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9

Pressure Vessels

9.1 Design Criteria

Design Loads • Materials • Allowable Stress

9.2 Design Formulas

9.3 Opening Reinforcement

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Pressure vessels used in industry are leak-tight pressure containers, usually cylindrical or spherical in shape, with different head configurations. They are usually made from carbon or stainless steel and assembled by welding. Early operation of pressure vessels and boilers resulted in numerous explosions, causing loss of life and considerable property damage. Some 80 years ago, the American Society of Mechanical Engineers formed a committee for the purpose of establishing minimum safety rules of construction for boilers. In 1925 the committee issued a set of rules for the design and construction of unfired pressure vessels. Most states have laws mandating that these **Code** rules be met. Enforcement of these rules is accomplished via a third party employed by the state or the insurance company. These Codes are living documents in that they are constantly being revised and updated by committees composed of individuals knowledgeable on the subject. Keeping current requires that the revised Codes be published every three years with addendas issued every year. This chapter covers a very generalized approach to pressure vessel design based on the ASME Boiler and Pressure Vessel Code, Section VIII, Division 1: Pressure Vessels.

9.1 Design Criteria

The Code design criteria consist of basic rules specifying the design method, design load, allowable stress, acceptable material, and fabrication—inspection certification requirements for vessel **construction**. The design method known as "design by rule" uses design pressure,

allowable stress, and a design formula compatible with the geometry of the part to calculate the minimum required thickness of the part. This procedure minimizes the amount of analysis required to ensure that the vessel will not rupture or undergo excessive distortion. In conjunction with specifying the vessel thickness, the Code contains many construction details that must be followed. Where vessels are subjected to complex loadings such as cyclic, thermal, or localized loads, and where significant discontinuities exist, the Code requires a more rigorous analysis to be performed. This method is known as the "design by analysis" method. A more complete background of both methods can be found in Bernstein, 1988.

The ASME Code [1994] is included as a standard by the American National Standards Institute (ANSI). The American Petroleum Institute (API) has also developed codes for low-pressure storage tanks, and these are also part of the ANSI standards. The ASME Boiler and Pressure Vessel Code has been used worldwide, but many other industrialized countries have also developed boiler and pressure vessel codes. Differences in these codes sometimes cause difficulty in international trade.

Design Loads

The forces that influence pressure vessel design are internal/external pressure; dead loads due to the weight of the vessel and contents; external loads from piping and attachments, wind, and earthquakes; operating-type loads such as vibration and sloshing of the contents; and startup and shutdown loads. The Code considers design pressure, design temperature, and, to some extent, the influence of other loads that impact the **circumferential** (or hoop) and **longitudinal stresses** in shells. It is left to the designer to account for the effect of the remaining loads on the vessel. Various national and local building codes must be consulted for handling wind and earthquake loadings.

Materials

The materials to be used in pressure vessels must be selected from Code-approved material specifications. This requirement is normally not a problem since a large catalogue of tables listing acceptable materials is available. Factors that need to be considered in picking a suitable table are:

- Cost

- Fabricability

- Service condition (wear, corrosion, operating temperature)

- Availability

- Strength requirements

Several typical pressure vessel materials for a noncorrosive environment and for service temperatures between -50°F and 1000°F are shown in [Table 9.1](#).

Table 9.1 Acceptable Pressure Vessel Materials

Temperature Use Limit (\pm F)	Plate Material	Pipe Material	Forging Material
Down to -50	SA-516 ^a	SA 333	SA 350
	All grades	Gr. 1	Gr. LF1, LF2
+33 to +775	SA-285	SA-53	SA-181
	Gr. C	SA-106	Gr. I, II
	SA-515		
	Gr. 55, 60, 65		
+776 to +1000	SA-516		
	All grades		
	SA-204	SA-335	SA-182
	Gr. B, C	Gr. P1, P11,	Gr. F1, F11,
	SA-387	P12	F12
	Gr. 11, 12		
	Class 1		

^a Impact testing required.

Note: SA is a classification of steel used in the ASME Boiler and Pressure Vessel Code.

Allowable Stress

The allowable stress used to determine the minimum vessel wall thickness is based on the material tensile and yield properties at room and design temperature. When the vessel operates at an elevated temperature, the creep properties of the material must also be considered. These properties are adjusted by design factors which limit the **hoop membrane stress** level to a value that precludes rupture, and excessive elastic or plastic distortion and creep rupture. [Table 9.2](#) shows typical allowable stresses for several carbon steels commonly used for unfired pressure vessels.

