CASTI Guidebook to
ASME Section VIII
Div. 1 - Pressure Vessels

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Subject Index

Table of Contents

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CASTI Guidebook to
ASME Section VIII Div. 1 - Pressure Vessels

(Covering 2001 Edition of ASME Section VIII Div. 1)

Third Edition

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DEDICATIONS

This work is dedicated to my wife, Margaret Carter. Margaret's constant encouragement to travel to more places and climb more mountains inspired me to write this book.

Will J. Carter, Ph. D., P.E.

To those in the past and present who developed the safety philosophy of the Code, and to those of the future who will continue to develop and expand the philosophy.

Bruce E. Ball, Ph.D., P.Eng.

PREFACE

The American Society for Mechanical Engineers present their Boiler and Pressure Vessel Code with limited explanation and equally frugal examples. Users of the Code who do not have an extensive scientific or engineering knowledge may question the rules of the Code and not appreciate their minimalist nature. Consequently, the philosophy of the Code is lost to many users. As practicing engineers, we understand the need for brief precision and therefore do not find fault with the format of the Code. It is our wish that by writing this book, a broader appreciation for the philosophy of the Code will be achieved.

In this book we do not attempt to put forward new ideas and concepts, but rather to explain well established engineering practice that perhaps, because of its fundamental nature, is overlooked by many Code users. That this occurs is evident in some of the questions posed for Interpretations. If this book prevents only one instance of the Code being circumvented, and the safety of a pressure component being compromised, then our efforts have been worthwhile.

Bruce E. Ball
Will J. Carter

Editor's Note: Practical Examples of using the Code are shown throughout the guidebook in shaded areas. Each Practical Example is numbered and titled. When a calculator icon appears next to a mathematical equation within a Practical Example, it indicates that the equation is "active" in the CD-ROM version. *CASTI's* "active equations" allow the user to enter their own values into the equation and calculate an answer. The "active equations" can be used an unlimited amount of times to calculate and recalculate answers at the user's convenience.

TABLE OF CONTENTS

Chapter 1

INTRODUCTION

History of Boiler and Pressure Vessel Codes in the United States

Perhaps the earliest reference to the design of pressure vessels was made in about 1495 by Leonardo da Vinci in his Codex Madrid I. Quoting from a translation, Leonardo wrote "We shall describe how air can be forced under water to lift very heavy weights, that is, how to fill skins with air once they are secured to weights at the bottom of the water. And there will be descriptions of how to lift weights by tying them to submerged ships full of sand and how to remove the sand from the ships."[1]

Leonardo's pressurized bags of air, if implemented, did not kill or injure large numbers of people and therefore did not force the need for a pressure vessel code. That distinction must go to the early model steam generators.

During the 18th and 19th centuries, steam became the chief source of power and spurred the industrial revolution. By the early 20th century, steam boiler explosions in the United States were occurring at the rate of one per day and claiming about two lives per day. In 1907, after two catastrophic explosions, the state of Massachusetts enacted the first legislation dealing with the design and construction of steam boilers. The resulting regulations were three pages long.

Over the next four years several other states and cities enacted similar legislation. The enacted legislation and the prospect of additional laws and requirements, all with similar yet different requirements, prompted users and manufacturers to seek standardized rules for the design, construction, and inspection of boilers.

In 1911, the Council of the American Society of Mechanical Engineers (ASME) appointed a committee to formulate standard specifications for the construction of steam boilers and other pressure vessels and for their care in service. The first committee consisted of seven members and was assisted by an eighteen member advisory committee. The committee members represented all facets of design, construction, installation, and operation of steam boilers.

The first ASME Boiler Code was issued on February 13, 1915. Six additional sections followed during the next eleven years. The first rules for pressure vessels were issued in 1925. This publication was entitled "Rules for the Construction of Unfired Pressure Vessels," Section VIII.

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[1] Heydenreich, L.H., Dibner, B. and Reti, L., "Leonardo the Inventor," McGraw-Hill Book Company, New York, 1980.

A chronological listing of the year of publication and title of the initial eight sections of the ASME Boiler and Pressure Vessel Code follows:

Section I – Boiler Construction Code, 1914
Section III – Locomotive Boilers, 1921
Section V – Miniature Boilers, 1922
Section IV – Low Pressure Heating Boilers, 1923
Section II – Material Specifications, 1924
Section VI – Rules for Inspection, 1924
Section VIII – Unfired Pressure Vessels, 1925
Section VII – Care and Use of Boilers, 1926

ASME Unfired Pressure Vessel Code

The original Unfired Pressure Vessel Code, Section VIII as prepared by the ASME Boiler Code Committee was concerned largely with riveted construction. However, during the time steam became common place, the process of welding was also being perfected. By 1916, the oxyacetylene process was well developed, and the welding techniques employed then are still used today.

High temperature riveted vessels proved to be unsatisfactory in the chemical industry and particularly unsatisfactory in the petroleum industry. The deficiencies of riveted construction were painfully evident in pressure vessels constructed for the newly developed petroleum cracking process. The cracking process converted the heavy fraction of crude oil into gasoline by heating the crude to a high temperature under pressure. The pressures depended on the process and varied from 100 to 2,000 psi (690 to 13,780 kPa). In such operations, it was found that it was practically impossible to keep riveted vessels tight at high temperatures. The problem was aggravated if the vessel operation contained cycles of heating and cooling.

The first attempts to solve the problem consisted of arc welding the edges of the riveted joints and around the rivet heads. The arc welding available in the early days made use of a bare welding rod which exposed the very hot molten iron that was being deposited to the atmosphere, resulting in the formation of oxides and nitrides in the metal. The resulting weld deposit was usually hard and brittle and sometimes cracked under the conditions of use. This solution, therefore, while an improvement, proved unsatisfactory and led to the construction of vessels by fusion welding of the plates.

The brittle nature of welds made by arc welding resulted in the use of the oxyacetylene welding processes for most of the early welded vessels. This process consisted of heating the edges of the plates with an oxyacetylene flame and joining the surfaces by depositing melted welding rod directly on the surfaces. This process produced satisfactory joints. However, it was troublesome to weld very thick plates because of the difficulty of keeping the edges of the plates hot enough to allow the melted welding rod to fuse to them.

Oxyacetylene welding gave way to electric arc welding when the pressure vessel industry discovered several techniques for protecting the molten iron from the elements in air. The basic idea was to coat the welding rod with a material that kept the oxygen away from the hot molten metal. One of the early coatings used was composed largely of wood pulp which, in the process of welding, burned and formed a gaseous reducing atmosphere at the point of welding. This reducing atmosphere kept the air from combining with the iron. Other types of coating formed a protective slag that floated on the surface of the deposited metal, thereby serving the same purpose. In at least one automatic process, the flux was applied in the groove to be welded ahead of a bare wire rod. The arc was formed beneath the surface of the flux, which melted to form a protective slag coating.

Many welded vessels were constructed in the 1920's and 1930's period. However, the Boiler Code Committee was reluctant to approve the use of welding processes for fabrication of vessels. When the Committee finally approved welding requirements for pressure vessels, they were very restrictive, and required vessels so much heavier than those that had been found safe in practice that the Code requirements were universally ignored.

Later, there was considerable interest by jurisdictional authorities in adopting the ASME Unfired Pressure Code as mandatory requirements for pressure vessel construction. Engineers in the petroleum industry did not agree with many of the provisions of the then existing ASME Unfired Pressure Vessel Code which permitted many things that, in their experience, were unsafe. Also, the nominal safety factor of five required by ASME, the highest of any official code, was greater than had been found necessary in practice.

There was also a difference in philosophy between the ASME Code Committee and the petroleum and chemical industry. This philosophy, while not formally expressed in the codes and standards, had considerable influence on the nature of the code rules and regulations proposed. The petroleum industry had found that, in many cases, vessels experienced corrosion and other phenomena such as creep while operating. Consequently, the industry adopted the position that frequent and careful inspections were as essential to safety as design and construction.

Faced with the prospect of being legally forced to accept the ASME Unfired Pressure Vessel Code, the American Petroleum Institute formed a committee to prepare a code that embodied the successful practice of the industry. After a draft of this code was prepared, it was proposed that the code, when completed, be submitted to the American Standards Association for adoption as an American standard for the petroleum industry. The Boiler Code Committee countered with a suggestion that a joint committee of the American Petroleum Institute and ASME be formed to prepare a code that would be acceptable to both bodies.

The counter proposal was accepted and the joint API–ASME Committee published the first edition of the API–ASME Unfired Pressure Vessel Code in 1934. The new API–ASME code adopted a safety factor of four which, with some of the other improvements such as a requirement for formed heads and elimination of elliptical manways, etc., was felt to produce a vessel that would be initially stronger than many produced using the then existing ASME Code.

For the next seventeen years, two separate unfired pressure vessel codes existed. They were the ASME Section VIII, Unfired Pressure Vessel Code under the control of the ASME Boiler and Pressure Vessel Code Committee and the API–ASME Section VIII, Unfired Pressure Vessel Code under the control of the American Petroleum Institute.

The last API–ASME Unfired Pressure Vessel Code was issued in 1951 and, in 1952, the two unfired pressure codes were merged into one Section VIII. The resulting ASME Section VIII, Unfired Pressure Vessel Code continued until the 1968 edition. At that time it became ASME Section VIII, Division 1, Rules for Construction of Pressure Vessels.

ASME Boiler and Pressure Vessel Code Committee

The ASME Boiler and Pressure Vessel Code Committee consists of several book and service subcommittees. The book subcommittees, such as the Subcommittee on Power Boilers and the Subcommittee on Pressure Vessels, are responsible for publishing code books. The service subcommittees, such as the Subcommittees on Design, are normally staffed with a level of technical expertise not found on the book subcommittees and serve as consultants to the book committees. The two exceptions are the Subcommittee on Materials and the Subcommittee on Welding. These subcommittees serve as both book and service subcommittees.

The subcommittees have numerous subgroups, working groups, and task forces. The subgroups are usually responsible for a certain aspect of vessel construction or a particular technical area or item. For example, the Subcommittee on Pressure Vessels has a Working Group on Layered Vessels, which reports to the Subgroup on Fabrication and Inspection. As the name implies, the Working Group on Layered Vessels is responsible for all matters that relate to the construction of layered vessels.

Committee members volunteer their time and receive no compensation from ASME. They represent all facets of pressure vessel construction and operation. The Boiler and Pressure Vessel Committee meets four times a year to consider revisions and corrections to the Code. It is not unusual for some subgroups and task force groups to meet more or less often than the Main Committee.

The chart in Figure 1.1 shows the structure of the Boiler and Pressure Vessel Committee.

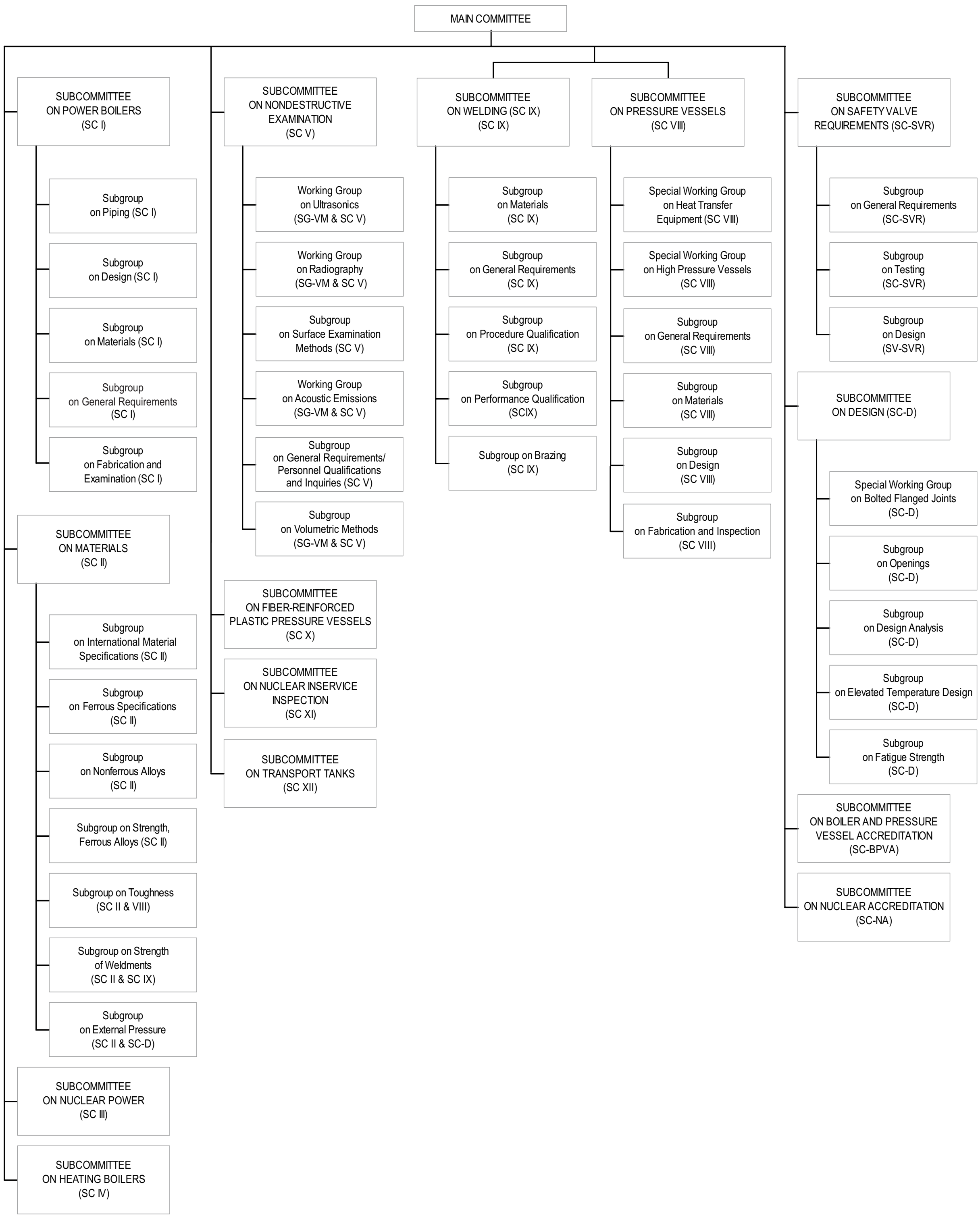


Figure 1.1 ASME Boiler and Pressure Vessel Committee, Subcommittees, and Subgroups.

The sections of the Boiler and Pressure Vessel Code have changed since the 1920s to reflect the needs and improvements in the pressure vessel industry. Section III is currently for nuclear vessels and Section V is Nondestructive Examination. Section VI is entitled Recommended Rules for the Care and Operation of Heating Boilers. The current sections and subsections are:

Sections

I Rules for Construction of Power Boilers

II Materials

- Part A – Ferrous Material Specifications
- Part B – Nonferrous Material Specifications
- Part C – Specifications for Welding Rods, Electrodes, and Filler Metals
- Part D – Properties

III Rules for Construction of Nuclear Power Plant Components

- Subsection NCA – General Requirements for Division 1 and Division 2

III Rules for Construction of Nuclear Power Plant Components – Division 1

- Subsection NB – Class 1 Components
- Subsection NC – Class 2 Components
- Subsection ND – Class 3 Components
- Subsection NE – Class MC Components
- Subsection NF – Supports
- Subsection NG – Core Support Structures
- Subsection NH – Class 1 Components in Elevated Temperature Service
- Appendices

III Rules for Construction of Nuclear Power Plant Components – Division 2

- Code for Concrete Reactor Vessels and Containments

III Rules for Construction of Nuclear Power Plant Components – Division 3

- Containment Systems for Storage and Transport Packaging of Spent Nuclear Fuel and High Level Radioactive Material and Waste

IV Rules for Construction of Heating Boilers

V Nondestructive Examination

VI Recommended Rules for the Care and Operation of Heating Boilers

VII Recommended Guidelines for the Care of Power Boilers

VIII Rules for Construction of Pressure Vessels – Division 1
Alternative Rules for Construction of Pressure Vessels – Division 2
Alternative Rules for Construction of High Pressure Vessels – Division 3

IX Welding and Brazing Qualifications

X Fiber-Reinforced Plastic Pressure Vessels

XI Rules for Inservice Inspection of Nuclear Power Plant Components

New editions of the sections of the ASME Boiler and Pressure Vessel Code are published every three years with the current edition being July 1, 2001. Two addenda, which include additions and revisions, will be issued in July of 2002 and 2003. A new edition consolidates all of the changes made by the addenda to the previous edition.

Code Interpretations

One of the orders of business when the ASME Boiler and Pressure Vessel Committee and its subcommittees meet is to consider Code Interpretations. Code subcommittees handle interpretations in various ways. The following comments relate to the current method used in the Subcommittee on Pressure Vessels.

Code Interpretations are a clarification of existing rules in the Code. They relate to a particular edition and paragraph of the Code and do not make new rules. The request for a Code Interpretation must be in the form of a question and reply. The question must address the paragraph of concern and should be composed in such a way that the reply can be either "yes" or "no." The subcommittee response will also be in the form of a question and reply, although most of the time not the same question and reply submitted by the inquirer.

Often the written rules are not clear and, while the committee members may agree on the intent, the words may not substantiate that intent. In those cases, the subcommittee will issue an intent interpretation. The intent interpretation is also in the form of a question and reply. However, it will have the word "intent" in the question or reply, and the proposed Code Interpretation and a proposed Code revision must be approved simultaneously.

Since interpretations do not make Code rules, the proposed Code revision accompanying an intent interpretation must only clarify an existing rule. Therefore, an intent interpretation is not issued until all levels of the Boiler and Pressure Vessel Code Committee agree that the proposed change is a clarification. The subcommittee's secretary may issue a regular interpretation, after its approval by the appropriate subcommittee members.

Interpretations usually result from a dispute between parties while designing or fabricating a vessel. Realizing this, the Subcommittee on Pressure Vessels attempts to answer all requests for an interpretation in an expeditious fashion. To accomplish this, the Subcommittee established a Special Committee on Interpretations. The Special Committee consists of the subcommittee chairman and vice-chairman and the chairman and a representative from each subgroup reporting to the subcommittee.

The staff secretary of the subcommittee composes a question and reply for all requested interpretations. The staff secretary decides which subgroup has responsibility for the part of the Code that deals with the proposed interpretation and sends the proposed question and reply to the subgroup representatives on the Special Committee on Interpretations. A copy of the proposed question and reply is also sent to the chairman and vice-chairman of the subcommittee.

Unanimous agreement on the proposed question and reply, or a variation of the proposal, results in a speedy reply to the inquirer. If the four members of the Special Committee do not agree, then the proposal is discussed at the next meeting. Lack of a unanimous agreement at the meeting results in the item being sent to the subgroup where it enters the regular Code revision process.

When an inquiry appears on the agenda of a subgroup, the subgroup prepares and votes on a question and reply and sends all subgroup-approved interpretations to the Subcommittee on Pressure Vessels for approval. If approved at the subcommittee level, a reply is sent to the inquirer. Unapproved proposals are sent back to the subgroup.

Interpretations are beneficial not just to the inquirer, but also to the pressure vessel community at large. Therefore, the Boiler and Pressure Vessel Code Committee publishes interpretations semiannually in July and December of each year.

Code Cases

Another form of technical inquiry is the request for a Code Case. Unlike an interpretation, Code Cases usually create new rules and provide the only mechanism for immediately publishing those new rules. They are used to clarify the intent of a Code requirement or to provide alternative requirements.

Code Cases are also written in the form of a question and reply and are usually intended to be incorporated into the Code at a later date. A request for a Code Case should provide a need for the immediate change and background information. The request must be in the form of a question and reply and must identify the applicable Code edition and addenda.

Because Code Cases often result in changes to the Code, they usually require a longer approval time than a Code Interpretation. Appendix 16 of Section VIII, Division 1 outlines the procedure for submitting a technical inquiry and requesting a Code Case.

2001 Edition

The 2001 edition of Section VIII marks a return of metric (SI) units. Prior to 1986, a metric edition of the Section was published. The current position of the Committee is that U.S. customary units are standard and metric units are provided for user convenience. As the metric units are primarily for information, the Code usually only provides a soft conversion. Code users attempting a hard conversion of U.S. customary units to metric or SI units will find that precision of the metric unit is less than the U.S. customary unit. The conversions are also not consistent throughout the Section and not every unit has been converted. This Guide follows the conversions used in the Code.

Appendix V presents conversions from U.S. customary units to metric values used in the Code. The U.S. customary unit is presented first and the metric value follows in parenthesis.

Chapter

2

SCOPE

Each article in ASME Section VIII, Division 1 is identified with an alphanumeric label. This labeling system is common to all the Boiler and Pressure Vessel Code (BPVC) sections. In Division 1 all article labels start with the letter U that symbolizes an article from the unfired vessel section of the Code. Another letter or letters symbolizing the information under discussion in the article follows this letter. The items starting UG come from the general requirements section of Division 1. Items UW are from the general requirements for welding of pressure vessels, UCS articles are from the requirements for fabrications from carbon and low alloy steel materials, and so on. A sequential number follows the alpha descriptors of the item. These numbers are not necessarily consecutive. The Division is continually being reviewed. Articles that are no longer applicable to the current state of the technology may be deleted, or new articles may be inserted that reflect the current state of knowledge. For example, articles dealing with riveted construction of pressure vessels are no longer present in the Division, while recent additions have been made to include further refinements on the use of carbon and low alloy steel materials to reduce the risk of catastrophic failure by brittle fracture.

U–1 Scope

The scope of ASME Section VIII, Division 1 is presented on page 1 of the Division in article U–1. Any pressure retaining vessel, whether the pressure is internal or external to the container, can be designed to meet the requirements of the Division. However, there are specific pressure containers that are not considered under the scope of the Division. These specific pressure containers are:

- items covered by other sections of the Boiler and Pressure Vessel Code
- fired process tubular heaters
- pressure containers that are integral parts of rotating or reciprocating mechanical devices such as motors, pumps, compressors, hydraulic and pneumatic cylinders, and other similar mechanical devices
- piping systems
- pressure containers designed for human occupancy

The application of the Division is shown in Figure 2.1. Attachments made to the pressure container, even though they themselves may not be resisting pressure, are within the scope of the Division (Figure 2.2). The extent of a pressure container is defined by the first connection to that container and includes that connection. [Interpretation VIII–1–95–52 points out that for a welded nozzle consisting of a nozzle neck and a weld neck flange welded to a vessel, the designer may either include the nozzle in the definition of the pressure vessel or exclude the nozzle from the requirements of Division 1 by defining the extent of the pressure vessel either as the vessel to nozzle weld or the flange face of the nozzle.]

Containers that are designed, fabricated, and inspected to meet the requirements of Division 1 can be marked by the letter U when the fabricator is so authorized by ASME (see Application of Section VIII, Division 1 later in this chapter). Smaller containers designed and fabricated in accordance with the requirements of the Division may be exempt from inspection by an independent inspector and as such will be marked by the fabricator with the letters UM when so authorized by ASME. The size and pressure limits of these mass produced vessels is based on stored energy as defined by the following three volume and pressure points:

- 5 cubic feet and 250 psi (0.14 cubic meters and 1720 kPa)
- 3 cubic feet and 350 psi (0.08 cubic meters and 2410 kPa)
- 1½ cubic feet and 600 psi (0.04 cubic meters and 4140 kPa)

Straight line interpolation for intermediate volumes and pressures is permitted.

Containers that are exempt from the requirements of ASME Section VIII, Division 1 but are manufactured in accordance with the requirements of Division 1 by an authorized manufacturer may be marked with U or UM as applicable. This indicates to the user that the container complies with ASME Section VIII, Division 1.

Jurisdictions and owners may require construction in accordance with Division 1, even though the construction is exempt from this requirement. The Division does not prohibit such construction. Numerous Interpretations of article U–1 indicate this. [Interpretation VIII–1–86–132, in response to a query on the construction of a vessel operating at atmospheric pressure and 180°F (82°C), states "The need for determining if Code construction is required is the responsibility of the user or his designated agent."]

Some of the exemption qualifications are based upon the vessel volume. This is the active volume and not necessarily the volume enclosed by the pressure envelope. The volume of internals is excluded. [Interpretation VIII–1–89–23 indicates shell side volume of shell and tube heat exchangers excludes the tube volume, even if the tube side of the exchanger is not exempt by the Division.] The volume exemptions in the Division are based on a consideration of the energy stored within the process environment.

U–1 provides cautions when constructing Division 1 vessels with a maximum pressure greater than 3,000 psi (20,685 kPa). Vessels with design pressures greater than this pressure limit may require design and fabrication principles for thick wall construction. These are not given in the Division. (The Code user may wish to consult ASME Section VIII, Division 3, Alternative Rules for High Pressure Vessels.) However, if the vessel complies with all the requirements of Division 1, it can be marked to indicate the compliance.

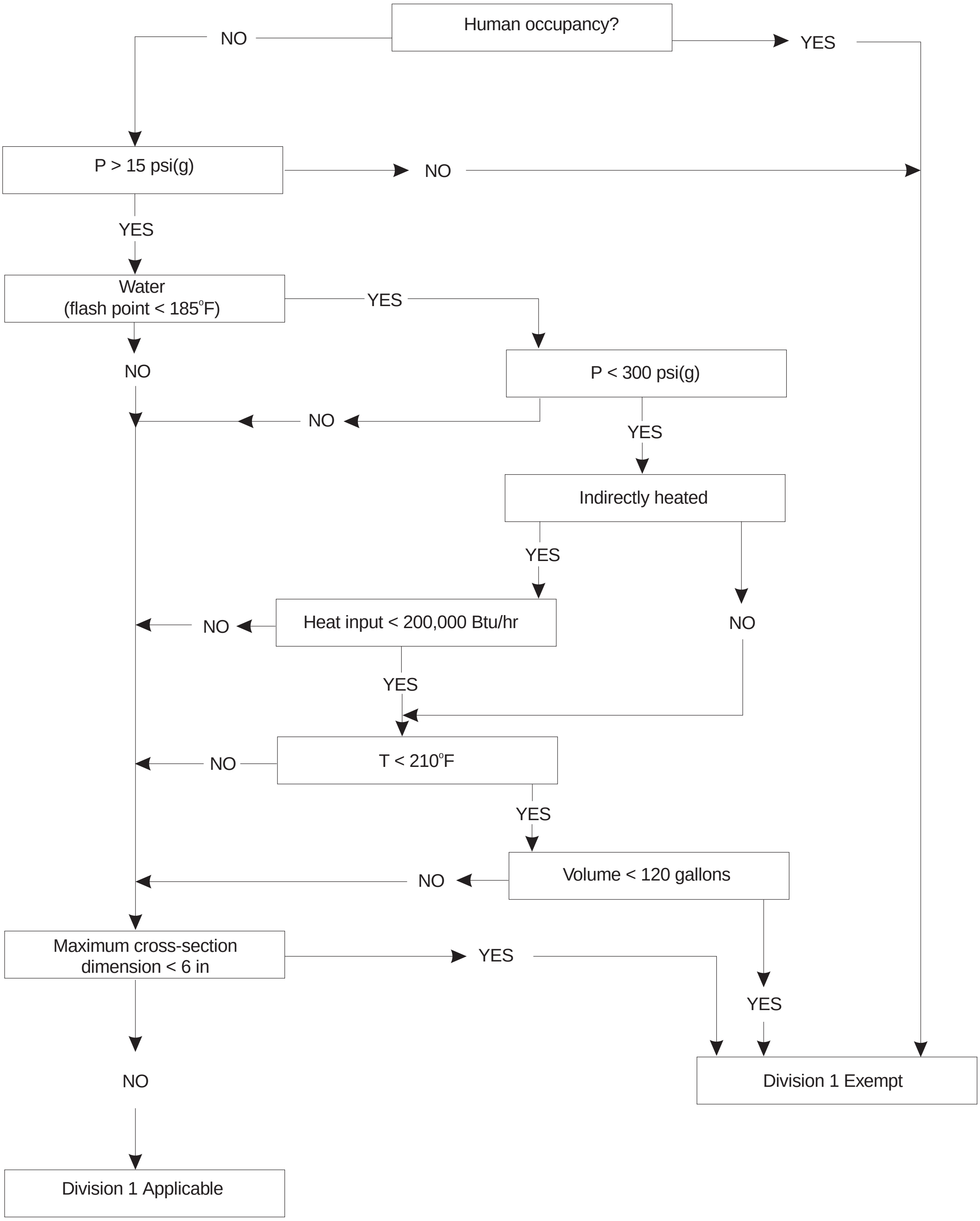


Figure 2.1 Application of Division 1.

Figure 2.2 While the insulation being applied to this pressure vessel is not within the scope of Division 1, the insulation anchors welded to the vessel shell and head must meet Division 1 requirements (UG-55).

Organization of Section VIII, Division 1 (U–1(b))

U–1(b) indicates that Division 1 is arranged into three primary subsections and two types of appendices. A vessel designer will have to refer to all the subsections of the Division as well as to any appendix dealing with the particular aspects of the vessel design. For the user who is unfamiliar with the Division, this usually means an examination of each paragraph in Subsection A (those items identified UG), followed by an examination of each item in the applicable method of fabrication in Subsection B (those items identified UW, UF, and UB). Next, the items in the applicable material type given in Subsection C (items UCS, UNF, UHA, UCI, UCL, UCD, UHT, ULW, and ULT) will need to be reviewed. Specific types of vessels or pressure retaining items within the vessel will have to be reviewed for conformance with the applicable mandatory appendix, if any. If there is an applicable nonmandatory appendix, it should be consulted.

U–1(b) also indicates that Section II, Part D of the Boiler and Pressure Vessel Code is required. This Code section is a recent addition to the Code volumes. Prior to 1992 the data in this section was contained in the applicable Code sections, such as Section VIII, Division 1, that require the data for design calculations. The technical data for materials approved for use in Division 1 vessels is found in Section II, Part D. This information includes maximum stress values, external pressure charts, elastic moduli, Poisson's ratio, and thermal properties for the materials approved for pressure vessel construction.

Application of Section VIII, Division 1

The rules given in the Division may be required for pressure vessel construction as determined by jurisdictional, contractual, or corporate requirements. Fabricators of pressure vessels or pressure relief valves can apply to ASME for certification as fabricators who construct in accordance with the rules of the Division. The certification requirements of ASME are given in UG–117. A quality control program that conforms to the rules of the Division (UG–90) and an inspection of the fabrications by an independent inspector (UG–91) are mandatory.

All pressure equipment built in accordance with Section VIII, Division 1 by an ASME certified fabricator must be permanently marked in accordance with the requirements of UG–115, UG–116, UG–118, UG–119, UG–129, and UG–130 as applicable. Part of the markings on the equipment are U or UM ASME registered trademark symbols which are used for vessels, or UV which is used for pressure relief devices. Figure 2.3 is an example of an approved form of marking for a full size vessel fabricated to Division 1 U requirements.

Figure 2.3 Data plate welded to vessel shell.

In addition to markings on the equipment, data reports as specified in UG–120 or relief valve flow rating reports in accordance with UG–131 must be prepared and provided to the equipment owner. Appendix W gives guidance on completing the required data reports.

Some jurisdictions require pressure vessel fabrication in accordance with the Division but do not require the fabricator to be registered with ASME. In such cases, no U or UM stamping is to be placed on the equipment, although data reports in accordance with the Division are usually required.

U–2 Code User Responsibilities

The organization or individual making use of Section VIII, Division 1 may have externally applied business or legal responsibilities and requirements that are inferred in the Division. There are, however, distinct responsibilities given in the Division to the user of the pressure vessel, the manufacturer of the vessel, and the individual authorized to inspect the vessel.

U–2 indicates that it is the responsibility of the user of the vessel to establish the design requirements for the vessel, always bearing in mind that Division 1 states minimum safety requirements that are not ideals to strive for, but rather are minimum requirements to be met. The manufacturer of the vessel is responsible for the design calculations that show that the maximum permitted material stresses have not been exceeded. In addition, manufacturers are responsible for their own workmanship as well as the workmanship of their subcontractors, including other services, materials, and components. The Authorized Inspector is responsible for ensuring that the vessel is designed and manufactured in accordance with the requirements of Division 1. These responsibilities are to be taken seriously.

U–2(g) indicates that all details of design and construction cannot possibly be covered by the rules of the Division. In such instances, the manufacturer is responsible for developing the principles and techniques required. [Interpretation VIII-1-01-12 indicates that if there are no applicable Code rules for a design, then finite element methods may be performed.] These are to be as safe as those given in the Division and are subject to approval by the Authorized Inspector. (Appendix 3 of this text provides information on some common design principles not detailed in Division 1.)

U–3 Other Standards

No section of the Boiler and Pressure Vessel Code stands alone, and the Code relies on other standards and authorities to qualify some of the requirements to make the Code complete.

U–3 lists the referenced standards that are external to the other sections of the BPVC as well as the particular editions of the standards that are applicable. Appendix 1 of this text gives definitions for terms and acronyms used in this text as well as those used in referencing other standards.

Chapter 3

DESIGN CONSIDERATIONS

As detailed in Chapter 2, Part UG of the Code contains the general requirements for all methods of construction and materials. These general requirements fall into five categories. They are:

- Materials – design aspects of materials such as dimensions, identification, and tolerances
- Design – formula for selection and sizing of vessels and vessel components
- Inspection and Testing – Code required inspection and pressure testing
- Marking and Reports – use of Code markings and stamp and required reports
- Pressure Relief Devices – selection, setting, and installation of pressure relief devices

This chapter outlines and explains the material and design aspects of Part UG. The UG requirements apply to all pressure vessels and vessel parts. These requirements are supplemented by additional requirements in Subsections B and C and the Mandatory Appendices.

MATERIALS

UG–4 through U–9

These paragraphs require that pressure-retaining materials conform to one of the specifications listed in Section II. They must also be listed in Subsection C of Division 1. Subsection C covers specific requirements for the classes of materials allowed in this Division. The Subsection C requirements actually limit the materials to those listed in the stress tables of Section II, Part D or to those covered in a Code case. There are some exceptions to this requirement. The exceptions are described in paragraphs UG–9, UG–10, UG–11, UG–15, and the Mandatory Appendices. These will be discussed later.

Materials may be dual marked or identified as meeting more than one specification or grade. However, the material must meet all the requirements of the identified material specification and grade [Interpretation VIII–1–89–65]. The Division acknowledges the fact that modern mills can and do produce materials capable of meeting several specifications. This is possible because many material specifications state chemical, physical, and mechanical requirements in terms of maximum, minimum, or a range.

Division 1 does not require that nonpressure part materials meet a Code specification if the nonpressure part is not welded to a pressure part. However, if the parts are attached by welding, then the material must be weldable. In addition, if the material does not conform to a Code specification, then the allowable stress value must not exceed 80% of the maximum allowable stress value permitted for a similar Code material. Skirts, supports, baffles, lugs, clips, and extended heat transfer surfaces are examples of nonpressure parts.

Section VIII, Division 1 allows the use of materials outside the size and thickness limits established in the material specification, provided all other requirements of the specification are satisfied. Thickness limits given in the stress tables may not be exceeded.

In general, materials that have not received Code approval may not be used to fabricate Division 1 vessels. If a manufacturer or user wants to use an unapproved material, then material data must be submitted to and approved by the Boiler and Pressure Vessel Committee. The two routes for this approval are a Code Case and Appendix 5 of section II, Part D. Code Case approval is quicker and is usually for a specific purpose. Such an approval does not carry with it inclusion in Section II. If the intent is to include the material in the Code for general use, then the process outlined in Appendix 5 must be followed.

UG–5 through UG–7 provide specific requirements for plate, forgings, and castings. Allowable stress values for these product forms are given in Section II, Part D. However, for castings, the values given must be multiplied by the casting factor given in UG–24 for all cast materials except cast iron.

UG–8 gives requirements for seamless or electric resistance welded pipe or tubes that conform to a specification in Section II. As with other product forms, the allowable stress values for pipe and tubes are given in Section II, Part D. Note that the values in Section II for welded pipe or tubes have been adjusted downward because of the weld. The Code allowable stress values for welded pipe or tubes are 85% of the values for the seamless counterparts.

Pipe or tubes fabricated by fusion welding with a filler metal are permitted in Division 1 construction only as a Code fabricated pressure part. As a pressure part, the fusion welded pipe or tubes must satisfy all the Division requirements for a Code part. Those requirements include material, welding, design, inspection, and testing. No further consideration for the weld is required in Division 1 calculations.

Paragraph UG–8 also sets forth the requirements for integrally finned tubes fabricated from tubes that conform to one of the specifications given in Section II. The integrally finned tubes may be used providing the following considerations are met:

- After finning, the tube has a condition that conforms to the specification or to the specified "as-fabricated condition."
- The allowable stress value for the finned tube is the value given in Section II for the tube before finning, or

- A higher value for the as-fabricated finned tube may be used if the appropriate mechanical tests demonstrate that the condition obtained conforms to one of those provided in the specification, and allowable stress values for that condition are in the allowable stress table found in ASME Section II, Part D.
- The maximum allowable internal or external working pressure is the smaller of the values based on either the finned or the unfinned section. Alternatively, Appendix 23 may be used to establish the maximum allowable external pressure.
- Each tube after finning shall either be pneumatically tested at not less than 250 psi for 5 seconds or hydrostatically tested per UG–99.

UG–9 is one of those exceptions to using a Code given specification. This paragraph points out the advantages of using a welding material listed in Section II, Part C. When the welding material does not comply with a specification in Section II, then the material marking or tagging must be identifiable with the welding material used in the welding procedure specification. [Interpretation VIII–83–343 indicates that individual welding materials need not be separately tagged but may be taken from a tagged container provided that the manufacturer's quality control system has provision for maintaining the material identity.]

Example 3.1 Dual Markings of Materials

The following table lists two chemicals and two tensile requirements of plate material SA–516.

Table 3.1 Selected Requirements for SA–516

Requirement	**Grade 55**	**Grade 60**	**Grade 65**	**Grade 70**
Carbon, max. – ½ in. and under	0.18%	0.21%	0.24%	0.27%
Manganese – ½ in. and under	0.60–0.90%	0.60–0.90%	0.85–1.20%	0.85–1.20%
Tensile strength - ksi	55–75	60–80	65–85	70–90
Yield strength, min. - ksi	30	32	35	38

Careful examination reveals that a material with a maximum carbon content of 0.18%, manganese content of 0.90%, 70 ksi tensile strength, and a yield strength of 38 ksi will satisfy the requirements for all grades of SA–516. If the material also meets all other requirements of the specification, then it may be marked for all four grades of SA–516. When the designer selects the appropriate grade, the complete design must be based on the selected grade.

UG–10 Material Identified with or Produced to a Specification Not Permitted or a Material Not Fully Identified

This paragraph provides a procedure for using materials that are not properly certified or are unidentified. The noncertified or unidentified materials may fall into three classes:

- identified materials with complete certification from the material manufacturer. This class includes materials to a specification not permitted or materials procured to a chemical composition only;
- material identified to a particular production lot as required by a specification permitted by this Division, but for which complete certification is not available, and
- material not fully identified.

The rules require that mechanical and chemical properties be obtained for each piece of material and that the properties satisfy a permitted specification. Deoxidation practice, metallurgical structure, and heat treatment requirements, as applicable, must also satisfy the requirements of the permitted specification. Once it is established that the unidentified material does satisfy these requirements, it may then be used as the material of the permitted specification.

UG–11 Prefabricated or Preformed Pressure Parts

The manufacturer of a pressure vessel may use prefabricated or preformed parts such as shown in Figure 3.1. Parts subjected to pressure must comply with all applicable requirements of the Code and be furnished to the vessel manufacturer with a Partial Data Report.

There are three exceptions to the above requirement. They are:

- Cast, forged, rolled, or die formed standard pressure parts do not require shop inspection, materials identification per UG–93(a), material certification per UG–93(b), or a Partial Data Report. Standard pressure parts are pipe fittings, flanges, nozzles, welding necks, welding caps, manhole frames, covers, and similar parts. The part may comply with an accepted ANSI standard. Alternatively, the part may be manufactured to a manufacturer's standard as long as the part is marked with a traceable identifying mark and is made from permitted materials.
- Cast, forged, rolled, or die formed nonstandard pressure parts such as shells, heads, removable doors, and pipe coils may be supplied as materials. These parts must be marked with the name of the part manufacturer and identifying marks traceable to the material.
- Welded standard pressure parts for use other than the shell or heads of vessels do not require in-shop inspection, materials identification per UG–93(a), material certification as specified in UG–93(b), or a Partial Data Report. If the part is made in accordance with an accepted ANSI standard, then the material may be acceptable. The part manufacturer must mark the part as above and, if radiography or postweld heat treatment is required, certify that it was done according to the Code.

Manufacturers who are not authorized by ASME to use the U or UM stamp may furnish parts that comply with the above exceptions. However, parts that are to be used for fabrication in Canada must be registered with and approved by the jurisdictional authority. This is a requirement of the Canadian Standards Association Standard CSA B51 which is incorporated into the Boiler and Pressure Vessel Regulations of all Canadian jurisdictions.

Figure 3.1 This large shop fabricated head is being used in the field fabrication of a catalytic cracker for a refinery.

UG–12 and UG–13 Fasteners

Fasteners may be used to attach removable pressure vessel parts. All fastener components must be manufactured in accordance with a Code specification and comply with all Code requirements. Bolts and studs must be selected to ensure that the maximum stress does not exceed the maximum allowable stress values given in Section II, Part D.

Additional dimensional requirements for studs are given. They must either have full length threads, or the unthreaded part may be machined down to the root diameter of the threads if the threaded portion is at least 1½ diameters long.

The unthreaded portion of long studs, that is studs with a length greater than eight diameters, may be reduced to the nominal diameter of the thread if the reduced portion is at least 0.5 diameters long and dynamic loads on the stud are considered. Suitable transitions must be provided between sections. The dynamic load consideration usually requires a fatigue analysis, the procedures for which are not given in the Division.

Nuts must engage the bolt threads for the full depth of the nuts. Washers are optional. A threaded fastener can have a complex load distribution. In most cases, the first three engaged threads transfer the load, and of these, the first engaged thread is the most significantly stressed. The usual threaded connection requires four engaged threads to ensure complete load transfer without overloading any one thread in a properly made up threaded connection. Standard nuts are sized to ensure a minimum of four threads when the connection is made for the full depth of the nut. Washers are used to ensure nut seating (see Chapter 6), or, for a high tensile bolt, to limit the loss of tightening torque from galling. Code bolting rules do not allow high bolt tension, and thus galling on bolt tightening should not occur. Hence the optional requirement stated for washers.

Missing Grade of a Product Specification

UG–15 provides for the use of grades of a material when that grade is not listed in the particular specification in Subsection C, but is listed in another specification in Subsection C. The paragraph states that the material with no grade given in the desired specification may be used provided it can be shown to have the required chemical and physical properties, tolerances, and other engineering data required for the grade which is given elsewhere in permitted materials. Mill test reports must reference the specification used in producing the material and paragraph UG–15. [Interpretation VIII–1–89–194 provides an example of acceptance to UG–15. Alloy 317L is available as plate and is listed under SA–240. It can be used as bar or pipe materials under SA–479 and SA–312, even though it is not listed under these specifications. The material must meet the chemical and physical requirements of SA–240 and the product and quality requirements of the applicable product specification.]

UG–16 General

The formulas throughout the Code represent dimensions in the corroded condition.

The minimum thickness of shells and heads after forming, regardless of material and product form, is $\frac{1}{16}$ inch (1.6 mm) exclusive of corrosion allowance. Exceptions are:

- heat transfer plates of plate type heat exchangers;
- inner pipe of double pipe heat exchangers or tubes of shell type heat exchangers where the pipes or tubes are NPS 6 (DN 150) or less;
- unfired steam boilers where the minimum thickness shall be $\frac{1}{4}$ inch (6 mm) exclusive of corrosion allowance; and
- shells and heads in compressed air , steam, and water service made from materials listed in UCS–23 which shall have a minimum thickness of $\frac{3}{32}$ inch (2.4 mm) exclusive of corrosion allowance.

Vessels made of plate furnished with a mill undertolerance of not more than the smaller of 0.01 inch (0.3 mm) or 6% of the ordered thickness may be considered as full design thickness. In most designs this does not present a problem since the general specification for rolled plates, SA–20, limits the mill thickness undertolerance to 0.01 inch (0.3 mm) for all plates up to 15 inches (381 mm) thick.

Pipe and tube material may be ordered by its nominal wall thickness. However, the manufacturing undertolerance must be taken into account when designing or ordering the component. The undertolerance need not be considered when designing nozzle wall reinforcement.

Example 3.2 Selection of Pipe Schedule

A pressure vessel is to be fabricated from 20 inch diameter SA–53 pipe. The required thickness is 0.35 inch. No corrosion allowance is required.

Solution: Specification SA–53 allows a minimum wall thickness of 87.5% of the specified nominal thickness. For economical reasons, pipe for pressure use is commonly supplied as a standard series of wall thicknesses called pipe schedules. The nominal and minimum thicknesses for three schedules of 20 inch diameter SA–53 pipe are given below.

Schedule	Nominal Wall Thickness	Minimum Wall Thickness
Standard	0.375 inch	0.328 inch
Schedule 30	0.500 inch	0.437 inch
Schedule 40	0.594 inch	0.520 inch

While the standard wall pipe has sufficient nominal wall thickness, Division 1 states that manufacturing undertolerance must be taken into account when ordering pipe. Therefore, the designer must order at least Schedule 30 pipe, which is the smallest schedule with a minimum wall thickness greater than 0.35 inch. The minimum wall thickness of the schedule 30 pipe is 0.437 inch.

UG–19 Special Construction

Section VIII, Division 1 allows a manufacturer to design and fabricate a vessel of any configuration, size (see Figure 3.2), shape, or combination thereof. Paragraph UG–19 provides general guidance for nontraditional designs.

If a vessel contains independent pressure chambers, then each chamber must be designed to withstand the most severe condition or combination of coincident pressures and temperatures expected during normal operation. Normal operation includes both startup and shutdown of the vessel, but does not include hydrostatic testing. (The Code user should consult UG–99 for further information on hydrostatic testing multichamber vessels.)

When design rules are not provided in the Division, then the vessel or part may be designed under the provisions set forth in U–2. Those provisions allow the manufacturer, subject to the acceptance of the inspector, to use engineering principles to provide a design that will be as safe as one produced using the rules of Division 1. If the strength of the vessel or part cannot be calculated with satisfactory accuracy, then the maximum allowable working pressure may be determined using a proof test. Proof test procedures and requirements are given in paragraph UG–101.

Figure 3.2 This 17 ft (5.2 m) diameter by 180 ft (54.9 m) long horizontal mounted vessel dwarfs the person standing beside it at midlength. The vessel is awaiting transport to the installation site.

UG–20 Design Temperature

Paragraph UG–20 establishes the requirement for maximum design temperature and minimum design metal temperature. This paragraph requires that UCS–66 be used for carbon and low alloy materials to determine if impact testing is required (see Chapter 6).

The maximum design temperature shall be equal to or greater than the mean metal temperature expected during operation. The maximum temperature at any point shall not exceed the maximum temperature listed in the stress tables for the specification, or the external pressure chart, if the vessel is under external pressure.

Division 1 states that any combination of coincident pressure and temperature during a specific operation is permissible, provided the combination does not constitute a more severe condition than that assumed in the design.

The Code does not explain what constitutes a "more severe condition." However, it is generally assumed that any combination of coincident pressure and temperature that produces a stress greater than the allowable stress for that temperature is a more severe condition. All Code requirements must be satisfied for all combinations. Stamping the vessel for all combinations of coincident pressure and temperature is optional. Footnote 37 in paragraph UG–116 states that more than one combination of coincident pressure and temperature may be added to the vessel markings.

The minimum design metal temperature shall be the lowest temperature expected in service. The lowest metal temperature shall be the mean temperature of the metal and must consider the lowest operating temperature, operational upsets, autorefrigeration, atmospheric temperature, and other sources of cooling.

The minimum design metal temperature marked on the nameplate must correspond to the greatest value of the maximum allowable working pressure marked on the nameplate.

The material parts, such as Part UCS or Part UHA, specify when to use impact testing. Part UCS materials are more prone to brittle fracture than other materials and therefore must often be impact tested. However, impact testing is not mandatory when all of the following are met:

- the material is P–No. 1 Group 1 or Group 2;
- the nominal thickness is equal to or less than ½ inch (13 mm) for materials listed in Curve A of Fig. UCS–66;
- the nominal thickness is equal to or less than 1 inch (25 mm) for materials listed in Curve B, C, or D of Fig. UCS–66;
- the complete vessel is hydrostatically tested per UG–99(b); UG–99(c) or App. 27-3
- the design metal temperature is -20°F (-29°C) minimum and 650°F (343°C) maximum, occasional operating temperatures colder than -20°F (-29°C) are acceptable if due to lower seasonal atmospheric temperature;
- thermal or mechanical shock loads are not a controlling design condition (UG–22); and
- cyclical loading is not a controlling design condition (UG–22(e)).

Division 1 does not provide guidance on determining the design temperatures other than to state that they may be computed or measured if there is equipment in service under equivalent operating conditions. The selected maximum design temperature should provide maximum flexibility for future changes in operating requirements at a minimum cost premium.

Once the maximum anticipated operating temperature has been determined from a thorough analysis of the process, the following should be considered in arriving at the maximum design temperature.

Vessels in hot service (above ambient temperature) should have a design temperature not less than the maximum anticipated operating temperature plus a 25 to 50°F (15 to 30°C) allowance. A reconsideration of the amount of the allowance and the stability of the process may be necessary when this requirement results in an increase in the material grade or flange rating. If no allowance is provided, then the maximum operating temperature value should be set high enough to allow for process upsets and should be on the conservative side.

The maximum allowable stress values for certain alloy materials fall off rapidly with increasing temperature. For example, the allowable stress value for 316 stainless steel plate at 400°F (204°C) is 31% less than the allowable stress value at 100°F (38°C). When this occurs, the temperature allowance and operating process stability should be reconsidered.

The maximum allowable stress value for many carbon and low alloy steels is constant between -20 and 400°F (-29 to 204°C). To take advantage of this, the design temperature can be upgraded in most cases. When the vessel design temperature for service is less than the maximum temperature of the constant allowable stress, the design temperature appearing on the nameplate may be upgraded to the highest value allowed by the Code for the as-built condition.

When different temperatures can be predicted for different zones of a vessel, these variant temperatures may be taken into account in the design.

Example 3.3 Design Temperature Selection

Consider a carbon steel vessel with a 150 psig design pressure and a 250°F operating temperature.

Solution: Normally the design temperature for this vessel would be 275°F or 300°F. The values are determined by adding 25°F or 50°F to the operating temperature. However, since the allowable stress value for this vessel at 500°F would not be less than the allowable value at 300°F, selection of a 500°F design temperature and ASME/ANSI B16.5 Class 150 flanges would be appropriate.

Note that a design temperature of 650°F would necessitate the use of Class 300 flanges for the vessel since the pressure rating of a Class 150 flange at 650°F is 125 psig less than the vessel design pressure. In this case, assigning a design temperature less than 500°F would unnecessarily restrict the use of the vessel.

Design Pressure

UG–21 states that vessels must be designed for the most severe condition of coincident pressure and temperature expected in normal operation. The Code also states that the maximum pressure difference between the inside and outside or between chambers of a vessel during operation and test must be considered.

Beyond the guidelines stated above, no Code guidance is given for determining the design pressure. In general, the design pressure is established by adding an allowance to the maximum internal or external pressure at which the vessel operates while fulfilling its normal function. The design pressure must be sufficiently higher than any expected operating conditions to permit satisfactory operation without activating the overpressure protection devices.

For vessels subjected to internal pressure only, the design pressure is normally not less than the maximum operating pressure plus approximately 10% or 25 psi (170 kPa), whichever is greater. Vessels subject to external pressure should be designed for full vacuum where practical. Vessels designed for vacuum should also be designed and stamped for a small internal pressure. Internal pressure of 25 to 50 psi (170 to 345 kPa) is common.

Many vessels are designed with chambers subjected to different pressures. Common examples are heat exchangers and jacketed vessels. One must give much thought and consideration to the operating conditions, including startup and shutdown, before establishing differential pressure values. As a general guideline, unless the process is very stable, jacketed vessels operating under external pressure as a result of the jacket fluid should be designed for the maximum anticipated jacket pressure plus 10% with no credit for any internal pressure in the vessel.

Example 3.4 Design Pressure Selection

Process design has requested that a vessel be designed to accommodate a new process. The vessel must be a jacketed vessel. The process is stable and the normal operating pressure of the inner vessel is 425 psi and the normal operating pressure of the jacket is 500 psi. During startup, the inner vessel must be pressurized to 350 psi before pressure can be applied to the jacket. Once the pressure in the inner vessel reaches 350 psi, the outer chamber is pressurized to 425 psi and then both chambers are slowly pressurized up to their operating pressures while maintaining a differential pressure between chambers of 75 psi or less. During shutdown the process is reversed. The process is relatively stable and computer controlled. Select a set of design pressures for the vessel.

Solution: The inner vessel is exposed to both internal and external pressure. The maximum operating external pressure on the inner vessel during startup, normal operation, and shutdown is 75 psig. Therefore, the external design pressure should be 75 psi plus an allowance. Utilizing the previous recommendation:

inner vessel external design pressure = 75 psi + 25 psi = 100 psi.

The maximum internal pressure in the inner vessel must be based on the internal pressure during startup and the differential pressure during normal operation. In both cases this value is 350 psi.

inner vessel internal design pressure = 350 psi + (0.10)(350 psi) = 385 psi.

The jacket is subjected to internal pressure only. The maximum internal operating pressure is 500 psi.

jacket internal design pressure = 500 psi + (0.10)(500 psi) = 550 psi.

This shows that design pressure and temperature are highly dependent on operating procedures.

Other Loadings

UG–22 requires that the designer consider all loads acting on the vessel. Loads shall include those from:

- internal and external pressure,
- weight of vessel plus normal and test contents including static head,
- weights of attachments (see Figure 3.3),
- internal attachments,
- vessel support attachments,
- cyclic and dynamic reactions due to pressure, thermal, or mechanical loads,
- wind, snow, and seismic loads,
- impact reactions such as those due to fluid shock, and
- temperature gradients and differential thermal expansion.
- abnormal pressures, including those caused by deflagration.

This is a commonly overlooked requirement of the Division. Since the Code does not provide design guidance or formulas for loads other than internal and external pressure, many designers overlook this requirement. [Interpretation VIII–1–86–191 indicates that static head is to be considered in a horizontal pressure vessel and that the Division provides no specific rules for how this is to be done. Interpretation VIII–1–89–143 indicates that the design thickness of a vessel shell shall be sufficient to accommodate the stresses imposed by damage protection devices attached to the shell. Interpretation VIII–1–95–82 indicates that the design must consider all loads, including those of the hydrostatic test.]

Figure 3.3 These ladders and platforms place loads on the vessel which the designer must consider (UG–22).

A batch reaction vessel provides an example of significant nonpressure loads. Such operations often include both temperature and pressure cycles. In such cases, the designer should be aware that, depending on the number and magnitude of the pressure and temperature cycles, the vessel could potentially fail from fatigue if these cycles are not recognized in the design.

UG–23 Maximum Allowable Stress Values

Allowable Tensile Stress

Materials which may be used to construct Section VIII, Division 1 vessels are listed in Subsection C. In general, the maximum allowable tensile stresses for these materials are given in Section II, Part D. The values are established so that when a vessel operates below the creep and stress rupture range, the material does not experience general yielding or stresses near the tensile strength of the material. At temperatures in the creep and stress rupture range, allowable tensile stresses are selected to ensure a low creep rate and reasonably long stress life. Allowable stresses for UCI, UCD, and ULT materials are listed in their respective parts of Division 1.

The criteria for wrought or cast ferrous and nonferrous materials, except strength enhanced bolting materials, structural grades of ferrous plate, cast iron, and cast ductile iron follow.

At temperatures below the creep/rupture range, the smallest of:

- $\frac{1}{3.5}$ of the specified minimum tensile strength,
- $\frac{1}{3.5}$ of the tensile strength at temperature,
- $\frac{2}{3}$ of the specified minimum yield strength, or
- $\frac{2}{3}$ of the yield strength at temperature.

The factor on yield strength may be increased to 0.9 for certain austenitic stainless steels and nickel and nickel alloys where greater deformation can be tolerated. Bolting and flanges are excluded.

At temperatures in the creep and stress rupture range, the smallest of:

- the average stress to produce a creep rate of 0.01% in 1,000 hours (assumed to equal 1% in 100,000 hours),
- 80% of the minimum stress to produce rupture at the end of 100,000 hours, or
- 67% of the average stress to produce rupture at the end of 100,000 hours.

Specific criteria for other materials and product forms are given in Appendix 1 of Section II, Part D. Tables 1A and 1B of Section II, Part D present the allowable tensile stress for each material at design temperature increments of 50°F (approximately 28°C). See Table 3.2.

Table 3.2 Allowable Stress Table, Excerpt from Section II, Part D, Table 1A

Line No	Nominal Comp.	Product Form	Spec No.	Type/ Grade	Alloy Design./ UNS No.	Class/ Cond./ Temper	Size/ Thickness, in.	P–No.	Group No.
23	Carbon steel	Forgings	SA-508	1	K13502	---	---	1	2
24	Carbon steel	Forgings	SA-508	1A	K13502	---	---	1	2
25	Carbon steel	Forgings	SA-541	1	K03506	---	---	1	2
26	Carbon steel	Forgings	SA-541	1A	K03506	---	---	1	2
27	Carbon steel	Cast Pipe	SA-660	WCB	J03003	---	---	1	2
28	Carbon steel	Forgings	SA-765	II	K03047	---	---	1	2
29	Carbon steel	Plate	SA-515	70	K03101	---	---	1	2
30	Carbon steel	Plate	SA-516	70	K02700	---	---	1	2
31	Carbon steel	Wld. Pipe	SA-671	CB70	K03101	---	---	1	2
32	Carbon steel	Wld. Pipe	SA-671	CC70	K02700	---	---	1	2
33	Carbon steel	Wld. Pipe	SA-672	B70	K03101	---	---	1	2
34	Carbon steel	Wld. Pipe	SA-672	C70	K02700	---	---	1	2

Line No.	Min. Tensile Strength, ksi	Min. Yield Strength, ksi	Application & Max. Temperature Limits (NP = Not Permitted) (SPT = Supports Only)			External Pressure Chart No.	Notes
			I	III	VIII-1		
23	70	36	NP	700	1000	CS-2	G10, T2
24	70	36	NP	700	1000	CS-2	G10, T2
25	70	36	NP	700	1000	CS-2	G10, T2
26	70	36	NP	700	1000	CS-2	G10, T2
27	70	36	1000	700	NP	CS-2	G1, G10, G17, G18, S1, T2
28	70	36	NP	NP	650	CS-2	---
29	70	38	1000	700	1000	CS-2	G10, S1, T2
30	70	38	850	700	1000	CS-2	G10, S1, T2
31	70	38	NP	700	NP	CS-2	S5, W10, W12
32	70	38	NP	700	NP	CS-2	S6, W10, W12
33	70	38	NP	700	NP	CS-2	S5, W10, W12
34	70	38	NP	700	NP	CS-2	S6, W10, W12

Line No.	Maximum Allowable Stress, ksi (Multiply by 1000 to obtain psi), for Metal Temperature, °F, not exceeding													
	-20 to 100	150	200	250	300	400	500	600	650	700	750	800	850	900
23	20.0	20.0	20.0	---	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3	6.7
24	20.0	20.0	20.0	---	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3	6.7
25	20.0	20.0	20.0	---	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3	6.7
26	20.0	20.0	20.0	---	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3	6.7
27	20.0	---	20.0	---	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3	6.7
28	20.0	20.0	20.0	---	20.0	20.0	19.6	18.4	17.8	---	---	---	---	---
29	20.0	20.0	20.0	---	20.0	20.0	20.0	19.4	18.8	18.1	14.8	12.0	9.3	6.7
30	20.0	20.0	20.0	---	20.0	20.0	20.0	19.4	18.8	18.1	14.8	12.0	9.3	6.7
31	20.0	---	20.0	---	20.0	20.0	20.0	19.4	18.8	18.1	---	---	---	---
32	20.0	---	20.0	---	20.0	20.0	20.0	19.4	18.8	18.1	---	---	---	---
33	20.0	---	20.0	---	20.0	20.0	20.0	19.4	18.8	18.1	---	---	---	---
34	20.0	---	20.0	---	20.0	20.0	20.0	19.4	18.8	18.1	---	---	---	---

Prior to 1998, the design factor for calculating the allowable stress was 4. In 1998, Code case 2278 provided for a maximum allowable stress based on a design factor of 3.5 with respect to the specified minimum tensile stress. The Code Committee also approved Code Case 2290 at the same time, and this Case listed the maximum allowable stress for common materials used in pressure vessel construction. The increase in the allowable stress is possible because of the improved quality in manufacture and inspection of today's wrought and cast materials. Materials such as bolting and cast irons, have other features that limit the allowable stress rather than strength alone and therefore these products do not directly benefit from modern manufacturing capabilities and their permitted stress values remain as before Code Case 2278.

In 1999, the Code Committee authorized the publication of the revised stress values in Section II Part D and Code Case 2290 was withdrawn. Code Case 2278 remains in effect because it presents the basis on which the higher maximum permitted stress values are determined. Nonmandatory Appendix P presents the basis for establishing allowable stress values based on a design factor of 4 which was the basis on which allowable stress values were determined before Code Case 2278. In some instances, for example, the design of large diameter flanges (greater than 40 inches (1016 mm)), the potential for leaks at the flange increases with the use of permitted stress values derived from a design factor of 3.5. The Code user may wish to use the permitted stress values derived by using the design factor of 4 for vessels in lethal or flammable service.

Allowable Longitudinal Compressive Stress

The maximum allowable longitudinal compressive stress for a Section VIII, Division 1 vessel is the smallest of the maximum allowable tensile stress value permitted in the stress table or a factor B value taken from the external pressure charts of Subpart 3 of Section II, Part D. The factor B is a function of the outside radius and thickness of the shell and is derived from buckling considerations. A flow chart for the complete procedure is given in Figure 3.4.

The thickness of a vessel must be determined such that any combination of loads listed in UG–22 that occurs simultaneously during normal operation will not produce a general primary membrane stress greater than the allowable tensile stress.

If bending exists, then the loads shall not produce a combined maximum primary membrane stress plus primary bending stress across the thickness greater than 1½ times the maximum allowable stress in tension.

When random loads, such as wind or seismic, combine with other loads, the maximum general primary membrane stress may be increased to 1.2 times the maximum allowable tensile or compressive stress.

Seismic and wind loads need not be considered to act simultaneously. The maximum allowable stress values must be those given for the temperature that the metal will be at when loaded. The maximum stress values in the tables and the B factors on the external pressure charts may be interpolated for intermediate temperatures.

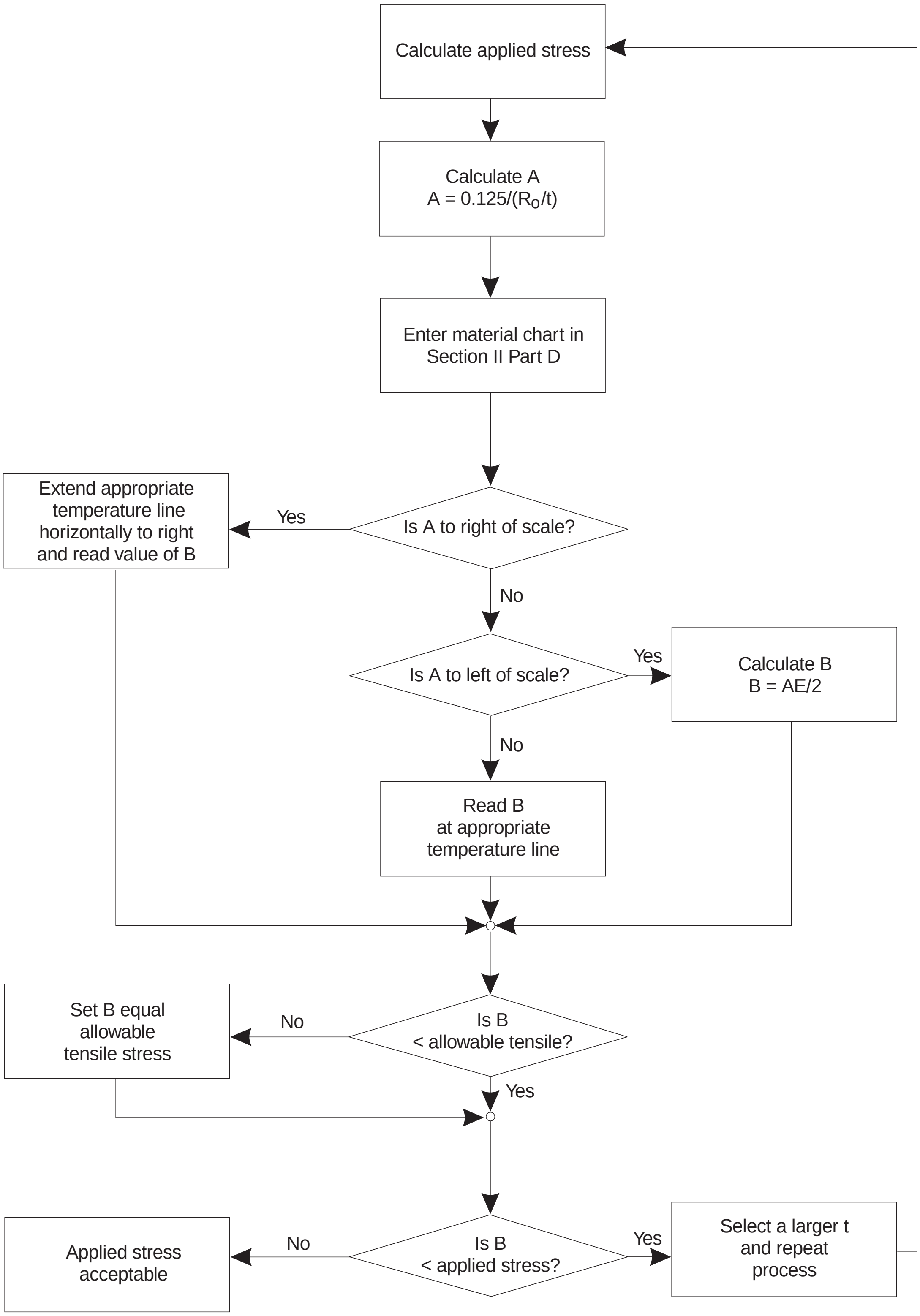


Figure 3.4 Maximum Allowable Longitudinal Stress.

Just as the allowable stress for tensile loading has been increased through the application of Code Case 2278, so the allowable stress for compressive loading can similarly be increased. Again this is only applicable by using a Code Case, Case 2286. However, as allowable compressive stresses are determined primarily from elastic buckling, which is geometry controlled, the restrictions for increased compressive loading as given in Code Case 2286 must be considered. The enhancement of permitted compressive loads is not applicable without the Code user specifying the application of the Code Case and this may not be acceptable to all owners and jurisdictional authorities.

UG–24 Castings

UG–24 details the supplementary machining and examinations that must be performed before using a casting quality factor greater than 80%. The casting quality factor must be used in the Division 1 equations that require an efficiency or E value. The requirements for using a quality factor greater than 80% are listed below.

- All surfaces of centrifugal castings shall be machined after heat treatment to a finish not coarser than 250 microinch (6.35 μm), and a factor not exceeding 85% shall be applied.

- For nonferrous and ductile cast iron materials, a factor not greater than 90% may be applied if, in addition to the above:
  1. all surfaces of each casting are subjected to a thorough defect free examination,
  2. at least three pilot castings from the first lot of five for a new or altered design are sectioned or radiographed at all critical sections without revealing any defects,
  3. one additional casting taken at random from every subsequent lot of five is sectioned or radiographically examined without revealing any defects, and
  4. all castings other than those sectioned or radiographed are examined by magnetic particle or liquid penetrant techniques.

- For nonferrous and ductile cast iron materials a factor not to exceed 90% may be used for a single casting if it is radiographically examined at all critical sections and found to be free of defects.

- For nonferrous and ductile cast iron materials, a factor not greater than 90% may be applied if the casting has been machined to the extent that all critical sections may be examined for the full wall thickness as in a tubesheet with the holes spaced no farther apart than the thickness. This exam may be conducted in lieu of a radiographic or destructive exam.

- For carbon, low alloy, or high alloy steels, the following factors may be used if the additional examinations are performed:
  1. For centrifugal castings, a factor of 90% may be used if the casting is magnetic particle or liquid penetrant examined.
  2. For static and centrifugal castings, a factor of 100% may be applied if the castings are examined in accordance with all the requirements of Appendix 7.

- The following requirements apply to castings in vessels containing lethal substances:
    1. Cast iron and cast ductile iron are prohibited.
    2. Each casting of nonferrous material must be radiographed at all critical sections without revealing any defects. A quality factor of 90% may be used.
    3. Steel castings shall be examined according to Appendix 7 for severe service applications. The quality factor shall not exceed 100%.

- When defects have been repaired by welding, the completed repair shall be reexamined. To obtain a 90% or 100% quality factor, the repaired casting must be stress relieved.

- Each casting for which a quality factor greater than 80% is applied shall be marked with the name, trademark, or other identification of the manufacturer as well as the casting identification including the quality factor and material designation.

UG–25 Corrosion

Provisions must be made to ensure the desired life of a vessel or part when it is subjected to thinning due to corrosion, erosion, or mechanical abrasion. The action may consist of a corrosion allowance, which is an increase in the thickness of the material over that required by the design formulas, or some other means of accommodating material loss such as a metallic or nonmetallic lining.

Vessels subjected to corrosion must be provided with a drain or drain pipe positioned to relieve liquid accumulation at the lowest point of the vessel.

Small holes, having a diameter of $\frac{1}{16}$ inch to $\frac{3}{16}$ inch (1.6 to 4.8 mm) and a depth not less than 80% of the equivalent thickness for a seamless shell, may be used to detect thickness loss. Such holes, called telltale holes, while allowed by the Code, are not recommended. Telltale holes are located on the surface opposite the surface experiencing the metal loss.

