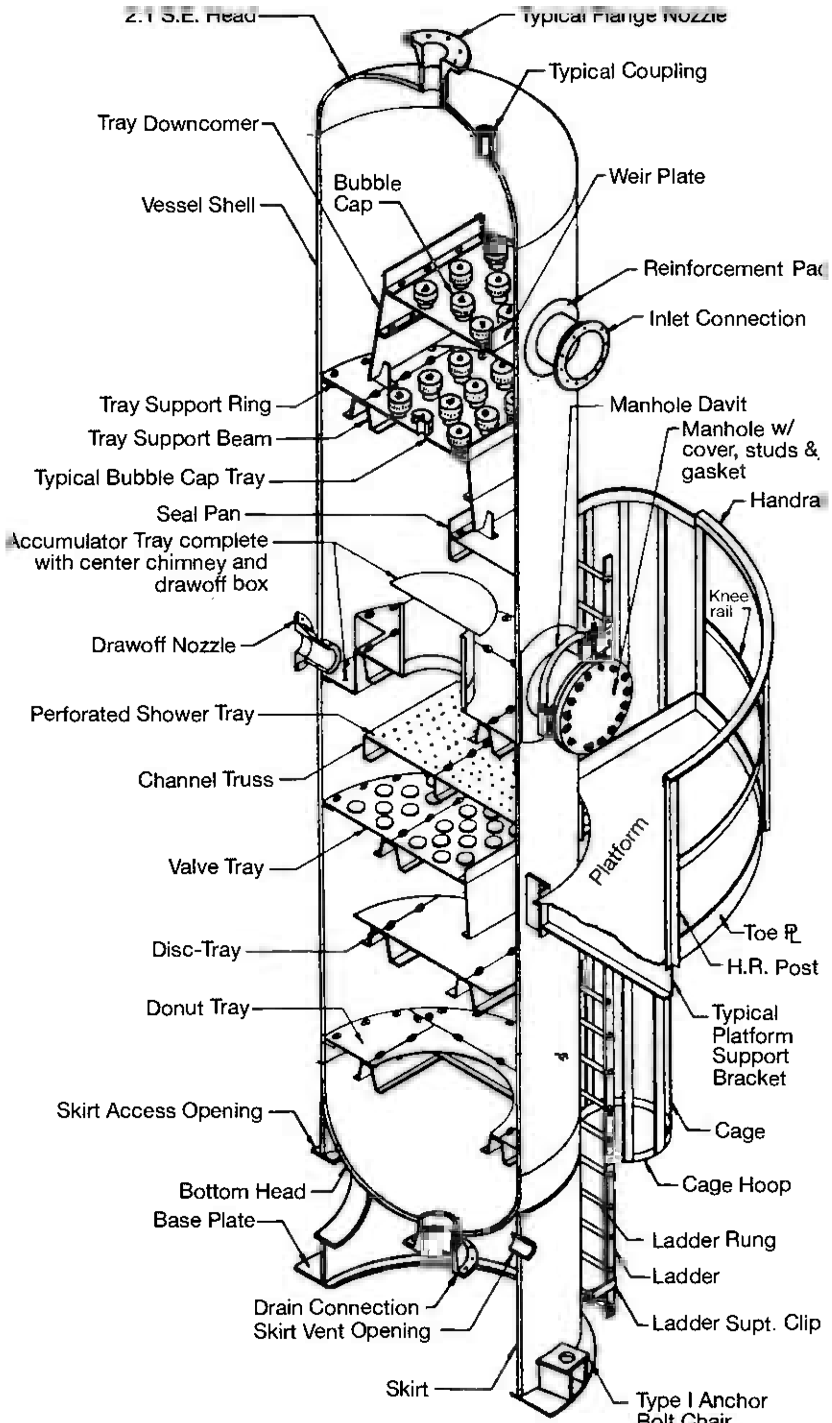


Figure F-1. Typical trayed column.

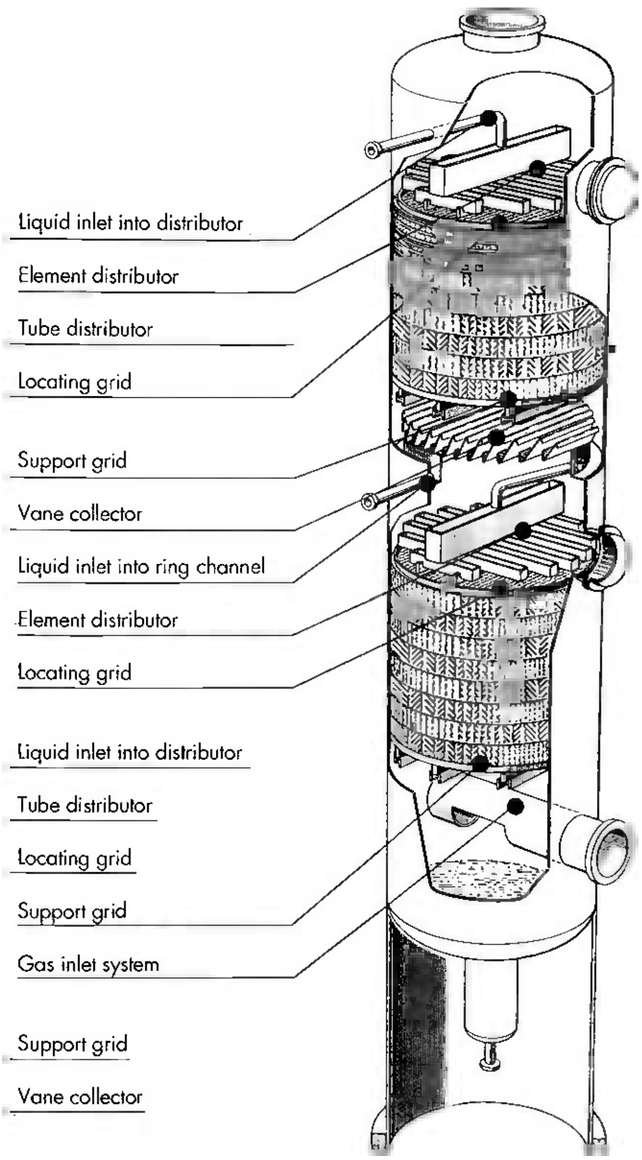


Figure F-2. Typical packed column.

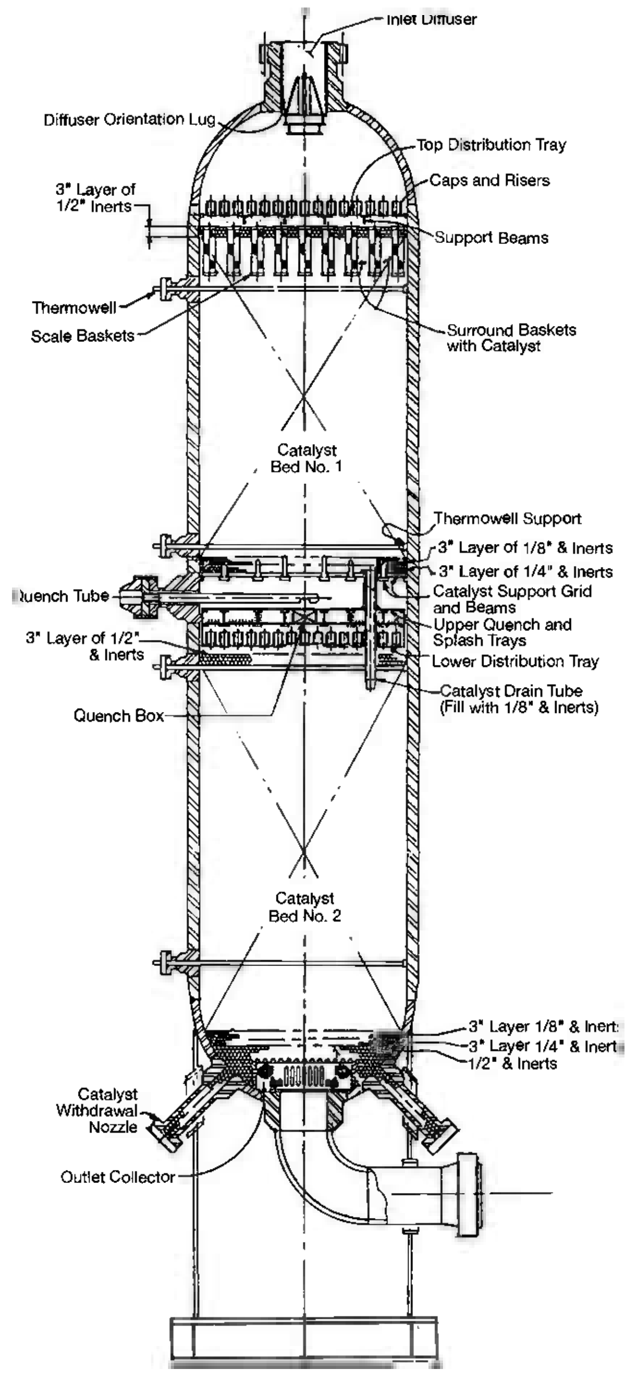
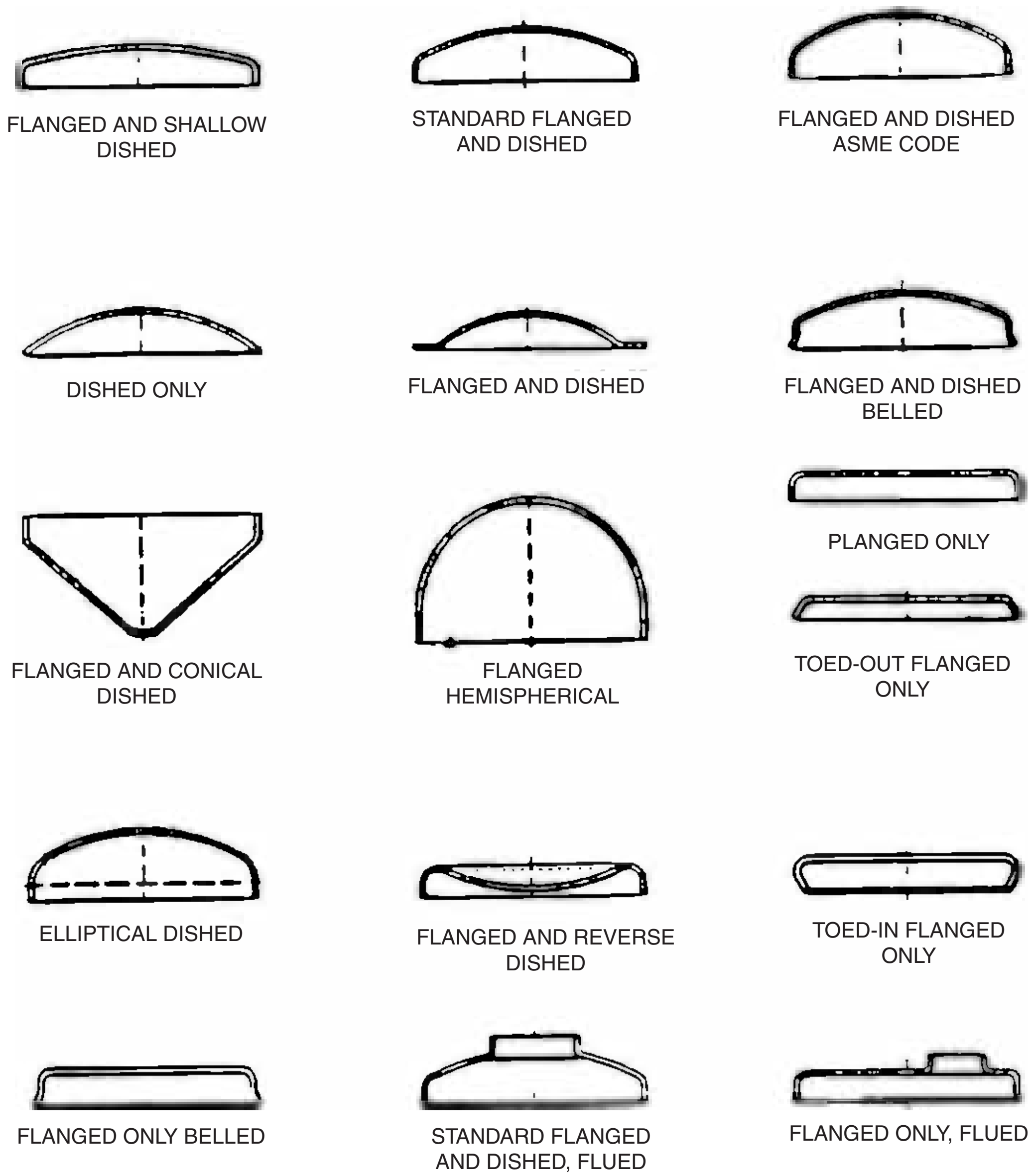
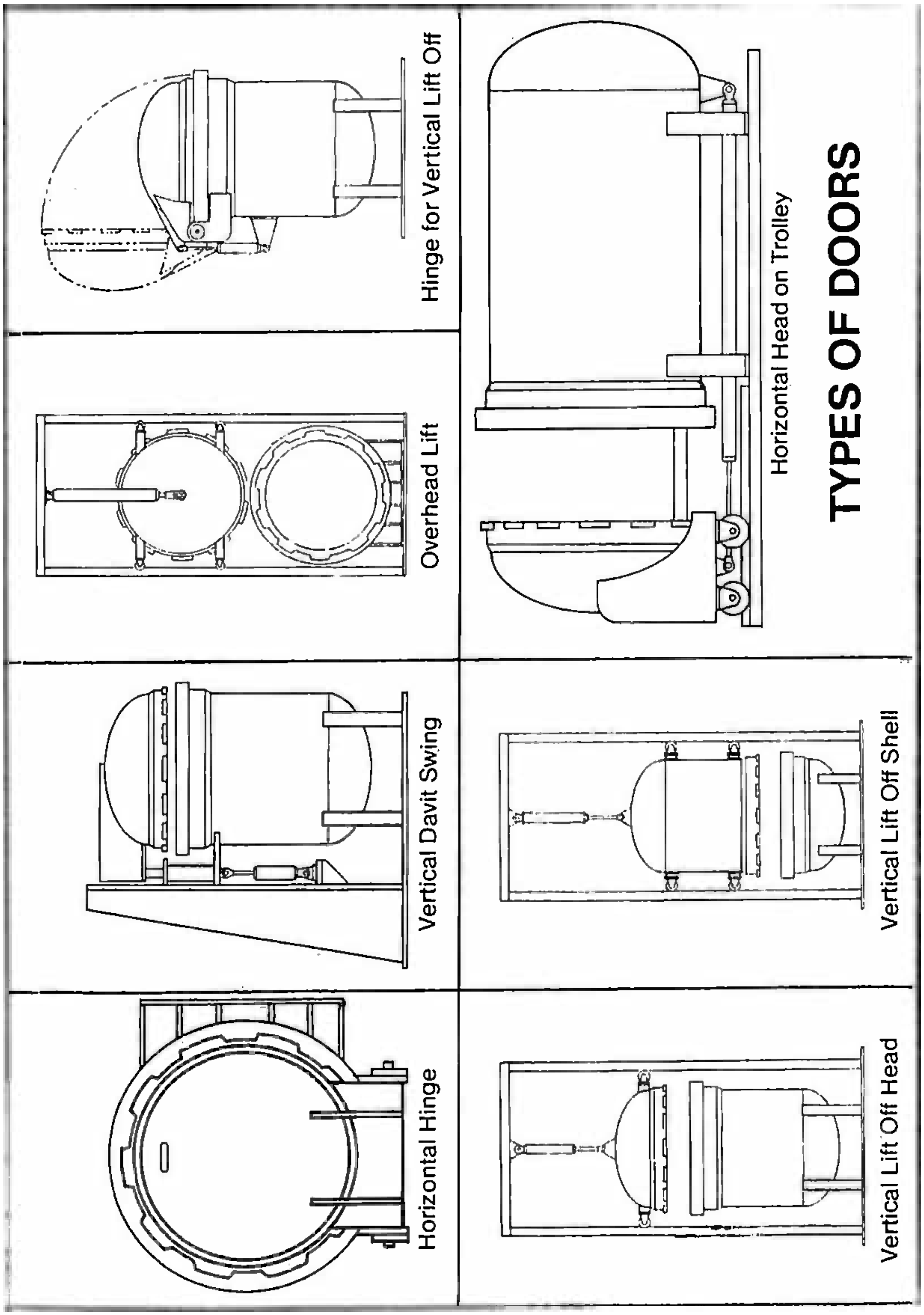


Figure F-3. Typical reactor internals.



Commonly used formed closure heads


TYPES OF DOORS
Hinge for Vertical Lift Off
Overhead Lift
Vertical Davit Swing
Horizontal Hinge
Horizontal Head on Trolley
Vertical Lift Off Shell
Vertical Lift Off Head

Glossary of Vessels Parts

Anchor Bolt Chairs: Gussets and plates welded to base plate and skirt to provide for anchor bolt attachment.

Anchor Bolts: Bolts embedded in concrete foundation and bolted to vessel anchor bolt chairs.

Base Plate: Flat plate welded to the bottom of vessel supports and bearing on the foundation.

Chimney Tray: A tray composed of chimneys extending above the liquid level of the tray, permitting passage of the vapors upward. The tray collects and removes all liquid product from a specific portion of the vessel.

Column Davit: A hoisting device attached by means of a socket to the top of fractionation columns. Used for handling relief valves, bubble trays, bubble caps, etc.

Conical Head: Head formed in the shape of a cone.

Coupling: A fitting welded into the vessel to which the piping is connected either by screwing or welding. This type of fitting is generally used for pipe sizes 1½ in. and smaller.

Distributor Tray: A perforated tray that provides equal distribution of liquid over the vessel area. Risers on the tray extend above the liquid level to permit passage of vapors rising upward.

Downcomers: Rectangular flat plates bolted, welded or clamped to shell and trays inside of fractionation columns. Used to direct process liquid and to prevent bypassing of vapor.

Flanged and Dished (Torispherical) Head: Head formed using two radii, one radius called crown radius, and another called knuckle radius, which is tangent to both the crown radius and the shell.

Flanges (or Pipe Flanges): Fittings used to connect pipes by bolting flanges together.

Flat Head (or Cover Plate): Flat plate welded or bolted to the end of a shell.

Fractionating Trays: Circular flat plates bolted, welded or clamped to rings on the inside of fractionation columns. Used to obtain vapor liquid contact, which results in fractionation.

Head: The end closure of a vessel.

Hemispherical Head: Head formed in the shape of a half sphere.

Insulation Rings: Rings made of flat bar or angle attached around the girth (circumference) of vertical vessels. Used to support the weight of the vessel insulation.

