

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

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Third Edition on CD-ROM™



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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

***CASTI* Guidebook Series - Vol. 4**

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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.

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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.

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.

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.

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.

- 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)).

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 $\frac{1}{4}$ the thickness of the thinner part, or by $\frac{1}{8}$ 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

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 qualified 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.

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.

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

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 $\frac{5}{8}$ 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 $\frac{5}{8}$ 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 $4\frac{1}{2}$ 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, 5⁄8 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 1⁄8 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.)

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 | <ul style="list-style-type: none"> • 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 | <ul style="list-style-type: none"> • 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 | <ul style="list-style-type: none"> • 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. |

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:

$$S_C = \frac{RP}{t} \quad (7.1)$$

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.

$$S_L = \frac{RP}{2t} \quad (7.2)$$

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.

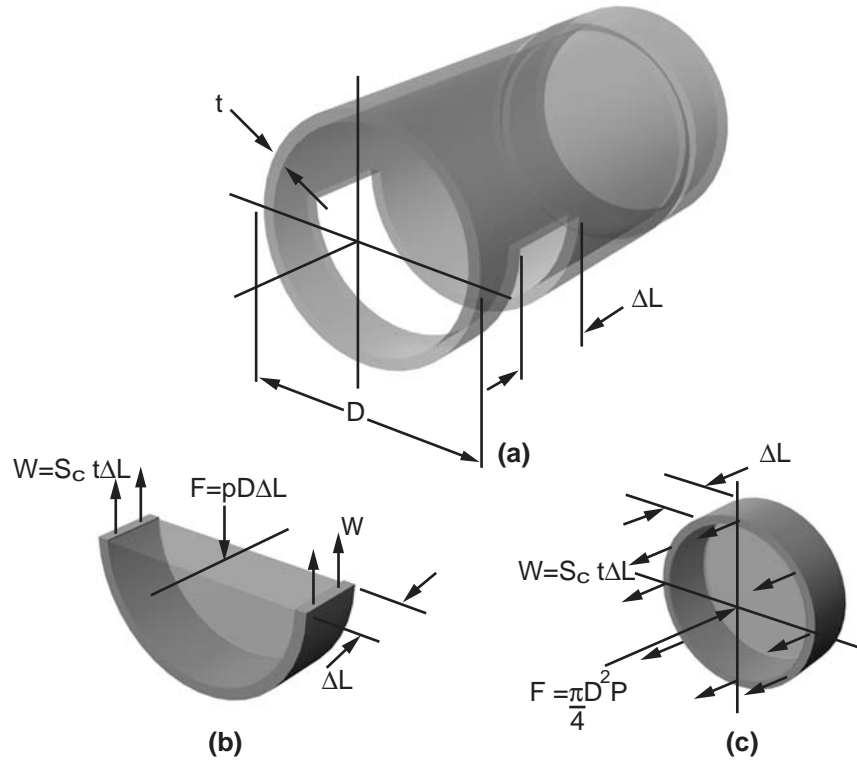


Figure 7.1 Forces and stresses in a pressurized cylinder.

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.

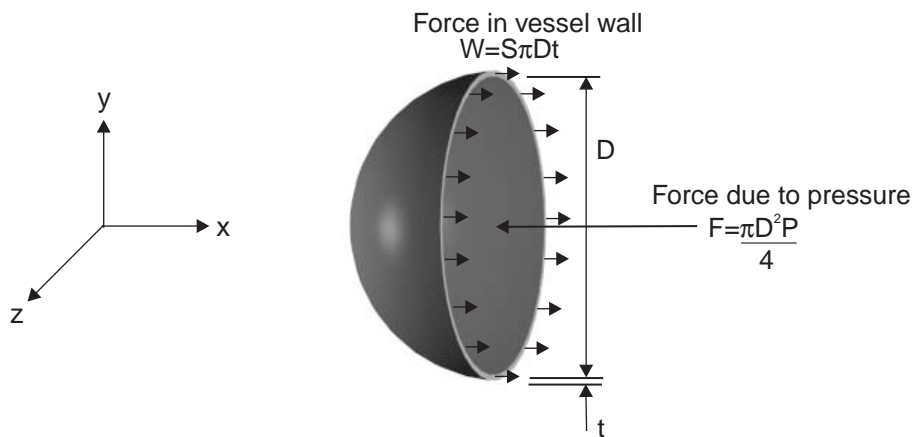


Figure 7.2 Forces and stresses in a sphere under pressure.