The user or his agent must specify the corrosion allowances. When no corrosion allowance is provided, this must be indicated on the Data Report.

The strength contribution of corrosion resistant or abrasion resistant linings shall not be considered unless the lining is designed in accordance with Part UCL.

Chapter 4

FABRICATION

The general requirements for fabrication are given in UG–75 through UG–83. UG–76 allows material to be cut to size by thermal or mechanical means (Figure 4.1). The complete material marks or other means to clearly identify what those marks are is to be assigned to all the pieces as stated in UG–77 and shown in Figure 4.2. Quality requirements for roundness of formed shapes are given in UG–79, UG–80, and UG–81. When material imperfections are detected they can be repaired as approved by the inspector (UG–78).

As indicated in Chapter 2, Section VIII, Division 1 is split into three subsections. Subsection B presents the rules applicable to the methods of welding, forging, and brazing fabrication of vessels. These methods can be used together or alone.

Figure 4.1 An oxygen cutting torch is used to cut the shaped plate to length. After cutting, the slag and metal discoloration will have to be mechanically removed (UG–76).

Figure 4.2 Die stamp identification marks on a section of carbon steel plate that has been formed into a vessel shell.

Fabrication by Welding

Part UW (Unfired Welded) contains the rules for construction of pressure vessels by welding. These rules are used in tandem with the general requirements of Subsection A and the material requirements of Subsection C of the Division.

Weld Processes

UW–9 allows only butt welds to be made using the pressure welding processes listed in UW–27(b), namely flash welding, induction welding, resistance welding, thermit pressure welding, gas pressure welding, inertia welding, continuous drive friction welding, and explosive welding. In all these processes, pressure or blows are imparted to the materials during the fusion process.

Arc and gas welding can be used to make groove welds, fillet welds, and overlay welds. Arc welds are limited to the following processes given in UW–27(a):

- shielded metal arc welding (SMAW)
- submerged arc welding (SAW)
- gas metal arc welding (GMAW)
- gas tungsten arc welding (GTAW)
- plasma arc welding (PAW)
- electroslag welding (ESW)
- electrogas welding (EGW)
- electron beam welding (EBW)
- laser beam welding (LBW)
- hydrogen metal arc welding

Flux core arc welding (FCAW) is recognized as a variety of GMAW in Section IX of the Code and is therefore recognized as being acceptable for Division 1 welding fabrication (UW–27(c)).

The only gas weld procedure recognized by the Division is oxyfuel gas welding (OFW).

UW–28 specifies that all welding to the pressure envelop be done to a weld procedure specification (WPS) qualified in accordance with the rules of Section IX of the Code. Similarly, UW–29 requires welders and weld equipment operators to be qualified in accordance with Section IX. UW–47 and UW–48 require the inspector to verify the qualifications of the weld procedures and the welders and weld equipment operators. (Specific information on weld procedure and welder requirements is given in *Practical Guide to ASME Section IX* - Welding Qualifications, published by *CASTI* Publishing Inc.)

UW–3 Weld Joint Classification System

In Section VIII, weld requirements are given in accordance with the nature of the principal stress at the weld joint due to internal pressurization. Four conditions of stress geometry are defined and termed as Category A, Category B, Category C, and Category D joints.

UW–3 and its accompanying figure, Figure UW–3, reproduced below, give the definitions for the various joint locations. Category A joints are generally transverse to the maximum stress created by the pressure containment. Category B joints are oriented parallel to the direction of maximum stress created by pressure containment and, as well, join parts of one dimension symmetry whose axes of symmetry are parallel. Category C joints are oriented parallel to the direction of maximum stress created by pressure containment and join parts of generally one dimension symmetry whose major axes of symmetry are not parallel. Category D joints are the joints between an appurtenance and the main pressure containment vessel or vessel subcomponent.

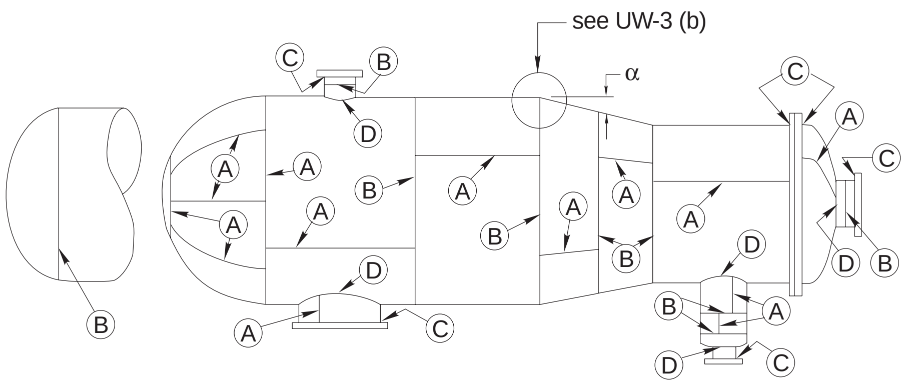


Figure 4.3 Section VIII Fig. UW–3.

Weld Joint Designs

Various weld joint configurations are permitted for arc and gas welding. These configurations include butt joints, lap joints, corner joints, tee joints, and edge joints. Unacceptable joint configurations are those that leave a crack-like configuration that would be subjected to tensile loading. The risk of failure at such configurations is significant. UW–9 makes it clear that groove welds must be designed to provide complete fusion and penetration.

There are very few restrictions on the joint detail in a WPS developed in accordance with Section IX of the Code (see *CASTI Guidebook to ASME Section IX* - Welding Qualifications published by *CASTI* Publishing Inc.). Inexperienced Code users are advised to restrict their designs to joint details presented pictorially in the figures of Subsections B and C of the Division as given here in Table 4.1.

Table 4.1 Weld Joint Details Given in Section VIII, Division 1

Typical Joint Connection	Applicable Figures in Division 1
Butt weld, plates of unequal thickness	UW–9, UW–13.1, ULW–17.1
Butt weld, weld necks to materials of unequal thickness	UW–13.4, ULW–17.1
Head to shell	UW–13.1, ULW–17.2, ULW–17.3
Nozzle or other appurtenance abutting a shell or head	UW–13.2, UW–13.3, UW–13.5, UW–16.1, UW–16.2,UHT–18.1, UHT–18.2, ULW–17.3, ULW–18.1
Stay bolts to shell or flange	UW–19.1
Tube to tubesheet	UW–20, ULW–17.3
Small fittings and couplings to shell or head	UW–16.1, UW–16.2

UW–9 specifies a minimum taper transition of 3 to 1 when joining materials of unequal thickness. This is illustrated in Fig. UW–9 where unequal thickness is quantified as two materials differing in thickness by ¼ the thickness of the thinner part, or by ⅛ inch (3.2 mm), whichever is less. Any change in material continuity serves to magnify the stress at the change. The more abrupt the change, the greater the stress magnification. In addition, stress concentrators in close proximity have a multiplying effect. A weld represents an interruption in metallurgical continuity and is therefore a stress magnifier. Weld reinforcement is an interruption in the geometry of the material surface, so a butt weld joining two materials of different thickness can be a very highly stressed area in a pressure vessel. Fig. UW–13.1(l) through (o) further illustrates the taper transition requirement. Figure 4.4 illustrates the stress concentration effect of various thickness taper angles.

UW–13 lists a number of special requirements for thickness transitions. The double transition thickness reduction specified in Fig. UW–13.4 is an important requirement that is frequently overlooked. This double transition occurs between heavy wall weld neck flanges and pipe nozzles. The double transition taper is an expensive machining operation that some manufacturers attempt to avoid by machining only a single long taper. If the length of the taper is too short, a significant increase in stress concentration can result.

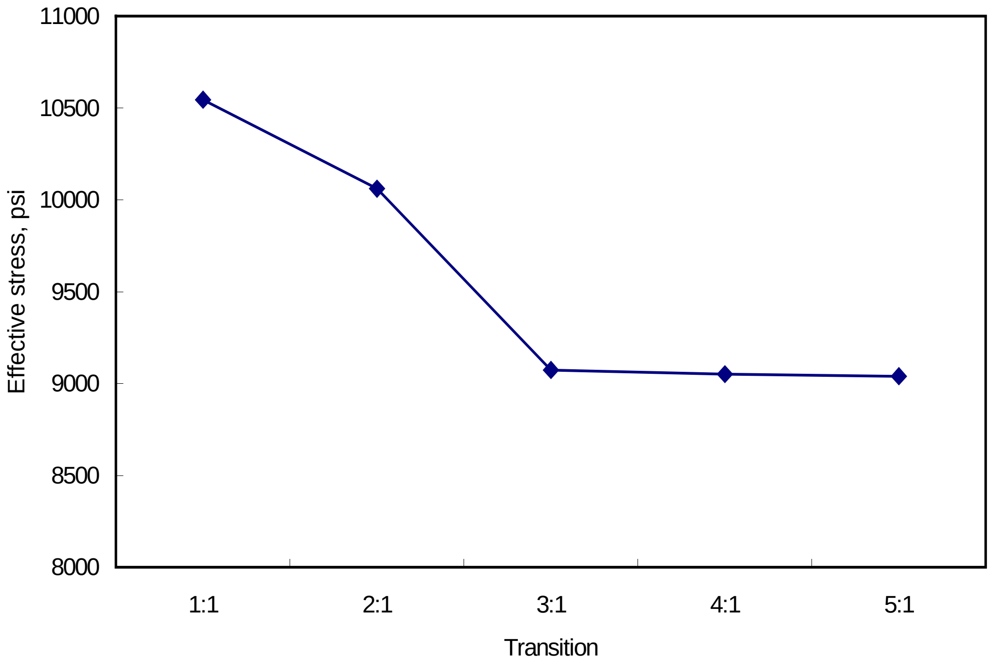


Figure 4.4 Effect of transition slope on a 1.25 to 1.00 inch (32 to 25 mm) transition in a vessel wall (FEA von Mises analysis).

Example 4.1 Stress Concentrations Can Cause Failure

An investigation into the cause of fracture at a weld joint between a weld neck valve and a pipe nozzle revealed that a single 30 degree thickness transition had been machined onto the weld neck transition right to the edge of the weld bevel preparation. This was found to represent an approximate 30% increase in stress magnification over the Code specified maximum. Failure was caused by bending stresses. Cracking originated at the toe of the weld cap on the valve side of the joint.

UW–9 (d) provides requirements for offset abutting longitudinal weld seams. There is a stress concentration at the junction of the longitudinal weld and the circumferential weld. Two abutting longitudinal welds in close proximity further amplify the stress. There is also a concern for fracture resistance as weld deposits and weld heat affected zones frequently are more susceptible to fracture because of imperfection concentration, residual stress, loss of ductility, and reduced fracture toughness. While alignment of longitudinal seams may appear correct to linear thinking, this arrangement represents a risk of failure and is to be avoided. When longitudinal joints abut across a circumferential joint within a distance less than 5 times the thickness of the thicker material, UW–9 requires the structural quality of the longitudinal welds to be verified by radiographic examination for a minimum 4 inch (100 mm) length on either side of the circumferential weld. This represents the reduction of only one of the failure risks and the designer may wish to consider other options to further reduce the risks.

UW–9(e) gives minimum lap dimensions for lap joint designs. This degree of overlap ensures adequate joint stiffness and is also a Division feature to reduce stress magnification because of the double change in material continuity created by the lap configuration.

Fillet weld joints must also meet certain minimum requirements. Throughout the Division, specific uses for fillet welds are given along with minimum fillet weld size requirements. In year 2001, rules for socket weld and slip-on flanges were added to the Division in UW-21 where the minimum size of the attaching fillet weld is specified and both sides of the weld joint for slip-on flanges is required to be welded. UW–9(g) requires that fillet weld sizes be calculated for maximum permitted stress. This will be discussed in Chapter 9.

The designer may consider using fillet welds to reduce surface contour stress risers. However, the fillet weld deposit also creates a stress magnification because of microstructure changes produced by the weld heat. This same heat also produces residual stress. Overwelding to increase weld transition taper should be used with caution.

UW–18 (a) contains a cautionary note indicating fillet weld joint details must be designed to ensure complete penetration at the weld root. This means that re-entrant corners of less than 30 degrees for SMAW fillet weld joint details should not be used. An even larger angle may be required for other weld processes.

Plug welds are a special joint configuration for a fillet weld. UW–17 gives special size limitations for plug welds where:

Φ = diameter of the plug weld joint preparation
t = thickness of the plate with the plug weld hole preparation
$t + 1/4$ inch $< \Phi < 2t + 1/4$ inch ($t + 6.4$ mm $< \Phi < 2t + 6.4$ mm)

Where plug welds are used for stayed construction (UG–47 through UG–50), UW–19 specifies a maximum plug weld joint preparation diameter of 1¼ inch (32 mm).

The plug weld design should specify the electrode size in relation to the hole size and material thickness to ensure that complete fusion is obtained at the weld root. Welders should be instructed to place the root fillet weld all around the hole before proceeding with any fill. For general use plug welds, UW–17 specifies that the minimum weld fill shall be the plate thickness for 5⁄16 inch (8 mm) and thinner plates, and 5⁄16 inch (8 mm) or ½ the plate thickness, whichever is the greater value, on plates thicker than 5⁄16 inch (8 mm). The joint can be completely filled, but the previous comments on overwelding are applicable. For stayed construction plug welds, weld filling is optional on thicknesses 3⁄16 inch (4.8 mm) or less when the hole is 1 inch (25 mm) diameter or less.

UW–12 Weld Joint Efficiency

The weld joint efficiency is a reliability factor assigned to the joint based on the joining process, the joint type and detail, and the degree of joint examination. This factor is based on the structural integrity of the material, with a forged product as the base standard representing an efficiency of 1. In the design calculations discussed in the subsequent chapters, the joint efficiency will be demonstrated and is represented both in this guide and in the Division by E.

UW–12, and in particular Table UW–12, lists the joint efficiencies to be used for various types of welds, joint details, and extent of radiographic examination. For weld joints loaded only in compression, radiographic examination is not required and the joint efficiency is a maximum of 1. Table UW–12 identifies six types of arc or gas weld details.

Type 1 welds are butt joints detailed to ensure high structural quality through the depth of the weld. They incorporate a detail requiring gouging or grinding of the root area of the second side of the weld. This is done to achieve sound metal before making the weld on the second side. Type 1 joints can be used to make all joint categories of butt welds and are shown pictorially in Figure 4.5. If these welds are examined radiographically for their entire length and they meet the quality requirements of the Division with or without repairs, they can be considered structurally equivalent to a forging and are accordingly assigned a joint efficiency factor of 1. If only a portion of the weld length is examined radiographically, then a maximum joint efficiency of 0.85 can be used in required thickness calculations. A maximum joint efficiency of 0.70 is to be used if only visual examination is carried out on the weld. All welds are to be visually examined whether or not they are also examined by other means.

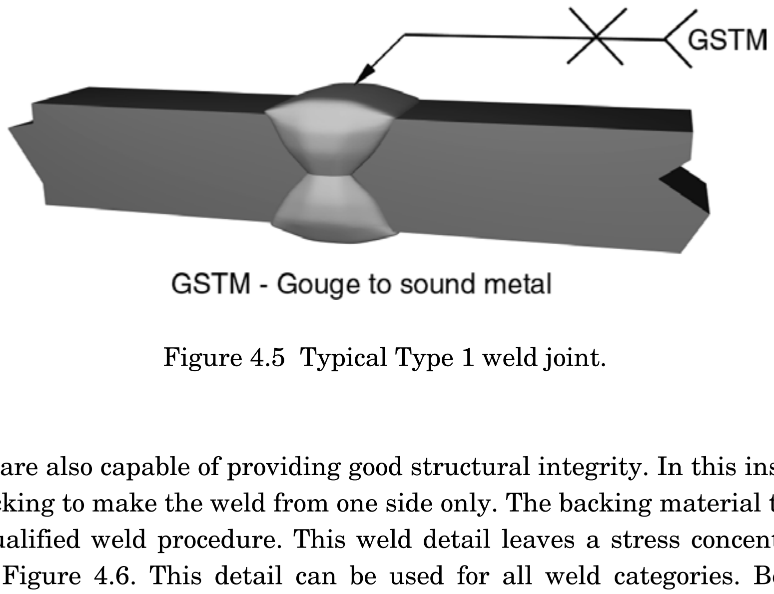


Figure 4.5 Typical Type 1 weld joint.

Type 2 butt welds are also capable of providing good structural integrity. In this instance, weld metal is deposited on backing to make the weld from one side only. The backing material to be used must be specified in the qualified weld procedure. This weld detail leaves a stress concentrator at the weld root as shown in Figure 4.6. This detail can be used for all weld categories. Because of reduced radiographic sensitivity and the stress concentration at the weld root, this detail has a lower efficiency. For full radiographic examination, a maximum value of 0.90 can be used, and, for spot radiographic examination, a maximum value of 0.80 can be used. With no radiographic examination, the maximum efficiency shall be 0.65. The backing strip detail is not good in corrosive environments regardless of the concentration of the corrodant. The crevice at the weld root can trap corrodants and this could result in concentration cell corrosion.

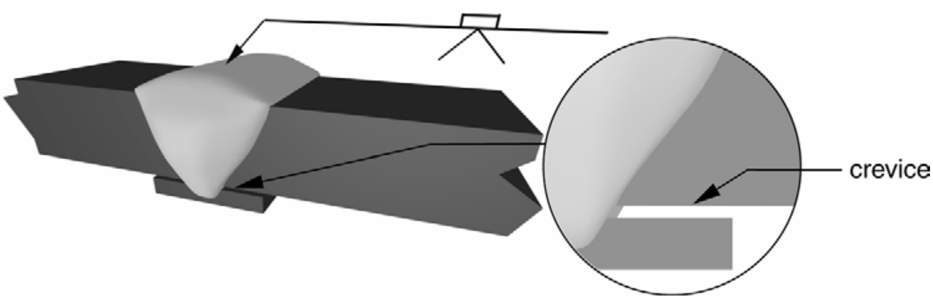


Figure 4.6 Typical Type 2 weld joint.

Type 3 welds are limited to circumferential butt welds for category A, B, and C joints. These welds are made by welding from only one side of the joint with no backing. This requires the molten deposited weld metal to freeze in the air to make a bridge between the two sides of the butt joint. As such, this type of joint is susceptible to poor or incomplete fusion of the weld metal at the root. This imperfection has the features of a crack and can be difficult to detect. Category D joints have low inspection sensitivity, and therefore welds made from one side without a backing strip (Type 3 welds) are not to be used for joining nozzles and communication chambers to the main vessel. A typical Type 3 weld is shown in Figure 4.7.

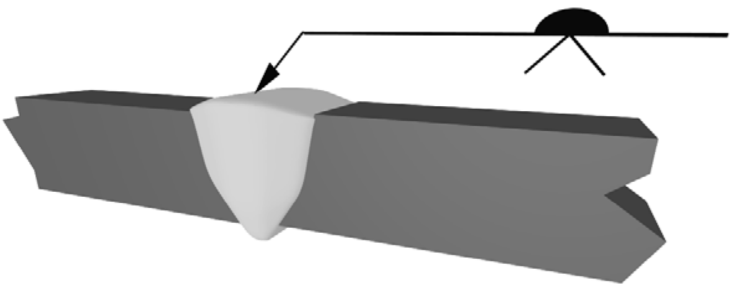
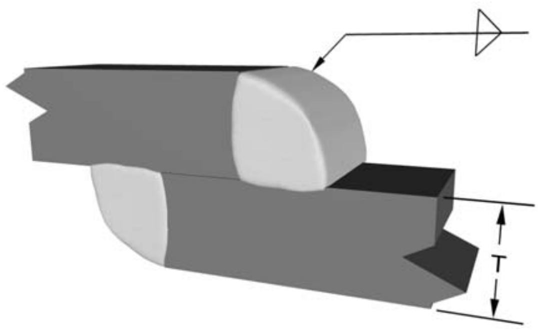


Figure 4.7 Typical Type 3 weld joint.

Type 4 welds are restricted to category A, B, and C joints and are made by overlapping material and placing fillet welds at either end face of the lap. This configuration, as illustrated in Figure 4.8, results in a radiographic film image that is difficult to interpret because of the thickness gradation in the area of interest. Accordingly, Type 4 joints are not considered radiographically examinable. A maximum efficiency of 0.55 is to be used for this joint configuration because of limited inspectability, the risks of incomplete weld root penetration, and the possibility of lamellar tearing. The requirement for weld root penetration is stated in UW–36. Lamellar tearing results from residual weld stress in the thickness direction of wrought materials causing cracking along the grain boundaries, particularly when the material has been cold worked, when it contains a significant concentration of nonmetallic contaminants, or when both conditions exist. Type 4 weld joints can only be used with materials that have sufficient working, caused by thickness reduction, to break up nonmetallic grain boundary films. Accordingly, maximum thicknesses are specified in Table UW–12 to ensure sufficient working.

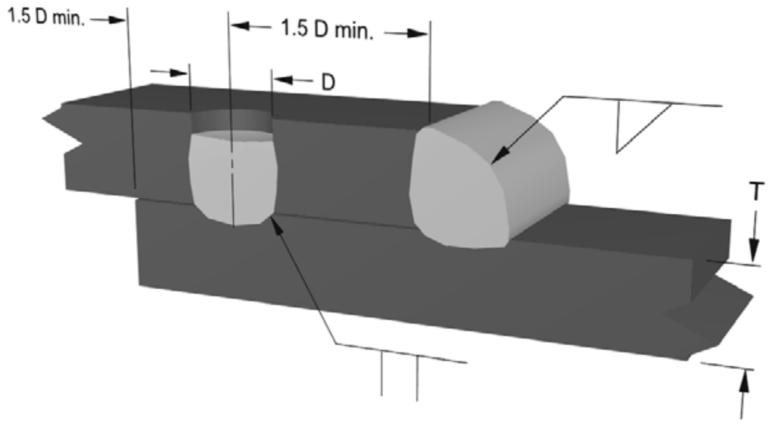


T = ⅜ inch (9.5 mm) maximum for longitudinal joints
T = ⅝ inch (16 mm) maximum for circumferential joints

Figure 4.8 Typical Type 4 weld joint.

For Type 4 joints used to make circumferential welds on thicknesses over ½ inch (13 mm), the Code user may wish to take extra precautions and follow the nondestructive examination requirements for corner joints given in UG–93(d)(3).

Type 5 joints are also lap joints, but are restricted to category B and C joint locations. They use a plug weld instead of a second fillet weld to secure one end of the lap. The plug weld does not make the joint stiff, and this permits some stressing of the fillet weld root area. Therefore, Type 5 joints are assessed a maximum efficiency of 0.5. The maximum diameter for attaching a head to a shell using a Type 5 joint is restricted to 24 inches (610 mm) (category B configuration). A Type 5 joint is not to be used for attaching hemispherical heads to shells. Hemispherical heads are made by weld construction and therefore have intersecting head, and shell-to-head, seam welds. This construction would cause additional stress magnification at the Type 5 shell-to-head joint. Minimum size requirements for a plug weld are given in UW–17 and discussed later in this chapter.



T = ½ inch (13mm) maximum for category B joint configurations
T = ⅝ inch (16mm) maximum for category C joint configurations

Figure 4.9 Typical Type 5 weld joint.

A Type 6 weld joint is a lap joint with a single fillet weld. It is applicable to either category A or B configurations. Because there is no resistance to bending at the weld root, the maximum efficiency is 0.45. When a single fillet lap joint is used to attach a head that is pressurized only on the convex side of the head, the maximum shell thickness shall be ⅝ inch (16 mm) or less and the fillet weld shall be inside the shell as shown in Figure 4.10. Alternatively, if the shell diameter is 24 inches (610 mm) or less and the shell thickness is ¼ inch (6.4 mm) or less, then the weld can be on the flange of the head as shown in Figure 4.11.

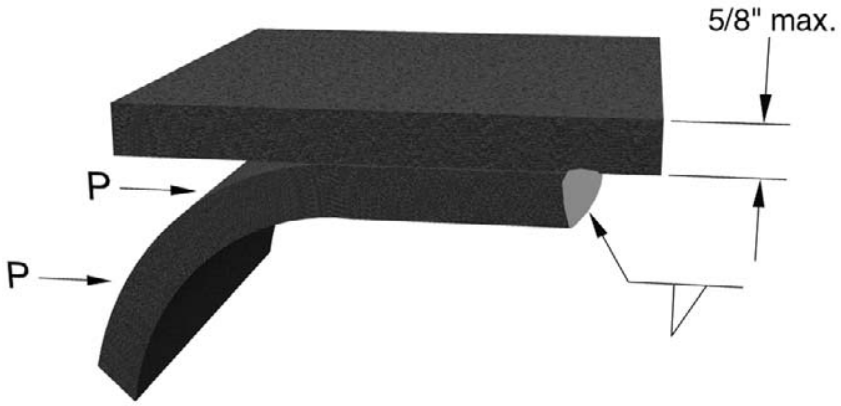


Figure 4.10 Typical Type 6 weld joint for shell-to-head.

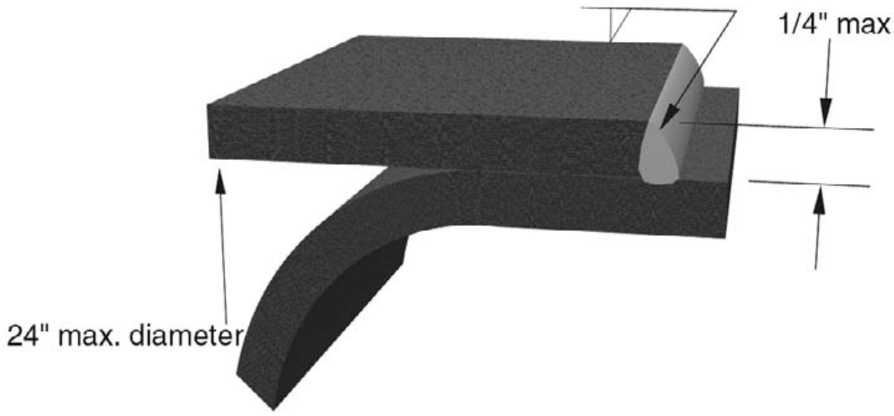


Figure 4.11 Alternative shell-to-head Type 6 weld joint.

P-Numbers

The Code places materials considered weldable into groups with similar weldability. These groups are assigned a parent material grouping number called the P-Number. Materials approved for welded construction in Division 1 will have a P-Number assigned to them in Section II of the Code. The P-Number assignment can also be found in Section IX. In both Sections II and IX, there are some materials listed that are not approved for Section VIII construction as these sections of the Code are also used for pressure piping, tanks, and other types of structures. These unapproved Section VIII materials will also have a P-Number assignment, or in some instances where there is a special interest use, an S-Number is assigned. These materials must not be used for pressure vessel construction as they are either unproven or have limitations that do not meet the criterion required for the construction of safe pressure vessels.

Within a P-Number assignment, the mechanical properties of some of the materials may be altered by weld heat. Subgroup numbers are assigned to identify those materials within a given P-Number where heat may alter the mechanical properties. This is a numerical assignment and usually considers the method of manufacture of the material. For example, the common carbon steels used in welded pressure vessel fabrication are assigned a P-Number 1. Within this category of materials there will be rolled, cast, and forged products. The internal energy and grain structures of these products will vary because of process history, even if the chemistry is very similar. This affects the fracture toughness of the material and heat input into the material will alter that toughness. Consequently, P-Number 1 materials are broken into 3 subgroups. This subgrouping may affect weld procedure specifications. (For further information, see chapter 4 of the *CASTI Guidebook to ASME IX* – Welding Qualifications, published by *CASTI* Publishing Inc.) For example, SA–106 Grade B is a common pipe material used in pressure vessel fabrication. This is categorized by the Code Committee as a P-Number 1 Group 1 material. This pipe may have a weld neck flange welded to it such as a SA–105 flange. This flange is categorized as a P-Number 1 Group 2 material. Section IX requires the weld procedure for joining the pipe to the flange be developed for joining P-Number 1 Group 1 material to P-Number 1 Group 2 material.

Weld Procedure and Welder Qualifications

All welding done on the pressure containment envelope must be done to a qualified procedure and by qualified personnel. UW–26 makes it very clear that this is the responsibility of the manufacturer and, furthermore, no welding is to be done until the qualifications are in place. UW–28 and UW–29 require that qualifications be done in detail in accordance with the provisions of Section IX of the BPVC.

Authorized manufacturers can subcontract to other welders or manufacturers who may or may not have certificates of authorization provided all welding is done in accordance with the weld procedures of the authorized manufacturer, and only qualified welders are used as stated in UW-26(d) [Interpretations VIII–1–86–192 and VIII–1–89–79]. In addition, the manufacturer's approved quality control system must provide for the supervision of weld construction even if that supervision is provided by a subcontractor [Interpretation VIII–1–89–247]. The approved quality control program must indicate the manufacturer's responsibility for all welding work.

UW–28 requires all welding done on the pressure loaded parts be done to a qualified weld procedure. This means that groove welds, fillet welds, plug welds, spot welds, stud welds, and even tack welds require a qualified weld procedure. An often overlooked requirement of the Division is UW–31(c), where it is specified that welders making tack welds are to be qualified. Welding heat causes metallurgical changes in the base material. These changes are stress concentrators. The weld heat input also results in residual stress at the weld location. Since welding is a manual skill, only those individuals who have been tested as having proven skill should weld on a structure whose integrity could be compromised by welding.

UW–29 requires each qualified welder and weld equipment operator to be assigned a unique identification mark by the manufacturer. This mark is used to identify the welds made by the welder. UW–37 gives the requirements for identifying the welds. Generally, the mark is to be die-stamped into the vessel adjacent to the applicable weld at 3 foot (900 mm) intervals. This stamp is a stress concentrator and must be applied with care. On thin materials and ductile or soft materials, severe deformation may result from the stamping process. UW–37 contains a number of restrictions in which alternative means of welding identification must be used. Paper records are the most common alternative.

Weld Fabrication Quality Requirements

Throughout various paragraphs of the Code, workmanship requirements are specified (see Table 4.2). Many of these are the common requirements given in welding fabrication standards and codes. Suggested preheats for various P-numbers are given in Table 4.3.

The use or absence of preheating before welding is an essential variable requirement of Section IX. Section VIII, Division 1 provides guidance for the preheat temperature. This information is given in Nonmandatory Appendix R.

The method of preheating is not restricted and therefore can be achieved by direct heating, such as flame heating shown in Figure 4.12, or indirect heating such as heating the material in a box furnace. The base metal temperature must be controlled within the requirements of the qualified weld procedure.

Figure 4.12 A torch is used to preheat this thick walled material prior to making the shell-to-head weld.

Appendix R also contains precautionary information on interpass temperatures. The quenched and tempered materials in P-Number 10C Group 3, and P-Number 11, all groups, may experience deterioration of strength and toughness at elevated temperatures. In such cases, a maximum interpass temperature should also be adhered to. This is particularly important in thinner materials. The maximum interpass temperature is not suggested. Generally, the Code user would be advised to weld as close to the preheat temperature as possible and to avoid temperatures in excess of 600°F (315°C). For the quenched and tempered P-Number 10D Group 4 and P-Number 10E Group 5 materials, a maximum interpass temperature of 450°F (230°C) is suggested in the appendix.

Table 4.2 Welding Workmanship Requirements Specified in Division 1

Item	Division Article	Comments
ambient temperature	UW–30	Recommendation only - no welding at ambient temperatures less than 0°F, preheat below 32°F to above 60°F before starting. (This should be followed for welding materials in which mechanical properties are altered by weld cooling rate.)
ice, rain, or snow	UW–30	Recommendation only - no welding when surfaces are wet. (This produces accelerated cooling rates and copious amounts of hydrogen. Hydrogen embrittles some metals.)
wind	UW–30	Recommendation only - no welding in high wind. (Weld pools are shielded from oxidation by the gases released on welding. This gas blanket protection can be blown away. Accelerated cooling rates also occur.)
joint cleanliness	UG–76, UW–31, UW–32	Remove nonmetallic contaminants before welding. (These can be trapped in the weld and may also contain hydrogen generating contaminants.)
joint fit-up and alignment	UW–31, UW–33	Section IX has joint alignment as an unlisted variable. For Division 1 welds, alignment tolerance must be in accordance with Table UW-33.
joint penetration	UW–35, UW–36, UW–37	Section IX has joint penetration as an unlisted variable. For Division 1 welds, joint penetration and fusion are required.
weld fill	UW–35	Joint fill is an unlisted requirement in Section IX. For Division 1 welds there are minimum requirements for joint throat and maximum requirements for weld reinforcement. (The weld must develop the strength of the base material. Weld reinforcement is a change in surface contour and therefore is a stress concentrator: the more severe the build-up, the greater the stress magnification. Division 1 does not require weld reinforcement--only complete joint fill and controlled maximum reinforcement.)
undercut	UW–35, UW–36	Undercut (reduction in thickness) is a stress concentrator and must be controlled.
peening	UW-38	Peening is an essential variable in Section IX. Division 1 recognizes peening as a weld fabrication technique.
post weld heat treating	UW–40, UW–49, UCS–56, UNF–32, UHT–56, UHA–32	Post weld heat treating is an essential variable in Section IX. Procedures developed for pressure piping codes may not meet the post weld heat treatment requirements of Division 1 even though they were developed for a P-Number material applicable to Division 1.

Table 4.3 Suggested Weld Preheats

P-Number	Thickness (in.)	Suggested Preheat Temperature (°F)
P-No. 1 Groups 1, 2, 3 (carbon greater than 0.30%)	greater than 1	175
P-No. 1, materials not described above		50
P-No. 3 Groups 1, 2, 3 (specified tensile strength greater than 70,000 psi)	all	175
P-No. 3 Groups 1, 2, 3	greater than ⅝	175
P-No. 3, materials not described above		50
P-No. 4 Groups 1, 2 (specified tensile strength greater than 60,000 psi)		250
P-No. 4 Groups 1, 2	greater than ½	250
P-No. 4, materials not described above		50
P-No. 5A, P-No. 5B Group 1 (specified tensile strength greater than 60,000 psi)		400
P-No. 5A, P-No. 5B Group 1 (chromium greater than 6%)	greater than ½	400
P-No. 5A, P-No. 5B, materials not described above		300
P-No. 6		400
P-No. 7		none
P-No. 8		none
P-No. 9A		250
P-No. 9B		300
P-No. 10A		175
P-No. 10B		250
P-No. 10C		175
P-No. 10D, P-No. 10E		300
P-No. 11A Group 1		none
P-No. 11A Groups 2, 3		400
P-No. 11A Group 4		250
P-No. 11B Groups 1, 2, 3, 4, 5		175
P-No. 11B Groups 6, 7		400

Radiography of Welds

UW–11 specifies radiographic requirements for welds mandated to be radiographed beyond the considerations given for weld joint efficiency. These requirements are for weld quality, and they consider the fracture toughness of the weld. The more sensitive a weld filler metal or weld heat affected zone is to brittle fracture, the more significant is the requirement for the weld to have fewer imperfections. The purpose of radiography is to identify imperfections so that those that may reduce the integrity of the weld can be removed, as shown in Figure 4.13. As the thickness of most materials increases, so does the propensity for brittle fracture. All butt welds of vessels in lethal service or in steam boilers exceeding 50 psi (345 kPa) are to be radiographically examined along their entire length. Similarly, all butt welds over 1½ inch (38 mm) are to be radiographically examined except those welds in the materials listed in Table 4.4 which are to be examined when the thickness exceeds that given in the table.

Figure 4.13 A grinder is used to remove a weld imperfection.

Table 4.4 Radiographic Requirements for Welds

P-Number	Group	Thickness Above Which Welds are to be Radiographed (in.)	Comments
1	1,2,3	1¼	
1	4	0	All Type 1 welds and all nozzle attachment welds except set-on and set-through nozzles of 2 inch inside diameter and smaller. Category B and C welds in nozzles and communication chambers of NPS 10 and smaller and 1⅛ inch wall and smaller are exempt.
3	1,2	¾	
3	3	¾	All Type 1 welds and all nozzle attachment welds except set-on and set- through nozzles of 2 inch inside diameter and smaller for SA–533 materials shall be examined in all thicknesses. Category B and C welds in nozzles and communication chambers of NPS 10 and smaller and 1⅛ inch wall and smaller are exempt.
4	1,2	⅝	
5A	1	0	
5B	1	0	
6	1	0	Alloy 410 only.
6	2	0	Alloy 429 only.
6	4	0	All Type 1 welds and all nozzle attachment welds except set-on and set-through nozzles of 2 inch inside diameter and smaller.
7	1	0	Applicable to alloy 405 welded with A numbers 6 and 7 electrodes only.
7	2	0	Alloy 430 only.
9A	1	⅝	
9B	1	⅝	
10A	1	¾	
10B	2	⅝	
10C	1	⅝	
10F	6	¾	

Table 4.4 Radiographic Requirements for Welds (Continued)

P-Number	Group	Thickness Above Which Welds are to be Radiographed (in.)	Comments
11A	1,2,3,4,5	0	All Type 1 welds and all nozzle attachment welds except set-on and set-through nozzles of 2 inch inside diameter and smaller. Category B and C welds in nozzles and communication chambers of NPS 10 and smaller and 1⅛ inch wall and smaller are exempt.
11B	1,2,3,4,6,8,10	0	All Type 1 welds and all nozzle attachment welds except set-on and set-through nozzles of 2 inch inside diameter and smaller. Category B and C welds in nozzle and communication chambers of NPS 10 and smaller and 1⅛ inch wall and smaller are exempt.
42		⅜	Alloys 400 and 401 are exempt.
43		⅜	Alloy 600 is exempt.
44		⅜	
45		⅜	
46		⅜	
47		⅜	
51		0	Category A and B welds.
52		0	Category A and B welds.
53		0	Category A and B welds.
61		0	Category A and B welds.
62		0	Category A and B welds.

The radiographic requirements for low temperature vessels and layered vessels are unique and are discussed in Chapter 6.

The weld joint efficiency affects the minimum required wall thickness. This is discussed in Chapter 7. Radiography may be cheaper than thicker material, so radiography may be specified even when it is not required by the Division. When radiography is specified only for joint efficiency enhancement, UW–11 requires that Category B and C butt welds that intersect a Category A weld shall at least be spot radiographed in accordance with UW–52. Such examination will only qualify the joint efficiency examination of the Category A weld examination. If spot radiography is specified for all the welds of a vessel for purposes of joint efficiency, Category B and C welds of nozzles and communication chambers less than NPS 10 (DN 250) and 1⅛ inch (29 mm) thickness do not require radiographic examination. This exemption is permitted since spot radiography is a quality control measure only and does not provide assurance of integrity throughout the vessel. Spot radiography is based on 50 foot (15 m) weld length intervals and only 1% of the weld length is radiographically examined. Smaller diameter and thinner wall appurtenances usually do not represent a significant failure risk when considered with the other Division mandated quality controls (see UW–52 and UG–97(c)) and are therefore exempt.

Table UW–12 identifies two forms of radiographic examination: full radiography and spot radiography. UW–51 specifies that radiographic examination shall be done in accordance with the requirements of Section V of the Code; specifically with Article 2 of this Section. All radiography shall be done by personnel who have been trained, examined, and qualified to the written practice of their employer. This written practice shall follow either the requirements given in the 1998 Addenda of the 1996 edition of Specification SNT–TC–1A of the American Society for Nondestructive Testing (ASNT), or ASNT Central Certification program (ACCP, 1997), or ASNT CP-189, 1995. As an alternative, nondeastructive testing personnel may be tested and certified by the American Society for Nondestructive Testing through their Central Certification Program (ACCP).

UW–51 gives the rejection criteria for welds examined along their full length (full radiography). An examination less than the full length of the weld is a spot radiographic examination. UW–52 gives the minimum requirements for spot radiography. The minimum length of weld that must be examined is 6 inches (150 mm) in each 50 feet (15 m) of weld made by each welder. The rejection criteria for spot radiography is less stringent than for full radiography, and these criteria are also given in UW–52. The location to be examined is to be selected by the Authorized Inspector.

Special Requirements for Welded Fabrications

Lethal Service

UW–2 and its footnote definition contain important information on special requirements for vessels containing lethal substances. As previously indicated (Chapter 2, U–2), it is the user of the pressure vessel who is responsible for designating a vessel for lethal service. When additional fabrication requirements and restrictions apply, increased cost of vessel construction will be realized. Users are cautioned not to let this cloud their judgment on specifying a particular substance as lethal. Ignoring or avoiding this requirement has both moral and, in many areas, legal ramifications. If the substances in the vessels are of such a nature that "a very small amount mixed or unmixed with air is dangerous to life when inhaled," they are defined in Section VIII, Division 1 as being lethal. Unfortunately, the definition leaves open the interpretation of "a very small amount."

What is lethal service? From the surface, it might appear that a vessel is in lethal service. Note that the Code does not designate which vessels are in lethal service. It leaves the decision to the user. Should a vessel processing H_2 be designated a lethal service vessel? What about H_2S, HF, or NH_4? A brief history can help answer these questions.

Since 1931 the ASME Code for pressure vessels has included special construction requirements for pressure vessels containing lethal gases or liquids. A companion code, the API–ASME Pressure Vessel Code, published from 1934 to 1951 and widely used by the petroleum industry through about 1956, did not include any special requirements for vessels containing lethal substances. A comparison of the two codes clearly shows that the API–ASME Code provided a higher quality welded vessel by requiring higher weld quality control, material limitations, radiography, and heat treatment.

The API–ASME Code was withdrawn at the end of 1956 and from that point on most pressure vessels (unfired) for the petroleum industry in the United States and Canada were constructed to the requirements of the ASME Section VIII Pressure Vessel Code. Initially the ASME Code recognized that this additional requirement was not needed in the petroleum industry and included a footnote to that effect.

However, in 1972 the Code was revised to exclude any definition or guidance on lethal versus nonlethal substances for pressure vessel construction and made such determination the responsibility of the user and/or the user's designated agent.

The footnote that existed in the Code in 1972 follows. The underlined part was deleted that year.

"By 'lethal substances' are meant poisonous gases or liquids of such a nature that a very small amount of the gas or of the vapor of the liquid mixed or unmixed with air is dangerous to life when inhaled. For purposes of this Division, this class includes substances of this nature which are stored under pressure or may generate a pressure if stored in a closed vessel. <u>Some such substances are hydrocyanic acid, carbonyl chloride, cyanogen, mustard gas, and xylyl bromide. For design purposes under this Division, chlorine, ammonia, natural or manufactured gas, any liquefied petroleum gas such as propane, butane, butadiene, and vapors of any other petroleum products are not classified as lethal substances."</u>

Because of the history outlined above, many industries normally do not designate a vessel as being in lethal service, but specify special requirements for some services. Those special requirements are:

- 100% radiography,
- post weld heat treatment,
- impact testing, and
- hardness limits on base material and weldments.

When the substance in the vessel is designated as lethal, all butt welds used to manufacture the vessel shall be fully examined using radiographic techniques. This means that lethal service vessels made using electric resistance welded or pressure forge welded pipe or tube also require radiographic examination for the weld seam length, even if the manufacturing specification for the pipe or tube required ultrasonic testing. [This is indicated in the revised interpretation VIII–1–83–77.] There are two exceptions to this requirement. UW–2(a)2 excludes the radiographic requirement for the welded seam in tubes or pipes fully enclosed within a vessel in which all butt welds are radiographically examined for the lethal service requirement. UW–2(a)3 excludes lethal service radiographic requirements for the portion of heat exchangers that are not in lethal service, provided the exchanger tubes are made from seamless pipe or tubing. If the pipes or tubes are butt welded, they must be welded using a procedure that does not employ a filler metal. In this instance, the tubes shall be hydrostatically tested as required by the Division and, in addition, pneumatically tested. The weld seam shall also be nondestructively tested using either ultrasonic or eddy current techniques.

In addition to the radiographic requirement, welded fabrications of carbon or low alloy steel intended for lethal service shall be post weld heat treated.

Weld types for lethal service shall be restricted as follows:

- Category A – Type 1 weld only
- Category B – Type 1 or Type 2 welds only
- Category C – Type 1 or Type 2 welds only
- Category D – Type 1 or Type 2 welds only

Cold Temperature Vessels

UW–2 gives joint type restrictions for cold service vessels. Cold service definitions are given as follows:

- UCS–68: steel listed in UCS–23 at a design temperature less than -55°F (-48°C) and stressed in excess of 35% of maximum allowable joint stress (SE value)
- UHA–51: high alloy steels listed in UHA–23 at a design temperature lower than listed in UHA–51

The weld types for cold service are restricted to the following:

- Category A – Type 1 welds only, except austenitic stainless steel materials which may use Type 2 welds when the design temperature is above -320°F (-195°C) and the weld procedure meets the requirements given in UHA-51(f)
- Category B – Type 1 or Type 2 welds only
- Category C – Type 1, Type 2, or Type 3 welds only
- Category D – Type 1 or Type 2 welds only

These weld types result in full penetration welds. In cold temperature service, steel materials are more susceptible to brittle fracture because of a reduction in fracture toughness of steels with reduced temperature. Weld imperfections such as incomplete weld penetration may simulate a crack. The tolerable size of a crack before it may propagate catastrophically is proportional to the square of the fracture toughness of the material.

In general terms, this relationship can be written as follows:

$$a = \frac{Q \times K^2}{\sigma^2}$$

where: a is the maximum tolerable depth of incomplete penetration,
Q is a shape and geometric factor,
K is the fracture toughness of the weld metal, and
σ is the stress transverse to the incomplete penetration.

Unfired Steam Boilers

Unfired steam boilers with design pressures exceeding 50 psi (343 kPa) must meet the requirements of UW–2(c). The weld types are restricted as follows:

- Category A – Type 1 joints only
- Category B – Type 1 or Type 2 joints only

In addition, all vessels constructed of carbon or low alloy steel (materials listed in UCS–23) shall be post weld heat treated. All butt welds shall also be radiographically examined except category B welds on nozzles and communication chambers of NPS 10 (DN 250) and smaller and 1⅛ inch (29 mm) thickness and smaller when constructed from ferritic steels whose properties are enhanced by heat treatment (materials listed in UHT–23). These requirements are again based on fracture toughness considerations. In this instance, the steam is an expansive vapor. When released, the pressure (stress) does not completely and rapidly dissipate. Even though the toughness may be reasonably high because of the warm temperature of the steam, a crack may grow to catastrophic size under the influence of the sustaining stress. Post weld heat treatment lowers the residual stress at the welds. Radiography reduces the possibility of flaws that may act as cracks. Ferritic steels whose mechanical properties are enhanced by the heat treat procedures of austenitizing followed by quenching from the austenitizing temperature and a tempering heat treatment (materials listed in UHT–23) usually have enhanced toughness properties. The radiographic requirements on these materials are slightly relaxed without compromising safety.

Direct Fired Vessels

UW–2(d) presents the special requirements for vessels heated by applying heat from combustion directly to the vessel. The weld types are restricted as follows:

- Category A – Type 1 only
- Category B – Type 1 or Type 2 joints for thicknesses greater than ⅝ inch (16 mm); Type 4 and Type 5 joints may be used for thicknesses ⅝ inch (16 mm) and less

Post weld heat treatment is required for P-Number 1 materials over ⅝ inch (16 mm) thick and for all thicknesses of P-Numbers 3, 4, 5, 9, and 10. As with the other special weld construction requirements, reduction of fracture risk is the primary concern in the special requirements. In thinner sections, and because of the elevated temperature from firing, the P–Number 1 materials should have reasonable fracture toughness. The low alloy steels, however, are susceptible to developing brittle microstructures on heating above 1300°F (700°C). This temperature is readily exceeded during welding. This brittleness may be reduced or removed by heat treating the steel below 1300°F (700°C).

Fabrication by Forging

Part UF (Unfired Forged) contains the rules for construction by forging. These rules are used in tandem with the general requirements, Subsection A, and the material requirements, Subsection C, of the Division. In addition to those vessels completely fabricated by forging, this section also applies to vessel components fabricated by forging, except for those components traditionally classified as fittings. Many of the vessels made entirely by forging are exempt from Section VIII because of the small size of the vessel (see Chapter 2 discussion on UM vessels). Such vessels, however, may be fabricated to Division requirements when specified by other codes and standards, legislation, or contract requirements.

Materials

UF–12 indicates that vessels or components may be made by forging ingots, slabs, billets, plate, pipe, or tubes provided the material is listed in Section II of the BPVC (UF–6). Further restrictions on materials are given in UF–5 and UF–6. Forgings to be strength welded are to be restricted to a maximum carbon content of 0.35%. Materials with a carbon content up to 0.50% can be seal welded. These materials can also be welded if the weld is an attachment weld of a minor nonpressure retaining part or a repair weld to a forging. UF–32 gives specific limitations for the attachment weld of the nonpressure retaining part. This is to be a fillet weld with a ¼ inch maximum throat. There also are special requirements for the qualification of the weld procedure and the welder. These carbon content restrictions result from the hardening and loss of ductility caused by welding carbon containing materials, particularly steels. Forging results in high energy being retained in the forged part and, when this is combined with a loss of ductility, the forged part is more susceptible to failure by brittle fracture. UF–6 restricts forging of pipe or tube to seamless products. This is not only a consideration of the loss of ductility in welded pipe and tube, but also a lower joint efficiency (reliability) inherent in welded products. UF–6 also restricts further forging of what were originally considered to be final forged products to the mild carbon steel based materials listed in UCS–23 and UHA–23. Again, this is a consideration of the loss of ductility and resulting increase in the potential for brittle fracture of some of the other material categories.

UF–5 and UF–12 both contain special requirements for some of the materials listed in the material specification SA–372. Ductility will be restored by following these requirements. In addition, it is necessary to inspect for flaws caused by a loss of ductility and to use construction practices that do not produce significant stress concentration. The details for the nondestructive testing requirements for SA–372 Type VIII material are given in UF–55, while the nondestructive testing requirements for SA–372 liquid quench and temper materials are given in UF–31. UF–7 specifies that steam heated steel rolls used to make corrugated cardboard be fabricated in accordance with material specification SA–649. Both specification SA–372 and SA–649 are given in Section II Part A of the Code.

Quality Requirements for Forged Vessels and Forged Components

Forged vessels and forged components must meet the dimensional requirements given in Subsection A. The surface roughness of a forged product is frequently irregular. This roughness will give variations in thickness. UF–30 permits variations in thickness that result in localized areas that are less than those required by the design, provided there is sufficient reinforcement in the surrounding material. The surrounding material is defined in UG–40, and calculations are required to verify that the surplus material in the surrounding area meets or exceeds the material deficit at the thin area (see Chapter 9). If approved by the Authorized Inspector, thin areas can be repaired by weld build-up as indicated in UF–37(b). The irregular surface may also result in imperfections. UF–45 requires that the manufacturer examine the forged surface for defects and measure the thickness for thin areas.

The mechanical energy used in a forging operation may alter the mechanical properties of the material being forged. Materials used in pressure vessel construction are required to conform to a specification that gives the manufacturing history. This ensures that the mechanical properties are predictable and consistent. UF–31 specifies that after forging, each vessel or vessel component be heat treated in accordance with the specification for the forged material. When weld fabrication of forged components is carried out, post weld heat treatment in accordance with UW–40 is required. UF–37 specifies requirements for post weld heat treatment when the welding was done to repair a forge defect or to build up a thin area. All carbon steel materials listed in UCS–23 shall be post weld heat treated if such heat treatment is specified for the material in UCS–56 regardless of any thickness limitations given in UCS–56, fillet welds excluded (see Chapter 6). Nondestructive testing requirements for repair welds are also given in UF–37. For materials with a carbon content greater than 0.35%, the extent of examination is greater and more rigorous than for those materials with 0.35% carbon and less. The extent of examination is also a function of the depth and area of repair. Higher carbon content materials are more susceptible to weld related cracks. The more extensive the weld repair, the greater will be the residual stress caused by the weld heat input. Stress can result in cracks. The emphasis in nondestructive testing, therefore, is to examine for cracks.

Fabrication by Brazing

Part UB (Unfired Brazed) contains the rules for construction by brazed assembly. These rules are used in tandem with those of the general requirements, Subsection A, and the material requirements in Subsection C of the Division. UB–1 defines brazing as a metal joining process where joining is achieved using a nonferrous filler metal with a melting temperature greater than 840°F (449°C) but less than the metals being joined. The spacing between the materials being joined must be small enough to ensure that capillary forces develop sufficient joint strength. Soldering is a joining process that uses filler metals with melting temperatures below 840°F (449°C). Soldering is not permitted for pressure vessel construction.

Approved Brazing Processes

UB–16 permits any joint design to be used provided the joint strength equals or exceeds the specified minimum strength of the parent materials when tested in accordance with Section IX of the BPVC. This is reiterated in UB–10 where, in addition, the designer is responsible for ensuring that the brazing procedure selected will develop the required joint strength. The approved methods of heating to melt the filler metal are given in UB–1 as follows:

- torch heating,
- furnace heating,
- induction heating,
- electrical resistance heating,
- salt bath heating, and
- flux bath heating.

As the joint strength is developed by capillary forces, the filler metal must wet the surfaces of the metals being joined. Joint cleanliness is therefore important (UB–34). As well, a fluxing agent may have to be used before or during brazing to clean the joint or to improve the surface wetability or both (UB–7). The width and depth of the joint will not only affect how readily the filler metal actually fills the space, but will also affect the joint strength (UB–35). Gravity also plays a part in determining the ease of joint fill. UB–18, UB–30, UB–31, UB–32, and UB–42 specify that braze procedures and brazing operators be qualified in accordance with the requirements of Section IX. UB–6 restricts the filler metals that can be used for pressure vessel construction to those listed in SFA-5.8 of Section II Part C of the BPVC. [This position has been restated in Interpretation VIII–1–86-228.] These filler metals are listed in Table UB–2.

Braze Joint Efficiency

Just as for welding, the reliability of the braze joint is assigned an efficiency value. In brazing, this efficiency or E value is determined by the extent to which it can be confirmed that the filler metal has filled the joint. UB–14 permits a maximum E of 1.0 when visual examination can be done to ensure the filler metal has filled the joint. If the joint cannot be completely examined visually, then a maximum E of 0.5 shall be used.

Brazing is not a common means for pressure vessel assembly. One reason for this is the restricted design service temperature range that can be specified. The relatively low melting temperature of the approved filler metals results in braze joints having a poor creep performance at lower elevated temperatures. The maximum permitted temperatures are presented in Column 1 of Table UB–2. Impact testing for cold service vessels is required (UB–22).

Corrosion consideration is also a factor that limits brazing use. The filler metals frequently have a different chemical composition than the parent materials. UB–13 requires that the vessel designer consider the possibility of galvanic corrosion caused by the dissimilar metals. Many of the flux materials function by forming an intermediate compound between the filler metal and the base material. This is done by a chemical reaction which, under other circumstances, may be considered to be corrosion. UB–36 requires joint cleaning after brazing, not only to remove the corroding flux, but also to clean for visual examination.

Quality Requirements for Brazed Vessels and Vessel Components

UB–37 recognizes that brazed joints may be repaired if they are found to be defective. UB–44 presents the acceptance criteria for a brazed joint. Repairs are not permitted for a crack in the base metal adjacent to the joint. UB–44 requires a visual examination of all brazed joints to be done by the Authorized Inspector.

Chapter 5

SPECIAL FABRICATION TECHNIQUES

The Mandatory Appendices 17, 18, 19, 20, 22, 27, 28, and 29 contain articles that provide instruction on special constructions. For the most part, these Appendices are short and reiterate the requirements of Subsections A for design, and C for materials, and present the fabrication requirements for special vessel or component configurations.

Plate Heat Exchangers

Plate heat exchangers are made by stacking thin dimpled or embossed plates together. Appendix 17 presents the special requirements for this type of construction. Paragraph 17–1 identifies the rules of the Appendix as being applicable to construction conducted by welding through one or more members to secure it (them) to another member. Such joining is done by either a spot weld process or an electric resistance seam weld. To achieve an annular space between plates for fluid or gas flow, one or more of the joined plates will be dimpled or embossed. (Embossing and dimpling are usually achieved by stamping plate material protuberances.) Figures 17–1 through 17–6 illustrate the typical designs for this type of construction.

The materials approved for plate heat exchanger construction are listed in Table 17–3, while Tables 17–4.1 and 17–4.2 give the thickness range for these materials. The thickness range is determined by the weld process selected to join the materials. Weld processes shall be in accordance with the requirements of Appendix 17. Paragraph 17–6 presents special essential variables for the welded construction. These variables affect the structural integrity of the exchanger because a change in the listed variables of weld spacing, material type, material thickness, or electrode size will affect the reliability of the weld joint. Paragraph 17–7 requires the weld procedure and welder qualification to be done by proof testing a fabricated assembly and a test coupon(s) made at the same time as the test panel. This coupon is to be subjected to mechanical and metallographic examinations in accordance with 17–7 and Fig. 17.7 through 17.15. Special examinations are also required of the test panel after it has been tested to failure. Paragraph 17–7 also specifies essential variables for the weld process. Section IX weld procedures and welder qualifications are not applicable to the spot and seam welds for plate exchanger construction. Owners should be aware of this special requirement when contemplating repair or modification of plate exchangers.

Weld quality verification testing is required during construction. The specific tests to be carried out and the test frequency are given in 17–8.

Data Plates

UG–119 requires nameplates (data plates) to be attached to the vessel. Normally this would be done by welding the plate to the vessel. When the plate is to be attached by adhesive gluing, Appendix 18 is applicable. Paragraph 18–2 requires that adhesive attachment be carried out according to a written procedure that is part of the manufacturer's quality control system. The specific requirements for the glued nameplate are given in the Appendix.

Jacketed Steam Kettles

The requirements for direct heated, jacketed steam kettles are given in Appendix 19. The steam pressure of the jacket is to be 50 psi (345 kPa) maximum as indicated in 19–2, and no steam or water is to be drawn from the jacket. The control requirements for the steam jacket are listed in 19–7 and include a pressure gauge, a sight glass, a vent, and low level controls.

Machined Plate

Plate is occasionally used to manufacture parts by machining the plate to achieve the desired shape. Tubesheet hubs, as pictured in Figure 5.1, and flat heads are two common structures machined from plate. The applicable rules for this form of component construction are given in Appendix 20. The plate is to be nondestructively examined to ensure that it contains no injurious discontinuities that could affect the structural reliability of the plate. These examinations are primarily for the detection of lamination type flaws. Paragraph 20–3 requires an ultrasonic volumetric examination of the plate, as well as either a liquid penetrant or a magnetic particle examination of the surface. At welded connections transverse to the rolling plane of the plate, either a radiographic or an ultrasonic examination is required of the area of the plate that may be susceptible to lamellar tearing. Paragraph 20–3 requires that the examination area extend at least ½ inch (13 mm) beyond the weld fusion line. The Code user should, however, examine a width that extends beyond the weld heat affected zone, as most lamellar tears occur at the edge of the heat affected zone. The risk of lamellar tearing is reduced if the material is made to a fine grain practice. Paragraph 20–2 requires tensile tests to be made on samples taken adjacent to the proposed hub to be machined from the plate. The location for sampling is shown in Fig. UW–13.3. In addition to the yield strength, tensile strength, and elongation conforming to those values of the plate specification, the test specimen must have a reduction of area value of 30% or greater. This requirement provides some assurance that the material has sufficient ductility to accommodate the shrinkage stresses from welding that cause lamellar tearing.

Integrally Forged Vessels

Appendix 22 provides for higher stress values than those in Part UF for integrally forged vessels. These vessels are unit body constructed so that the shell and heads are one piece of material. Paragraph 22–2 restricts the fabrication to SA–372 material types while 22–3 restricts the vessel to 24 inches (610 mm) inside diameter.

Figure 5.1 A machinist confirms the ligament size on a thick plate being drilled to make a tubesheet.

Enamel Lined Vessels

Enamel materials are ceramics or glasses and do not possess the ductility inherent in most metals. Appendix 27 provides alternative rules from Part UCL that are meant to accommodate the poor ductility of the lining. Enamel materials may be applied hot or may be baked on. This can result in vessel distortion. The rules of Appendix 27 are designed to take this into consideration. Out-of-round conditions outside the tolerance limits of UG–80 can be tolerated in glass lined vessels up to a maximum of 3%, provided the rated pressure is reduced to accommodate the bending stress created by the out-of-round condition. Paragraph 27–2 provides the simplified mathematical calculations required for verifying the maximum safe design pressure. Because of the heat required to apply the linings, 27–4 restricts glass lined pressure vessels to SA–106, SA–285, SA–414, SA–516, and SA–836 materials. Even so, these materials must demonstrate mechanical properties in accordance with their specifications after undergoing heat treat cycles such as those they would be exposed to during installation of the lining.

Weld procedure qualifications are to be in accordance with ASME Section IX, and 27–6 requires the weld test coupon to be heat treated to the lining application temperatures.

Heat Exchanger Box Headers

Air-cooled heat exchangers (fin fan coolers and similar items) frequently use a rectangular box configuration for the headers. UW–13 requires that box corners have a minimum weld size based on the size of the minimum thickness requirement for the shell. To achieve this by welding only from one side, both members must be beveled to create a V or U groove joint. Appendix 28 provides an alternative requirement on box headers for air-cooled heat exchangers so that the headers can be constructed by a full penetration, single bevel corner weld that is not otherwise recognized by the Division. Paragraph 28–2 requires a cross section macro examination of a sample of the joint design. Measurements made on the cross section are used to determine an effective weld throat dimension, a_2. The value of a_2 divided by the nominal thickness of the shell, less the corrosion allowance, shall not be less than 0.29 unless tensile test results are available for the through thickness of the thicker plate. In this case, a lower value is permitted, but only in accordance with a specific minimum value of the ratio of the through thickness tensile strength to the specified minimum strength of the plate. The weld procedure used to make the joint shall be in accordance with ASME Section IX with the additional essential variables listed in 28–2.

Interlocking Layered Vessels

While the use of interlocking layered construction is covered in Part ULW, which is discussed in Chapter 6, there is a layered construction technique that employs interlocking layers applied hot as a helical wrap and on subsequent air quenching gives a high strength, heat shrunk reinforced vessel. Appendix 29 presents the requirements for this construction technique. The material used for the helical wrap is a controlled chemistry, low alloy steel, the composition of which is given in Table 29–2–1. The heat treating and fabrication processes are described in 29–3, while 29–4 gives the mechanical test requirements for the resulting layer. Paragraphs 29–6 and 29–7 give the permitted stress values for the wrap material. The material is a weldable grade of steel, and a weld procedure in accordance with ASME Section IX is required (29–5).

Chapter 6

MATERIALS

The construction materials permitted for pressure containment vessels and the applicable rules for using these materials are given in Subsection C of the Division. In this Subsection rules are presented in accordance with specific material classifications, namely:

- carbon and low alloy steels – Part UCS
- nonferrous materials – Part UNF
- high alloy steels – Part UHA
- cast irons other than ductile iron – Part UCI
- clad materials, weld overlaid materials – Part UCL
- ductile iron castings – Part UCD
- ferritic steels with tensile properties enhanced by heat treatment – Part UHT
- alternative rules for low temperature materials – Part ULT
- materials for vessels built by layered construction – Part ULW

The rules for the materials of construction are to be used in conjunction with those given in Subsections A and B of the Division.

The BPVC does not specifically list applicable materials for a given chemical environment. It does, however, forbid the use of a material in an environment where it is known that the material presents an unacceptable risk. Guidance in applying the material groups is given in appendices at the end of each chapter.

The materials approved for use in Division 1 construction are given in article 23 of each of the applicable parts. The actual material specification list for carbon and low alloy steels, nonferrous materials, high alloy steels, and ferritic steels with strength enhanced by heat treatment is presented in Table 23 which is found out of sequence in Section VIII, Division 1, and at the back of Subsection C.

The ASME Code Subcommittee on Materials does not develop detailed material specifications. Those material standards developed by the American Society for Testing and Materials (ASTM) and the American Welding Society (AWS) that represent the levels of reliability deemed necessary for safe construction of pressure vessels are adopted in whole or with slight modifications. The adopted standards retain the identification of the originating society but are prefixed by the letter S. Both ASTM and AWS identify their material standards with an alphanumeric designation. No particular meaning is to be construed from these designations. The Code Committee republishes the adopted and, in some instances, modified material standards in Section II, Parts A, B, and C of the Code.

Only those materials listed in these three volumes of the Code are to be used for pressure vessel construction. The Materials Subcommittee also considers the applicable safe stress level for materials used in pressure vessel construction. The maximum stress levels developed by the Subcommittee are published in Section II, Part D of the Code. It is essential for most Section VIII, Division 1 Code users to have copies of Section II. Direct use of the material standards published by ASTM and AWS should be avoided, as some of the ASME adopted standards have been modified. [Interpretation VIII–1–92–149 clarifies the relationship between the material standards. ASTM material produced to a specification that the Subcommittee has adopted from ASTM without modification can be used without recertification of the material to the ASME standard. AWS materials (welding filler materials) can be used for pressure vessel construction provided a qualified weld procedure is developed in accordance with Section IX of the BPVC.] The use of ASTM materials is qualified in Appendix A of Section II, Part A as follows:

> "Materials for Code use should preferably be ordered, produced, and documented on this basis (material specifications in Section II); however, material produced under an ASTM Specification may be used in lieu of the corresponding ASME Specification as listed in this Appendix. Material produced to an ASME or ASTM Specification with requirements different from the requirements of the corresponding Specification may be used in accordance with the above, provided the material manufacturer or vessel manufacturer certifies with evidence acceptable to the Authorized Inspector, that the corresponding Specification requirements have been met."

Material requirements are different from product requirements. A material specification for pressure vessel use must give the method(s) of melting and the melting practice, heat treatment(s), chemical and mechanical properties, and minimum quality requirements. For example, SA–105, "Forgings, Carbon Steel, for Piping Components," is a carbon steel material used for making forged components for pressure vessel use. The material standard, SA–105, is found in Section II, Part A. The acceptable methods of melting are listed in the standard as open hearth furnace, basic oxygen process, or electric arc furnace, and the only acceptable steel making practice is killed (full scavenge of free oxygen in the melt). The material is to be supplied in the annealed or normalized, or normalized and tempered, or quenched and tempered heat treat condition (purchaser's preference). The chemical composition limits are given, as are the mechanical requirements of strength and ductility. The methods to be used to determine the chemical and mechanical requirements are stated. The final surface condition and the dimensional tolerances are also stated in the standard. The material standard does not identify the product form in which the material is to be used (flange, fitting, or valve component). The product form is given in other standards that are not developed by the Code Committee unless they are material standards for plate, pipe, or bolts. The acceptable product standards are given in U–3 and were discussed in Chapter 2. A flange made to American National Standards Institute Specification B16.5 is acceptable for pressure vessel use. One of the permitted materials in this standard is SA–105. (Since 1987 SA–105 has been identical to A–105, so an ANSI B16.5 flange would be acceptable for pressure vessel use if made from either of these materials.)

The permitted stress value for a given material is derived on the basis of the Committee policy given in Nonmandatory Appendix P. This Appendix reveals that the minimum factor of safety is based on mechanical strength parameters and, in simplistic terms, can usually be assumed to be 4. This Appendix is being revised by the Committee due to the change of the design factor to 3.5 in the 1999 Addenda of the Code. Code users may wish to employ more conservative values but are cautioned to make certain they understand what the strength parameter is that controls at a given temperature. The controlling parameters are specified minimum ultimate tensile strength, specified minimum yield strength, stress rupture life, or steady state creep rate. Appendix P provides for ductility only in those materials known to have essentially nonductile behavior. These provisions are discussed in the articles following specific material classifications.

Carbon and Low Alloy Steels

The American Iron and Steel Institute (AISI) definition for a carbon steel is an iron base alloy that contains carbon up to 2 weight percent (w/o), manganese up to 1.65 w/o, silicon up to 0.6 w/o, and copper up to 0.60 w/o. The effect of carbon on the strength of iron is dramatic and is what makes iron the most popular base metal for construction. Unfortunately, carbon detracts from the ductility of iron and reduces its elongation, its reduction of cross sectional area, and its fracture toughness. Manganese is always added to steel, since in quantities up to 2 w/o it increases the strength of the steel and improves ductility and toughness. It also has another important function in readily reacting with sulfur to form a nonmetallic compound that has a higher melting temperature than the sulfur. Sulfur and phosphorus are impurities in the original iron ore that are very difficult to remove and therefore remain as residual elements in the steel. Similarly, copper has become an impurity in the steel making feed stock (resulting from metal recycling). These elements are undesirable because they embrittle the steel and melt at a very low temperature (even when they are part of the steel alloy). Accordingly, steel manufacturing processes attempt to limit the quantity of sulfur, phosphorus, and copper to as low a concentration as can economically be achieved. The usual maximum quantities for sulfur and phosphorus are limited to less than 0.04 w/o. Silicon is usually added to steel in small quantities to improve toughness. It also improves the fluidity of molten steel making it easier to cast a billet of high quality. Silicon is normally added in a quantity up to 0.3 w/o. The standard carbon steel is thus an alloy of iron, carbon, manganese, and silicon with sulfur, phosphorus, and copper impurities. The AISI definition for a low alloy steel is the same as for a carbon steel but with the addition of aluminum and chromium up to 3.99 w/o, and cobalt, columbium, molybdenum, nickel, titanium, tungsten, vanadium, zirconium, or other alloying elements added to obtain a desired alloying effect. The Nonmandatory Appendix item UCS–150 astutely points out that the properties of the steels in this section of the Code are "influenced by the processing history (cast, forged, rolled, etc.), heat treatment, deoxidization practice (during the melt stage of the steel), and level of residual elements (impurities)."

Carbon and Low Alloy Steel Products

UCS–6 indicates that plate materials listed in Table UCS–23 can be used in an unrestricted manner for pressure retaining parts except for SA–36 and SA–283 plate materials. These particular material specifications do not require the steel to be made to give a fine grain structure (fine grain practice), nor for the steel to be deoxidized (killed). Steels that are not made to a fine grain practice are more susceptible to catastrophic fracture and consequently are judged to represent an unacceptable risk for pressure equipment in lethal service and unfired steam boilers. (Unfired boilers generate an expansive gas at a relatively cold temperature where there is a greater risk of catastrophic fracture. The Code user should also consider avoiding these materials where other expansive gases may be generated.) Material not made to a fine grain practice must be ⅝ inch (16 mm) thick or less for use in welded pressure vessel construction. This is a consideration for both toughness (thick materials have lower toughness) and for lamellar tearing (large grain material is more susceptible). Steels that are not killed can experience solid state reactions that will cause them to become embrittled. This may occur at the time of manufacture but more commonly this embrittlement will occur over time. Embrittled materials have very poor toughness.