Ladders and Cages: Rung-type ladders with cages built of structural shapes to prevent a person from falling when climbing the ladder. Bolted to and supported by clips on the outside of the vessel. Used for vertical access to the platforms.

Manhole Hinges or Davits: Hinges or davits attached to manhole flange and cover plate which allow cover plate to swing aside from the manhole opening.

Mist Eliminator (or Demister): A wire mesh pad held in place between two light grids. The mist eliminator disengages liquids contained in the vapor.

Nozzle: Generally consists of a short piece of pipe welded in the shell or head with a flange at the end for bolting to the Piping.

Pipe Supports and Guides: Supports and guides for attached piping that is bolted to clips, which are welded to the vessel.

Platforms: Platforms bolted to and supported by clips on the outside of the vessel. Generally located just below a manhole, at relief valves, and other valves or connections that need frequent service.

Reinforcing Pad: Plate formed to the contour of shell or head, welded to nozzle and shell or head.

Saddles: Steel supports for horizontal vessels.

Seal Pans: Flat plates bolted, welded, or clamped to rings inside of fractionation column shell below downcomer of lowest tray. Used to prevent vapor from bypassing up through the downcomer by creating a liquid seal.

Shell: The cylindrical portion of a vessel.

Skirt: Cylinder similar to shell, which is used for supporting vertical vessels.

Skirt Access Opening: Circular holes in the skirt to allow workers to clean, inspect, etc., inside of skirt.

Skirt Fireproofing: Brick or concrete applied inside and outside of skirt to prevent damage to skirt in case of fire.

Skirt Vents: Small circular holes in the skirt to prevent collection of dangerous gases within the skirt.

Stub-end: A short piece of pipe or rolled plate welded into the vessel to which the piping is connected by welding.

Support Grid: Grating or some other type of support through which vapor or liquid can pass. Used to support tower packing (catalyst, raschig rings, etc.).

Support Legs: Legs made of pipe or structural shapes that are used to support vertical vessels.

Toriconical Head: Head formed in the shape of a cone and with a knuckle radius tangent to the cone and shell.

2:1 Semi-Elliptical Head: Head formed in the shape of a half ellipse with major to minor axis ratio of 2:1.

Vacuum Stiffener Rings: Rings made of flat bar or plate, or structural shapes welded around the circumference of the vessel. These rings are installed on vessels operating under external pressure to prevent collapse of the vessel. Also used as insulation support rings.

Vessel Manhole: Identical to a nozzle except it does not bolt to piping and it has a cover plate (or blind flange), which is bolted to the flange. When unbolted it allows access to the inside of the vessel. Generally 18 in. or larger in size.

Vortex Breaker: A device located inside a vessel at the outlet connection. Generally consisting of plates welded together to form the shape of a cross. The vortex breaker prevents cavitation in the liquid passing through the outlet connection.

Definitions

1. Butt Joint: A butt joint is a connection between two members with a full penetration weld.
2. Corner Joint: A corner joint is a connection between two members at right angles to each other that is made with a full penetration weld, partial penetration weld or fillet welds.
3. Angle Joint: An angle joint is a connection between the edges of two members with a full penetration weld with one of the members consisting of a transition of diameter.
4. Spiral weld: A weld joint having a helical seam.
5. Fillet weld: A fillet weld is a weld that is approximately triangular in cross section that joins two surfaces at approximately right angles to each other.
6. Gross Structural Discontinuity: A gross structural discontinuity is a source of stress or strain intensification which affects a relatively large portion of a structure and has a significant effect on the overall stress or strain pattern or on the structure as a whole. Examples of gross structural discontinuities are as follows;
   a. Head to shell junctions
   b. Flange to shell junctions
   c. Nozzles
   d. Junctions between shells of different diameters or thickness
7. Lightly Loaded Attachments: A lightly loaded attachment is one where the weld stress due to mechanical loads is not over 25% of the allowable stress for fillet welds and the temperature difference between the shell and attached member is not more than 14° C (25° F).
8. Minor Attachments: Minor attachments are small parts attached to the pressure boundary that carry no load or insignificant load. They are less than 3/8 in (10 mm) in thickness or less than 5 in^3 (82 cm^3) in size. Some examples are nameplates, insulation supports and locating lugs.
9. Major Attachments: Any part that is not a minor attachment or lightly loaded by definition.
10. Design Pressure: The pressure used in the design of a vessel component together with the coincident design metal temperature, for the purpose of determining the minimum permissible thickness or physical characteristics of the different zones of the vessel. Where applicable, the static head and other static or dynamic loads shall be included in addition to the specified design pressure in the determination of the minimum permissible thickness or physical characteristics of a particular zone of the vessel.
11. Maximum Allowable Working Pressure (MAWP): The maximum gage pressure permissible at the top of a completed vessel in its normal operating position in the hot and corroded condition. The pressure is the least of the values for the internal or external pressure to be determined by the Code rules for any of the pressure boundary parts, considering static head, using nominal thicknesses.
12. Maximum Allowable Pressure (MAP): Not an ASME Code requirement or definition. This is an "optional" pressure that was historically used as a basis for calculating test pressures. The MAP is the maximum pressure allowed in the new and cold condition (N&C). This is calculated from the ASME equations using the new nominal thickness and the allowable stress at ambient temperature.
13. Test Pressure: The test pressure is the pressure applied at the top of the vessel during the test. The test can be either hydrotest or pneumatic test.
14. Maximum Allowable Temperature (MAT): This is not an ASME Code definition and is not required to be calculated by the ASME Code or stamped on the nameplate. However, certain client specifications request that the stamping of the nameplate include the highest permissible temperature for which the vessel is acceptable. The MAT is the maximum allowable temperature allowed for the vessel and shall be the lesser of the following;
    a. The highest temperature corresponding to the maximum allowable tensile stress. Used when tensile stress is governing all thicknesses.

b. The highest temperature corresponding to the maximum allowable compressive stress.
c. The highest temperature allowed based on the flange pressure and temperature ratings.

15. Minimum Design Metal Temperature (MDMT): The MDMT is an ASME Code requirement and must be stamped on the nameplate. the lower of the following;
    a. The lowest temperature expected in normal service (Service MDMT).
    b. The qualification of temperature based on charpy impact testing of the base materials (Qualified MDMT).
    c. The lowest ambient temperature of the atmospheric conditions (Arbitrary MDMT).
16. Nil Ductility Transition Temperature (NDT): Also known as DBTT (ductile-brittle transition temperature): The temperature above which the material is predominantly ductile and below which it is predominantly brittle. The NDT represents the point at which the fracture energy passes below a predetermined point, i.e. 15 ft-lbs (20 joules) for ordinary steel or 40 ft-lbs (54 joules) for Cr-Mo steels.
17. Fracture Appearance Transition Temperature (FATT): On a toughness curve (a plot of impact values at various temperatures), it is the point at which the specimen fracture surface is 50% shear (brittle fracture) and 50% cleavage (brittle fracture).
18. Minimum Pressurization Temperature (MPT): This is not an ASME Code requirement and is not stamped on the nameplate. This is the lowest temperature that will allow full pressurization for a vessel that has been subject to the long term effects of embrittlement. Embrittlement can be the result of temperature, hydrogen or irradiation. The temperature is either determined by calculation or testing methods.

Bolting Definitions

19. ***Cut Threads***: Threads may be either cut or rolled. Cut threads are machined from a piece of bar stock by removal of material by machining. The distinctive feature being that the shank of the bolt is the same diameter as the OD of the threads.
20. ***Rolled Threads***: Rolled threads are formed from a piece of round bar stock by die rolling or upsetting the threads. Rolled threads are cheaper to manufacture than cut threads. No machining, therefore no waste. The distinctive feature is that the shank is always slightly smaller in diameter than the OD of the threads. Thus, to make a 1 inch dia bolt, you start with a 0.913 inch diameter bar.
21. ***Strain Hardening***: This effect is produced in austenitic stainless steels by reducing over-sized bars to the desired size by cold drawing or other process. Strain hardening is the increase in strength and hardness that results from plastic deformation below the recrystallization temperature (cold work). The degree of strain hardening achievable in any alloy is limited by its strain hardening characteristics and the amount of reduction.
22. ***Carbide Solution Treated***: Carbide solution treatment is a heat treatment for austenitic stainless steel materials equivalent to solution annealing. The purpose is to reheat materials to a level that will cause chromium carbides to go into solution. This is followed by a rapid quench that prevents the precipitation of carbides into the grain boundaries where it would have a detrimental effect.
23. ***Quench & Tempered***: A two step heat treatment for carbon and low alloy steels. The first step is to quench the material to develop hardness and strength. This is followed by a subsequent treatment that tempers the material to develop toughness. Since high hardness and strength are often brittle, the quenching reforms the material to restore toughness while retaining some of the strength, thus giving the material the right balance of properties.
24. ***Precipitation Hardening***: Precipitation hardening is only possible in certain alloys, such as 17-7 SST, maraging steel, various aluminum alloys, copper, nickel (Inconel 718 & Inconel x-750) and magnesium. It is hardening of a material caused by the precipitation of a constituent to form a separate phase which is an intermetallic compound. This can occur at room temperature or at elevated temperatures. The solute will precipitate (leave the supersaturated condition) by either migration or diffusion. By this method the tensile strength of 2024 Al can be doubled from 30 ksi to 60 ksi. The heat treatment is usually a two step process. Step one is a solution heat treatment followed by a rapid quench. Step two, is an aging or precipitation treatment to cause separation of a second phase.

Appendix G: Useful Formulas for Vessels [1,2]

1. Properties of circle. (See Figure G-1.)
   - *C.G. of area.*

$$e_1 = \frac{C^3}{12A_1}$$

$$e_2 = \frac{120C}{\alpha\pi}$$

$$e_3 = \frac{38.197\left(R^3 - r^3\right)\sin\phi/2}{(R^2 - r^2)\phi/2}$$

   - *Chord, C.*

$$C = 2R\sin\frac{\theta}{2}$$

$$C = 2\sqrt{2bR - b^2}$$

   - *Rise, b.*

$$b = 0.5C\tan\frac{\theta}{4}$$

$$b = R - 0.5\sqrt{4R^2 - C^2}$$

   - *Angle, θ*

$$\theta = 2\arcsin\frac{C}{2R}$$

   - *Area of sections.*

$$A_1 = \frac{\theta\pi R^2 - 180C(R - b)}{360}$$

$$A_2 = \frac{\pi R^2\alpha}{360}$$

$$A_3 = \frac{\left(R^2 - r^2\right)\pi\phi}{360}$$

2. Properties of a cylinder.
   - *Cross-sectional metal area, A.*

$$A = 2\pi R_m t$$

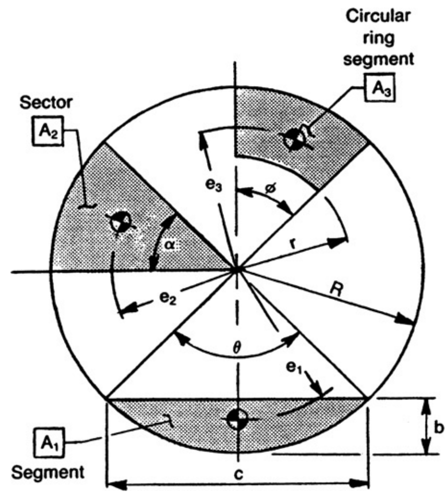


Figure G-1. Dimensions and areas of circular sections.

   - *Section modulus, Z.*

$$Z = \pi R_m^2 t = \frac{\pi D_m^2 t}{4} = \frac{\pi\left(D^4 - d^4\right)}{32d}$$

   - *Polar moment of inertia, J.*

$$J = \frac{\pi\left(D^4 - d^4\right)}{32}$$

   - *Moment of inertia, I.*

$$I = \pi R_m^3 t = \frac{\pi D_m^3 t}{8} = \frac{\pi\left(D^4 - d^4\right)}{64}$$

   - *Radius of gyration, r.*

$$r = \sqrt{\frac{I}{A}}$$

3. Radial displacements due to internal pressure.
   - *Cylinder*.

$$\delta = \frac{PR^2}{Et}(1 - 0.5v)$$

   - *Cone*.

$$\delta = \frac{PR^2}{Et \cos \alpha}(1 - 0.5v)$$

   - *Sphere/hemisphere*.

$$\delta = \frac{PR^2}{2Et}(1 - v)$$

   - *Torispherical/ellipsoidal*.

$$\delta = \frac{R}{E}\left(\sigma_\phi - v\sigma_x\right)$$

where P = internal pressure, psi
R = inside radius, in.
t = thickness, in.
v = Poisson's ratio (0.3 for steel)
E = modulus of elasticity, psi
α = ½ apex angle of cone, degrees
σ_ϕ = circumferential stress, psi
σ_x = meridional stress, psi

4. Longitudinal stress in a cylinder due to longitudinal bending moment, M_L.
   - *Tension*

$$\sigma_x = \frac{M_L}{\pi R^2 t}$$

   - *Compression*

$$\sigma_x = (-)\frac{M_L}{\pi R^2 t}$$

where
R = inside radius, in.
M_L = bending moment, in.-lb
t = thickness, in.

5. Thickness required heads due to external pressure.

$$t_h = \frac{L}{\sqrt{\frac{E}{16P_e}}}$$

where L = crown radius, in.
P_e = external pressure, psi
E = modulus of elasticity, psi

6. Equivalent pressure of flanged connection under external loads.

$$P_e = \frac{16M}{\pi G^3} + \frac{4F}{\pi G^2} + P$$

where P = internal pressure, psi
F = radial load, lb
M = bending moment, in.-lb
G = gasket reaction diameter, in.

7. Bending ratio of formed plates.

$$\% = \frac{100t}{R_f}\left(1 - \frac{R_f}{R_o}\right)$$

where R_f = finished radius, in.
R_o = starting radius, in. (∞ for flat plates)
t = thickness, in.

8. Stress in nozzle neck subjected to external loads.

$$\sigma_x = \frac{PR_m}{2t_n} + \frac{F}{A} + \frac{MR_m}{I}$$

where R_m = nozzle mean radius, in.
t_n = nozzle neck thickness, in.
A = metal cross-sectional, area, in.^{2}
I = moment of inertia, in.^{4}
F = radial load, lb
M = moment, in.-lb
P = internal pressure, psi

9. Circumferential bending stress for out of round shells [2].

$$D_1 - D_2 > 1\%\ D_{nom}$$

$$R_1 = \frac{D_1 + D_2}{2}$$

$$R_a = \frac{D_1 + D_2}{4} + \frac{t}{2}$$

$$\sigma_b = \frac{1.5PR_1t(D_1 - D_2)}{t^3 + 3\left(\frac{P}{E}\right)R_1R_a^2}$$

where D_1 = maximum inside diameter, in.
D_2 = minimum inside diameter, in.
P = internal pressure, psi
E = modulus of elasticity, psi
t = thickness, in.

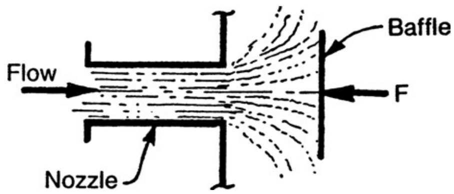


Figure G-2. Typical nozzle configuration with internal baffle.

10. Equivalent static force from dynamic flow.

$$F = \frac{V^2Ad}{g}$$

where F = equivalent static force, lb
V = velocity, ft/sec
A = cross-sectional area of nozzle, ft^2
d = density, lb/ft^3
g = acceleration due to gravity, 32.2 ft/sec^2

11. Allowable compressive stress in cylinders [1].

$$\text{If } \frac{t}{R} \le 0.015,\ X = \frac{10^6 t}{R}\left(2 - \frac{200t}{3R}\right)$$

$$\text{If } \frac{t}{R} > 0.015,\ X = 15{,}000$$

$$\text{If } \frac{L}{R} \le 60,\ Y = 1$$

$$\text{If } \frac{L}{R} > 60,\ Y = \frac{21{,}600}{18{,}000 + \left(\frac{L}{R}\right)^2}$$

$$F_a = \frac{Q}{A} = XY$$

where t = thickness, in.
R = outside radius, in.
L = length of column, in.
Q = allowable load, lb
A = metal cross-sectional area, in.^{2}
F_a = allowable compressive stress, psi

12. Unit stress on a gasket, S_g.

$$S_g = \frac{A_b S_a}{.785\left[(d_o - .125)^2 - d_i^2\right]}$$

where A_b = area of bolt, in.^{2}
d_o = O.D. of gasket, in.
d_i = I.D. of gasket, in.
S_a = bolt allow. stress, psi

13. Determine fundamental frequency of a vertical vessel on skirt, f.

$$I = \frac{\pi D_m^3 t}{8}$$

$$m = \frac{\pi D_m t d}{g}$$

$$f = \frac{.560}{(12H)^2}\sqrt{\frac{EI}{m}}$$

where I = moment of inertia, in.^{4}
D_m = mean vessel diameter, in.
t = vessel thickness, in.
d = density of steel 0.2833 lbs/in.^{3}
g = acceleration due to gravity, 386 in./sec^2
E = modulus of elasticity, psi
H = vessel height, ft
m = mass of vessel per unit length, lbf-sec^2/in.^{2}
f = fundamental frequency, Hertz (cycles/second)

14. Maximum quantity of holes in a perforated circular plate.

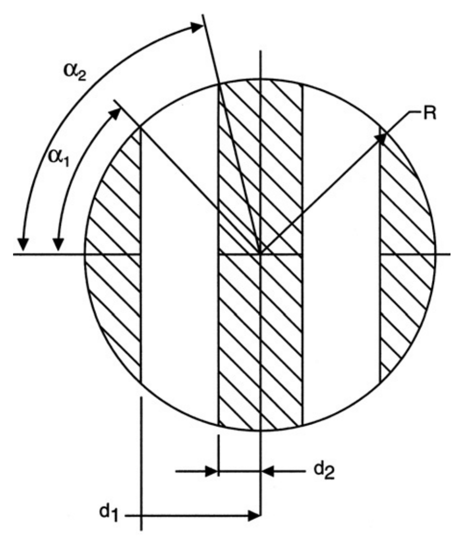
A = area of circular plate, in.^{2}
D = diameter of circular plate, in.
d = diameter of holes, in.
p = pitch, in.
Q = quantity of holes
K = constant (0.86 for triangular pitch)
R = practical physical radius to fully contain all holes

$$A = \pi R^2$$

$$R = \frac{D - d}{2}$$

$$Q = \frac{A}{Kp^2}$$

15. Divide a circle into "N" equal number of parallel areas.

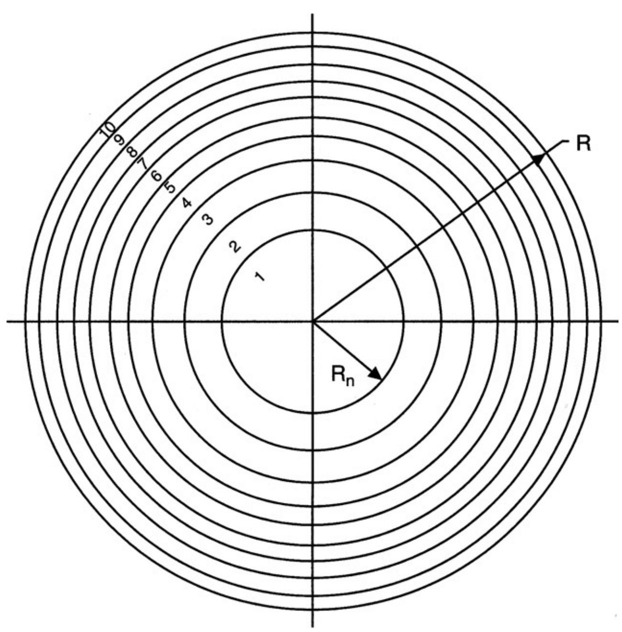


Multiply d_n times R to get actual distances.