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:

$$\begin{array}{|c|} \hline \\ \hline \end{array} t = t_{\text{nominal}} - CA$$

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

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

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.

$$\begin{array}{|c|} \hline \\ \hline \end{array} P = \frac{SEt}{R + 0.6t}$$

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

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

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. 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.

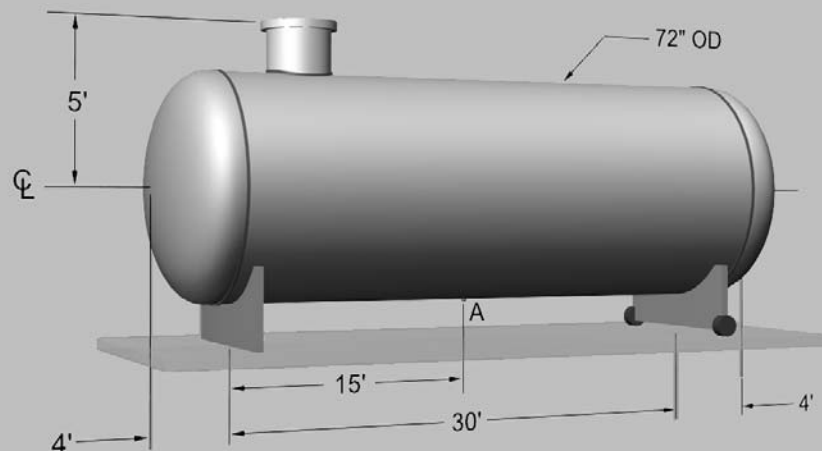


Figure 7.4 Vessel in Example 7.3.

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.

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

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

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:

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

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

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 $1\frac{1}{4}$ inch nominal thickness. Since 1.25 inch is equal to $1\frac{1}{4}$ 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 $1\frac{1}{4}$ inch. Since 1.25 inch is equal to $1\frac{1}{4}$ 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.

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

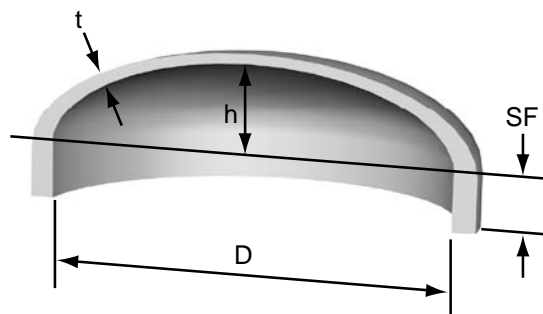


Figure 8.1 Ellipsoidal head.

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

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

or

$$P = \frac{2SEt}{D + 0.2t} \quad (8.1)$$

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

$$P = \frac{2SEt}{KD + 0.2t} \quad (8.2)$$

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

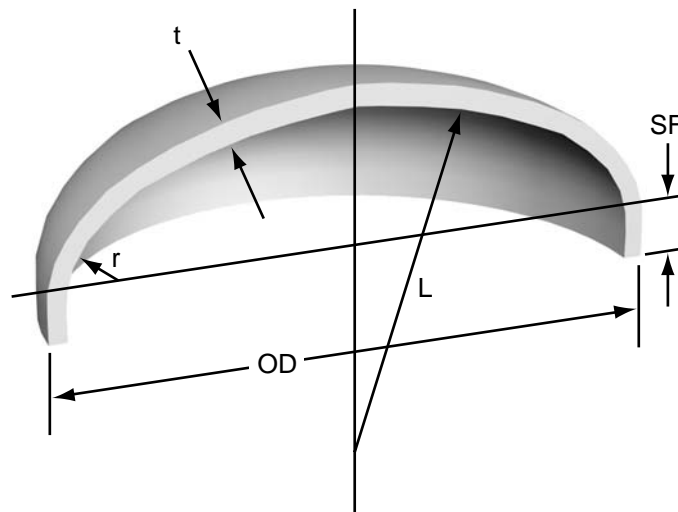


Figure 8.2 Torispherical head.