Plate materials can be formed (forged or rolled) into shapes such as heads and shells for pressure vessel construction. Figure 6.1 shows the forming of a plate into a round shell. UCS–79 indicates that such forming shall not be done cold by blows. Such an operation can result in forging bursts that are cracks in the material. These bursts may be on the surface or internal to the plate. Cold forming can be done, but not by blows. Cold forming detracts from the available ductility in the material. This processing will make the material more prone to failure by fracture and corrosion. UCS–79 requires that carbon steel and low alloy steel materials be given a stress relief heat treatment if the cold forming strain in the outside fibers exceeds 5% when:

- the material is used in lethal service,
- other Code rules require impact testing,
- the original part thickness exceeds ⅝ inches (16 mm), or
- the reduction in part thickness resulting from the forming operation exceeds 10%.

(The heat treat requirements are given in UCS–56. The exemptions in UCS–56 do not apply.)

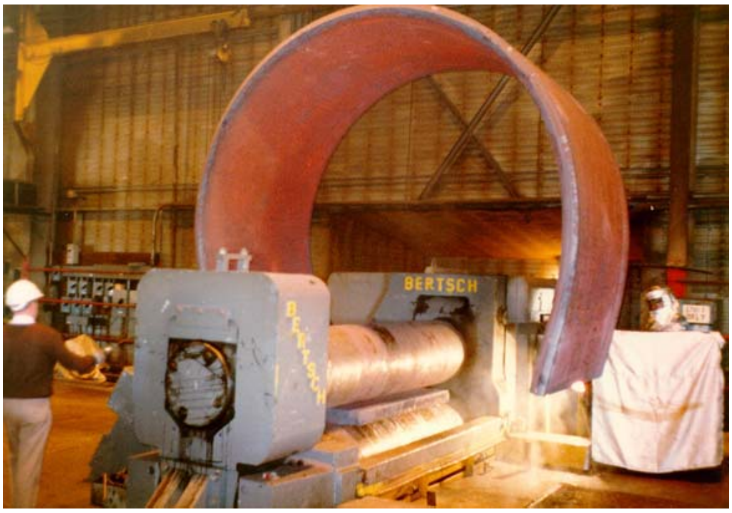


Figure 6.1 A 4½ inch (115 mm) thick plate being hot roll formed into a cylindrical section for a pressure vessel.

Diffusion of residual elements in carbon and low alloy steels can embrittle the material at temperatures as low as 250°F (120°C). Forging at temperatures between 250°F and 900°F (120 and 480°C), sometimes referred to as warm forming, can result in embrittled materials. UCS–79 requires that materials formed at this temperature be given a stress relief heat treatment in accordance with UCS–56. This heat treatment should be given immediately after the forming operation as embrittlement increases with time. Except for the special requirements stated above, carbon steel P-Number 1 classification plate materials can be formed without stress relief heat treatment, up to an outer fiber elongation of 40% (UCS–79). The formulas for determining outer fiber stress are given in UCS–79. [Interpretation VIII–1–83–81 specifies that the radius to be used in the formula for calculating the maximum fiber stress in a double curvature such as a head, is the radius of the knuckle.]

UCS–7 indicates that forged shapes can be used in pressure vessel manufacture provided they are made from a material listed in Table UCS–23. Bolts and nuts are special cases of forged products. UCS–10 only refers to use of bolts listed in Table UCS–23, while UCS–11 indicates that nuts must conform to the general manufacturing standards, SA–194 and SA–563, or to the requirements given for nuts in the bolting specifications in Table UCS–23. Nuts can either be forged or machined from forged, drawn, or rolled bar stock. Nuts must be of the ANSI B18.2.2 Heavy Series classification (heavy hex) or equivalent and must develop the strength of the bolt. When nonstandard ANSI nuts are used, proof of the adequacy of the nut must be provided. Similarly, when bolt-up configurations are used that employ bolt holes with clearances in excess of those used for a standard ANSI flange, then proof of the adequacy of the nut stiffness in bearing is required (U–2). Washers are not required in a bolted connection, but when they are used they shall be made from a listed material. The washer shall be as hard or harder than the nut to reduce galling between the nut and the washer. (When galling occurs it is more difficult to determine the loading in the bolt.) For applications up to 900°F (480°C), carbon steel bolts and nuts shall be used, but at temperatures greater than 900°F (480°C), only alloy steel materials shall be used.

Figure 6.2 Schematic illustrating (a) proper nut seating and (b) improper nut seating due to an oversize bolt hole.

Bars and shapes (hexagonal, rectangular, or square) are another special class of forged product. UCS–12 indicates that only those materials listed in Table UCS–23 may be used for pressure vessel manufacture. Machine shop practices of using AISI bar products, for example 1020, 4140, 4340, and similar specifications, are not directly applicable to pressure vessel construction as these designations are for chemical composition only. Some of the AISI products can be used, but only when they are a grade of material listed under an ASME material specification. The AISI materials that can be used for pressure vessel construction are indicated in the *CASTI Guidebook to ASME Section II* - Materials Index published by *CASTI* Publishing Inc.

Example 6.1 Manufacture of Bolts

SA–193 Grade B7 material is the common bolting material for pressure vessel use. One of the materials that can be heat treated to meet the property requirements of this Specification is AISI 4140.

Pipes and tubes are also a special class of forged product and, again, only those materials listed in Table UCS–23 can be used for pressure vessel manufacture as stated in UCS–9. Seamless and electric resistance welded pipe and tube can be used as the shell component of a pressure vessel as provided for in UCS–27 if the material is made in a basic oxygen, electric arc, or open hearth furnace.

UCS–8 provides direction for the use of steel castings. Only those materials listed in Table UCS–23 can be used for pressure vessel construction.

Welding Carbon and Low Alloy Steels

Not all carbon or low alloy steels for pressure vessel use are considered weldable. Those materials considered to be of weldable quality have been assigned a P–Number. These assignments are found in both Section II and Section IX of the Code. UCS–57 gives the radiographic requirements for the various P–Number assignments for carbon and low alloy steels. UCS–19 permits only joint Types 1 or 2 for weld categories A and B when radiography is required as these weld configurations are less likely to have nonfusion at the weld root. Radiography can have a low detection sensitivity for nonfusion.

UCS–56(f) introduces the concept of temper bead welding. In this paragraph temper bead welding is given as a means of conducting weld repairs after post weld heat treatment. Many Code users have also found the technique useful in maintenance applications. (Section VIII, Div. 1 is a construction standard and does not provide for operation and maintenance.) Temper bead welding is not applicable to new vessels designed for lethal service or service at temperatures below -55°F (-48°C). It is also not an acceptable repair procedure for surface restoration of new construction. The acceptable temper bead weld procedure follows.

- The vessel owner must approve use of the procedure.
- The procedure is restricted to: P-Number 1 Groups 1, 2, and 3, 1½ inch (38 mm) maximum thickness; P-Number 3 Groups 1, 2, and 3, ⅝ inch (16 mm) maximum thickness.
- A weld procedure qualified in accordance with Section IX is required.
- Only SMAW using low hydrogen electrodes in the conditioned state shall be used.
- Only stringer bead weld passes shall be used. (Electrode manipulation is restricted to a weave width of 4 times the electrode wire core diameter. For example, for ⅛ inch (3.2 mm), ASME SFA 5.1 Classification E7018 electrode, the maximum weave width is ½ inch (13 mm).
- Remove the defect and verify removal by nondestructive testing. (Although the Code does provide guidance on the defect removal technique, the Code user should consider that stressing of material may result from thermal removal techniques. Grinding or preheating prior to thermal removal should be considered.)

- Preheat and interpass temperature control is required: P-Number 1,200°F (93°C) minimum preheat and 200°F (93°C) minimum interpass temperature; P-Number 3,350°F (175°C) minimum preheat, 350°F (175°C) minimum interpass temperature and 450°F (230°C) maximum interpass temperature
- For P-Number 3 materials the maximum electrode size for the root pass is ⅛ inch (3.2 mm). (The Code user should consider this for P-Number 1 as well.)
- Grind root deposit to approximately half the deposited thickness.
- Maximum electrode size for all other passes is 5⁄32 inch (4 mm).
- Apply an additional pass above the final weld thickness, reinforcement included. (This is called a temper bead.) This pass must be restricted to the weld deposit only and must not touch the base material.
- Heat the area between 400 and 500°F (200 and 260°C) and maintain for 4 hours minimum.
- Remove the temper bead pass by grinding.
- After a minimum of 48 hours, nondestructively inspect the surface of the repair area for cracks. Repairs deeper than ⅜ inch (10 mm) shall also be radiographically examined. (Radiography is not considered to be an effective surface examining technique.)

The requirement for inspection 48 hours after welding is to examine for delayed cracking caused by hydrogen penetration into the steel during the welding. Avoiding delayed cracking is also the reason for using electrodes conditioned by baking. Baking reduces the moisture in the flux coating of the electrode. Moisture can increase hydrogen penetration into the steel during welding. For thinner weld deposits, the hydrogen caused crack will probably break the surface of the material at the weld fusion line or the heat affected zone on either side of the weld, but for thicker weld deposits, the crack may remain confined to the internal volume. This crack is commonly called an underbead crack. In some instances the owner may want to have the volume around the repair examined using an ultrasonic technique as well as the mandated radiography, since radiographic examination may not be sensitive to detecting underbead cracking because of the orientation of the crack.

The temper bead weld procedure may adversely affect the toughness of the base material as may any other weld procedure. Therefore, on materials that are toughness controlled, impact tests of the temper bead weld procedure will be required. Use of temper bead welding on cold service vessels is not permitted because of the potential for reduction of toughness by this technique.

UCS–56, Heat Treatment of Carbon and Low Alloy Steels

UCS–56 presents the minimum requirements for post weld heat treatment of carbon and low alloy steels. While this article of the Division is labeled post weld heat treatment, the heat treatments prescribed are used in achieving stored internal energy redistribution and reduction. This energy results not only from welding, but from other fabrication means as well, such as forging or rolling. Heat treatment, as defined by Part UCS–85 of the Division, is a controlled exposure to a temperature greater than 900°F (480°C), but less than the transformation temperature for the steel (approximately 1300°F (700°C) for carbon steel but slightly higher for alloy steels, with greater alloy additions having a higher transformation temperature). At these temperatures, the metal will be red hot as shown in Figure 6.3.

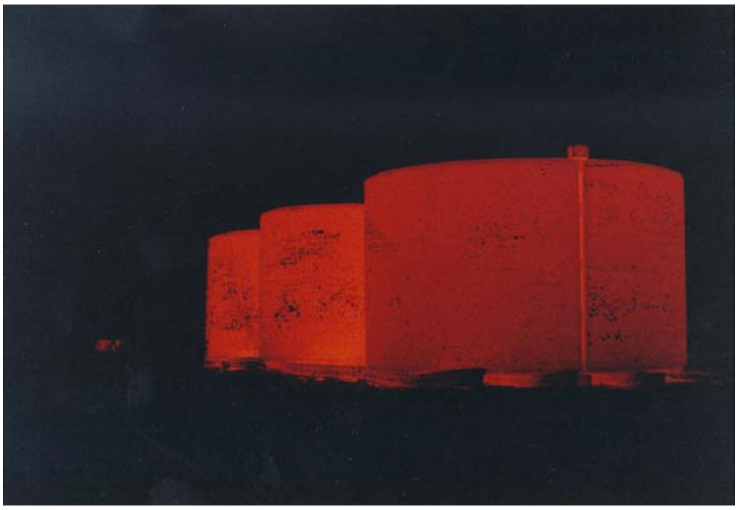


Figure 6.3 Thick wall shell sections glow red upon their removal from the heat treatment furnace.

The atoms of a material are in constant vibration about a low energy mean position. When the material is shaped by forging, rolling, or some other forming operation, or assembled by welding where thermal expansion is restrained at the weld joint by the bulk material around that joint, some of the atoms are trapped away from their low energy position. This results in energy being stored in the material, some of which can be measured as a strain or stress in the material. Such a material is said to contain residual stress. This energy makes the material more susceptible to failure, whether the mechanism of failure is fracture or corrosion. By lowering this energy, the risk of material failure can be reduced. This is the reason for post weld heat treatment, which is commonly referred to as stress relief heat treatment.

Heat input represents energy addition to the material. This causes an increase in the length of the path of the vibrating atoms and this in turn is measured as thermal expansion. The increasing vibration path lessens the atom-to-atom interaction and is measured as a decrease in the strength of the material. Some collisions or near collisions result between the vibrating atoms, particularly those pushed out of their low energy position. These collisions result in the atoms being moved around in the material until they find a new lower energy location. This motion of atoms in the solid material is called diffusion. Thus, in a post weld heat treatment, some atoms move by diffusion to lower energy positions. Other atoms are moved slightly out of their position because of the lower interaction strength between atoms. This gives a redistribution of the energy in the material to lower the energy or stress in some locations in the material. The result of diffusion and redistribution of energy is a lower total energy in the material, and in this manner the risk of material failure is reduced.

The movement of atoms is an exponential function of temperature. A slight temperature change usually affects the rate of atom movement significantly. A dwell time at the treatment temperature is required if maximum reduction of energy is to be achieved. The rate of atom movement is also a function of atom size. Large atoms from some alloy additions mean that higher temperatures are required. Alloy additions that are strong carbide formers have atom interactions that also require higher temperatures to permit weakening of the interaction. These heat treat features are addressed in Table UCS–56. The factors that affect weldability and result in ASME classifying materials in P-Number groupings also apply to heat treatment. Therefore, the heat treat dwell times and temperatures are given in Table UCS–56 in accordance with P-Number classifications.

The Code user should keep in mind that the heat treat requirements listed in Table UCS–56 are minimum requirements and while a number of exemptions are given for each P-Number listing, using the exemptions may not represent the lowest risk or the lowest cost for the pressure vessel. Exemptions are not applicable for some cold service vessels as defined in UCS–68 (service temperature less than -55°F (-48°C)), nor for electron beam welded vessels, or P-Number 3 Group 4, P-Number 5, and P-Number 10 materials welded by friction weld processes.

Example 6.2 Optional Post Weld Heat Treatment

A welded vessel designed for service as a discharge accumulator from a pump and constructed from ½ inch (13 mm) SA–516 Grade 70 material is exempt from post weld heat treatment (Table UCS–56, P-Number 1 material, exemption (2)). The pump discharge will be cyclically straining the vessel. This will cause fatigue damage to the material, and while the hoop and longitudinal stresses will be well below the fatigue limit for the material as specified by the maximum allowable stress value, S, the residual stress (energy) at the weld may be at material yield stress. The cyclic stress ripple combined with the residual stress may result in fatigue cracking of the vessel. A post weld heat treatment reduces this risk.

UCS–56 provides a set of heat treating rules that represents good practice for heat treating carbon and alloy steel products other than just pressure vessels.

- Heat treatment shall be done after all weld repairs and prior to hydrostatic testing (UW–40).
- The thickness of the weld throat, exclusive of permitted reinforcement, shall govern the time at the specified temperature (UW–40).
- The heat treated item shall not be placed in a furnace hotter than 800°F (427°C).
- During the holding period the furnace temperature shall not vary by more than 150°F (83°C) between the hottest and coldest areas in the furnace.
- The time at temperature (1 hour per inch (25 mm) of thickness up to 2 inches (50 mm), 15 minutes minimum) and the stated temperatures in Table UCS–56 are minimums. (For P-Number 1 Groups 1 and 2 materials, Table UCS–56.1 presents alternative lower temperatures with corresponding longer times at temperature.)
- The rate of heating above 800°F (427°C) shall be no more than 400°F (220°C) per inch (25 mm) of thickness, and the rate of cooling shall be no more than 500°F (278°C) per inch (25 mm) of thickness.
- The flame shall not directly impinge on the material.
- Furnace atmospheres shall be controlled to limit the amount of high temperature oxidation.

These rules allow the material to achieve a reduction in internal energy without introducing stress caused by thermal gradients. When components of a vessel are restricted from moving during heat treatment, such as might be encountered during a local heat treat procedure, or when large thermal gradients are present, then stress equal to or greater than that already in the part may be created by the process. The Code user is cautioned to consider the potential for creating this condition, particularly on treatments where the entire object is not being heated.

Toughness Requirements for Carbon and Low Alloy Steels

The ability to resist crack initiation and the catastrophic propagation of any crack or crack-like imperfections in a carbon or low alloy steel is an important consideration in the reliability of fabrications from these materials. This ability is a material property called toughness. Scientists and engineers have only begun to develop design and construction techniques based on toughness considerations, although the effects of insufficient toughness have been known for years. In the 1930s Charpy impact testing was developed to measure minimum toughness. This is the most popular and economical testing means developed so far and is the means the BPVC uses to control toughness in pressure vessel construction. This test is discussed in item UG–84.

Charpy impact testing as described in the rules given in UG–84 consist of a set of three specimens that are to be prepared and tested in accordance with SA–370 (from Section II Part A) and UG–84. The test specimen size is 10 x 10 mm cross section x 55 mm length. Specimens of less than 10 mm width are permitted when the base material is extremely tough, the material being tested is less than 10 mm thick, or the configuration of the material will not enable a 10 mm thickness with the 55 mm specified length. In such instances, Table UG–84.2 may apply and a reduction in test temperature may be required to compensate for the reduced test specimen size.

The three specimens are notched at mid span with a specified "V" configuration notch (see Figure 6.4, which is a UG–84, Charpy type A notch). This notch simulates a crack. Charpy impact specimens are then heated or chilled to the minimum design metal temperature for the vessel and hit with the hammer of a Charpy impact test machine. Such a machine is shown in Figure 6.5, while Figure 6.6 shows a Charpy specimen sitting in the anvil of the test machine. This machine will measure the energy absorbed in breaking the specimen. This energy must meet or exceed a minimum value based on the specified minimum yield strength for the material. This value is obtained from Fig. UG–84.1.

Progress on toughness design has been made through the area of study called fracture mechanics. The Committee made use of this advancement in the 1987 edition of Section VIII, Division 1 with the modification of items UCS–66 and UCS–67. Fracture mechanics can involve complex calculations and the consideration of advanced mechanical and metallurgical concepts. The Committee has been reasonably successful in reducing these considerations to permit use of the concepts without the complexities of detailed calculations.

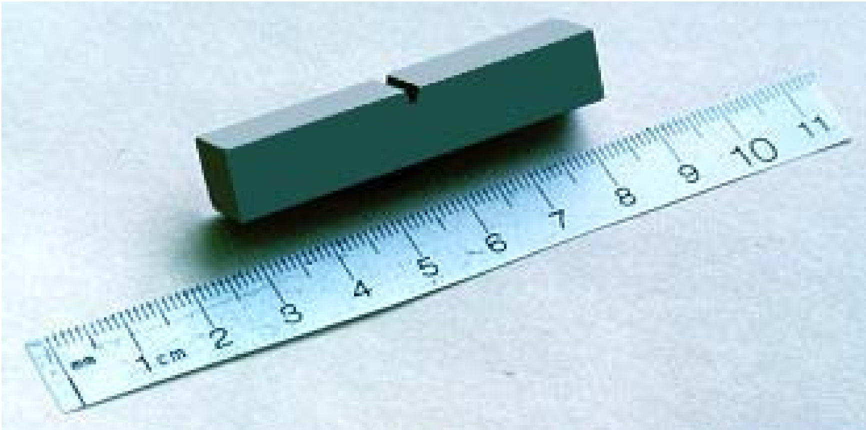


Figure 6.4 10 mm (full size) Charpy specimen with "V" notch.

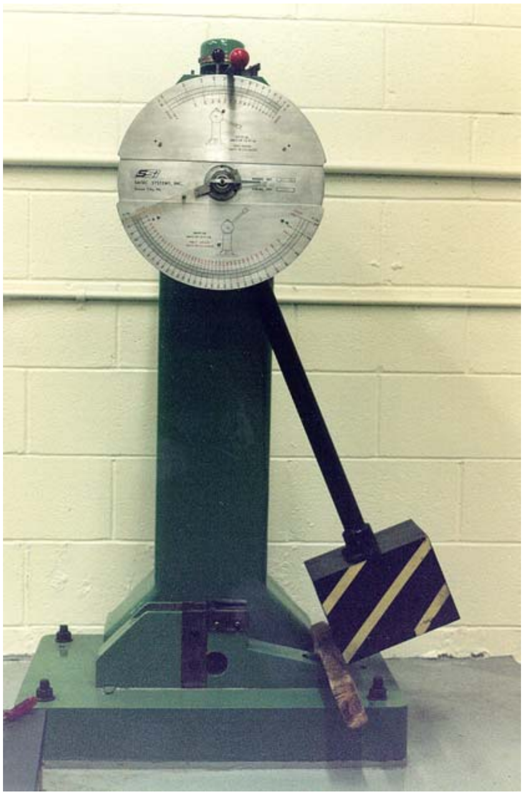


Figure 6.5 Charpy impact machine with the impact hammer supported on a wooden timber.

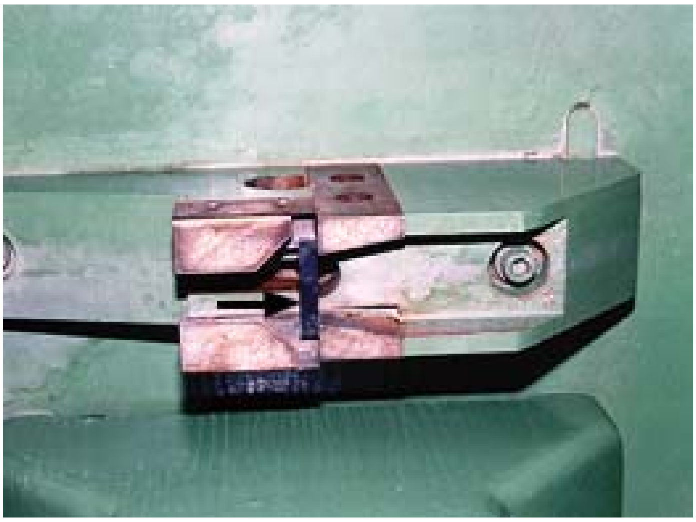


Figure 6.6 Top view of a Charpy test specimen (arrow) sitting in the anvil of the test machine ready to be tested.

UCS–66 gives the toughness design requirements for carbon and low alloy steels. Unless they are exempted by this item of the Division or item UG–20(f), materials require verification of toughness by Charpy impact testing. The exemption requirements are based on service temperature, material thickness, material chemistry, and steel making practice.

Table 6.1 Specific Carbon and Alloy Steel Product Exemptions from Impact Testing

Product Form	**Exemption Description**
Steel flanges	ANSI B16.5 and B16.47 are exempt at temperatures no colder than -20°F.
Nuts	UCS nut materials are exempt at temperatures no colder than -55°F.
UCS materials	Materials less than 0.10 inch thick are exempt at temperatures no colder than -55°F. Materials impact tested as per the material specification are exempt at temperatures no colder than the specification temperature.

[Interpretations VIII–1–89–76 and VIII–1–155R clarify that products made to the material requirements of ANSI flange standards do not qualify for exemption unless they meet all the requirements of the standard.]

Listed UCS materials may be exempt from impact testing when the minimum design metal temperature (MDMT) is greater than that determined from either Fig. UCS–66 or the tabular form of this figure, Table UCS–66. An exemption or lower minimum design metal temperature may be realized by lowering the stress in the vessel material.

[Interpretation VIII–1–86–230 makes it clear the exemption is applicable to a rotating vessel at a fixed location. Interpretation VIII–1–92–98 indicates that the exemption provision is applicable to truck, skid, or trailer mounted vessels provided the vessel is not in pressure service while being transported.]

This exemption is considered by calculating what the Committee calls the "coincident ratio." This is the ratio of the required minimum thickness for pressure retention (as calculated by part UG) times the minimum value of the joint efficiency or casting quality, as applicable, divided by the nominal thickness of the material less the corrosion allowance. Fig. UCS–66.1 can then be used to obtain a Fahrenheit temperature reduction that can be applied to Fig. UCS–66 or Table UCS–66. An alternative ratio to the coincident ratio can also be considered for Fig. UCS–66.1. The calculated primary membrane stress times the minimum value of the joint efficiency or casting quality as applicable, divided by the maximum allowable stress value times the joint efficiency can be used. For P–Number 1 materials, two further means of lowering the exemption requirement can be realized. By post weld heat treating the vessel (provided that other articles of the Division do not require it as indicated in UCS–68(c)), a temperature reduction of 30°F (17°C) may be realized from Fig. UCS–66 or Table UCS–66. UG–20(f) provides impact test exemption for P-Number 1 Groups 1 and 2 materials that meet the 5 requirements listed in this article.

Example 6.3 Charpy Impact Testing Exemption

A cylindrical vessel has a minimum required shell wall thickness of 2 inches if manufactured from SA–516 Grade 70 material. The service is not lethal, but the minimum metal temperature is -45°F. The material specification requires that this material be given a normalizing heat treatment. Accordingly, curve D of Fig. UCS–66 is applicable. The minimum Charpy impact test exemption from the figure is -4°F. Table UCS–56 requires post weld heat treatment for welded construction, so the temperature reduction option by heat treatment as per UCS–68(c) is not applicable. Since the vessel operating temperature and wall thickness exceed the requirements of UG–20(f), this option is also not available. To obtain an exemption from Charpy impact testing, the values of Fig. UCS–66.1 could be used. The maximum coincident ratio required from the figure is 0.59. By using the alternative ratio S*E*/SE, the required wall thickness can be determined. In this instance, E=E* and S* is the hoop stress in the heavier wall thickness. The designer must now consider the extra cost of fabricating with a thicker plate as opposed to the costs of Charpy impact testing. Other design options may also be available.

UCS–66(b)(2) requires that all carbon and low alloy steel materials be impact tested when the minimum design temperature is less than -55°F (-48°C) with the exception that if the coincident ratio is 0.35 or less, the material is exempt down to -155°F (-104°C). The Code User is cautioned that Charpy impact testing rules given in ASME VIII Division 1 are not accepted by all jurisdictional authorities. For example, the governing Standard for pressure vessel construction in Canada, CSA B51, requires impact testing when the minimum design temperature is less than -50°F (-46°C).

Material thickness is an important variable in fracture control. The thicker the material, the more susceptible it is to catastrophic fracture. In Fig. UCS–66 and Fig. UCS–66.1, thickness is an important variable in determining exemption from Charpy impact testing. In example 6.3, the controlling material thickness for the vessel shell is obvious. This may not be as apparent where thickness transitions occur. Generally, the thinner thickness in each component of the vessel is allowed to be the controlling thickness. It is this thinner thickness that usually has the ability to resist uncontrolled crack propagation. UCS–66 requires that each component that is essential to the structural integrity for pressure containment be considered for its own thickness. Thus, items like reinforcing pads must be considered separately from the shell, head, or nozzle materials; tubesheets considered separately from shell materials; and heads considered separately from shell materials. Figures to assist in determining the applicable thickness for welded construction are given in Fig. UCS–66.3 and are summarized in the following table.

Table 6.2 Governing Thickness for Charpy Impact Test Exemption

Configuration	**Exceptions**	**Governing Thickness**
Butt welds	castings, flat heads, tubesheets, and governing thickness greater than 4 inches and temperature less than 120°F	thickness of welded joint
Flat head butt welds	castings, governing thickness greater than 4 inches, and temperature less than 120°F	thinner of parts joined or ¼ of flat thickness
Tubesheet butt welds	castings, governing thickness greater than 4 inches, and temperature less than 120°F	thinner of parts joined or ¼ of tubesheet thickness
Corner welded joints	castings, governing thickness greater than 4 inches, and temperature less than 120°F	thinner of parts joined
Fillet weld joints	castings, governing thickness greater than 4 inches, and temperature less than 120°F	thinner of parts joined
Lap weld joints	castings, governing thickness greater than 4 inches, and temperature less than 120°F	thinner of parts joined
Multiple component welded assemblies	castings, governing thickness greater than 4 inches, and temperature less than 120°F	warmest of the individual components
Castings		largest nominal thickness
Governing thickness for welded fabrication greater than 4 inches and temperatures less than 120°F	castings	testing is mandatory
Nonwelded parts	castings, bolts, and governing thickness greater than 6 inches and temperature less than 120°F	¼ of the flat component thickness
Governing thickness for nonwelded parts greater than 6 inches and temperatures less than 120°F	castings and bolts	testing is mandatory

Thickness is not a normal descriptor for bolts and nuts. These product forms are provided for in the Notes associated with Fig. UCS–66. The commonly used SA–193 Grade B7 and B7M bolt materials are exempt to -40 and -55°F (-40 and –48°C) respectively, while SA–307 and SA–325 fasteners are exempt to -20°F (-29°C). Other bolting product exemptions are listed with Fig. UCS–66.

Even though the Committee has reduced the considerations required for classical fracture toughness calculations, there is considerable misunderstanding of the application of UCS–66 as indicated by the number of publications explaining the use of these rules as well as the significant number of Interpretation requests.

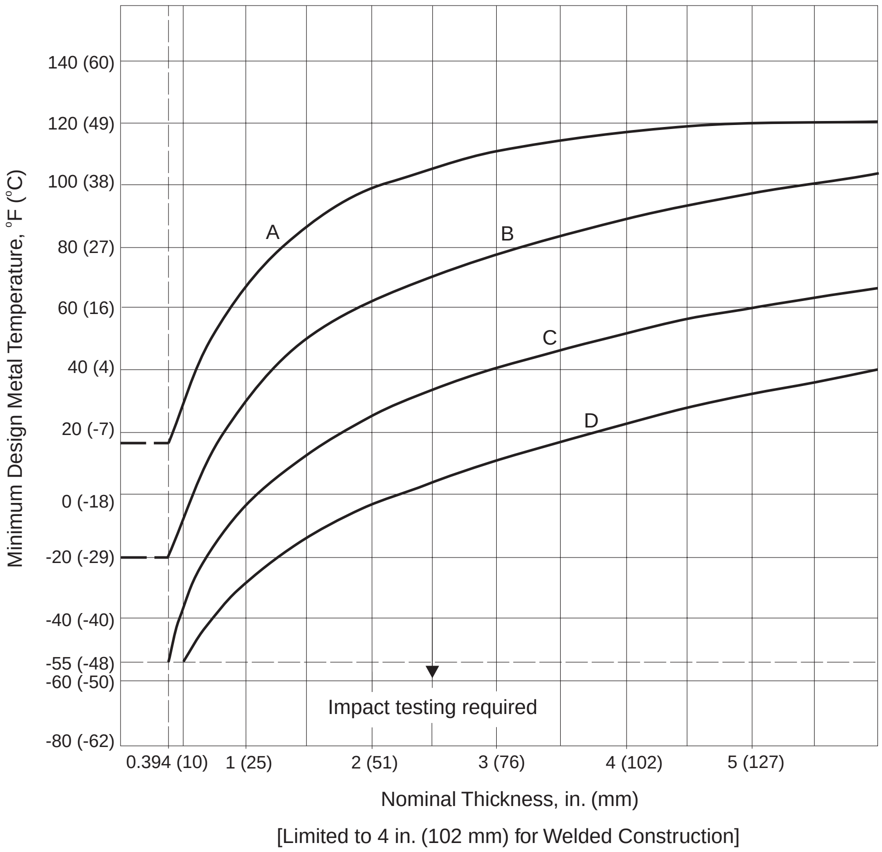


Figure 6.7 Fig. UCS–66 Impact Test Exemption Curves

GENERAL NOTES ON ASSIGNMENT OF MATERIALS TO CURVES:

(a) Curve A applies to:

(1) all carbon and all low alloy steel plates, structural shapes, and bars not listed in Curves B, C, and D below;

(2) SA-216 Grades WCB and WCC if normalized and tempered or water-quenched and tempered; SA-217 Grade WC6 if normalized and tempered or water-quenched and tempered.

(b) Curve B applies to:

(1) SA-216 Grade WCA if normalized and tempered or water-quenched and tempered

(b) Curve B applies to (continued):

SA-216 Grades WCB and WCC for thicknesses not exceeding 2 in. (51 mm), if produced to fine grain practice and water-quenched and tempered
SA-217 Grade WC9 if normalized and tempered
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-62 Grade B if not normalized;

(2) except for cast steels, all materials of Curve A if produced to fine grain practice and normalized which are not listed in Curves C and D below;

(3) all pipe, fittings, forgings 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 21 and 22 if normalized and tempered
SA-302 Grades C and D
SA-336 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 material of Curve B if produced to fine grain practice and normalized and not listed for Curve D below.

(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-662 if normalized
SA-738 Grade A
SA-738 Grade A with Cb and V deliberately added in accordance with the provisions of the material specification, not colder than -20°F (-29°C)
SA-738 Grade B not colder than -20°F (-29°C)

(e) For bolting and nuts, the following impact test exemption temperature shall apply:

Bolting

Spec. No.	Grade	Impact Test Exemption Temperature, °F (°C)
SA-193	B5	-20 (-29)
SA-193	B7 [2½ in. (64 mm) dia. and under]	-55 (-48)
	[Over 2½ in. to 7 in., (64 mm to 178 mm) incl.]	-40 (-40)
SA-193	B7M	-55 (-48)
SA-193	B16	-20 (-29)
SA-307	B	-20 (-29)
SA-320	L7, L7A, L7M, L43	Impact tested
SA-325	1, 2	-20 (-29)
SA-354	BC	0 (-18)
SA-354	BD	+20 (-7)
SA-449	…	-20 (-29)
SA-540	B23/24	+10 (-12)

Nuts

Spec. No.	Grade	Impact Test Exemption Temperature, °F (°C)
SA-194	2, 2H, 2HM, 3, 4, 7, 7M, and 16	-55 (-48)
SA-540	B23/24	-55 (-48)

(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 procedure necessary to obtain a fine austenitic grain size as described in SA-20.

NOTES:

(1) Tabular values for this Figure are provided in Table UCS-66.

(2) Castings not listed in General Notes (a) and (b) above shall be impact tested.

Example 6.4 Selection of Curve from UCS-66 for Fine Grain Material

[Interpretation VIII–1–92–134] Which curve is applicable for SA–216 Grade WCB made to a fine grain practice? SA–216 is a cast material. Note (b)(3) indicates cast materials normally will use curve B. However, Note (c)(2) allows curve B material made to a fine grain practice to use curve C. Even though SA–216 is not specifically listed as being applicable to any curve, a curve is applicable based on product form and this can be further modified because of steel making technique.

Example 6.5 Selection of Curve from UCS-66 for Unlisted Material

[Interpretation VIII–1–89–44] Which curve is applicable for SA–285 Grade C? SA–285 Grade C is a plate material that has no requirements for fine grain steel making practice nor normalized heat treatment. SA–285 is not listed as being applicable to any specific curve. The applicable curve therefore is A. Curve B cannot be used, as the material is not made to fine grain practice and is not normalized.

Example 6.6 Selection of Curve from UCS-66 for Charpy Tested Material

[Interpretation VIII–1–92–219] Which curve is applicable for SA–350 Grade LF2? SA–350 Grade LF2 is a forged material that has Charpy impact values specified for -50°F. None of the curves are applicable. Figure UCS–66 is applicable to -55°F and materials like SA–350 Grade LF2 (whose specification requires testing) are exempt down to their test temperature.

Not only must carbon and low alloy steel materials possess a minimum fracture toughness as determined by impact testing, but the welds made to join these materials and the resulting weld heat affected zones must also exhibit a minimum toughness (UG–84(f)). UCS–67 indicates that if impact testing of materials is required, then so is testing of the weld and heat affected zones in order to qualify the weld procedure. Testing of the weld metal may be exempt if the weld metal specification (SFA specification) requires testing at a temperature as cold or colder than the base metal requirements. When the minimum design temperature is less than -55°F (-48°C) and the parent materials are exempt from testing because of a low coincident ratio, the weld is still to be tested because the coincident ratio exemption does not apply to weld metal.

The toughness of base materials is usually reduced by weld heat input. The control of weld heat (preheat, amps, and weld travel speed) is required to reduce the destruction of the heat affected zone toughness. UCS–67 accordingly places more stringent requirements on heat affected zone testing. Impact testing is required when more than ½ inch (13 mm) weld metal is deposited in one pass and the design metal temperature is less than 70°F (21°C). Impact testing is also required when the design temperature is less than -55°F (-48°C), even though the base metals may be exempt. The heat affected zone of welds made without filler metal addition shall be impact tested if the thickness of the weld exceeds ½ inch (13 mm) at all temperatures, and if the thickness exceeds 5⁄16 inch (8 mm) for temperatures less than 50°F (10°C).

The toughness requirements given in the Division do not consider all aspects related to fracture control. In some instances the specified requirements may be incomplete or insufficient. For unusual designs involving cold temperatures or impact loading or both, the Code user may wish to conduct a more rigorous fracture mechanics analysis. The Division requirements for impact testing are minimum requirements.

Nonferrous Materials

Nonferrous metals do not contain iron as the major constituent of the alloy. These materials are copper and copper alloys, nickel and high nickel content alloys, aluminum and aluminum alloys, titanium and titanium alloys, and zirconium alloys. These materials are usually more expensive than steel materials and therefore are used in pressure vessel construction primarily for corrosion resistance or extreme cold temperature toughness. Chapter UNF presents the rules for construction with nonferrous materials.

Nonferrous Material Products

The approved materials for nonferrous construction are listed in Table UNF–23. This table is located at the end of Subsection C. Nonferrous materials are available as plate (UNF–6), forgings (UNF–7), castings (UNF–8), rods, bars and shapes (UNF–14), and bolt products (UNF–12 and UNF–13). In some instances bolts are manufactured by machining rather than by a forging process. UNF–12 specifies that the maximum stress for the bolt is to be based on the product form of the material (casting, forging) prior to the machining.

Fabrication of Nonferrous Materials

Welding or brazing fabrication of nonferrous materials is to be done to a procedure qualified in accordance with Section IX. Those nonferrous materials considered to be weldable or brazeable have a P-Number assignment.

Some of the nonferrous materials become embrittled when their alloy composition is not closely controlled. Since welding results in a mixing of materials at the weld, titanium and zirconium are only to be welded to themselves or each other to avoid embrittlement in the weld (UNF–19(b)). UNF–78 requires that these materials be welded only by gas shielded or vacuum environment weld processes as these processes are less likely to result in weld contamination. UNF–95 requires a production test plate for welds in these materials. Bend test coupons from the test plate are to be tested to qualify the weld(s) represented by the plate. UNF–19(a) requires that category A and B weld joints in titanium or zirconium fabrications be Type 1 or Type 2. UNF–57 requires radiographic examination of these joints because of the embrittlement risk and the need to detect any resulting cracks. As in other instances of crack susceptibility at weld zones, the Code user may also wish to conduct an ultrasonic examination even though the rules of the Division do not require this examination. The ultrasonic examination cannot be substituted for radiographic examination.

Some of the nickel based alloys are susceptible to incomplete weld fusion at the root of the joint because of oxide stability at high temperature. UNF–19 requires that category A and B welds in alloy 625 be either Type 1 or Type 2 and if the design temperature is 1000°F (538°C) or hotter, category C and category D welds also are to be either Type 1 or Type 2 welds. These weld types reduce the risk of incomplete fusion at the weld root.

UNF–57 requires radiography on welds involving a number of nickel alloy materials when the design thickness is greater than ⅜ inch (10 mm), which again, is a requirement to limit nonfusion type imperfections. The exempt nickel materials that are not prone to the nonfusion limitation are alloys 200, 201, 400, 401, and 600.

UNF–77 gives the requirements for forming nonferrous materials. Local deformations can be corrected by cold or warm pressing or by hammering. The use of heat is restricted depending on the material type. Advice on the temperature limitations for various materials is given in the NF Appendix located at the end of the Subsection. Further information can be obtained from the material manufacturer or may be deduced from the stress tables in Section II, Part D.

Toughness of Nonferrous Materials

Nonferrous materials are generally tougher than steel materials. Consequently, low temperature vessels are frequently constructed from them. UNF–65 gives the minimum service temperatures without further testing for the nonferrous materials.

Table 6.3 Minimum Service Temperature for Nonferrous Materials Without Further Testing

Material	Minimum Untested Temperature, °F
Aluminum alloys (wrought)	-452
Aluminum alloys (cast)	-325
Copper and copper alloys	-325
Nickel and nickel alloys	-325
Zirconium	-75
Titanium	-75

Finned Copper Tubes

Copper materials, SB–359, can be used for pressure equipment construction even though this material is not listed in Table UNF–23. These materials are copper and copper alloy tubes with integral fins. They have application in heat exchangers where the pressure may be internally or externally applied. For internal pressure, the rules in Part UNF apply. Mandatory Appendix 23 provides for the use of these products in external pressure applications. These tubes are restricted to pressures of 450 psi (3103 kPa) maximum and temperatures of 150°F (65°C) for external pressure application (23–4). The integral fins provide reinforcement, so the maximum allowable external pressure is to be determined by hydrostatic collapse testing in accordance with 23–3.

Work Hardened Nickel

The almost pure nickel alloys produced to SB–162 can have their strength enhanced by cold working. On vessels to be steam heated by an external steam jacket fabricated from nickel (typically food processing kettles), strengthening of the nickel against structural collapse from the external steam pressure is of value. Mandatory Appendix 21 gives the rule for the design and construction of jacketed work hardened nickel vessel construction. Because work hardening reduces the ductility of the nickel, it becomes more sensitive to stress concentrations. Appendix 21–3 requires weld metal reinforcement to be removed.

High Alloy Steels

Steels that contain alloy additions in excess of the AISI definition for a low alloy steel classification are classed as high alloy steels. In this classification there are a number of common materials used in the construction of pressure vessels, namely austenitic stainless steels, martensitic stainless steels, duplex stainless steels, and maraging steels. These materials are selected for toughness and corrosion resistance.

High Alloy Steel Products

The applicable material product forms are given by the material list in UHA–23, which is located at the end of Subsection C of the Division. All the product forms found for carbon and low alloy steels are also available in high alloy steels. The applicable permitted stress levels are found in Section II, Part D.

Fabrication of High Alloy Steels

Most of the high alloy steels are readily weldable. The high chromium steels and the martensitic stainless steels are susceptible to cracking as a result of welding because of the high hardening tendency created by weld heating. The Code user should be aware of the potential for fabrication difficulty with these materials because of cracking. Strict attention to weld preheat requirements and post heat requirements, as well as following good weld practices, is necessary to reduce cracking risks.

UHA–33 makes radiographic examination mandatory for welds in the martensitic stainless steel types 410, 429, and 430 because of the cracking risk. Also, if the weld filler metal used to fabricate the martensitic stainless steel, including type 405, is classed as straight chromium (no nickel alloy addition), then these welds shall also be radiographed. To enhance the sensitivity of the examination, UHA–21 requires that category A and B butt welds be Type 1 or Type 2 joints. Cracking can be latent in welds made with straight chromium filler metals. Therefore, the acceptance radiography is to be done after any post weld heat treatment. The Code user should consider radiographic examination of such welds prior to post weld heat treatment so that repairs can be done before heat treatment, as there is no temper bead approved weld repair after heat treatment.

The material will require reheat treatment and subsequent complete radiographic examination after the repair as required by UHA–32(e). [Interpretation VIII–1–86–16 indicates that spot radiography is not applicable. UHA–33(b) requires full radiography.]

UHA–33 also requires that any radiography done on austenitic stainless steel, whether it be Code mandated or design selected, be done after any post weld heat treatment. This, too, is a requirement necessitated by the enhanced cracking potential from heating but, in this case, the potential results from heat making the weld area sensitive to stress corrosion cracking. When austenitic or duplex stainless steels over ¾ inch (19 mm) are welded, the accessible surfaces of the weld area shall be examined for cracks by the liquid penetrant method of nondestructive testing (UHA–34). This examination is to be done after any post weld heat treatment.

Welding of high alloy steels is to be done in accordance with a weld procedure qualified to Section IX. UHA–33 gives the requirements for post weld heat treatment. Post weld heat treatment is not required for the high ductility materials in the high alloy steel classification, and, in some instances, post weld heat treatment is undesirable as it may deteriorate corrosion resistance or ductility, or both. Post weld heat treatment at the temperatures used for carbon and low alloy steels may also be detrimental to the high alloy steel because of the more complex metallurgical composition of many of these materials.

At high temperatures, solid state reactions may take place between the various alloy contents, and these may result in a deterioration of material properties. Some of the nonmandatory post weld heat treatments given in UHA–33 are not tempering or stress relief heat treatments, but are normalizing or solution heat treatments. These treatments reduce the residual stress in the material and also homogenize the metallurgical composition to give a material with a lower internal energy. Some of the treatments require rapid cooling which can create residual stress in the material and result in distortion of the fabrication. These high cooling rates are necessary if dwell times at certain temperatures are to be avoided. Solid state reactions occur at certain temperatures and can result in metallurgical phases that detract from the material's properties. The Code user is cautioned on arbitrary use of the optional heat treatments given in this paragraph.

The Code user should consult UHA–32 for exemptions from heat treatment for those materials listed in Table 6.4.

Table 6.4 Mandatory Post Weld Heat Treatments for High Alloy Steels

P-Number	Generic Description	Temperature, °F
P-No. 6 Groups 1,2,&3	403, 410, 429, CA15B,CA15C, CA15M-A	1250
P-No. 7 Groups 1&2	405, 408, 409, 430, 444	1350
P-No. 10E Group 1	446	1250
P-No. 10I Group 1	446, XM27, XM33	1350

When handling high alloy and nonferrous materials, a clean environment or dedicated work space is required to prevent material contamination. Figure 6.8 shows a stainless steel shell section that is rusting because of contamination by carbon steel dust.

Figure 6.8 Rust on the surface of a stainless steel section due to carbon steel dust contamination.

Toughness Requirements for High Alloy Steels

High alloy steels are frequently used in the construction of pressure vessels for cold temperature applications. As with other materials previously discussed, heating of the materials may detract from their toughness. UHA–51 requires impact testing of stainless steels when, during fabrication, they have been subjected to temperatures known to possibly result in embrittlement. This testing is required for vessel service at temperatures less than 70°F (21°C).

The requirements for material acceptability for the high alloy steels are based on lateral expansion of the Charpy impact test specimen, and not impact energy values. Lateral expansion is the recognized method of judging toughness acceptance in materials that normally exhibit higher absorbed energy values, even in an embrittled state. The lateral expansion more clearly demonstrates the ability of the material to deform and resist crack initiation and propagation in these more ductile materials than does an absorbed energy value. UHA–51 gives the acceptance criterion for lateral expansion in high alloy steels. This value varies with the test temperature.

UHA–51 gives exemptions for impact testing of high alloy steels. When vessels are made by welding fabrication using SMAW, SAW, GMAW, GTAW, and PAW, the production welds are exempt from testing at temperatures -320°F (-195°C) and warmer when all of the following conditions are met:

- weld procedure qualified in accordance with Section IX has Charpy impact tests meeting the vessel temperature requirements,
- weld metal carbon content is 0.10% maximum, and
- filler metal conforms to SFA–5.4, SFA 5.9, SFA–5.11, SFA–5.14, or SFA–5.22 and meets the special provisions of UHA–51(f)(4). (See Section II, Part C for SFA requirements.)

As with other materials, when the coincident ratio is less than 0.35. the materials of construction are exempt from impact tests provided the fabrication has not been heat treated in the temperature ranges described in Table 6.5. The other exemptions permitted by the Division are summarized in Table 6.6.

Table 6.5 Materials Requiring Impact Testing When Heated During Fabrication and to be Used with Service Temperatures Less Than 70°F (21°C)

Material Type	Typical Materials	Heat Exposure Range, °F
austenitic stainless steel	301, 302, 304, 308, 309, 310, 316, 317, 321, 304L, 308L, 316L	900 to 1650
duplex stainless steel	318, 323, 324, 325, 329	600 to 1750
ferritic stainless steel	405, 409, 429, 430, 446	800 to 1350

Table 6.6 Exemptions from Impact Testing for High Alloy Steels

Material Description	Specific Material Types	Subqualifications	Minimum Exempt Temperature, °F
stainless steel	all types	min. thickness < 0.099 in. and not heat treated	min. design metal temp.
austenitic chromium-nickel stainless steel	304, 304L, 316, 316L	carbon < 0.10%	-320
	321, 347	carbon > 0.10%	-55
	weld	carbon < 0.10% & welded without filler metal	-155
		carbon < 0.10% austenitic filler metals SFA–5.4, SFA–5.9, SFA–5.11, SFA–5.14, SFA–5.22.	-155
		carbon < 0.10% and austenitic filler metals SFA–5.4, SFA–5.9, SFA–5.11, SFA–5.14, SFA–5.22.	-50
austenitic chromium-manganese-nickel stainless steel	200 series	carbon < 0.10%	-320
		carbon > 0.10%	-55
duplex stainless steel		⅜ in. thickness and less	-20
	weld	≤ ⅜ in. thickness and less using filler metal of similar chemistry	-20
ferritic chromium stainless steels		≤ ⅛ in. thickness	-20
	weld	≤ ⅛ in. thickness using filler metal of similar chemistry	-20
martensitic chromium stainless steels		≤ ¼ in. thickness	-20
	weld	≤ ¼ in. thickness using filler metal of similar chemistry	-20

Cast Irons

Cast irons represent an economical material choice for some pressure vessel fabrications. Generally these materials are extremely brittle. The Code user should therefore use these materials with caution. Generically, the approved materials are gray cast iron, malleable iron, ductile iron, or nodular iron, and mixtures of gray and white cast iron whose elongation over a 2 inch (50 mm) gauge length is less than 15%. These materials are chemically similar to carbon and low alloy steels except that they all contain high carbon content that forms in the material matrix as graphite. They also contain a relatively high silicon content that reduces the melting temperature of ferrous materials and improves the flow characteristics of the molten metal making it easy to cast.

When the graphite is present as flakes in the material, it is called gray cast iron. The graphite flakes interrupt the continuity of the metal matrix and can simulate crack-like flaws. When the graphite is present as spherical particles, the material is called ductile iron or nodular cast iron. While this material does not have crack-like phases present, the nodules interrupt the metal continuity and thereby reduce the material toughness. The malleable cast irons have their graphite in the form of splats or shattered spheres and therefore have toughness intermediary between the gray and the nodular irons.

There is no free carbon in the white cast iron phase. The carbon content is present as carbides that are brittle materials. All cast irons are relatively brittle and, because of the brittle feature, cast iron vessels are usually restricted to relatively low pressure service and warmer temperatures.

Cast irons are available only as casting forms or as machined components. The material specifications for the acceptable grades of cast iron are given in Table UCI–23. The permitted stress values are also given in the table and in item UCI–23. The permitted stress value is only $1/10$ of the specified minimum tensile strength for cast iron materials.

Castings can have intricate shapes that can be difficult to mathematically analyze for stresses. In such instances, UG–101 allows for proof testing to demonstrate the adequacy of the design. For cast iron materials, the requirements of the proof test are different from other materials in that a burst test is required. UCI–101 presents the criterion for determining design adequacy based on the burst test.

Table 6.7 Pressure–Temperature Limitations for Cast Iron Vessels and Components

Component	Service	Maximum Pressure, psi	Maximum Temperature, °F
vessels	gas, steam, vapor	160	450
vessels and components	liquid	160	375
vessels and components	liquids at less than boiling point at design pressure and temperature	250	120
bolted heads and closures		300	450
vessels and components SA–278 class 40 , 45, 50, 55, and 60 stress relief annealed material	uniform temperature in the material	250	650
vessels and components SA–476 stress relief annealed material		250	450
flanges and flanged fittings ANSI B16.1 Class 125 and 250		ANSI B16.1 rating	450

UCI–2 prohibits cast iron construction for vessels in lethal service, vessels in flammable service, unfired steam boilers, and direct fired vessels. Cast iron is commonly used in steam containment vessels (steam chests) such as rollers and hot tables. UCI–3 places restrictions on the pressure, temperature, and service use of cast iron vessels and components.

Fabrication of Cast Irons

Cast iron materials generally are not to be welded for pressure vessel use (UCI–36). Cast irons are not considered to be weldable materials because of the extremely brittle phases and cracks that form in the weld heat affected zone. UCI–78 permits welding of cast iron to affect a repair only when a threaded or driven plug cannot be used. If welding is undertaken, then the weld area must be examined for crack-like flaws using either the magnetic particle or liquid penetrant nondestructive examining techniques. The requirements for making repairs using plugs are rigorously specified in UCI–78 where the intent is to reduce stress concentration. UCI–37 also presents requirements to reduce stress concentrations by requiring a “liberal radius” for all corners and changes in surface contour in cast iron.

The standard hydrostatic test for cast iron vessels is given in UCI–99. For pressures less than 30 psi (207 kPa), a hydrostatic test pressure of 2½ times the rated pressure up to 60 psi (414 kPa) maximum is to be used. At pressure ratings greater than 30 psi (207 kPa), a pressure of 2 times the rated pressure is to be used.

While the general methods of design given in item UG of the Division are applicable for most fabrications, UCI–29 has special requirements for dual cast iron fabrications (gray and white iron material). Vessels made from SA–667 or SA–748 materials shall have a minimum wall thickness of 5 inches (127 mm) and a diameter not greater than 36 inches (914 mm).

Cast Ductile Iron

The material specification SA–395 for ductile iron that has an elongation requirement in excess of 15% over a 2 inch (50 mm) gauge length has its own separate material section: UCD. Many of the service restrictions applicable to cast iron also apply for ductile iron. UCD–2 prohibits ductile iron from being used in the construction of vessels for lethal service but not for flammable service, which results from the more ductile performance of this material. Ductile iron is not permitted for unfired steam boilers or for direct fired vessels. UCD–3 restricts the service temperature range to -20 to 650°F (-29 to 343°C) and the pressure to 1000 psi (6900 kPa). This material is not to be welded (UCD–36 and UCD–78). UCD–78 requires that repairs to ductile iron be made by plugging.

Vessels made from SA–395 ductile iron shall be hydrostatically tested in accordance with UCD–99 to two times the rated pressure, and, as with new designs for cast iron in general, UCD–101 requires a burst test to prove incalculable designs.

Quenched and Tempered Steels

Approved pressure vessel steels, whose tensile properties are enhanced by quenching and tempering, are of the carbon and low alloy steel classification. Because of the rapid cooling from the austenitizing temperature (in the order of 1550°F (840°C) and hotter), their properties are significantly different from those of conventional carbon and low alloy steels.

Requirements for the use of these steels are presented in article UHT. The rules given in item UHT are not applicable to carbon and low alloy steels that are thick and have been quenched or rapidly cooled for fine grain practice. Quenching or rapid cooling can result in properties defined in the material specifications of Table UCS–23. The values given in UHT are also not applicable to integrally forged vessels that are quenched and tempered (UHT–1). The steels listed in Table UHT–23, which is found immediately prior to the Mandatory Appendices, are intended for welded construction. These steels are often incorrectly referred to as T1 steel. (T1 is US Steel's proprietary designation for a steel of the quenched and tempered class.) The quenched and tempered steels have enhanced strengths and improved toughness over carbon and low alloy steels. Consequently, they are frequently used for the fabrication of mobile pressure vessels such as the one shown in Figure 6.9.

Figure 6.9 This transport truck carries pressurized gas in a pressure vessel configured as a trailer.

Quenched and Tempered Steel Products

The approved quenched and tempered steel products listed in Table UHT–23 are either in plate, pipe, casting, or forged component form. These materials can only be used up to the maximum thickness given in the listed material specification. The listed specifications give relatively confined ranges of chemical composition for these steels to improve the effects that rapid cooling has upon strength development.

Quenching creates a very high internal energy within the steel so that, in the quenched condition, the steel is very susceptible to brittle fracture. As with stress relief heat treatment of welded carbon and low alloy steels, a reduction of this internal energy can be achieved. The quenched and tempered steels are always used in the tempered condition (stress relieved) where, even though a decrease in material strength is realized from the quenched state, a very significant increase in toughness is achieved. This makes these steels useful for pressure vessel construction.

Heating a quenched and tempered steel beyond its tempering temperature (in the range of 1200°F (650°C)) results in a destruction of strength, and, in many cases, a loss of toughness. UHT–5 requires that these materials be toughness tested at the minimum service temperature and that tests be done to ensure that welding, warm forming, or post weld heat treatments have not destroyed the strength and toughness of the material. The toughness test criteria are given in UHT–6. The minimum test requirement is a Charpy impact test with a lateral expansion acceptance criterion. Figure 6.10 shows lateral expansion being measured. Some of the materials also require a drop weight test to demonstrate that the material can withstand a prescribed blow without cracking or fracture at the minimum service temperature of the vessel.

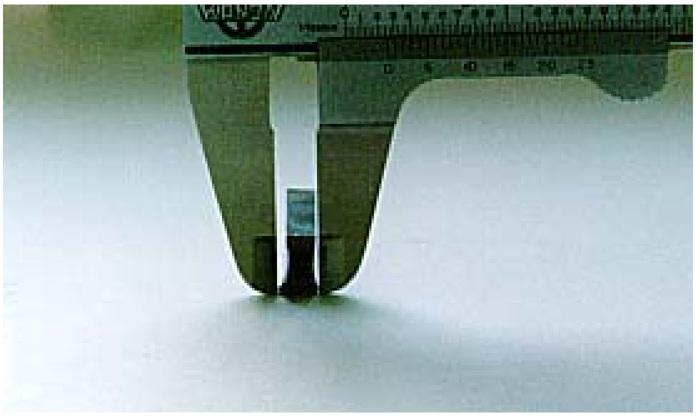


Figure 6.10 Vernier calipers being used to measure lateral expansion on a 5 mm (½ size) Charpy impact specimen.

Fabricating Quenched and Tempered Steels

The steels in item UHT are intended primarily for welded fabrication. A weld procedure in accordance with Section IX, and possibly as modified by UHT–82, is required. UHT–17 specifies that all welded joints must be Type 1 except category B welds in SA–333 Grade 8, SA–334 Grade 8, SA–353, SA–522, SA–553, and SA–645 materials, where Type 2 joints may also be used. In these same exempt steels, category C and D welds are specified to be full penetration.

UHT–18 and Fig. UHT–18.1 and 18.2 describe acceptable joint configurations for nozzle attachments to achieve the Type 1 joint. The Type 1 joint is required to facilitate the radiographic examination of welds as specified in UHT–57. All Type 1 welds shall be radiographically examined except nozzles with a 2 inch (51 mm) inside diameter and smaller nozzle attachment welds, in which instance, they are to be nondestructively examined using either the magnetic particle method or the liquid penetrant method. Magnetic particle or liquid penetrant examinations are to be done after hydrostatic testing, as surface flaws are more likely to grow during hydrostatic testing and therefore will be more readily detectable.

The Code user should consider the risk of brittle fracture during hydrostatic testing and conduct the nondestructive examination before hydrostatic testing as well as the mandatory examination after. In addition, consideration should also be given to doing this testing before post weld heat treatment, since any cracks will require repair. There is no provision for weld repair of quenched and tempered steels after post weld heat treatment without following the welding by another heat treatment.

Quenched and tempered steels have chemical composition designed to facilitate hardening upon fast cooling from temperatures greater than 1350°F (730°C). One of the consequences of nonuniform cooling of the entire metal piece can be cracks. This is the reason for the extensive nondestructive testing requirement of welds in quenched and tempered steels. UHT–83 requires that surfaces prepared by thermal cutting be either incorporated into a weld or be removed by machining or grinding, and the area examined for cracks by either the magnetic particle or liquid penetrant methods. UHT–85 requires temporary welds to be removed and the area examined for cracks using either the magnetic particle or liquid penetrant methods.

In addition to the essential variables given for weld procedures developed and tested in accordance with Section IX, UHT–82 presents essential requirements for weld procedures for the UHT steels.

UHT–56 requires that all the listed materials in Table UHT–23, regardless of thickness, be subjected to post weld heat treatment when the material is welded by the inertial and continuous drive friction weld processes. This is necessary because of the limited control on heat input on these processes rendering the materials susceptible to areas of high hardness.

The post weld heat treatment is to be in accordance with the temperatures and times at temperature listed in Table UHT–56. All other forms of welding may also require post weld heat treatment. As previously indicated, the heat treatment temperature must not exceed the tempering temperature of the material, otherwise the mechanical properties of the material will be altered.

The rates of heating and cooling of the material are given in UCS–56. Those material specifications that require accelerated cooling in the tempering cycle for the material shall also be cooled from the post weld heat treatment temperature using that same accelerated cooling rate.

The requirements for post weld heat treatment are given in Table UHT–56 along with the exemptions. The governing thicknesses are much more stringent for these materials than they are for carbon and low alloy steels as is indicated in UHT–56.

Table 6.8 Essential Variables in Addition to Those of Section IX for Weld Procedures for Quenched and Tempered Steels

Material	Essential Variable
All UHT material to be PWHT	• Weld filler metal shall contain less than 0.06% vanadium. (All listed UHT materials to be post weld heat treated.)
SA–508 and SA–543	• Increase in maximum preheat or interpass temperature. • Preheat temperature to be a minimum of 100°F for ½ inch and less, 200°F for over ½ inch to and including 1½ inch, 300°F above 1½ inch. • Decrease in minimum preheat or interpass temperature. • Range of preheat temperatures is not to exceed 150°F. • Heat treatment shall be identical to that done to the vessel or component (soak temperature and time, and cooling rate). • A change in weld heat input (change in voltage, amperage, or travel speed). • An increase in base material beyond that used in the qualification test for materials that are quenched and temper heat treated after welding. The minimum thickness qualified is ¼ inch. • For materials that are not quench and temper heat treated after welding, the minimum thickness qualified for a test coupon thickness of less than ⅝ inch shall be the coupon thickness. For thickness ⅝ inch and greater, the minimum thickness qualified shall be ⅝ inch. In all cases the maximum thickness qualified shall be two times the thickness of the test coupon. • SMAW electrodes shall conform to SFA-5.5 and shall be taken from undamaged hermetically sealed containers, or shall be baked at 700 to 800°F for 1 hour. • SMAW electrodes of a strength less than E100XX shall have a maximum moisture content in the coating of 0.2% by weight. • SMAW electrodes shall be used within ½ hour of removal from a hermetically sealed container or an electrode storage oven operating at least at 250°F, otherwise they shall be dried at 700 to 800°F for 1 hour.
SA–517 and SA–592	• Increase in maximum preheat or interpass temperature. • Decrease in minimum preheat or interpass temperature. • Range of preheat temperatures is not to exceed 150°F. • Heat treatment shall be identical to that given to the vessel or component (soak temperature and time, and cooling rate). • A change in weld heat input (change in voltage, amperage, or travel speed). • SMAW electrodes shall conform to SFA-5.5 and shall be taken from undamaged hermetically sealed containers or shall be baked at 700 to 800°F for 1 hour. • SMAW electrodes of a strength less than E100XX shall have a maximum moisture content in the coating of 0.2% by weight. • SMAW electrodes shall be used within ½ hour of removal from a hermetically sealed container or an electrode storage oven operating at least at 250°F, otherwise they shall be dried at 700 to 800°F for 1 hour.

For clad or weld overlaid vessels, the governing thickness shall include the clad or weld thickness. The governing thickness at connections and attachments is to be the maximum thickness at the point of attachment. UHT–82 provides an exemption from post weld heat treatment for SA–517 and SA–592 materials when they:

- are 1¼ inches (32 mm) thick or less,
- have been welded using a minimum preheat of 200°F (93°C),
- have been welded using a maximum interpass temperature of 400°F (204°C),
- are held at 400°F (204°C) for a minimum of 4 hours after welding and without any cooling between welding and the low temperature tempering, and
- are nondestructively examined in accordance with the requirements of part UHT.

While the quenched and tempered steels have good toughness, their high strength makes them more vulnerable to stress concentrations. UHT–19 has special requirements for the skirt length on a conical section to reduce stress concentration compounding caused by a circumferential weld and a taper transition. The weld joint alignment requirements are more restrictive than for other materials and are given in UHT–20. UHT–84 requires that the maximum weld reinforcement on welds made to SA–517 material be limited to 10% of the material thickness or ⅛ inch (3.2 mm), whichever is less.

Although UHT–84 is not specific about the governing thickness, it follows that it will be the minimum thickness of the joint. UHT–34 requires that welds occurring in or adjacent to a taper section attachment for a hemispherical head be tapered and finished in a manner which allows the uniform run out of the taper. Again, this is to reduce the compounding effects of stress concentrators in close proximity. Some component configurations have inherent stress concentration compounding or higher inherent stress concentration than is desirable for the higher strength steels. For this reason, UHT–32 generally limits head designs to ellipsoidal or hemispherical configurations.

Construction Techniques Requiring Special Material Considerations

In Subsection C there are three items that are concerned with construction techniques. Use of these techniques, however, has significant consequences for the materials involved. Consequently, the Committee has decided to place these techniques in the materials subsection rather than in Subsection B, Fabrication Techniques. These techniques are construction with a material that has an integral corrosion resistant layer, item UCL; construction of vessels in material layers, item ULW; and construction with materials having a higher allowable stress at low temperatures, item ULT.

Vessel Construction with Integral Clad Corrosion Resistant Linings

Clad vessel construction can be an economical alternative to constructing a vessel entirely from a high alloy steel or nonferrous material. In this instance, a more economical steel is integrally joined with a more expensive corrosion resistant material, the thickness of which is limited to that necessary to provide the corrosion resistant life of the vessel. The more economical material is usually selected to provide the mechanical strength for the vessel. In many instances, the corrosion resistant materials are not as strong as the less corrosion resistant materials, so a clad vessel can be used to reduce the vessel wall thickness, weight, and cost.

There are three material specifications for clad plate materials: SA–263, SA–264, and SA–265. These specifications are for an ASME approved steel material that is clad with either a chromium steel, a chromium-nickel steel, or nickel or a nickel alloy steel. The method of joining the two materials is the manufacturer's choice. Explosive bonding, diffusion bonding, forge bonding, and roll bonding are some of the methods that may be used. Section UCL is not limited to these specified materials. Materials can be made using weld overlay as shown in Figures 6.11 and 6.12 when the welding is done in accordance with a procedure conforming to Section IX. Other materials can be made by welding sheet linings onto the substrate steel. However, item UCL is applicable only to linings that are integral or are attached by welding to the base material [Interpretation VIII–1–83–136].

Figure 6.11 A welder is placing a layer of austenitic stainless steel as an overlay onto the inner surface of a large nozzle fabricated from carbon steel.

UCL–3 includes an important cautionary note concerning the weld attachment of clad linings that are of significant chemical difference. Brittle fusion line phases may be formed in the weld. This can affect the corrosion resistance and mechanical performance of the clad material. This is particularly possible for some welds between nickel alloys and carbon steel. Another important note is given with UCL–32. In this instance, the note is a reminder of the possible difficulties that may result when the two constituents have significantly different thermal expansion behavior. Restraint at joints must be carefully analyzed by the Code user.

Because cladding is usually for corrosion protection only, the Division requirements for the base metal apply in most circumstances. UCS–23 provides for the thickness of the corrosion resistant layer contribution to the base metal strength only for integral linings and when the thickness of the corrosion resistant layer exceeds that provided as a corrosion allowance. Then, a ratio of the permitted stress values may be used to calculate the effective contribution of the surplus corrosion resistant material to the strength of the vessel. UCL–24 requires that when the lining contributes to the vessel strength, the maximum service temperature shall be the lower permitted temperature of the two materials considered separately.

Figure 6.12 The initial few passes of the stainless steel being deposited in Figure 6.11. The final surface may be machined, ground smooth, or left as is.

UCL–34 requires post weld heat treatment in accordance with the requirements for the base material. An important cautionary note goes with this requirement regarding the post weld heat treatment of chromium-nickel cladding. This material can experience loss of chromium in solid solution because of carbide precipitation at temperatures above 800°F (425°C). Chromium reduction from solid solution can destroy the corrosion resistance of the clad material. UCL–36 has special requirements for welds on chromium stainless steel cladding. Welds made with chromium filler metal require examination along their full length. When these welds are in continuous contact with the base metal welds, they are to be radiographically examined. When the chromium filler metal welds only cross base metal welds, they require examination by any test means that is sensitive to surface cracks. Weld metal dilution may result in alloy contents in the chromium weld deposit that make the deposit susceptible to the formation of brittle microstructures. The possibility of cracks in this brittle material increases with stress that is usually greater at welds. Welds made with chromium-nickel filler metals or non-air hardening nickel-chromium-iron filler metals are not susceptible to hard microstructure formation. UCL–36 only requires a spot radiographic examination in accordance with UW–52 for these filler metals when they are used to join chromium stainless steel cladding.

Information and guidance is provided in the Division for lining installations in general. Aside from the rules given in UCL, Nonmandatory Appendix F provides general advice on linings.

F–2 indicates that the lining thickness for corrosion resistant applications should be at least twice the estimated material corrosion loss to be expected between inspection periods. Consideration should also be given to providing a corrosion allowance for the base material in case of lining failure or penetration under the lining.

F–3 advises on the limitations of paint coatings for corrosion protection and suggests that a corrosion allowance should be used as if there were no protection at all. Paint coatings frequently have pinholes and thin spots regardless of the extent of inspection conducted for these imperfections. F–2 further advises of hydrogen cracking problems on welded linings installed on carbon steels, and encourages following good welding practices by welding only on clean substrate materials.

F–4 provides cautionary advice for a lining that fails during hydrostatic testing when test fluid gets trapped behind the lining. This fluid needs to be drained and the interface dried out before repairs are done so that high vapor pressure between the lining and base material will not occur if the vessel is warmed.

For more information on cladding products and their application, consult *CASTI Handbook of Cladding Technology* published by *CASTI* Publishing Inc.

Layered Construction of Vessels

Layered construction is similar in some aspects to cladding. This construction technique may be employed for corrosion design, although it is usually used to develop enhanced strength of the vessel. ULW–2 identifies an inner vessel that is used for pressure containment and outer layers used for opening reinforcement or thickness transition. Many of the requirements for the fabrication methods for single wall vessels remain with only minor modifications to accommodate the unique features of the layered construction. ULW–32 provides additional requirements for weld procedures beyond those given in Section IX. ULW–54 provides alternatives for radiographic examination to reduce the loss of inspection sensitivity caused by the unbonded layer intersections with butt weld joints. ULW–76 requires vent holes between layers to prevent pressure accumulation between layers. Such an occurrence could result in the failure of one of the layers. Accordingly, ULW–77 specifies undertakings to limit the gap that may develop between layers during construction, and also specifies acceptable limits on gaps that do form.