Table G-1
Dimensions for equal areas

"N" Areas	α_1	α_2	α_3	d_1	d_2	d_3
3	74.65	NA	NA	0.2647	NA	NA
4	66.18	NA	NA	0.4038	NA	NA
5	60.55	80.9	NA	0.4917	0.1582	NA
6	56.4	74.65	NA	0.5534	0.2647	NA
7	53.2	69.6	83.55	0.599	0.3485	0.1123
8	50.63	66.18	78.6	0.6343	0.4038	0.1977

16. Divide a circle into "N" equal number of circular areas.
    A_T = total area, in.^{2}
    A_n = area of equal part, in.
    R = radius to circle, in.
    R_n = radius to equal part, in.
    N = number of equal parts



$$A = \pi R^2$$

$$A_n = \frac{A_T}{N}$$

$$R_n = \sqrt{\frac{A_n N}{\pi}}$$

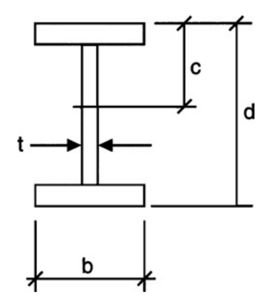
Example: Divide a circle into (10) equal areas.
Answer:
$R_1 = 0.3163R$
$R_2 = 0.4472R$
$R_3 = 0.5477R$
$R_4 = 0.6325R$
$R_5 = 0.7071R$
$R_6 = 0.7746R$
$R_7 = 0.8367R$
$R_8 = 0.8944R$
$R_9 = 0.9487R$
$R_{10} = R$

17. Maximum allowable beam-to-span ratios for beams.
    L = unsupported length, in.
    d = depth of beam, in.
    b = width of beam, in.
    t = thickness of compression flange, in.

If $\frac{Ld}{bt} \le 600$, then the allowable stress = 15,000 psi
If $\frac{Ld}{bt} > 600$,
then the allowable stress $= \frac{9,000,000}{Ld/bt}$

18. Properties of a built-up "I" beam.

$$Z = \frac{td}{6}(6b + d)$$
$$I = ZC$$
t = same thk. for all parts



19. Volume required for gas storage.
    V = volume, in.^{3}
    m = mole weight of contents
    R = gas constant
    T = temperature, Rankine
    P = pressure, psi

$$V = \frac{mRT}{P}$$

Appendix H: Metric Guidelines and Conversions

Table H-1
Metric guideline

Quantity		Typical US Units	SI Units
1	Force	Lbs	N (Newtons)
2	Mass	Lbs	Kg (Kilograms)
3	Distance	Ft, In	M or mm (Meter or Millimeter)
4	Section Modulus	In^3	mm^3
5	Moment of Inertia	In^4	mm^4
6	Moment	Ft-Lbs, In- Lbs	N-M or KN-M (Newton-Meter or Kilo Newton-Meter)
7	Pressure	PSI	Bar, Pa, KPa, Kg/cm^2, N/cm^2
8	Thermal Conductivity	BTU-in/hr/ft^2/°F	W / M- °C (Watts per Meter-Centigrade)
9	Thermal Expansion	In/In/°F	mm/mm/°C
10	Density	PCI, PCF	Kg / mm^3, Kg/M^3
11	Area	Ft^2, In^2	mm^2, M^2
12	Modulus of Elasticity	PSI	MPa, N/M^2, N/mm^2
13	Stress	PSI	Mpa
14	Torque	Ft-Lbs, In-Lbs	N-M
15	Mass per Area	PSF	Kg/M^2
16	Mass per length	Lbs / Ft	Kg / M
17	Volume	Ft^3	M^3
18	Temperature	°F	°C
19	Misc	NA	NA

Table H-2
Metric conversions

Quantity		Unit	Multiply by…	To Obtain…
1	Force	Lbs	4.448222	Newtons
		Kg	9.80666	Newtons
		KN	224.8089	Lbs
		KN	102	Kgs
		N	0.1019716	Kgs
		N	0.2248089	Lbs
2	Mass	Lbs	0.4536	Kg
		Kg	2.204623	Lbs
		Metric Ton (MT)	1000	Kg

(*Continued*)

Table H-2
Metric conversions—cont'd

Quantity		Unit	Multiply by…	To Obtain…
		Metric Ton	2204.623	Lbs
		Metric Ton	1.1023	US Ton
		US Tons	2000	Lbs
		US Tons	907.2	Kg
		US Tons	0.90718	Metric Ton
3	Distance	In	25.4	mm
		Ft	0.3048	M
		M	3.2808	Ft
		M	39.37	In
4	Section Modulus	In^3	16,387.06	mm^3
		mm^3	6.1163 (10^{-5})	In^3
5	Moment of Inertia	In^4	416,231.43	mm^4
		mm^4	2.4025 (10^{-6})	In^4
6	Moment	In-Lbs	11.52124	Kg-mm
		In-Lbs	1.35582	N-M
		Ft-lbs	1.35582 (10^{-3})	KN-M
		N-M	8.85075	In-Lbs
		N-M	0.737561	Ft-Lbs
		KN-M	737.561	Ft-Lbs
		Kg-M	7.233	Ft-Lbs
		N-mm	8.85075 (10^{-3})	In-Lbs
		N-mm	7.375 (10^{-4})	Ft-Lbs
7	Pressure	Bar	1.019716	Kg/Cm^2
		Bar	0.010197	Kg/mm^2
		Bar	14.50377	PSI
		Bar	0.1	Mpa
		Bar	100	Kpa
		Pascals	0.000145038	PSI
		Pascals	1	N/M^2
		KPa	0.1450377	PSI
		KPa	4.014743	In-H_2O
		MPa	145.0377	PSI
		Kg/mm^2	1422.334	PSI
		Kg/mm^2	9806.65	KPa
		Kg/Cm^2	98.0665	KPa
		Kg/Cm^2	393.7115	In-H_2O
		PSI	6.894757	KPa

(*Continued*)

Table H-2
Metric conversions—cont'd

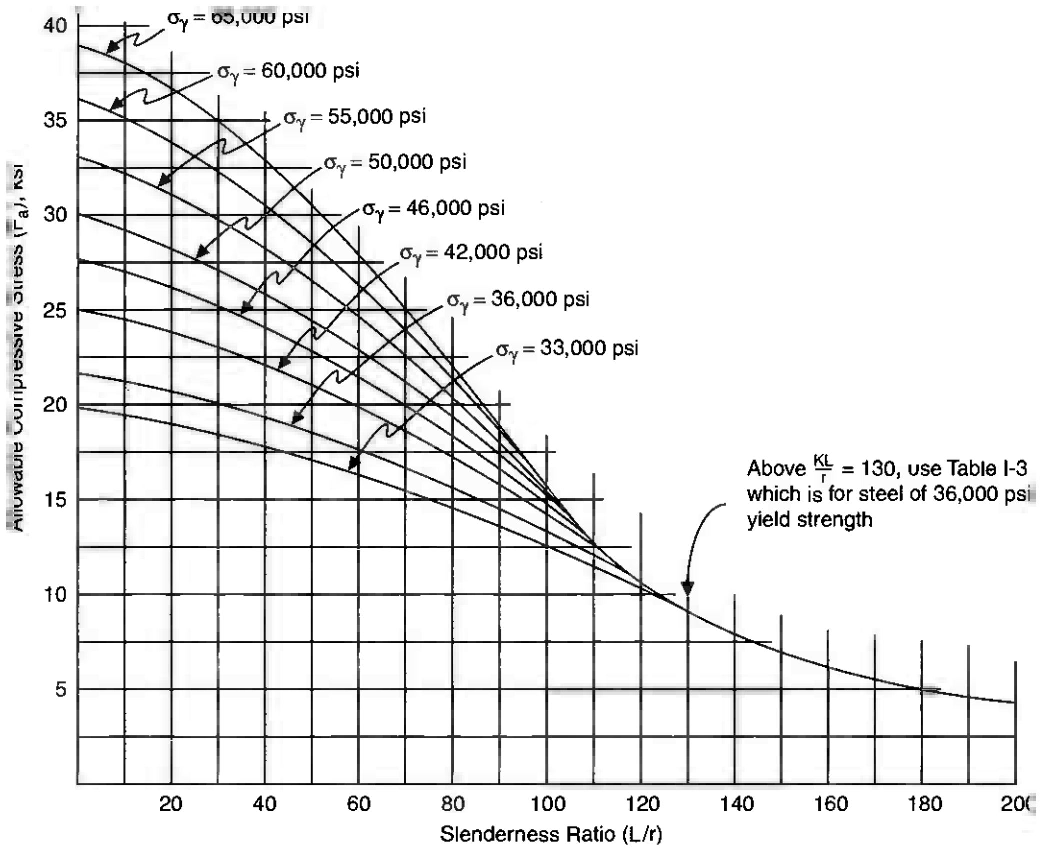
Quantity		Unit	Multiply by…	To Obtain…
		PSI	0.006894757	MPa
		PSI	0.070307	Kg/Cm^2
		PSI	0.000703	Kg/mm^2
		PSI	0.068947	Bar
		Millibar	0.0001	MPa
		Millibar	0.1	KPa
		Millibar	100	Pa
		Millibar	0.01450377	PSI
		Millibar	0.001	Bar
		Millibar	0.02952999	In-Hg
		Millibar	0.401474	In-H2O
		$Newton/Cm^2$	1.450377	PSI
		$Newton/Cm^2$	10	KPa
		$Newton/Cm^2$	0.01	Mpa
		$Newton/mm^2$	145.0377	PSI
		$Newton/mm^2$	1000	KPa
		$Newton/mm^2$	1	MPa
8	Thermal Conductivity	BTU-ft/hr-ft^2-°F	1.731	W/m·K
		W/m·K	0.5777	BTU-ft/hr-ft^2-°F
9	Thermal Expansion	In/In/ °F	1.8	mm/mm/ °C
		mm/mm/ °C	0.5556	In/In/ °F
10	Density	PCI	2.768 (10^{-5})	Kg/mm^3
		PCF	16.01846	Kg/M^3
		Kg/mm^3	36045	PCI
		Kg/Cm^3	62,427.96	PCF
		Kg/M^3	0.062427	PCF
11	Area	mm^2	1.55 (10^{-3})	In^2
		Cm^2	0.155	In^2
		M^2	1550	In^2
		M^2	10.764	Ft^2
		In^2	64,516	mm^2
		In^2	6.4516	Cm^2
		Ft^2	0.0929	M^2
12	Modulus of Elasticity	PSI	0.006894757	MPa
		MPa	145.0377	PSI
13	Stress	PSI	0.006894757	MPa
		MPa	145.0377	PSI

(*Continued*)

Table H-2
Metric conversions—cont'd

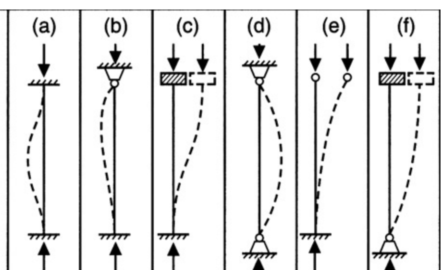
Quantity		Unit	Multiply by…	To Obtain…
14	Torque	SEE MOMENT		
15	Mass / Area	Kg/mm^2	1422.334	PSI
		Kg/Cm^2	204,386.69	PSF
		Kg/M^2	0.2048	PSF
		Kg/M^2	0.001422	PSI
		Kg/M^2	0.0001	Kg/Cm^2
		Kg/M^2	0.001422334	PSI
		Kg/M^2	0.2048161	PSF
		PSF	4.882428	Kg/M^2
		PSI	$7.0307\ (10^{-4})$	Kg/mm^2
		PSI	0.070307	Kg/Cm^2
		N/M^2	1	Pa
16	Mass / Length	Lbs/Ft	14.59385	N/M
		Lbs/ In	0.17513	KN/M
		N/M	0.068522	Lbs/Ft
		KN/M	68.523	Lbs/Ft
17	Volume	In^3	16,387.06	mm^3
		Ft^3	0.028317	M^3
		mm^3	$6.1163\ (10^{-5})$	In^3
		Cm^3	0.061024	In^3
		M^3	35.31467	Ft^3
		M^3	61023.74	In^3
18	Temperature	^oF	1.8 C + 32	
		^oC	.556 (F -32)	
		^oK	C + 273.18	
		^oR	F + 459.72	
19	Misc	Weight of Steel	490 PCF	
			.2833 PCI	
			$7.858\ (10^{-6})$ Kg / mm^3	
			$7.858\ (10^{-3})$ Kg / Cm^3	
		Weight of Water	62.4 PCF	
			1001 Kg/ M^3	
			8.33 Lbs Water /Gal	
		Volume of Contents	7.481 Gallons/ Ft^3	
			5.614 Ft^3/BBL	
			6.29 BBL/ M^3	
			.1781 BBL/ Ft^3	
			42 Gallons/ BBL	
			264.189 Gallons/ M^3	

Appendix I: Allowable Compressive Stress for Columns, F_A



Reprinted by permission of the James F. Lincoln Arc Welding Foundation

Table I-1
End connection coefficients

	(a)	(b)	(c)	(d)	(e)	(f)
Buckled shape of column is shown by dashed line						
Theoretical K value	0.5	0.7	1.0	1.0	2.0	2.0
Recommended design value when ideal conditions are approximated	0.65	0.80	1.2	1.0	2.10	2.0
End condition code		Rotation fixed and translation fixed Rotation free and translation fixed Rotation fixed and translation free Rotation free and translation free				

Reprinted by permission of AISC.

Table I-2
33,000-psi-Yield steel

L/r Ratio		1	2	3	4	5	6	7	8	9
		19,770	19,730	19,690	19,660	19,620	19,580	19,540	19,500	19,460
10	19,410	19,370	19,320	19,280	19,230	19,180	19,130	19,080	19,030	18,980
20	18,930	18,880	18,820	18,770	18,710	18,660	18,600	18,540	18,840	18,420
30	18,360	18,300	18,240	18,180	18,110	18,050	17,980	17,920	17,850	17,780
40	17,710	17,640	17,570	17,500	17,430	17,360	17,290	17,220	17,140	17,070
50	16,990	16,920	16,840	16,760	16,680	16,600	16,520	16,440	16,360	16,280
60	16,200	16,120	16,030	15,950	15,860	15,780	15,690	15,610	15,520	15,430
70	15,340	15,250	15,160	15,070	14,980	14,890	14,800	14,700	14,610	14,510
80	14,420	14,320	14,230	14,130	14,030	13,930	13,840	13,740	13,640	13,530
90	13,430	13,330	13,230	13,130	13,020	12,920	12,810	12,710	12,600	12,490
100	12,380	12,280	12,170	12,060	11,950	11,830	11,720	11,610	11,900	11,380
110	11,270	11,150	11,040	10,920	10,800	10,690	10,570	10,450	10,330	10,210
120	10,090	9,996	9,840	9,720	9,590	9,470	9,340	9,220	9,090	8,960
130	8,830	8,700	8,570	8,440	8,320	8,190	8,070	7,960	7,840	7,730
140	7,620	7,510	7,410	7,300	7,200	7,100	7,010	6,910	6,820	6,730
150	6,640	6,550	6,460	6,380	6,300	6,220	6,140	6,060	5,980	5,910
160	5,830	5,760	5,690	5,620	5,550	5,490	5,420	5,350	5,290	5,230
170	5,170	5,110	5,050	4,990	4,930	4,880	4,820	4,770	4,710	4,660
180	4,610	4,560	4,510	4,460	4,410	4,360	4,320	4,270	4,230	4,180
190	4,140	4,090	4,050	4,010	3,970	3,930	3,890	3,850	3,810	3,770
200	3,730									

Above L/r of 130, the higher-strength steels offer no advantage as to allowable compressive stress (f_a). Above this point, use Table I-3 for the more economical steel of 36,000-psi-yield strength.

Table I-3
36,000-psi-Yield steel

L/r Ratio		1	2	3	4	5	6	7	8	9
		21,560	21,520	21,480	21,440	21,390	21,350	21,300	21,250	21,210
10	21,160	21,100	21,050	21,000	20,950	20,890	20,830	20,780	20,720	20,660
20	20,600	20,540	20,480	20,410	20,350	20,280	20,220	20,150	20,080	20,010
30	19,940	19,870	19,800	19,730	19,650	19,580	19,500	19,420	19,350	19,270
40	19,190	19,110	19,030	18,950	18,860	18,780	18,700	18,610	18,530	18,440
50	18,350	18,260	18,170	18,080	17,990	17,900	17,810	17,710	17,620	17,530
60	17,430	17,330	17,240	17,140	17,040	16,940	16,840	16,740	16,640	16,530
70	16,430	16,330	16,220	16,120	16,010	15,900	15,790	15,690	15,580	15,470
80	15,360	15,240	15,130	15,020	14,900	14,790	14,670	14,560	14,440	14,320
90	14,200	14,090	13,970	13,840	13,720	13,600	13,480	13,350	13,230	13,100
100	12,980	12,850	12,720	12,590	12,470	12,330	12,200	12,070	11,940	11,810
110	11,670	11,540	11,400	11,260	11,130	10,990	10,850	10,710	10,570	10,430
120	10,280	10,140	9,990	9,850	9,700	9,550	9,410	9,260	9,110	8,970
130	8,840	8,700	8,570	8,440	8,320	8,190	8,070	7,960	7,840	7,730
140	7,620	7,510	7,410	7,300	7,200	7,100	7,010	6,910	6,820	6,730
150	5,640	6,550	6,460	6,380	6,300	6,220	6,140	6,060	5,980	5,910
160	5,830	5,760	5,690	5,620	5,550	5,490	5,420	5,350	5,290	5,230
170	5,170	5,110	5,050	4,990	4,930	4,880	4,820	4,770	4,710	4,660
180	4,610	4,560	4,510	4,460	4,410	4,360	4,320	4,270	4,230	4,180
190	4,140	4,090	4,050	4,010	3,970	3,930	3,890	3,850	3,810	3,770
200	3,730									

Above L/r of 130, the higher-strength steels offer no advantage as to allowable compressive stress (f_a). Above this point, use this table.