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.

$$t_{\text{req}} = 1.350 \text{ in.} + 0.125 \text{ in.}$$

$$t_{\text{req}} = 1.475 \text{ inch}$$

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.

$$L = 0.80[120 \text{ in.} + 2(0.88 \text{ in.})]$$

$$L = 97.4 \text{ inches}$$

$$r = 0.10(121.8 \text{ in.})$$

$$r = 12.18 \text{ inches}$$

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

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

$$\text{☐} \quad M = \frac{1}{4} \left(3 + \sqrt{\frac{97.4 \text{ in.}}{12.18 \text{ in.}}} \right)$$

$$M = 1.46$$

M may also be obtained from Section VIII, Division 1, Table 1-4.2.

$$t = \frac{150 \text{ psi}(97.4 \text{ in.})(1.46)}{2(12,000 \text{ psi})(1) - 0.2(150 \text{ psi})}$$

$$t = 0.89 \text{ inch}$$

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 $\frac{1}{2}$, 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)

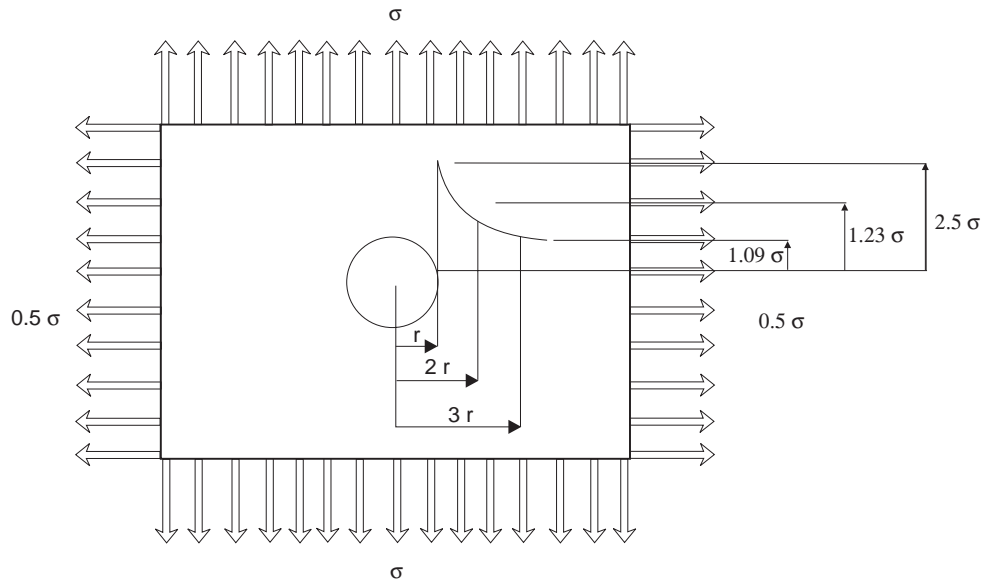


Figure 9.1 Stress distribution in flat plate with circular opening.

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 $\frac{3}{8}$ inch (10 mm) thickness or less, the maximum opening is $3\frac{1}{2}$ inches (89 mm) diameter
 - in plate greater than $\frac{3}{8}$ inch (10 mm) thickness, the maximum opening is $2\frac{3}{8}$ inches (60 mm) diameter
- (b) For threaded, studded, or expanded connections:
- in all plate thicknesses, the maximum opening is $2\frac{3}{8}$ 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).

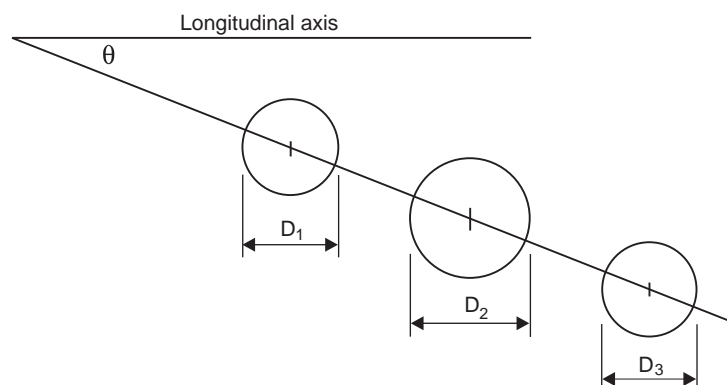


Figure 9.2 Center spacing of openings.