Low Temperature Vessels with Higher Allowable Stress

Materials known to exhibit good toughness can be used at higher stress values than those given in Section II, Part D when their maximum design temperature is less than 100°F (38°C). ULT–2 gives the conditions for higher level stress, specifically: the safety relief valve must be set at the pressure of the liquid at maximum operating temperature, the allowable stress at 100°F (38°C) shall be used for gaseous pressures caused by the liquid static head, and the vessel shall be externally insulated. Only type 304 stainless steel; 5%, 8%, and 9% nickel steels; and aluminum alloy 5083 as listed in ULT–23 can be used at higher stress levels at low temperatures.

Generally, the requirements given in Sections A and B, as well as those given in UNF and UHA, apply. Stress concentrators are significant because of the higher permitted nominal stress in the material. The fit up and alignment tolerances given in UHT are applicable (ULT–17). ULT–57 requires complete radiographic examination of butt welds. All other pressure retaining welds are to be examined either by ultrasonic examination or liquid penetrant examination. ULT–82 requires the weld procedures for fabrication of higher stress low temperature vessels to include a list of additional essential variables.

Material Selection

The Code does not require specific materials for certain environments. It does, however, restrict the use of some materials in environments where the use of the material poses an unacceptable risk. It is unreasonable to expect the Code to provide material specifications for the millions of environmental service conditions. The owner and designer are best qualified to determine what materials within the scope of the Division provide the most reliable, secure, and economic performance. The Division provides rules and guidance for material selection. At the end of sections UCS, UNF, and UHA, there are Nonmandatory Appendices CS, NF, and HA, respectively that provide guidance on the use of the materials discussed in those items; a miniature text on the metallurgical shortcomings of the various material groups.

For guidance on material selection for corrosion resistance, the Code user should consult sources other than ASME. For example, the National Association of Corrosion Engineers (NACE International) and ASM International publish books and technical papers that provide data and experience with materials in corrosive environments. This information is available in both hard copy and computer software.

Example 6.7 Material Selection in Corrosive Environment

An owner wishes to have a tube in a tube heat exchanger designed and built that will carry a liquid stream containing phenols, acetates, and chlorides in dilute concentration at 700°F and 2500 psi. The pH of the system can be as low as 5.5. Some size restraints for the equipment are also in effect. Heat transfer calculations indicate that an NPS 1½ inner tube and an NPS 2 outer tube would be the best size arrangement. The designer calculates that SA–106 Grade C schedule 80 pipe will work for the outer tube and that SA–106 Grade B schedule 40 pipe will be satisfactory for the inner tube in accordance with the requirements of UG–27 and UG–28. A corrosion allowance of 0.040 inch is provided. For high alloy materials, an austenitic stainless steel is considered, such as alloy 20. It is anticipated that the product stream will actually have phenolic and acetic acid because of the pH. A review of the corrosion literature reveals no reported data for the operating conditions of the exchanger. The available information, of which there is a significant quantity, indicates that phenolic and acetic acid are extremely corrosive to carbon steel in small concentrations and at all temperatures and pressures studied. Austenitic stainless steel and alloy 20 are reported to be relatively immune to these acids at the reported conditions. UHA–102 and UHA–103 provide cautionary information on the performance of high alloy materials that may be susceptible to stress corrosion cracking. The corrosion literature identifies chloride solutions as causing stress corrosion cracking in highly stressed or welded austenitic stainless steel. Alloy 20 is reported to be resistant, but again, there is no information for the service conditions for the exchanger. Alloy 20 also represents a significant cost premium over stainless steel. The owner opts for carbon steel construction. In accordance with the available information, the designer's knowledge, and the requirements of UG–25, the designer specifies an expected life for the exchanger of 1 year continuous operation, or 4 years on a 10% duty cycle.

Material selection also has to be considered for ease of fabrication, both for cost considerations and for failure resistance. Cast irons have been previously discussed where there is a certain ease of construction for some vessel designs. Thus, cast iron can be an economical choice. The designer has to consider the Code stress restrictions because of failure risk, as well as directly considering the risk that may be induced by the service conditions on this material. Considerations for impact loads or thermal gradients should be considered, as indicated in U–2.

Materials, particularly carbon and low alloy steels, need to be considered for toughness. Some of the classic failures of pressure vessels have resulted from insufficient toughness. Although the Division mandates toughness requirements, considerable evaluation is still required to select the optimum material at the most economical cost.

Example 6.8 Material Selection for Cold Service

A carbon steel vessel is to be built for fixed service at a maximum temperature of 260°F. This vessel will be subject to occasional cold startup. The coldest mean daily temperature at the vessel operating site is -20°F. SA–516 Grade 70 material is readily available, as is SA–106 Grade B pipe and SA–105 ANSI B16.5 flanges. These materials are initially considered for fabrication of the shell, head, and nozzles of the vessel. Calculations in accordance with UG–27 and UG–32 indicate that a minimum wall thickness of 0.45 inch for the head, 0.46 inch for the shell, and 0.31 inch for the NPS 4 nozzles is required. UG–20 requires the minimum design metal temperature (MDMT) to be -20°F. The requirement for impact testing must be evaluated in accordance with UCS–66 and Fig. UCS–66. SA–516 materials are mandated to be in the normalized condition in thicknesses of 1 inch and over. Testing of the shell and head material can be evaluated using either curves B or D in Fig. UCS–66. The maximum metal thickness at -20°F for SA–516 Grade 70 material in the hot rolled condition without Charpy impact tests to prove toughness is 0.394 inch. For the material in the normalized heat treated condition, the maximum metal thickness is 1.25 inches as given in Table UCS–66. The material selection choices are to use the normalized product or to test the as-rolled product. Another alternative is to increase the metal thickness beyond the minimum calculated in UG–27 and UG–32, plus the material added for corrosion allowance (and any other required allowance, such as thread allowance). From Table UCS–66, a temperature reduction of approximately 13°F is required to exempt Charpy impact testing. From Fig. UCS–66.1, this temperature reduction can be achieved by increasing the wall thickness by about 0.08 inch for both the head and shell. For the nozzles, curve A of Fig. UCS–66 is applicable. This material will either have to be tested, normalized, or increased in wall thickness to obtain a 38°F temperature reduction from Fig. UCS–66.1. UCS–66(c) specifies that no testing is required for the ANSI B16.5 flange.

Unlisted Materials

The Boiler and Pressure Vessel Committee makes it clear that only listed materials shall be used for pressure vessel construction. The Committee will consider unlisted materials for eventual listing in accordance with the general procedure given in Appendix 5 of Section II Part D. Appendix 5 gives the Code policy on new materials. These must either be already listed as an ASTM or AWS standard, or a request for consideration by the Committee can be made. This request must be coincident with a request for Standard Specification to either ASTM or AWS, as applicable. The request for approval must include extensive data and information as listed in the Appendix. The Committee may then list the material for a three year period as a Code Case.

Chapter

7

CYLINDRICAL AND SPHERICAL PARTS SUBJECTED TO INTERNAL AND EXTERNAL PRESSURE

Theory

The primary purpose of a pressure vessel is to separate two or more areas of different pressures. In most cases the vessels are subjected to an internal pressure that is greater than the atmospheric or ambient pressure on the outside of the vessel.

The pressure difference between the inside and outside of the vessel produces a stress in the vessel walls. The design process involves selecting an economic wall thickness such that the vessel can safely operate with the produced stress. In order to accomplish this, formulas that relate the pressure, stress, and wall thickness must be utilized.

Figure 7.1(b) shows a thin walled cylindrical section of length ΔL subjected to an internal pressure P. The section is in equilibrium. A thin walled vessel has a small ratio of wall thickness to radius so that the distribution of the normal stress across the wall thickness is essentially uniform. The force F is the force caused by the pressure, and the force W, is the resultant internal force on the section. Summing the forces in the y direction gives an equation for the circumferential stress S_C. The stresses in the wall are given as:

$$S_C = \frac{RP}{t} \tag{7.1}$$

where: R = inside radius of the cylinder
t = thickness of cylinder
P = internal pressure

The stresses in the longitudinal direction can be determined by analyzing the forces in the x direction. Figure 7.1(c) is a free body diagram of the forces in the x direction. Summation of forces gives an equation for the longitudinal stress S_L in the cylinder's wall.

$$S_L = \frac{RP}{2t} \tag{7.2}$$

The notation in Equation 7.2 is the same as that in Equation 7.1.

Equation 7.1 shows that the stress across a longitudinal plane is twice the stress across a circumferential plane. Stated another way, a longitudinal joint must be twice as strong as a circumferential joint. The above formulas assume that the cylinder walls are thin compared to the radius and that the stresses are uniform across the wall thickness.

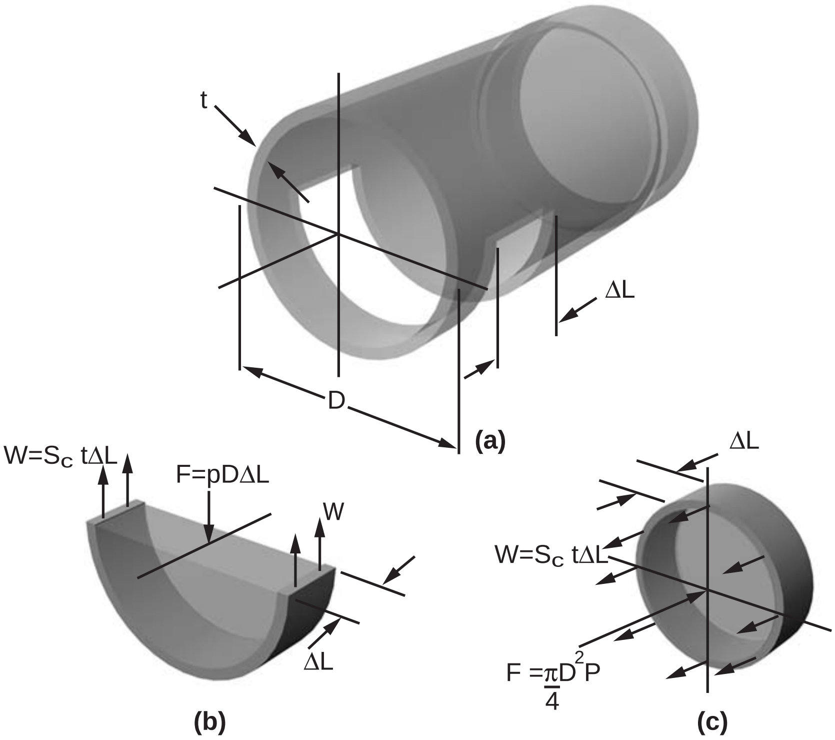


Figure 7.1 Forces and stresses in a pressurized cylinder.

Note that Figure 7.2 displays the forces which exist in a sphere if one passes a plane through its center. The stress in a thin wall sphere is given by Equation 7.2.

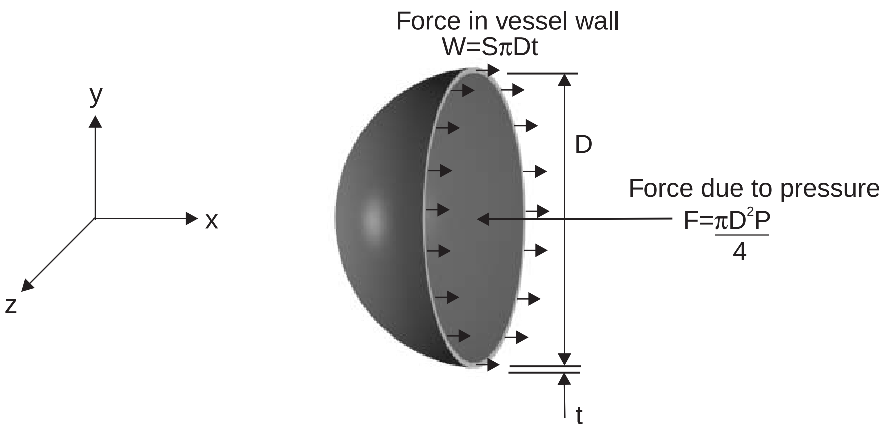


Figure 7.2 Forces and stresses in a sphere under pressure.

Thickness of Shells Under Internal Pressure

Code paragraph UG–27 gives the Code formulas for determining the thickness of cylindrical and spherical vessel shells designed to resist internal pressure. The nonpressure loads listed in UG–22 must also be considered, and, if necessary, the thickness must be increased or stiffeners and other means must be provided to prevent overstress.

Code Formulas for Circumferential (Hoop) Stress

The Code formulas in paragraph UG–27 are the same as Equations 7.1 and 7.2 with the exception of a correction factor in the denominator. The correction factors extend the thickness over radius ratio limitation of the formulas making them applicable to a larger range of vessel design thicknesses.

Two formulas are given for each circumstance: one for determining the minimum thickness based on the design pressure, and another for determining the pressure based on a given thickness.

The Code formulas for determining the thickness or pressure in a longitudinal joint (circumferential stress) of a cylinder when the thickness does not exceed one-half the inside radius or P does not exceed 0.385SE are:

$$t = \frac{PR}{SE - 0.6P} \text{ or } P = \frac{SEt}{R + 0.6t} \tag{7.3}$$

where:

t = minimum required thickness of shell in inches
P = internal design pressure, in pounds per square inch (psi)
R = inside radius of the shell course under consideration in inches
S = maximum allowable tensile stress value in psi as given in Section II, Part D. See Table 7.1.
E = joint efficiency of the appropriate joint in the shell or the efficiency of the ligaments between openings, whichever is less. Use Code Table UW–12 for welded joints. See Chapter 4.

The 0.6 factor is the correction that extends the applicability of the thin wall formula for circumferential stress to a thickness equal to one-half the inside radius. The formulas in Appendix 1 of the Code must be used if the thickness exceeds one-half the radius or P exceeds 0.385SE.

Table 7.1 Allowable Stress Table, Excerpt from Section II, Part D, Table 1A

Line No	Nominal Comp.	Product Form	Spec No.	Type/ Grade	Alloy Design./ UNS No.	Class/ Cond./ Temper	Size/ Thickness, in.	P–No.	Group No.
23	Carbon steel	Forgings	SA-508	1	K13502	---	---	1	2
24	Carbon steel	Forgings	SA-508	1A	K13502	---	---	1	2
25	Carbon steel	Forgings	SA-541	1	K03506	---	---	1	2
26	Carbon steel	Forgings	SA-541	1A	K03506	---	---	1	2
27	Carbon steel	Cast Pipe	SA-660	WCB	J03003	---	---	1	2
28	Carbon steel	Forgings	SA-765	II	K03047	---	---	1	2
29	Carbon steel	Plate	SA-515	70	K03101	---	---	1	2
30	Carbon steel	Plate	SA-516	70	K02700	---	---	1	2
31	Carbon steel	Wld. Pipe	SA-671	CB70	K03101	---	---	1	2
32	Carbon steel	Wld. Pipe	SA-671	CC70	K02700	---	---	1	2
33	Carbon steel	Wld. Pipe	SA-672	B70	K03101	---	---	1	2
34	Carbon steel	Wld. Pipe	SA-672	C70	K02700	---	---	1	2

Line No.	Min. Tensile Strength, ksi	Min. Yield Strength, ksi	Application & Max. Temperature Limits (NP = Not Permitted) (SPT = Supports Only)			External Pressure Chart No.	Notes
			I	III	VIII-1		
23	70	36	NP	700	1000	CS-2	G10, T2
24	70	36	NP	700	1000	CS-2	G10, T2
25	70	36	NP	700	1000	CS-2	G10, T2
26	70	36	NP	700	1000	CS-2	G10, T2
27	70	36	1000	700	NP	CS-2	G1, G10, G17, G18, S1, T2
28	70	36	NP	NP	650	CS-2	---
29	70	38	1000	700	1000	CS-2	G10, S1, T2
30	70	38	850	700	1000	CS-2	G10, S1, T2
31	70	38	NP	700	NP	CS-2	S5, W10, W12
32	70	38	NP	700	NP	CS-2	S6, W10, W12
33	70	38	NP	700	NP	CS-2	S5, W10, W12
34	70	38	NP	700	NP	CS-2	S6, W10, W12

Line No.	Maximum Allowable Stress, ksi (Multiply by 1000 to obtain psi), for Metal Temperature, °F, not exceeding													
	-20 to 100	150	200	250	300	400	500	600	650	700	750	800	850	900
23	20.0	20.0	20.0	---	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3	6.7
24	20.0	20.0	20.0	---	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3	6.7
25	20.0	20.0	20.0	---	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3	6.7
26	20.0	20.0	20.0	---	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3	6.7
27	20.0	---	20.0	---	20.0	20.0	19.6	18.4	17.8	17.2	14.8	12.0	9.3	6.7
28	20.0	20.0	20.0	---	20.0	20.0	19.6	18.4	17.8	---	---	---	---	---
29	20.0	20.0	20.0	---	20.0	20.0	20.0	19.4	18.8	18.1	14.8	12.0	9.3	6.7
30	20.0	20.0	20.0	---	20.0	20.0	20.0	19.4	18.8	18.1	14.8	12.0	9.3	6.7
31	20.0	---	20.0	---	20.0	20.0	20.0	19.4	18.8	18.1	---	---	---	---
32	20.0	---	20.0	---	20.0	20.0	20.0	19.4	18.8	18.1	---	---	---	---
33	20.0	---	20.0	---	20.0	20.0	20.0	19.4	18.8	18.1	---	---	---	---
34	20.0	---	20.0	---	20.0	20.0	20.0	19.4	18.8	18.1	---	---	---	---

Code Formulas for Longitudinal Stress

The Code formulas for longitudinal stress in the circumferential joints of a cylindrical shell are:

$$t = \frac{PR}{2SE + 0.4P} \text{ or } P = \frac{2SEt}{R - 0.4t} \tag{7.4}$$

Formula 7.4 is limited to use when the thickness is less than or equal to one-half of the inside radius or P does not exceed 1.25SE. The variables are the same as those in Formulas 7.3. Formulas 7.3 and 7.4 reflect the fact that the longitudinal joint must be stronger than the circumferential joint of a cylindrical vessel. Footnote 16 of Part UG points out that normally the circumferential stress acting on the longitudinal joint will govern. However, this is not the case when the circumferential joint efficiency is less than one-half the longitudinal joint efficiency or when nonpressure loads cause high stresses in the longitudinal direction.

Code Formulas for Spherical Shells

When the thickness does not exceed 0.356R, or P does not exceed 0.665SE, the following formulas for determining the thickness and internal pressure of a sphere apply:

$$t = \frac{PR}{2SE - 0.2P} \text{ or } P = \frac{2SEt}{R + 0.2t} \tag{7.5}$$

The variables are the same as those in Formulas 7.3. As with Formulas 7.3 and 7.4, the formulas in Code Appendix 1 must be used if the thickness or pressure exceeds the limitations. The above formulas represent dimensions in the corroded conditions (UG–16), as do all Code formulas.

Example 7.1 Thickness of a Cylindrical Shell

The upper cylindrical part of the vessel shown in Figure 7.3 has a 120 inch (3050 mm) inside diameter, Type 1 butt welds, and a corrosion allowance of 0.125 inch. The design temperature is 800°F and the design pressure is 150 psi. The material is SA–516 Grade 70, and full radiography will be performed.

Find the minimum thickness of the section:

(a) with the inside radius corrected for the corrosion allowance and
(b) without the correction.

Solution: Use Code UG–27 Formulas 1 and 2 which are reproduced above as formulas (7.3) and (7.4). The allowable stress in Part D of Section II, Table 1A, for SA–516 Grade 70 (Table 7.1) at 800°F is 12,000 psi. Table UW–12 states that the joint efficiency for a fully radiographed Type 1 butt joint is 1.0. P is less than 0.385SE and t is less than $R/2$.

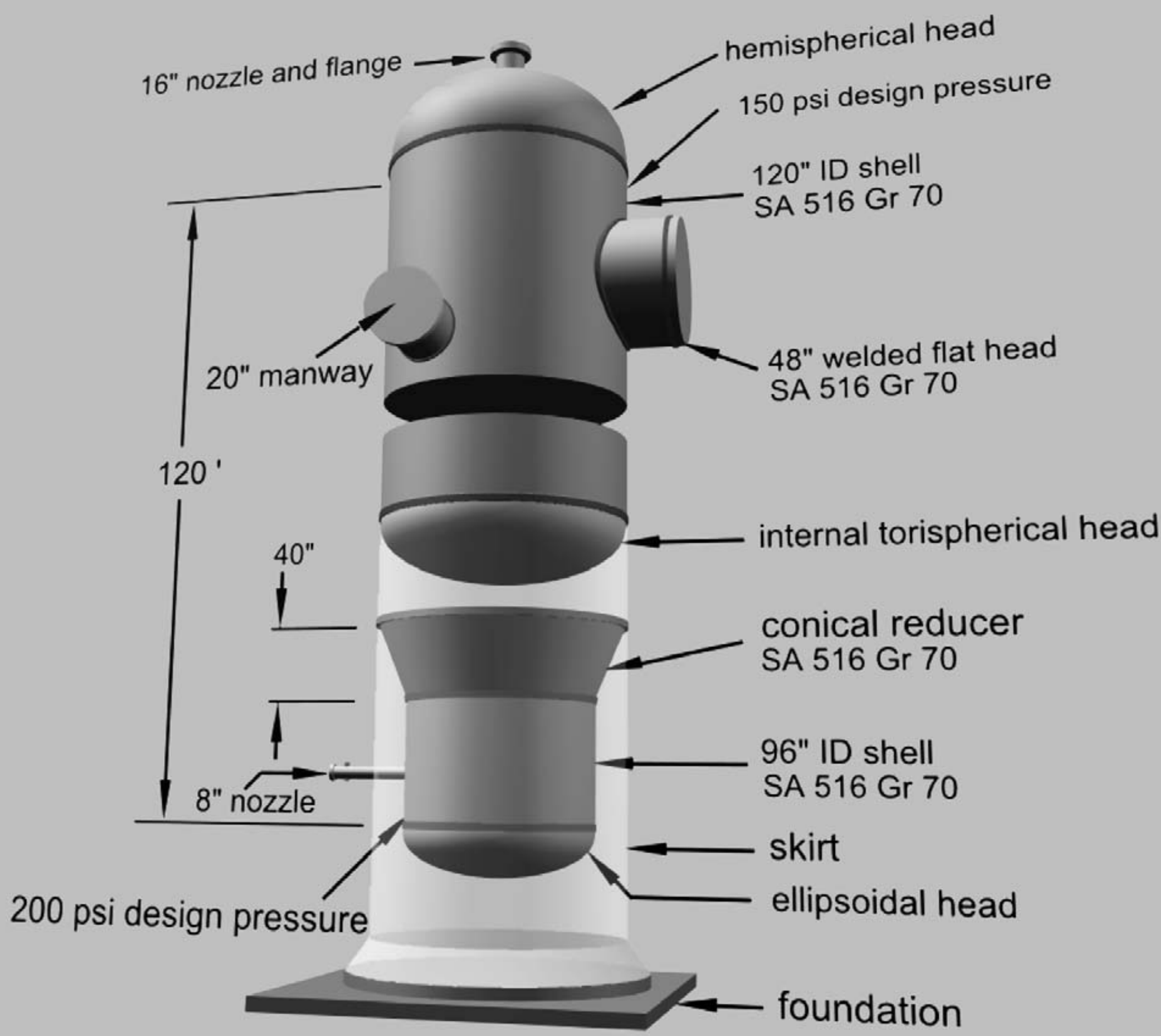


Figure 7.3 Example vessel.

Circumferential Stress (longitudinal joint)

To accurately reflect the corroded condition, the inside radius R in the stress formulas must be changed to R + CA as follows.

(a) Inside radius corrected for the corrosion allowance.

$$t = \frac{P(R + CA)}{SE - 0.6P}$$

where: P = design pressure, UG–21
R + CA = inside radius plus corrosion allowance, UG–16 and UG–25
S = maximum allowable stress from Section II, Part D for SA-516, Grade 70 @ 800°F. (Observe notes.)
E = joint efficiency for a Type 1, fully radiographed longitudinal joint in a shell. (from Table UW-12)

$$t = \frac{150\text{ psi}\left(\dfrac{120\text{ in.}}{2} + 0.125\text{ in.}\right)}{12{,}000\text{ psi}\,(1) - 0.6\,(150\text{ psi})}$$

t = 0.757 inch

t_{req} = t + CA = 0.757 in. + 0.125 in. = 0.882 inch

Therefore, the SA–516 plate must be 0.882 inch thick or thicker.

(b) Solution without corrosion allowance correction.

$$t = \frac{PR}{SE - 0.6P}$$

$$t = \frac{150\text{ psi}\left(\dfrac{120\text{ in.}}{2}\right)}{(12{,}000\text{ psi})(1) - 0.6\,(150\text{ psi})}$$

t = 0.756 inch

t_{req}= 0.756 in. + 0.125 in. = 0.881 inch

This variation requires that the plate be 0.881 inch thick or thicker. The thickness of 0.756 inch is required to resist the pressure. The designer would most likely select a 1 inch thick plate for the construction. The following items must be checked and verified before final selection:

- The post weld heat treatment and preheat requirements.

  UCS–56, Table 56 note (2): Post weld heat treatment or preheat required if thickness is greater than 1¼ inch nominal thickness. Since 1 inch is less than 1¼ inch, preheat and post weld heat treatment is not required.

- The radiography requirements require checking.

  UCS–57, Table 57 requires that P-Number–1, Group 2 materials be fully radiographed if the butt joint thickness is greater than 1¼ inch. Since 1 inch is less than 1¼ inch, spot radiography would be sufficient and full radiography is not required. The E value, however, is less if spot radiography is used and the thickness calculations would have to be redone.

UG–16 states that all Code formulas are in the corroded condition. Since the inside radius of a corroded vessel is greater than the inside radius of the vessel in the uncorroded condition, the uncorroded radius must be increased to reflect the corrosion allowance. While the correction is necessary to satisfy UG–16, Example 7.1 shows that it has a negligible effect on the results. Since this is true in all but rare cases, the correction will be ignored in future examples.

Longitudinal Stress (circumferential joint)

Use Formula 7.4 which is Formula 2 from UG-27 of the Code. P is less than 1.25SE and t is less than $^{R}/_{2}$.

$$t = \frac{PR}{2SE + 0.4P}$$

where: P = design pressure, UG–21
R = inside radius
S = maximum allowable stress from Section II, Part D for SA-516, Grade 70 @ 800°F. (Observe notes.)
E = joint efficiency for a Type 1, fully radiographed circumferential joint in a shell. (from Table UW-12)

$$t = \frac{150\text{ psi}\,(60\text{ in.})}{2\,(12{,}000\text{ psi})\,(1) + 0.4\,(150\text{ psi})}$$

t = 0.375 inch

t_{req} = 0.375 in. + 0.125 in. = 0.500 inch

The girth or circumferential joints must be 0.375 inch thick or thicker in order to adequately resist the pressure. Note that this thickness is ½ the required thickness for the longitudinal joints.

Example 7.2 Maximum Allowable Working Pressure

The fabricator elected to use 1 inch plate to satisfy the design requirements in Example 7.1. The density of the fluid is negligible. Determine the maximum allowable working pressure of the cylindrical section.

Solution: The thickness to be used in this example must be the thickness that resists the pressure. Therefore:

$$t = t_{nominal} - CA$$

$$t = 1.00 \text{ in.} - 0.125 \text{ in.}$$

$$t = 0.875 \text{ inch}$$

Use the second part of formula 1 in UG-27. P is less than 0.385SE and t is less than $^R/_2$. The radius, allowable stress, and efficiency are taken from Example 7.1.

$$P = \frac{SEt}{R + 0.6t}$$

$$P = \frac{(12{,}000\,\text{psi})(1.0)(0.875\,\text{in.})}{60\,\text{in.} + 0.6(0.875\,\text{in.})}$$

$$P = 173.5 \text{ psi}$$

As you can see, the maximum allowable working pressure (MAWP) for the cylindrical section is 23.5 psi greater than the design pressure. The greater pressure is due to the excess thickness available in the plate. Note that if this vessel operated under a static head, then the design pressure and MAWP must be corrected for the static head. This correction for the design pressure is shown in Example 7.3.

Example 7.3 Thickness of a Cylindrical Shell Considering Internal Pressure and Bending

A horizontal pressure vessel, 60 inch inside diameter, is to be fabricated from SA–516 Grade 70 material. The design pressure at the top of the vessel is 490 psi at 600°F. All longitudinal joints shall be Type 1 and spot radiographed in accordance with UW–52. Circumferential joints are Type 1 with no radiography other than the spot radiography overlap for the longitudinal joints. The vessel operates full of liquid. The density of the liquid is 62.4 pounds per cubic foot and the distance from the center line to the uppermost part of the vessel is 5 feet. Determine the required thickness at point A. Neglect the weight of the vessel.

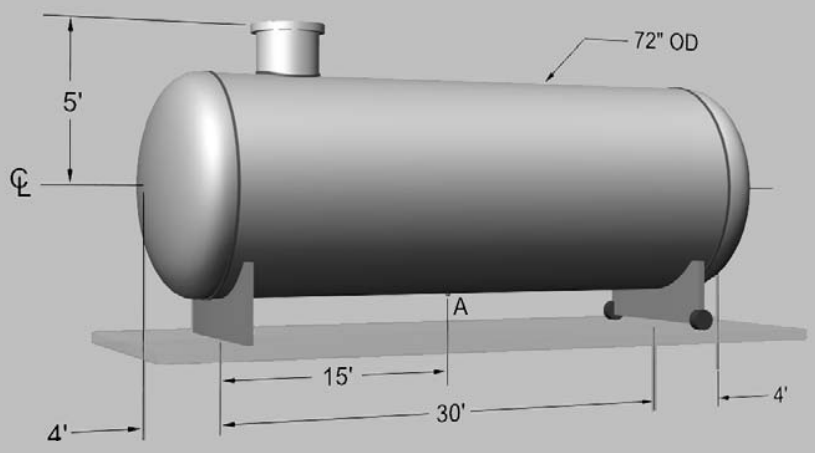


Figure 7.4 Vessel in Example 7.3.

Solution: As stated in UG–22, the static head of the liquid must be included in P. The design pressure is less than 0.385SE and t is less than $R/2$. The allowable stress of SA–516 Grade 70 at 600°F is 19,400 psi as indicated in Table 1D of Section II, Part D. Column B of Table UW–12 gives an E of 0.85 for Type 1 spot radiographed joints. No corrosion allowance is given.

Calculate the thickness of longitudinal joints based on pressure, using Formula 1 of UG-27 [Formula 7.3].

$$t = \frac{PR}{SE - 0.6P}$$

From the sketch, the inside radius R equals 36 inches.

P is the total pressure acting on the section and must equal 490 psi plus the static head created by eight feet (the total height above the section) of liquid.

$$P = 490\,\text{psi} + \frac{(8.0\,\text{ft})\left(62.4\,\text{lb}/\text{ft}^3\right)}{144\,\text{in}^2/\text{ft}^2}$$

$$P = 493.5\text{ psi}$$

Therefore:

$$t = \frac{493.5\text{ psi}\,(36\text{ in.})}{19{,}400\text{ psi}\,(0.85) - 0.6\,(493.5\text{ psi})}$$

$$t = 1.10\text{ inch}$$

Check the stress in the longitudinal direction.

The stress in the longitudinal direction is the sum of the stress due to internal pressure and the bending stress. Formula 2 of UG–27 may be used for the contribution due to pressure. The bending stress may be computed using the basic formula for flexural stress in a beam. When one considers the joint efficiency, the resulting formula is:

$$S_L = \frac{MR_o}{IE} + \frac{P(R - 0.4t)}{2Et}$$

where R_o is the outside radius.

Summation of moments at A gives a bending moment of 183,900 foot-pounds. The moment of inertia, I, for a thin wall cylinder is $\pi r^3 t$ where r is the mean radius of the shell.

For Type 1 joints with no radiography, E = 0.7. Therefore, the stress in the longitudinal direction is:

$$S_L = \frac{(183{,}900\text{ ft - lb})\,(12\text{ in.}/\text{ft})\,(37.2\text{ in.})}{(\pi)\,(36.6\text{ in.})^3\,(1.10\text{ in.})\,(0.7)} + \frac{493.5\text{ psi}\,(36\text{ in.} - 0.4\,(1.10\text{ in.}))}{(2)\,(0.7)\,(1.10\text{ in.})}$$

$$S_L = 693\text{ psi} + 11{,}395\text{ psi} = 12{,}088\text{ psi}$$

Check post weld heat treatment and preheat requirements.

Assume the nominal thickness of the plate is 1.25 inches.

- UCS–56, Table 56 note (2): Post weld heat treatment or preheat is required if thickness is greater than 1¼ inch nominal thickness. Since 1.25 inch is equal to 1¼ inch, preheat and post weld heat treatment is not required.

Check radiography requirement.

- UCS–57, Table 57 requires that P-Number 1 Group 2 materials be fully radiographed if the butt joint thickness is greater than 1¼ inch. Since 1.25 inch is equal to 1¼ inch, spot radiography is sufficient and full radiography is not required.

This example illustrates that the longitudinal stress is small even when other loads are considered and a lower efficiency is used for the circumferential joints.

UG–28 Thickness of Shells and Tubes Under External Pressure

As observed in the beginning of this chapter, tensile stresses occur when a thin wall vessel is subjected to an internal pressure. Fracture is controlled by limiting the tensile stress to some fraction of the yield or tensile strength of the material. When the same vessel is subjected to an external pressure, the primary stress becomes compressive, and buckling instability can result at a stress level below the yield strength of the material.

The buckling limit depends on the stress-strain curve of the material and the geometry of the vessel. In fact, if the vessel falls outside the realm of a thin wall vessel, failure may occur by yielding of the material.

The Code rules for designing cylindrical shells with or without stiffening rings, tubes, and spherical shells for external pressure are given paragraph UG–28. The procedures for determining the minimum thickness are based on a geometric chart, Fig. G (Figure 7.5), and a series of material charts, similar to Fig. CS–2 (Figure 7.6), all of which are given in Subpart 3 of Section II, Part D. The tabular data for the charts are also given.

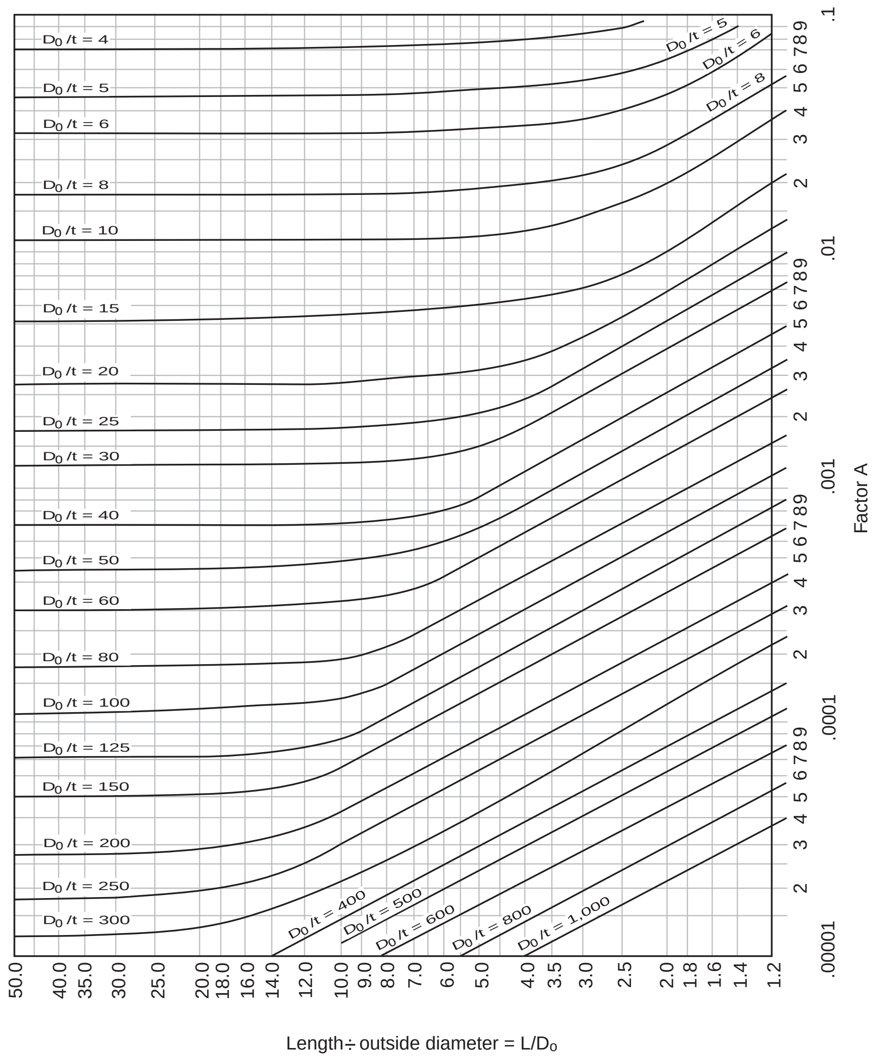


Figure 7.5 Geometric chart for components under external or compressive loadings (for all materials). Fig. G of Section II, Part D.

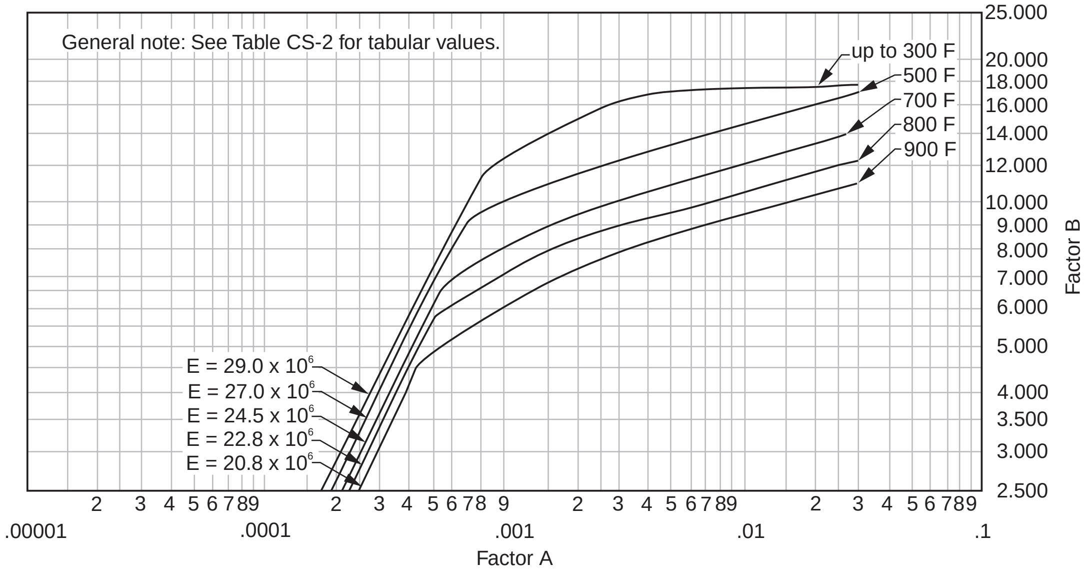


Figure 7.6 Chart for determining shell thickness of components under external pressure when constructed of carbon or low alloy steels (specified minimum yield strength 30,000 psi and over except for materials within this range where other specific charts are referenced) and type 405 and type 410 stainless steels. Fig. CS–2 of Section II, Part D.

The procedure in UG–28 for determining the maximum allowable external working pressure (MAEWP) of a cylinder is given in two parts: the first part is for cylinders having D_o/t values ≥ 10 and the second part is for cylinders having D_o/t values < 10. In each part, an allowable external pressure is calculated based on a factor "B" which depends upon the material, temperature, and geometry of the vessel.

When a vessel is subjected to an external pressure, the pressure boundary is established by the outside surface dimension D_o. The ratio D_o/t, where t is the thickness, is a measure of the stiffness of the vessel in the circumferential direction. The stiffness in the longitudinal direction is represented with the L/D_o ratio. The relationship between these geometric ratios is represented by the factor A.

Cylinders with $D_o/t \geq 10$

The steps for determining the maximum allowable external working pressure for a cylinder with D_o/t greater than or equal to 10 are given in Figure 7.7 The symbols are defined as follows:

A = a geometric factor either determined by formula or from Figure G in Subpart 3 of Section II, Part D.
B = a pseudostress factor determined from the applicable material chart at the design metal temperature, psi.
D_o = outside diameter, inches.
E = modulus of elasticity of material at temperature. Value must be taken from the applicable material chart, psi.
L = unsupported length of vessel section, Fig. UG–28 (Figure 7.8) and Fig. UG–28.1, or length of tube between tubesheets, inches.
P = external design pressure, psi.
P_a = maximum allowable external working pressure for value of t, psi.
R_o = outside radius, inches.
t = minimum required thickness, inches.
t_s = nominal thickness, inches.

Step 1. To determine the D_o/t required in the fist step, the value of t must be available. This initial t is often the thickness required to satisfy internal pressure requirements (UG–27). However, if that thickness is unavailable, then t must be assumed. L, the unsupported length, must also be assumed if stiffeners are to be provided.
Step 2a. The values of D_o/t and L/D_o are used in Fig. G to determine A. L/D_o equal to 50 is the upper limit of Fig. G. This step compares the calculated value of L/D_o to the upper limit and if it is greater than the limit, then the value of L/Do is set to 50.
Step 2b. 0.050 is the L/D_o lower limit in Fig. G. This step tests the calculated value for the lower limit and, if reached, sets the calculated L/D_o to the lower limit.
Step 3. The factor A is read from Fig. G in this step. Interpolation is allowed for values of Do/t that do not fall on the designated curves.
Step 4. The applicable external pressure chart is entered with the value of A. Interpolation is allowed for intermediate temperatures. Tables 1A and 1B, the allowable stress tables, of Section II, Part D list the applicable external pressure chart for each material.
Step 5. Go to step 7 if the value of A falls to the left of the temperature line. If the value of A falls to the right of the end of the temperature line, assume an intersection with the horizontal projection of the upper limit of the line.
Step 6. The maximum allowable external pressure may be calculated with the value B and the given formula.
Step 7. Use this formula if B cannot be read from the chart because A falls to the left of the curves. Note that E in this formula is the modulus of elasticity at temperature and must be taken from the applicable external pressure chart.

If the maximum allowable external pressure calculated in steps 6 or 7 is less than the required external pressure P, then the value of t must be increased or the value of L decreased and the procedure repeated until the value of the allowable external pressure is equal to or larger than P.

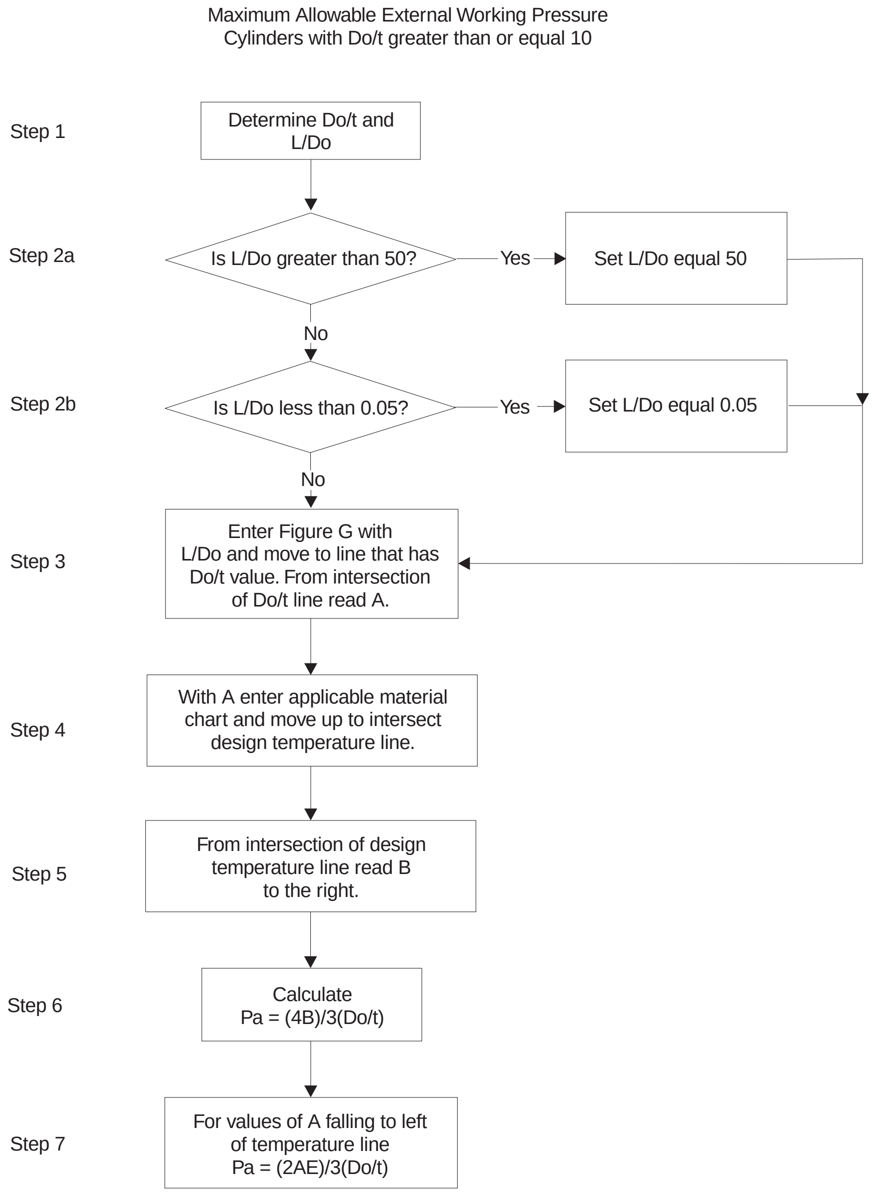


Figure 7.7 Fig. UG–28 of ASME Section VIII, Division 1.

Cylinders with $D_o/t < 10$

The steps for determining the maximum allowable external working pressure for a cylinder with D_o/t less than 10 are given in Figure 7.9. The symbols are the same as those for a cylinder with D_o/t greater than or equal to 10. As the D_o/t value decreases, the possibility of buckling decreases and the possibility of the material yielding increases. Therefore, the Code procedures for this range of D_o/t require that both buckling instability and yielding be checked. A description of the steps in Figure 7.9 that illustrates how to check for these parameters follows.

Step 1. Determine the D_o/t and L/D_o ratios. t may be the required thickness or assumed for internal pressure.

Step 2. Check to determine if D_o/t is less than the Figure G lower limit of 4. If it is, then use the formula for A. If the calculated A is greater than the upper limit of 0.10, then set A equal 0.10.

Step 3. If D_o/t is equal to or greater than 4, then use steps 1 to 4 of Figure 7.7 to determine A.

Step 4. The applicable external pressure chart is entered with the value of A. Interpolation is allowed for intermediate temperatures. Tables 1A and 1B, the allowable stress tables, of Section II, Part D list the applicable external pressure chart for each material.

Step 5. If the value of A falls to the right of the end of the temperature line, assume an intersection with the horizontal projection of the upper limit of the line.

Step 6. Calculate P_{a1}, a maximum allowable external pressure with the factor B.

Step 7. For thick cylinders, the possibility that the material will be overstressed before buckling is great. Therefore, a maximum allowable external pressure based on the smaller of the allowable tensile stress and yield strength must be calculated. The allowable tensile stress at temperature is taken from Tables 1A or 1B of Section II, Part D.

Step 8. B* equals the value of B obtained by projecting horizontally the upper limit of the design temperature line.

Step 9. S is the smallest value of S1 and S2.

Step 10. Calculate P_{a2}, a maximum allowable external pressure based on the allowable tensile stress and yield strength.

Step 11. The maximum allowable external pressure for the vessel is the smaller of P_{a1} or P_{a2}.

As with the procedure for $D_o/t \geq 10$, this procedure must be repeated with a larger value of t or smaller L if the allowable external pressure is less than the required design external pressure.

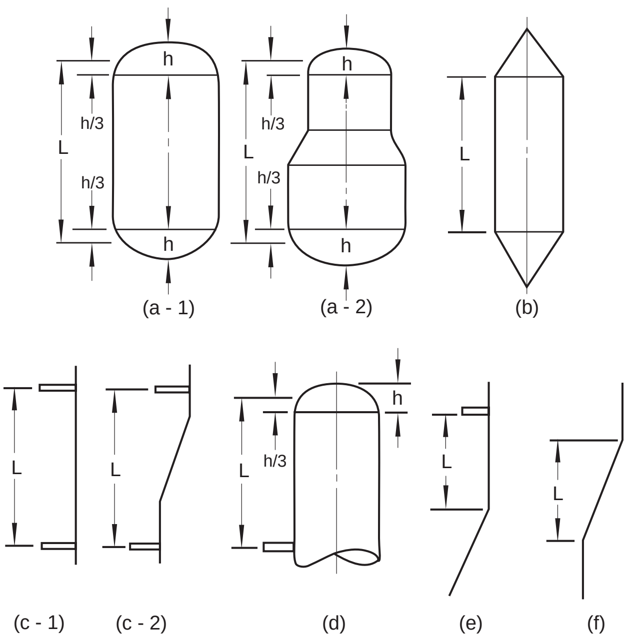


Figure 7.8 Unsupported length of vessels subjected to external pressure. (Fig. UG-28.1)

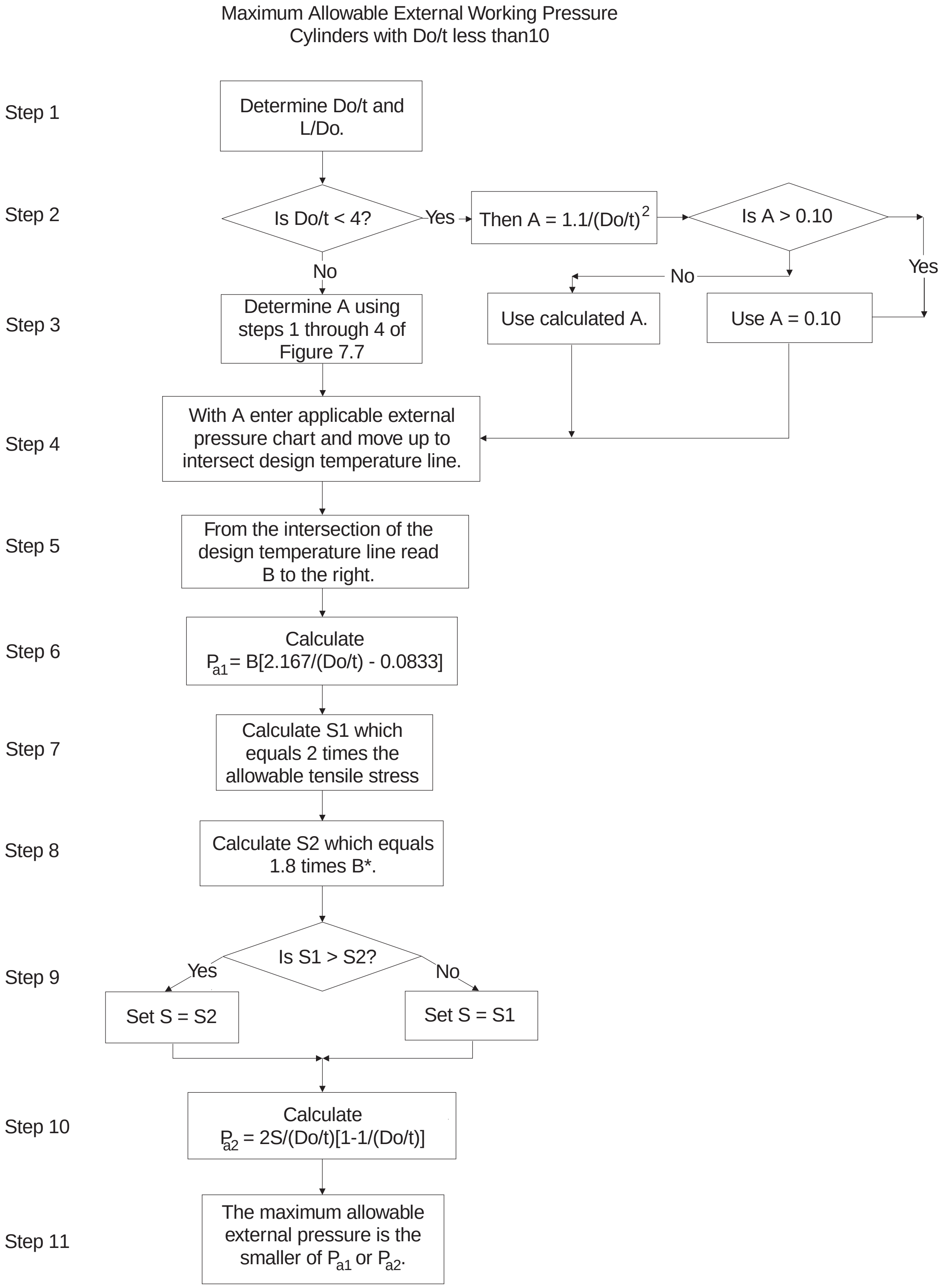


Figure 7.9 Calculation of external pressure for cylinders with D_0/t less than 10.

Spherical Shells

The procedure for calculating the minimum required thickness for a sphere under external pressure is similar to that for cylinders. A geometric factor A is calculated and a factor B is read from the applicable external pressure chart.

Apply the following steps to determine the minimum required thickness of a spherical shell under external pressure:

1. Assume a thickness, t, and calculate A where:

   $A = \frac{0.125}{(R_o/t)}$ and Ro is the outside radius.

2. Using the value of A, enter the applicable material chart of Section II, Part D and move vertically to an intersection with the appropriate temperature line. If the value of A falls to the right of the material/temperature line, extend the line to the right and read B at the intersection of the line and the vertical axis. See step 5 for values of A falling to the left of the material/temperature line.

3. From the intersection of A and the material/temperature line, move horizontally to the right and read the value of B on the vertical axis.

4. Using the value of B, calculate the maximum allowable external working pressure P_a where: $P_a = \frac{B}{(R_o/t)}$.

5. For values of A falling to the left of the material/temperature line, P_a is calculated with $P_a = \frac{0.0625E}{(R_o/t)^2}$ where E is the modulus of elasticity at temperature taken from the external pressure material chart in Section II, Part D.

If the value of P_a is smaller than the required design value, then increase t and repeat the process until the desired value is obtained.

General Requirements

UG–28 requires the maximum allowable external working pressure to be not less than the maximum expected difference in operating pressure between the outside and inside of the vessel at any time. If the vessel is a cylinder and has a longitudinal lap joint, or if the vessel is a sphere and has a lap joint, then the external thickness calculations shall be based on 2P instead of P.

Stiffeners or other means of support may be used to prevent overstress or large distortions. Pressure chambers subjected to external pressure with shapes other than cylinders, spheres, or formed heads must have staybolts or be proof tested in accordance with UG–101(p).

Example 7.4 Cylindrical Shell Thickness for External Pressure

A vessel has a 96 inch inside diameter and Type 1 butt welds. No corrosion allowance is required. The design temperature is 800°F and the internal design pressure is 150 psi. The external design pressure is 15 psi. The material is SA–516 Grade 70 and full radiography will be performed. The unsupported length of the section L, including the contribution from the head, is 360 inches. Find the minimum required thickness and the maximum allowable external pressure of the section.

Solution: The allowable stress in Part D of Section II, Table 1A, for SA–516 Grade 70 at 800°F is 12,000 psi. Table UW–12 gives a joint efficiency of 1.0 for fully radiographed Type 1 joints.

A) Determine the thickness, t, required for internal pressure. Use Formula 1 of UG–27 or 7.3.

$$t = \frac{PR}{SE - 0.6P}$$

$$t = \frac{150\text{ psi}\left(96\text{ in.}/2\right)}{12{,}000\text{ psi}(1) - 0.6(150\text{ psi})} = 0.604\text{ inch}$$

$t < R/2$ and $P < 0.385SE$. Therefore, 0.604 inch is the minimum required thickness for internal pressure.

B) Determine the maximum allowable external pressure for the section with t equal 0.604 inch.

Calculate ratios L/D_o and D_o/t where D_o is the outside diameter and t is 0.604 inch.

D_o = 96 in. + 2 x t

D_o = 97.21 inch

D_o/t = 97.21 in./0.604 in.

D_o/t = 160.9. D_o/t is greater than 10. Therefore, use procedure in UG–28(c)(1) which is diagrammed in Figure 7.8.

L/D_o = 360 in./97.21 in.

L/D_o = 3.70

From Figure G of Section II, Part D, with L/D_o and D_o/t

A = 0.00017

Table 1A of Section II, Part D lists External Pressure Chart No. CS-2 for SA–516 Grade 70 material. Chart CS-2 is shown in Figure 7.6. Entering Chart CS–2, A = 0.00017 falls to the left of the temperature line. Therefore the allowable pressure is calculated using the formula given in Step 7 of Figure 7.7. Chart CS–2 gives a value of 22,800,000 psi for E, the modulus of elasticity at 800°F.

$$P_a = \frac{2\,(0.00017)\,(22{,}800{,}000\text{ psi})}{3\,(160.9)} = 16.1\text{ psi}$$

P_a is greater than the external design pressure. Therefore the minimum thickness of 0.604 inch is acceptable.

C) Check the above external pressure results using the Section II, Part D tabular data for Figure G and Chart CS-2.

From Figure G tabular data for D_o/t = 150 and L/D_o = 2, Factor A = 0.000349. Also, for D_o/t = 150 and L/D_o = 4, Factor A = 0.000168. Interpolation is permitted, so A = 0.000195 for D_o/t = 150 and L/D_o = 3.7.

For D_o/t = 200 and L/D_o = 2, Factor A = 0.000227. For D_o/t = 200 and L/D_o = 4, Factor A = 0.000110. Interpolation gives A = 0.000128 for D_o/t = 200 and L/D_o = 3.7.

Interpolation between the sets of D_o/t = 150 and, Factor A = 0.000195 and D_o/t = 200, Factor A = 0.000128 for D_o/t = 160.9, gives A = 0.00018.

The Chart CS-2 tabular values at 800°F for factors A and B are:

A	**B**
0.00000223	250
0.000505	5710

Interpolation is permitted, therefore,

$$B = 250 + \frac{(5710 - 250)\,(0.00018 - 0.0000223)}{(0.000505 - 0.0000223)} = 2{,}034\text{ psi}$$

With B, the equation in Step 6 of Figure 7.7 may be used to calculate the maximum allowable external working pressure.

$$P_a = \frac{4\,(2034\text{ psi})}{3\,(160.9)} = 16.9\text{ psi}$$

As in part B above, this value of P_a is greater than the required value of 15 psi. Therefore the thickness 0.604 inch is acceptable.

One should note that the charts, figures, and tabular data may not provide identical results but will give values reasonably close to each other. If the maximum allowable pressure obtained is less than that required, then the designer must increase the thickness or reduce the unsupported length L and repeat the process. Stiffeners may be required to reduce L.

Example 7.5 Minimum Thickness for a Spherical Shell under External Pressure

A butt welded spherical vessel having an inside diameter of 120 inches is made of SA–516 Grade 60 material. The vessel must be designed to withstand an external design pressure of 30 psi at 400°F. What is the minimum thickness?

Solution: SA–516 Grade 60 is checked in Section II, Part D and found to be on External Pressure Chart CS–2. Assume a thickness and calculate factor A.

Assume t = 1.0 inch

$$R_o = \frac{120\,\text{in.}}{2} + 1.0\,\text{in.}$$

R_o = 61.0 inch

$$A = \frac{(0.125)(1.0\,\text{in.})}{61\,\text{in.}} = 0.0020$$

Enter Figure CS–2 of Section II, Part D and move vertically from 0.002 up to the intersection of the 500°F and 300°F temperature lines. Move horizontally to the right and read the B values for each temperature.

B_{500} = 11,500 psi @ 500°F

B_{300} = 15,000 psi @ 300°F

$$B = \frac{(11{,}500 + 15{,}000)}{2}$$

$B = 13{,}250\,\text{psi}$ @ 400°F

Calculate the maximum allowable external pressure using the formula for spherical shells.

$$P_a = \frac{B}{(R_o/t)}$$

$$P_a = \frac{13{,}250\,\text{psi}}{61\,\text{in.}/1\,\text{in.}} = 217\text{ psi}$$

The calculated maximum allowable external pressure of 217 psi is much greater than the required 30 psi. Therefore assume a thinner section and redo calculations.

Assume t = 0.25 inch

$$R_o = \frac{120\,\text{in.}}{2} + 0.25\,\text{in.} = 60.25 \text{ inch}$$

$$A = \frac{(0.125)(0.25\,\text{in.})}{60.25\,\text{in.}} = 0.00052$$

From Figure CS–2, B = 7,500 psi. Therefore:

$$P_a = \frac{7{,}500\,\text{psi}\,(0.25\,\text{in.})}{60.25\,\text{in.}} = 31 \text{ psi}$$

The calculated maximum allowable external pressure of 31 psi is just slightly greater than the required value of 30 psi. Therefore, the minimum required thickness is 0.25 inch.

UG–29 Stiffening Rings for Cylindrical Shells Under External Pressure

The Code provides two methods for sizing stiffening rings. Method (a) is based on the stiffening ring providing all additional stiffening, and method (b) is based on a combination of the stiffening ring and a part of the shell providing the additional stiffening. Both methods require that one assume an initial size and shape for the ring.

(a) Stiffening ring alone:

1) Determine B, where:

$$B = 0.75\left(\frac{PD_o}{t + A_S/L_S}\right)$$

A_S = cross-sectional area of stiffening ring, square inches.
L_S = distance between support lines on both sides of the stiffening ring, inches.
D_o = outside diameter of shell, inches.

2) Enter appropriate external pressure chart in Section II, Part D and determine Factor A that corresponds to the calculated Factor B.

3) Determine the required moment of inertia of the stiffening ring only, I_S:

$$I_S = \frac{\left[D_o^2 L_S\left(t + A_S/L_S\right)A\right]}{14}$$

4) Determine actual moment of inertia of ring only, I, in.^{4}

5) I must be equal to or greater than I_S

(b) The stiffening ring/shell combination:

1) Determine B, where:

$$B = 0.75\left(\frac{PD_o}{t + A_S/L_S}\right)$$

A_S = cross-sectional area of stiffening ring, square inches.
L_S = distance between support lines on both sides of the stiffening ring, inches.
D_o = outside diameter of shell, inches.

2) Enter appropriate external pressure chart in Section II, Part D and determine Factor A that corresponds to the calculated Factor B.

3) Determine the required moment of inertia of the ring/shell combination, I_S', in.^{4}

$$I_S = \frac{\left[D_o^2 L_S \left(t + {}^{A_S}\!/_{L_S}\right) A\right]}{14}$$

$$I_S' = \frac{\left[D_o^2 L_S \left(t + {}^{A_S}\!/_{L_S}\right) A\right]}{10.9}$$

4) Determine the moment of inertia of the combined ring and shell section acting together, I', inch^4. The length of shell used in the calculation shall not be greater than $1.10\ (D_o t)^{1/2}$. No overlap of contribution is allowed.

5) I' shall be equal to or greater than I_S'.

Note that the formulas for B are identical. If the value of B falls below the left end of the material/temperature curve, then A is calculated using A = 2B/E where E is the modulus of elasticity. If different materials are used for the shell and stiffening ring, then use the external pressure chart that gives the lowest value of A.

General Requirements

Stiffening rings must extend completely around the circumference of the cylinder. All joints and connections between rings must maintain the required moment of inertia of the combined ring-shell section. Inside and outside rings are allowed and stiffening rings may be continued on the opposite surface. See Figure 7.10**.**

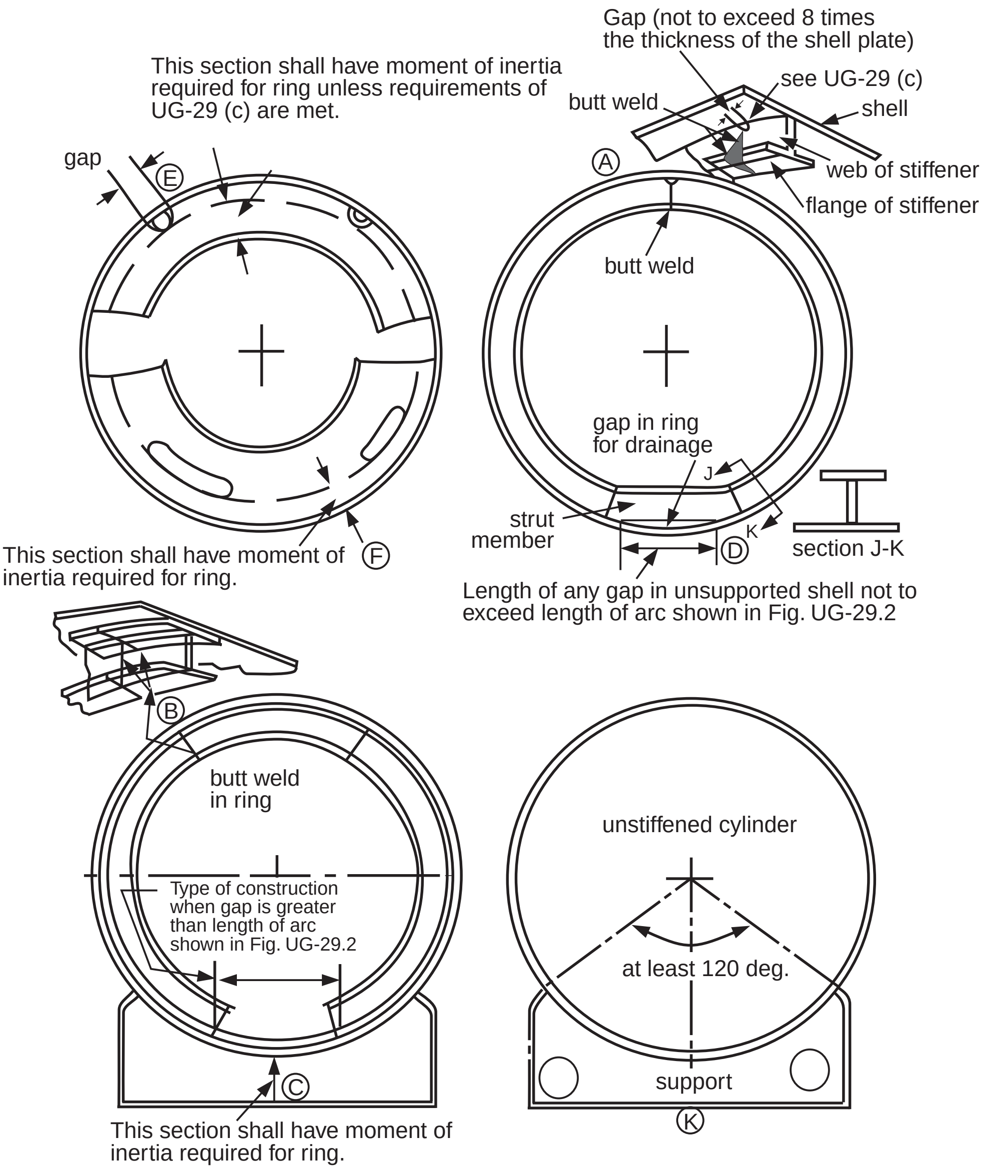


Figure 7.10 Various arrangements of stiffening rings for cylindrical vessels subjected to external pressure. Fig. UG–29.1.

Limited gaps in stiffening rings adjacent to the shell are allowed. The allowed gap length depends on L/D_o and D_o/t and is given by Figure 7.11. Gaps greater than the allowed length must be reinforced on the opposite surface or comply with additional requirements given in UG–29. The additional requirements are:

1. limit the unsupported shell arc to 90 degrees,
2. stagger unsupported shell arcs in adjacent stiffening rings, and
3. increase the unsupported shell length L to the distance between alternate rings or at heads to the second ring from the head.

Baffles, trays, tray supports, or other internal structures perpendicular to the axis of the longitudinal axis and welded to the cylinder may function as stiffeners provided they satisfy all the requirements in paragraphs UG–29 and UG–30.

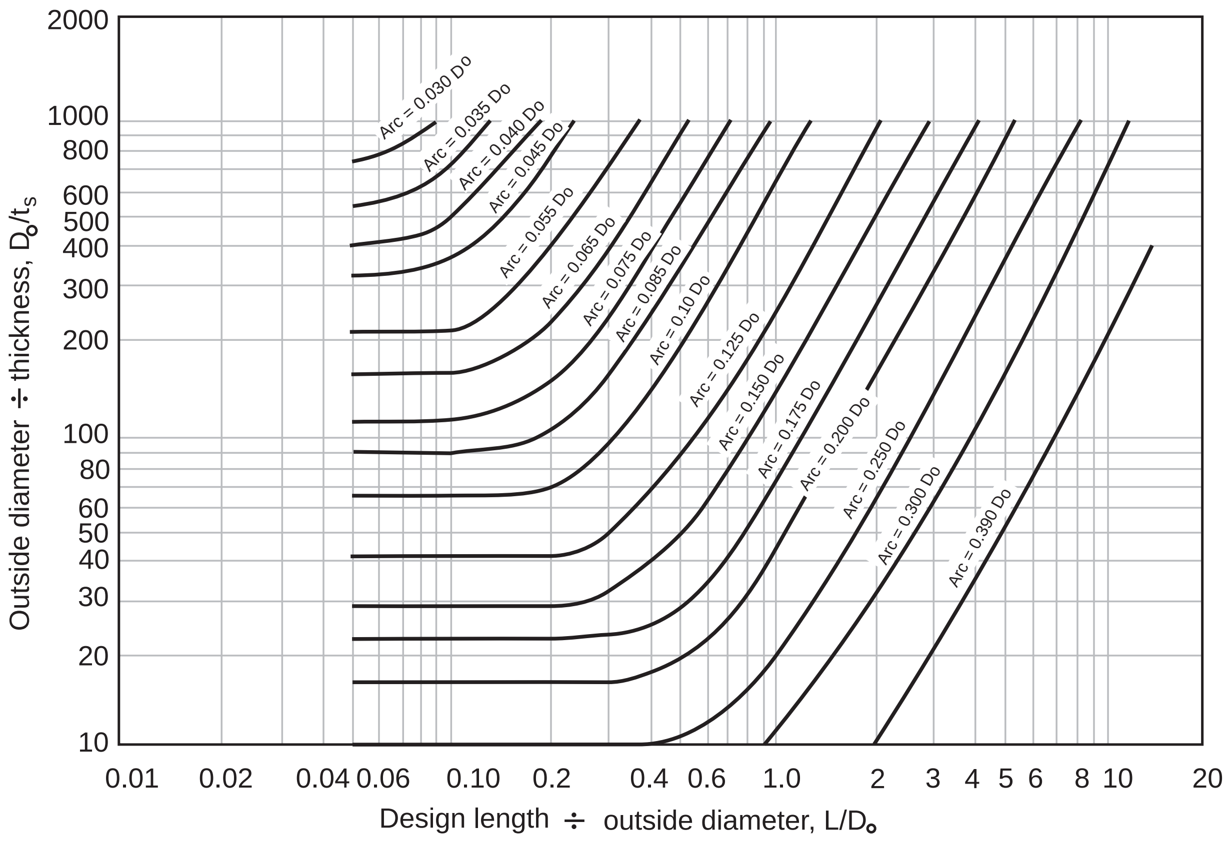


Figure 7.11 Maximum arc of shell left unsupported because of gap in stiffening ring of cylindrical shell under external pressure. Fig. UG–29.2.