Table I-4
42,000-psi-Yield steel

L/r Ratio		1	2	3	4	5	6	7	8	9
		25,150	25,100	25,050	24,990	24,940	24,880	24,820	24,760	24,700
10	24,630	24,570	24,500	24,430	24,360	24,290	24,220	24,150	24,070	24,000
20	23,920	23,840	23,760	23,680	23,590	23,510	23,420	23,330	23,240	23,150
30	23,060	22,970	22,880	22,780	22,690	22,590	22,490	22,390	22,290	22,190
40	22,080	21,980	21,870	21,770	21,660	21,550	21,440	21,330	21,220	21,100
50	20,990	20,870	20,760	20,640	20,520	20,400	20,280	20,160	20,030	19,910
60	19,790	19,660	19,530	19,400	19,270	19,140	19,010	18,880	18,750	18,610
70	18,480	18,340	18,200	18,060	17,920	17,780	17,640	17,500	17,350	17,210
80	17,060	16,920	16,770	16,620	16,470	16,320	16,170	16,010	15,860	15,710
90	15,550	15,390	15,230	15,070	14,910	14,750	14,590	14,430	14,260	14,090
100	13,930	13,760	13,590	13,420	13,250	13,080	12,900	12,730	12,550	12,370
110	12,190	12,010	11,830	11,650	11,470	11,280	11,100	10,910	10,720	10,550
120	10,370	10,200	10,030	9.870	9,710	9,560	9,410	9,260	9,110	8,970

Above L/r of 130, the higher-strength steels offer no advantage as to allowable compressive stress (f_a). Above this point, use Table I-3 for the more economical steel of 36,000-psi-yield strength.

Table I-5
46,000-psi-Yield steel

L/r Ratio		1	2	3	4	5	6	7	8	9
		27,540	27,480	27,420	27,360	27,300	27,230	27,160	27,090	27,020
10	26,950	26,870	26,790	26,720	26,630	26,550	26,470	26,380	26,290	26,210
20	26,110	26,020	25,930	25,830	25,730	25,640	25,540	25,430	25,330	25,230
30	25,120	25,010	24,900	24,790	24,680	24,560	24,450	24,330	24,210	24,100
40	23,970	23,850	23,730	23,600	23,480	23,350	23,220	23,090	22,960	22,830
50	22,690	22,560	22,420	22,280	22,140	22,000	21,860	21,720	21,570	21,430
60	21,280	21,130	20,980	20,830	20,680	20,530	20,370	20,220	20,060	19,900
70	19,740	19,580	19,420	19,260	19,100	18,930	18,760	18,600	18,430	18,260
80	18,080	17,910	17,740	17,560	17,390	17,210	17,030	16,850	16,670	16,480
90	16,300	16,120	15,930	15,740	15,550	15,360	15,170	14,970	14,780	14,580
100	14,390	14,190	13,990	13,790	13,580	13,380	13,170	12,960	12,750	12,540
110	12,330	12,120	11,900	11,690	11,490	11,290	11,100	10,910	10,720	10,550
120	10,370	10,200	10,030	9,870	9,710	9,560	9,410	9,260	9,110	8,970

Above L/r of 130, the higher-strength steels offer no advantage as to allowable compressive stress (f_a). Above this point, use Table I-3 for the more economical steel of 36,000-psi-yield strength.

Table I-6
50,000-psi-Yield steel

L/r Ratio		1	2	3	4	5	6	7	8	9
		29,940	29,870	29,800	29,730	29,660	29,580	29,500	29,420	29,340
10	29,260	29,170	29,080	28,990	28,900	28,800	28,710	28,610	28,510	28,400
20	28,300	28,190	28,080	27,970	27,860	27,750	27,630	27,520	27,400	27,280
30	27,150	27,030	26,900	26,770	26,640	26,510	26,380	26,250	26,110	25,970
40	25,830	25,690	25,550	25,400	25,260	25,110	24,960	24,810	24,660	24,510
50	24,350	24,190	24,040	23,880	23,720	23,550	23,390	23,220	23,060	22,890
60	22,720	22,550	22,370	22,200	22,020	21,850	21,670	21,490	21,310	21,120
70	20,940	20,750	20,560	20,380	20,190	19,990	19,800	19,610	19,416	19,210
80	19,010	18,810	18,610	18,410	18,200	17,990	17,790	17,580	17,370	17,150
90	16,940	16,720	16,500	16,290	16,060	15,840	15,520	15,390	15,170	14,940
100	14,710	14,470	14,240	14,000	13,770	13,530	13,290	13,040	12,800	12,570
110	12,340	12,120	11,900	11,690	11,490	11,290	11,100	10,910	10,720	10,550
120	10,370	10,200	10,030	9,870	9,710	9,560	9,410	9,260	9,110	8,970

Above L/r of 130, the higher-strength steels offer no advantage as to allowable compressive stress (f_a). Above this point, use Table I-3 for the more economical steel of 36,000-psi-yield strength.
Reprinted by permission of the James F. Lincoln Arc Welding Foundation.

Appendix J: Design of Flat Plates

Table J-1
Flat plate formulas

Shape	Loading	Edge Fixation	Maximum Fiber Stress, f, psi	Center Deflection, Δ, in.	Remarks
Circle radius R.	Uniform p	Fixed	$0.75p\frac{R^2}{t^2}$	$0.17\left(\frac{p}{E}\right)\frac{R^4}{t^3}$	f max. at edge
		Simply Supported	$1.24p\frac{R^2}{t^2}$	$0.695\left(\frac{p}{E}\right)\frac{R^4}{t^3}$	f max. at center
	Central concentrated P on r	Fixed	$1.43\left[\log_{10}\left(\frac{R}{t}\right)+0.11\left(\frac{r}{R}\right)^2\right]\frac{P}{t^2}$	$0.22\left(\frac{P}{E}\right)\frac{R^2}{t^3}$	P uniform over circle, radius r Center stress
		Simply Supported	$1.43\left[\log_{10}\left(\frac{R}{t}\right)+0.334+0.06\left(\frac{r}{R}\right)^2\right]\frac{P}{t^2}$	$0.55\left(\frac{P}{E}\right)\frac{R^2}{t^3}$	As above center stress
Ellipse 2A × 2a a < A	Uniform p	Fixed	$\frac{6}{3n^4+2n^2+3}p\frac{a^2}{t^2}$	$\frac{1.365}{3n^4+2n^2+3}\left(\frac{p}{E}\right)\frac{a^4}{t^3}$	$n=\frac{a}{A}$, exact solution
		Simply Supported	$\frac{3}{0.42n^4+n^2+1}p\frac{a^2}{t^2}$ (1)		$n=\frac{a}{A}$, approximate fits n = 0 and n = 1
	Central concentrated p	Fixed	$\frac{50}{3n^4+2n^2+12.5}\frac{P}{t^2}$ (2)		$n=\frac{a}{A}$, approximate Fits n = 0 and n = 1 Load over 0.01% of area
		Simply Supported	$\frac{13.1}{0.42n^4+n^2+2.5}\frac{P}{t^2}$ (2)		$n=\frac{a}{A}$, approximate Fits n = 1 Load over 0.01% of area
Rectangle B × b b < B	Uniform p	Fixed	$B_1p\frac{b^2}{t^2}$	$\phi_1\left(\frac{p}{E}\right)\frac{b^4}{t^3}$	ϕ_1 and B_1 depend on B/b, See Table J-2.
		Simply Supported	$B_2p\frac{b^2}{t^2}$	$\phi_2\left(\frac{p}{E}\right)\frac{b^4}{t^3}$	ϕ_2 and B_2 depend on B/b. See Table J-2.
	Central concentrated p	Fixed	$\frac{4.00}{1+2n^2}\frac{P}{t^2}$ (3)	$\phi_3\left(\frac{p}{E}\right)\frac{b^2}{t^3}$	$\frac{b}{B}=n$, approximate Fits n=1 and n=0
		Simply Supported	$\frac{5.3}{1+2.4n^2}\frac{P}{t^2}$ (3)		$\frac{b}{B}=n$, approximate Fits n=1 and n=0
Square B × B	Uniform p	Fixed	$0.308p\frac{B^2}{t^2}$	$0.0138\left(\frac{p}{E}\right)\frac{B^4}{t^3}$	f max. at center of side
		Simply Supported	$0.287p\frac{B^2}{t^2}$	$0.0443\left(\frac{p}{E}\right)\frac{B^4}{t^3}$	f max. at center
	Central concentrated p	Fixed	$1.32\frac{P}{t^2}$		As above deflection nearly exact
		Simply Supported	$1.58\frac{P}{t^2}$	$0.125\left(\frac{P}{E}\right)\frac{B^2}{t^3}$	Approximate for f area of contact not too small
Flat Stayed Plate	Uniform p	Staybolts spaced at corners of square of side S	$0.228p\frac{S^2}{t^2}$	$0.0284\left(\frac{p}{E}\right)\frac{S^4}{t^3}$	If plate as a whole deforms, superimpose the stresses and deflections on those for flat plate when loaded
Circular Flanged	Uniform p	Fastened to shell	$p\left[\frac{r}{2t}+\phi\left(\frac{R-\frac{r}{2}1+\frac{r}{R}}{t}\right)^2\right]$		ϕ varies with shell and joint stiffness from 0.33 to 0.38 knuckle radius, r’

1. Formula of proper form to fit circle and infinite rectangle as n varies from 1 to 10.
2. Formulas for load distributed over 0.0001 plate area to match circle when n = 1. They give reasonable values for stress when n = 0. Stress is lower for larger area subject to load.
3. Formulas of empirical form to fit Hutte values for square when n = 1. They give reasonable values when n = 0. Assume load on 0.01 of area.
4. Only apparent stresses considered.
5. These formulas are not to be used in determining failure.

Table J-2
Flat plate coefficients

Stress Coefficients—Circle with Concentrated Center Load											
r/R	1.0	0.10	0.09	0.08	0.07	0.06	0.05	0.04	0.03	0.02	0.01
Fixed[1]	0.157	1.43	1.50	1.57	1.65	1.75	1.86	2.00	2.18	2.43	2.86
Supported[2]	0.563	1.91	1.97	2.05	2.13	2.23	2.34	2.48	2.66	2.91	3.34
Stress and Deflection Coefficients—Ellipse											
A/a	1.0	1.2	1.4	1.6	1.8	2.0	2.5	3.0	4.0	5.0	∞
Uniform Load											
Fixed											
Stress[3]	0.75	1.03	1.25	1.42	1.54	1.63	1.77	1.84	1.91	1.95	2.00
Deflection[4]	0.171	0.234	1.284	0.322	0.350	0.370	0.402	0.419	0.435	0.442	0.455
Uniform Load											
Supported[5]	1.24	1.58	1.85	2.06	2.22	2.35	2.56	2.69	2.82	2.88	3.00
Central Load											
Fixed[6]	2.86	3.26	3.50	3.64	3.73	3.79	3.88	3.92	3.96	3.97	4.00
Supported[7]	3.34	3.86	4.20	4.43	4.60	4.72	4.90	5.01	5.11	5.16	5.24
Stress and Deflection Coefficients—Rectangle											
B/b	1.0	1.25	1.5	1.6	1.75	2.0	2.5	3.0	4.0	5.0	∞
Stress B_1	0.308	0.399	0.454		0.490	0.497					0.500
Stress B_2	0.287	0.376	0.452	0.517	0.569	0.610	0.650	0.713	0.741	0.748	0.750
$\frac{4}{1+2n^2}$	1.33	1.75	2.12	2.25	2.42	2.67	3.03	3.27	3.56	3.70	4.00
$\frac{5.3}{1+2.4n^2}$	1.56	2.09	2.56	2.74	2.97	3.31	3.83	4.18	4.61	4.84	5.30
Deflection ϕ_1	0.0138	0.0199	0.0240		0.0264	0.0277					0.0284
Deflection ϕ_2	0.0443	0.0616	0.0770	0.0906	0.1017	0.1106	0.125	0.1336	0.1400	0.1416	0.1422
Deflection ϕ_3	0.1261		0.1671			0.1802		0.1843	0.1848		0.1849

[1]Values of 1.43 [$\log_{10}R/r + 0.11\,(r/R)^2$]
[2]Values of 1.43 [$\log_{10}R/r + 0.334 + 0.06(r/R)^2$]
[3]Values of $6/(3n^4 + 2n^2 + 3)$
[4]Values of $1{,}365/(3n^4 + 2n^2 + 3)$
[5]Values of $3/(0.42n^4 + n^2 + 1)$
[6]Values of $50/(3n^4 + 2n^2 + 12.5)$
[7]Values of $13.1/(0.42n^4 + n^2 + 2.5)$

Reprinted by permission of AISI, Committee of Steel Plate Producers and Steel Plate Fabricators Association, Inc. from Steel Plate Engineering Data, Volume 2

Appendix K: Time Required to Drain Vessels

Notation

q = discharge rate, cu ft/sec
g = acceleration due to gravity, ft/sec
D = diameter of vessel, ft
R = radius of sphere, ft
L = length of horizontal vessel, ft
H = height of liquid in vessel, ft
d = diameter of drain, in.
c = coefficient of discharge
T = time to drain, min

Notes

1. It is assumed that the flow has a Reynolds number greater than 1000.

   General Equation.

   $$q = dc\sqrt{2gH}$$

- *For sphere.*

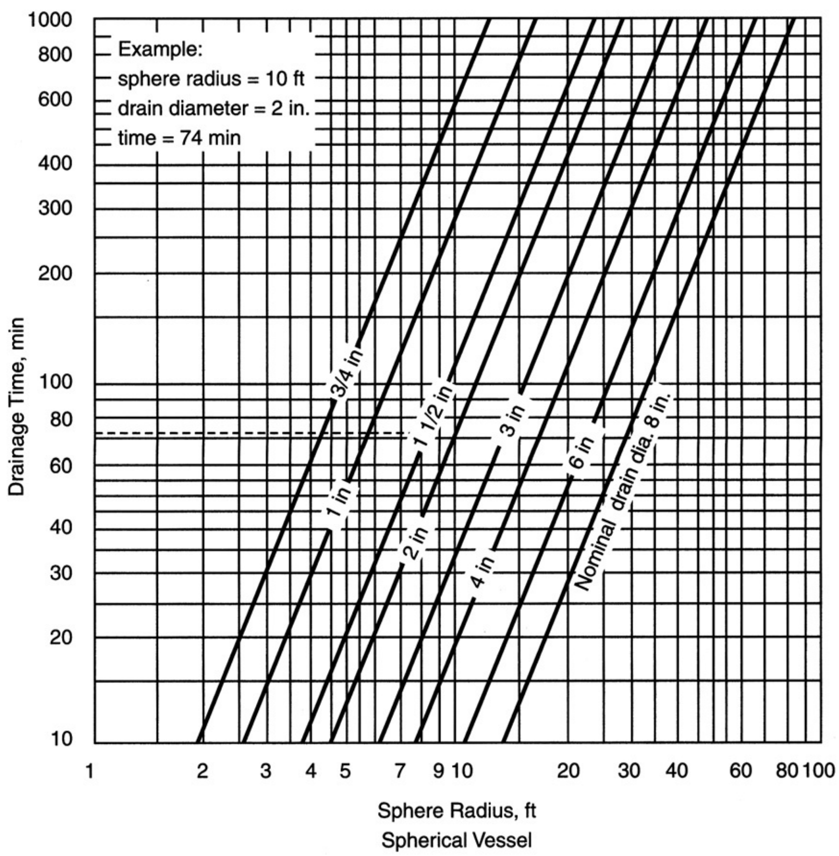
$$T = \frac{R^{2.5}}{d^2}$$

- *For horizontal vessel*

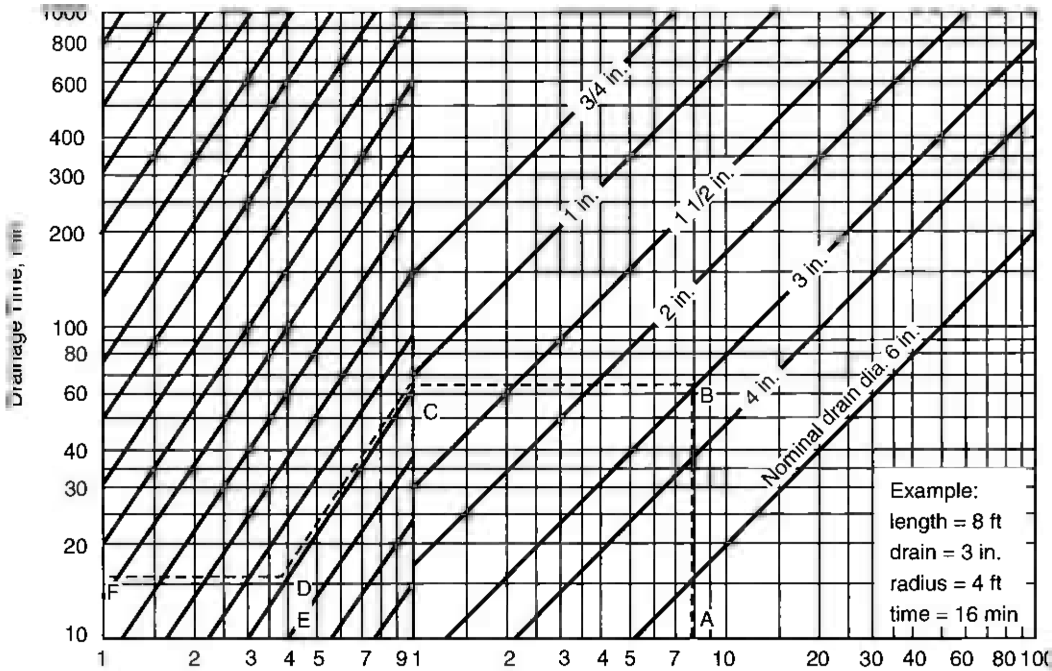
$$T = 2.4\left(\frac{L \cdot D^{1.5}}{d^2}\right)$$

- *For vertical vessels.*

$$T = D^2\sqrt{\frac{H}{D^2}}$$

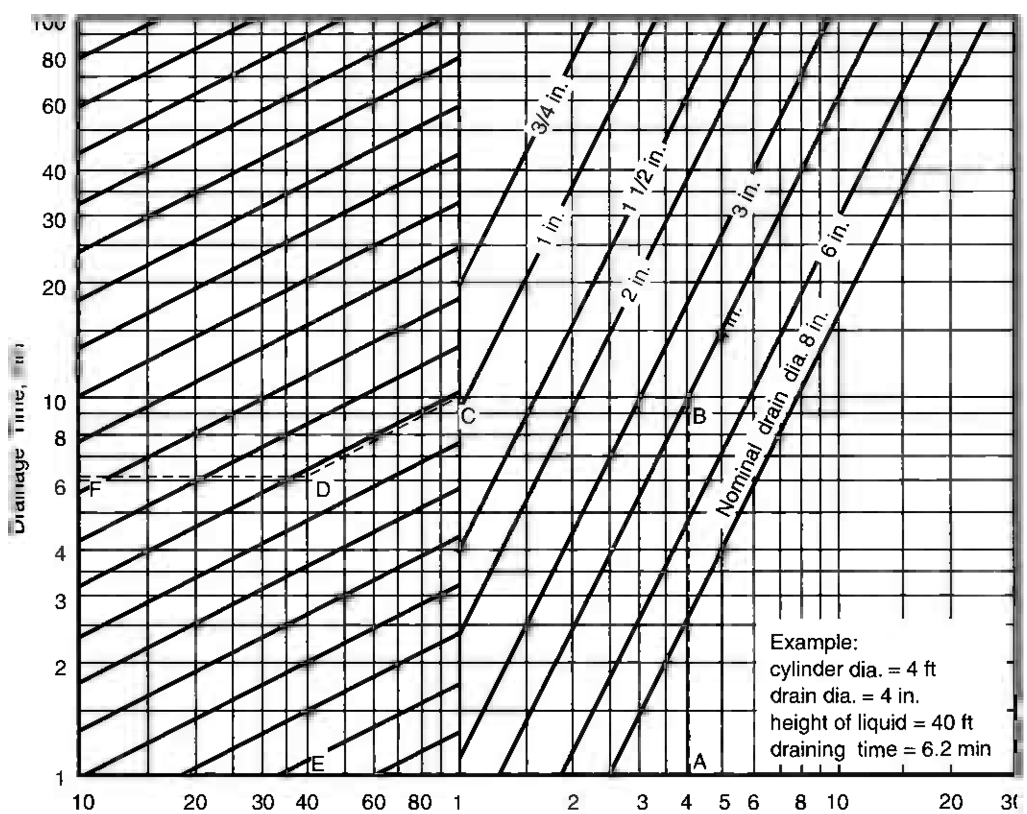


Source: Ray Elshout,Union Oil Co, Brea,CA. *Repriented by Permission of Gulf Publishing Co.*



Horizontal Cylinder

Reprinted by permission of Gulf Publishing Co.