Table 9.2 Typical Allowable Stresses for Use in Pressure Vessel Design

Material Specification	Temperature Use Limit (\pm F)	Allowable Stress (psi)
SA-515 Gr. 60	700	14 400
	800	10 800
	900	6 500
SA-516 Gr. 70	700	16 600
	800	14 500

	900	12 000
SA-53 Gr. A	700	11 700
	800	9 300
	900	6 500
SA-106 Gr. B	700	14 400
	800	10 800
	900	6 500
SA-181 Gr. I	700	16 600
	800	12 000
	900	6 500

9.2 Design Formulas

The design formulas used in the "design by rule" method are based on the principal stress theory and calculate the average hoop stress. The principal stress theory of failure states that failure occurs when one of the three principal stresses reaches the yield strength of the material. Assuming that the radial stress is negligible, the other two principal stresses can be determined by simple formulas based on engineering mechanics.

The Code recognizes that the shell thickness may be such that the radial stress may not be negligible, and adjustments have been made in the appropriate formulas. [Table 9.3](#) shows various formulas used to calculate the wall thickness for numerous pressure vessel geometries.

Table 9.3 Code Formulas for Calculation of Vessel Component Thickness

Cylindrical shell	$t = \frac{PR}{SE_j 0.6P}$
Hemispherical head or spherical shell	$t = \frac{PR}{2SE_j 0.2P}$
2:1 ellipsoidal head	$t = \frac{PD}{2SE_j 0.2P}$
Flanged and dished head	$t = \frac{1.77PL}{2SE_j 0.2P}$
Flat head	$t = d \sqrt{\frac{CP}{SE}}$

where

t = Minimum required thickness (in.)

P = Design pressure (psi)

R = Inside radius (in.)

S = Allowable stress (psi)

D = Inside diameter (in.)

L = Inside spherical crown radius (in.)

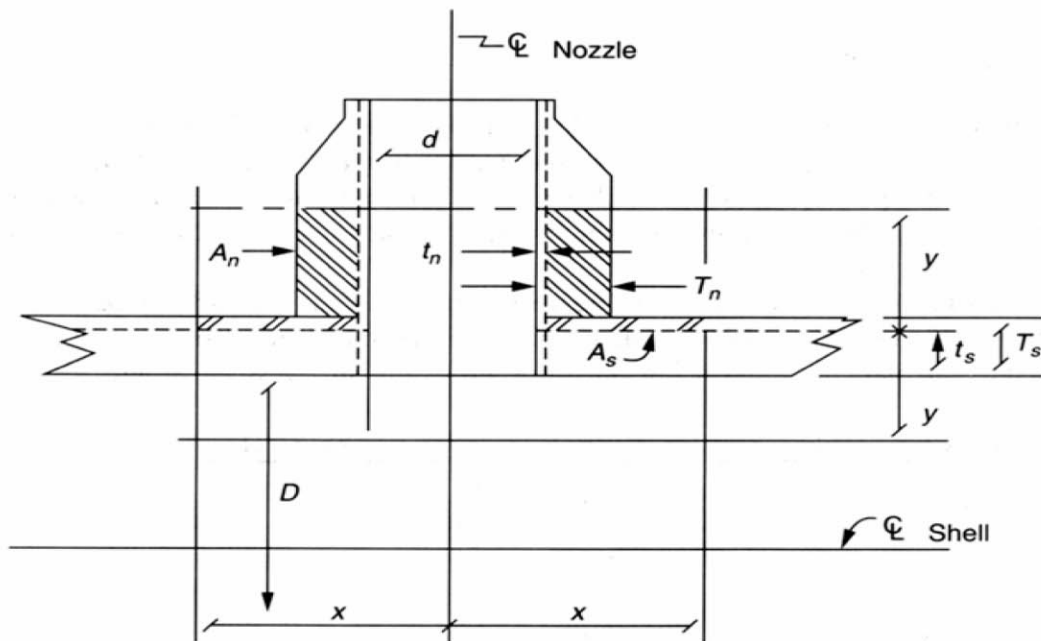
E = Weld joint efficiency factor, determined by joint location and degree of examination