UG–30 Attachment of Stiffening Rings

To insure that stiffening rings are effective, UG–30 provides minimum welding requirements. Stiffeners may be welded or brazed to the shell. The rings may be attached with continuous, intermittent, or a combination of continuous and intermittent welds or brazes. See Figure 7.12 for some acceptable details.

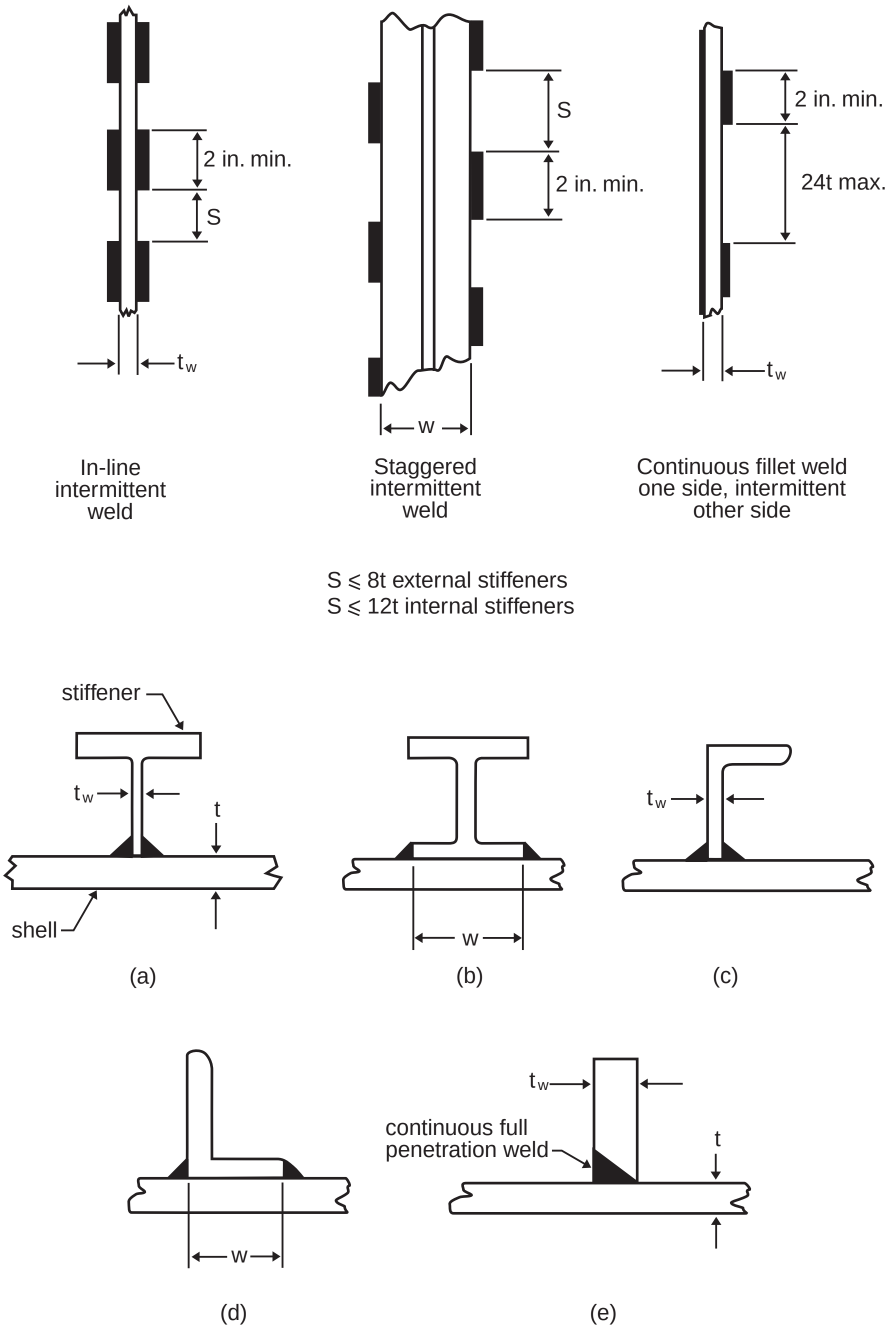


Figure 7.12 Some acceptable methods of attaching stiffening rings. Fig. UG–30.

Example 7.6 Minimum Thickness of a Thick Cylinder Under External Pressure

A forged 7.5 inch inside diameter PI vessel is exposed to an external design pressure of 1,700 psi at 700°F. The material is SA-182 Type F321. The unsupported length is 18 feet. Find the minimum required thickness for the vessel.

Solution: Table 1A of Section II, Part D gives an allowable tensile stress of 13,000 psi for SA-182 Type F321 material. The applicable external pressure chart in Section II, part D is HA-2. The solution procedure is shown in Figure 7.9.

A) Calculate D_o/t and L/D_o ratios.

Assume t = 1.0 inch

D_o = 7.5 in. + 2 (1.0 in.)

D_o = 9.5 inch

D_o/t = 9.5 in./1 in.

D_o/t = 9.5

L/D_o = (18 ft) (12 in./ft)/9.5 in.

L/D_o = 22.7

D_o/t is less than 10 but greater than 4. Therefore, use the procedure outlined in Figure 7.9. From Figure G of Section II, Part D, with D_o/t = 9.5 and L/D_o = 22.7, A = 0.014.

Chart HA-2 gives a factor B value of 11,800 psi for A = 0.014 and 700°F.

B) Calculate P_{a1} using the formula in Step 6 of Figure 7.9.

$$P_{a1} = 11{,}800\,\text{psi}\left(\frac{2.167}{9.5} - 0.0833\right)$$

$$P_{a1} = 1{,}708\,\text{psi}$$

C) Calculate P_{a2} using the formula in Step 10 of Figure 7.9.

The allowable tensile stress for the material is 13,000 psi. Therefore:

S1 = 2 (13,000 psi) = 26,000 psi.

B* is one-half the yield strength at temperature and is equal to the maximum value of B for the material at 700°F.

B* = 13,100 psi

S2 = 1.8 (13,100) psi

S2 = 23,580 psi

S is the smaller of S1 and S2 and equals 23,580 psi, therefore,

$$P_{a2} = \frac{2\,(23{,}580\,\text{psi})}{9.5}\left[1 - \frac{1}{9.5}\right]$$

$$P_{a2} = 4{,}442\,\text{psi}$$

D) The maximum allowable external pressure is the smaller of P_{a1} and P_{a2}. Therefore, P_a for the 1 inch thick section is 1,708 psi, which is greater than the required 1,700 psi.

Example 7.7 Design of Shell and Stiffening Rings

The lower cylindrical part of the vessel shown in Figure 7.3 has a 96 inch inside diameter and Type 1 butt welds. The corrosion allowance is 0.125 inch. The design temperature is 800°F and the internal design pressure is 200 psi. The external design pressure is 15 psi. The material is SA–516 Grade 70 and full radiography will be performed. The minimum unsupported length of the section L is 360 inches. Find the minimum required thickness and the maximum allowable external pressure of the section.

Solution: The allowable stress in Part D of Section II, Table 1A, for SA–516 Grade 70 at 800°F is 12,000 psi. Table UW–12 gives a joint efficiency of 1.0 for fully radiographed Type 1 joints.

A) Determine the thickness t required for internal pressure.

$$t = \frac{PR}{SE - 0.6P}$$

$$t = \frac{200\,\text{psi}\,(96\,\text{in.}/2)}{12{,}000\,\text{psi}\,(1) - 0.6\,(200\,\text{psi})}$$

t = 0.81 inch

t < R/2 and P < 0.385SE. Therefore, 0.81 inch is the minimum required thickness for internal pressure.

t_{req} = t + 0.125 in.

t_{req} = 0.81 in. + 0.125 in.

t_{req} = 0.94 inch. Use 1 inch plate.

B) Determine the maximum allowable external pressure for the section with t = 0.81 inch and t_s = 1.0 inch.

Calculate ratios L/D_o and D_o/t where D_o is the outside diameter and t is 0.81 inch.

D_o = 96 in. + 2 x t_s

D_o = 96 in. + 2 (1 in.) = 98.0 inch

D_o/t = 98.0 in./0.81 in.

D_o/t = 121.0. D_o/t is greater than 10. Therefore, use procedure in UG–28(c)(1) which is diagrammed in Figure 7.8

L/D_o = 360 in./98 in.

L/D_o = 3.70

From Figure G of Section II, Part D, with L/D_o and D_o/t

A = 0.00026

Table 1A of Section II, Part D lists External Pressure Chart No. CS-2 for SA–516 Grade 70 material. Chart CS-2 gives a B factor of 3,200 psi for A = 0.00026 at 800°F.

Therefore using the formula in Step 6 of Figure 7.8:

$$P_a = \frac{4\,(3{,}200)\text{ psi}}{3\,(121.0)}$$

P_a = 35.3 psi

P_a is much greater than the required 15 psi. The value may be reduced by decreasing the thickness or increasing L. The thickness is required for internal pressure. Therefore, increase L and redo the procedure.

Assume L = 612 inches

L/D_o = 612 in./98 in.

L/D_o = 6.24

A = 0.00014. A is to the left of the temperature line on Chart CS–2

E = 22,800,000 psi

The allowable external pressure must be calculated using the formula in Step 7 of Figure 7.8.

$$P_a = \frac{2\,(0.00014)\,(22{,}800{,}000\,\text{psi})}{3\,(121.0)}$$

P_a = 17.6 psi

P_a is greater than the required 15 psi. Therefore use the one inch shell plate and space stiffeners at 612 inches.

C) Size the stiffening rings assuming rings act alone. Use the procedure from UG–29.

Determine B. Assume a 1 x 8 inch bar of SA–516 Grade 70 material.

A_S = 8 in.^{2}

L_S = 612 inches

t = 0.810 inch

P = 15 psi

D_O = 98 inches

$$B = 0.75\left(\frac{(15)\,(98)}{(0.81 + 8/612)}\right)$$

B = 1,339 psi

B is to the left of the curve on Figure CS-2. Therefore A must be calculated using the following equation:

$$A = \frac{2B}{E} = \frac{(2)\,(1{,}339\,\text{psi})}{22.8\,\text{x}\,10^6\ \text{psi}} = 0.0001175$$

Determine the required moment of inertia for the stiffener.

$$I_S = \frac{\left[D^2 L\left(t + \frac{A_S}{L}\right)A\right]}{14}$$

$$I_S = \frac{\left[612\,\text{in.}(98\,\text{in.})^2\left(0.81\,\text{in.} + \frac{8\,\text{in.}^2}{612\,\text{in.}}\right)(0.0001175)\right]}{14}$$

I_S = 40.6 inch^4

Determine the actual moment of inertia of the 1 inch x 8 inch bar.

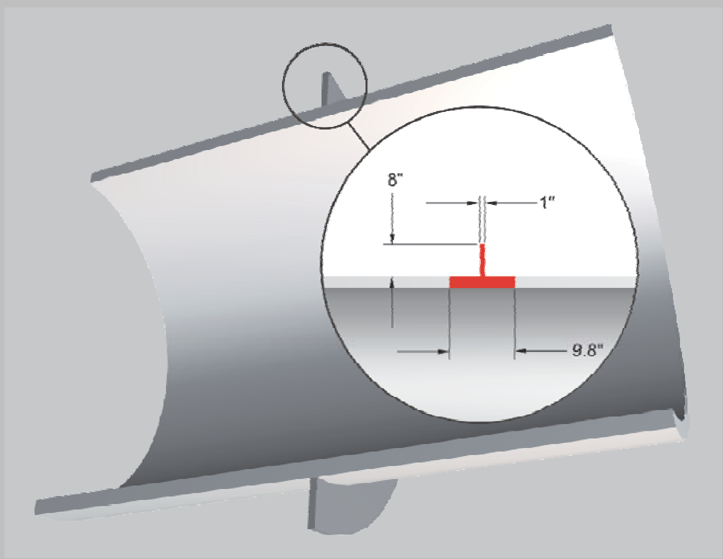


Figure 7.13 Stiffening ring.

$$I = \frac{bd^3}{12} = \frac{(1\,\text{in.})(8\,\text{in.})^3}{12} = 42.7\,\text{inch}^4$$

$I > I_S$. Therefore a 1 inch x 8 inch bar may be used. Attach per Fig. UG-30.

D) Size stiffening ring assuming ring and shell act together.

Note that B does not change. Therefore calculate the required moment of inertia for the combined section assuming a 1 inch x 8 inch bar.

$$I_S = \frac{\left[D^2 L\left(t + \frac{A_S}{L}\right)A\right]}{10.9}$$

$$I_S = \frac{\left[(98\text{ in.})^2\, 612\text{ in.}\left(0.81\text{ in.} + \dfrac{8\text{ in.}^2}{612\text{ in.}}\right)(0.0001175)\right]}{10.9}$$

$I_S = 52.15$ inch^4

Allowed length of shell contribution is 1.1 $(Dt)^{½}$ or 9.8 inches. I for the 1 inch x 8 inch combination section is 120.4 in.^{4}, which is much larger than required I_s. Therefore, assume a 1 inch x 6 inch ring. The I_S for the 6 inch ring combination is 51.94 in.^{4}. The centroid of the composite section (Figure 7.14) consisting of 6 inch ring and shell is 1.871 inches from the ID of the shell.

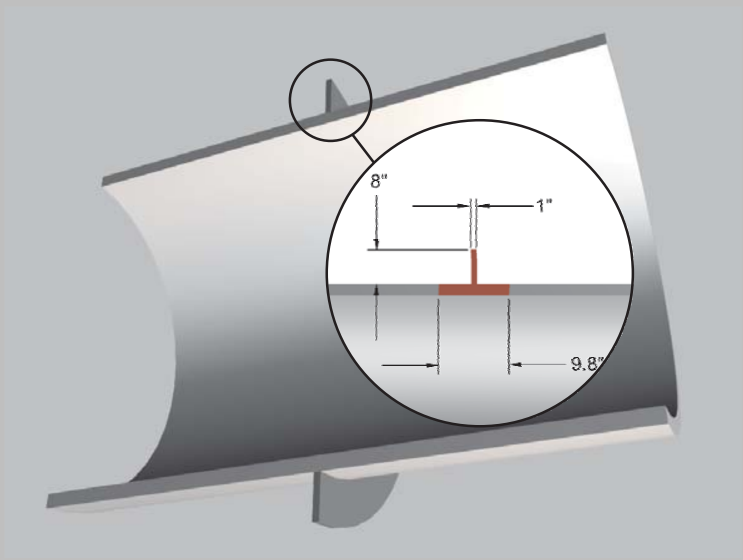


Figure 7.14 6 inch Stiffening ring and portion of shell.

Calculate the actual moment of inertia of the combined 6 inch ring and shell combination.

$$I = (1\text{ in.})(6\text{ in.})^3 \frac{1}{12} + (9.80\text{ in.})(0.81\text{ in.})(1.466\text{ in.})^2 + (6\text{ in.})(1\text{ in.})(1.939\text{ in.})^2$$

$I = 57.62$ inch^4

$I > I_S$ therefore, use a 6 inch x 1 inch ring. Attach per Fig. UG–30.

Note that a smaller ring is required in part D. If the stiffeners are designed as a combination ring and shell section, then paragraph UG–29 does not contain additional requirements other than calculating the moment of inertia of the combined section. The material savings will often outweight the additional design cost.

UG–31 Tubes and Pipe When Used as Tubes or Shells

When designing tubes or shells using tube or pipe, the designer must use the rules in UG–27 for internal pressure and the rules in UG–28 for external pressure. The allowable stress values for both seamless and welded pipe are in Section II, Part D. They are not interchangeable. The values for welded products have been reduced due to the weld. Therefore, the allowable tensile stress for welded products must be taken from the welded product specification line and not from the seamless product line.

When using Formula 1 of paragraph UG–27, paragraph UW–12 stated that the joint efficiency, E=1, when the spot radiography requirements of UW–11(a)(5) are met. Otherwise E = 0.85. No increase in the allowable stress or E is allowed for radiography of the longitudinal joint.

The thickness required to comply with paragraphs UG–27 and UG–28 shall be increased where tube ends are threaded by 0.8/n inch, where n is the number of threads per inch. The calculated thickness may also require an increase due to rolling, expanding, corrosion, erosion, or other wear mechanisms.

The Division does not give dimensions for tubing or standard welded and seamless wrought steel pipe. It does require that the manufacturing undertolerance be considered in the design process. The designer must refer to a handbook or another standard for dimensions.

Example 7.8 Shell Design Using Pipe

The 8 inch nozzle at the bottom of the example vessel in Figure 7.3 is to be fabricated from SA–53 Grade E/A pipe. The corrosion allowance is 0.125 inch and no radiography is performed. The design pressure is 200 psi and the design temperature is 800°F. Select the pipe schedule.

Solution: From Section II, Part D, the allowable stress at 800°F is 7,900 psi. (This allowable stress includes the efficiency value of 0.85. See note G24 in Section II, Part D.) Assuming standard schedule 8 inch pipe, the outside diameter is 8.625 inches and the thickness is 0.322 inch. Using Formula 1 of UG–27 gives:

$$t = \frac{PR}{SE - 0.6P}$$

where,

$$R = \frac{[8.625\,\text{in.} - (2)(0.322\,\text{in.})]}{2} = 3.99\ \text{inches}$$

so:

$$t = \frac{(200\ \text{psi})(3.99\ \text{in.})}{(7{,}900\ \text{psi}) - 0.6\,(200\ \text{psi})}$$

$$t = 0.102\ \text{inch}$$

t_{req} = 0.102 in. + 0.125 in.
t_{req} = 0.227 inch

This value is less than 0.282 inch which is the minimum wall thickness of standard wall 8 inch pipe.

Paragraph UG–16 requires that pipe be corrected for manufacturing tolerance on the thickness of the pipe. Therefore, the minimum wall thickness of 8 inch standard wall pipe is 87.5% of 0.322 inch or 0.282 inch.

Chapter

8

HEADS AND TRANSITION SECTIONS

UG–32 Formed Heads and Sections, Pressure on Concave Side

The most common type of end closure for a cylindrical shell is a formed head. Paragraph UG–32 contains the design requirements for formed heads subjected to internal pressure. There are five types of formed heads: ellipsoidal, torispherical, hemispherical, conical, and toriconical. Conical and toriconical sections are also used as transition sections between shell sections of different diameters.

The required thickness at the thinnest point after forming an ellipsoidal, torispherical, hemispherical, conical, or toriconical section under internal pressure is given by Formulas 8.1, 8.3, 8.5, 8.6 and 8.4, respectively. The symbols in the formulas are:

t = minimum required thickness after forming, inches
P = internal design pressure or maximum allowable working pressure, psi
D = inside diameter or inside length of the major axis of the head, inches
D_i = inside diameter of the conical portion of a toriconical head at its point of tangency to the knuckle, inches
= $D - 2r(1 - \cos \alpha)$
r = inside knuckle radius, inches
S = maximum allowable tensile stress from Section II, Part D, psi
E = lowest efficiency of any point in the head
L = inside spherical or crown radius, inches
α = one-half of the apex angle of the cone at the center line of the head

Formulas using outside dimensions and formulas for heads of other proportions are given in Section 4 of Appendix 1 of ASME Boiler and Pressure Vessel Code, Section VIII, Division 1.

Ellipsoidal Heads

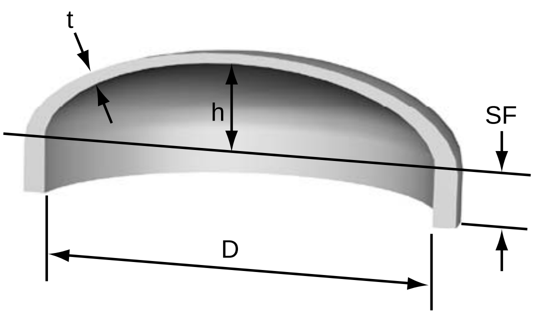


Figure 8.1 Ellipsoidal head.

The required minimum thickness for a 2:1 ellipsoidal head is:

$$t = \frac{PD}{2SE - 0.2P}$$

or

$$P = \frac{2SEt}{D + 0.2t} \tag{8.1}$$

A 2:1 ellipsoidal head has one-half the minor axis, h, equal to one-fourth of the inside diameter of the head skirt, D. SF is the skirt length required by UG–32(l). A 2:1 ellipsoidal head may be approximated with a head containing a knuckle radius of 0.17D and a spherical radius of 0.90D.

Appendix 1–4 gives the following formulas for ellipsoidal heads with D/2h ratios other than 2:1.

$$t = \frac{PKD}{2SE - 0.2P}$$

or

$$P = \frac{2SEt}{KD + 0.2t} \tag{8.2}$$

The K factor is given in Table 1-4.1 of Appendix 1 and depends upon the D/2h ratio of the head. If the D/2h ratio is greater than 2 and the minimum tensile strength of the material is greater than 70,000 psi (483 MPa), then the allowable tensile stress, S, shall equal 20,000 psi (138 MPa) at room temperature, or 20,000 psi (138 MPa) times the ratio of the material's maximum allowable stress at temperature divided by the material's allowable stress at room temperature.

Torispherical Heads

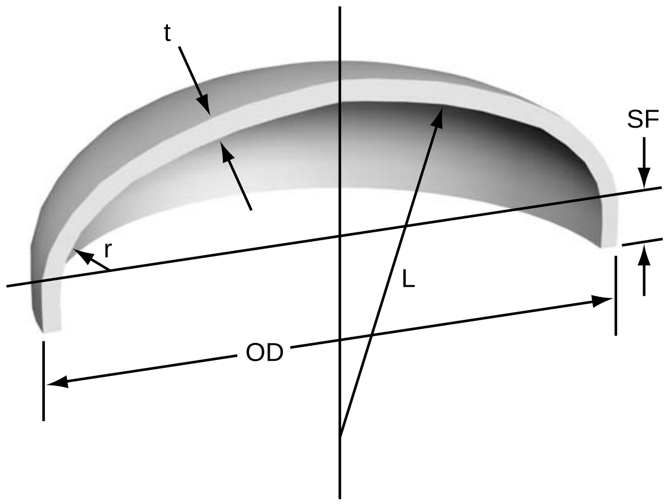


Figure 8.2 Torispherical head.

The formulas given in UG–32 for the torispherical head apply only to what is commonly referred to as the standard torispherical head. The standard torispherical head has a knuckle radius, r, of 6% of the inside crown radius, L, and the inside crown radius equals the outside diameter of the skirt, OD. The minimum required thickness and maximum allowable working pressure for this head are:

$$t = \frac{0.885PL}{SE - 0.1P}$$

or

$$P = \frac{SEt}{0.885L + 0.1t} \qquad (8.3)$$

Formulas for a knuckle radius other than 6% of the inside crown are in Appendix 1–4. They are:

$$t = \frac{PLM}{2SE - 0.2P}$$

or

$$P = \frac{2SEt}{LM + 0.2t} \qquad (8.4)$$

M is a factor that depends on the L/r ratio of the torispherical head. M is determined using the following formula:

$$M = \frac{1}{4}\left(3 + \sqrt{\frac{L}{r}}\right)$$

The allowable tensile stress value, S, for torispherical heads is also limited. If the minimum tensile strength of the material is greater than 70,000 psi (483 MPa), then the allowable tensile stress shall equal 20,000 psi (138 MPa) at room temperature or 20,000 (138 MPa) psi times the ratio of the material's maximum allowable stress at temperature divided by the material's allowable stress at room temperature.

Hemispherical Heads

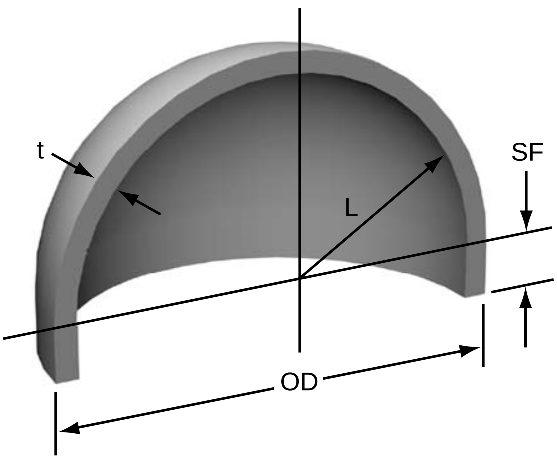


Figure 8.3 Hemispherical head.

The minimum required thickness after forming and the maximum allowable working pressure for hemispherical heads are:

$$t = \frac{PL}{2SE - 0.2P} \tag{8.5a}$$

or

$$P = \frac{2SEt}{L + 0.2t} \tag{8.5b}$$

Formulas 8.5a and 8.5b are applicable only when the thickness is less than or equal to 0.356L or P does not exceed 0.665SE. The formulas for a thick spherical shell, given in Appendix 1–3, must be used if the thickness or pressure exceed these limits.

Conical Heads and Sections (Without Transition Knuckles)

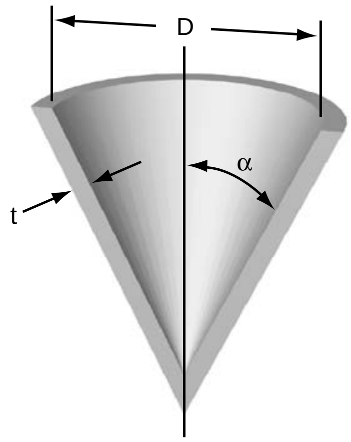


Figure 8.4 Conical head.

The most common use of a conical shell is as a transition between cylindrical shells of different diameters. This is usually the case, even when it is used as a head. The conical head usually terminates with an opening instead of an apex weld. The minimum required thickness and the maximum allowable working pressure of a conical shell section that has a half apex angle, α, equal to or less than 30 degrees are:

$$t = \frac{PD}{2\cos\alpha\,(SE - 0.6P)} \tag{8.6a}$$

or

$$P = \frac{2SEt\cos\alpha}{D + 1.2t\cos\alpha} \tag{8.6b}$$

Formulas 8.6 are applicable only if there is no transition knuckle with the conical head or section.

The angle joint between the conical section and the cylinder shall have a smooth contour and be equivalent to a Type 1 welded joint. It may be necessary to increase the thickness of the cylinder to get a smooth contour at the joint.

Reinforcement may be required, depending on the magnitude of the discontinuity stresses at the cone-to-cylinder junctions. The reinforcement can be in the form of excess shell thickness or reinforcing rings and may be needed for both small and large apex angles.

The following procedure for designing reinforcement when the half apex angle is equal to or less than 30° is presented in Appendix 1, item 5. The procedure must be applied at both the small and large end of the conical section.

Calculate the angle, Δ, using the appropriate P/S_sE_1 and Tables 1-5.1 or 1-5.2 in Appendix 1. No reinforcement is required if Δ is equal to or greater than α. S_s is the allowable stress of the cylinder at design temperature and E_1 is the efficiency of the longitudinal joints in the cylinder.
If Δ is less than α, then reinforcement is required and must be at least equal to:

$$A_r = \frac{kQ_iR_i}{S_SE_1}\left(1-\frac{\Delta}{\alpha}\right)\tan\alpha \tag{8.7}$$

R_i is the inside radius of the cylinder at the end of the cone, Q_i is the algebraic sum of $PR_i/2$ plus the axial load in the cylinder due to loads other than pressure. To use the Code procedure, Q must be a tensile load. If Q_i is compressive, the designer must use paragraph U–2(g). The ratio of SE/S_rE_r is k in Formula 8.7, and SE is the allowable stress and modulus of elasticity of the shell if the ring is placed on the shell or of the cone when the ring is placed on the cone. S_rE_r is the allowable stress and modulus of elasticity of the reinforcing ring.

The effective areas of available reinforcement in the cylindrical and conical sections are:

$$A_e = (t_s - t)(\sqrt{R_Lt_s}) + (t_c - t_r)\sqrt{R_Lt_c/\cos\alpha} \tag{8.8}$$

at the large end, and

$$A_e = 0.78\sqrt{R_st_s}\,\left[(t_s - t) + (t_c - t_r)/\cos\alpha\right] \tag{8.9}$$

at the small end.

Additional reinforcement, if required, must be within a distance of $(RT_S)^{½}$ from the junction, and the centroid of the added area must be within a distance of $0.25(RT_S)^{½}$ from the junction.

If the half apex angle is greater than 30 degrees, Appendix 1(g) requires that a special analysis, such as the beam-on-elastic-foundation analysis of Timoshenko, Hetenyi, or Watts and Lang, be done. At the junction between cone and the cylinder, there is an interruption in the continuity of the hoop stress and the introduction of stresses due to the moment caused by the variation in the stressed plane. This discontinuity stress must be considered. The Division does not provide direction as to how this is to be done, and it is therefore necessary to use other references for this evaluation. The discontinuity stresses calculated by such an analysis shall not exceed the following:

- Membrane hoop stress plus average discontinuity hoop stress shall be equal to or less than 1½ SE. The average discontinuity hoop stress is the average hoop stress across the wall thickness due to the discontinuity at the junction. Disregard the contribution due to Poisson ratio multiplied by the longitudinal stress.
- Membrane longitudinal stress plus discontinuity longitudinal stress due to bending shall be equal to or less than 4SE.

Toriconical Heads and Sections

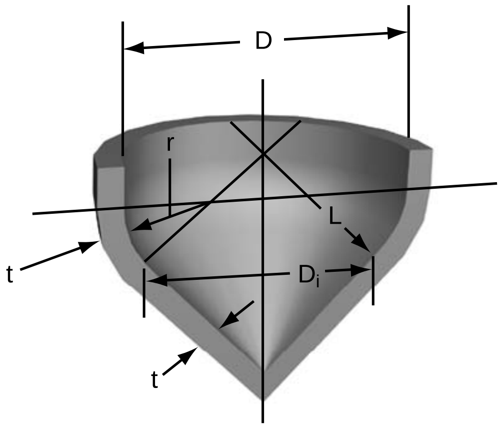


Fig. 8.5 Toriconical head.

Formulas 8.6 with D_i instead of D must be used to calculate the minimum required thickness of the conical portion of a toriconical head or section. The formulas are valid only if the knuckle radius is equal to or greater than 6% of the outside diameter of the head skirt and also equal to or greater than three times the knuckle thickness.

Formulas 8.4 with $L = D_i/2\cos\alpha$ must be used to calculate the minimum required thickness of the knuckle in a toriconical head.

Toriconical heads and sections may be used when $\alpha \leq 30$ degrees and are required when α is > 30 degrees unless the design satisfies paragraph 1-5(g) of Appendix 1.

Additional Requirements for Heads

In addition to the above formulas, the thickness of an unstayed ellipsoidal or torispherical head must be equal to or greater than the required thickness of a seamless hemispherical head divided by the efficiency of the head-to-shell joint.

Ellipsoidal, torispherical, hemispherical, conical, or toriconical heads with thicknesses less than those required by the formulas given in UG–32 must be designed as stayed flat surfaces as given in the rules of UG–47.

The inside crown radius of a dished head must be less than or equal to the outside diameter of the skirt of the head, and the inside knuckle radius must be equal to or greater than 6% of the outside diameter of the skirt of the head. The inside knuckle radius must also be equal to or greater than 3 times the head thickness (UG–32(J)).

When a skirt is required or provided, its thickness shall be equal to or greater than that of a seamless shell of the same diameter. Formed heads that are to be welded to the shell and are thicker than the shell are required to have a skirt. The skirt length of welded heads shall be in accordance with Fig. UW–13.1. Heads that are brazed to the shell shall have a skirt length in accordance with Part UB.

The diameter of flat spots in formed heads shall not exceed that permitted by Formula 1 in UG–34, using a C value of 0.25.

UG–33 Formed Heads, Pressure on Convex Side

Ellipsoidal Heads

The required thickness at the thinnest point of an ellipsoidal head is the largest value of t from:

- the procedure in UG–32(d) for internal pressure using a design pressure equal to 1.67 times the external design pressure, or
- the procedure in UG–28(d) with $R_o = K_oD_o$. K_o is given in Table UG–33.1.

Torispherical Heads

The required thickness at the thinnest point of a torispherical head is the largest value of t from:

- the procedure in UG–32 (e) for internal pressure using a design pressure equal to 1.67 times the external design pressure, or
- the procedure in UG–28(d) with R_o equal to the outside radius of the crown portion of the head.

Hemispherical Heads

Use the procedure from UG–28(d) for spherical shells, which is given in Chapter 7.

Conical Heads and Sections

UG–33(f) provides rules for designing conical heads and sections when subjected to external pressure. The rules are based on the half apex angle, α.

When α is less than or equal to 60 degrees, the procedure for designing a conical head or section for external pressure is the same as that in UG–28 for a cylindrical shell. An equivalent thickness and length is used to reflect the section geometry. The equivalent length L_e equals $(L/2)(1 + D_S/D_L)$. The effective thickness t_e equals $t \cos \alpha$. Figure 8.6 provides guidance in determining t_e and L_e.

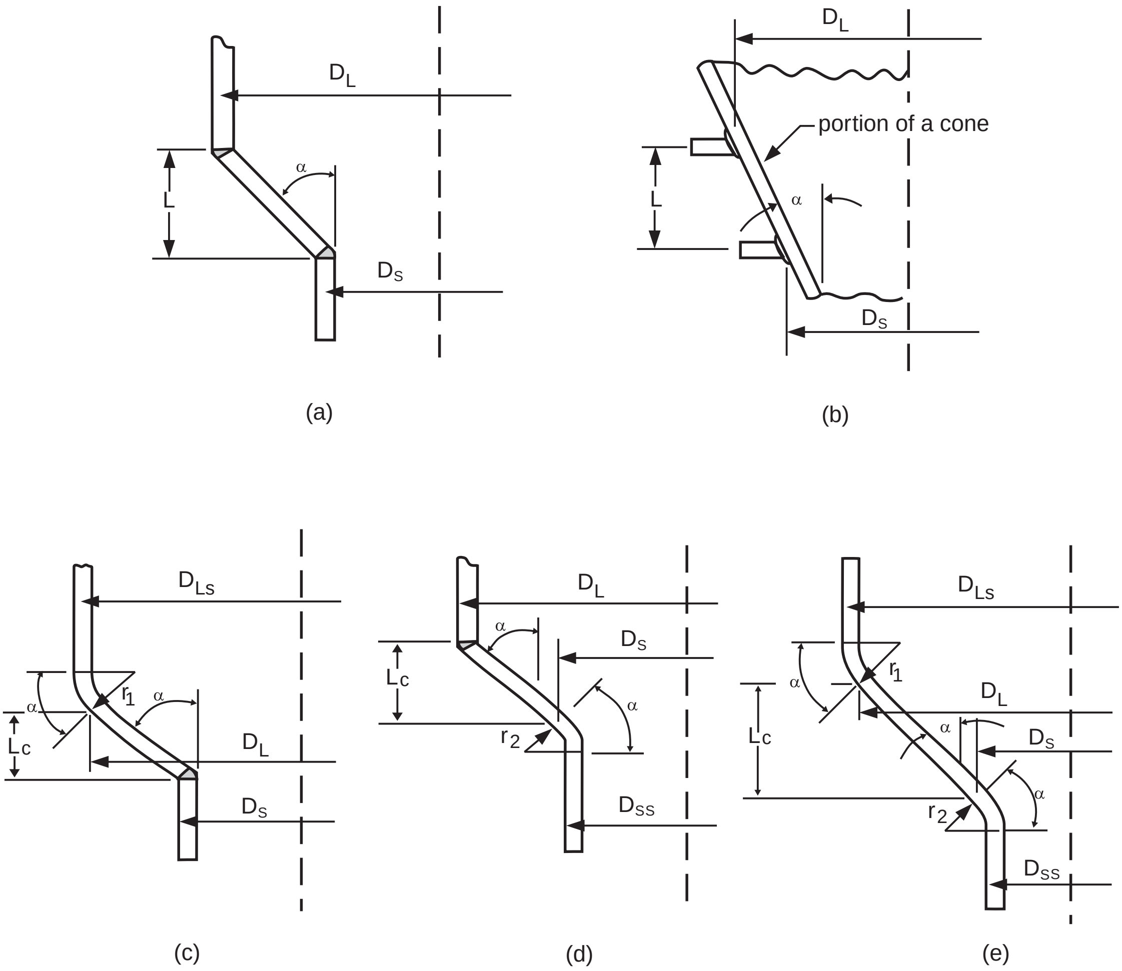


Figure 8.6 Length L of some typical conical sections for external pressure.
Fig. UG–33.1

When α is greater than 60 degrees, the thickness of the cone shall be the same as for an unstayed flat head under external pressure, assuming a diameter equal to the large diameter of the cone.

Unstayed Flat Heads and Covers

The formulas in the Division for the required thickness of flat unstayed heads, covers, and blind flanges provide safe construction considering stress only. A greater thickness may be necessary if deflection causes leakage at the threaded or gasketed joint. In the following formulas, the coefficient, C, depends on the configuration of the head and must be obtained from Figure 8.7. The value of d is the short span that is also defined in Figure 8.7 for various head configurations. W is the bolt clamping forces in psi and h_G is the gasket moment arm also defined in Figure 8.7.

For a flat unstayed circular head, cover, or blind flange use:

$$t = d\sqrt{CP/SE} \tag{8.10}$$

except when attached by bolts causing an edge moment, in which case use:

$$t = d\sqrt{CP/SE + 1.9Wh_G/SEd^3} \tag{8.11}$$

Formula 8.11 must be applied for both operating conditions and gasket seating. Use the largest t from the two conditions.

For a square, rectangular, elliptical, obround, segmental, etc. cover, use:

$$t = d\sqrt{ZCP/SE} \tag{8.12}$$

where: $Z = 3.4 - \dfrac{2.4}{D}$ but ≤ 2.5

and D is the long span, in inches, of the noncircular head. Reference to the d value in Figure 8.7 can be made to define D.

When attached by bolts causing an edge moment use:

$$t = d\sqrt{ZCP/SE + 6Wh_g/SELd^2} \tag{8.13}$$

where L is the perimeter, in inches, of the noncircular bolted head measured along the centers of the bolt holes.

Formula 8.13 must be applied to both operating conditions and gasket seating and the largest resulting t used.

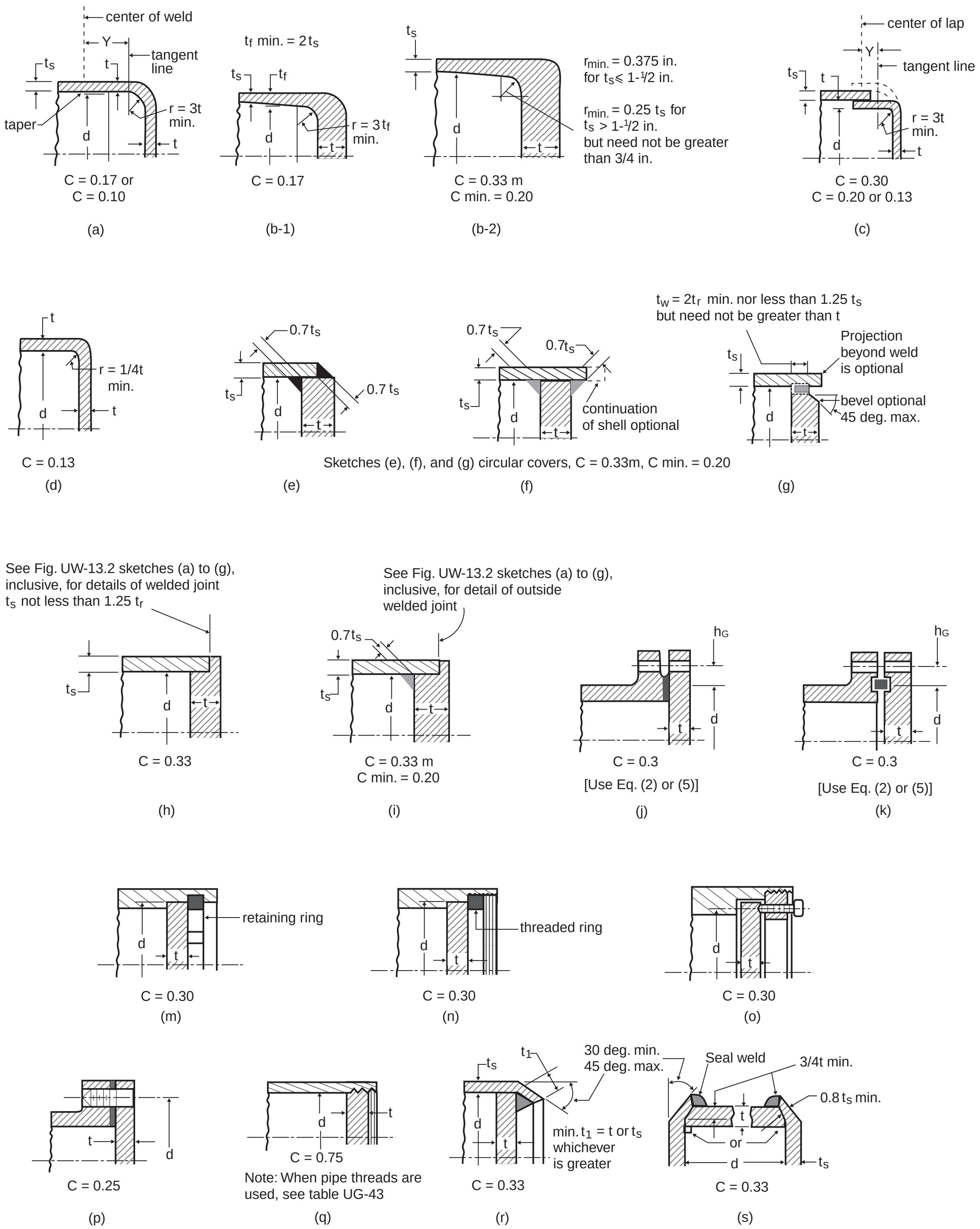


Figure 8.7 Some acceptable types of unstayed flat heads and covers. Fig. UG–34

Example 8.1 Design of a Standard Torispherical Head

Select a thickness for the internal torispherical head on the vessel shown in Figure 7.3. Design a head with an inside crown radius equal to the outside diameter of the skirt and a 6% knuckle radius. The vessel is fabricated such that the head skirt and cylinder have the same outside diameter. The internal design pressure is 150 psi at 800°F. Neglect the external pressure acting on the head. The vessel has full radiography. The material is SA-516 grade 70. The corrosion allowance on the concave side is 0.125 inch.

Solution: The allowable stress for the material is 12,000 psi. The dished head is seamless and designed according to UG-32. From UW-12, the joint efficiency E is 1.0.

$$t = \frac{0.885PL}{SE - 0.1P}$$

The inside crown radius is equal to the inside diameter plus 2 times the thickness of the shell. The shell thickness is calculated in Example 7.1 and is 0.882 inch. Therefore:

$$L = 120 \text{ in.} + 2(0.88 \text{ in.})$$

$$L = 121.8 \text{ inch}$$

$$t = \frac{0.885(150\text{psi})(121.8\text{in.})}{(12{,}000\text{psi})(1) - 0.1(150\text{psi})}$$

$$t = 1.350 \text{ inch}$$

Add corrosion allowance to the minimum thickness.

$$t_{req} = 1.350 \text{ in.} + 0.125 \text{ in.}$$

$$t_{req} = 1.475 \text{ inch}$$

Check post weld heat treatment and preheat requirements.

- UCS-56, Table 56 note (2): Post weld heat treatment or preheat is required if thickness is greater than 1¼ inch nominal thickness. Since 1.475 inches is greater than 1¼ inches and not over 1½ inches, a minimum of 200°F preheat maybe used in lieu of post weld heat treatment.

Check radiography requirement.

- UCS-57, Table 57 requires that P-1, Group 2 materials be fully radiographed if the butt joint thickness is greater than 1¼ inches. Since 1.475 inches is greater than 1¼ inches, full radiography is required.

Check minimum knuckle radius.

$r = 0.06(121.8\text{ in.})$

$r = 7.3\text{ inch}$

$r_{min} = 3.0(1.350\text{ in.})$

$r_{min} = 4.05\text{ inch}$

$r > r_{min}$ therefore, meets the requirements of UG-32(j).

Use 1.475 inch or thicker SA-516 Grade 70 plate. All weld seams must be Type 1 and fully radiographed. Preheat or post weld heat treatment is required for all weld seams.

Example 8.2 Design of a Non-Standard Torispherical Head

Rework Example 8.1 assuming that the inside crown radius is 80% of the skirt outside diameter and the inside knuckle radius is 10% of the skirt outside diameter. All other data is the same as that given in Example 8.1.

Solution: The new proportions require the use of Formula 8.4.

$L = 0.80[120\text{ in.} + 2\,(0.88\text{ in.})]$

$L = 97.4\text{ inches}$

$r = 0.10\,(121.8\text{ in.})$

$r = 12.18\text{ inches}$

$$t = \frac{PLM}{2SE - 0.2P}$$

where $M = \frac{1}{4}\left(3 + \sqrt{\frac{L}{r}}\right)$

$$M = \frac{1}{4}\left(3 + \sqrt{\frac{97.4\text{ in.}}{12.18\text{ in.}}}\right)$$

$M = 1.46$

M may also be obtained from Section VIII, Division 1, Table 1-4.2.

$$t = \frac{150\text{ psi}\,(97.4\text{ in.})(1.46)}{2\,(12{,}000\text{ psi})(1) - 0.2\,(150\text{ psi})}$$

$t = 0.89\text{ inch}$

Add corrosion allowance to the minimum thickness.

t_{req} = 0.89 in. + 0.125 in.

t_{req} = 1.02 inches

Check post weld heat treatment and preheat requirements.

- UCS–56, Table 56 note (2): Post weld heat treatment or preheat is required if thickness is greater than 1¼ inch nominal thickness. Since 1.02 inches is less than 1¼ inches, no preheat or post weld heat treatment is required.

Check radiography requirement.

- 1.02 inches is less than 1¼ inches. Full radiography is not required but is specified (see Example 8.1).

Check minimum length of knuckle radius.

r_{min} = 3.0 (0.89 in.)

r_{min} = 2.67 inches

$r > r_{min}$ therefore, meets the requirements of UG–32(j)

Note that the non-standard proportioned head is about ⅔ the thickness of the standard torispherical head. Additional cost savings are possible since preheat, post weld heat treatment, and full radiography are not required.

Example 8.3 Design of a Conical Transition Section

Design the conical transition section between the 120 inch and 96 inch shell sections shown in Figure 7.3. The design pressure is 200 psi at 800°F. The material is SA–516 Grade 70 and full radiography will be performed. Both shell sections are 1 inch thick. The Code required shell thickness for the smaller diameter is calculated to be 0.808 inch. The corrosion allowance is 0.125 inch. The geometry of the section is shown below.

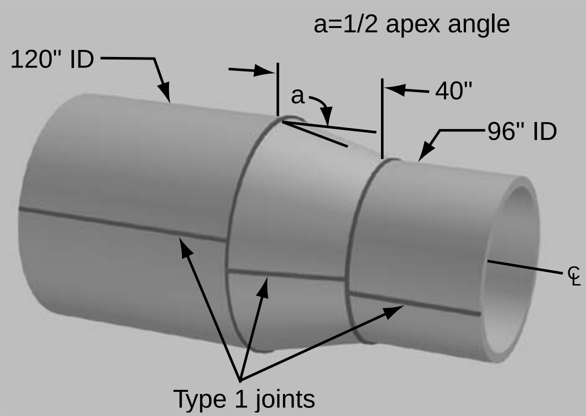


Figure 8.8 Conical transition for Example 8.3.

Solution: Use Formulas 8.6 through 8.9.

Determine the minimum required thickness.

From the sketch, the half apex angle is:

$$\mathrm{Tan}^{-1}\, a = \frac{(120\,\text{in.} - 96\,\text{in.})/2}{40\,\text{in.}}$$

a = 17° < 30° therefore use of Formula 8.6 is valid

$$t = \frac{PD}{2\cos a\,(SE - 0.6P)}$$

where: P = design pressure, UG–21
D = inside diameter plus corrosion allowance, UG–16 and UG–25
S = maximum allowable stress from Section II, Part D for SA-516, Grade 70 @ 800°F. (Observe notes.)
E = joint efficiency for a Type 1, fully radiographed longitudinal joint in a shell. (from Table UW-12)

$$t = \frac{200\text{ psi}\,[120\text{ in.} + (2)(0.125\text{ in.})]}{2\cos 17\,(12{,}000\text{ psi})(1.0) - 0.6\,(200\text{psi})}$$

t = 1.058 inches

t_{req} = 1.058 in. + 0.125 in. = 1.183 inch. Use 1¼ inch plate.

Check reinforcement requirement for large end.

Determine P/S_SE_L, Δ, and k where S is allowable stress of the cylinder and E is cylinder joint efficiency.

P/S_SE_L = 200 psi/12,000 psi = 0.0167

From Table 1–5.1 for P/S_SE_L = 0.0167, Δ is 30°. Δ equals the angle indicating need for reinforcement at the cone-to-cylinder junction when the half apex angle is less than or equal to 30°. No reinforcement is required when Δ is equal to or greater than the half apex angle.

Δ = 30° ≥ 17°, the half apex angle. No reinforcement is required at the large end of the conical section.

Check reinforcement requirement at the small end.

P/S_SE_L = 0.0167 from above

Interpolation of values in Table 1–5.2 is required to find Δ for the small end.

P/S_SE_1	Δ
0.010	9°
0.0167	Δ_S
0.020	12.5°

$\Delta_S = 11.35°$

11.35° is less than 17°. Therefore, reinforcement is required at the small end.

Determine k where k = Y/S_rE_r, but not less than 1.0. Y equals the greater of S_SE_S or S_cE_c. S is the allowable stress and E is the modulus of elasticity.

$S_SE_S = S_cE_c$ = (12,000 psi)(30 x 10^6 psi) = 3.6 x 10^{11} psi•psi

$$k = \frac{3.6 \times 10^{11}\ \text{psi} \bullet \text{psi}}{(12{,}000\ \text{psi})(30 \times 10^6\ \text{psi})}$$

k = 1.0

Determine the required area of reinforcement.

$$A_{rs} = \frac{PR_S^2k}{2S_SE_1}(1 - \frac{\Delta_S}{a})\tan(a)$$

$$= \frac{(200\,\text{psi})(96\,\text{in./2})^2(1.0)}{2(12{,}000\text{psi})}(1 - \frac{11.35}{17})\tan 17° = 1.95\ \text{inches}^2$$

When the nominal thickness less corrosion allowance of the cylinder and cone exceed that required by the applicable required minimum thickness formula, the excess thickness may be used to satisfy reinforcement requirements. The effective excess thickness is:

$$A_{es} = 0.78\sqrt{(R_st_s)}\left[(t_s - t) + \frac{(t_c - t_r)}{\cos a}\right]$$

$$A_{es} = 0.78\sqrt{(48\ \text{in.} \times 1\ \text{in.})}\left[(1.00\ \text{in.} - 0.808\ \text{in.}) + \left(\frac{1.25\ \text{in.} - 0.125\ \text{in.} - 1.058\ \text{in.}}{\cos 17°}\right)\right]$$

= 1.41 inch^2 < 1.95 inches^2

Therefore, a reinforcement ring is required. The area of the ring must be equal to or greater than:

A_{ring} = 1.95 in.^{2} - 1.41 in.^{2} = 0.54 inches^2

Use a ½ inch by 2 inch ring (1.00 inch^2) attached at the junction.

Example 8.4 External Pressure Design of a Torispherical Head

Determine the minimum required thickness, based on external pressure, of the internal torispherical head in Figure 7.3. The inside crown radius equals the outside diameter of the skirt and the knuckle radius is 6% of the inside crown radius. The vessel is fabricated with the head skirt and cylinder having the same outside diameter. The vessel must be designed for an external pressure of 50 psi at 800°F. The vessel has full radiography. The material is SA–516 Grade 70. The corrosion allowance is 0.125 inch on the concave side only.

Solution: The external design procedure in the Division requires an initial or assumed thickness. The required thickness for internal pressure is the most obvious starting thickness. However, in this example, it is assumed that the designer elects to do the external design first.

UG–33 states that the required minimum thickness for external pressure is the larger of the calculated thicknesses in (a) and (b).

(a) Design according to the internal pressure procedure (Formula 8.3) with an internal pressure equal to 1.67 times the external design pressure.

$$t = \frac{0.885PL}{SE - 0.1P}$$

$$t = \frac{0.885(50\,\text{psi})(1.67)(121.8\,\text{in.})}{(12{,}000\,\text{psi})(1.0) - 0.1(1.67)(50\,\text{psi})}$$

t = 0.75 inch

(b) Design with the external pressure procedure for a spherical shell where R_o is equal to the outside radius of the crown.

Assume t = 0.75 inch

R_o = 121.8 in. + 0.75 in.

R_o = 122.55 inch

Calculate the factor A for a sphere.

$$A = \frac{(0.125)(0.75\,\text{in.})}{122.51\,\text{in.}} = 0.00077$$

For A = 0.00077 and design temperature of 800°F, B = 6,500 psi as in Figure CS–2 of Section II, Part D.

Calculate the maximum allowable external pressure for B = 6,500 psi.

$$P_a = \frac{B}{R_o/t}$$

$$P_a = \frac{6{,}500\,\text{psi}}{(122.55\,\text{in.}/0.75\,\text{in.})} = 39.8\ \text{psi}$$

P_a = 39.8 psi < 50 psi. Therefore, repeat the process with a greater thickness.

Assume t = 1.0 inch

R_o = 121.8 in. +1.0 in.

R_o = 122.8 inch

$$A = \frac{(0.125)(1.0\,\text{in.})}{122.8\,\text{in.}} = 0.0010$$

Figure CS–2 of Section II, Part D gives a value of B = 7,100 psi at 800°F for A = 0.0010.
Calculate the maximum allowable external pressure for B = 7,100 psi.

$$P_a = \frac{7{,}100\,\text{psi}}{(122.8\,\text{in.}/1.00\,\text{in.})}$$

P_a = 57.8 psi > 50 psi, therefore, it is acceptable.

t = 1.00 inch

Step (b) produces t = 1.00 inch, which is the largest minimum required value to withstand an external pressure of 50 psi. Adding corrosion allowance gives a required thickness of:

t_{req} = 1.00 in. + 0.125 in.

t_{req} = 1.125 inches

The minimum thickness for this head should be the largest of the thicknesses required for internal and external design. Example 8.1 shows that the internal design conditions require a thicker head. Therefore, the minimum thickness of the head should be 1.475 inches, the thickness to satisfy internal design pressure requirements.

The Code user should observe that in many cases the external design process can be shortened if the initial thickness is assumed to be the thickness required to satisfy the internal design pressure condition. In this example, 1.475 inches, minus the corrosion allowance, fulfills all external design requirements.

Example 8.5 Design of an Unstayed Flat Head

Design the 48 inch flat head shown in Figure 7.3. The design pressure is 150 psi at 800°F. The corrosion allowance is 0.125 inch. The vessel is fabricated from SA–516 Grade 70 with full radiography. The uncorroded thickness of the nozzle is 0.50 inch. The head-to- shell configuration is shown in Fig. UG–34(f).

Solution: The thickness must be determined using Formula 8.10, which is taken from UG–34. The constant C must be computed according to Fig. UG-34(f) and paragraph UG–34(d).

$$t = d\sqrt{CP/SE} \quad \text{where } C = 0.33M \geq 0.20$$

$$M = t_R/t_S$$

t_R is the required thickness of a seamless shell of the same diameter, and t_S is the actual thickness of the shell. Remember all Division formulas are in the corroded condition.

$$t_R = \frac{PR}{SE - 0.6P}$$

$$t_R = \frac{(150\ \text{psi})\,(48\ \text{in.}/2)}{(12{,}000\ \text{psi})\,(1.0) - 0.6\,(150\ \text{psi})}$$

$$t_R = 0.30\ \text{inch}$$

$$M = \frac{0.30\ \text{in.}}{0.50\ \text{in.} - 0.125\ \text{in.}}$$

$$M = 0.8$$

$$C = 0.33(0.80) = 0.264$$

$$t = 48.0\ \text{in.}\sqrt{(0.264)(150\ \text{psi})/(12{,}000\ \text{psi})\,(1.0)}$$

$$t = 2.76\ \text{inches}$$

$$t_{req} = 2.76\ \text{in.} + 0.125\ \text{in.}$$

$$t_{req} = 2.885\ \text{inches}$$

Make the head 3 inches thick.

Note that the fillet welds must have a throat equal to or greater than $0.7t_s$, which equals 0.26 inch. Also, paragraph UG–34(d) states that if a value of M less than 1 is used in calculating the head thickness, the shell thickness t_s must be maintained for at least a distance of $2(dt_s)^{½}$. This requirement is satisfied since the thickness of the nozzle throughout its length is t_s.

Chapter 9

OPENINGS AND REINFORCEMENTS

Vessels have openings to accommodate manholes, handholds, and nozzles. Openings vary in size from small drain nozzles to full vessel size openings with body flanges. When an opening is cut into a symmetrical shell or head, the load normally carried by the removed metal must be carried by the wall adjacent to the opening. This added load increases stresses in the vessel wall adjacent to the opening. The increased stress will produce stresses higher than allowed by the Code unless the component has excess thickness.

Figure 9.1 is a plot of the stress variation in a flat plate with a hole. The bi-directional stress ratio is ½, which represents the ratio of longitudinal to circumferential stress found in a cylindrical shell. Note that the stress varies from a maximum of 2.5 times the nominal stress at the edge of the opening to 1.09 times the nominal stress at a distance of 3r from the center. At a distance of one diameter from the center, the stress in the unreinforced opening is 1.23 times the nominal stress.

Code reinforcement rules are based on replacing the metal area removed by the opening. The rules consider only internal and external pressure and are given in both the main body of the Division and the Appendices. Area required to resist external loads such as moments and forces caused by dead load or piping is not addressed. The designer must use U–2(g) to analyze the effect of nonpressure loads on openings.

UG–36 Openings in Pressure Vessels

All openings in pressure vessels shall meet the requirements for reinforcement given in paragraphs UG–36 through UG–42 and Appendix 1–7 if required by size limits. The weld size requirements of UW–16 must also be satisfied. The ligament rules given in UG–53 may be used for multiple openings in lieu of paragraphs UG–36 to UG–42, unless exempted by size, type, or special applications.

Openings may be of any shape. However, Division 1 states a preference for circular, elliptical, or obround openings. All corners must have a radius. When the long dimension of an elliptical or obround opening exceeds twice the short dimensions, the reinforcement across the short dimension must be increased to prevent distortion due to the twisting moment.

If the following opening size limits are exceeded, the supplemental rules given in Appendix 1–7 shall be satisfied:

- for vessels 60 inches (1520 mm) diameter or less; one-half the vessel diameter, or 20 inches (508 mm)
- for vessels over 60 inches (1520 mm) diameter; one-third the vessel diameter, or 40 inches (1000 mm)

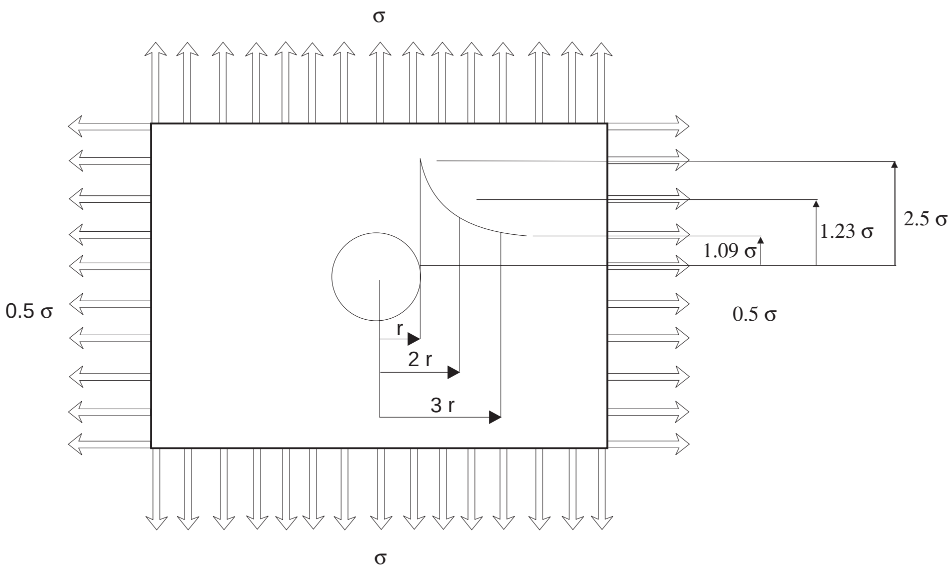


Figure 9.1 Stress distribution in flat plate with circular opening.

UG–36(c)(3) exempts openings from reinforcement calculations if the vessel is not subject to rapid pressure fluctuations and the following limits are not exceeded:

(a) For welded, brazed or flued connections:
- in plate of ⅜ inch (10 mm) thickness or less, the maximum opening is 3½ inches (89 mm) diameter
- in plate greater than ⅜ inch (10 mm) thickness, the maximum opening is 2⅜ inches (60 mm) diameter

(b) For threaded, studded, or expanded connections:
- in all plate thicknesses, the maximum opening is 2⅜ inches (60 mm) diameter

[Interpretation VIII–1–95–124 indicates that the exemptions are applicable only for a perpendicular nozzle.]

No two openings described in (a) or (b) above shall have their centers closer to each other than the sum of their diameters.

No two openings in a cluster of three or more as described in (a) and (b) above shall have their centers closer than as follows:

For cylinders and cones:

$$(1 + 1.5 \cos \theta)(d_1 + d_2)$$

For doubly curved shells or heads:

$$2.5 (d_1 + d_2)$$

where:

θ is the angle between a line connecting the center of the openings and the longitudinal axis of the shell;

d_1 and d_2 are the finished diameters of the adjacent openings (Figure 9.2).

Except for category B joints, openings in butt welded joints or openings with some reinforcement over butt welded joints must use the weld joint efficiency in the calculation for available area in the vessel wall and shall meet the additional requirements in UW–14. A joint efficiency of 1 may be used for openings in category B weld joints or in solid plate. UW–14 requires full radiography of the seam for a length of 3 times the diameter of the opening for all openings exempted from reinforcement by paragraph UG–36(c)(3). The center of the radiographed area must coincide with the center of the opening.

UG–36 provides rules for openings designed as reducer sections. Such openings must also satisfy the requirements of UG–32 if they are subjected to internal pressure and UG–33 when subjected to external pressure. Reverse curve reducers must be designed according to U–2(g).

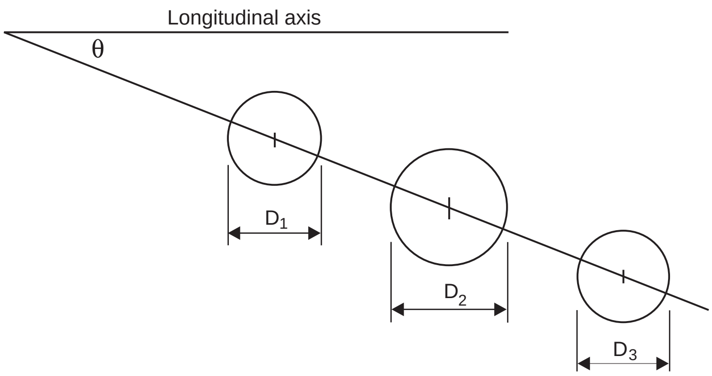


Figure 9.2 Center spacing of openings.

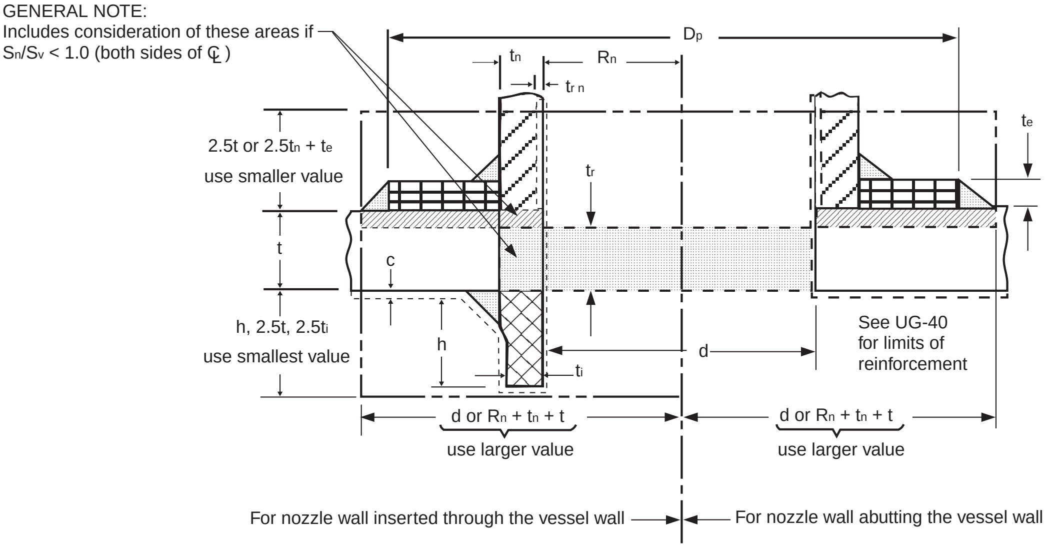
UG–37 Reinforcement Required for Openings in Shells and Formed Heads

UG–37 contains the reinforcement requirements for openings not exempted from reinforcement calculations by UG–36. The requirements are based on the area replacement rule and apply for both internal and external pressure. The area replacement rule states that excess or extra metal must be available to carry the load that would normally be carried by the missing metal. This excess material must also be within certain limits in order to be effective.

The excess metal can be surplus material in the shell and nozzle or an extra reinforcing pad. The reinforcement pad may be placed inside or outside of the vessel. The amount of missing metal that must be replaced equals the minimum required thickness of the shell times the diameter of the opening. The requirements for internal and external pressure are the same, except that the minimum required thickness for internal pressure of a cylindrical shell is according to UG–27, while UG–28 must be used to determine the minimum required thickness for external pressure. If the component is designed exclusively for external pressure, then only 50% of the required area needs to be replaced.

When the opening and its reinforcement are entirely within the spherical part of a torispherical head, the required thickness must be based on the formula, in Appendix 1–4(d), using M = 1. If the opening and its reinforcement are entirely within the central 80% of an ellipsoidal head, the required thickness must be calculated using the formula for a seamless sphere of radius K_1D, where D is the shell diameter and K_1 is given in Table UG–37.

Figure 9.3 contains a drawing, nomenclature, and formulas for reinforced openings. Note that the required area A in any given plane equals $dt_rF + 2t_nt_rF(1–f_{r1})$. If the opening is not symmetrical, then consideration for reinforcement requirements must be made in all planes. The variable F is a stress correction factor and varies from 1 for the longitudinal plane to 0.5 on the circumferential plane. The variation corresponds to the variation in stress as one proceeds from the longitudinal to the circumferential plane in a cylindrical shell. F = 1 for all configurations except Fig. 37 may be used for integrally reinforced openings in shells and cones. Reinforcement is not integral if a reinforcing pad is welded to the vessel. The strength reduction factor f_{r1} will be discussed later.



Without Reinforcing Element

= A	$= d\,t_r\,F + 2t_n\,t_r\,F(1 - f_{r1})$	Area required
= A_1	$= d(E_1 t - F t_r) - 2t_n (E_1 t - F t_r)(1 - f_{r1})$ $= 2(t + t_n)(E_1 t - F t_r) - 2t_n (E_1 t - F t_r)(1 - f_{r1})$	Area available in shell; use larger value
= A_2	$= 5(t_n - t_{rn}) f_{r2}\, t$ $= 5(t_n - t_{rn}) f_{r2}\, t_n$	Area available in nozzle projecting outward; use smaller value
= A_3	$= 5t\, t_i\, f_{r2}$ $= 5t_i\, t_i\, f_{r2}$ $= 2h\, t_i\, f_{r2}$	Area available in inward nozzle; use smallest value
= A_{41}	= outward nozzle weld = $(\text{leg})^2 f_{r2}$	Area available in outward weld
= A_{43}	= inward nozzle weld = $(\text{leg})^2 f_{r2}$	Area available in inward weld

If $A_1 + A_2 + A_3 + A_{41} + A_{43} \geqslant A$ — Opening is adequately reinforced

If $A_1 + A_2 + A_3 + A_{41} + A_{43} < A$ — Opening is not adequately reinforced so reinforcing elements must be added and/or thicknesses must be increased

With Reinforcing Element Added

A	= same as A, above	Area required
A_1	= same as A_1, above	Area available
A_2	$= 5(t_n - t_{rn}) f_{r2}\, t$ $= 2(t_n - t_{rn})(2.5t_n + t_e) f_{r2}$	Area available in nozzle projecting outward; use smaller area
A_3	= same as A_3, above	Area available in inward nozzle
= A_{41}	= outward nozzle weld = $(\text{leg})^2 f_{r3}$	Area available in outward weld
= A_{42}	= outer element weld = $(\text{leg})^2 f_{r4}$	Area available in outer weld
= A_{43}	= inward nozzle weld = $(\text{leg})^2 f_{r2}$	Area available in inward weld
= A_5	$= (D_p - d - 2t_n)\, t_e\, f_{r4}$ [Note (1)]	Area available in element

If $A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 \geqslant A$ — Opening is adequately reinforced

NOTE:
(1) This formula is applicable for a rectangular cross-sectional element that falls within the limits of reinforcement.