Vertical Cylinder

Reprinted by permission of Gulf Publishing Co.

Appendix L: Vessel Surge Capacities and Hold-Up Times

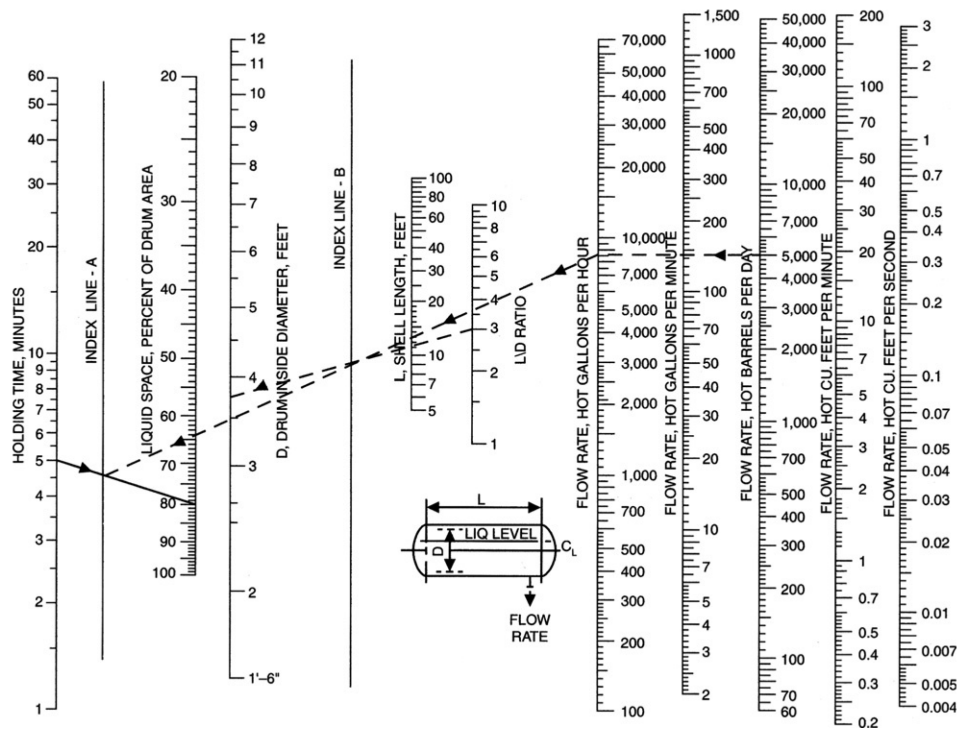


Figure L-1. Nomograph to find drum size for holding time. *Reprinted by permission of Gulf Publishing Co.*

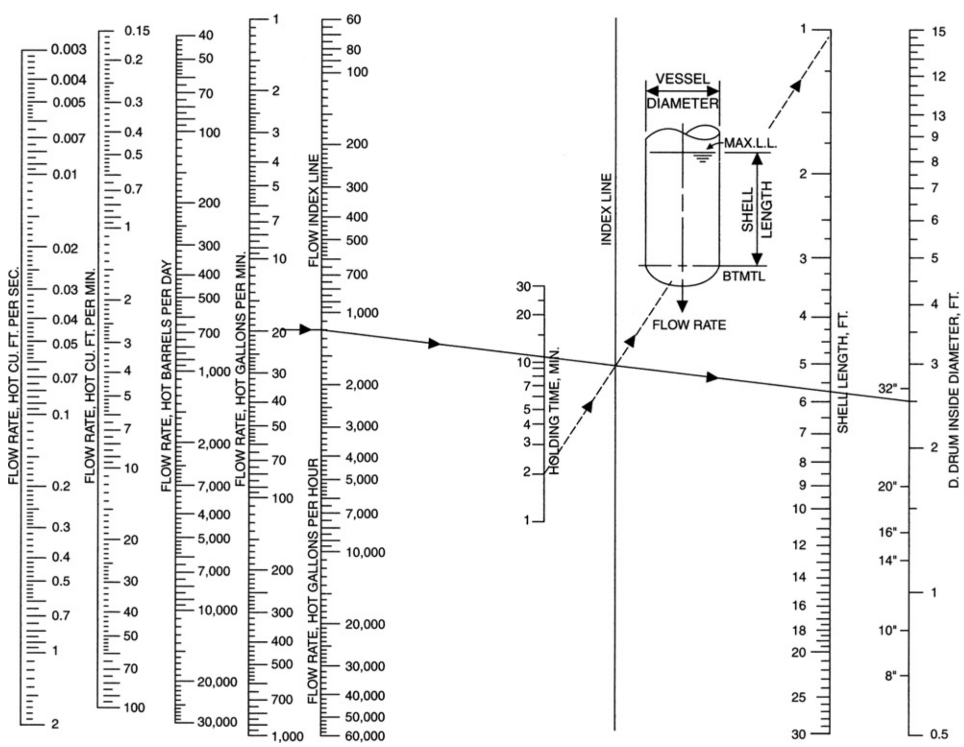
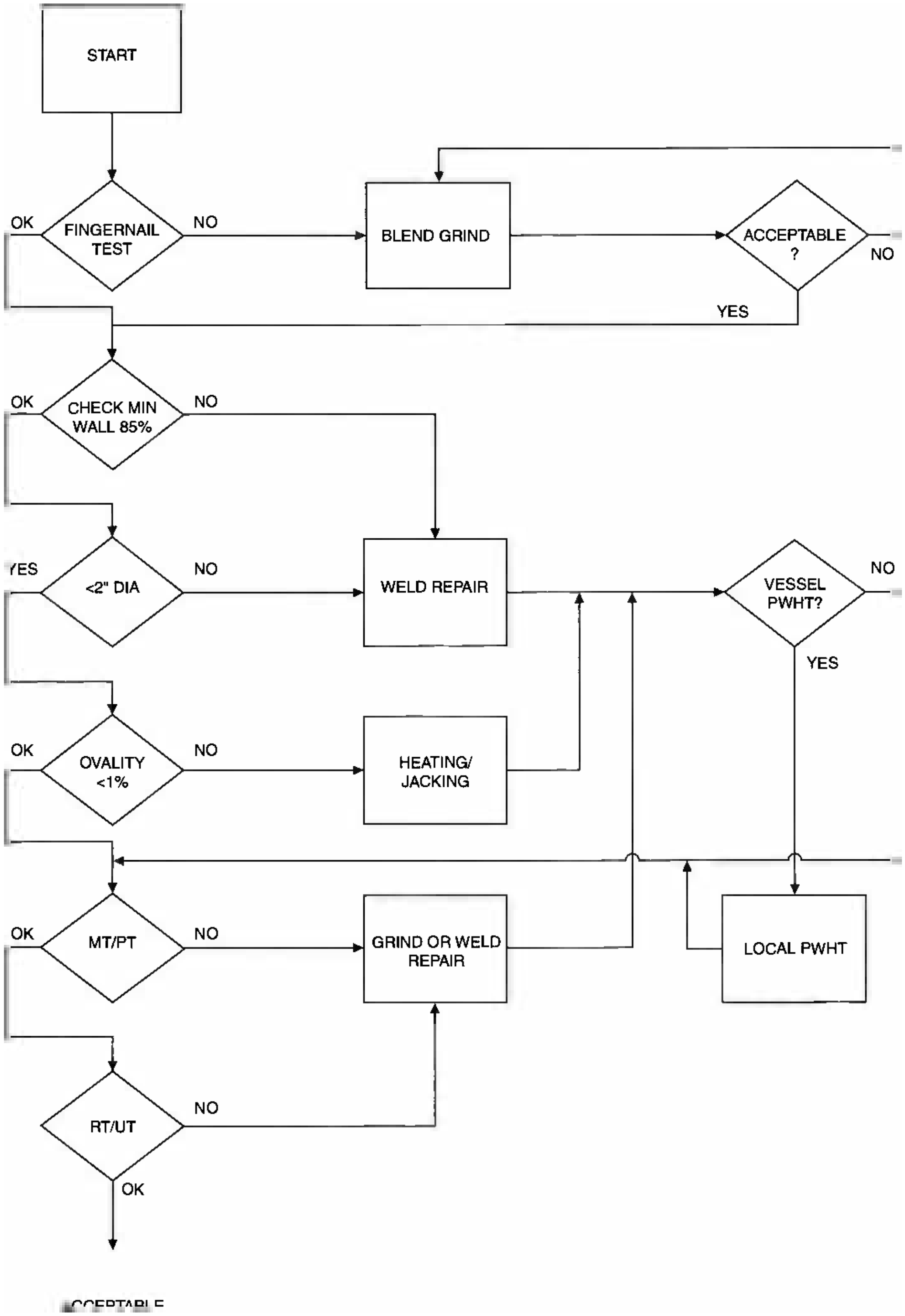
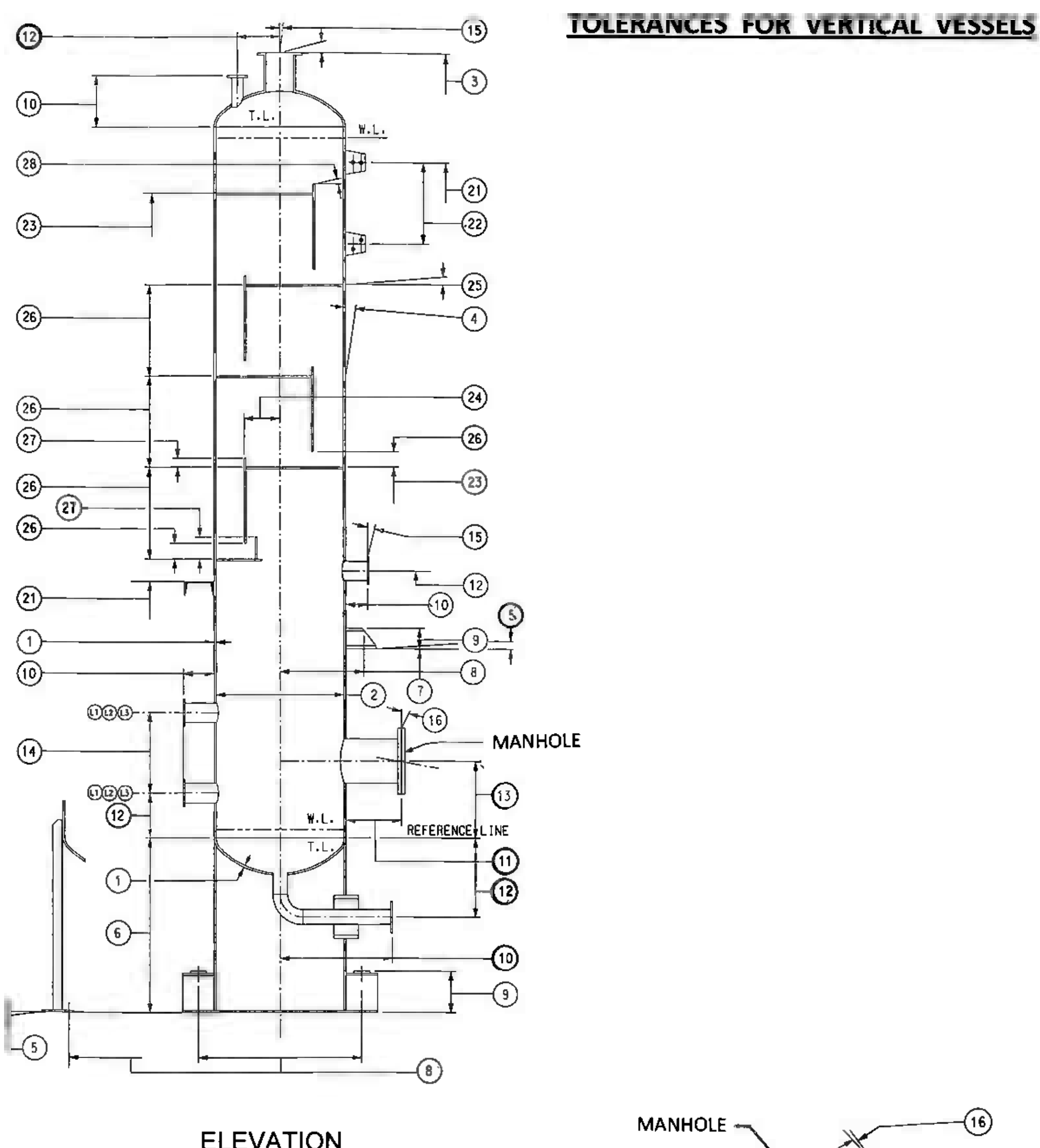


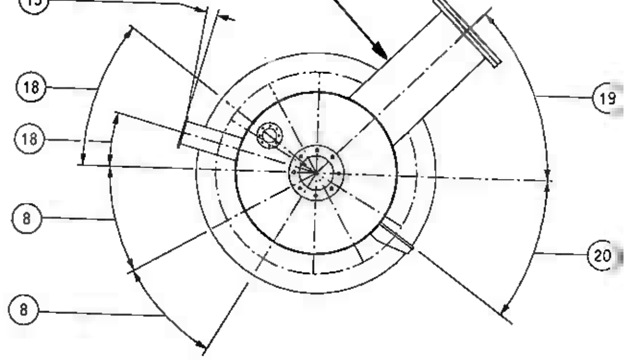
Figure L-2. Nomograph to find shell length for desired holding time. *Reprinted by permission of Gulf Publishing Co.*

Appendix M: Minor Defect Evaluation Procedure



Appendix N: Tolerances





Tolerance for Vertical Vessels

Notes:

1. Minimum thickness as specified.
2. Out of roundness is defined by ASME VIII-1, Para. UG 80 and is +/- 1% of the diameter. The following tolerances are recommended to provide stricter control of diametral difference;

SHELL DIA	<48" (1200)	48" - 84" (1200 - 2100)	84"- 180" (2100- 4800)	>180" (4800)
TOLERANCE	± .125" (3mm)	± .188" (4.5mm)	± .25" (6mm)	± .313(7.5mm)

3. Distance from reference tangent line and top flange Face: 0.015 in / Ft (1mm / 1m) < 0.5 in (12mm)
4. Shell & skirt tolerance: Max slope from straight line is 0.125 in / 10 feet (3mm / 3050mm). The total maximum deviation allowed is as follows; (also see table for maximum permissible bow)

Tan-Tan Length	Total Max Deviation
<50' (15240 mm)	.5" (13 mm)
50' - 100' (15240 mm - 30480 mm)	.75" (19 mm)
>100' (30480 mm)	1" (25 mm)

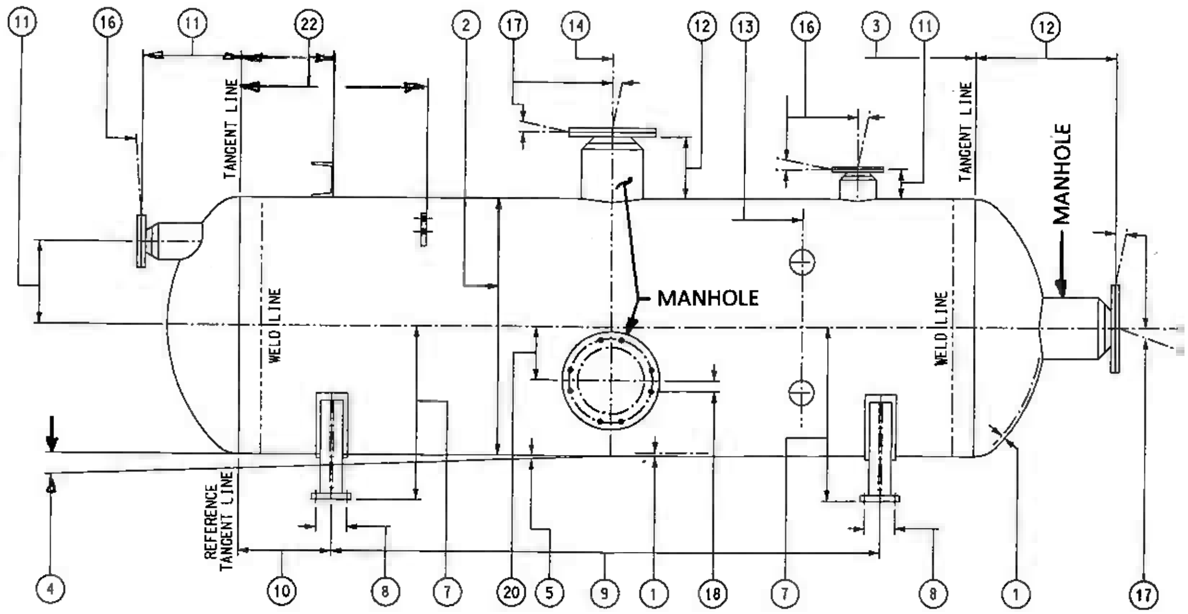
5. Out of levelness over a horizontal plane;