C = Factor depending upon method of head-to-shell attachment

9.3 Opening Reinforcement

Vessel components are weakened when material is removed to provide openings for nozzles or access. High **stress concentrations** exist at the opening edge and decrease radially outward from the opening, becoming negligible beyond twice the diameter from the center of the opening. To avoid failure in the opening area, compensation or reinforcement is required. Some ways in which this can be accomplished are: (a) increase the vessel wall thickness, (b) increase the wall thickness of the nozzle, or (c) use a combination of extra shell and nozzle thickness. The Code procedure is to relocate the removed material to an area within an effective boundary around the opening. [Figure 9.1](#) shows the steps necessary to reinforce an opening in a pressure vessel. Numerous assumptions have been made with the intent of simplifying the general approach.

Figure 9.1 Opening Reinforcement Requirements.



x = Larger of d or $R_n + t_n + T_n$
 y = Smaller of $2\frac{1}{2}T_s$ or $2\frac{1}{2}T_n$
 d = Diameter of circular opening (in.)
 D = Inside diameter of shell (in.)
 t_s = Required thickness of shell (in.)
 T_s = Actual thickness of shell (in.)
 t_n = Required thickness of nozzle (in.)
 T_n = Actual thickness of nozzle (in.)
 R_n = Inside radius of nozzle = $d/2$ (in.)

A_r = Area of required reinforcement (in.²)
 A_s = Area available in the shell (in.²)
 A_n = Area available in the nozzle (in.²)
 $A_r = (d)(t_s)$
 A_s = Larger of: $d(T_s - t_s) - 2T_n(T_s - t_s)$ or
 $2(T_s + t_n)(T_s - t_s) - 2t_n(T_s - t_s)$
 A_n = Smaller of: $2[2\frac{1}{2}(T_s)(T_n - t_n)]$ or
 $2[2\frac{1}{2}(T_n)(T_n - t_n)]$
 $A_r < (A_s + A_n)$: Acceptable configuration

The example shown in [Fig. 9.2](#) uses the design approach indicated by the Code to perform a simple sizing calculation for a typical welded carbon steel vessel. [Figure 9.3](#) shows a typical shell-nozzle juncture and head-shell juncture which meet the code requirements. Design specifications for the many associated vessel parts such as bolted flanges, external attachments, and saddle supports can be found in the reference materials.

Figure 9.2 Sample vessel calculation.

DESIGN SPECIFICATION

Design pressure = 700 psi

Design temperature = 700° F

Material:

Shell SA-516 Gr. 70

Head SA-181 Class 70

Nozzle SA-106 Gr. B

Weld efficiency factor = 1.0 = E
(full radiographic examination)

Shell Thickness

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

$$= \frac{700(30)}{16600(1.0) - 0.6(700)}$$

$$= 1.30 \text{ in. Use } 1\frac{1}{2}'' = T_s$$

P = 700 psi

R = 30 in.

E = 1.0

S = 16 600 psi (SA-516 Gr. 70, Table 9.2)

Hemispherical Head Thickness

$$t_h = \frac{PR}{2SE - 0.2P}$$

$$= \frac{700(30)}{2(16600)(1.0) - 0.2(200)}$$

$$= 0.64 \text{ in. Use } 1'' = T_h$$

Nozzle Thickness

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

$$= \frac{700(4)}{16600(1.0) - 0.6(700)}$$

$$= 0.17 \text{ in. Use } 1\frac{3}{4}'' = T_n$$

P = 700 psi

R = 4 in.

E = 1.0

S = 16 600 psi (SA-106 Gr. B, Table 9.2)