Figure 9.3 Nomenclature and formulas for reinforced openings. Fig. UG–37.1.

The following areas are available for reinforcement:

A_1 – excess thickness in the shell, Figure 9.4. If the nominal thickness of the shell is greater than the minimum required to resist the MAWP, the excess thickness may be used for reinforcing the opening. Two formulas are given, and the larger value must be used. [Interpretation VIII–1–89–171 clarifies that the calculated wall thickness for reinforcement surplus is based on using E = 1 regardless of the joint efficiency determined by the rules of UW–12, provided that the opening is not on or in a weld. Interpretation VIII–1–92–69R points out that the mill tolerance must be accounted for in shells made from pipe.]

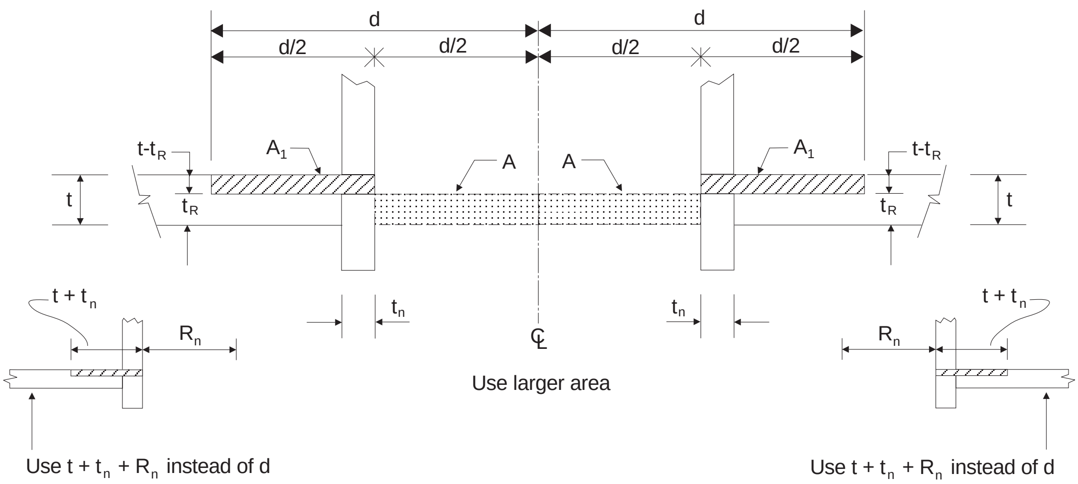


Figure 9.4 Required area A and available shell area A_1.

A_2 – excess thickness in the nozzle, Figure 9.5. The extra thickness in the nozzle beyond that required to resist the pressure may be used as reinforcement. Two formulas are also given for this contribution, and the smaller of the two must be used. [Interpretation VIII–1–95–123 indicates that surplus thickness in a nozzle that is threaded or expanded into an opening does not contribute to the reinforcement. Interpretation VIII–1–83–359 states that if a threaded nozzle does not extend through the thickness of the opening, t_n = 0. Interpretations VIII–1–92–25R and VIII–1–83–240 allow the seamless pipe stress values for nozzles made from ERW pipe.]

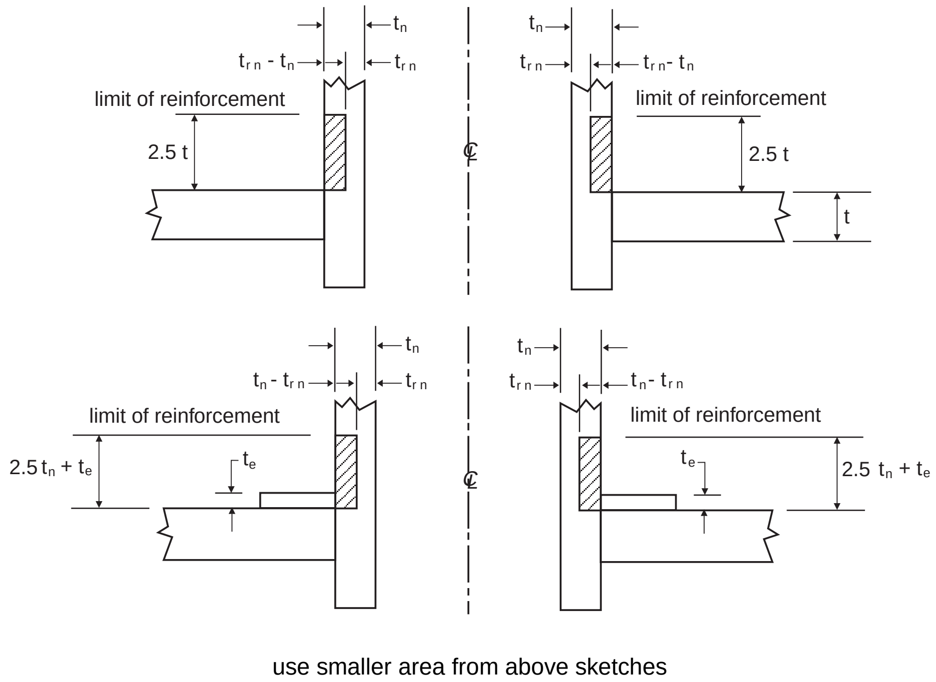


Figure 9.5 Outward projecting nozzle area A_2.

A_3 – inward nozzle area, Figure 9.6. The portion of the nozzle that projects into the vessel does not resist pressure. Therefore, with the exception of corrosion allowance, all of the projected area may be used as reinforcement.

A_{41} and A_{43} – fillet weld area, Figure 9.6. Both fillet weld areas, if they are within the reinforcement limits, may be used as reinforcement.

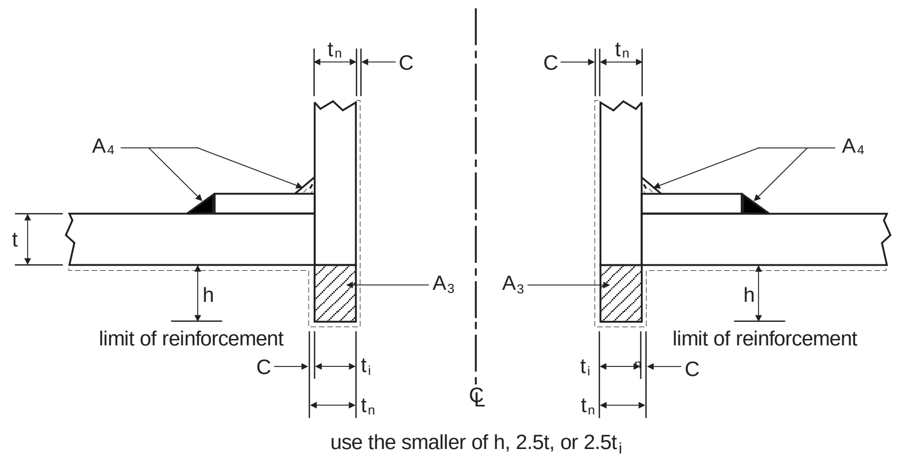


Figure 9.6 Inward nozzle available area A_3 and weld areas A_{41} and A_{42}.

A_5 – reinforcement pad, Figure 9.7. The cross-sectional area of the reinforcement pad within the reinforcement limits may be used as reinforcement. [Interpretation VIII–1–86–18 clarifies that even though Fig. UG–37.1 shows the reinforcing pad as a flat element, contoured pads are applicable.]

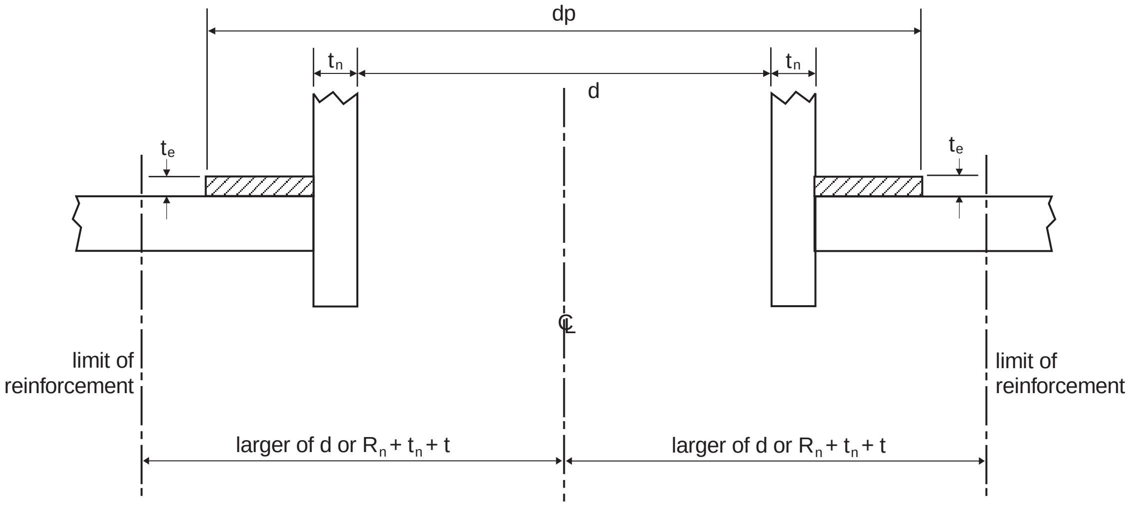


Figure 9.7 Reinforcement pad area A_5.

The sum of the excess areas A_1, A_2, A_3, A_{41}, $A_{43,}$ and A_5 must be equal to or greater than the required area A. If this condition is not satisfied, then additional excess areas must be provided.

UG–39 Reinforcement Required for Openings in Flat Heads

In general, the Division offers two ways of providing adequate reinforcement for openings in flat heads. The first consists of providing extra area equal to ½ of the removed area. The ½ requirement considers that flat heads are in bending, making the stress a function of the section modulus. The other option is to compensate for the opening by increasing the thickness of the head.

The rules in UG–39 apply to all openings except those small openings exempted by UG–36(c)(3).

1. Single Opening

When the opening does not exceed one-half the head diameter or shortest span, the following formula applies:

$$A = 0.5dt + tt_n\,(1\text{-}f_{r1})$$

When the opening exceeds one-half the head diameter, the rules of Appendix 14 apply. Appendix 14 addresses only single circular and centrally located openings. If these conditions do not apply, then U–2(g) is applicable.

As an alternative, the thickness of the flat head may be increased by using the flat head formulas in UG–34 with adjusted C factors. The adjusted C factors are given in UG–39(d).

2. Multiple Openings

Multiple openings with diameters equal to or less than ½ the head diameter, and no pair with an average diameter greater than ¼ of the head diameter may be reinforced as single openings using the following formula to determine the required reinforcement:

$$A = 0.5dt + tt_n\,(1\text{-}f_{r1})$$

Also, the spacing between any pair of openings must be equal to or greater than two times their average diameter. As an alternative to reinforcement, a thicker head based on UG–34 and the adjusted C factors in UG–39(d) may be used.

When the spacing between adjacent openings is less than two times their average diameter, but equal to or greater than 1.25 times the average diameter of the pair, the required reinforcement for each opening in the pair may be based on the above formula and shall be added together and distributed such that 50% of the sum is located between the two openings. As an alternative to reinforcement, an intermediate head thickness may be calculated using UG–34 and the adjusted C factors in UG–39(d). The final thickness is to be the intermediate thickness times h, where h equals $(0.5/e)^{1/2}$, and e is the smallest value of $[p\text{-}d_{ave}/p]$,.where p is the center to center spacing of two adjacent openings, and d_{ave} is the average diameter of the same two openings.

When the spacing is less than 1.25 times the average diameter, use paragraph U–2(g).

In no case shall the ligament between pairs of openings be less than ¼ the diameter of the smaller opening. Also, the ligament between the edge of an opening and the edge of the flat head must be equal to or greater than ¼ the diameter of the smaller opening.

3. Rim Openings

Openings may be located in the rim surrounding a central opening as shown in Figure 9.8 (Fig. UG–39). Rim openings must satisfy requirements 1 and 2 above or the head thickness must be calculated per Appendix 14 and increased by 1.414 for single openings and e for multiple opening where e is defined in the section on *Multiple Openings* above.

Rim openings shall not be larger than ¼ the difference between the head diameter and the central opening diameter. The ligament widths shown in Fig. UG–39 must be equal to or greater than ¼ the diameter of the smallest opening diameter.

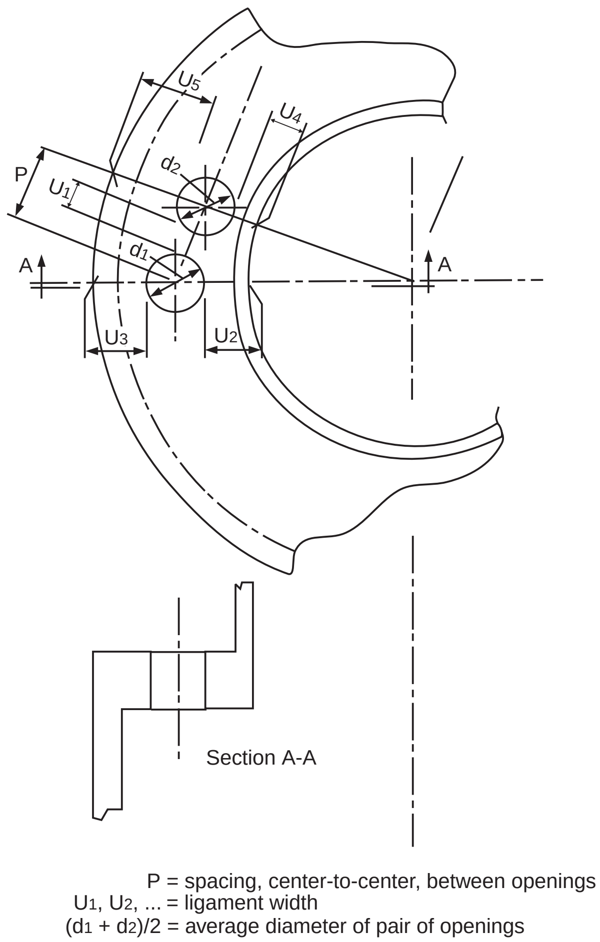


Figure 9.8 Multiple openings in rim of heads with a large central opening. Fig. UG–39.

Limits of Reinforcement

Figure 9.1 shows that the stress at the edge of an opening drops off rapidly. At a distance of one diameter from the center of the opening, the stress is about 1.23 times the remote stress. The overstress is less than 10% of the remote stress at 1½ diameters from the center. Therefore, to be effective, reinforcement must be close to the opening. The boundaries of the cross sectional area where reinforcement is effective are designated as the limits of reinforcement.

The limits of reinforcement normal to the shell are shown in Figures 9.5 and 9.6. The limits of reinforcement parallel to the shell are shown in Figure 9.7. Only metal that is in excess of the thickness required to resist pressure, other loads (UG–22), plus the thickness specified as corrosion allowance and also within the limits of reinforcement, may be counted as reinforcement. [Interpretation VIII–1–89–83 indicates reinforcement can extend into another component of the vessel. For example, reinforcement can extend from a shell onto the head.]

UG–41 Strength of Reinforcement

Paragraph UG–41 deals with the allowable stress of reinforcement and the strength of the attachment joining the vessel wall and the reinforcement. Section VIII, Division 1 states that the reinforcement preferably should have an allowable stress equal to or greater than the allowable stress of the material in the vessel wall. When either the nozzle or pad has a lower allowable stress than the shell, the reinforcement contribution from that nozzle or pad must be reduced by multiplying the actual area by the ratio of the allowable stresses. However, if the nozzle or pad has a higher allowable stress than the shell, no credit may be taken for the additional strength. The strength reduction factor f_r applies this correction to the area formulas in Fig. UG–37.1.

Attachment welds also have a strength reduction factor. Fillet weld areas are credited with an allowable tensile stress equal to the weaker of the parts joined by the weld. Groove welds such as vessel-to-nozzle or pad-to-nozzle welds may be credited with an allowable tensile stress equal to that of the vessel wall or pad as appropriate.

In order to be effective, reinforcement pads and nozzles must be attached with welds that are capable of transmitting the load from the shell to the pad and nozzle. Fig. UG-41.1 shows the weld strength paths and the loads that must be transmitted through each path.

The failure paths are based on the concept of tearing open the shell along a plane passing through the opening, while the reinforcing pad and nozzle, with the exception of a shear failure, remain intact. The weld load results from the tensile stress acting on the reinforcing elements that transmit load from the shell to the reinforcing pad and nozzle. This concept is illustrated with Figure 9.9. A reinforced opening is shown and there is no excess thickness in the shell or nozzle and the allowable stress (S_v) is the same for all components. The reinforcing pad A_5 and fillet welds A_{41} and A_{42} provide all reinforcement. Therefore, the load W that must be transmitted by the reinforcement equals:

$$W = d t_r S_v$$

The tensile force, which would normally be in the removed area A, is transmitted by areas A_5, A_{41}, and A_{42}. Consider failure path 1. The shell opens like a fish mouth carrying to one side the reinforcing pad and nozzle intact. To prevent this type of failure weld, A_{42} and the nozzle wall must be able to support a shear load equal to W. Figure 9.10 is an exaggerated view of the failure along path 1. Note that both the nozzle and weld areas fail around one-half of their circumference.

Failure along path 2 is similar to failure on path 1. The weld A_{41} and the nozzle wall fail in shear resulting in movement of the nozzle and shell relative to the reinforcing pad. Failure along path 3 is through welds A_{41} and A_{43}. Weld A_{41} is in shear. However, for the nozzle to separate from the shell, weld A_{43} must fail in tension.

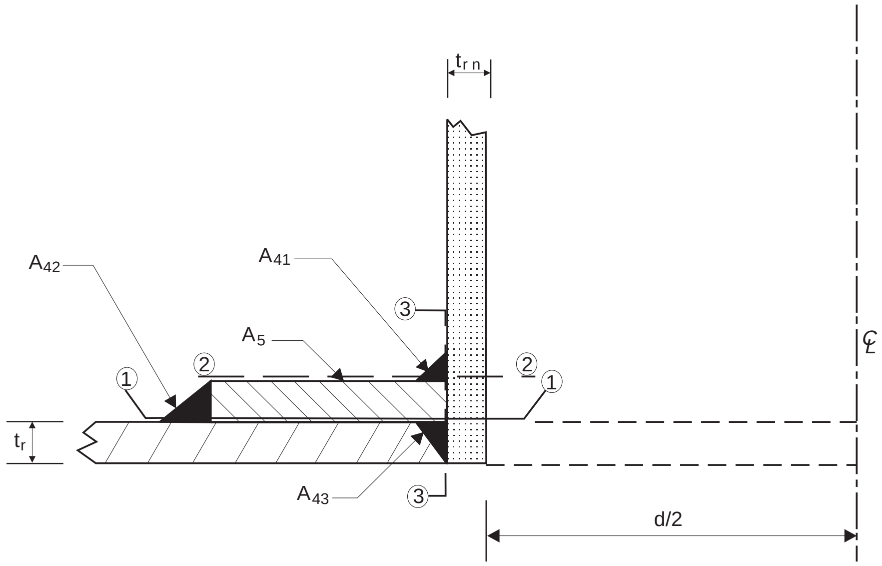


Figure 9.9 Weld failure paths and area load.

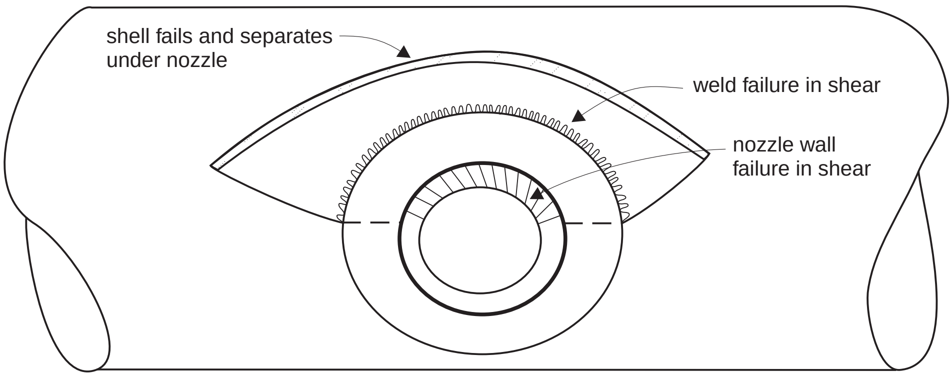


Figure 9.10 Weld failure path 1–1.

Fig. UG–41.1 gives the nozzle attachment weld loads as four formulas. W is the total weld load. It equals the required area A minus the reinforcement contribution from the shell times the allowable stress of the shell. $W_{1\text{-}1}$, $W_{2\text{-}2}$, and $W_{3\text{-}3}$ are the tensile loads the failure paths 1, 2, and 3 must resist.

UW–15 gives exemptions from weld strength calculations and provides the allowable weld tensile and shear stress as a fraction of the material allowable stress. Openings designed in accordance with the ligament rules of UG–53 and those openings exempted from reinforcement in UG–36(c)(3) as well as several sketches in Fig. UW–16.1 are exempt from weld strength calculations. [Interpretation VIII–1–89–196R qualifies the UG-36(c)(3)(a) exemption from calculations when the exemption for Fig. UW–16.1 minimum weld size is used in accordance with UW–16(f)(2).] The maximum permitted tensile stress must be multiplied by the following factors to obtain the appropriate allowable stress in the weld allowables:

Groove-weld tension	0.74
Groove-weld shear	0.60
Fillet-weld shear	0.49

The maximum permitted shearing stress for the nozzle neck is 70% of the allowable tensile stress of the nozzle material.

UG–42 Reinforcement of Multiple Openings

When the spacing between two openings is less than two times their average diameter, the reinforcement must satisfy the requirements for multiple openings. There are three general procedures for reinforcing multiple openings. They are: (1) standard reinforcement practice with restrictions on the distribution of the reinforcement, (2) reinforcement based on a large opening that encompasses the multiple openings, and (3) the ligaments rules of UG–53.

Procedure (1)

When any two openings are spaced such that their reinforcement overlaps, the reinforcement in the plane connecting the centers of the two openings must be equal to or greater than the sum of the area required for each opening and must satisfy all standard reinforcement requirements (UG–37, 38, 40, and 41).

The overlap area shall be proportioned between the two openings by the ratio of their diameters. Available cross sectional areas may be considered as reinforcement contributions to only one of the openings. If the reinforcement area between the openings is less than 50% of the required area, the rules of paragraph 1–7 must be applied.

Several openings on the same center line may be treated as successive pairs.

When several openings are spaced at less than two times, but equal to or more than 1⅓ times their average diameter, and when they are not on the same center line, then at least 50% of the total required reinforcement for the two openings must be placed between the two openings. No reinforcement credit may be taken for any surplus material between the two openings. One of the other procedures must be used if the distance between the centers of the openings is less than 1⅓ times their average diameter.

Procedure (2)

Any number of openings in any arrangement may be treated as one large opening with a diameter that includes the entire pattern. The limit of reinforcement for the assumed opening shall be the assumed diameter, parallel to the shell, and 2½ times the shell thickness less corrosion allowance, normal to the shell. The nozzle walls shall not be used as reinforcement.

Paragraph 1–7 must be used if the assumed opening diameter exceeds the requirements for large openings in UG–36(b)(1).

Procedure (3)

Paragraph UG–53, Ligaments, may be used for two or more openings arranged in a regular pattern.

UG–43 Methods of Attaching Pipe and Nozzle Necks to Vessel Walls

Pipe and nozzles may be attached to the wall of a vessel with welded, brazed, studded, threaded, and expanded connections (UG–43). The welded connections must satisfy the requirements of UW–15 and UW–16. The brazed connections must comply with the requirements of UB–17 through UB–19.

The studs of a studded connection must be in holes that are drilled and tapped in a flat surface, machined on the shell or on a built up pad or attached plate or fitting. The remaining thickness at the bottom of the hole must be equal to or greater than ¼ of the required wall thickness after deducting the corrosion allowance. Metal may be added to the inside surface of the vessel to maintain the ¼ wall thickness requirement.

The thread engagement of the studs shall be equal to or greater than the larger of d_s, or

$$0.75 d_s \frac{S_s}{S_t}$$

where d_s is the nominal diameter of the stud, S_s is the allowable stress of the stud material at design temperature, and S_t is the allowable stress of the tapped material at design temperature.
The thread engagement need not be greater than 1½ d_s.

Threaded connections that conform to ANSI/ASME B1.20.1 may be screwed into a threaded hole in a vessel wall. Table 9.1 contains the thread engagement and minimum plate thickness requirements. The threads must be standard taper pipe threads. A straight thread of equal strength may be used provided there is some sealing mechanism. A built up pad, plate, or fitting may be used to satisfy the minimum thickness and thread engagement requirements of Table UG–43.

Table 9.1 Minimum Number of Pipe Threads for Connections (Table UG–43)

Size of Pipe Connection, NPS (DN)	Threads Engaged	Minimum Plate Thickness Required, in. (mm)
½ & ¾ (DN 15 & 20)	6	0.43 (11.0)
1, 1¼ & 1½ (DN 25, 32 & 40)	7	0.61 (15.5)
2 (DN 50)	8	0.70 (17.8)
2½ & 3 (DN 65 & 80)	8	1.0 (25.4)
4-6 (DN 100-150)	10	1.25 (31.8)
8 (DN 200)	12	1.5 (38.1)
10 (DN 250)	13	1.62 (41.2)
12 (DN 300)	14	1.75 (44.5)

Threaded connections are limited to 4 inch pipe size (DN 100) for vessels that contain flammable vapors, flammable liquids at temperatures above their boiling point, and liquids with a flash point below 110°F (43°C).

Threaded connections are also limited to 3 inch pipe size (DN80) for vessels with a maximum allowable working stress greater than 125 psi (861 kPa). This limitation does not apply to plug closures used for inspection openings, end closures, or forgings that comply with UF–43.

Two inch pipe size connections may be expanded into unreinforced openings. A pipe or fitting of not more than 6 inch (152 mm) outside diameter may be expanded into reinforced openings. The connection may be made with several combinations of rolled, beaded, welded or flared connection methods. Expanded connections must be seal welded if the vessel is in flammable gas or noxious gas and liquid service.

UG–45 Nozzle Neck Thickness

The minimum thickness of a nozzle neck or any other connection is the thickness required to resist the applicable loads plus corrosion, threading, or other allowance. See UG–22 for a listing of loads that must be considered.

In addition to the above, the nozzle neck thickness, except for access and inspection openings, must also be equal to or greater than the smallest of the following plus corrosion allowance:

- The shell thickness required for resisting pressure at the attachment point of the nozzle. Both internal and external pressure must be considered. A joint efficiency of 1 may be used. Minimum thicknesses per UG–16(b) must be used if greater than the calculated thickness.
- The nominal thickness of standard wall pipe of the same pipe size as the nozzle. Use the next higher pipe size if the pipe size is not listed in the standard (ANSI/ASME B36.10M). Use the largest size listed in the standard for nozzle pipe sizes greater than those listed.

UG–53 Ligaments

When there is a series of openings in the shell of a vessel and it is not practical to reinforce them, the shell thickness should be increased to compensate for the openings. Figure 9.11 shows a shell with three longitudinal rows of openings. The metal between the holes is called the ligament, and the area ratio of the perforated shell to the unperforated shell is called the ligament efficiency. UG–53 provides rules for determining ligament efficiency. Once the minimum ligament efficiency is determined, the shell thickness is calculated using UG–27 with E equal to the minimum ligament efficiency.

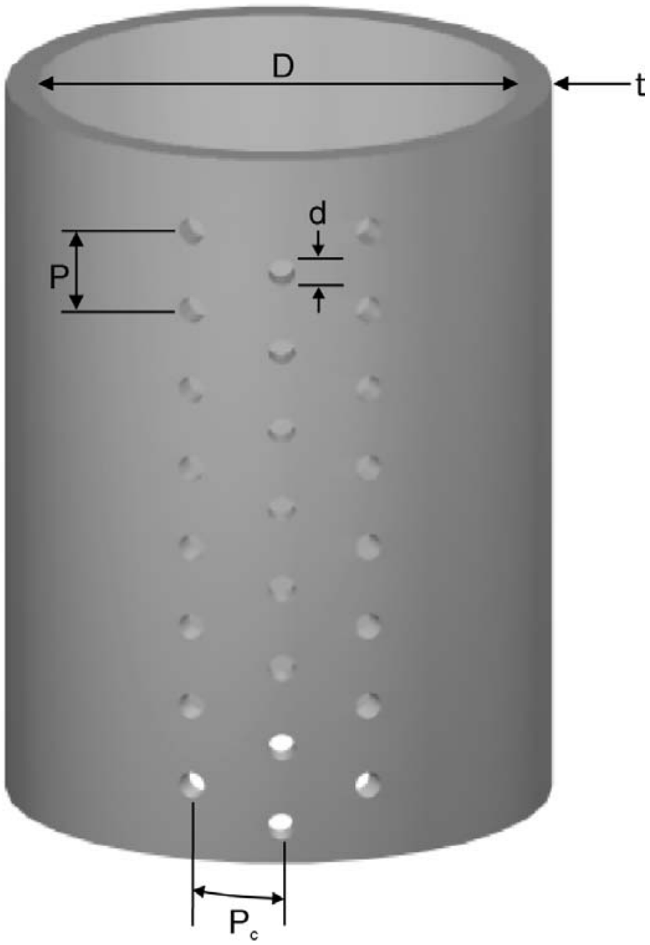


Figure 9.11 Shell with a series of openings.

UG–53(b)(1) gives the ligament efficiency when the pitch of the tube holes on every row is the same, where p is the longitudinal pitch in inches, and d is the diameter of the holes in inches.

$$E = \frac{p - d}{p}$$

The following formula must be used when the pitch of holes on any row is not equal:

$$E = \frac{p_1 - nd}{p_1}$$

where p_1 is the pitch of the pattern in inches and n is the number of holes in the pattern, Figure 9.12. In Figure 9.12, p_1 equals 8 and n is 3.

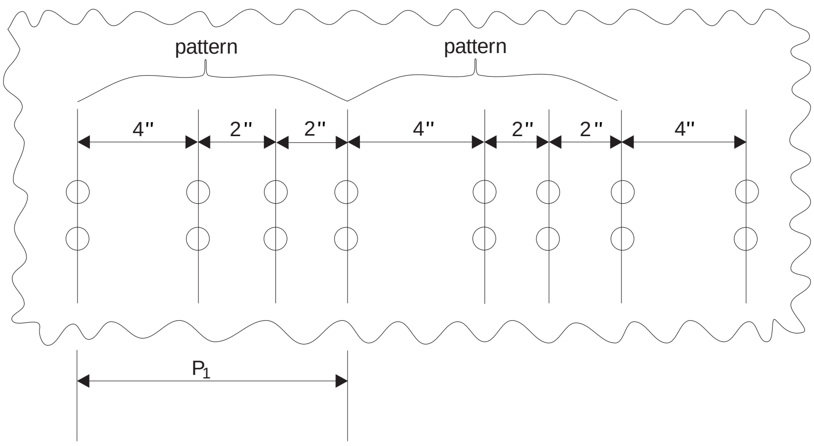
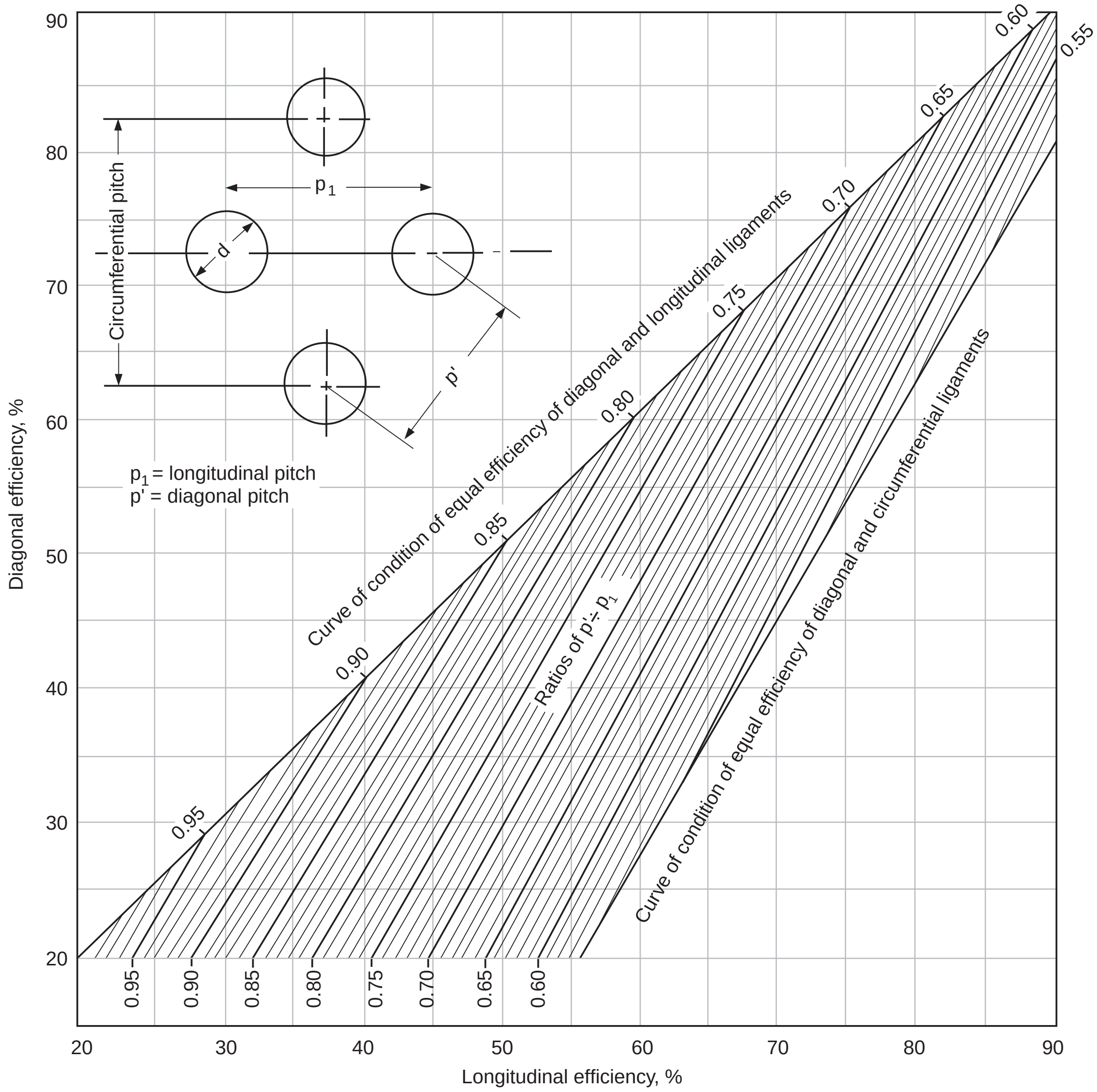


Figure 9.12 A hole pattern with an unequal pitch in the pattern.



Notes:

(1) Equations are provided for the user's option in Notes (2), (3), and (4) below. The use of these equations is permitted for values beyond those provided by Fig. UG-53.5.

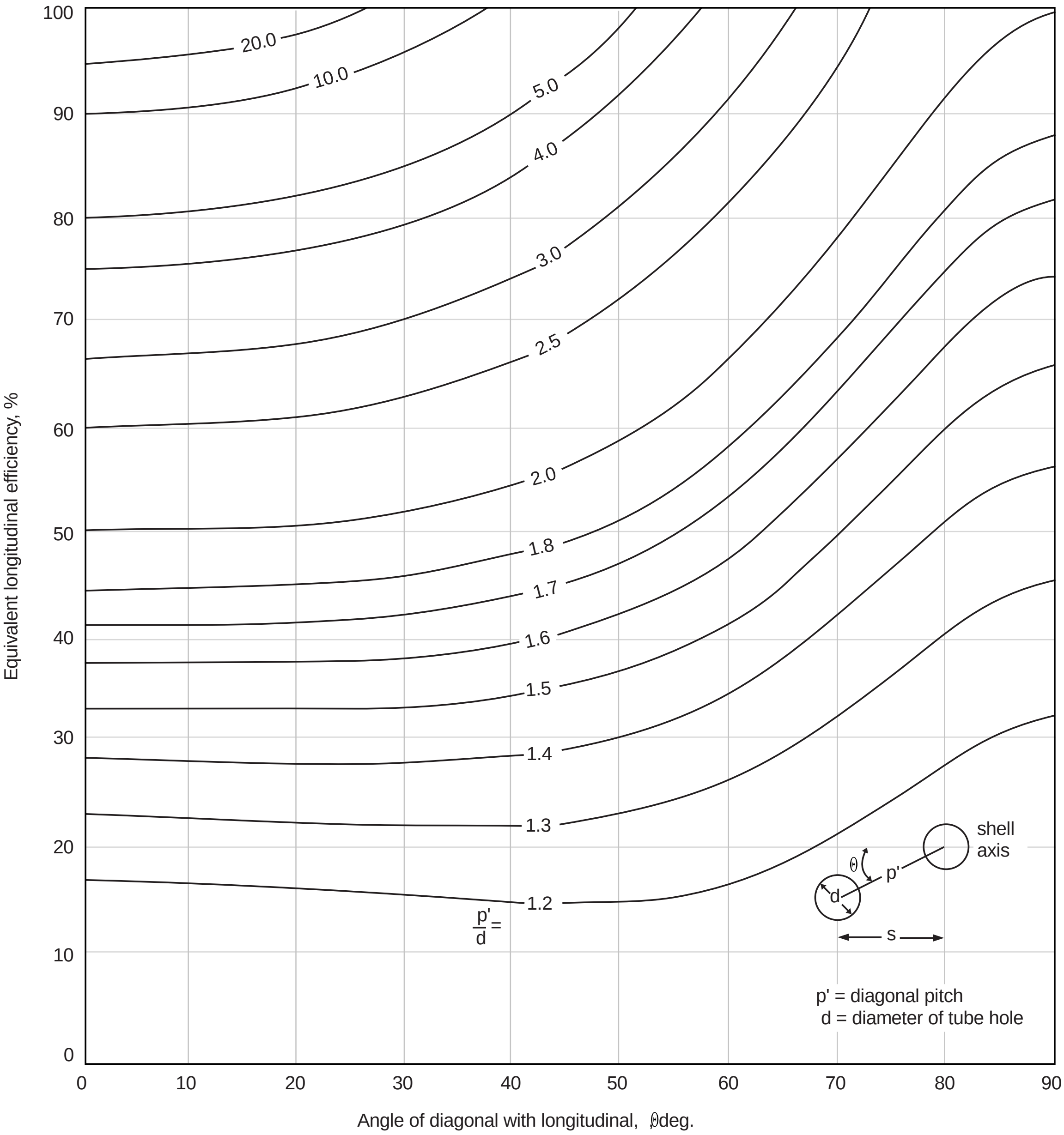
(2) Diagonal efficiency, % = $\dfrac{J + 0.25 - (1 - 0.01E_{long.})\sqrt{0.75 + J}}{0.00375 + 0.005\,J}$, where $J = (p'/p_1)^2$

(3) Curve of condition of equal efficiency of diagonal and circumferential ligaments,

diagonal efficiency, % = $\dfrac{200M + 100 - 2(100 - E_{long.})\sqrt{1 + M}}{(1 + M)}$, where $M = [(100 - E_{long.})/(200 - 0.5E_{long.})]^2$

(4) Longitudinal efficiency, % = $E_{long.} = [(p_1 - d)/p_1]100$

Figure 9.13 Diagram for determining the efficiency of longitudinal and diagonal ligaments between openings in cylindrical shells. Fig. UG–53.5.



Notes:

(1) The equation in Note (2) below is provided for the user's option. The use of the equation is prohibited beyond the range of the abscissa and ordinate shown.

(2) Equivalent longitudinal efficiency,

$$\% = \frac{\sec^2\theta + 1 - \left(\dfrac{\sec\theta}{p'/d}\right)\sqrt{3 + \sec^2\theta}}{0.015 + 0.005\sec^2\theta}$$

Figure 9.14 Diagram for determining equivalent longitudinal efficiency of diagonal ligaments between openings in cylindrical shells. Fig. UG-53.6.

UG–53 provides two figures for determining the efficiency of diagonal ligaments. Figure 9.13 is to be used when the openings are placed on diagonal lines. This figure gives a diagonal efficiency that must be used if it is less than the longitudinal efficiency. Figure 9.14 is to be used when the holes are in a longitudinal pattern but are not all on the same longitudinal line. Figure 9.14 gives an equivalent longitudinal efficiency.

The lowest efficiency from all sources, weld joint efficiency, casting factor, or ligament efficiency, must be used when determining the minimum required thickness and maximum allowable working pressure. However, when the ligament efficiency in a welded pipe or tube is less than 85% (longitudinal) or 50% (circumferential), the allowable tensile stress may be increased by 18%.

Example 9.1 16 inch Nozzle and Reinforced Opening

The 16 inch outlet nozzle at the top of the vessel shown in Figure 7.3 is fabricated from SA-106 Grade B material. The finished diameter of the opening is 15 inches. The nozzle wall is 0.500 inch thick and it is attached by welding to a ⅝ inch thick hemispherical head. The inside diameter of the head is 120 inches, and the material is SA–516 Grade 70. The opening does not pass through a Category A joint. The MAWP of the vessel is 150 psi at 800°F. The corrosion allowance is 0.125 inch. The SA–516 Grade 60 reinforcing element is ⅜ inch thick and 5.0 inches wide. The configuration is similar to Fig. UW–16.1 (h). The repad-to-nozzle weld leg is ⅜ inch and the repad-to-head weld leg is 5⁄16 inch. S_n = 10,800 psi, S_v = 12,000 psi, S_p = 10,800 psi. Is the reinforcement adequate for the intended MAWP and temperature?

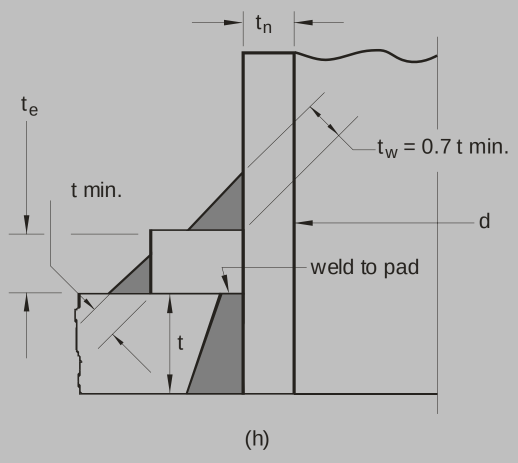


Figure 9.15 Nozzle design for Example 9.1.

Solution:

Design temperature = 800°F
Design pressure = 150 psi
Joint efficiency = 1.0
Head allowable stress S_v = 12,000 psi
Nozzle allowable stress S_n = 10,800 psi
Reinforcing pad allowable S_p = 10,800 psi

Strength Reduction Factors:

$f_{r1} = S_n/S_v$ for nozzle wall inserted through the vessel wall

$$f_{r1} = \frac{10{,}800\,\text{psi}}{12{,}000\,\text{psi}} = 0.90$$

$f_{r2} = S_n/S_v$ for nozzle material

$f_{r2} = f_{r1} = 0.90$

$f_{r4} = S_p/S_v$ for reinforcing pad material

$$f_{r4} = \frac{10{,}800\,\text{psi}}{12{,}000\,\text{psi}} = 0.90$$

f_{r3} = (lesser of S_n or S_p)/S_v

$f_{r3} = f_{r4} = 0.90$

Nozzle thickness t_n = 0.500 in. - 0.125 in. = 0.375 inch
Nozzle inside radius R_n = 7.50 in. + 0.125 in. = 7.625 inches
Vessel head thickness t = 0.625 in. - 0.125 in. = 0.500 inch
Head inside radius R_v = 60.0 inches
Reinforcing pad thickness t_e = 0.375 inch
Reinforcing pad diameter D_p = 26.0 inches
Repad-to-nozzle weld leg = 0.375 inch
Repad-to-shell weld leg = 0.3125 inch
Reinforcement is not integral, therefore F = 1.0
Opening is not in a weld, therefore E_1 = 1.0

a) Calculate limits of reinforcement:

Parallel to head, use larger of:

$d = 15.25$ inches

$R_n + t_n + t = 7.625$ in. + 0.375 in. + 0.500 in. = 8.500 inches

Use 15.25 inches

Normal to head, use smaller of:

$2.5t = 2.5 \times 0.500$ in. = 1.250 inches

$2.5t_n + t_e = 2.5\ (0.375$ in.) + 0.375 in. = 1.313 inches

Use 1.250 inches

b) Head and nozzle required thickness:

Required head thickness:

$$t_r = \frac{PR_v}{2\,(SE) - 0.2(P)}$$

$$t_r = \frac{(150.0\text{ psi})(60.125\text{ in.})}{2(12{,}000\text{ psi})\,(1) - 0.2\,(150\text{ psi})}$$

$$t_r = 0.376\text{ inch}$$

Required nozzle thickness:

$$t_{rn} = \frac{PR_n}{(SE - 0.6P)}$$

$$t_{rn} = \frac{(150\text{ psi})\,(7.625\text{ in.})}{[(10{,}800\text{ psi})\,(1) - 0.6\,(150\text{ psi})]}$$

$$t_{rn} = 0.107\text{ inch}$$

c) Determine required replacement area:

$$A = dt_rF + 2t_nt_rF\ (1\text{-}f_{r1})$$

$$A = (15.250\text{ in.})\ (0.376\text{ in.})(1.0) + 2\ (0.375\text{ in.})\ (0.376\text{ in.})\ (1.0)\ (1.0\text{-}0.90) = 5.762\text{ inches}^2$$

d) Calculate area contributions:

Area available in the head, use larger value:

$A_{11} = d\,(E_1t - Ft_r) - 2t_n\,(E_1t - Ft_r)\,(1 - f_{r1})$

A_{11} = 15.25 in.[(1)(0.500 in.)-(1)(0.376 in.)]-(2)(0.375 in.)[(1)(0.500 in.)-(1)(0.376 in.)][1.0 - 0.90]

A_{11} = 1.882 inches^2

$A_{12} = 2(t + t_n)\,(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)\,(1 - f_{r1})$

A_{12} = 2 (0.500 in. + 0.375 in.) [(1.0) (0.500 in.) - (1.0) (0.376 in.)] - 2 (0.375 in.) [(1.0) (0.500 in.) - (1.0) (0.376 in.)] (1.0-0.90)

A_{12} = 0.208 inch^2

Use A_1 = 1.882 inches^2

Area available in nozzle projecting outward, use smaller value:

$A_{21} = 5\,(t_n - t_{rn})\,(f_{r2})\,(t)$

A_{21} = 5 (0.375 in. - 0.107 in.) (0.90) (0.500 in.)

A_{21} = 0.603 inch^2

$A_{22} = 2\,(t_n - t_{rn})\,(2.5t_n + t_e)\,(f_{r2})$

A_{22} = 2 (0.375 in. - 0.107 in.) [(2.5) (0.375 in.) + 0.375 in.] (0.90)

A_{22} = 0.633 inch^2

Use A_2 = 0.603 inch^2

No inward nozzle projection, therefore A_3 = 0.00

Area available in welds:

$A_{41} = (leg)^2\,(f_{r3})$

A_{41} = (0.375 in.)^2 (0.90) = 0.126 inch^2

$A_{42} = (leg)^2 (f_{r4})$

$A_{42} = (0.3125)^2 (0.90)$

$A_{42} = 0.088 \text{ inch}^2$

$A_4 = 0.126 \text{ in.}^2 + 0.088 \text{ in.}^2$

$A_4 = 0.214 \text{ inch}^2$

Area in reinforcing pad:

$A_5 = (D_p - d - 2t_n)\, t_e f_{r4}$

$A_5 = [26.00 \text{ in.} - 15.25 \text{ in.} - (2)(0.375 \text{ in.})] (0.375 \text{ in.}) (0.90)$

$A_5 = 3.375 \text{ inches}^2$

e) Total area available:

$A_1 + A_2 + A_3 + A_{41} + A_{42} + A_5 = A_t$

$1.882 + 0.603 + 0.0 + 0.214 + 3.375 = 6.074 \text{ inch}^2 > 5.762 \text{ inches}^2$

The available reinforcing area of 6.074 inches^2 is greater than the required area of 5.762 inches^2. Therefore, area requirement is satisfied. If the available area happens to be less than the required area, then the diameter or thickness of the repad should be increased. In this example, the outside diameter of the repad can be increased to 30.5 inches and the maximum useful thickness may be increased to 1.25 inches.

Also note that once the reinforcement limits are established, both values for A_1 and A_2 need not be calculated. The value of A_1 and A_2 that correspond to the reinforcement limits will be the larger and smaller values respectfully.

f) Check weld size requirements per UW–16:

Repad to nozzle weld: Per Fig. UW–16(h) the minimum throat is $t_w = 0.7t_{min}$ where t_{min} is the smaller of ¾ inch or the thickness of the thinner of the two parts joined by the fillet. The thinner of the two parts joined is 0.375 inch. Therefore, t_{min} = 0.375 inch.

Requirement is t_w = 0.7 (0.375 in.) = 0.263 inch
Actual throat = sin 45° (0.375 in.) = 0.707 (0.375 in.) = 0.265 inch > 0.263 inch
Therefore, the weld size is acceptable.

Repad-to-head weld: from the figure, the minimum throat is ½ t_{min}. The thinner of the shell thickness of (0.500 inch) and repad thickness (0.375 inch) is 0.375 inch. Therefore t_{min} is 0.375 inch.

Requirement is 0.5(0.375 in.) = 0.188 inch
Actual throat is 0.707(0.313 in.) = 0.221 inch > 0.188 inch
Therefore, the weld size is acceptable.

The selected nozzle configuration requires an overlay fillet with a minimum throat of t_c where t_c is not less than ¼ inch or 0.7 t_{min}. For this fillet t_{min} is the smaller of 0.500 inch or 0.375 inch. t_{min} is 0.375 inch, which is greater than ¼ inch. Therefore t_c is 0.25 inch.

Weld size (leg) = 0.250 in./sin 45° = 0.250 in./0.707 = 0.354 inch

Use ⅜ inch weld.

g) Check nozzle neck thickness as specified in UG–45.

UG–45 requires that the actual nozzle thickness be equal to or greater than t_{rn} (0.107 inch from item B above), and the smaller of t_r (0.376 inch from item B above) and the thickness of 16 NPS standard wall pipe corrected for under tolerance (0.328 inch). Therefore, the actual wall thickness must be equal to or greater than 0.107 inch and 0.328 inch in the corroded condition. t_n equals 0.375 inch. Therefore, UG–45 is satisfied.

h) Calculate loads to be carried by welds:

Load to be carried by welds (Fig. UG-41.1.a):

$$W = [A - A_1 + 2t_nf_{r1} (E_1t - Ft_r)]\ S_v$$

$$W = [5.762\text{ in.}^2 - 1.880\text{ in.}^2 + (2)(0.375\text{ in.})(0.90)(0.500\text{ in.} - 0.376\text{ in.})](12{,}000\text{ psi})$$

$$W = 47{,}600\text{ pounds}$$

$$W_{1\text{-}1} = (A_2 + A_5 + A_{41} + A_{42})\ S_v$$

$$W_{1\text{-}1} = (0.603\text{ in.}^2 + 3.375\text{ in.}^2 + 0.214\text{ in.}^2)\ (12{,}000\text{ psi})$$

$$W_{1\text{-}1} = 50{,}304\text{ pounds}$$

$$W_{2\text{-}2} = (A_2 + A_3 + A_{41} + A_{43} + 2t_n\,tf_{r1})\ S_v$$

$$W_{2\text{-}2} = [0.603\text{ in.}^2 + 0.0\text{ in.}^2 + 0.126\text{ in.}^2 + 0.126\text{ in.}^2 + (2)(0.375\text{ in.})(0.50\text{ in.})(0.90)](12{,}000\text{ psi})$$

$$W_{2\text{-}2} = 14{,}310\text{ pounds}$$

$W_{3\text{-}3} = (A_2 + A_3 + A_5 + A_{41} + A_{42} + A_{43} + 2t_n\, tf_{r1})\, S_v$

$W_{3\text{-}3}$ = [0.603 in.^{2}+0.0 in.^{2}+3.375 in.^{2}+0.214 in.^{2}+0.126 in.^{2}+(2)(0.375 in.)(0.50 in.)(0.90)] (12,000 psi)

$W_{3\text{-}3}$ = 55,866 pounds

Since the weld load W is smaller than $W_{1\text{-}1}$ and $W_{3\text{-}3}$, W may be used in place of $W_{1\text{-}1}$ and $W_{3\text{-}3}$

i) Calculate strength of connection elements:

Load path 1 consists of the head-to-pad weld in shear and the nozzle neck in shear.
No credit is taken for the part of the groove weld attaching the pad to the shell.

Strength of the fillet weld is:
½ π (inside diameter of the weld)(minimum leg size)(stress factor)(allowable tensile stress)

Head-to-pad weld shear = ½ π (26.00 in.) (0.3125 in.) (0.49) (10,800 psi) = 67,540 pounds

Because this is greater than the required strength, it is not necessary to continue. However, calculations will be continued to illustrate the procedure.

The strength of the nozzle wall in shear is:
½ π (mean diameter of nozzle) (t_n) (stress factor) (allowable tensile stress)

Nozzle wall shear = ½ π (15.625 in.) (0.375 in.) (0.70) (10,800 psi) = 69,580 pounds

Total strength of load path 1 = 67,540 lb + 69, 580 lb = 137,120 pounds

Load path 2 consists of the repad-to-nozzle weld in shear, the overlay fillet at the bottom in shear, and the groove weld in tension.

Overlay fillet in shear = ½ π (16.00 in.) (0.375 in.) (0.49) (10,800 psi) = 49,870 pounds

The strength of the groove weld in tension is:

½ π (nozzle O.D.) (t) (stress factor) (allowable tensile stress)

Groove weld tension = ½ π (16.00 in.) (0.500 in.) (0.74) (12,000 psi) = 111,590 pounds

Strength of the repad-to-nozzle weld = ½ π(16.00 in.) (0.375 in.) (0.49) (10,800) = 49,870 pounds

Total strength of load path 2 = 49,870 lb + 49,870 lb + 111, 590 lb = 211,280 pounds

Load path 3 consists of the repad-to-shell weld in shear the groove weld in tension, and the overlay fillet weld in shear. Therefore from above:

Repad-to-shell weld shear = 67,540 pounds

Groove weld tension = 111,590 pounds

Overlay fillet in shear = 49,870 pounds

Total strength of load path 3 = 229,000 pounds

All paths are stronger than the required strength 47,600 pounds

The nozzle design would be incomplete without the selection of a flange. An ASME/ANSI B16.5 welding neck flange will be selected. For the design conditions of 150 psi at 800°F, ASME/ANSI B16.5 requires a class 300 flange and, as a minimum, a flange material of grade 1.4 (SA–350 Grade LF1, Class 1) meets these requirements.

Example 9.2 Design of the Manway Nozzle in Figure 7.3

The 20 inch manway nozzle shown in Figure 7.3 is fabricated from 1 inch thick SA–516 Grade 70 plate. The corrosion allowance is 0.125 inch and the inside diameter is 18 inches. The nozzle does not pass through a weld seam. The repad-to-nozzle weld is ½ inch and the repad to shell weld is 7⁄16 inch. The repad is SA–516 Grade 70, ½ inch thick plate. The design pressure is 150 psi at 800°F. Determine the reinforcement and attachment welds.

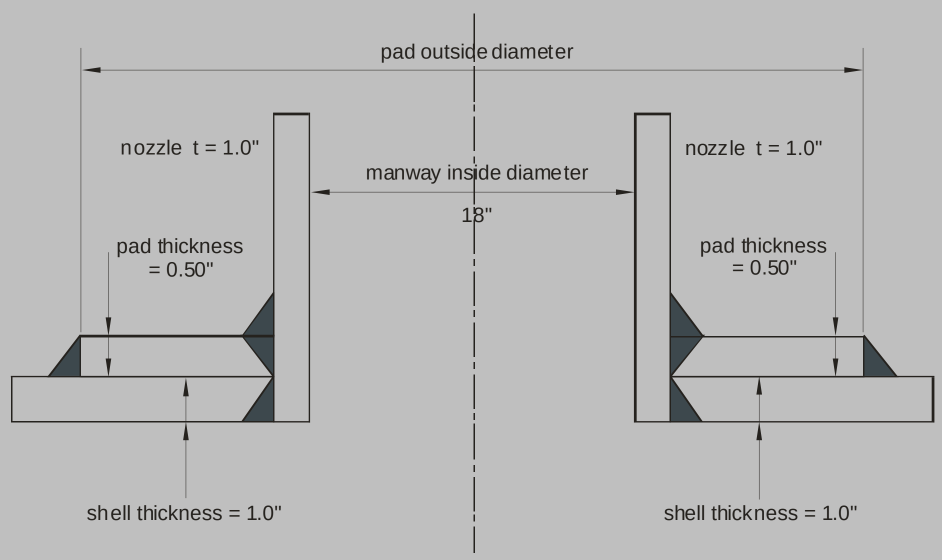


Figure 9.16 Manway nozzle reinforcement.

Solution:

Design temperature = 800°F
Design pressure = 150 psi
Joint efficiency = 1
Head allowable stress S_v = 12,000 psi
Nozzle allowable stress S_n = 12,000 psi
Reinforcing pad allowable S_p = 12,000 psi

Strength reduction factors:

$f_{r1} = f_{r2} = f_{r3} = 1$

Nozzle thickness t_n = 1.000 in. - 0.125 in. = 0.875 inch
Nozzle inside radius R_n = 9.000 in. + 0.125 in. = 9.125 inches
Vessel shell thickness t =1.00 in. - 0.125 in. = 0.875 inch
Shell inside radius R_v = 60.0 inches
Reinforcing pad thickness t_e =0.500 inch
Reinforcing pad diameter D_p = 30.0 inches
Repad-to-nozzle weld leg = 0.5 inch
Repad-to-shell weld leg = 0.4375 inch
Reinforcement is not integral, therefore F = 1
Opening is not in a weld, therefore E_1 = 1

a) Calculate limits of reinforcement:

Parallel to shell, use larger of:

d = 18.25 inches

$R_n + t_n + t$ = 9.125 in. + 0.875 in. + 0.875 in. = 10.875 inches

Use 18.25 inches

Normal to shell, use smaller of:

2.5t = 2.5 x 0.875 in. = 2.188 inches

$2.5t_n + t_e$ = 2.5 x 0.875 in. + 0.500 in. = 2.688 inches

Use 2.188 inches

b) Shell and nozzle required thickness:

Required shell thickness:

$$t_r = \frac{PR_v}{(SE - 0.6P)}$$

$$t_r = \frac{(150\text{ psi})(60.0\text{ in.})}{[(12{,}000\text{ psi})(1) - 0.6(150\text{ psi})]}$$

$$t_r = 0.756\text{ inch}$$

Required nozzle thickness:

$$t_{rn} = \frac{PR_n}{(SE - 0.6P)}$$

$$t_{rn} = \frac{(150\text{ psi})(9.125\text{ in.})}{[(12{,}000\text{ psi})(1) - 0.6(150\text{ psi})]}$$

$$t_{rn} = 0.115\text{ inch}$$

c) Determine required replacement area:

$$A = dt_rF + 2t_nt_rF\ (1\text{-}f_{r1})$$

$$A = (18.250\text{ in.})\ (0.756\text{ in.})\ (1.0)$$

$$A = 13.80\text{ inches}^2$$

d) Calculate area contributions: Area available in the shell, use larger value.

$$A_{11} = d(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1})$$

$$A_{11} = 18.25\text{ in.}[(1)(0.875\text{ in.}) - (1)(0.756\text{ in.})]$$

$$A_{11} = 2.172\text{ inches}^2$$

$$A_{12} = 2(t + t_n)(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1})$$

$$A_{12} = 2(0.875\text{ in.} + 0.875\text{ in.})[(1.0)(0.875\text{ in.}) - (1.0)(0.756\text{ in.})]$$

$$A_{12} = 0.417\text{ inch}^2$$

Use $A_1 = 2.172\text{ inches}^2$

Area available in nozzle projecting outward, use smaller value.

$A_{21} = 5(t_n - t_{rn})(f_{r2}t)$

$A_{21} = 5(0.875 \text{ in.} - 0.115 \text{ in.})(1.00)(0.875 \text{ in.})$

$A_{21} = 3.325 \text{ inches}^2$

$A_{22} = 2(t_n - t_{rn})(2.5t_n + t_e)(f_{r2})$

$A_{22} = 2(0.875 \text{ in.} - 0.115 \text{ in.})[(2.5)(0.875 \text{ in.}) + 0.500 \text{ in.}](1.00)$

$A_{22} = 4.085 \text{ inches}^2$

Use $A_2 = 3.325 \text{ inches}^2$

No inward nozzle projection, therefore $A_3 = 0.00$

Area available in welds:

$A_{41} = (\text{leg})^2 (f_{r3})$

$A_{41} = (0.500 \text{ in.})^2 (1.00)$

$A_{41} = 0.250 \text{ inch}^2$

$A_{42} = (\text{leg})^2 (f_{r4})$

$A_{42} = (0.438 \text{ in.})^2 (1.00)$

$A_{42} = 0.192 \text{ inch}^2$

$A_4 = 0.250 \text{ inch}^2 + 0.192 \text{ inch}^2$

$A_4 = 0.442 \text{ inch}^2$

Area in reinforcing pad:

$A_5 = (D_p - d - 2t_n)(t_e)(f_{r4})$

$A_5 = [30.00 \text{ in.} - 18.25 \text{ in.} - (2)(0.875 \text{ in.})] (0.500 \text{ in.}) (1.00) = 5.000 \text{ inches}^2$

e) Total area available:

$$A_1 + A_2 + A_3 + A_{41} + A_{42} + A_5 = A_t$$

$$2.172 \text{ in.}^2 + 3.325 \text{ in.}^2 + 0.0 + 0.442 \text{ in.}^2 + 5.000 \text{ in.}^2 = 10.939 \text{ inch}^2 < 13.790 \text{ inches}^2$$

f) Increase repad diameter to 36.00 inches:

$$A_5 = [36.00 \text{ in.} - 18.25 \text{ in.} - (2)(0.875 \text{ in.})](0.500 \text{ in.})$$

$$A_5 = 8.00 \text{ inches}^2$$

g) Revised total area available:

$$2.172 \text{ in.}^2 + 3.325 \text{ in.}^2 + 0.0 + 0.442 \text{ in.}^2 + 8.000 \text{ in.}^2 = 13.939 \text{ inch}^2 > 13.790 \text{ inches}^2$$

h) Check weld size requirements:

The minimum throat of the repad–to-nozzle weld is t_c where t_c equals 0.25 inch.
Actual throat is 0.707(0.500 in.) = 0.354 inch.
Weld size is acceptable.

The minimum throat of the repad-to-shell weld is ½ t_{min} where t_{min} equals 0.500 inch.
Actual throat is 0.707(0.438 in.) = 0.310 inch.
Weld size is acceptable.

i) Calculate load paths:

Nozzle configuration is similar to Fig. UW-16(a-1). UW-15(b) exempts Fig. UW-16(a-1) from strength calculations.

The flange for this nozzle will be designed in Chapter 10.

Example 9.3 Maximum Design Pressure Using Ligament Efficiency

The inside diameter of the vessel in Figure 9.17 is 40 inches and the material is SA–516 Grade 60. The wall thickness is ⅞ inch in both the drilled and non-drilled section of the vessel. The diameter of the holes is 3.53 inches, the longitudinal pitch is 6 inches, and the circumferential pitch is 4.5 inches. The vessel is spot radiographed according to UW–52. The design temperature is 750°F. The corrosion allowance is 0.10 inch.

Determine the maximum design pressure of the vessel.

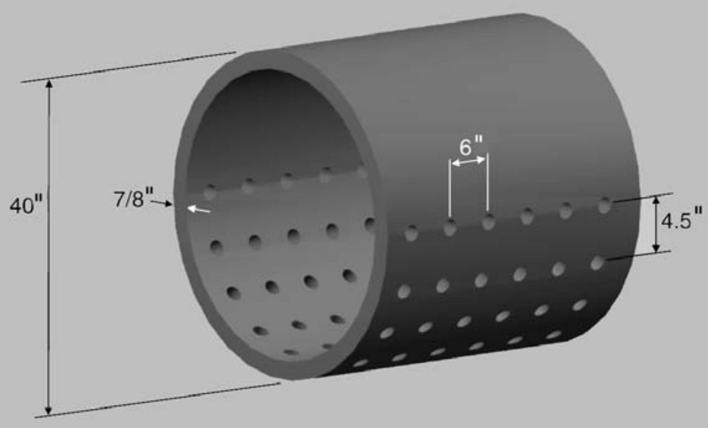


Figure 9.17 40 inch vessel with 3.53 inch drilled holes on lower half.

Solution:

a) Determine the ligament efficiency using UG–53:

Longitudinal efficiency:

$$E = \frac{p - d}{p}$$

$$E = \frac{6.0\text{ in.} - 3.53\text{ in.}}{6.0\text{ in.}}$$

$$E = 0.412$$

Circumferential efficiency:

$$E_c = \frac{p_c - d}{p_c}$$

$$E_c = \frac{4.50\text{ in.} - 3.53\text{ in.}}{4.50\text{ in.}}$$

$$E_c = 0.216$$

b) Determine design pressure using UG–27. The allowable stress S equals 13,000 psi from Table 1A of Section II, Part D. The weld joint efficiency for spot radiography is 0.85 from UW–12.

Non-drilled shell:

$$P = \frac{SEt}{R + 0.6t}$$

$$P = \frac{13{,}000\ \text{psi}\ (0.85)\ (0.875\ \text{in.} - 0.10\ \text{in.})}{[20.0\ \text{in.} + 0.6\ (0.875\ \text{in.} - 0.10\ \text{in.})]}$$

$$P = 418\ \text{psi}$$

Drilled shell section, circumferential stress:

$$P = \frac{SEt}{R + 0.6t}$$

$$P = \frac{13{,}000\ \text{psi}\ (0.412)\ (0.875\ \text{in.} - 0.10\ \text{in.})}{[20.0\ \text{in.} + 0.6\ (0.875\ \text{in.} - 0.10\ \text{in.})]}$$

$$P = 203\ \text{psi}$$

Drilled shell section, longitudinal stress:

$$P = \frac{2SEt}{R - 0.4t}$$

$$P = \frac{2\ (13{,}000\ \text{psi})\ (0.216)\ (0.875\ \text{in.} - 0.10\ \text{in.})}{[20.0\ \text{in.} - 0.4\ (0.875\ \text{in.} - 0.10\ \text{in.})]}$$

$$P = 221\ \text{psi}$$

The lowest value of P, 203 psi, is the maximum design pressure of the vessel. Note that the often used practice of calculating an equivalent longitudinal efficiency in the circumferential direction of $2E_c$ or 2(0.216) = 0.432, and selecting the smaller longitudinal efficiency can be misleading. Different results are obtained because of corrosion allowance and the correction factors in the stress equations.

Example 9.4 Thickness Determination for Ligaments

Determine the minimum required thickness of a 40 inch ID cylindrical shell using the ligament efficiency rules of UG–53. The holes have a 3.53-inch diameter and are on a 37.5° diagonal. The design pressure is 200 psi at 750°F. The corrosion allowance is 0.125 inch. The material is SA–515 Grade 70.

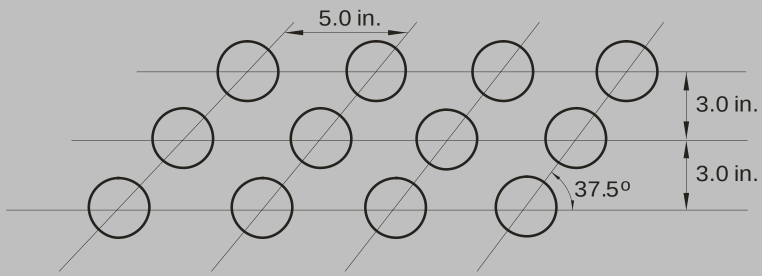


Figure 9.18 Holes in shell for Example 9.4.

Solution:

a) Determine the efficiency:

diagonal pitch = P' = 3.00 in./sin 37.5° = 4.93 inches

P'/P_l = 4.93 in./5.00 in. = 0.986

$E_l = (P_l - d)/P_l$ = (5.00 in. - 3.53 in.)/5.00 in.

$E_l = 0.294$

In Fig. UG—53.5, the vertical line of E_l = 0.294 intersects the P'/P_l value of 0.986 above the curve of equal efficiency. Therefore, the longitudinal efficiency is less than the diagonal efficiency. Use E_l in calculations.

b) Determine thickness using UG-27:

$$t = \frac{PR}{SE - 0.6P} + CA$$

$$t = \frac{(200\text{ psi})(20.125\text{ in.})}{(14{,}800\text{ psi})(0.294) - (0.6)(200\text{ psi})} + 0.125\text{ in.}$$

$$t = 1.08\text{ inches}$$

The formula in note 2 of Fig. UG—53.5 may be used to calculate the diagonal efficiency. For this example the diagonal efficiency is 0.343. Note that the value referred to as the diagonal efficiency is the equivalent longitudinal efficiency of the diagonal direction.

Example 9.5 Openings in a Flat Head

The nominal thickness of the flat head in Example 8.5 is 3.00 inches. Two SA–106 Grade B schedule 60 pipes are inserted as nozzles. The nozzle attachment is similar to UW–16.1(e). All fillet welds are ¾ inch. Design the reinforcement. The corrosion allowance is 0.125 inch.