VESSEL DIAMETER	<48" (1200 mm)	48" -84" (1200-2100)	84" - 120" (2100-3000)	>120" (3000 mm)
TOLERANCE	± .06" (1.5 mm)	± .125" (3 mm)	± .188" (5 mm)	± .25" (6 mm)

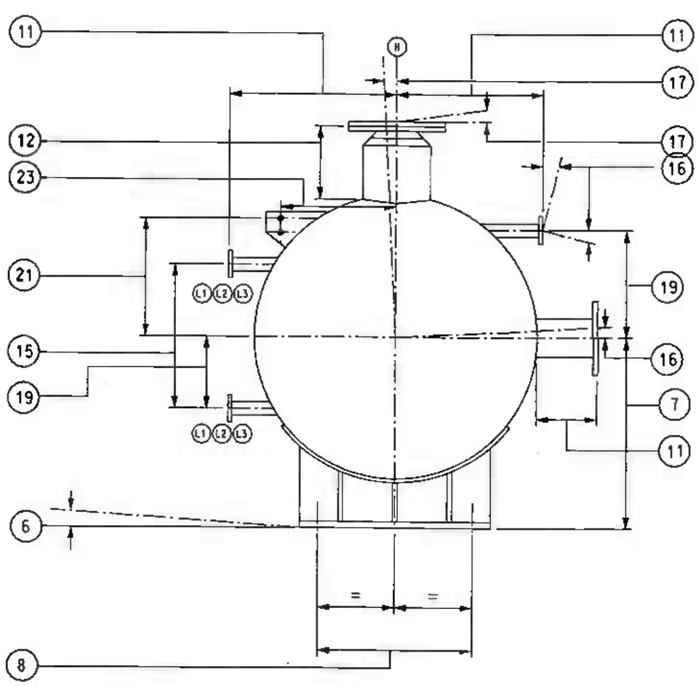
		± inches	± mm
6.	Height of Skirt or Support Legs	+ 0 (-) .125	+ 0 (-) 3
7.	Levelness of Support Lugs	0.125	3
8.	Bolt Circle Diameter for Anchor Bolts	0.125	3
9.	Height of Anchor Chairs	0.125	3

10.	Nozzle projection	0.125	3
11.	Projection of Manhole	0.25	6
12.	Location of Nozzle	0.125	3
13.	Location of Manhole	0.25	6
14.	Distance beween matched instrument connections	0.04	1
15.	Tilt of Nozzle	+/- .5 Degree	
16.	Tilt of Manhole	+/- 1 Degree	
17.	Permissible deviation of bolt holes from C.L.	0.06	1.5
18.	Orientation of Nozzle	0.125	3
19.	Orientation of Manhole	0.25	6
20.	Orientation of Clips & Gussets	0.125	3
21.	Location of Stiffening rings	0.25	6
22.	C-C location of bolt holes in clips	0.125	3
23.	Elevation of Tray Supports from Ref Tan Line	0.25	6
24.	Location of Downcomer Support bars from C.L.	+ .375 (-) .125	+ 9 (-) 3
25.	Out of level of Tray Support Rings	See Tolerances for Trays	
26.	Distance beween trays or DC Clearance	0.125	3
27.	Weir height	0.06	1.5
28.	Levelness of Downcomer Supports	0.125	3

TOLERANCES FOR HORIZONTAL VESSELS



ELEVATION



END VIEW

Tolerance for Horizontal Vessels

Notes:

1. Minimum thickness as specified.
2. Out of roundness is defined by ASME VIII-1, Para. UG 80 and is +/- 1% of the diameter. The following tolerances are recommended to provide stricter control of diametral difference;

Shell dia	<48" (1200)	48" - 84" (1200 - 2100)	>84" (2100)
Tolerance	± .125" (3)	± .188" (4.5)	± .25" (6)

3. Distance between Reference Line and far end of shell.
4. Deviation from horizontal plane.

Shell Length	< 50' (15000)	>50' (15000)
Tolerance	.015"/Ft (1 mm/M)	.01"/Ft (.8 mm / M)

5. Maximum Permissible Bow: <.15% of length <1.5" (40 mm) (See also Table for Maximum Bow)
6. Out of levelness of supports;

Vessel Diameter	< 48" (1200 mm)	48" -84" (1200-2100)	84" - 120" (2100-3000)	>120" (3000 mm)
Tolerance	± .06" (1.5 mm)	± .125" (3 mm)	± .188" (5 mm)	± .25" (6 mm)

		+/- inches	+/- mm
7.	Height of supports	0.13	3
8.	Distance between hole centers for Anchor Bolts	0.06	1.5
9.	Distance between supports	0.13	3
10.	Distance from C.L. Support to Reference Tangent Line	0.13	3
11.	Nozzle projection	0.13	3
12.	Projection of Manhole	0.25	6
13.	Location of Nozzle	0.13	3
14.	Location of Manhole	0.25	6
15.	Distance beween matched instrument connections	0.04	1
16.	Tilt of Nozzle	+/- .5 Degree	
17.	Tilt of Manhole	+/- 1 Degree	

(*Continued*)

18.	Permissible deviation of bolt holes from C.L.	0.04	1
19.	Location of Nozzle from C.L.	0.13	3
20.	Location of Manhole from C.L.	0.25	6
21.	Location of bolt holes of clip from C.L.	0.13	3
22.	Location of clips and stiffening rings from Reference Tangent Line	0.25	6
23.	Projection of bolt holes in clips from C.L.	0.13	3

Tolerances for trayed columns fabricated in multiple sub-assemblies

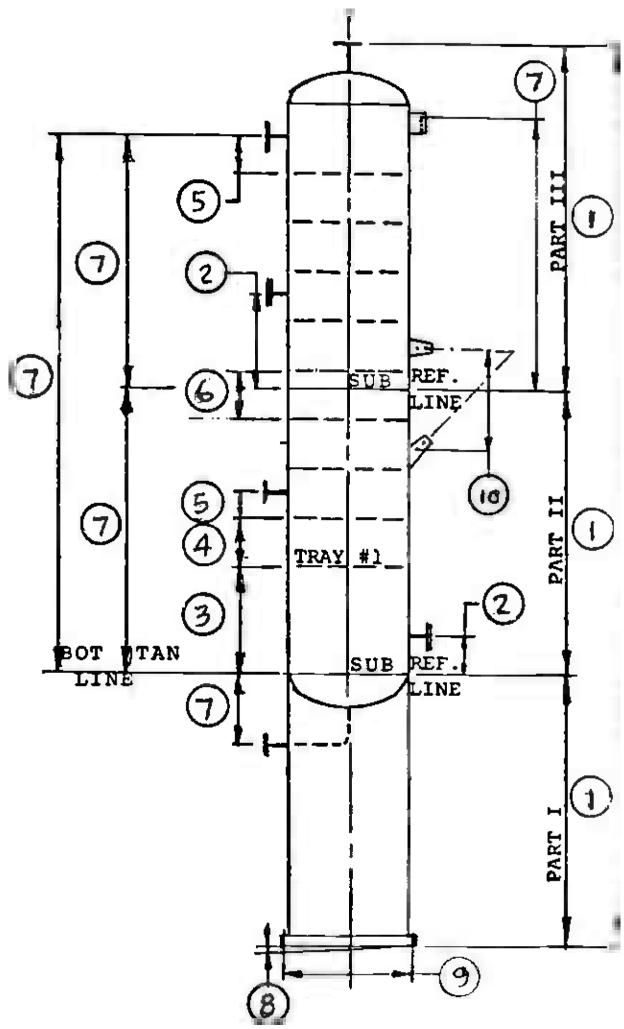
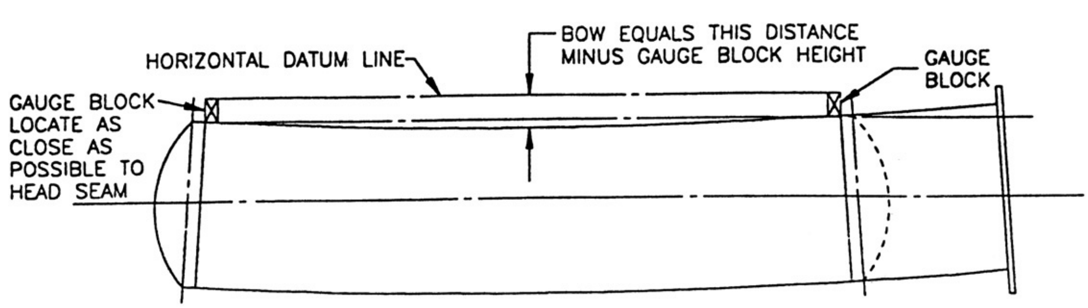


Table N-1
Tolerances for trayed columns fabricated in multiple sub-assemblies

Note	Item	Description/Notes		Tolerance	
				INCHES	mm
1	Sub-Assy	Length			
		Ft	mm		
		<5	<1525	+/- .125	+/- 6
		5-10	1525-3045	+/- .25	+/- 6
		10-15	3045-4575	+/- .375	+/- 6
		>15	>4575	+/- .5	+/- 6
2	Manways	From Reference Tangent Line		+/-0.375	+/- 10
3	First Tray	Elevation		+/- .25	+/- 6
4	Tray Spacing	Non-Cumulative		+/- .125	+/- 3
5	Nozzle	Above Tray		+/- .125	+/- 3
6	Tray Spacing	At Girth Seam Sub-Assembly		+/- .375	+/- 10
7	Nozzle Clips & Elevations	LENGTH			
		Ft	mm		
		<5	<1525	+/- .125	+/- 3
		5-10	1525-3045	+/- .25	+/- 6
		10-15	3045-4575	+/- .375	+/- 10
		15-20	4575-6100	+/- .5	+/- 13
		20-40	6100-12200	+/- .75	+/- 19
		40-80	12200-24400	+/- 1	+/- 25
		80-160	24400-48800	+/- 1.5	+/- 38
		>160	>48800	+/- .75%	+/- .75%
8	Out of level Slope	For Skirt Diameter >20' (6 M)		1/500	1/500
9	Anchor Bolt Circle	.5" (12 mm) Max Total		+/- .25	+/- 6
10	Bolt Hole Location-Clips	Non-Cumulative		+/- .25	+/- 6

Maximum Permissible Bow



Note: For permissible bow see Tables below. Measurements shall be taken at approximately 10' (3000 mm) apart, lengthwise, and 90° apart around the circumference.

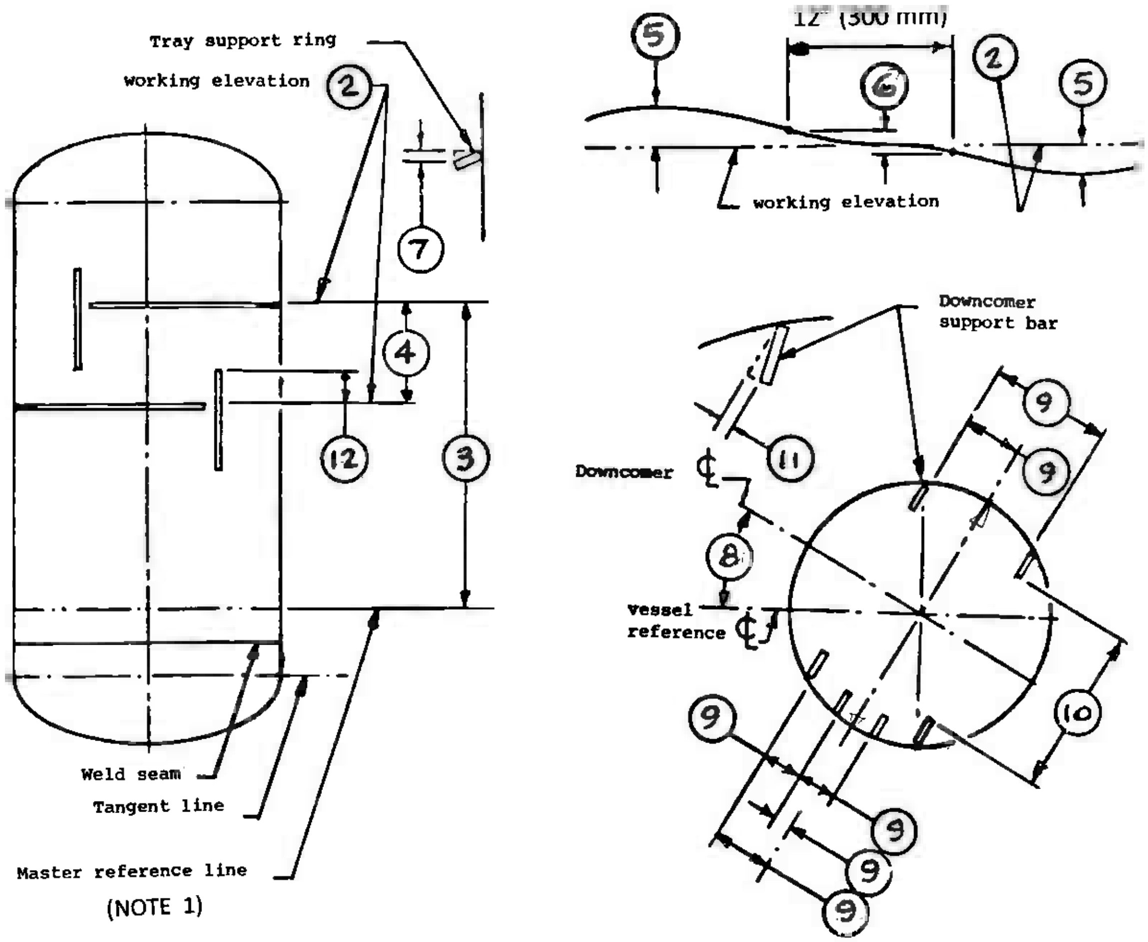
Table N-2
US units

Vessel Length Tan-Tan (Ft)	Vessel Diameter & Maximum Permissible Bow			
	<48"	48" to 72"	72" to 96"	>96"
<10	0.12	0.09	0.06	0.06
10-20	0.25	0.18	0.15	0.1
20-30	0.35	0.25	0.23	0.15
30-40	0.5	0.34	0.3	0.2
40-50	0.6	0.4	0.35	0.25
50-60	0.7	0.5	0.4	0.3
60-70	0.85	0.6	0.45	0.35
70-80	1	0.67	0.53	0.4
80-90	1.1	0.8	0.6	0.45
90-100	1.2	0.9	0.67	0.5
>100	1.5	1	0.75	0.55

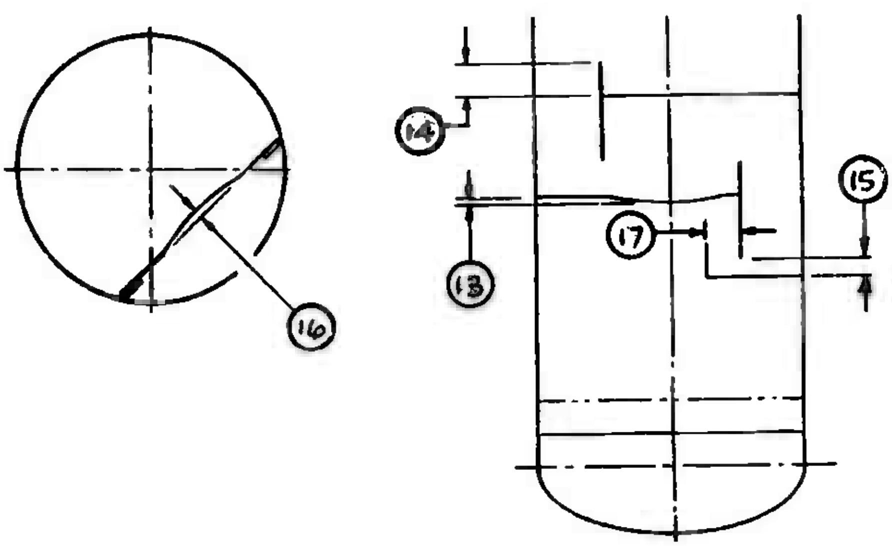
Table N-3
Metric units

Vessel Length Tan-Tan (mm)	Vessel Diameter & Maximum Permissible Bows			
	<1200	1200-1800	1800-2400	>2400
<3000	4	2.4	1.6	1.6
3000-6000	8	4.7	4	2.4
6000-9000	11	8	5.5	4
9000-12000	15	10	8	4.7
12000-15000	19	13	9.5	6
15000-18000	23	15	11	8
18000-21000	27	17	13.5	9
21000-24000	30	21	15	10
24000-27000	34	23	17	11
>27000	38	25	19	13

Tolerances for Trays, Tray Support Rings and Downcomers



Vessel Trays Supports
(Weld in Parts)



TRAY TOLERANCES

Tolerances for Trays, Tray Support Rings and Downcomers

1. The Master Reference Line (MRL) shall be established by the vessel manufacturer and clearly marked inside and outside of the vessel prior to attaching the bottom head. It shall be parallel to the root land of the bottom seam weld, and perpendicular to the longitudinal axis of the vessel.
2. The working elevation of a tray support ring shall be the elevation specified on the outline drawing. It shall be a level plane parallel to the MRL.
3. The distance from the MRL to the working elevation of any tray support ring shall be within +/- 1/4 in. (6mm) of the nominal distance.
4. The distance between the working elevations of any two adjacent tray support rings shall be within +/- 1/8 in. (3mm) of the nominal distance, and the accumulated tolerances between tray support rings shall not exceed the tolerances in Note 3.
5. The highest and lowest points of a tray support ring, measured adjacent to the vessel shell, shall not exceed the following deviations from the working elevations;

Vessel Dia	Tolerance
48” (1200 mm) and smaller	1/16” (1.5 mm)
48” to 84” (1200 – 2100 mm)	3/32” (2.5 mm)
Over 84”	1/8” (3 mm)

6. Tray support rings shall not have a waviness exceeding 1/16 in. (1.5mm) for any 1 foot (300mm) of circumferential length.
7. Tilt of a tray support ring over its width shall not exceed 1/16 in. (1.5mm)
8. Orientation of the downcomer centerline shall be within 1/8 in. (3mm) of its nominal distance from the Vessel Reference Centerline, VRC, for vessels up to 72 in. (1800mm) dia and within ¼” (6mm) for vessels over 72 in. (1800mm) dia.
9. Distance from downcomer support bar to VRC shall be within +/- 1/8 in. (3mm) of the nominal distance. On a multipass tray, the distance between downcomer support bars shall also be within +/- 1/8 in. (3mm) of the nominal distance.
10. Deviation from nominal distance between downcomer support bars shall be subject to vessel out-of-roundness tolerance. Facility for adjustment must be provided in the downcomer and weir plates.
11. Tilt of downcomer support bars over its width shall not exceed 1/16 in. (1.5mm)
12. Distance between top of downcomer (weir) support bar and top of tray support ring shall be within +/- 1/16 in. (1.5mm) of the nominal distance.
13. Tray decks shall have a flat surface within 1/16 in. (1.5mm) measured across a 1 ft (300mm) square surface.
14. The difference in height between the highest and lowest points of the weir, measured to a level plane, shall not exceed;

Vessel Dia	Tolerance
72” (1800 mm) and smaller	1/8” (3 mm)
72” to 108” (1800 – 2700 mm)	3/16” (5 mm)
Over 108”	1/4” (6 mm)

15. Clearance between bottom of downcomer and top of tray or seal pan below shall not deviate from the nominal clearance by more than;

Vessel Dia	Tolerance
72” (1800 mm) and smaller	+/- 1/8” (3 mm)
72” to 108” (1800 – 2700 mm)	+/- 3/16” (5 mm)
Over 108”	+/- 1/4” (6 mm)

16. The bowing of a downcomer in a horizontal plane shall not deviate from the straight by more than +/- 1/8 in. (3mm)
17. Downcomer horizontal clearances, measured from the bottom edge of downcomer to recessed seal pan or inlet weir, shall be within +/- 1/8 in. (3mm) of nominal.