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.]

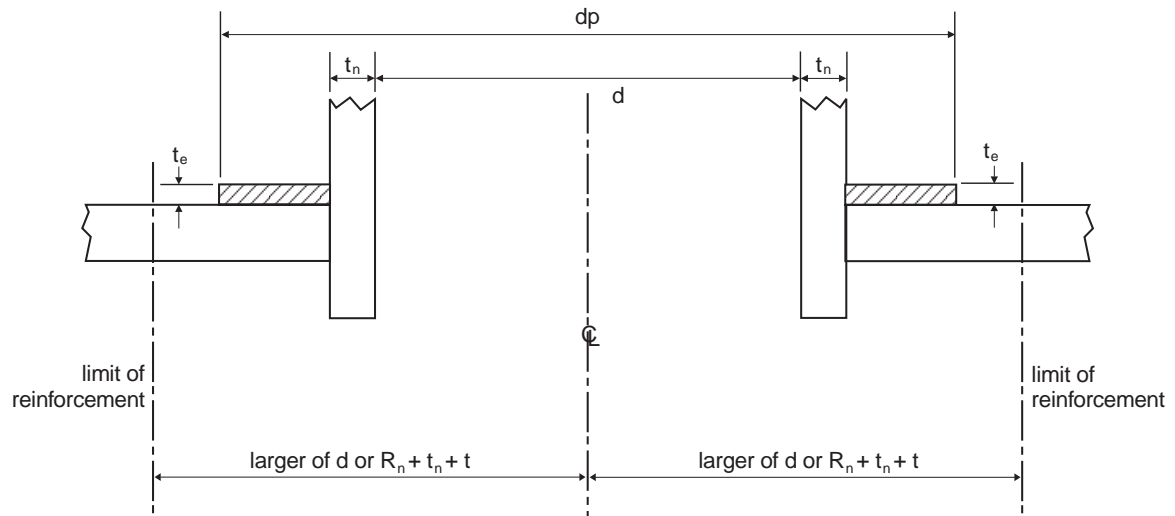


Figure 9.7 Reinforcement pad area A_5 .

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 $\frac{1}{2}$ of the removed area. The $\frac{1}{2}$ 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:

$$A = 0.5dt + tt_n(1-f_{r1})$$

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 $\frac{1}{2}$ the head diameter, and no pair with an average diameter greater than $\frac{1}{4}$ of the head diameter may be reinforced as single openings using the following formula to determine the required reinforcement:

$$A = 0.5dt + tt_n (1-f_{r1})$$

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 $[p-d_{ave}/p]$, 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 $\frac{1}{4}$ 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 $\frac{1}{4}$ 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 $\frac{1}{4}$ 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 $\frac{1}{4}$ the diameter of the smallest opening diameter.