Opening Reinforcement Calculation (see Fig. 9.1)*

$$A_{req'd} = (d)(t_s) = (8)(1.3) = 10.4 \text{ in.}^2$$

$$A_n = \text{Smaller of: } 2[2\frac{1}{2}(T_s)(T_n - t_n)] \text{ or}$$

$$2[2\frac{1}{2}(T_n)(T_n - t_n)]$$

$$T_s < T_n \text{ Use } T_s$$

$$2[2\frac{1}{2}(1\frac{1}{2})(1\frac{3}{4} - 0.17)] = 11.7 \text{ in.}^2$$

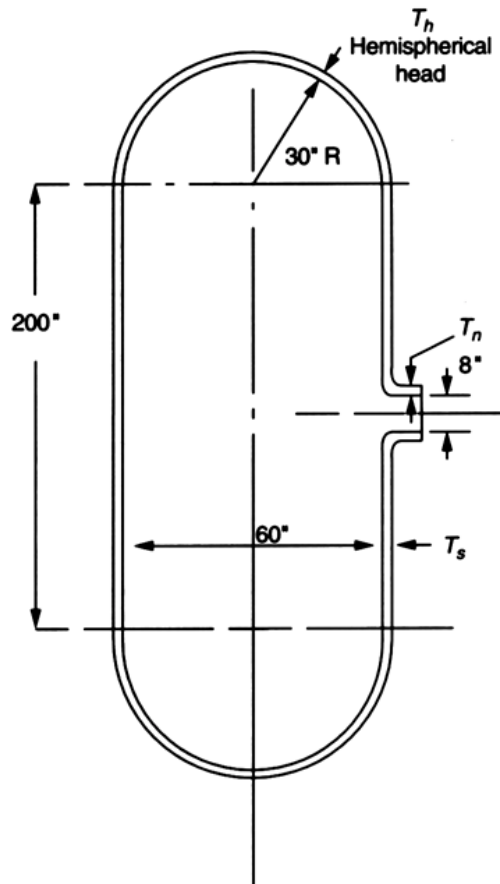
$$d = 8 \text{ in.}$$

$$t_s = 1.3 \text{ in.}$$

$$T_s = 1\frac{1}{2} \text{ in.}$$

$$T_n = 1\frac{3}{4} \text{ in.}$$

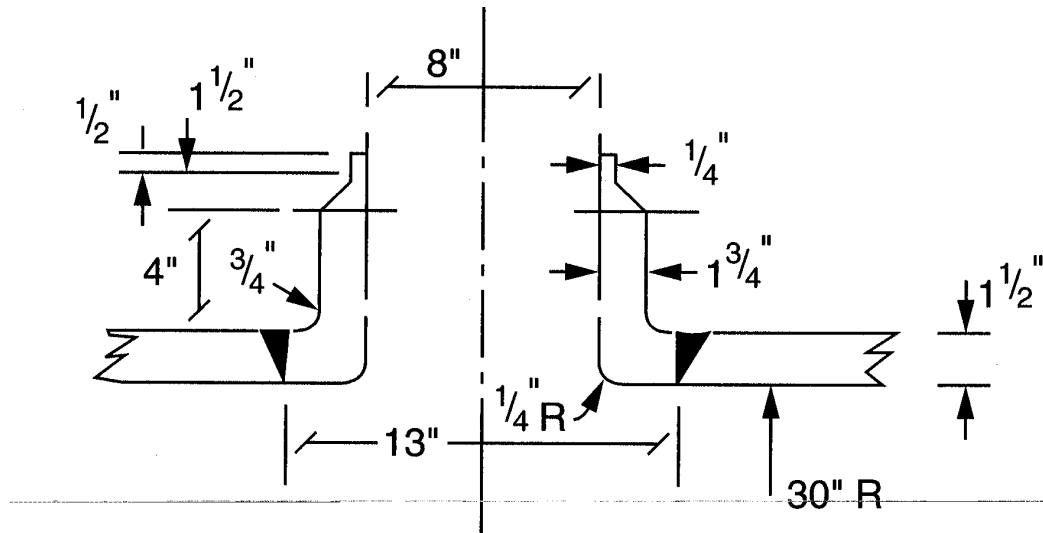
$$t_n = 0.17 \text{ in.}$$



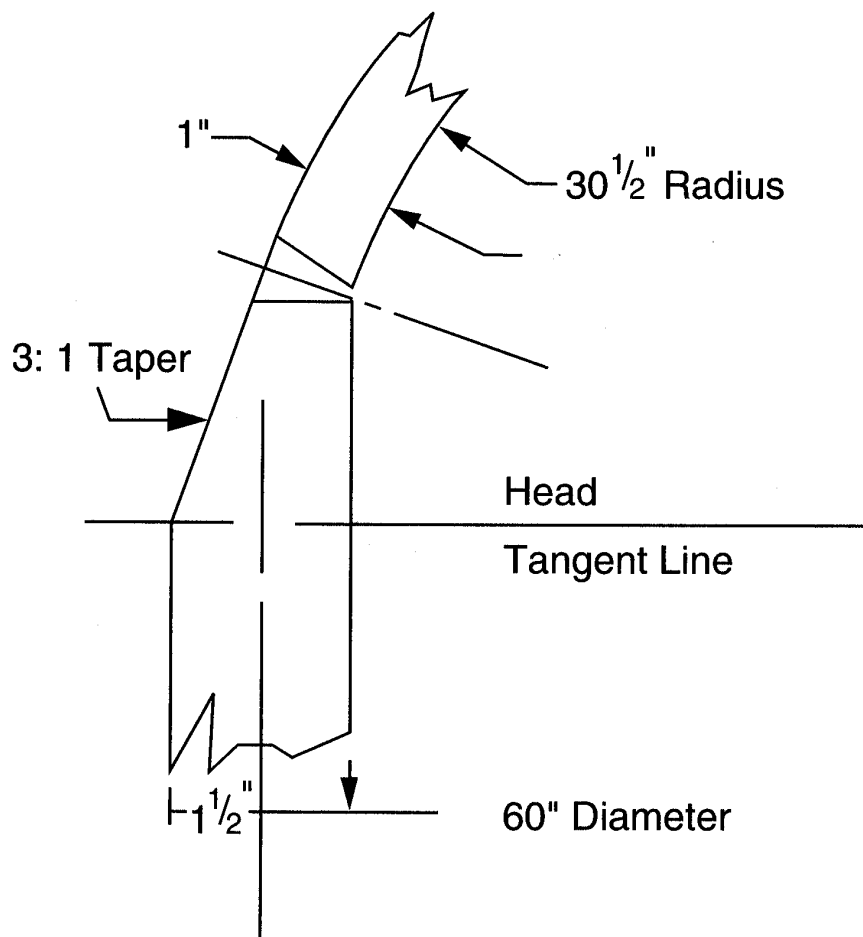
*This example places all the required reinforcement in the nozzle wall since it is more economical than increasing the shell thickness.

$$A_n = 11.7 \text{ in.}^2 > A_{req'd} = 10.4 \text{ in.}^2 \text{ — OK}$$

Figure 9.3 Fabrication details.



Nozzle-Shell Junction



Head-Shell Junction

Defining Terms

Code: The complete rules for construction of pressure vessels as identified in ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, Pressure Vessels.

Construction: The complete manufacturing process, including design, fabrication, inspection, examination, hydrotest, and certification. Applies to new construction only.

Hoop membrane stress: The average stress in a ring subjected to radial forces uniformly distributed along its circumference.

Longitudinal stress: The average stress acting on a cross section of the vessel.

Pressure vessel: A leak-tight pressure container, usually cylindrical or spherical in shape, with pressure usually varying from 15 psi to 5000 psi.