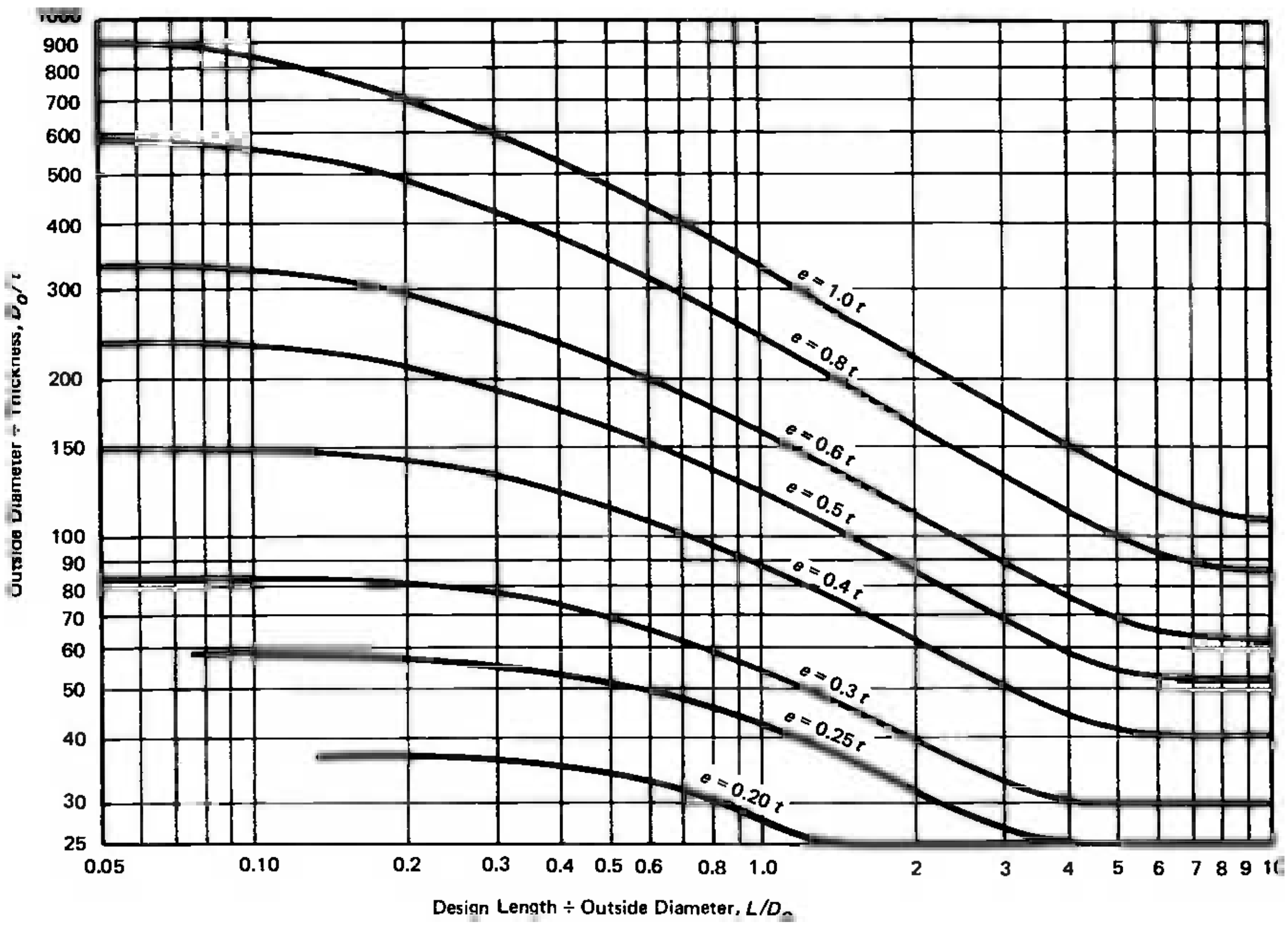


Figure N-1. Maximum permissible deviation from a circular form e for vessels under external pressure. *Source: Reprinted by permission of ASME.*

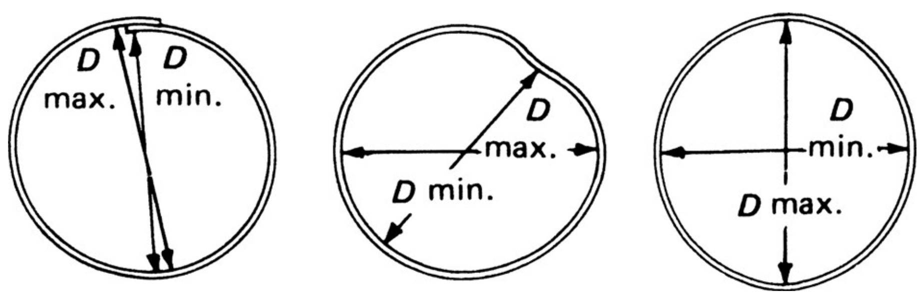
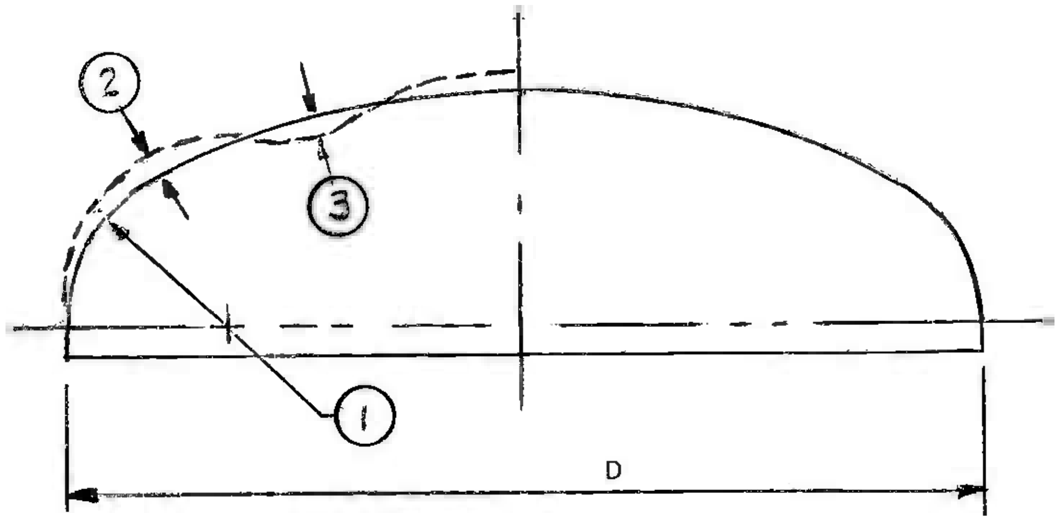


Figure N-2. Example of differences between maximum and minimum inside diameters in cylindrical, conical, and spherical shells. *Source: Reprinted by permission of ASME.*

Tolerances for Shape of Heads

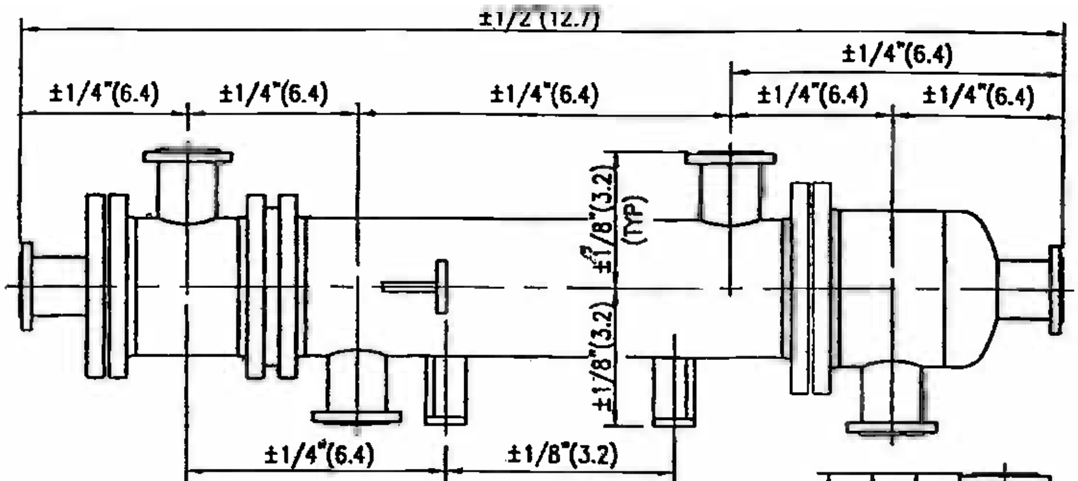


(1) Knuckle Radius: Not less than minimum specified

(2) Outside Surface: .0125 D

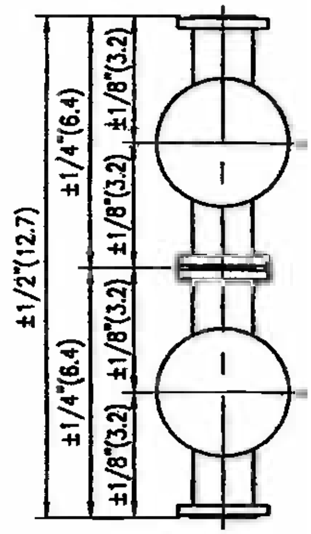
(3) Inside Surface: .00625 D

Tolerances for Heat Exchangers

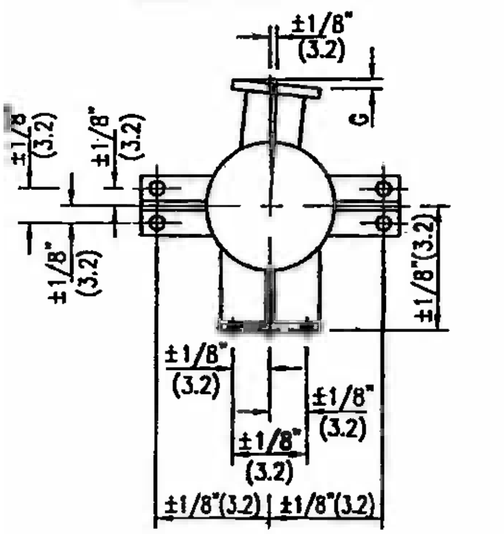


NOMINAL NOZZLE SIZE	G MAX
2"–4" INCLUSIVE	1/16"(1.6)
6"–12" INCLUSIVE	3/32"(2.4)
14"–36" INCLUSIVE	3/16"(4.8)
OVER 36"	1/4"(6.4)

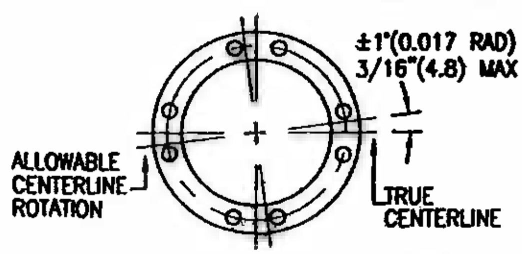
NOTE: This table applies to nozzles connecting to external piping only.



STACKED EXCHANGERS



CONNECTION NOZZLE ALIGNMENT AND SUPPORT TOLERANCES



ROTATIONAL TOLERANCE ON NOZZLE FACES AT BOLT CIRCLE

Reprinted by Permission of TEMA

Table N-4
Tolerances for flanges

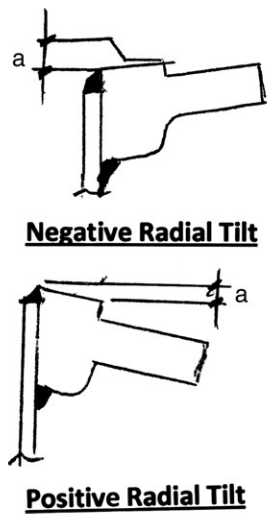
Threaded, Slip-on, Lap Joint and Blind			
Outside Diameter	OD 24" & Less	+/- .06	
	OD > 24"	+/- .125	
Inside Diameter	Threaded	To Standard Gage Limits	
	SLIP-ON & LAP JOINT	10" & Smaller, +.03, -0	
		12" & Larger, +.06, -0	
Outside Diameter of Hub	12" & Smaller	+.09, -.06	
	14" thru 42"	+/- .125	
Diameter of Contact Face	.0625 RF	+/- .03	
	.25" RF, T&G, M&F	+/- .015	
Diameter of Counterbore	10" & Smaller	+.03, -0	
	12" thru 42"	+.06, -0	
Drilling	Bolt Circle	.5" thru 24", +/- .06	26" thru 42", +/- .06
	Bolt Hole Spacing	+/- .03	+/- .03
	Eccentricity of Bolt Circle with Respect to Bore	.03 Max	
	Eccentricity of Facing with Respect to Bore		
Thickness	18" & Smaller	+.125, -0	
	20" thru 42"	+.188, -0	
Length Thru Hub	10" & Smaller	+/- .06	
	12" thru 42"	+/- .125	
Ring Joint Facing			
Depth	Dim L	+.015, -0	
Width	Dim D	+/-0.008	
Pitch Diameter	Dim P	+/-0.005	
Radius at Bottom	Dim r	Max	
Angle	23° Angle	+/- .5°	

Welding Neck			
Outside Diameter	OD 24" & Less	+/- .06	
	OD > 24"	+/- .125	
Inside Diameter	10" & Smaller	+/- .03	
	12" thru 18"	+/- .06	
	20" thru 42"	+.125, - .06	
Diameter of Contact Face	.0625 RF	+/- .03	
	.25" RF, T&G, M&F	+/- .015	
Diameter of Hub at Base	When "X" is 24" or Smaller	+/- .06	
	When "X" is over 24"	+/- .25	
Diameter of Hub at Point of Welding	5" & Smaller	+.09, -.03	
	6" & Larger	+.156, -.03	
Drilling	Bolt Circle	.5" thru 24", +/- .06	26" thru 42", +/- .06
	Bolt Hole Spacing	+/- .03	+/- .03
	Eccentricity of Bolt Circle with Respect to Bore	.03 Max	
	Eccentricity of Facing with Respect to Bore		
Thickness	18" & Smaller	+.125, -0	
	>18"	+.188, -0	
Length thru Hub	10" & Smaller	+/- .06	
	12" thru 42"	+/- .125	
Ring Joint Gasket			
Ring Width	Dim A	+/-0.008	
Ring Depth	Dim B&H	+/-0.015	
Width Octagonal Flat	Dim C	+/-0.008	
Pitch Diameter	Dim P	+/-0.007	
Radius	Dim r	+/-0.015	
Angle	23° Angle	+/-0.5°	

Notes:

1. All tolerances per Tube-Turn Catalog
2. See ASME B16.5 for flange dimensions and tolerances

Table N-5
Flange face tolerances



Maximum Tolerances			
Flange Rating	**Nominal Size**	**Waviness T.I.R. (In)**	**Positive Radial Tilt (a)**
150	<24"	0.016	0.009
	>24"	0.012	0.012
300-600	All Sizes	0.006	0.012
900-2500	All Sizes	0.005	0.003
75, 125, 175, 250, 350	>26"	0.006	0.018

Notes:

1. Negative radial tilt is not allowed

Table N-6
Maximum permissible offset in butt welding

Thickness		**Joint category, UW-33**			
		A		**B, C, D**	
Inches	mm	Inches	mm	Inches	mm
<.5	<12.5	.25 T	.25 T	.25 T	.25 T
.5 to .75	12.5 to 19	0.125	3.2	.25 T	.25 T
.75 to 1.5	19 to 38	0.125	3.2	0.188	4.8
1.5 to 2	38 to 50	0.125	3.2	0.125 T	.125 T
>2	>50	.06 T < b < .37	.06 T < b < .75	.125 T < b < .75	.125 T < b < 19

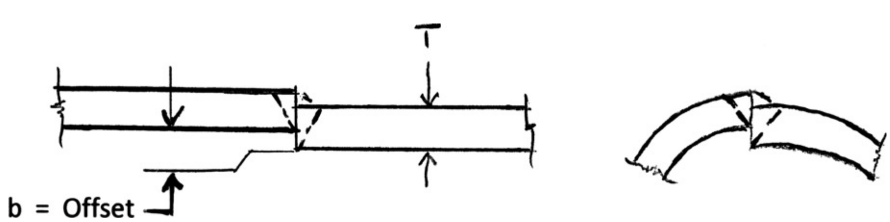
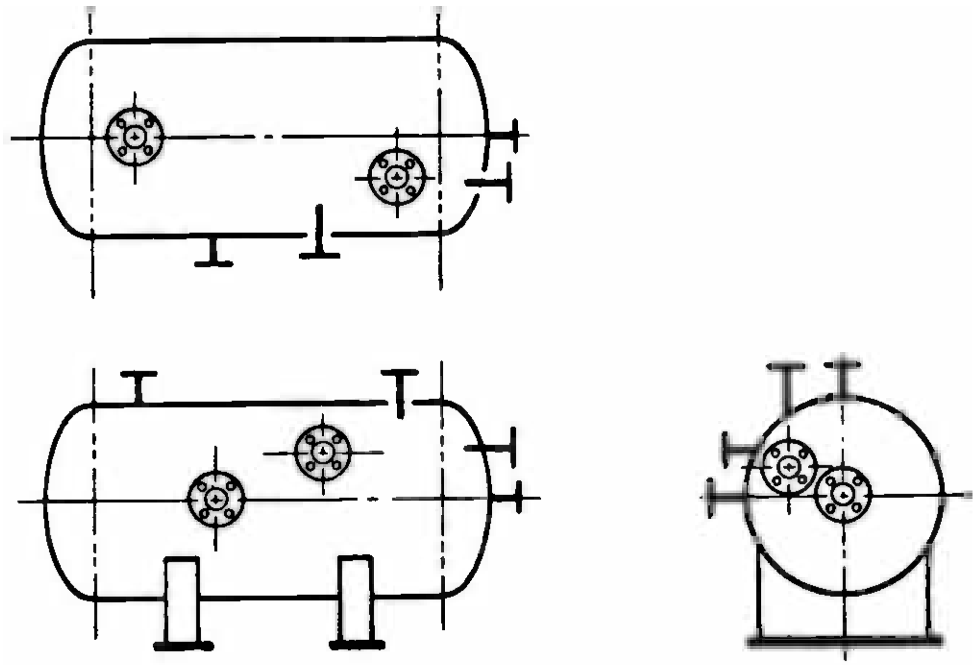