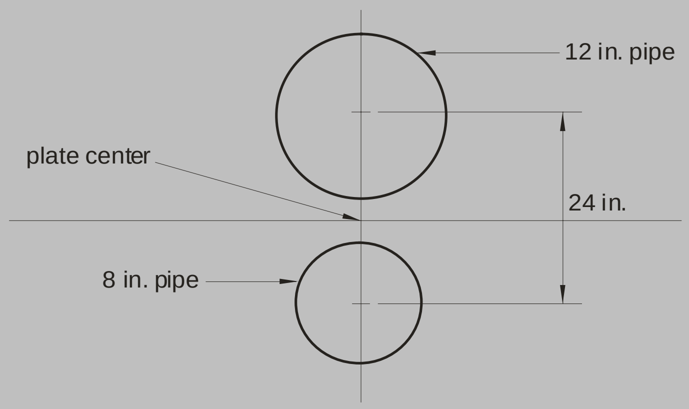


Figure 9.19 Holes in flathead for Example 9.5.

Solution:

a) Check diameter and spacing requirements of UG–39(b)(2):

Head diameter/2 = D/2 = 48 in./2 = 24 inches

Head diameter/4 = D/4 = 48 in./4 = 12 inches

Both pipes have a nominal thickness of 0.406 inch. Actual thickness can be 12½% less because of material specification permitted under tolerance for pipe. Therefore, opening diameters are:

d_{12} = 12.75 in. - 2[(0.406 in.) (0.875) + 0.125 in.]

d_{12} = 11.790 inches

d_8 = 8.625 in. - 2[(0.406 in.) (0.875) + 0.125 in.]

d_8 = 7.665 inches

Average diameter of openings is (11.790 in. + 7.665 in.)/2 = 9.73 inches

Both diameters are less than D/2 and the average diameter of the openings is less than D/4. Therefore, they may be reinforced as single openings, but at least 50% of the total reinforcement for the openings must be placed between the two openings.

b) Determine the required area:

From Example 8.5, t_r for the flat plate is 2.76 inches.
The allowable stress for the nozzles is 10,800 psi.

$f_{r1} = S_n/S_v$ for nozzle wall inserted through the vessel wall

$$f_{r1} = \frac{10{,}800\,\text{psi}}{12{,}000\,\text{psi}} = 0.90$$

$f_{r3} = f_{r2} = f_{r1} = 0.90$

$A = 0.5dt_r + t_r t_n(1 - f_{r1})$

$A_{12} = 0.5\ (11.790\text{ in.})\ (2.76\text{ in.}) + (2.76\text{ in.})\ (0.281\text{ in.})\ (1 - 0.90)$

$A_{12} = 16.348\text{ inches}^2$

$A_8 = 0.5\ (7.665\text{ in.})\ (2.76\text{ in.}) + (2.76\text{ in.})\ (0.281\text{ in.})\ (1 - 0.90)$

$A_8 = 10.655\text{ inches}^2$

Total $A = 10.655\text{ inch}^2 + 16.348\text{ inch}^2 = 27.003\text{ inches}^2$

c) Calculate limits of reinforcement:

Limits for 12 inch nozzle.

Parallel to head, use larger of two values:

d = 11.79 inches

$R_n + t_n + t = 5.90\text{ in.} + 0.281\text{ in.} + 2.875\text{ in.} = 9.056\text{ inches}$

Use 11.79 inches

Normal to head, use smaller of:

$2.5t = 2.5 \times 2.875\text{ in.} = 7.188\text{ inches}$

$2.5t_n + t_e = 2.5 \times 0.281\text{ in.} + 0 = 0.702\text{ inch}$

Use 0.702 inch

Limits for 8 inch nozzle.

Parallel to head, use larger of two valves:

d = 7.665 inches

$R_n + t_n + t = 3.833 \text{ in.} + 0.281 \text{ in.} + 2.875 \text{ in.} = 6.989$ inches

Use 7.665 inches

Normal to head, use smaller of:

2.5t = 2.5 x 2.875 in. = 7.188 inches

$2.5t_n + t_e = 2.5 \text{ x} 0.281 \text{ in.} + 0 = 0.702$ inch

Use 0.702 inch

d) Calculate area contributions:

Required thickness for the 12 inch nozzle.

$$t_{rn} = \frac{PR_n}{(SE - 0.6P)}$$

$$t_{rn} = \frac{(150 \text{ psi})(5.90 \text{ in.})}{[(10{,}800 \text{ psi})(1) - 0.6\,(150 \text{ psi})]}$$

$$t_{rn} = 0.083 \text{ inch}$$

Required thickness for the 8 inch nozzle.

$$t_{rn} = \frac{PR_n}{(SE - 0.6P)}$$

$$t_{rn} = \frac{(150 \text{ psi})\,(3.833 \text{ in.})}{[(10{,}800 \text{ psi})\,(1) - 0.6\,(150 \text{ psi})]}$$

$$t_{rn} = 0.054 \text{ inch}$$

Area available in the head where A_{11} is for the 12 inch nozzle and A_{12} is for the 8 inch nozzle.

$$A_1 = d(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1})$$

$$A_{11} = 11.79\text{ in.}[(1)(2.875\text{ in.}) - (1)(2.760\text{ in.})] - (2)(0.281\text{ in.})[(1)(2.875\text{ in.}) - (1)(2.760\text{ in.})][1 - 0.90]$$

$$A_{11} = 1.349\text{ inches}^2$$

$$A_{12} = 7.665\text{ in.}[(1)(2.875\text{ in.}) - (1)(2.760\text{ in.})] - (2)(0.281\text{ in.})[(1)(2.875\text{ in.}) - (1)(2.760\text{ in.})][1 - 0.90]$$

$$A_{12} = 0.875\text{ inch}^2$$

Area available in nozzle projecting outward.

$$A_{21} = 5\,(t_n - t_{rn})(f_{r2})(t_n)$$

$$A_{21} = 5(0.281\text{ in.} - 0.083\text{ in.})(0.90)(0.281\text{ in.})$$

$$A_{21} = 0.250\text{ inch}^2$$

$$A_{22} = 5\,(t_n - t_{rn})(f_{r2})(t_n)$$

$$A_{22} = 5(0.281\text{ in.} - 0.083\text{ in.})(0.90)(0.281\text{ in.})$$

$$A_{22} = 0.250\text{ inch}^2$$

No inward nozzle projection, therefore $A_3 = 0.00$.

Area available in welds.

$$A_{41} = (\text{leg})^2\,(f_{r3})$$

$$A_{41} = (0.750\text{ in.})^2\,(0.90)$$

$$A_{41} = 0.506\text{ inch}^2$$

e) Total area available without repad:

For the 12 inch nozzle:

$$A_t = A_1 + A_2 + A_3 + A_{41}$$

$$A_t = 1.349 \text{ in.}^2 + 0.250 \text{ in.}^2 + 0.0 \text{ in.}^2 + 0.506 \text{ in.}^2 = 2.105 \text{ inch}^2$$

$$A_t < 16.348 \text{ inches}^2$$

For the 8 inch nozzle:

$$A_t = A_1 + A_2 + A_3 + A_{41}$$

$$A_t = 0.875 \text{ in.}^2 + 0.250 \text{ in.}^2 + 0.0 \text{ in.}^2 + 0.506 \text{ in.}^2 = 1.631 \text{ inches}^2$$

$$A_t < 10.655 \text{ inches}^2$$

The reinforcement available in the head and nozzle is less than the amount necessary to adequately reinforce the openings. Additional reinforcement in the form of a reinforcing pad or a thicker head as outlined in UG–39(d)(e) is necessary. If a reinforcement pad is added, 50% of the total area (13.5 inch^2) must be placed between the nozzles.

f) Design a thicker head according to UG–39(e) to compensate for the openings:

Space between openings = 24.00 in. - (5.900 in. + 3.833 in.) = 14.267 inch > 1¼ average diameter

UG–39(d) requires the use of Formula 1 in UG–34(c) with a value of C equal to the smaller of 2C or 0.50 for Fig. UG–34(f). The value of C from Example 8.5 is 0.264. Therefore, use formula with a modified C value of 0.50.

$$t_{Base} = 48.0\,\text{in.}\sqrt{(0.500)(150\,\text{psi})/12{,}000\,\text{psi}}$$

$$t_{Base} = 3.795 \text{ inches}$$

Paragraph UG–39(e)(2) requires that t_{BASE} be multiplied by h.

$$e = \frac{24.00 - 9.73}{24.00}$$

$$e = 0.594$$

$$h = \sqrt{\frac{0.50}{e}}$$

$$h = \sqrt{\frac{0.50}{0.594}}$$

$$h = 0.917$$

$$t_{req} = 0.917\ (3.795\text{ in.})$$

$$t_{req} = 3.480\text{ inches}$$

The corrosion allowance of 0.125 inch must be added to this thickness to determine the minimum nominal thickness.

$$t = 3.480\text{ in.} + 0.125\text{ in.}$$

$$t = 3.610\text{ inches}$$

The presence of holes in this flat head requires a 26% increase in required thickness if one elects the option of a thicker head.

Chapter

10

APPENDIX 2 - RULES FOR BOLTED FLANGE CONNECTIONS WITH RING TYPE GASKETS

General

Section VIII, Division 1 presents several design procedures and methods in both the Mandatory and Nonmandatory Appendices. Design procedures in the appendices differ from those in the body of the Code in that most of the complex procedures in the appendices are illustrated with examples. An exception to this rule is Appendix 2. It is fairly complex and does not have examples.

Appendix 2 provides the Code method for designing multibolted flanges. Paragraph 2–1(c) recommends that bolted flange connections in accordance with the standards listed in UG–44 be used within the material, size, and pressure-temperature ratings listed in UG–44. UCI–3 and UCD–3 also have restrictions on the pressure-temperature ratings of standard flanges.

UG–44 lists the following standards as acceptable:

- ASME/ANSI B16.5, Pipe Flanges and Flanged Fittings
- ASME B16.20 Metallic Gaskets for Pipe Flanges–Ring–Joints, Spiral–Wound, and Jacketed
- ASME B16.24, Cast Copper Alloy Pipe Flanges and Fittings, Class 150, 300, 400, 600, 900, 1500 and 2500
- ASME/ANSI B16.42, Ductile Iron Pipe Flanges and Flanged Fittings, Class 150 and 300
- ASME B16.47, Large Diameter Steel Flanges, NPS 26 Through NPS 60.

Table U–3 must be referenced for the correct edition of the above standards. Paragraph UG–44 and Appendix 2 reference UG–11 which provides for manufacturer standards.

The design procedure in Appendix 2 must be used when a standard flange is not available or is inadequate, when the standard pressure-temperature ratings are not adequate, or when special design conditions such as materials, gaskets, or loads must be considered. The design procedure considers only pressure and cautions the user that allowances must be made if external loads other than pressure exist.

The Appendix 2 procedure is a trial and error method for designing a bolted flange system consisting of bolting, gasket, and flanges. The designer must select the type of gasket, materials, and all flange dimensions including flange thickness. Table 2–5.1 lists recommended Gasket

Factors, m, and Minimum Design Seating Stress, y, for different types of gaskets. The m and y values are only recommendations and may be replaced with other values if appropriate.

Flanges are designed either as loose type flanges, integral type flanges, or optional type flanges (see Fig. 2–4 Types of Flanges). Loose type means no attachment to the pipe or, if attached, then no ability to transfer load through the attachment. Integral type means that the ring, hub, and pipe are one continuous component. Optional type flanges are those which by choice can be designed as integral or loose type.

Design Procedure

After determining the design pressure and design temperature, the following seven steps should be followed to design a flange:

1. Select the flange material and bolt material and determine the allowable stress at both ambient and operating temperature.
2. Estimate the dimensions, including thickness, and select the flange facing and gasket details.
3. Determine an equivalent pressure if external loads exist by converting the external loads to a pressure and adding it to the internal pressure. Calculate the required bolt area and select the bolt size.
4. Calculate all flange loads, moment arms, and moments for both gasket seating and operating conditions.
5. With the flange dimensions, calculate the shape constants and read the appropriate stress factors from the curves given in Fig. 2–7.1, 2–7.2, 2–7.3, 2–7.4, 2–7.5, and 2–7.6. The stress factors may be calculated using the formulas given in Table 2–7.1.
6. Calculate the longitudinal hub stress, radial flange stress, tangential flange stress, and the required combinations.
7. Compare the calculated stresses to the allowable stresses. If the calculated or actual stresses are greater than the allowables, adjust the dimensions and repeat the process until the stresses are within an acceptable range.

A single split loose flange ring shall be designed as if it were a solid ring using 200% of the total moment M_0. If the flange consists of two split rings, then each ring shall be designed as if it were a solid ring using 75% of the total moment M_0. The splits should be 90° from each other.

Noncircular shaped flanges with a circular bore shall be designed as a circular flange with the outside diameter A equal to the diameter of the largest circle inscribed within the outside edges of the flange and concentric with the bore. The equivalent bolt circle must pass through the center of the outermost bolt holes. The same may be applied to noncircular bores.

The outside diameter A of flanges that have slotted bolt holes may be taken as the diameter of a circle tangent to the inner edge of the slots.

As stated, Appendix 2 requires that appropriate allowances be made for external loads. An often used conservative approach is to calculate an equivalent pressure P_e using the following formula:

$$P_e = \frac{16M}{\pi G^3} + \frac{4F}{\pi G^2} + P$$

where:
M = bending moment, in.-lb (bending and torsional moments must be considered separately)
F = radial force, lb
G = diameter, inches, at location of gasket load reaction
P = internal design pressure, psi.

The first example, Example 10.1, is the design of a bolted flat head for the 48 inch nozzle shown in Figure 7.3. The bolted flat head, covered in UG–34, requires that W, the total bolt load, be calculated for both gasket seating and operating conditions.

Example 10.1 Design of a Bolted Flat Head

Design a bolted flat head cover (blind flange) for the 48 inch nozzle on the vessel shown in Figure 7.3. The design pressure is 150 psi at 800°F. The corrosion allowance is 0.125 inch. The head, Figure 10.1, is fabricated from SA–516 Grade 70 material with full radiography. The bolting material is SA–193 Grade B7. The gasket is one inch wide spiral-wound steel, filled with asbestos. Figure 10.1 shows the head and shell configuration.

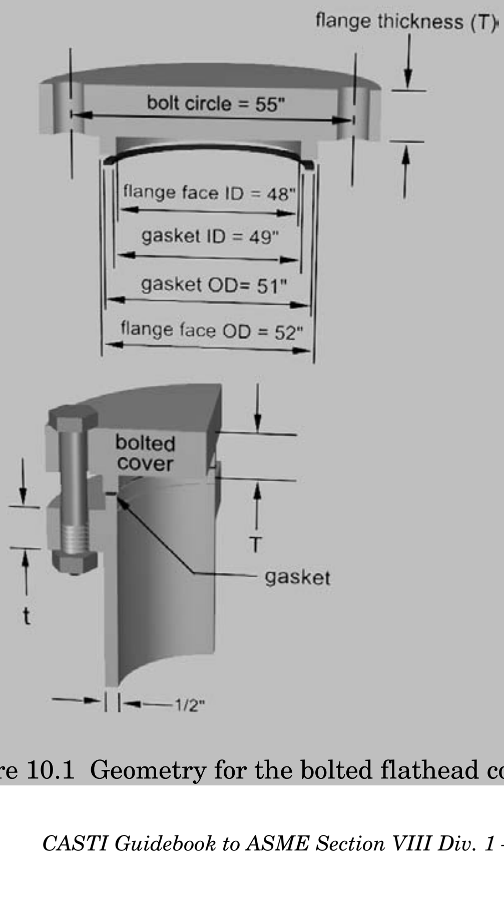


Figure 10.1 Geometry for the bolted flathead cover.

Solution:

From Table 2–5.1, the gasket factor is m = 2.5 and y = 10,000 psi. From Section II, Part D, the allowable stress for the bolts at ambient and operating are Sb_a = 25,000 psi and Sb_o = 21,000 psi. The allowable stress for the flange material is 20,000 psi at ambient and 12,000 psi at operating.

Determine the effective gasket seating width and gasket load diameter. Use Tables 2-5.1 and 2-5.2.

N = 1.0 inch from Table 2-5.2
b_o = N/2 = 1.0 in./2 = 0.5 inch from Table 2-5.2

$$b = 0.5\sqrt{b_o} = 0.5\sqrt{0.5\text{ in.}}$$

b = 0.3535 inch

$$G = G_{OD} - 2b$$

G = 51.0 in. -2(0.3535 in.)
G = 50.293 inches

Determine bolt loads per paragraph 2-5.

$$H = 0.785G^2p$$

$H = 0.785\ (50.293\text{ in.})^2\ (150\text{ psi})$
H = 297,800 pounds

$$H_p = 2b\pi Gmp$$

$H_p = 2(0.3535\text{ in.})\ \pi\ (50.293\text{ in.})\ (2.5)\ (150\text{ psi})$
H_p = 41,900 pounds

$W_{m1} = H + H_p$ = 297,800 lb + 41,900 lb
W_{m1} = 339,700 pounds

$W_{m2} = \pi bGy = \pi\ (0.3535\text{ in.})\ (50.293\text{ in.})\ (10{,}000\text{ psi})$
W_{m2} = 558,500 pounds

Calculate total bolt area and determine the number and size of bolts.

$$A_{m1} = \frac{W_{m1}}{Sb_o}$$

$$A_{m1} = \frac{339{,}700\text{ lb}}{21{,}000\text{ psi}} = 16.178\text{ inches}^2$$

$$A_{m2} = \frac{W_{m2}}{Sb_a}$$

$$A_{m2} = \frac{558{,}500 \text{ lb}}{25{,}000 \text{ psi}} = 22.340 \text{ inches}^2$$

The largest required area A_m = 22.34 in^2. Use 36 1½-inch bolts. A_b = 32.04 inches^2.

Determine gasket seating design load.

$$A = 0.5\,(A_m + A_b)$$
$$A = 0.5\,(23.34 \text{ in.}^2 + 32.04 \text{ in.}^2) = 27.69 \text{ inches}^2$$

$$W_a = A\,(S_a)$$
$$W_a = 27.69 \text{ in.}^2\,(25{,}000 \text{ psi}) = 692{,}250 \text{ pounds}$$

Determine the minimum thickness using Equation 2 in UG–34.

$$t = d\sqrt{\frac{cP}{SE} + \frac{1.9Wh_G}{SEd^3}}$$

c = 0.30, d = G, and C = flange OD

$$h_G = 0.5\,(C - G) = 0.5\,(55.00 \text{ in.} - 50.293 \text{ in.})$$
$$h_G = 2.3535 \text{ inches}$$

For gasket seating only, P = 0 and W = W_a therefore:

$$t = 50.293 \text{ in.}\sqrt{\frac{1.9\,(692{,}250 \text{ lb})(2.3535 \text{ in.})}{(20{,}000 \text{ psi})(1)(50.293 \text{ in.})^3}}$$
$$t = 1.754 \text{ inches}$$

For operating conditions:

$$t = 50.293 \text{ in.}\sqrt{\frac{(0.30)(150 \text{ psi})}{(12{,}000 \text{ psi})(1)} + \frac{1.9\,(339{,}700 \text{ lb})(2.3535 \text{ in.})}{(12{,}000 \text{ psi})(1)(50.293 \text{ in.})^3}}$$
$$t = 3.464 \text{ inches}$$

$$t_{req} = t + CA$$
$$t_{req} = 3.464 \text{ in.} + 0.125 \text{ in.} = 3.589 \text{ inches}$$

Therefore, the operating condition controls and the minimum required thickness, including corrosion allowance, is 3.589 inches. Note that the bolted flat head is thicker than the welded head in Example 8.5.

Example 10.2 Design of a Non-Standard Manway Flange

What is the minimum thickness of the manway flange in Figure 10.2? The nozzle and opening are designed in Example 9.2. The flange material is SA–516 Grade 70 and 24 1¼ inch bolts are used. The bolting material is SA–193 Grade B7. The one inch gasket is carbon steel spiral wound asbestos filled. The flange is an integral type as shown in Fig. 2–4, item 7 of ASME Section VIII, Division 1. See the following figure for dimensions. The design information is repeated for convenience:

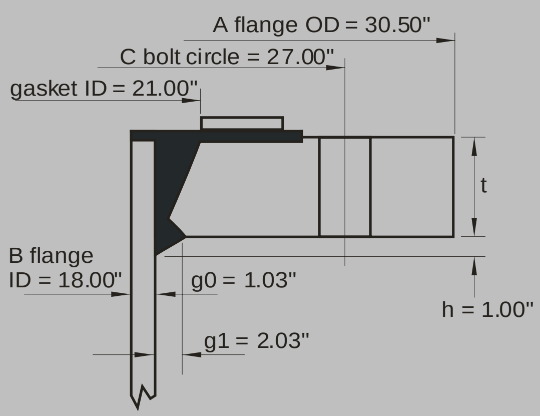


Figure 10.2 Twenty inch manway.

Design temperature = 800°F
Design pressure = 150 psi
Bolt-up temperature = 85°F
Allowable flange stress at design = 12,000 psi
Allowable flange stress at bolt-up temperature = 20,000 psi
Allowable bolt stress at design = 21,000 psi
Allowable bolt stress at bolt-up temperature = 25,000 psi
Gasket factor m = 2.5
Gasket design seating stress y = 10,000 psi

Solution:

Calculate gasket dimensions.

N = 1.0 inch From Table 2-5.2

$b_o = N/2 = 1.0 \text{ in.}/2 = 0.5$ inch From Table 2–5.2

Gasket OD = 21.00 in. + 2(1 in.) = 23.00 inches

$$b = 0.5\sqrt{b_o}$$
$$b = 0.5\sqrt{0.5 \text{ in.}}$$
$$b = 0.3535 \text{ inch}$$

$G = G_{OD} - 2b$

$G = 23.0\text{ in.} - 2\,(0.3535\text{ in.})$

$G = 22.293\text{ inches}$

Calculate the bolt loads per paragraph 2-5.

$H = 0.785G^2p$

$H = 0.785\,(22.293\text{ in.})^2\,(150\text{ psi})$

$H = 58{,}520\text{ pounds}$

$H_p = 2b\pi Gmp$

$H_p = 2(0.3535\text{ in.})\,\pi\,(22.293\text{ in.})\,(2.5)\,(150\text{ psi})$

$H_p = 18{,}570\text{ pounds}$

$W_{m1} = H + H_p = 58{,}520\text{ lb} + 18{,}570\text{ lb}$

$W_{m1} = 77{,}090\text{ pounds}$

$W_{m2} = \pi bGy = \pi\,(0.3535\text{ in.})\,(22.293\text{ in.})\,(10{,}000\text{ psi})$

$W_{m2} = 247{,}580\text{ pounds}$

Calculate total bolt area and compare to actual bolt area.

$$A_{m1} = \frac{W_{m1}}{Sb_o}$$

$$A_{m1} = \frac{77{,}090\text{ lb}}{21{,}000\text{ psi}}$$

$$A_{m1} = 3.67\text{ inches}^2$$

$$A_{m2} = \frac{W_{m2}}{Sb_a}$$

$$A_{m2} = \frac{247{,}580\text{ lb}}{25{,}000\text{ psi}}$$

$$A_{m2} = 9.903\text{ inches}^2$$

The largest required area is 9.903 inches^2. The actual area is 22.296 inches^2. Therefore, bolting is adequate.

Determine gasket seating design load.

$A = 0.5\,(A_m + A_b)$

$A = 0.5\,(9.903\text{ in.}^2 + 22.296\text{ in.}^2)$

$A = 16.10\text{ inches}^2$

$W_a = A\,(Sb_a)$

$W_a = 16.10\text{ in.}^2\,(25{,}000\text{ psi})$

$W_a = 402{,}500\text{ pounds}$

Calculate the flange moment for bolt-up condition.

Moment arm

$h_G = 0.5\ (C\text{-}G)$

$h_G = 0.5\ (27.00\text{ in.} - 22.293\text{ in.})$

$h_G = 2.354$ inches

Flange load

$H_G = W_a$

$H_G = 402{,}500$ pounds

Flange moment

$M_{GS} = H_G\ (h_G)$

$M_{GS} = (402{,}500\text{ lb})\ (2.354\text{ in.})$

$M_{GS} = 947{,}485$ inch-pounds

Calculate the flange moment at operating condition.

Flange loads

$$H_D = \frac{\pi}{4} B^2 p = \frac{\pi}{4} (18.00\,\text{in.})^2\ (150\,\text{psi})$$

$H_D = 38{,}170$ pounds

$H_G = H_p = 18{,}570$ pounds

$H_T = H - H_D = 58{,}520\text{ lb} - 38{,}170\text{ lb}$

$H_T = 20{,}350$ pounds

Moment arms

$R = 0.5\ (C - B) - g_1$

$R = 0.5\ (27.00\text{ in.} - 18.00\text{ in.}) - 2.03\text{ in.}$

$R = 2.47$ inches

$h_D = R + 0.5 g_1$

$h_D = 2.47\text{ in.} + 0.5\ (2.03\text{ in.})$

$h_D = 3.49$ inches

$h_G = 0.5\ (C - G)$

$h_G = 0.5\ (27.00\text{ in.} - 22.293\text{ in.})$

$h_G = 2.354$ inches

$h_T = 0.5\ (R + g_1 + h_G)$

$h_T = 0.5\ (2.47\text{ in.} + 2.03\text{ in.} + 2.354\text{ in.})$

$h_T = 3.427$ inches

Flange moment

$M_D = H_D\ (h_D)$

$M_D = (38{,}170\text{ lb})\ (3.49\text{ in.})$

$M_D = 133{,}210$ inch-pounds

$M_G = H_G\ (h_G)$

$M_G = (18{,}570\text{ lb})\ (2.354\text{ in.})$

$M_G = 43{,}710$ inch-pounds

$M_T = H_T\ (h_T)$

$M_T = (20{,}350\text{ lb})\ (3.43\text{ in.})$

$M_T = 69{,}800$ inch-pounds

$M_0 = M_D + M_G\ M_T$

$M_0 = 246{,}720$ inch-pounds

Calculate the shape constants and read the stress factors from the appropriate figure in Appendix 2 of ASME Code Section VIII, Division 1.

$$K = \frac{A}{B}$$

$$K = \frac{30.50}{18.00}$$

$$K = 1.694$$

From Fig. 2-7.1: T = 1.63; Z = 2.07; Y = 3.86; U = 4.24

$$\frac{g_1}{g_0} = \frac{2.03\text{ in.}}{1.03\text{ in.}} = 1.971$$

$$h_0 = \sqrt{Bg_0}$$

$$h_0 = \sqrt{(18.00\text{ in.})(1.03\text{ in.})}$$

$$h_0 = 4.306\text{ inches}$$

$$\frac{h}{h_0} = \frac{1.00\text{ in.}}{4.306\text{ in.}} = 0.232$$

From Fig. 2-7.2, F = 0.884; from Fig. 2-7.3, V = 0.343; and from Fig. 2-7.6, f = 2.375

$$e = \frac{F}{h_0}$$

$$e = \frac{0.884}{4.306\text{ in.}}$$

$$e = 0.205\text{ inch}^{-1}$$

$$d = \frac{Uh_0 (g_0)^2}{V}$$

$$d = \frac{(4.24)(4.306\text{ in.})(1.03\text{ in.})^2}{0.343}$$

$$d = 56.47\text{ inches}^3$$

Assume a flange thickness of 2.37 inches and calculate the stresses.

$$L = \frac{(te+1)}{T} + \frac{t^3}{d}$$

$$L = \frac{((2.37\text{ in.})(0.205\text{ in.}^{-1})+1)}{1.627} + \frac{(2.37\text{ in.})^3}{56.45\text{ in.}^3}$$

$$L = 1.149$$

Gasket Seating

Longitudinal Hub Stress

$$S_H = \frac{fM}{L(g_1)^2 B}$$

$$S_H = \frac{(2.375)(947{,}485\text{ in.}\cdot\text{lb})}{(1.149)(2.03\text{ in.})^2(18.00\text{ in.})}$$

$$S_H = 26{,}410\text{ psi}$$

Radial Flange Stress

$$S_R = \frac{(1.33te+1)M}{Lt^2 B}$$

$$S_R = \frac{\left((1.33)(2.37\text{ in.})(0.205\text{in.}^{-1})+1\right)(947{,}485\text{ in.}\cdot\text{lb})}{(1.149)(2.37\text{ in.})^2(18.00\text{ in.})}$$

$$S_R = 13{,}480\text{ psi}$$

Tangential Flange Stress

$$S_T = \frac{YM}{t^2 B} - ZS_R$$

$$S_T = \frac{(3.86)(947{,}485\text{ in.}\cdot\text{lb})}{(2.37\text{ in.})^2(18.00\text{ in.})} - (2.069)(13{,}480\text{ psi})$$

$$S_T = 8{,}320\text{ psi}$$

Allowable Stresses for Gasket Seating

$S_H = 1.5S_f$

$S_H = 1.5\ (20{,}000\text{ psi})$

$S_H = 30{,}000$

S_H > 26,410 psi therefore, acceptable.

$S_R = S_f$ = 20,000 > 13,480 psi therefore, acceptable.

$S_T = S_f$ = 20,000 > 8,320 psi therefore, acceptable.

Operating Condition

Longitudinal Hub Stress

$$S_H = \frac{fM}{L(g_1)^2 B}$$

$$S_H = \frac{(2.375)(246{,}720\ \text{in.-lb})}{(1.149)(2.03\ \text{in.})^2(18.00\ \text{in.})}$$

$$S_H = 6{,}880\ \text{psi}$$

Radial Flange Stress

$$S_R = \frac{(1.33te + 1)M}{Lt^2 B}$$

$$S_R = \frac{((1.33)(2.37)(0.205\ \text{in.}^{-1}) + 1)\ (246{,}720\ \text{in.-lb})}{(1.149)(2.37\ \text{in.})^2(18.00\ \text{in.})}$$

$$S_R = 3{,}510\ \text{psi}$$

Tangential Flange Stress

$$S_T = \frac{YM}{t^2 B} - ZS_R$$

$$S_T = \frac{(3.86)(246{,}720\ \text{in.-lb})}{(2.37\ \text{in.})^2(18.00\ \text{in.})} - (2.069)(3{,}510\ \text{psi})$$

$$S_T = 2{,}180\ \text{psi}$$

Allowable stresses for gasket seating.

$S_H = 1.5S_f$
S_H = 1.5 (12,000)
S_H = 18,000

S_H > 6,880 psi therefore, acceptable.

$S_R = S_f$ = 12,000 > 3,510 psi therefore, acceptable.

$S_T = S_f$ = 12,000 > 2,180 psi therefore, acceptable.

Therefore, the minimum required flange thickness t, is 2.37 inches.

Flange Rigidity

The flange design procedure in Appendix 2 is based on an acceptable stress limit and does not consider deformation. Flanges designed in accordance with the procedure may be within the allowable stress limits, but may not be sufficiently rigid to prevent excessive deformations, which leads to leaks. Paragraph S–2 provides a method for checking flange rigidity.

The method is based on calculating a rigidity index, J, and insuring that it is ≤ 1.0. The formula for J for integral and optional-type flanges designed as integral is:

$$J = \frac{52.14 M_0 V}{LE(g_0)^2 h_0 K_l}$$

J for loose type flanges with hubs:

$$J = \frac{52.14 M_0 V_L}{LE(g_0)^2 h_0 K_L}$$

J for loose type flanges without hubs and optional flanges designed as loose type:

$$J = \frac{109.4 M_0}{Et^3 \ln(K) K_L}$$

E is the modulus of elasticity. K_l and K_L are given below and the other symbols are defined in Appendix 2. If the flange rigidity is less than or equal to one, then the probability of the flange leakage in service is minimized. If the calculated J is greater than one, then consideration should be given to increasing the flange thickness and possibly other dimensions such as h, g_0, and g_1. Experience suggests that a K_L value of 0.2 for loose type flanges and a K_l of 0.3 for integral and optional flanges is sufficient for most services. Other valves may be used.

Example 10.3 Flange Rigidity

Calculate the flange rigidity of the flange in Example 10.2. The controlling moment, M_0 = 947,485 in.-lb. The flange thickness in the corroded condition is 2.37 inches. V = 0.343, L = 1.149, g_0 = 1.03 inches, h_0 = 4.306 inches, and the modulus of elasticity E = 29,800,000 psi.

Solution: From Appendix S-2, K = 0.3 for integral flanges, therefore,

$$J = \frac{52.14 M_0 V}{LE(g_0)^2 h_0 K_l}$$

$$J = \frac{52.14\,(947{,}485 \text{ in.-lb})(0.343)}{(1.149)(29{,}800{,}000 \text{ psi})(1.03 \text{ in.})^2(4.306 \text{ in.})(0.3)}$$

$$J = 0.331$$

J is less than one which minimizes the possibility of flange leakage.

Influence of Bolt Properties

Formula 4 of Appendix 2 provides a margin against abuse of the flange from overbolting. This is accomplished by calculating a gasket seating force, W, based on the average of the actual and required bolt areas. Footnote 2 cautions that the flange should be designed based on the total actual area if additional safety against abuse is desired.

If the controlling moment, M_0, is based on gasket seating, then bolt area and the allowable stress of the bolts at ambient temperature dominate the design in what may appear to be a confusing manner. In this instance, if either the allowable bolt stress, S_a, or actual bolt area, A_a, are increased, the value of W increases, resulting in higher flange stresses. Of course, the reverse is true. If gasket seating controls a flange design, then flange stresses may be reduced by reducing the actual bolt area or reducing the ambient allowable stress of the bolts.

The following example illustrates the effect of bolts on the design of a flange and also demonstrates the procedure for external loads.

Example 10.4 Design of a Flange Exposed to an External Moment

The 16 inch nozzle at the top of the vessel shown in Figure 7.3 is to be equipped with the SA–105 forging shown in Figure 10.3. The flange has an external bending moment of 8,790 foot-pounds. The carbon steel spiral-wound asbestos filled gasket is 1 inch wide.

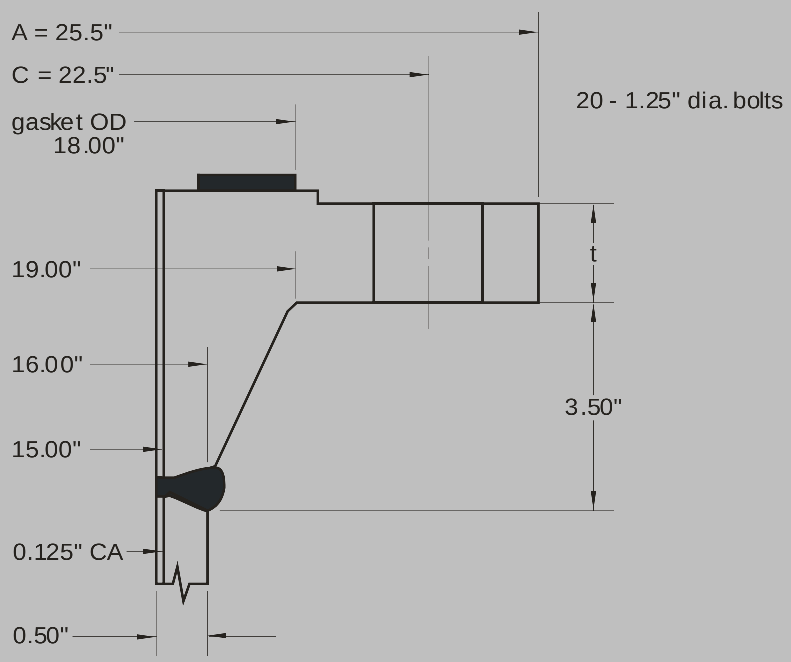


Figure 10. 3 Flange dimensions for 16 inch nozzle.

Additional design information:

Design temperature = 800°F	Allowable flange stress at design temperature = 12,000 psi
Design pressure = 150 psi	Allowable flange stress at bolt-up temperature = 20,000 psi
Gasket factor m = 2.5	Gasket design seating stress y = 10,000 psi
Corrosion allowance = 0.125 inch	

Part (a). Determine the flange thickness using SA–193 Grade B7M bolts. The allowable bolt stresses at bolt-up temperature and design temperature are 20,000 and 18,500 psi respectively.

Part (b). With the flange thickness determined in (a), calculate the flange stresses assuming the bolt material is SA–193 Grade B7. The allowable stress for SA–193 Grade B7 is 21,000 psi at design temperature and 25,000 psi at the bolt-up temperature. Note that SA–193 Grade B7 is a higher strength material than SA–193 Grade B7M.

Solution for Part (a):

The flange is an integral type similar to the one in sketch 6 of Fig. 2–4 of Appendix 2.

$B = 15.25$ inches $g_1 = 1.875$ inches

$g_0 = 0.375$ inches $h = 3.50$ inches

1. Calculate gasket dimensions.

N = 1.0 inch from Table 2-5.2

$b_0 = N/2$

$b_0 = 1.0 \text{ in.}/2$

$b_0 = 0.5 \text{ in.}$

Gasket OD = 18.00 inches

$$b = 0.5\sqrt{b_0}$$
$$b = 0.5\sqrt{0.5}$$
$$b = 0.3535 \text{ inch}$$

$$G = G_{OD} - 2b$$
$$G = (18.0 \text{ in.}) - 2\,(0.3535 \text{ in.})$$
$$G = 17.293 \text{ inches}$$

2. Calculate the equivalent pressure.

The radial force, F = 0

$$P_e = \frac{16M}{\pi G^3} + \frac{4F}{\pi G^2} + P$$
$$P_e = \frac{(16)\,(8{,}790 \text{ x } 12 \text{ in.-lb})}{\pi (17.29 \text{ in.})^3} + 150 \text{ psi}$$
$$P_e = 254 \text{ psi}$$

3. Calculate the bolt loads.

H = the hydrostatic end force = 59,660 pounds

H_P = joint contact surface compression load = 24,390 pounds

W_{m1} = minimum required operating bolt load

$W_{m1} = H + H_P$ = 84,050 pounds

W_{m2} = minimum required bolt load for gasket seating = 192,080 pounds

4. Calculate the required bolt area.

A_m = minimum required bolt area

A_m = 192,080 lb/20,000 psi

A_m = 9.604 inches^2

5. Determine the flange moment for bolt-up condition.

A_B = the actual bolt area = 18.58 inches^2

W_a = flange design bolt load = 281,840 pounds

h_G = flange load moment arm = 2.605 inches

M_{GS} = gasket seating moment = 734,190 inch-pounds

6. Calculate flange moment for operating condition.

H_D = hydrostatic end force on area inside of flange = 46,390 pounds

H_G = gasket load = 24,390 pounds

H_T = difference between hydrostatic end forces = 13,260 pounds

h_D = moment arm for H_D = 2.688 inches

h_G = moment arm for H_G = 2.604 inches

h_T = moment arm for H_T = 3.114 inches

M_D = moment due to H_D = 124,680 inch-pounds

M_G = moment due to H_G = 63,510 inch-pounds

M_T = moment due to H_T = 41,300 inch-pounds

M_0 = total operating moment = 229,500 inch-pounds

7. The shape constants and stress factors are:

$K = 1.672$	$h/h_0 = 1.46$	$g_1/g_0 = 5.0$
$V = 0.026$	$Y = 3.95$	$d = 56.90$
$T = 1.637$	$f = 1.000$	$h_0 = 2.391$
$F = 0.564$	$e = 0.236$	$Z = 2.114$
$U = 4.341$		

8. Assume a thickness of 1.95 inches. Calculate the stresses and compare to the allowables.

Gasket Seating Condition

STRESS	ALLOWABLE	ACTUAL
Longitudinal hub stress (S_h)	30,000	13,390
Radial flange stress (S_r)	20,000	19,940
Tangential flange stress (S_t)	20,000	7,780
Average S_h and S_t	20,000	10,580
Average S_h and S_r	20,000	16,660

The radial flange stress during gasket seating controls the design, producing a minimum thickness of 1.95 inches.

Solution for Part (b):

Items 1, 2, 3, 6, and 7 of the part (a) solution are unchanged. Items 4, 5, and 8 must be revised:

4. Calculate total bolt area and compare to actual bolt area.

A_m = minimum required bolt area

A_m = 192,080/25,000

A_m = 7.680 inches^2

5. Determine the flange moment for bolt-up condition.

A_B, actual bolt area = 18.58 inches^2

The allowable bolt stress is 25,000 psi instead of 20,000 psi therefore,

W_a, flange design bolt load =328,250 pounds

h_G, flange load moment arm = 2.605 inches

M_{GS}, gasket seating moment = 855,090 inch-pounds

8. The flange thickness is 1.95 inches. Calculate the stresses and compare to the allowables.

Gasket Seating Condition

STRESS	ALLOWABLE	ACTUAL
Longitudinal hub stress (S_h)	30,000	15,590
Radial flange stress (S_r)	20,000	23,260
Tangential flange stress (S_t)	20,000	9,060
Average S_h and S_t	20,000	12,320
Average S_h and S_r	20,000	19,420

The higher strength B7 bolts produce a radial flange stress that is 16% greater than the allowable. This clearly indicates that higher strength bolts are not a solution for an overstressed flanged if bolt-up conditions control the design. A similar condition exists if a larger bolt area is used.

Reversed Flanges

Paragraph 2–13 of ASME Section VIII, Division 1 also has rules for the design of reversed flanges. Reversed flanges, Figure 10.4, are often used in combination with a standard flange to form a reducer.

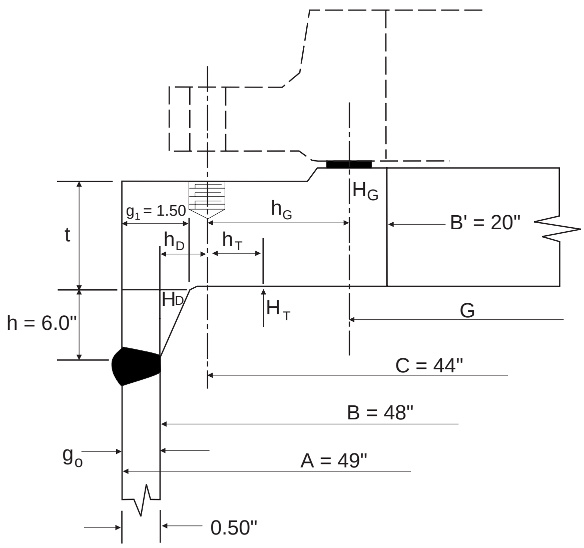


Figure 10.4 Reversed flange.

The design procedure for a reversed flange is the same as that for a standard flange with some minor modifications. The rules are mandatory for A/B' ≤ 2. The procedure becomes increasingly conservative and is optional when A/B' > 2.

As with the standard flange, moments must be calculated for both gasket seating and operating conditions. h_0 and K are redefined to reflect the inside diameter of the reversed flange and a new term, α_r, is introduced. α_r is used to modify stress factors T, U, and Y. An additional stress S_{T2}, the tangential flange stress at the inside bore of the flange is also introduced.

Example 10.5 Design of a Reversed Flange

Design a 20 inch inside diameter reversed flange for the 48 inch nozzle in Figure 7.3. The non-corroded dimensions of the reversed flange are given in Figure 10.4. The flange material is SA–105. There are 32 1¼ inch diameter SA–193 Grade B7 bolts. A one inch wide spiral-wound, carbon steel, asbestos filled gasket is used. Additional design data from Figure 7.3 is given below:

Design temperature = 800°F	Allowable flange stress at design temperature = 12,000 psi
Design pressure = 150 psi	Allowable flange stress at bolt-up temperature = 20,000 psi
Gasket factor m = 2.5	Gasket design seating stress y = 10,000 psi
Corrosion allowance = 0.125 inch	Gasket inside diameter = 22.0 inches

Solution:

The gasket dimensions and gasket load diameter, G are:

N = 1.0 inch from Table 2-5.2

b_o = N/2 = 1.0 in./2 = 0.5 inch from Table 2-5.2

Gasket OD = 22.00 in. + 2 (1 in.) = 24.00 inches

$$b = 0.5\sqrt{b_0}$$
$$b = 0.5\sqrt{0.5}$$
$$b = 0.3535 \text{ inch}$$

$$G = G_{OD} - 2b$$
$$G = 24.0 \text{ in.} - 2\,(0.3535 \text{ in.})$$
$$G = 23.293 \text{ inches}$$

Determine the minimum required bolt area for operating and gasket seating per paragraph 2–5 of ASME Section VIII, Division 1.

$$H = 0.785G^2p$$
$$H = 0.785\ (23.293 \text{ in.})^2\ (150 \text{ psi}) = 63{,}890 \text{ pounds}$$

$$H_p = 2b\pi Gmp$$
$$H_p = 2\ (0.3535 \text{ in.})\ (\pi)\ (23.293 \text{ in.})\ (2.5)\ (150 \text{ psi}) = 19{,}400 \text{ pounds}$$

$$W_{m1} = H + H_p = 63{,}890 \text{ pounds} + 19{,}400 \text{ pounds} = 83{,}290 \text{ pounds}$$

$$W_{m2} = \pi bGy = \pi\ (0.3535 \text{ in.})\ (23.293 \text{ in.})\ (10{,}000 \text{ psi}) = 258{,}680 \text{ pounds}$$

The required bolt area is the larger of A_{m1} and A_{m2} where A_{m1} and A_{m2} are:

$$A_{m1} = \frac{W_{m1}}{Sb_o}$$
$$A_{m1} = \frac{83{,}290 \text{ lb}}{21{,}000 \text{ psi}}$$
$$A_{m1} = 3.97 \text{ inches}^2$$

$$A_{m2} = \frac{W_{m2}}{Sb_a}$$
$$A_{m2} = \frac{258{,}680 \text{ lb}}{25{,}000 \text{ psi}}$$
$$A_{m2} = 10.34 \text{ inches}^2$$

The minimum required area is 10.34 $inches^2$. The actual area is 29.728 $inches^2$. Therefore bolting is acceptable.

The gasket seating design load is a function of the actual bolt area and the required bolt area:

$A = 0.5\ (A_m + A_b)$

$A = 0.5\ (10.34\ inches^2 + 29.728\ inches^2)$

$A = 20.03\ inches^2$

$W_a = A\ (Sb_a)$

$W_a = 20.03\ inches^2\ (25{,}000\ psi)$

$W_a = 500{,}890$ pounds

The flange moment during bolt-up is:

Moment arm

$h_G = 0.5\ (C - G)$

$h_G = 0.5\ (44.00\ in. - 23.293\ in.)$

$h_G = 10.354$ inches

The bolt-up load is W_a and the flange moment is:

$M_{GS} = H_G(h_G)$

$M_{GS} = (500{,}890\ lb)\ (10.354\ in.)$

$M_{GS} = 5{,}186{,}200$ inch-pounds

The flange loads, moment arms, and moments during operating conditions are:

$$H_D = \frac{\pi}{4}B^2p$$

$$H_D = \frac{\pi}{4}(48.25\ in.)^2(150\ psi)$$

$H_D = 274{,}100$ pounds

$H_G = W_{m1} - H$

$H_G = 83{,}290\ lb - 63{,}890\ lb$

$H_G = 19{,}400$ pounds

$H_T = H - H_D$

$H_T = 63{,}890\ lb - 274{,}100\ lb$

$H_T = -210{,}200$ pounds

$h_D = 0.5\ (C + g_1 - 2g_0 - B)$

$h_D = 0.5\ (44.00\ in. + 1.375\ in. - 2\ (0.375\ in.) - 48.25\ in.)$

$h_D = -\ 1.813$ inch

$h_G = 0.5\ (C - G)$
$h_G = 0.5\ (44.00 \text{ inches} - 23.293 \text{ in.})$
$h_G = 10.354 \text{ inches}$

$$h_T = 0.5\left(C - \frac{B+G}{2}\right)$$
$$h_T = 0.5\left(44.00 \text{ in.} - \frac{48.25 \text{ in.} + 23.293 \text{ in.}}{2}\right)$$
$h_T = 4.114 \text{ inches}$

Flange moment
$M_D = H_D\ (h_D)$
$M_D = (274{,}100 \text{ pounds})\ (-1.813 \text{ in.})$
$M_D = -496{,}800 \text{ inch·pounds}$

$M_G = H_G\ (h_G)$
$M_G = (19{,}400 \text{ pounds})\ (10.354 \text{ in.})$
$M_G = 200{,}900 \text{ inch·pounds}$

$M_T = H_T\ (h_T)$
$M_T = (-210{,}200 \text{ pounds})\ (4.114 \text{ in.})$
$M_T = -864{,}800 \text{ inch·pounds}$

$M_0 = |M_D + M_G + M_T|$
$M_0 = |(-496{,}800 \text{ inch·pounds}) + (200{,}900 \text{ inch·pounds}) + (-864{,}800 \text{ inch·pounds})|$
$M_0 = 1{,}160{,}700 \text{ inch·pounds}$

The flange shape constants and stress factors are shown below:

$$K = \frac{A}{B'}$$
$$K = \frac{49.00 \text{ in.}}{20.25 \text{ in.}}$$
$K = 2.42$

From Fig. 2-7.1 of ASME Section VIII, Division 1: T = 1.36 Z = 1.412 Y = 2.33 U = 2.56

$$\frac{g_1}{g_0} = \frac{1.375 \text{ in.}}{0.375 \text{ in.}} = 3.67$$

$$h_0 = \sqrt{Ag_0}$$
$$h_0 = \sqrt{(49.00 \text{ in.})(0.375 \text{ in.})}$$
$h_0 = 4.287 \text{ inches}$

$$\frac{h}{h_0} = \frac{6.00 \text{ in.}}{4.287 \text{ in.}} = 1.40$$

From Fig. 2-7.2, F = 0.605; from Fig. 2-7.3, V = 0.044; and from Fig. 2-7.6, f = 1.00

$$e_r = \frac{F}{h_0}$$
$$e_r = \frac{0.605}{4.287\text{ in.}}$$
$$e_r = 0.141\text{ inch}^{-1}$$

$$\alpha_r = \frac{1 + 0.668\dfrac{K+1}{Y}}{K^2}$$
$$\alpha_r = \frac{1 + 0.668\,\dfrac{2.42+1}{2.33}}{(2.42)^2}$$
$$\alpha_r = 0.338$$

$$U_r = \alpha_r U$$
$$U_r = 0.338\ (0.256)$$
$$U_r = 0.866$$

$$Y_r = \alpha_r Y$$
$$Y_r = 0.338\ (2.33)$$
$$Y_r = 0.788$$

$$T_r = \frac{Z+0.3}{Z-0.3}\alpha_r T$$
$$T_r = \frac{1.412+0.3}{1.412-0.3}(0.338)(1.36)$$
$$T_r = 0.709$$

$$d_r = \frac{U_r h_0 (g_0)^2}{V}$$
$$d_r = \frac{(0.866)(4.287\text{ in.})(0.375\text{ in.})^2}{0.044}$$
$$d_r = 11.87\text{ inches}^3$$

The stresses are calculated using the following formulas where M_0 is the gasket seating or operating moment:

$$S_H = \frac{f\,M_0}{L_r (g_1)^2 B'}$$

$$S_R = (1.33\,te_r + 1)\frac{M_0}{L_r t^2 B'}$$

$$S_{T1} = \frac{Y_r M_0}{t^2 B'} - Z S_R \frac{0.67\, te_r + 1}{1.33\, te_r + 1}$$

$$S_{T2} = \frac{M_0}{t^2 B'}\left(Y - \frac{2K^2(1 + 0.667\, te_r)}{(K^2 - 1)\, L_r}\right)$$

where:

$$L_r = \frac{te_r + 1}{T} + \frac{t^3}{d_r}$$

By successive recalculations, a flange thickness t = 5.25 inches is determined.

The stresses with t = 5.25 are summarized in the following table:

	Operating		Seating	
Stress (psi)	**Allowable**	**Actual**	**Allowable**	**Actual**
Longitudinal hub stress, S_H	18,000	2,060	30,000	9,220
Radial flange stress, S_R	12,000	280	20,000	1,260
Tangential flange stress, S_{T1}	12,000	1,340	20,000	6,000
Largest of $0.5(S_H+S_R)$ and $0.5(S_H+S_{T1})$	12,000	1,700	20,000	7,610
Tangential stress at the inside diameter of the flange, S_{T2}	12,000	4,340	20,000	19,410

The tangential flange stress at the inside diameter of the flange controls the design.

Chapter 11

QUALITY CONTROL

Quality Control and Inspection

The need for a quality control program is given in UG–90 and was discussed in Chapter 2. There are a number of Mandatory Appendices associated with Section VIII, Division 1 that list the specific requirements for a quality system. Appendix 10 discusses the minimum requirements in a pressure vessel construction or safety valve manufacturing quality program. An effective quality system includes definitions, and Mandatory Appendix 3 provides some of these. Inspection is a major element in a quality system. Mandatory Appendices 4, 6, 7, 8, and 12 provide the inspection requirements as well as acceptance criterion. Nonmandatory Appendix K provides guidance for the destructive sectioning of welds when such examination is required by construction contract.

Quality Control Programs

Quality control programs specify the manufacturing requirements practiced by a company to ensure that all applicable Sections of the Code are complied with. This is separate from quality assurance which defines inspections, examinations, and tests done to assure a structurally reliable vessel or component. Quality assurance is an integral part of a quality control program.

The Division requirements for a quality control manual predate the development of today's much discussed ISO 9000 Standards and their corresponding national equivalents. The Code requirements for quality control have remained relatively constant for many years. Manufacturers who so wish may have their quality control program in an ISO format provided all of the items in Appendix 10 are adequately presented. [Interpretation VIII–1–92–203 indicates that ISO programs may be acceptable.] In some aspects, an ISO program may be more demanding than a Code required quality control program. [Interpretations VIII–1–83–82 and VIII–1–83–82R make it clear that traceability of test equipment to a national standard is not required in a Code quality control program. This is not the case in an ISO program.] In the Code, the manufacturer is ultimately responsible for all aspects of conforming to the Code, and this responsibility cannot be lessened by others or secondary requirements that may suggest otherwise. Manufacturers who wish to have an ISO quality control program can have a separate quality program for Code manufacture listed as special process or a work procedure within the ISO program in lieu of using the program as the direct means of quality control.

Mandatory Appendix 10 identifies the essential elements of a quality control program. Paragraph 10–2 requires that the program be written down, and 10–1 requires that the Authorized Inspector approve the methods by which conformance to the Code is achieved. Changes to the program must not be enacted until approved by the Authorized Inspector. Items 10–3 and 10–4 present elements essential to any quality control program: a commitment by management to make the program company policy, and the assignment of authority and responsibility to those charged with ensuring compliance of work to the program. Paragraphs 10–15 and 10–16 contain essential items for a Code required quality program. In an ISO program, the responsible inspector may be identified in more generic terms, but in a quality program for Code work, the quality control manual must directly reference the Authorized Inspector.

Occasionally, an error may occur whereby it may not be possible to comply with the Code. This possibility is recognized in 10–8. Nonconformance with the Code must be brought to the attention of the Authorized Inspector and a mechanism must be provided to achieve agreement with the Inspector on how the nonconformity will be corrected or eliminated.

Other essential elements for a quality control program are given in 10–5, 10–6, 10–7, 10–9, 10–10, 10–11, and 10–13. These items require statements and procedures on how the minimum requirements given in the Code are to be adhered to through the work procedures followed by the manufacturer. Paragraph 10–12 requires a provision in the quality control program for the calibration of measuring, testing, and examining devices and tools.

Appendices 6, 8, and 10 require that nondestructive testing by magnetic particle, liquid penetrant, and ultrasonic techniques be done to written procedures, and that these procedures be certified by the manufacturer as meeting the requirements of the Division. These documents are also essential to the quality control program as indicated in 10–10. UW–51 does not require that radiographic examination be conducted to a written procedure. This same item however requires that all personnel performing nondestructive testing be qualified to a written procedure.

Nondestructive Testing

Section VIII, Division 1 recognizes the nondestructive examining techniques of radiographic, ultrasonic, magnetic particle, and liquid penetrant. The Division also requires visual examinations. While visual examination is nondestructive in nature, it is not referenced as a nondestructive examination technique in the Division (Figure 11.1).

Mandatory Appendix 6 gives the requirements for magnetic particle examination. Paragraph 6–1 references Article 7 of Section V for the methods and procedures that are applicable, while 6–2 gives the requirements for personnel doing magnetic particle examination and interpreting the results. The definitions of relevant indications are given in 6–3, and the accept or reject criteria are given in 6–4. Emphasis is placed upon linear indications that can be considered as cracks or crack-like defects. Such flaws render the material or part susceptible to failure.

Figure 11.1 This inspector is checking mill certificates against permanent die stampings on the fittings to be used for pressure vessel fabrication.

Mandatory Appendix 8 gives the requirements for liquid penetrant examination. Paragraph 8–1 references Article 6 of Section V for the acceptable techniques and procedures. The remaining items in Appendix 8 parallel the corresponding items in Appendix 6 regarding personnel skill, rejection criterion and repair requirements. The Code user should be aware that the two techniques do not mean detection equivalency, and that expanded knowledge of application techniques is required to select the technique offering the appropriate sensitivity in a given set of circumstances.

Mandatory Appendix 12 gives the requirements for ultrasonic examination. Paragraph 12–1 references Article 5 of Section V for the acceptable techniques and procedures. As with the other techniques, 12–2 specifies the training and competency requirements for personnel doing ultrasonic examination. The acceptance criterion for imperfection indications are different than for other nondestructive examination techniques, as ultrasonic examination gives an electronic signal that requires interpretation compared with the visual image provided by the other techniques. Paragraph 12–3 allows for interpretation by the equipment operator on two bases. If the operator can identify the flaw as a crack, lack of fusion, or incomplete penetration by virtue of the relative location of the flaw and the general form of the received signal, then the flaws are classed as unacceptable. If the flaw cannot be identified as one of the previously described imperfections, then it is judged on the basis of length and signal size relative to the signal from a manufactured calibration standard. Ultrasonic examination requires knowledge of manufacturing processes and part geometry. When employed by a skilled examiner, it can be a sensitive examination technique. Trust in the examiner's skill and integrity is required. Because the test results are not available in a format that can readily be verified by the Authorized Inspector, the Division does not permit ultrasonic examination to be substituted for radiographic examination except under the requirement of Code Case 2235. UW–51 does permit ultrasonic examination of repairs provided the original imperfection was also detected by ultrasonics. Paragraph 12–4 requires that the inspection technique and all examination findings be available for review by the Authorized Inspector.

Mandatory Appendices 4 and 7, and UW–51 and UW–52, give definitions and acceptance criterion for radiographic examination findings. Appendix 3 contains a definition for radiographic examination that makes it clear that radiographic images must be permanently recorded. Real time examinations, where images are seen on a video display without being recorded, do not meet the requirements of the Division for mandated radiographic examination. When round images are detected in a weld, they are to be interpreted as a round image even though the image may result from slag, porosity (gas), tungsten, or any other imperfection (Appendix 4, 4–2). Paragraph 4–3 states that image density is not relevant when assessing the acceptability of the imperfection. These are common misunderstandings held by radiograph interpreters. The Division deals only in images of cracks, incomplete fusion, incomplete penetration, elongated images, and rounded images. The first three of these listed images are either cracks or crack-like imperfections that have the potential to propagate vessel failure. Rounded images have a lesser significance unless they appear in clusters or in alignment. Paragraph 4–3 and Table 4-1, and Fig. 4-2 through 4–8 give the maximum acceptance limits for rounded images. The radiographic requirements given in Appendix 7 apply to ferrous castings.

UG–24 assesses a quality factor, E, for castings. The value of this quality factor varies with material type and the extent of destructive or nondestructive examination. Appendix 7 presents the magnetic particle, liquid penetrant, ultrasonic, and radiographic examination requirements for ferrous castings that are to have a higher quality factor than that assigned by UG–24. The acceptance criterion for each technique is given in the Appendix. Reference is made to indications categorized in accordance with standards in Section II Part A or ASTM standards. Castings that have been nondestructively examined shall be marked with the applicable quality factor in accordance with the requirements given in UG–24 and 7–5.

Pressure Testing

Regardless of the inspections and nondestructive tests conducted on a fabrication, the Division requires demonstration that the fabricated vessel can safely retain the maximum allowable working pressure. This is accomplished by pressure testing. The specified test is a hydrostatic test. However, in those circumstances where the fabrication cannot be adequately supported when filled with the liquid used to do the hydrostatic test, when it is essential that the equipment not be liquid wetted, or when the equipment cannot be dried, then a pneumatic test is permitted. (The Code user should note that while the Division may permit pneumatic testing, the jurisdictional authority may not.)

The pressure testing requirement is necessary since nondestructive testing techniques and visual examination have both scientific and physical limitations, as well as skill limitations of the inspector or equipment operator. The materials used in fabrication may also have undetected deficiencies. Equipment that conforms to all aspects of the Division except for the pressure test are possibly unsafe. A pressure test is required to verify the integrity of the equipment.

UG–99 gives the standard requirements for hydrostatic pressure testing. Hydrostatic testing is to be done only after all fabrications, heat treatment, inspections, and other tests have been done. The equipment is to be pressurized to 1.3 times the maximum allowable pressure for the vessel for the design temperature. This means that the equipment may actually be tested at a pressure greater than 1.3 times the design pressure. The Division requires that the maximum allowable pressure be multiplied by 1.3 and by the ratio of the material permitted stress value at the test temperature to that at the vessel design temperature. This requirement is made to account for the reduction in strength of metallic materials with increasing temperature.

Vessels designed to operate under vacuum conditions are also required to be hydrostatically tested. In this instance the test pressure is to be 1.3 times the difference between atmospheric pressure and the minimum internal pressure. Multi-chamber vessels shall have each chamber separately tested, or, in instances where the design is based on a pressure differential between chambers, the test shall be based on the maximum pressure differential. The acceptance criterion for a hydrostatic test is the absence of leaks at all joints and connections as determined by a visual examination.

Example 11.1 Design of Hydrostatic Test

The vessel in Figure 7.3 is designed as a multichamber vessel. The upper chamber is designed for 150 psi at 800°F and the lower chamber is designed for 200 psi at 800°F. All materials are SA–516 Grade 70. Design the hydrostatic test.

Solution: The vessel will require two separate tests. Tests will be done at shop temperature of 70°F.
The permitted stress value at the test temperature is 20,000 psi.
The permitted stress value at the design temperature is 12,000 psi.

(a) Calculate the test pressure for the upper chamber:

Hydro P = MAWP (1.3) (S(70)/S(800))
Hydro P = 1.3 (150 psi) (20,000 psi/12,000 psi) = 325 psi

(b) Calculate the test pressure for the lower chamber:

Hydro P = 1.3 (200 psi) (20,000 psi/12,000 psi) = 433 psi

The rules for pneumatic testing are given in UG-100. Wherever possible pneumatic test should be avoided because the expansive nature of a gas could maintain stress on a leak and cause it to grow, thus creating the potential for an explosive failure during the test. For this reason the preinspection requirements for a pneumatic test including UW-50, are rigorous; the rate of pressurization is to be controlled, and the test pressure is restricted to 1.1 times the maximum allowable operating pressure.

Occasionally equipment is designed where mathematical analysis of the stress in the material cannot be made or is uncertain. In such instances a hydrostatic proof test is conducted to validate the reliability of the design. Such testing is given in UG–101 and a vessel that was proof tested to failure is shown in Figure 11.2.

Fig 11.2 Fracture path of a vessel proof tested to failure. Strain gauges were used to monitor the stress at key locations in the vessel.

Pressure Relief Devices

Regardless of the good intentions of the designer, fabricator, inspector, and the user of pressure equipment, there remains an inherent risk that the design parameters may be exceeded in service. UG–125 requires that all pressure vessels be equipped with a pressure relieving device or devices. This is the only provision the Division makes for equipment operation. The specific requirements for pressure relief are given in sections UG–125 through UG–136. It is the pressure vessel user's responsibility to ensure that a device or devices are installed that prevent the pressure from rising more than 10% or 3 psi, whichever is greater, above the maximum allowable working pressure. If multiple pressure relief devices are used, then the devices must prevent the pressure from rising more than 16% or 4 psi, whichever is greater, above the maximum allowable working pressure. If supplemental pressure relief devices, or devices to protect against overpressure due to external heat, are installed, then the devices must prevent the pressure from rising more than 21% above the maximum allowable working pressure. In all cases, at least one pressure relief device must be set at or below the vessel's maximum allowable working pressure.

Appendix 1

TERMS AND ABBREVIATIONS

The following is a list of terms and abbreviations frequently used in this guide.

AI	Authorized Inspector
API	American Petroleum Institute
ASME	American Society of Mechanical Engineers
ASTM	American Society for Testing and Materials
AWS	American Welding Society
B31	The Pressure Piping Code of ASME
B31.1	The Power Piping Code Section of ASME
BPVC	ASME Boiler and Pressure Vessel Code
Code	(With an upper case C) ASME Boiler and Pressure Vessel Code, particularly Section VIII, Division 1
code	(With a lower case c) All other codes and standards
Code user	The organization responsible for the application of Section VIII
construction	An all-inclusive term comprising materials, design, fabrication, repair, examination, inspection, testing, certification, and pressure relief
FCAW	A GMAW process with flux contained in a tubular electrode
GMAW	Gas metal-arc welding
GTAW	Gas tungsten-arc welding
HAZ	Heat affected zone
Inspection Authority	As established by the jurisdiction or construction code
Inspector	Authorized Inspector
MAEWP	Maximum allowable external working pressure
MAWP	Maximum allowable working pressure
MDMT	Minimum design metal temperature
MT	Magnetic Particle Examination
NBCI	National Board Commissioned Inspector
P-number	An ASME classification of material into groups according to weldability
PT	Penetrant testing
PWHT	Post weld heat treatment
PQR	Procedure Qualification Record
RT	Radiographic examination
SA	Designation of an ASTM A metal adopted by ASME as SA
SB	Designation of an ASTM B metal adopted by ASME as SB
SAW	Submerged arc welding
Section V	Nondestructive Examination Section of the ASME Boiler and Pressure Vessel Code
Section VIII	Pressure Vessel Section of the ASME Boiler and Pressure Vessel Code
Section IX	Welding and Brazing Qualification Section of the ASME Boiler and Pressure Vessel Code
SMAW	Shielded metal arc welding
UT	Ultrasonic testing
welder	One who performs a manual or semiautomatic welding operation

Appendix 2

QUALITY CONTROL MANUAL

For those manufacturers wanting to have a U or UM authorization, the requirements for a quality control system are clearly stated at numerous locations in Section VIII. The basic element of a quality control program is the quality control manual. This is the master document that states the manufacturer's commitment to manufacture in accordance with the Code requirements and defines how the Code requirements are to be met. As indicated in Chapter 11, a quality control manual developed in accordance with ISO 9001 can be used to meet the Code requirements. In this Appendix, a sample quality control manual is presented that addresses the basic Code requirements. This manual is not intended to apply to individual cases without modification. The manual must represent how the Code user is to meet the Code requirements. The essential elements that must be addressed are included in the manual. How these elements are addressed is up to the Code user and the Authorized Inspector.

Some sections of this sample manual have little or no text. This format is designed to simplify preparation and documentation of the manual. Where commentary is used for explanation and is not meant to be part of the actual manual, it is enclosed in parentheses.

Quality Control Manual
for the
Fabrication of Pressure Vessels
Conforming to the Requirements of
ASME Section VIII, Division 1

PVCo. Inc.
Anywhere

Manual number_______________
Assigned to_______________

rev. no._____

(Each page of the manual should be identified with the date of issue or a revision number. This identifies the edition of the manual that is being used. It also facilitates updating the manual without requiring a complete replacement of the document every time a minor change is made.)

Table of Contents

(The Table of Contents listing is not a requirement, but it does facilitate use of the document.)

rev. no.____

Scope

This manual is presented as an outline of the procedures and policies that will be followed by PVCo. Inc. in the manufacture of pressure vessels and pressure retaining components that are to conform to the requirements of the American Society for Mechanical Engineers' Boiler and Pressure Vessel Code; specifically ASME Section VIII, Division 1. This manual recognizes the authority of the Authorized Inspector as defined in ASME Section VIII, Division 1, item UG–91. The Authorized Inspector is not an employee of PVCo. Inc.

Standards and specifications other than those of the American Society for Mechanical Engineers' Boiler and Pressure Vessel Code (BPVC) that are referenced or otherwise required in the manufacture of pressure vessels and pressure retaining components shall only be those listed in the BPVC. Only the BPVC approved editions of the listed standards and specifications shall be used.

PVCo. Inc. shall manufacture only those vessels and components for which there is the necessary skill, qualifications, and equipment.

The quality control program is to be administered by the Quality Control Manager who is authorized to resolve all issues of quality. All employees are responsible for ensuring that their work and the materials and equipment they work with meet the minimum requirements of the quality control program. Matters affecting quality that cannot be resolved by the Quality Control Manager shall be brought to the President's attention for resolution.

This manual and the quality control program has the support of PVCo. Inc.'s management and is company policy.

____

President
PVCo. Inc.

(Code users should use this section of the manual to present their commitment to build in accordance with the requirements of the Division. The Authorized Inspection Agency may also have requirements for the scope of the document, particularly where the manufacture of pressure vessels and components is controlled by law in the jurisdiction of manufacture.

It is important that all the Code user's personnel understand the quality control program and the company's commitment to manufacture in accordance with the policies and procedures stated in the manual. Therefore, the manual should be available to all employees.)

rev. no_____

Definitions

Acceptance - review of the subject in accordance with the requirements of this manual, and after such review, signing the applicable document to indicate agreement with the proposals or workmanship.

ASME - American Society for Mechanical Engineers

Authorized Inspector - an individual regularly employed by an Authorized Inspection Agency as defined in UG-91, having been qualified by written examination to conduct the inspections required in ASME Section VIII, Division 1.

BPVC - Boiler and Pressure Vessel Code. Those documents produced by ASME forming the basis for the safe design, construction, and use of pressurized equipment.

Calibrate - correlation of the performance of a measuring tool with a standard.