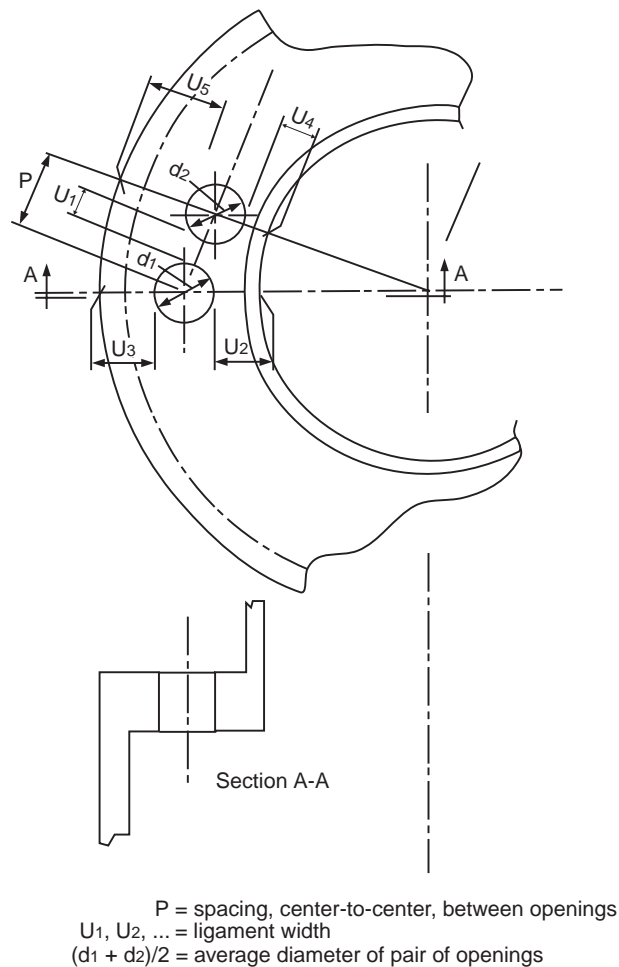


Figure 9.8 Multiple openings in rim of heads with a large central opening. Fig. UG-39.

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 $1\frac{1}{2}$ 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-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 $\frac{5}{8}$ 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 $\frac{3}{8}$ inch thick and 5.0 inches wide. The configuration is similar to Fig. UW-16.1 (h). The repad-to-nozzle weld leg is $\frac{3}{8}$ inch and the repad-to-head weld leg is $\frac{5}{16}$ 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?

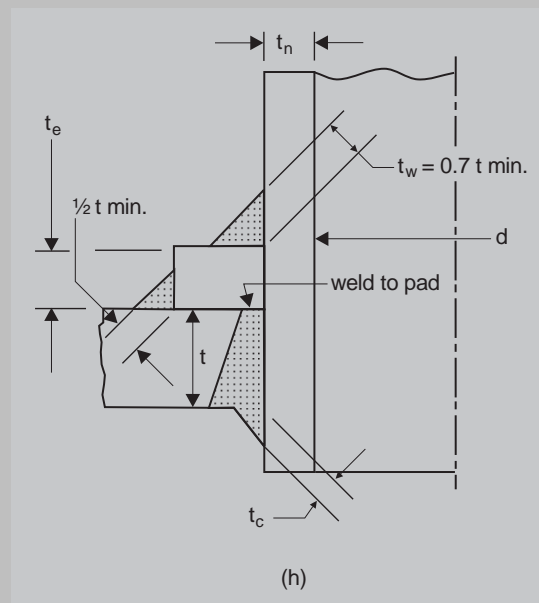


Figure 9.15 Nozzle design for Example 9.1.

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.

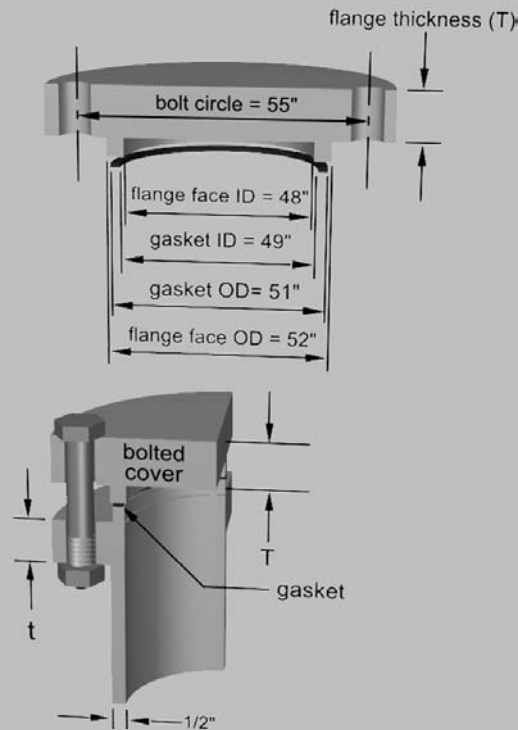


Figure 10.1 Geometry for the bolted flathead cover.