Table N-7
General tolerances: For machined or fabricated parts

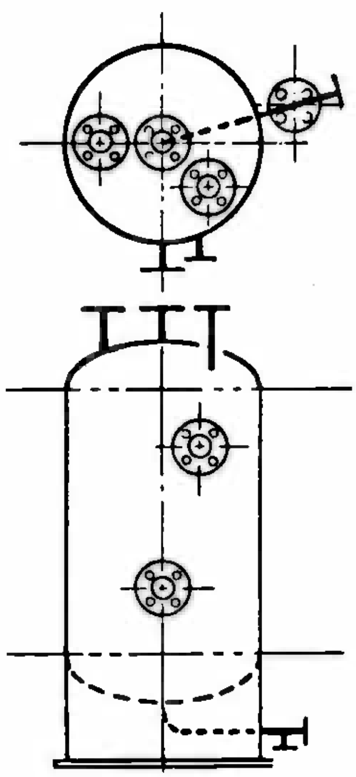
1.0 Machined Parts												
1.1 Linear Dimensions and Diameters												
Distance, mm	.5 to 3	3 to 6	6.1 to 30	31 to 120	121 to 400	401 to 1000	1001 to 2000	2001 to 4000	4001 to 8000	8001 to 12000	12001 to 16000	16001 to 20000
Tolerance + or - mm	0.15	0.2	0.5	0.8	1.2	2	3	4	5	6	7	8
Distance, Inches	.2" to .125"	.126 to .25	0.26 to 1.25	1.26 to 4.75	4.76 to 15.75	15.76 to 39.5	39.51 to 78.75	78.76 to 157.5	157.51 to 315	315.1 to 472	472.1 to 630	630.1 to 787
Tolerance + or - inches	0.01	0.01	0.02	0.03	0.05	0.08	0.12	0.16	0.2	0.24	0.28	0.3
1.2 Radii												
Radii, mm	.5 to 3	3 to 6	6 to 30	30 to 120	120 to 400	400 to 1000	1000 to 2000	2000 to 4000				
Tolerance + or - mm	0.2	0.5	1	2	4	6	8	10				
Radii, Inches	.2" to .125"	.126 to .25	0.26 to 1.25	1.26 to 4.75	4.76 to 15.75	15.76 to 39.5	39.51 to 78.75	78.76 to 157.5				
Tolerance + or - inches	0.01	0.01	0.04	0.08	0.16	0.24	0.3	0.39				
1.3 Angular Dimensions												
Angle, Degrees	0 to 10	10.1 to 20	20.1 to 40	40.1 to 60	60.1 to 90							
Tolerance + or - minutes	15	20	30	45	90							
1.4 Straightness/Planeness (Maximum distance between a straight line and actual line or plane surface and actual surface)												
Length, mm	0 to 6	6.1 to 30	31 to 120	121 to 400	401 to 1000	1001 to 2000	2001 to 4000	4001 to 8000	Above 8001			
Tolerance + or - mm	0.1	0.25	0.5	1	1.5	2.5	3.5	5	7			
Length, Inches	0 to .25	.26 to 1.25	1.26 to 4.75	4.76 to 15.75	15.76 to 39.5	39.6 to 78.75	78.76 to 157.5	157.6 to 315	Above 315.1			
Tolerance + or - inches	0.005	0.01	0.02	0.04	0.06	0.1	0.14	0.2	0.28			
1.5 Rectangularity (Maximum difference in length between the diagonals)												
Distance, mm	0 to 120	121 to 400	401 to 1000	1001 to 2000	2001 to 3000	3001 to 4000	4001 to 5000	5001 to 6000	6001 to 7000	7001 to 8000	8001 to 9000	9001 to 10000
Tolerance + or - mm	2	3	4	7	10	14	18	21	24	28	31	35
Distance, Inches	0 to 4.75	4.76 to 15.75	15.78 to 39.5	39.6 to 78.75	78.76 to 118	118.1 to 157	157.1 to 197	197.1 to 236	236.1 to 275	275.1 to 315	315.1 to 355	355.1 to 395
Tolerance + or - inches	0.08	0.12	0.16	0.28	0.4	0.55	0.7	0.8	1	1.1	1.2	1.4
2.0 Fabricated Parts												
2.1 Linear Dimensions and Diameters												
Distance, mm	0 to 30	31 to 120	121 to 400	401 to 1000	1001 to 2000	2001 to 4000	4001 to 8000	8001 to 12000	12001 to 16000	16001 to 20000	Above 20001	
Tolerance + or - mm	1	2	2	3	4	6	8	10	12	14	16	
Distance, Inches	0 to 1.25	1.26 to 4.75	4.76 to 15.75	15.76 to 39.5	39.6 to 78.75	78.76 to 157.5	157.6 to 315	315.1 to 475	475.1 to 630	630.1 to 790	Above 791	

Tolerance + or - inches	0.04	0.08	0.08	0.12	0.16	0.24	0.31	0.39	0.47	0.55	0.63	
2.2 Radii												
Radii, mm	30 to 120	121 to 400	401 to 1000	1001 to 2000	2001 to 4000							
Tolerance + or - mm	3	6	9	12	15							
Radii, Inches	1.26 to 4.75	4.76 to 15.75	15.76 to 39.5	39.51 to 78.75	78.76 to 157.5							
Tolerance + or - inches	0.12	0.25	0.38	0.5	0.6							
2.3 Angular Dimensions												
Angle, Degrees	0 to 10	10.1 to 20	20.1 to 40	40.1 to 60.	60.1 to 90							
Tolerance + or - minutes	30	45	60	90	120							
2.4 Straightness/Planeness (Maximum distance between a straight line and actual line or plane surface and actual surface)												
Length, mm	30 to 120	121 to 400	401 to 1000	1001 to 2000	2001 to 4000	4001 to 8000	8001 to 12000	12001 to 16000	Above 16001			
Tolerance + or - mm	1	1.5	3	4.5	6	8	10	12	14			
Length, Inches	1.2 to 4.75	4.76 to 15.75	15.76 to 39.5	39.51 to 78.75	78.76 to 157.5	157.6 to 315	315.1 to 472	472.1 to 630	Above 631			
Tolerance + or - inches	0.04	0.08	0.12	0.18	0.24	0.31	0.4	0.47	0.55			
2.5 Rectangularity (Maximum difference in length between the diagonals)												
Distance, mm	30 to 120	121 to 400	401 to 1000	1001 to 2000	2001 to 3000	3001 to 4000	4001 to 5000	5001 to 6000	6001 to 7000	7001 to 8000	8001 to 9000	9001 to 10000
Tolerance + or - mm	2	3.5	7	14	21	28	35	42	49	56	63	70
Distance, Inches	1.2 to 4.75	4.76 to 15.75	15.76 to 39.5	39.6 to 78.75	78.76 to 118	118.1 to 157	157.1 to 197	197.1 to 236	236.1 to 275	275.1 to 315	315.1 to 355	355.1 to 395
Tolerance + or - inches	0.08	0.14	0.28	0.55	0.83	1.1	1.38	1.65	1.93	2.2	2.5	2.75

Flange Bolt Hole Orientation



HORIZONTAL VESSEL



VERTICAL VESSEL

Stress Due to Out of Tolerance Condition

Notation

d = Peaking or banding measurement
d = for peaking;
$.5\ (a_1 + a_2)$ or
$.25\ (b_1 + b_2)$
d = For banding;.
$.5\ (a_1 + b_1)$
e = Amount of offset, in
E = Modulus of elasticity, PSI
L = Half the length of gauge, in
P = Internal pressure, PSIG
R_m = Mean vessel radius, in
t = Vessel thickness, in
β = Factor
ν = Poisson's ratio
σ_P = Stress due to peaking, PSI
σ_O = Stress due to offset, PSI
σ_X = Longitudinal stress, PSI
σ_ϕ = Circumferential Stress, PSI

Note: Peaking and banding are for longitudinal seams only. Offset can be for longitudinal seams or circumferential seams.

Calculations

STRESS DUE TO PEAKING OR BANDING

- Circumferential stress due pressure, σ_ϕ

$$\sigma_\phi = P\ R_m/t$$

- Factor β

$$\beta = \left[2\ L/t\right]\ \left[\left(3\ \left(1-\nu^2\right)\ \sigma_\phi\right)/E\right]^{1/2}$$

For $\nu = .3$;

$$\beta = \left[3.3\ L/t\right]\ \left[\sigma_\phi/E\right]^{1/2}$$

- Stress due to peaking or banding, σ_P

$$\sigma_P = \left(6\sigma_\phi d\ \tanh\beta\right)/\left(t\left(1-\nu^2\right)\beta\right)$$

For $\nu = .3$;

$$\sigma_P = \left(6.59\ \sigma_\phi d\ \tanh\beta\right)/\left(t\,\beta\right)$$

STRESS DUE TO OFFSET

- Circumferential stress due pressure, σ_ϕ

$$\sigma_\phi = P\ R_m/t$$

- Longitudinal stress due pressure, σ_X

$$\sigma_X = P\ R_m/2\ t$$

- Stress due to offset, σ_O

$$\sigma_O = \left(3\ e\,\sigma\right)/\left(t\ \left(1-\nu^2\right)\right)$$

For $\nu = .3$;

$$\sigma_O = \left(3.3\ e\,\sigma\right)/t$$

where σ = Applicable case of σ_X or σ_ϕ

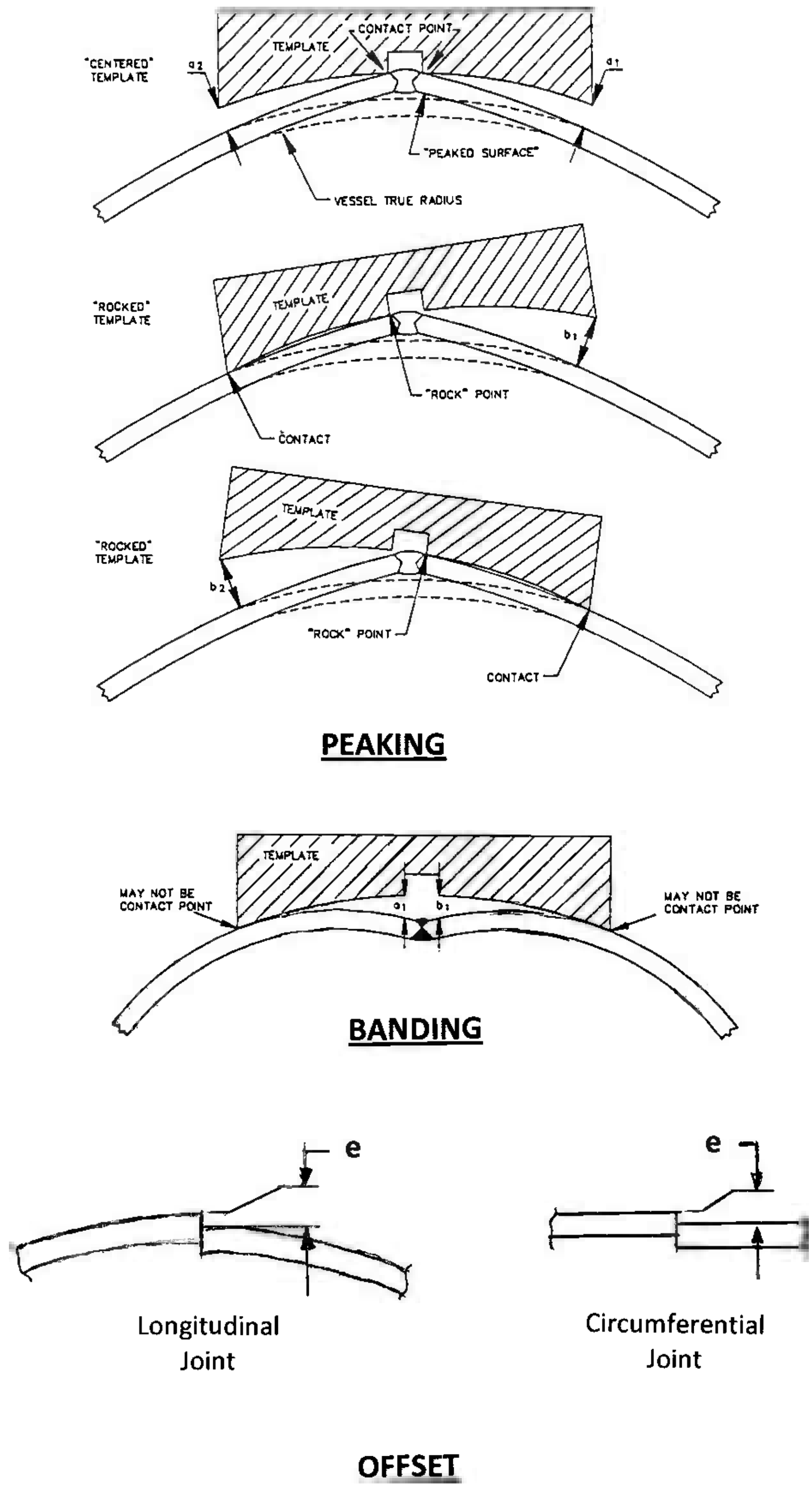
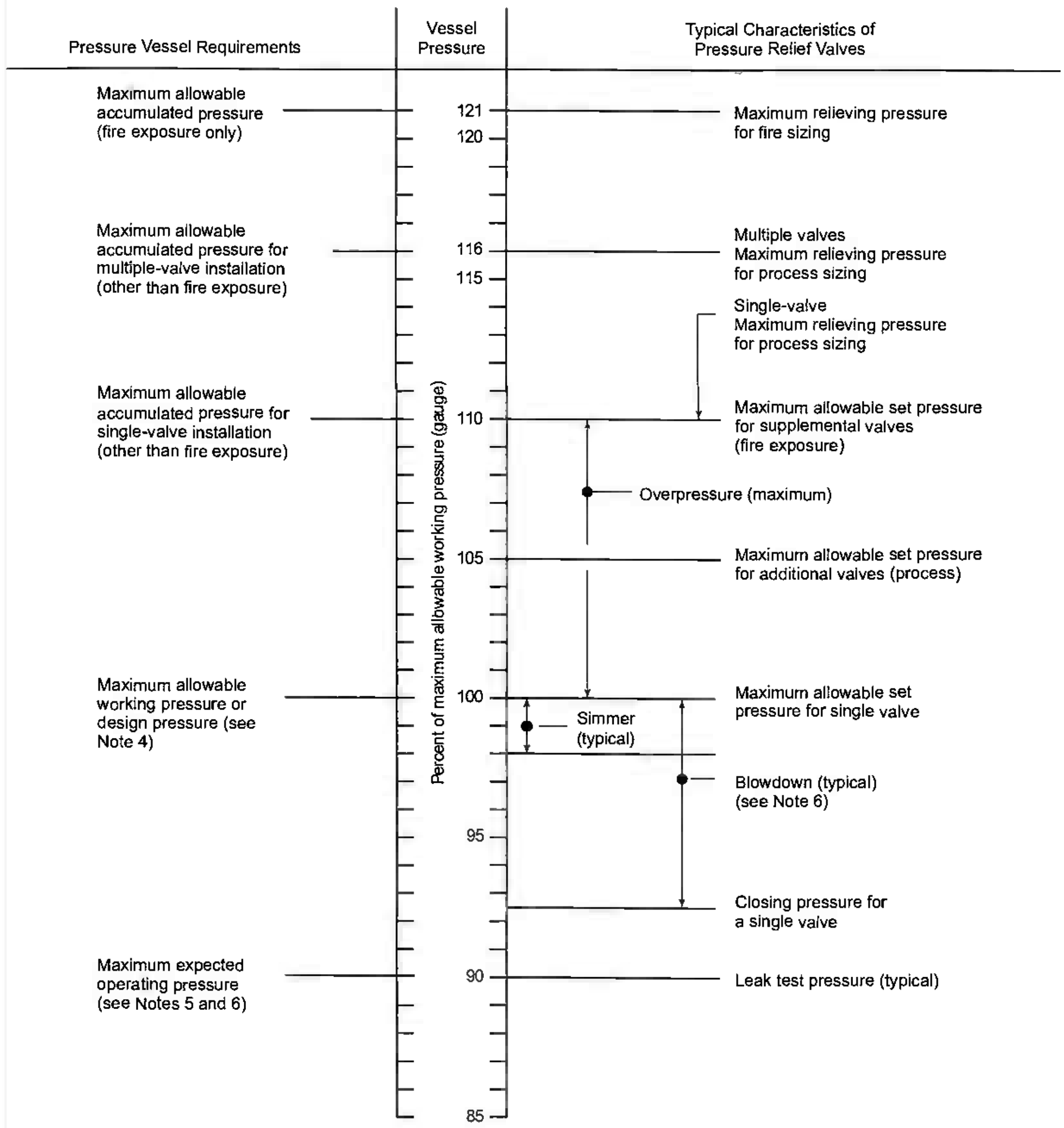


Figure N-3. Measurement of out of tolerance conditions.

Appendix O: Guideline for Application of Pressure Relief Values (PRVs)



NOTES

1. This figure conforms with the requirements of Section VIII of the ASME *Boiler and Pressure Vessel Code* for MAWPs greater than 30 psig.
2. The pressure conditions shown are for pressure relief valves installed on a pressure vessel.
3. Allowable set-pressure tolerances will be in accordance with the applicable codes.
4. The maximum allowable working pressure is equal to or greater than the design pressure for a coincident design temperature.
5. The operating pressure may be higher or lower than 90 %.
6. Section VIII, Division 1, Appendix M of the ASME Code should be referred to for guidance on blowdown and pressure differentials.

Reprinted by permission from API 521.

References

[1] Roark RJ. Formulas for Stress and Strain. 5th Edition. McGraw-Hill Book Co; 1975.
[2] ASME Code, Section VIII, Division 2.
[3] ASME Code, Section VIII, Division 1.

Index

Note: Page numbers with "*f*" denote figures; "*t*" tables.