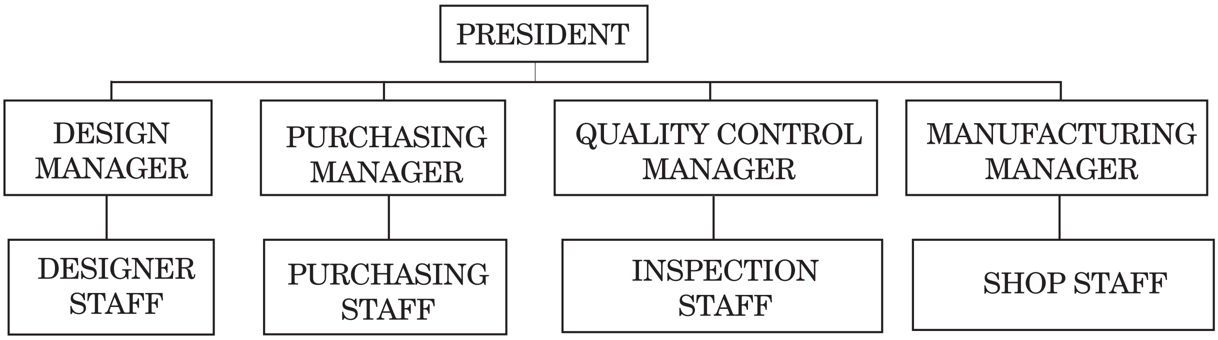
Fabricate - processes involved in the manufacture of pressure vessels, their parts, and appurtenances, including material handling, assembly, inspection, and proof testing.

Nonconformance - a deviation from the requirements of this manual or the BPVC.

Nondestructive testing - processes followed to examine materials or products for imperfections, which are carried out in a manner that neither alters nor changes the material or product examined.

rev. no_____

Organization



(In a small fabrication business, it is possible for two or more of the positions listed above to be filled by one individual. The positions may be filled by contractors. The names of the individuals filling given positions need not be identified in the document. However, those who have the authority and responsibility for the various job functions described in the Quality Control Manual need to be clearly identified. This can be done in a file in the quality control records.)

rev. no ____

Authority and Responsibility

1. Quality Control Manager

Authority–The Quality Control Manager is authorized by the President to manage the Quality Control Program of PVCo. Inc. The Quality Control Manager has the authority to make changes in the program, to investigate nonconformances with the program, and to resolve issues of nonconformance in accordance with the provisions of this program and the requirements of ASME Section VIII, Division 1.

Responsibility–The Quality Control Manager shall report to the President and is responsible for:

1.1 distributing, maintaining, and updating the Quality Control Manual and quality control work procedures;
1.2 evaluating all nonconformities and implementing remedial action;
1.3 liaising with the Authorized Inspector;
1.4 approving all subcontractors;
1.5 reviewing all designs for conformance with ASME Section VIII, Division 1;
1.6 calibrating all measuring and test instruments;
1.7 maintaining and storing all quality control documents;
1.8 examining all materials and fabrications for conformance with ASME Section VIII, Division 1;
1.9 preparing Manufacturer's Data Report in accordance with the requirements of ASME Section VIII, Division 1;
1.10 marking the manufactured vessel or component in accordance with ASME Section VIII, Division 1;
1.11 providing information on the latest editions and addenda of the applicable ASME Boiler and Pressure Vessel Code Sections to all employees; and
1.12 indoctrinating all employees in the policies and procedures of PVCo. Inc.

2. Design Manager

Authority–The Design Manager is authorized by the President to design pressure vessels and components for manufacture by PVCo. Inc.

Responsibility–The Design Manager shall report to the Quality Control Manager on design and quality issues. The Design Manager is responsible for:

2.1 designing pressure vessels and pressure retaining components in accordance with ASME Section VIII, Division 1;
2.2 designing support structures and nonpressure components in accordance with ASME Section VIII, Division 1 and the building code in the jurisdiction of installation for the equipment;
2.3 designing weld procedures in accordance with ASME Section IX for pressure vessels and pressure retaining components, support structures, and nonpressure retaining components welded to pressure retaining components;
2.4 preparing generic and project specific work procedures as specified by customers, the Authorized Inspector, and BPVC requirements; and
2.5 submitting designs to the Quality Control Manager for review for Code conformance.

rev. no.____

3. Purchasing Manager

Authority–The Purchasing Manager is authorized by the President to acquire those materials specified by the approved designs and the manufacturing equipment and supplies specified by the Manufacturing Manager.
Responsibility–The Purchasing Manager shall report to the Quality Control Manager on quality issues. The Purchasing Manager is responsible for:
3.1 purchasing the specified materials, equipment, and supplies.

4. Manufacturing Manager

Authority–The Manufacturing Manager is authorized by the President to manufacture pressure vessels and pressure retaining components in accordance with the approved designs.
Responsibility–The Manufacturing Manager shall report to the Quality Control Manager on quality issues. The Manufacturing Manager is responsible for:
4.1 using only those materials listed in the approved design;
4.2 employing only those work procedures that conform to the requirements of the approved design and ASME Section VIII, Division 1;
4.3 training manufacturing staff in the methods of material handling and manufacture processing;
4.4 using only trained and qualified personnel for manufacturing processes; and
4.5 monitoring the performance of the manufacturing process.

(The authorities given in this manual are specific to quality control for the requirements of the Code only. The Code user may find it appropriate to give other authorities (such as hiring and firing) to the various managers, along with cost control and other matters that the individual circumstance requires to implement the imposed responsibilities effectively. A quality control program that assigns responsibilities without providing authority to enforce the assigned responsibility is not effective. This is an easy feature to detect in a quality program and could result in a challenge to the company's commitment to provide a conforming product.)

rev. no._____

Document Control

1. Quality Control Manual
1.1 The Quality Control Manager is responsible for distributing this manual and maintaining records of issuance and all revisions. Form 1 shall be used to record the manual distribution and Form 2 shall be used to record all revisions and their approval by the Authorized Inspector. The original issue and all revisions of this manual shall remain in the possession of the Quality Control Manager.
1.2 Two types of manual distributions are to be used. Uncontrolled manuals do not require notification of revision to the manual. Controlled manuals require revision updates and remain the property of PVCo. Inc. Form 3 shall be used to verify that controlled manual copies are current. Uncontrolled manuals shall be clearly labeled as such.
1.3 Quality Control is the responsibility of all employees of PVCo. Inc. Revisions to the program and the Quality Control Manual may be proposed by any employee. Such proposals are to be submitted in writing to the Quality Control Manager. If approved by the Quality Control Manager, the revision shall be submitted to the Authorized Inspector. Revisions shall not be enacted unless approved by the Authorized Inspector.
1.4 The Quality Control Manager shall review the Quality Control Manual for conformance with addenda of ASME Section VIII, Division 1 as they are released. This review shall be done within 3 months of issuance of each addendum. Form 4 shall be used to verify this review.

2. Work Procedures
2.1 The Design Manager, in conjunction with the Quality Control and Manufacturing Managers, is responsible for preparing work procedures. These procedures shall conform to the requirements of ASME Section VIII, Division 1 and are part of the Quality Control Program.
2.2 The Quality Control Manager is responsible for the distribution and maintenance of the work procedures. These procedures are to be available to all workers and the Authorized Inspector.
2.3 The Quality Control Manager shall review the work procedures for conformance with the latest edition of the referencing standard as approved in the current edition of the relevant section of the ASME Boiler and Pressure Vessel Code.

3. Design Documents
3.1 The Design Manager shall assign a sequential number from the PVCo. Inc. job list for each pressure vessel or pressure component construction. This number shall be used to track the progress of the job and shall be the serial number assigned by PVCo. Inc. to the vessel or component. The Design Manager shall initiate a file for each job number. This file shall contain all design calculations, drawings, test results, inspection reports, data reports, nondestructive test reports, approvals, client data, contract documents, and other documents and information relating to the manufacture of the item.

4. Calibration Records
4.1 The Quality Control Manager shall maintain a permanent record of all calibrations of test and measuring equipment. Form 5 shall be used and referred to as the Calibration Log. The Quality Control Manager shall review the Calibration Log monthly and arrange for calibration of repaired equipment and equipment that is due for calibration.

rev. no_____

5. Job Files
5.1 The Quality Control Manager shall retain the job records for at least five years. The job file is to be the permanent record of manufacture. The minimum records and reports to be retained in the job file are:

- design calculations,
- design drawing,
- records of approvals,
- material list,
- mill certificates and partial data reports,
- job record,
- nondestructive examination reports,
- heat treat record,
- nonconformity reports, and
- manufacturer's data reports.

(The technique of using separate work procedures rather than detailing these in the Quality Control Manual is used here to develop a quality program with flexibility. The commitment to quality manufacturing and the basic program to achieve control of the quality are given in the manual with the intent that there will be few changes. The work procedures may well change with the project variety, and incorporating them in the Quality Control Manual, which is a controlled document, may inhibit the scope of work that a company undertakes. For example, weld procedure specifications as required by the Code are actually work procedures. These are frequently subject to revision. They could be included in a Quality Control Manual but their presence may require more updating work on the manual than is reasonable. Section VIII, Division 1 does not require nor imply that weld procedure specifications be included in the Quality Control Manual. They must, however, be available for the purpose of providing instruction to employees and information to the Authorized Inspector. Weld procedure specifications are considered to be work procedures that are part of the quality control system but are not presented in the Quality Control Manual. The same is true for procedures such as those required for nondestructive testing, heat treatment, material purchasing, etc.

Work procedures need not be complex documents, nor should they be. Weld procedure specifications are commonly written in a cryptic form. A work procedure may consist of a form or checklist. The intent is to ensure that all items required by the Code are tested, examined, or used as specified by the minimum requirements of the Code. Customers or manufacturers may have more requirements or more rigorous requirements than the Code, and routine use of work procedures helps to ensure that the special and nonstandard requirements are not missed.

The Division requires that data reports be retained for five years (UG–120). Radiograph examination records are to be retained at least until the Data Reports are signed by the Authorized Inspector (UW–51). No other records need be retained for Code compliance.)

rev. no____

Design Control

The Design Manager shall ensure that the design of the vessel or component and any auxiliary supports or nonpressure components conform to the requirements of ASME Section VIII, Division 1.

The Design Manager shall present the completed design and calculations to the Quality Control Manager for review. Once approved, the design and calculations shall be available for review and approval by the client and the Authorized Inspector. The complete design shall give the applicable date of Section VIII, Division 1 and addenda, material specifications, design pressure limits, design temperature limits, nondestructive examination requirements, heat treat requirements, work procedure requirements, weld joint requirements, impact test requirements, hydrostatic test requirements, and any other requirements specified by the customer or ASME Section VIII, Division 1.

The Quality Control Manager shall mark as approved and distribute copies of the approved design to the Purchasing Manager and the Manufacturing Manager. Form 1 shall be used to record the distribution of the design and any revisions. Superseded design information shall be destroyed when so indicated on Form 1.

rev. no____

Material Control

The Purchasing Manager shall purchase only those materials and components listed on the approved design. Mill certifications and partial data reports and certifications shall be requested in accordance with the requirements of Work Procedure 1.

The Manufacturing Manager is responsible for material receipt. Received material shall be placed in the quarantine bay until released by the Quality Control Manager.

The Quality Control Manager shall examine all incoming materials for conformance with the purchasing documents, the job design, and ASME Section VIII, Division 1. This examination shall be done in accordance with Work Procedure 2. Approved material shall be so marked prior to release to the Manufacturing Manager.

The Manufacturing Manager shall maintain material identification at all times. Identification shall be done in accordance with the requirements of ASME Section VIII, Division 1 and Work Procedure 3.

(Work Procedure 1 should contain instructions for purchasing only those materials, with the required documents, as required for Code compliance. Some of the Section VIII, Division 1 articles that may be applicable are U-3, UG–8, UG–9, UG–10, UG–11, UG–24 and UG–93. Work Procedure 2 may require a form to assist in the examination of incoming materials and documents. Some of the Section VIII, Division 1 articles that may be applicable are UG–8, UG–9, UG–10, UG–11, UG–16, UG–24, UG–32, UG–81, UG–90, UG–93 and UF–27. Work Procedure 3 will need to consider UG–77, UG–90, UG–94, UW–31, and UW–32.)

rev. no.____

Fabrication Requirements

The Quality Control Manager is responsible for preparing the Job Progress Record using Form 6. It shall contain examination hold points for nondestructive tests and visual examinations. The Quality Control Manager shall advise the Authorized Inspector of the job, and the Authorized Inspector shall indicate, on the Job Progress Record, any specific hold points for examinations.

The Quality Control Manager is responsible for examining the materials and workmanship during all stages of manufacture for conformance with ASME Section VIII, Division 1. Work Procedure 4 shall be followed in conducting these examinations.

The Manufacturing Manager shall make the fabrication available for all examinations, scheduled or otherwise, made by the Quality Control Manager or the Authorized Inspector.

The Manufacturing Manager shall ensure that manufacturing methods are in accordance with the applicable work procedure listed on the Job Progress Record. The Quality Control Manager shall ensure that the approved work procedure is available.

The Manufacturing Manager shall maintain records of worker qualifications using Form 7. Only those workers qualified in the manufacturing procedure shall be used. Workers shall record their work involvement in accordance with the requirements of ASME Section VIII, Division 1 and Work Procedure 5.

The Manufacturing Manager shall conduct the standard hydrostatic test listed on the job design.

(Work Procedure 4 specifies how in-process examinations and inspections by the manufacturer are to be done. Some of the Section VIII, Division 1 articles that may need to be considered are UG–76, UG–77, UG–79, UG–80, UG–81, UG–82, UG–84, UG–85, UG–94, UG–96, UG–97, UG–99,UG–100, UW–11, UW–13, UW–26, UW–28, UW–29, UW–30, UW–31, UW–32, UW–33, UW–35, UW–41, UW–47, UW–48, UW–49, UW–50, UW–51, UW–52, UF–27, UF–45, UB–31, UB–32, UB–44, UCS–79, UHT–19, UHT–20, UHT–81, and UHT–84. Work Procedure 5 needs to specify how welders are to identify their work in accordance with the requirements of UW-37.)

rev. no____

Examination and Nondestructive Test Requirements

The Quality Control Manager is responsible for conducting those examinations listed on the Job Progress Report as well as those examinations he/she believes are necessary or are requested by the Authorized Inspector to ensure a safe fabrication.

The Quality Control Manager is responsible for conducting those nondestructive tests required by the design and ASME Section VIII, Division 1, as well as those he/she believes are necessary or are requested by the Authorized Inspector to ensure a safe fabrication. Nondestructive tests shall be conducted in accordance with a written work procedure. The qualifications of nondestructive testing personnel shall be in accordance with Work Procedure 6.

(Appendices 6, 8, and 12 reference nondestructive tests to be done in accordance with a written procedure. Work Procedure 6 gives the training, experience, qualification, and certification of nondestructive testing personnel required in UW–51. This procedure will have to meet the requirements of SNT–TC–1A, 1998 addenda of 1996 edition. The work procedure should also make provision for keeping qualifications records.)

rev. no.____

Equipment Calibrations

The Quality Control Manager is responsible for ensuring that only calibrated measuring and test equipment is used. A log of all such equipment and its calibration frequency shall be maintained using Form 5.

Measuring and testing equipment shall be calibrated in accordance with the equipment manufacturer's specifications. Where no such specifications exist, calibration shall be in accordance with written procedures prepared by the Quality Control Manager.

Calibrations shall be done whenever equipment is repaired or suspected of being in error, or on a frequency determined by the Quality Control Manager.

rev. no.____

Nonconformances

Nonconformances occur when there is a deviation in material or manufacturing tolerance or a failure to meet the minimum requirements in either ASME Section VIII, Division 1, or this Manual. It is the responsibility of all employees to report nonconformances. Nonconformances shall be reported immediately to the Quality Control Manager using Form 8.

The Quality Control Manager shall quarantine all nonconforming material, components, and fabrications by tagging the item with a Stop Work Tag, Form 9. No subsequent work shall be done on a tagged item.

The Quality Control Manager is responsible for resolving all nonconformances. For those items that do not conform to ASME Section VIII, Division 1, the resolution of the Authorized Inspector is required and shall be the final authority.

rev. no.____

Form 1

Document Transmittal

To:

The following documents are attached

- ( ) for your information
- ( ) for your working copy
- ( ) for your review and comments
- ( ) for your review and acceptance
- ( ) for material purchase
- ( ) for construction

Item no.	Job no.	Rev. no.	No. of copies	Description

Comments: ____

____

____

Previous documents are to be ( ) destroyed ( ) returned to ____________________

This form is to be ( ) signed and returned to ____________________

Issued by ______________________________ Date ____________

rev. no._____

Form 2

Manual Revision Record

Rev. no.	Date	Description	QC Manager	Inspector

rev. no._____

Form 3

Quality Control Manual Distribution

Manual no.	Assigned to	Date	Rev. no.	Rev. no.	Rev. no	Rev. no.	Rev. no.

rev. no._____

Form 4

Manual Review Record

Section VIII Edition	Revision	Issue Date	Review Date

There shall be no more than 90 days from date of issue to manual review date.

rev. no._____

Form 5

Equipment Calibration Log

Description ____

Serial/Identification number ______________________________

Manufacturer's recommended calibration frequency ______________________

Calibration frequency and justification ____________________________

____

Date Calibrated	Calibrated by	Date Calibrated	Calibrated by	Date Calibrated	Calibrated by

rev. no________

Form 6

Job Progress Record

Job Number ________________ Description ____

Drawing Number ____________ Rev. no. ______

Hold points are indicated with an asterisk (*). Work must not proceed until indicated by the Quality Control Manager. This is achieved by the Quality Control Manager signing and dating this form at the hold point position.

Item	Notes	QC	AI
design review *	AI review before proceeding		
material inspect, certificate review			
formed part inspect			
shell to head fit-up inspect			
nozzle fit-up inspect			
internal fit-up inspect			
in-process weld inspect			
weld size and profile inspect			
internal and external inspect *	AI review before proceeding		
radiography			
magnetic particle			
other NDE			
NDE result review *	AI review before proceeding		
repairs and rework inspect			
heat treatment review			
hydrostatic test *	AI review before proceeding		
nameplate inspect			
data reports completed			

rev. no._____

Form 7

Welder Qualification Record

Symbol	Name	Weld Process	Weld Procedure	Date of Qualification	Date of Last Requalification

rev. no._____

Form 8

Nonconformance Report

Job Number ______________________________ **Date** ________________________

Description of Nonconformity ____

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Reviewed by ______________________________ **Date** ________________________
(Quality Control Manager)

Proposed Rework or Repair ____

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Review and Approvals
Design Manager __________________________ **Date** ________________________
Quality Control Manager ___________________ **Date** ________________________
Owner's Representative ___________________ **Date** ________________________
Authorized Inspector _____________________ **Date** ________________________

Review and Approval of Rework
Quality Control Manager ___________________ **Date** ________________________
Owner's Representative ___________________ **Date** ________________________
Authorized Inspector _____________________ **Date** ________________________

rev. no.______

Form 9

Copy of Stop Work Tag

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STOP ALL WORK ON THIS ITEM

Contact Quality Control Manager if you have any questions.

rev. no._____

Form 10

(The code user should include a copy or copies of the applicable forms from Appendix W, as required by UG–120.)

rev. no._____

Form 11

(The code user should include a copy of the data plate that the company will use, as required by UG–118 and 119. This form and Form 10 could be referenced in the Manual as a work procedure.)

rev. no._____

Appendix 3

DESIGN METHODS NOT GIVEN IN DIVISION 1

Aside from the design methods given in the Appendices, Section VIII, Division 1 provides only rules for safe design and construction. UG–22 provides a list of loads that must be considered when designing a pressure vessel. The list includes all loads that might act on the vessel. U–2(g) specifies that the manufacturer is responsible for the principles and techniques used for the design details not provided directly in the Division. Appendix G provides guidance in designing these structural details. This appendix contains suggested methods for some of the more common design approaches used.

UG–54 Vessel Support

Vessels are commonly mounted with their major axis either in the horizontal or the vertical plane. For horizontal mounting, the vessel shell is normally set on two saddles. The analysis commonly used was initially developed by L. P. Zick[(1)] where the stresses in the vessel are determined for longitudinal bending, tangential shear, and hoop stresses due to vessel and vessel charge weight. The formulas consider the stress concentration caused by the supports. The longitudinal bending stress plus the longitudinal stress due to pressure must not exceed the allowable stress for the shell material at any circumferential weld. Similarly, the tangential shear stress must not exceed the allowable stress for the shell material at any circumferential weld. The hoop stress will be in compression and must be considered at the bottom of the saddle and at the end of the saddle (called the horn of the saddle). At the bottom of the saddle the stress should not exceed 0.5 times the compressive yield stress for the shell material, while at the end of the saddle the stress must not exceed 1.5 times the allowable stress for the material (UG–23(c)).

Vertically mounted vessels are frequently supported on a vessel skirt. Since the skirt is welded to the vessel, the design rules of the Division are applicable. Usually the thickness of the skirt is controlled by the skirt-to-shell weld detail as the stress allowables for the weld and the joint efficiency are applicable. The skirt must be sized to accommodate the compressive load created by the weight of the vessel and its charge plus the stress caused from bending created by wind load. Openings in the skirt create a stress concentration that may necessitate reinforcement. The discontinuity stress at the shell-to-skirt weld will also require evaluation.

Saddle supports, skirt bolt rings, anchor bolts, and foundations are normally designed to procedures and allowable stress values given in national standards for steel construction and related building codes.

Environmental Loads

Wind load must be considered for both horizontal and vertical vessels. Not only does wind cause bending stresses in the vessel, but vortex shedding may create vibration. The wind acts as an external unidirectional pressure on the vessel and this can be determined using the wind loading charts and methods given in the National Building Code. The vessel is evaluated as a beam with a uniformly distributed load.

Wind induced vibration may result in fatigue damage to the vessel. The first critical vibration frequency of the vessel should be determined and this can be compared with the first critical wind velocity, V.

$$V = 3.4\frac{D}{T}$$

where: V is the first critical wind velocity in miles per hour,
D is the outside diameter of the vessel in feet, and
T is the first period of vibration of the vessel in second per cycle.

The stresses resulting from wind load are added to the other static and operating stresses and the resulting stress must not exceed 1.2 times the allowable stress for the material as described in UG–23(d).

In some environments and for some process applications, snow and ice loading will also have to be considered, and the stress resulting from these environmental loads is to be added to the other static and operating stresses. National Building Codes can be used for projected snow loads.

Seismic Loads

In earthquake active areas the vessel will require evaluation for stability during an earthquake. The analysis is usually done in accordance with National Building Code requirements where the overturning moment for the vessel is considered as well as the horizontal shear stress. UG–23(d) provides for such stresses up to 1.2 times the maximum stress value for the material.

UG–55 Piping and Other External Equipment

Pipe and other external equipment such as ladders and platforms usually create a bending moment stress at the point of attachment. This stress is the result of the weight of the material and, in some instances, also results from differential thermal expansion. Generally these stresses are small and can be neglected.

[1] L. P. Zick, "Stresses in Large Horizontal Cylindrical Pressure Vessels on Two Supports," *Welding Journal Research Supplement*, September 1951.

Appendix 4

APPLICATIONS OF SECTION VIII, DIVISION 1 TO OPERATING PRESSURE VESSELS

Application of Section VIII, Division 1

Section VIII, Division 1 is a design and fabrication standard with no provision for operating guidelines. However, an operating pressure vessel usually requires maintenance and may require modification or repair. The jurisdictional authority usually specifies these requirements and frequently the rules of Section VIII, Division 1 are specified. These rules may be specified directly or indirectly through the specification of another code or standard. The more commonly used standards are:

- API 510, Pressure Vessel Inspection Code
- ANSI NB–23, A Manual for Boiler and Pressure Vessel Inspectors
- CSA B51, Boiler, Pressure Vessel, and Pressure Piping Code

It is important for owners and operators to understand the new requirements for their pressure vessel equipment so they do not compromise the safety features that were designed for and built into the equipment. The Code is not retroactive. Vessels should therefore be maintained, repaired, and modified, in so far as possible, in accordance with the edition of the Code that was applicable when the vessel was manufactured. This information is given on the data plate and the data sheets for the vessel. The owner or operator who makes repairs or modifications to a vessel in accordance with the current edition of the Code should review the entire vessel design if it was designed and built to an earlier edition. The significance of this requirement is particularly important as the permitted stress values with the 1999 edition of Section VIII are based upon the enhanced material quality available for new construction from 1999 and onwards. This quality may not be prevalent in the material of the existing vessel and to assume so may compromise the safety of the vessel. Similarly, prior to 1999, hydrostatic testing of vessels was done at a stress basis of 1.5 times the design pressure. In 1999 the stress basis was lowered to 1.3 times the design pressure. The reasons for hydrostatic testing of existing equipment should be carefully reviewed to determine what test pressure will be appropriate. The jurisdiction should be consulted for the requirements. (Old editions of the Code should be retained for servicing dated equipment.)

API–510

The American Petroleum Institute (API) has provided a set of rules for the inspection, repair, or alteration of pressure vessels since 1934. At one time these rules were part of the API/ASME Code For Unfired Pressure Vessels. When these rules were no longer presented in the BPVC, API published them as Recommended Practice 510. In 1989, API 510 was expanded to include provisions for the certification of inspectors who examine pressure vessels that have been in service. This has been popular with jurisdictional authorities and facilitated assignment of responsibility for proper maintenance of pressure vessels to owners and users.

API 510 is not only applicable to vessels built to the BPVC, but to other codes and standards as well. It does, however, place emphasis on maintaining ASME BPVC standards of safety.

Besides the 510 Standard, API publishes or will soon publish a number of supporting documents as Recommended Practices. These are:

- API RP 571, Conditions Causing Deterioration and Failure
- API RP 572, Inspection of Pressure Vessels
- API RP 573, Inspection of Fired Boilers and Heaters
- API RP 576, Inspection of Pressure Relieving Devices

There are other Standards and Recommended Practices published by API that provide guidance in special areas, but are not necessarily part of the API 510 program. Some of these are given below.

- API RP 574, Inspection of Piping, Valves, and Fittings
- API RP 575, Inspection of Atmospheric and Low-Pressure Storage Tanks
- API RP 2201, Procedures for Welding or Hot Tapping on Equipment in Service

Together with ASME Section VIII, these documents provide guidance to the owner or user of pressure equipment on when to inspect, what to look for, how to evaluate potential defects, and what repair procedures may be applicable.

ANSI NB–23

This is an American National Standard prepared by the National Board of Pressure Vessel Inspectors. The National Board consists of the Chief Inspectors of any state or city of the United States, and the provinces of Canada that adopt any or all sections of the ASME Boiler and Pressure Vessel Code. This organization promotes uniform administration and enforcement of the Code. Standards are developed and published to assist in meeting this objective, one of which is NB–23.

NB–23 provides rules to maintain the integrity of boilers and pressure vessels in service. While API is primarily for the owner or user, NB–23 is for the jurisdictional authority, but it does provide guidance to owners and users, and it has a broader base of vested interest than does API 510. Beside providing guidance in conducting inspections, the Standard provides guidance on making repairs and alterations to pressure vessels and boilers, including those not constructed in accordance with the BPVC.

NB–23 provides for Owner-User programs wherein owners or users are authorized by the jurisdictional authority to inspect their own equipment independently and make certain repairs and alterations without the presence of a jurisdictional inspector.

CSA B51

The ASME Boiler and Pressure Vessel Code is a referenced standard in CSA B51 which is the legislated requirement in Canadian jurisdictions. While individual Canadian jurisdictions may have slightly different requirements for the administration of pressure vessel safety, CSA B51 provides a basic uniform application of the Code in Canada.

One of the items CSA B51 provides for is Owner-User programs for the inservice inspection and maintenance of pressure equipment. This is the same program given in NB–23.

Appendix 5

ENGINEERING DATA

ASME Boiler and Pressure Vessel Code

I	Rules for Construction of Power Boilers
II	Materials
	▪ Part A – Ferrous Material Specifications
	▪ Part B – Nonferrous Material Specifications
	▪ Part C – Specifications for Welding Rods, Electrodes, and Filler Metals
	▪ Part D – Properties
III	Rules for Construction of Nuclear Power Plant Components
	▪ Subsection NCA – General Requirements for Division 1 and Division 2
III	Rules for Construction of Nuclear Power Plant Components – Division 1
	▪ Subsection NB – Class 1 Components
	▪ Subsection NC – Class 2 Components
	▪ Subsection ND – Class 3 Components
	▪ Subsection NE – Class MC Components
	▪ Subsection NF – Supports
	▪ Subsection NG – Core Support Structures
	▪ Subsection NH – Class 1 Components in Elevated Temperature Service
	▪ Appendices
III	Rules for Construction of Nuclear Power Plant Components – Division 2
	▪ Code for Concrete Reactor Vessels and Containments
III	Rules for Construction of Nuclear Power Plant Components – Division 3
	▪ Containment Systems for Storage and Transport Packaging of Spent Nuclear Fuel and High Level Radioactive Material and Waste
IV	Rules for Construction of Heating Boilers
V	Nondestructive Examination
VI	Recommended Rules for the Care and Operation of Heating Boilers
VII	Recommended Guidelines for the Care of Power Boilers
VIII	Rules for Construction of Pressure Vessels – Division 1
	Alternative Rules for Construction of Pressure Vessels – Division 2
	Alternative Rules for Construction of High Pressure Vessels – Division 3
IX	Welding and Brazing Qualifications
X	Fiber-Reinforced Plastic Pressure Vessels
XI	Rules for Inservice Inspection of Nuclear Power Plant Components

ASME Piping Codes

B31G - Manual for Determining the Remaining Strength of Corroded Pipelines
B31.1 - Power Piping
B31.2 - Fuel Gas Piping
B31.3 - Process Piping
B31.4 - Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids
B31.5 - Refrigeration Piping and Heat Transfer Components
B31.8 - Gas Transmission and Distribution Systems
B31.9 - Building Services Piping
B31.11 - Slurry Transportation Piping Systems

To Convert From	To	Multiply By	To Convert From	To	Multiply By
Angle			**Mass per unit length**		
degree	rad	1.745 329 E -02	lb/ft	kg/m	1.488 164 E + 00
Area			lb/in.	kg/m	1.785 797 E + 01
in.^{2}	mm^2	6.451 600 E + 02	**Mass per unit time**		
in.^{2}	cm^2	6.451 600 E + 00	lb/h	kg/s	1.259 979 E - 04
in.^{2}	m^2	6.451 600 E - 04	lb/min	kg/s	7.559 873 E - 03
ft^2	m^2	9.290 304 E - 02	lb/s	kg/s	4.535 924 E - 01
Bending moment or torque			**Mass per unit volume (includes density)**		
lbf - in.	N - m	1.129 848 E - 01	g/cm^3	kg/m^3	1.000 000 E + 03
lbf - ft	N - m	1.355 818 E + 00	lb/ft^3	g/cm^3	1.601 846 E - 02
kgf - m	N - m	9.806 650 E + 00	lb/ft^3	kg/m^3	1.601 846 E + 01
ozf - in.	N-m	7.061 552 E - 03	lb/in.^{3}	g/cm^3	2.767 990 E + 01
Bending moment or torque per unit length			lb/in.^{3}	kg/m^3	2.767 990 E + 04
lbf - in./in.	N - m/m	4.448 222 E + 00	**Power**		
lbf - ft/in.	N - m/m	5.337 866 E + 01	Btu/s	kW	1.055 056 E + 00
Corrosion rate			Btu/min	kW	1.758 426 E - 02
mils/yr	mm/yr	2.540 000 E - 02	Btu/h	W	2.928 751 E - 01
mils/yr	μ/yr	2.540 000 E + 01	erg/s	W	1.000 000 E - 07
Current density			ft - lbf/s	W	1.355 818 E + 00
A/in.^{2}	A/cm^2	1.550 003 E - 01	ft - lbf/min	W	2.259 697 E - 02
A/in.^{2}	A/mm^2	1.550 003 E - 03	ft - lbf/h	W	3.766 161 E - 04
A/ft^2	A/m^2	1.076 400 E + 01	hp (550 ft - lbf/s)	kW	7.456 999 E - 01
Electricity and magnetism			hp (electric)	kW	7.460 000 E - 01
gauss	T	1.000 000 E - 04	W/in.^{2}	W/m^2	1.550 003 E + 03
maxwell	μWb	1.000 000 E - 02	**Pressure (fluid)**		
mho	S	1.000 000 E + 00	atm (standard)	Pa	1.013 250 E + 05
Oersted	A/m	7.957 700 E + 01	bar	Pa	1.000 000 E + 05
Ω - cm	Ω - m	1.000 000 E - 02	in. Hg (32 F)	Pa	3.386 380 E + 03
Ω circular - mil/ft	μΩ - m	1.662 426 E - 03	in. Hg (60 F)	Pa	3.376 850 E + 03
Energy (impact other)			lbf/in.^{2} (psi)	Pa	6.894 757 E + 03
ft - lbf	J	1.355 818 E + 00	torr (mm Hg, 0 C)	Pa	1.333 220 E + 02
Btu (thermochemical)	J	1.054 350 E + 03	**Specific heat**		
cal (thermochemical)	J	4.184 000 E + 00	Btu/lb - F	J/kg - K	4.186 800 E + 03
kW - h	J	3.600 000 E + 06	cal/g - C	J/kg - K	4.186 800 E + 03
W - h	J	3.600 000 E + 03	**Stress (force per unit area)**		
Flow rate			tonf/in.^{2} (tsi)	MPa	1.378 951 E + 01
ft^3/h	L/min	4.719 475 E - 01	kgf/mm^2	MPa	9.806 650 E + 00
ft^3/min	L/min	2.831 000 E + 01	ksi	MPa	6.894 757 E + 00
gal/h	L/min	6.309 020 E - 02	lbf/in.^{2} (psi)	MPa	6.894 757 E - 03
gal/min	L/min	3.785 412 E + 00	MN/m^2	MPa	1.000 000 E + 00
Force			**Temperature**		
lbf	N	4.448 222 E + 00	F	C	5/9 (F - 32)
kip (1000 lbf)	N	4.448 222 E + 03	R	K	5/9
tonf	kN	8.896 443 E + 00	**Thermal conductivity**		
kgf	N	9.806 650 E + 00	Btu - in./s - ft^2 - F	W/m - K	5.192 204 E + 02
			Btu/ft - h - F	W/m - K	1.730 735 E + 00

To Convert From	To	Multiply By	To Convert From	To	Multiply By
Force per unit length			**Thermal conductivity (Con't)**		
lbf/ft	N/m	1.459 390 E + 01	Btu - in./h . ft^2 - F	W/m - K	1.442 279 E - 01
lbf/in.	N/m	1.751 268 E + 02	cal/cm - s - C	W/m - K	4.184 000 E + 02
Fracture toughness			**Thermal expansion**		
ksi√in.	MPa√m	1.098 800 E + 00	in./in. - C	m/m - K	1.000 000 E + 00
Heat content			in./in. - F	m/m - K	1.800 000 E + 00
Btu/lb	kJ/kg	2.326 000 E + 00	**Velocity**		
cal/g	kJ/kg	4.186 800 E + 00	ft/h	m/s	8.466 667 E - 05
Heat input			ft/min	m/s	5.080 000 E - 03
J/in.	J/m	3.937 008 E + 01	ft/s	m/s	3.048 000 E - 01
kJ/in.	kJ/m	3.937 008 E + 01	in./s	m/s	2.540 000 E - 02
Length			km/h	m/s	2.777 778 E - 01
Å	nm	1.000 000 E - 01	mph	km/h	1.609 344 E + 00
µin.	µm	2.540 000 E - 02	**Velocity of Rotation**		
mil	µm	2.540 000 E + 01	rev/min (rpm)	rad/s	1.047 164 E - 01
in.	mm	2.540 000 E + 01	rev/s	rad/s	6.283 185 E + 00
in.	cm	2.540 000 E + 00	**Viscosity**		
ft	m	3.048 000 E - 01	poise	Pa - s	1.000 000 E - 01
yd	m	9.144 000 E -01	stokes	m^2/s	1.000 000 E - 04
mile	km	1.609 300 E + 00	ft^2/s	m^2/s	9.290 304 E - 02
Mass			in.^{2}/s	mm^2/s	6.451 600 E + 02
oz	kg	2.834 952 E - 02	**Volume**		
lb	kg	4.535 924 E - 01	in.^{3}	m^3	1.638 706 E - 05
ton (short 2000 lb)	kg	9.071 847 E + 02	ft^3	m^3	2.831 685 E - 02
ton (short 2000 lb)	kg x 10^3	9.071 847 E - 01	fluid oz	m^3	2.957 353 E - 05
ton (long 2240 lb)	kg	1.016 047 E + 03	gal (U.S. liquid)	m^3	3.785 412 E - 03
kg x 10^3 = 1 metric ton			**Volume per unit time**		
Mass per unit area			ft^3/min	m^3/s	4.719 474 E - 04
oz/in.^{2}	kg/m^2	4.395 000 E + 01	ft^3/S	m^3/s	2.831 685 E - 02
oz/ft^2	kg/m^2	3.051 517 E - 01	in.^{3}/min	m^3/s	2.731 177 E - 07
oz/yd^2	kg/m^2	3.390 575 E - 02	**Wavelength**		
lb/ft^2	kg/m^2	4.882 428 E + 00	A	nm	1.000 000 E - 01

SI PREFIXES			
Prefix	**Symbol**	**Exponential Expression**	**Multiplication Factor**
peta	P	10^{15}	1 000 000 000 000 000
tera	T	10^{12}	1 000 000 000 000
giga	G	10^{9}	1 000 000 000
mega	M	10^{6}	1 000 000
kilo	k	10^{3}	1 000
hecto	h	10^{2}	100
deka	da	10^{1}	10
Base Unit	---	10^{0}	1
deci	d	10^{-1}	0.1
centi	c	10^{-2}	0.01
milli	m	10^{-3}	0.001
micro	µ	10^{-6}	0.000 001
nano	n	10^{-9}	0.000 000 001
pico	p	10^{-12}	0.000 000 000 001
femto	f	10^{-15}	0.000 000 000 000 001

DIMENSIONS OF WELDED AND SEAMLESS PIPE								
Nominal Pipe Size, in.	Outside Diameter	Nominal Wall Thickness (in.) for						
		Schedule 5S	Schedule 10S	Schedule 10	Schedule 20	Schedule 30	Schedule Standard	Schedule 40
⅛	0.405	---	0.049	---	---	---	0.068	0.068
¼	0.540	---	0.065	---	---	---	0.088	0.088
⅜	0.675	---	0.065	---	---	---	0.091	0.091
½	0.840	0.065	0.083	---	---	---	0.109	0.109
¾	1.050	0.065	0.083	---	---	---	0.113	0.113
1	1.315	0.065	0.109	---	---	---	0.133	0.133
1¼	1.660	0.065	0.109	---	---	---	0.140	0.140
1½	1.900	0.065	0.109	---	---	---	0.145	0.145
2	2.375	0.065	0.109	---	---	---	0.154	0.154
2½	2.875	0.083	0.120	---	---	---	0.203	0.203
3	3.5	0.083	0.120	---	---	---	0.216	0.216
3½	4.0	0.083	0.120	---	---	---	0.226	0.226
4	4.5	0.083	0.120	---	---	---	0.237	0.237
5	5.563	0.109	0.134	---	---	---	0.258	0.258
6	6.625	0.109	0.134	---	---	---	0.280	0.280
8	8.625	0.109	0.148	---	0.250	0.277	0.322	0.322
10	10.75	0.134	0.165	---	0.250	0.307	0.365	0.365
12	12.75	0.156	0.180	---	0.250	0.330	0.375	0.406
14 OD	14.0	0.156	0.188	0.250	0.312	0.375	0.375	0.438
16 OD	16.0	0.165	0.188	0.250	0.312	0.375	0.375	0.500
18 OD	18.0	0.165	0.188	0.250	0.312	0.438	0.375	0.562
20 OD	20.0	0.188	0.218	0.250	0.375	0.500	0.375	0.594
22 OD	22.0	0.188	0.218	0.250	0.375	0.500	0.375	---
24 OD	24.0	0.218	0.250	0.250	0.375	0.562	0.375	0.688
26 OD	26.0	---	---	0.312	0.500	---	0.375	---
28 OD	28.0	---	---	0.312	0.500	0.625	0.375	---
30 OD	30.0	0.250	0.312	0.312	0.500	0.625	0.375	---
32 OD	32.0	---	---	0.312	0.500	0.625	0.375	0.688
34 OD	34.0	---	---	0.312	0.500	0.625	0.375	0.688
36 OD	36.0	---	---	0.312	0.500	0.625	0.375	0.750
42 OD	42.0	---	---	---	---	---	0.375	---

See next table for heavier wall thicknesses. All units are inches.

DIMENSIONS OF WELDED AND SEAMLESS PIPE									
Nominal Pipe Size, in.	Outside Diameter	Nominal Wall Thickness (in.) for							
		Schedule 60	Extra Strong	Schedule 80	Schedule 100	Schedule 120	Schedule 140	Schedule 160	XX Strong
⅛	0.405	---	0.095	0.095	---	---	---	---	---
¼	0.540	---	0.119	0.119	---	---	---	---	---
⅜	0.675	---	0.126	0.126	---	---	---	---	---
½	0.840	---	0.147	0.147	---	---	---	0.188	0.294
¾	1.050	---	0.154	0.154	---	---	---	0.219	0.308
1	1.315	---	0.179	0.179	---	---	---	0.250	0.358
1¼	1.660	---	0.191	0.191	---	---	---	0.250	0.382
1½	1.900	---	0.200	0.200	---	---	---	0.281	0.400
2	2.375	---	0.218	0.218	---	---	---	0.344	0.436
2½	2.875	---	0.276	0.276	---	---	---	0.375	0.552
3	3.5	---	0.300	0.300	---	---	---	0.438	0.600
3½	4.0	---	0.318	0.318	---	---	---	---	---
4	4.5	---	0.337	0.337	---	0.438	---	0.531	0.674
5	5.563	---	0.375	0.375	---	0.500	---	0.625	0.750
6	6.625	---	0.432	0.432	---	0.562	---	0.719	0.864
8	8.625	0.406	0.500	0.500	0.594	0.719	0.812	0.906	0.875
10	10.75	0.500	0.500	0.594	0.719	0.844	1.000	1.125	1.000
12	12.75	0.562	0.500	0.688	0.844	1.000	1.125	1.312	1.000
14 OD	14.0	0.594	0.500	0.750	0.938	1.094	1.250	1.406	---
16 OD	16.0	0.656	0.500	0.844	1.031	1.219	1.438	1.594	---
18 OD	18.0	0.750	0.500	0.938	1.156	1.375	1.562	1.781	---
20 OD	20.0	0.812	0.500	1.031	1.281	1.500	1.750	1.969	---
22 OD	22.0	0.875	0.500	1.125	1.375	1.625	1.875	2.125	---
24 OD	24.0	0.969	0.500	1.218	1.531	1.812	2.062	2.344	---
26 OD	26.0	---	0.500	---	---	---	---	---	---
28 OD	28.0	---	0.500	---	---	---	---	---	---
30 OD	30.0	---	0.500	---	---	---	---	---	---
32 OD	32.0	---	0.500	---	---	---	---	---	---
34 OD	34.0	---	0.500	---	---	---	---	---	---
36 OD	36.0	---	0.500	---	---	---	---	---	---
42 OD	42.0	---	0.500	---	---	---	---	---	---

All units are inches.

APPROXIMATE HARDNESS CONVERSION NUMBERS FOR NONAUSTENITIC STEELS[a, b]								
Rockwell C 150 kgf Diamond HRC	Vickers HV	Brinell 3000 kgf 10mm ball[c] HB	Knoop 500 gf HK	Rockwell A 60 kgf Diamond HRA	Rockwell Superficial Hardness 15 kgf Diamond HR15N	Rockwell Superficial Hardness 30 kgf Diamond HR30N	Rockwell Superficial Hardness 45 kgf Diamond HR45N	Approx. Tensile Strength ksi (MPa)
68	940	---	920	85.6	93.2	84.4	75.4	---
67	900	---	895	85.0	92.9	83.6	74.2	---
66	865	---	870	84.5	92.5	82.8	73.3	---
65	832	739[d]	846	83.9	92.2	81.9	72.0	---
64	800	722[d]	822	83.4	91.8	81.1	71.0	---
63	772	706[d]	799	82.8	91.4	80.1	69.9	---
62	746	688[d]	776	82.3	91.1	79.3	68.8	---
61	720	670[d]	754	81.8	90.7	78.4	67.7	---
60	697	654[d]	732	81.2	90.2	77.5	66.6	---
59	674	634[d]	710	80.7	89.8	76.6	65.5	351 (2420)
58	653	615	690	80.1	89.3	75.7	64.3	338 (2330)
57	633	595	670	79.6	88.9	74.8	63.2	325 (2240)
56	613	577	650	79.0	88.3	73.9	62.0	313 (2160)
55	595	560	630	78.5	87.9	73.0	60.9	301 (2070)
54	577	543	612	78.0	87.4	72.0	59.8	292 (2010)
53	560	525	594	77.4	86.9	71.2	58.6	283 (1950)
52	544	512	576	76.8	86.4	70.2	57.4	273 (1880)
51	528	496	558	76.3	85.9	69.4	56.1	264 (1820)
50	513	482	542	75.9	85.5	68.5	55.0	255 (1760)
49	498	468	526	75.2	85.0	67.6	53.8	246 (1700)
48	484	455	510	74.7	84.5	66.7	52.5	238 (1640)
47	471	442	495	74.1	83.9	65.8	51.4	229 (1580)
46	458	432	480	73.6	83.5	64.8	50.3	221 (1520)
45	446	421	466	73.1	83.0	64.0	49.0	215 (1480)
44	434	409	452	72.5	82.5	63.1	47.8	208 (1430)
43	423	400	438	72.0	82.0	62.2	46.7	201 (1390)
42	412	390	426	71.5	81.5	61.3	45.5	194 (1340)
41	402	381	414	70.9	80.9	60.4	44.3	188 (1300)
40	392	371	402	70.4	80.4	59.5	43.1	182 (1250)
39	382	362	391	69.9	79.9	58.6	41.9	177 (1220)
38	372	353	380	69.4	79.4	57.7	40.8	171 (1180)
37	363	344	370	68.9	78.8	56.8	39.6	166 (1140)
36	354	336	360	68.4	78.3	55.9	38.4	161 (1110)
35	345	327	351	67.9	77.7	55.0	37.2	156 (1080)
34	336	319	342	67.4	77.2	54.2	36.1	152 (1050)
33	327	311	334	66.8	76.6	53.3	34.9	149 (1030)
32	318	301	326	66.3	76.1	52.1	33.7	146 (1010)
31	310	294	318	65.8	75.6	51.3	32.5	141 (970)
30	302	286	311	65.3	75.0	50.4	31.3	138 (950)
29	294	279	304	64.6	74.5	49.5	30.1	135 (930)
28	286	271	297	64.3	73.9	48.6	28.9	131 (900)
27	279	264	290	63.8	73.3	47.7	27.8	128 (880)
26	272	258	284	63.3	72.8	46.8	26.7	125 (860)
25	266	253	278	62.8	72.2	45.9	25.5	123 (850)

APPROXIMATE HARDNESS CONVERSION NUMBERS FOR NONAUSTENITIC STEELS[a, b] (Continued)

Rockwell C 150 kgf Diamond HRC	Vickers HV	Brinell 3000 kgf 10mm ball[c] HB	Knoop 500 gf HK	Rockwell A 60 kgf Diamond HRA	Rockwell Superficial Hardness 15 kgf Diamond HR15N	30 kgf Diamond HR30N	45 kgf Diamond HR45N	Approx. Tensile Strength ksi (MPa)
24	260	247	272	62.4	71.6	45.0	24.3	119 (820)
23	254	243	266	62.0	71.0	44.0	23.1	117 (810)
22	248	237	261	61.5	70.5	43.2	22.0	115 (790)
21	243	231	256	61.0	69.9	42.3	20.7	112 (770)
20	238	226	251	60.5	69.4	41.5	19.6	110 (760)

a. This table gives the approximate interrelationships of hardness values and approximate tensile strength of steels. It is possible that steels of various compositions and processing histories will deviate in hardness-tensile strength relationship from the data presented in this table. The data in this table should not be used for austenitic stainless steels, but have been shown to be applicable for ferritic and martensitic stainless steels. Where more precise conversions are required, they should be developed specially for each steel composition, heat treatment, and part.
b. All relative hardness values in this table are averages of tests on various metals whose different properties prevent establishment of exact mathematical conversions. These values are consistent with ASTM A 370-91 for nonaustenitic steels. It is recommended that ASTM standards A 370, E 140, E 10, E 18, E 92, E 110 and E 384, involving hardness tests on metals, be reviewed prior to interpreting hardness conversion values.
c. Carbide ball, 10mm.
d. This Brinell hardness value is outside the recommended range for hardness testing in accordance with ASTM E 10.

APPROXIMATE HARDNESS CONVERSION NUMBERS FOR NONAUSTENITIC STEELS[a, b]

Rockwell B 100 kgf 1/16" ball HRB	Vickers HV	Brinell 3000 kgf 10 mm HB	Knoop 500 gf HK	Rockwell A 60 kgf Diamond HRA	Rockwell Superficial Hardness 15 kgf 1/16" ball HR15T	30 kgf 1/16" ball HR30T	45 kgf 1/16" ball HR45T	Approx. Tensile Strength ksi (MPa)
100	240	240	251	61.5	93.1	83.1	72.9	116 (800)
99	234	234	246	60.9	92.8	82.5	71.9	114 (785)
98	228	228	241	60.2	92.5	81.8	70.9	109 (750)
97	222	222	236	59.5	92.1	81.1	69.9	104 (715)
96	216	216	231	58.9	91.8	80.4	68.9	102 (705)
95	210	210	226	58.3	91.5	79.8	67.9	100 (690)
94	205	205	221	57.6	91.2	79.1	66.9	98 (675)
93	200	200	216	57.0	90.8	78.4	65.9	94 (650)
92	195	195	211	56.4	90.5	77.8	64.8	92 (635)
91	190	190	206	55.8	90.2	77.1	63.8	90 (620)
90	185	185	201	55.2	89.9	76.4	62.8	89 (615)
89	180	180	196	54.6	89.5	75.8	61.8	88 (605)
88	176	176	192	54.0	89.2	75.1	60.8	86 (590)
87	172	172	188	53.4	88.9	74.4	59.8	84 (580)
86	169	169	184	52.8	88.6	73.8	58.8	83 (570)
85	165	165	180	52.3	88.2	73.1	57.8	82 (565)
84	162	162	176	51.7	87.9	72.4	56.8	81 (560)
83	159	159	173	51.1	87.6	71.8	55.8	80 (550)
82	156	156	170	50.6	87.3	71.1	54.8	77 (530)
81	153	153	167	50.0	86.9	70.4	53.8	73 (505)
80	150	150	164	49.5	86.6	69.7	52.8	72 (495)
79	147	147	161	48.9	86.3	69.1	51.8	70 (485)
78	144	144	158	48.4	86.0	68.4	50.8	69 (475)
77	141	141	155	47.9	85.6	67.7	49.8	68 (470)
76	139	139	152	47.3	85.3	67.1	48.8	67 (460)
75	137	137	150	46.8	85.0	66.4	47.8	66 (455)

APPROXIMATE HARDNESS CONVERSION NUMBERS FOR NONAUSTENITIC STEELS[a, b] (Continued)								
Rockwell B 100 kgf 1/16" ball HRB	**Vickers HV**	**Brinell 3000 kgf 10 mm HB**	**Knoop 500 gf HK**	**Rockwell A 60 kgf Diamond HRA**	**Rockwell Superficial Hardness 15 kgf 1/16" ball HR15T**	**30 kgf 1/16" ball HR30T**	**45 kgf 1/16" ball HR45T**	**Approx. Tensile Strength ksi (MPa)**
74	135	135	147	46.3	84.7	65.7	46.8	65 (450)
73	132	132	145	45.8	84.3	65.1	45.8	64 (440)
72	130	130	143	45.3	84.0	64.4	44.8	63 (435)
71	127	127	141	44.8	83.7	63.7	43.8	62 (425)
70	125	125	139	44.3	83.4	63.1	42.8	61 (420)
69	123	123	137	43.8	83.0	62.4	41.8	60 (415)
68	121	121	135	43.3	82.7	61.7	40.8	59 (405)
67	119	119	133	42.8	82.4	61.0	39.8	58 (400)
66	117	117	131	42.3	82.1	60.4	38.7	57 (395)
65	116	116	129	41.8	81.8	59.7	37.7	56 (385)
64	114	114	127	41.4	81.4	59.0	36.7	---
63	112	112	125	40.9	81.1	58.4	35.7	---
62	110	110	124	40.4	80.8	57.7	34.7	---
61	108	108	122	40.0	80.5	57.0	33.7	---
60	107	107	120	39.5	80.1	56.4	32.7	---
59	106	106	118	39.0	79.8	55.7	31.7	---
58	104	104	117	38.6	79.5	55.0	30.7	---
57	103	103	115	38.1	79.2	54.4	29.7	---
56	101	101	114	37.7	78.8	53.7	28.7	---
55	100	100	112	37.2	78.5	53.0	27.7	---
54	---	---	111	36.8	78.2	52.4	26.7	---
53	---	---	110	36.3	77.9	51.7	25.7	---
52	---	---	109	35.9	77.5	51.0	24.7	---
51	---	---	108	35.5	77.2	50.3	23.7	---
50	---	---	107	35.0	76.9	49.7	22.7	---
49	---	---	106	34.6	76.6	49.0	21.7	---
48	---	---	105	34.1	76.2	48.3	20.7	---
47	---	---	104	33.7	75.9	47.7	19.7	---
46	---	---	103	33.3	75.6	47.0	18.7	---
45	---	---	102	32.9	75.3	46.3	17.7	---
44	---	---	101	32.4	74.9	45.7	16.7	---
43	---	---	100	32.0	74.6	45.0	15.7	---
42	---	---	99	31.6	74.3	44.3	14.7	---
41	---	---	98	31.2	74.0	43.7	13.6	---
40	---	---	97	30.7	73.6	43.0	12.6	---
39	---	---	96	30.3	73.3	42.3	11.6	---
38	---	---	95	29.9	73.0	41.6	10.6	---
37	---	94	29.5	78.0	41.0	9.6		---
36	---	---	93	29.1	72.3	40.3	8.6	---
35	---	---	92	28.7	72.0	39.6	7.6	---
34	---	---	91	28.2	71.7	39.0	6.6	---
33	---	---	90	27.8	71.4	38.3	5.6	---
32	---	---	89	27.4	71.0	37.6	4.6	---
31	---	---	88	27.0	70.7	37.0	3.6	---
30	---	---	87	26.6	70.4	36.3	2.6	---

APPROXIMATE HARDNESS CONVERSION NUMBERS FOR NONAUSTENITIC STEELS[a, b] (Continued)

a. This table gives the approximate interrelationships of hardness values and approximate tensile strength of steels. It is possible that steels of various compositions and processing histories will deviate in hardness-tensile strength relationship from the data presented in this table. The data in this table should not be used for austenitic stainless steels, but have been shown to be applicable for ferritic and martensitic stainless steels. Where more precise conversions are required, they should be developed specially for each steel composition, heat treatment, and part.

b. All relative hardness values in this table are averages of tests on various metals whose different properties prevent establishment of exact mathematical conversions. These values are consistent with ASTM A 370-91 for nonaustenitic steels. It is recommended that ASTM standards A 370, E 140, E 10, E 18, E 92, E 110 and E 384, involving hardness tests on metals, be reviewed prior to interpreting hardness conversion values.

APPROXIMATE HARDNESS NUMBERS FOR AUSTENITIC STEELS[a]

Rockwell C	Rockwell A	Rockwell Superficial Hardness		
150 kgf, Diamond HRC	60 kgf, Diamond HRA	15 kgf, Diamond HR15N	30 kgf, Diamond HR30N	45 kgf, Diamond HR45N
48	74.4	84.1	66.2	52.1
47	73.9	83.6	65.3	50.9
46	73.4	83.1	64.5	49.8
45	72.9	82.6	63.6	48.7
44	72.4	82.1	62.7	47.5
43	71.9	81.6	61.8	46.4
42	71.4	81.0	61.0	45.2
41	70.9	80.5	60.1	44.1
40	70.4	80.0	59.2	43.0
39	69.9	79.5	58.4	41.8
38	69.3	79.0	57.5	40.7
37	68.8	78.5	56.6	39.6
36	68.3	78.0	55.7	38.4
35	67.8	77.5	54.9	37.3
34	67.3	77.0	54.0	36.1
33	66.8	76.5	53.1	35.0
32	66.3	75.9	52.3	33.9
31	65.8	75.4	51.4	32.7
30	65.3	74.9	50.5	31.6
29	64.8	74.4	49.6	30.4
28	64.3	73.9	48.8	29.3
27	63.8	73.4	47.9	28.2
26	63.3	72.9	47.0	27.0
25	62.8	72.4	46.2	25.9
24	62.3	71.9	45.3	24.8
23	61.8	71.3	44.4	23.6
22	61.3	70.8	43.5	22.5
21	60.8	70.3	42.7	21.3
20	60.3	69.8	41.8	20.2

a. All relative hardness values in this table are averages of tests on various metals whose different properties prevent establishment of exact mathematical conversions. These values are consistent with ASTM A 370-91 for austenitic steels. It is recommended that ASTM standards A 370, E 140, E 10, E 18, E 92, E 110 and E 384, involving hardness tests on metals, be reviewed prior to interpreting hardness conversion values.

APPROXIMATE HARDNESS CONVERSION VALUES FOR AUSTENITIC STEELS[a]						
Rockwell B 100 kgf 1/16" ball HRB	Brinell Indentation Diameter, mm	Brinell 3000 kgf 10 mm Ball HB	Rockwell A 60 kgf Diamond HRA	Rockwell Superficial Hardness 15 kgf 1/16" ball HR15T	30 kgf 1/16" ball HR30T	45 kgf 1/16" ball HR45T
100	3.79	256	61.5	91.5	80.4	70.2
99	3.85	248	60.9	91.2	79.7	69.2
98	3.91	240	60.3	90.8	79.0	68.2
97	3.96	233	59.7	90.4	78.3	67.2
96	4.02	226	59.1	90.1	77.7	66.1
95	4.08	219	58.5	89.7	77.0	65.1
94	4.14	213	58.0	89.3	76.3	64.1
93	4.20	207	57.4	88.9	75.6	63.1
92	4.24	202	56.8	88.6	74.9	62.1
91	4.30	197	56.2	88.2	74.2	61.1
90	4.35	192	55.6	87.8	73.5	60.1
89	4.40	187	55.0	87.5	72.8	59.0
88	4.45	183	54.5	87.1	72.1	58.0
87	4.51	178	53.9	86.7	71.4	57.0
86	4.55	174	53.3	86.4	70.7	56.0
85	4.60	170	52.7	86.0	70.0	55.0
84	4.65	167	52.1	85.6	69.3	54.0
83	4.70	163	51.5	85.2	68.6	52.9
82	4.74	160	50.9	84.9	67.9	51.9
81	4.79	156	50.4	84.5	67.2	50.9
80	4.84	153	49.8	84.1	66.5	49.9

a. All relative hardness values in this table are averages of tests on various metals whose different properties prevent establishment of exact mathematical conversions. These values are consistent with ASTM A 370-91 for austenitic steels. It is recommended that ASTM standards A 370, E 140, E 10, E 18, E 92, E 110 and E 384, involving hardness tests on metals, be reviewed prior to interpreting hardness conversion values.

APPROXIMATE HARDNESS CONVERSION NUMBERS FOR NICKEL & HIGH-NICKEL ALLOYS									
Vickers[a]	Brinell[b]	Rockwell Hardness Number[c]							
HV	HB	HRA	HRB	HRC	HRD	HRE	HRF	HRG	HRK
513	479	75.5	---	50.0	63.0	---	---	---	---
481	450	74.5	---	48.0	61.5	---	---	---	---
452	425	73.5	---	46.0	60.0	---	---	---	---
427	403	72.5	---	44.0	58.5	---	---	---	---
404	382	71.5	---	42.0	57.0	---	---	---	---
382	363	70.5	---	40.0	55.5	---	---	---	---
362	346	69.5	---	38.0	54.0	---	---	---	---
344	329	68.5	---	36.0	52.5	---	---	---	---
326	313	67.5	---	34.0	50.5	---	---	---	---
309	298	66.5	106	32.0	49.5	---	116.5	94.0	---
285	275	64.5	104	28.5	46.5	---	115.5	91.0	---
266	258	63.0	102	25.5	44.5	---	114.5	87.5	---
248	241	61.5	100	22.5	42.0	---	113.0	84.5	---
234	228	60.5	98	20.0	40.0	---	112.0	81.5	---
220	215	59.0	96	17.0	38.0	---	111.0	78.5	100.0

APPROXIMATE HARDNESS CONVERSION NUMBERS FOR NICKEL & HIGH-NICKEL ALLOYS (Continued)									
Vickers[a]	Brinell[b]	Rockwell Hardness Number[c]							
HV	HB	HRA	HRB	HRC	HRD	HRE	HRF	HRG	HRK
209	204	57.5	94	14.5	36.0	---	110.0	75.5	98.0
198	194	56.5	92	12.0	34.0	---	108.5	72.0	96.5
188	184	55.0	90	9.0	32.0	108.5	107.5	69.0	94.5
179	176	53.5	88	6.5	30.0	107.0	106.5	65.5	93.0
171	168	52.5	86	4.0	28.0	106.0	105.0	62.5	91.0
164	161	51.5	84	2.0	26.5	104.5	104.0	59.5	89.0
157	155	50.0	82	---	24.5	103.0	103.0	56.5	87.5
151	149	49.0	80	---	22.5	102.0	101.5	53.0	85.5
145	144	47.5	78	---	21.0	100.5	100.5	50.0	83.5
140	139	46.5	76	---	19.0	99.5	99.5	47.0	82.0
135	134	45.5	74	---	17.5	98.0	98.5	43.5	80.0
130	129	44.0	72	---	16.0	97.0	97.0	40.5	78.0
126	125	43.0	70	---	14.5	95.5	96.0	37.5	76.5
122	121	42.0	68	---	13.0	94.5	95.0	34.5	74.5
119	118	41.0	66	---	11.5	93.0	93.5	31.0	72.5
115	114	40.0	64	---	10.0	91.5	92.5	---	71.0
112	111	39.0	62	---	8.0	90.5	91.5	---	69.0
108	108	---	60	---	---	89.0	90.0	---	67.5
106	106	---	58	---	---	88.0	89.0	---	65.5
103	103	---	56	---	---	86.5	88.0	---	63.5
100	100	---	54	---	---	85.5	87.0	---	62.0
98	98	---	52	---	---	84.0	85.5	---	60.0
95	95	---	50	---	---	83.0	84.5	---	58.0
93	93	---	48	---	---	81.5	83.5	---	56.5
91	91	---	46	---	---	80.5	82.0	---	54.5
89	89	---	44	---	---	79.0	81.0	---	52.5
87	87	---	42	---	---	78.0	80.0	---	51.0
85	85	---	40	---	---	76.5	79.0	---	49.0
83	83	---	38	---	---	75.0	77.5	---	47.0
81	81	---	36	---	---	74.0	76.5	---	45.5
79	79	---	34	---	---	72.5	75.5	---	43.5
78	78	---	32	---	---	71.5	74.0	---	42.0
77	77	---	30	---	---	70.0	73.0	---	40.0

a. Vickers Hardness Number, Vickers indenter, 1.5, 10, 30-kgf load.

b. Brinell Hardness Number, 10 mm ball, 3000 kgf load. Note that in Table 5 of ASTM Test Method E 10, the use of a 3000-kgf load is recommended (but not mandatory) for material in the hardness range 96 to 600 HV, and a 1500-kgf load is recommended (but not mandatory) for material in the hardness range 48 to 300 HV. These recommendations are designed to limit impression diameters to the range 2.50 to 6.0 mm. The Brinell hardness numbers in this conversion table are based on tests using a 3000-kgf load. When the 1500-kgf load is used for the softer nickel and high-nickel alloys, these conversion relationships do not apply.

c. A Scale - 60-kgf load, diamond penetrator; B Scale - 100-kgf load, 1/16 in. (1.588 mm) ball; C Scale - 150-kgf load, diamond penetrator; D Scale - 100-kgf, diamond penetrator; E Scale - 100-kgf load, 1/8 in. (3.175 mm) ball; F Scale - 60-kgf load, 1/16 in. (1.588 mm) ball; G Scale - 150-kgf load, 1/16 in. (1.588 mm) ball; K Scale - 150-kgf load, 1/8 in. (3.175 mm) ball.

APPROXIMATE HARDNESS CONVERSION NUMBERS FOR NICKEL & HIGH-NICKEL ALLOYS							
Vickers[a]	**Brinell[b]**	**Rockwell Superficial Hardness[c]**					
HV	**HB**	**HR15-N**	**HR30-N**	**HR45-N**	**HR15-T**	**HR30-T**	**HR45-T**
513	479	85.5	68.0	54.5	---	---	---
481	450	84.5	66.5	52.5	---	---	---
452	425	83.5	64.5	50.0	---	---	---
427	403	82.5	63.0	47.5	---	---	---
404	382	81.5	61.0	45.5	---	---	---
382	363	80.5	59.5	43.0	---	---	---
362	346	79.5	58.0	41.0	---	---	---
344	329	78.5	56.0	38.5	---	---	---
326	313	77.5	54.5	36.0	---	---	---
309	298	76.5	52.5	34.0	94.5	85.5	77.0
285	275	75.0	49.5	30.0	94.0	84.5	75.0
266	258	73.5	47.0	26.5	93.0	83.0	73.0
248	241	72.0	44.5	23.0	92.5	81.5	71.0
234	228	70.5	42.0	20.0	92.0	80.5	69.0
220	215	69.0	39.5	17.0	91.0	79.0	67.0
209	204	68.0	37.5	14.0	90.5	77.5	65.0
198	194	66.5	35.5	11.0	89.5	76.0	63.0
188	184	65.0	32.5	7.5	89.0	75.0	61.0
179	176	64.0	30.5	5.0	88.0	73.5	59.5
171	168	62.5	28.5	2.0	87.5	72.0	57.5
164	161	61.5	26.5	-0.5	87.0	70.5	55.5
157	155	---	---	---	86.0	69.5	53.5
151	149	---	---	---	85.5	68.0	51.5
145	144	---	---	---	84.5	66.5	49.5
140	139	---	---	---	84.0	65.5	47.5
135	134	---	---	---	83.0	64.0	45.5
130	129	---	---	---	82.5	62.5	43.5
126	125	---	---	---	82.0	61.0	41.5
122	121	---	---	---	81.0	60.0	39.5
119	118	---	---	---	80.5	58.5	37.5
115	114	---	---	---	79.5	57.0	35.5
112	111	---	---	---	79.0	56.0	33.5
108	108	---	---	---	78.5	54.5	31.5
106	106	---	---	---	77.5	53.0	29.5
103	103	---	---	---	77.0	51.5	27.5
100	100	---	---	---	76.0	50.5	25.5
98	98	---	---	---	75.5	49.0	23.5
95	95	---	---	---	74.5	47.5	21.5
93	93	---	---	---	74.0	46.5	19.5
91	91	---	---	---	73.5	45.0	17.0
89	89	---	---	---	72.5	43.5	14.5
87	87	---	---	---	72.0	42.0	12.5
85	85	---	---	---	71.0	41.0	10.0
83	83	---	---	---	70.5	39.5	7.5
81	81	---	---	---	70.0	38.0	5.5
79	79	---	---	---	69.0	36.5	3.0
78	78	---	---	---	68.5	35.5	1.0

APPROXIMATE HARDNESS CONVERSION NUMBERS FOR NICKEL & HIGH-NICKEL ALLOYS (Cont.)							
Vickers[a]	Brinell[b]	Rockwell Superficial Hardness[c]					
HV	HB	HR15-N	HR30-N	HR45-N	HR15-T	HR30-T	HR45-T
77	77	---	---	---	67.5	34.0	-1.5

a. Vickers Hardness Number, Vickers indenter, 1.5, 10, 30-kgf load.

b. Brinell Hardness Number, 10 mm ball, 3000 kgf load. Note that in Table 5 of ASTM Test Method E 10, the use of a 3000-kgf load is recommended (but not mandatory) for material in the hardness range 96 to 600 HV, and a 1500-kgf load is recommended (but not mandatory) for material in the hardness range 48 to 300 HV. These recommendations are designed to limit impression diameters to the range 2.50 to 6.0 mm. The Brinell hardness numbers in this conversion table are based on tests using a 3000-kgf load. When the 1500-kgf load is used for the softer nickel and high-nickel alloys, these conversion relationships do not apply.

c. 15-N Scale - 15-kgf load, superficial diamond penetrator; 30-N Scale - 30-kgf load, superficial diamond penetrator; 45-N Scale - 45-kgf load, superficial diamond penetrator; 15-T Scale - 15-kgf load, 1/16 in. (1.588 mm) ball; 30-T Scale - 30-kgf load, 1/16 in. (1.588 mm) ball; 45-T Scale - 45-kgf load, 1/16 in. (1.588 mm) ball.

Subject Index

A

B

C

D

F

H

I

L

M

N

O

R

S

T

V

W

CODE PARAGRAPH INDEX

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Dr. Bruce Ball, is a professional engineer with over 25 years consulting experience in failure analysis and inspection of plant facilities and mechanical equipment. He has a Ph.D. in Metallurgical Engineering from the University of Alberta. He is a CSA W178.2 certified welding inspector and API 510 pressure vessel inspector. He has been awarded a fellowship by the Canadian Society for Nondestructive Testing and is an Adjunct Professor in the Department of Chemical and Materials Engineering at the University of Alberta. He is vice chairman of the Canadian Standards Association Committee on Weld Inspector and Weld Inspection Organization Certification.

Dr. Will Carter, a graduate from the Southern University, as well as the University of Illinois, has 32 years experience in failure analysis, design, repair and construction of process equipment. His expertise is in design, analysis, fabrication, failure investigation, trouble shooting, repair and upgrading of pressure vessels, tanks, pipelines and piping systems. Dr. Carter has been an active participant on pressure vessel and tank code and standards committees. Also, he is a registered mechanical engineer.

Dr. Carter is the past chairman of the ASME Boiler & Pressure Vessel Subcommittee on Design as well as a past member of the ASME Boiler and Pressure Vessel Main Committee and Subcommittee for Pressure Vessels. He was a past chairman of the API Subcommittee Pressure Vessels and Tanks, and has also chaired the API 653 Task Force on inspection, maintenance and repair of aboveground storage tanks.

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