Stress concentration: Local high stress in the vicinity of a material discontinuity such as a change in thickness or an opening in a shell.

Weld efficiency factor: A factor which reduces the allowable stress. The factor depends on the degree of weld examination performed during construction of the vessel.

References

- American Society of Mechanical Engineers. 1994. *ASME Boiler and Pressure Vessel Code, Section VIII Division 1, Pressure Vessels*. ASME, New York.
- Bednar, H. H. 1981. *Pressure Vessel Design Handbook*. Van Nostrand Reinhold, New York.
- Bernstein, M. D. 1988. Design criteria for boiler and pressure vessels in the U.S.A. *ASME J. F* -443.
- Jawad, M. H. and Farr, J. R. 1989. *Structural Analysis and Design of Process Equipment*, 2nd ed. John Wiley & Sons, New York.

Further Information

Each summer, usually in June, the Pressure Vessels and Piping Division of the American Society of Mechanical Engineers organizes an annual meeting devoted to pressure vessel technology. Usually 300 to 500 papers are presented, many of which are published by ASME in booklets called special publications. Archival papers are also published in the *Journal of Pressure Vessel Technology*.

Research for ASME Boiler and Pressure Code work is often conducted by the Pressure Vessel Research Council (PVRC). This research is normally published in Welding Research Council bulletins. These bulletins, which number about 400, are an excellent documentation of major research contributions in the field of pressure vessel technology. The Electric Power Research Institute (EPRI) located in Palo Alto, California, also conducts extensive research for the electric power industry, and this work includes some research on pressure