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.

$$N = 1.0 \text{ inch from Table 2-5.2}$$

$$b_o = N/2 = 1.0 \text{ in.}/2 = 0.5 \text{ inch from Table 2-5.2}$$

$$b = 0.5\sqrt{b_o} = 0.5\sqrt{0.5 \text{ in.}}$$

$$b = 0.3535 \text{ inch}$$

$$G = G_{OD} - 2b$$

$$G = 51.0 \text{ in.} - 2(0.3535 \text{ in.})$$

$$G = 50.293 \text{ inches}$$

Determine bolt loads per paragraph 2-5.

$$H = 0.785G^2p$$

$$H = 0.785 (50.293 \text{ in.})^2 (150 \text{ psi})$$

$$H = 297,800 \text{ pounds}$$

$$H_p = 2b\pi Gmp$$

$$H_p = 2(0.3535 \text{ in.}) \pi (50.293 \text{ in.}) (2.5) (150 \text{ psi})$$

$$H_p = 41,900 \text{ pounds}$$

$$W_{m1} = H + H_p = 297,800 \text{ lb} + 41,900 \text{ lb}$$

$$W_{m1} = 339,700 \text{ pounds}$$

$$W_{m2} = \pi bGy = \pi (0.3535 \text{ in.}) (50.293 \text{ in.}) (10,000 \text{ psi})$$

$$W_{m2} = 558,500 \text{ pounds}$$

Calculate total bolt area and determine the number and size of bolts.

$$A_{m1} = \frac{W_{m1}}{Sb_o}$$

$$A_{m1} = \frac{339,700 \text{ lb}}{21,000 \text{ psi}} = 16.178 \text{ inches}^2$$

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

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._____

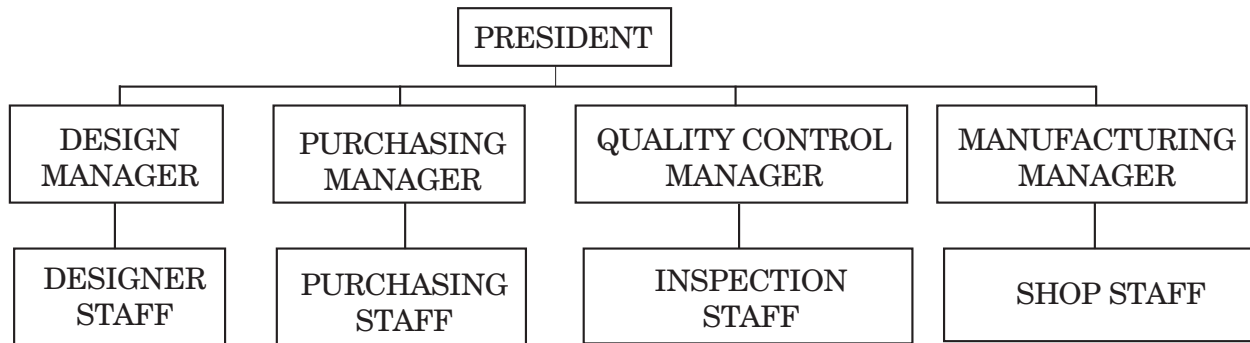
(Each page of the manual should be identified with the date of issue or a revision number. This identifies the edition of the manual that is being used. It also facilitates updating the manual without requiring a complete replacement of the document every time a minor change is made.)

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(The Table of Contents listing is not a requirement, but it does facilitate use of the document.)

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. _____

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.

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.)

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

| 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 |
| 1/8 | 0.405 | --- | 0.049 | --- | --- | --- | 0.068 | 0.068 |
| 1/4 | 0.540 | --- | 0.065 | --- | --- | --- | 0.088 | 0.088 |
| 3/8 | 0.675 | --- | 0.065 | --- | --- | --- | 0.091 | 0.091 |
| 1/2 | 0.840 | 0.065 | 0.083 | --- | --- | --- | 0.109 | 0.109 |
| 3/4 | 1.050 | 0.065 | 0.083 | --- | --- | --- | 0.113 | 0.113 |
| 1 | 1.315 | 0.065 | 0.109 | --- | --- | --- | 0.133 | 0.133 |
| 1 1/4 | 1.660 | 0.065 | 0.109 | --- | --- | --- | 0.140 | 0.140 |
| 1 1/2 | 1.900 | 0.065 | 0.109 | --- | --- | --- | 0.145 | 0.145 |
| 2 | 2.375 | 0.065 | 0.109 | --- | --- | --- | 0.154 | 0.154 |
| 2 1/2 | 2.875 | 0.083 | 0.120 | --- | --- | --- | 0.203 | 0.203 |
| 3 | 3.5 | 0.083 | 0.120 | --- | --- | --- | 0.216 | 0.216 |
| 3 1/2 | 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.

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