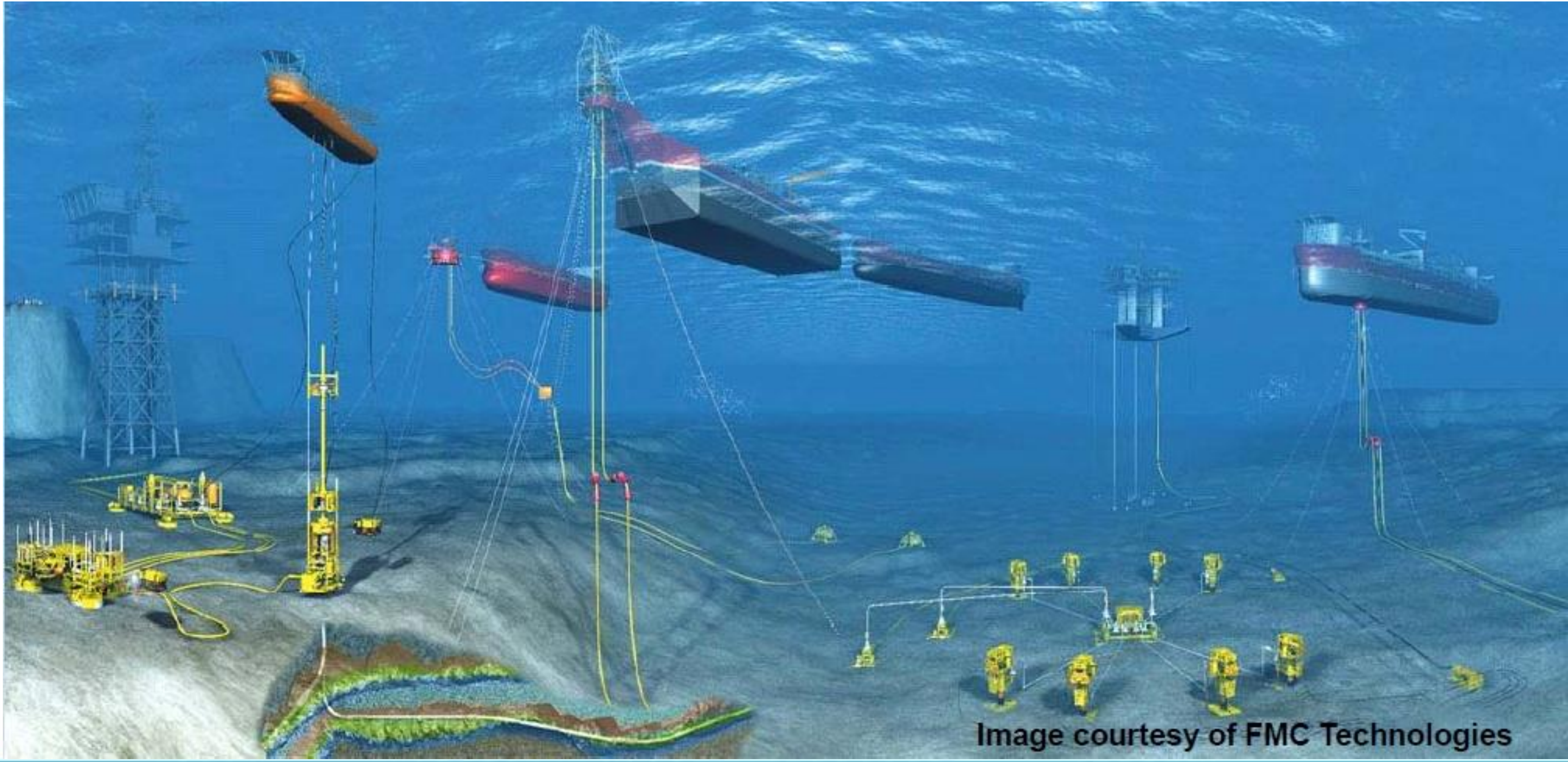


Offshore Equipment

Yutaek Seo

Mechanical design of pressure vessel

Introduction

- Chapters 4 and 5 discuss the concepts for determining the diameter and length of two-phase and three-phase vertical and horizontal separators.
- This chapter addresses the selection of design pressure rating and wall thickness of pressure vessels.
- The purpose of this chapter is to present an overview of simple concepts of mechanical design of pressure vessels that must be understood by a project engineer specifying and purchasing this equipment.
- Most pressure vessels used in the oil and gas industry are designed and inspected according to the American Society of Mechanical Engineers' Boiler and Pressure Vessel Code (ASME code).
- In particular, Section VIII of the code, "Pressure Vessels," is particularly important. Countries that do not use the ASME code have similar documents and requirements.
- The use of non-code vessels should be discouraged to assure vessel mechanical integrity.

Design considerations

Design temperature

- The maximum and minimum design temperatures for a vessel will determine the maximum allowable stress value permitted for the material to be used in the fabrication of the vessel.
- maximum temperature used in the design should not be less than the mean metal temperature expected under the design operating conditions.
- The minimum temperature used in the design should be the lowest expected in service except when lower temperatures are permitted by the rules of the ASME code.

  : lowest operating temperature, operational upset, auto-refrigeration, ambient temperature, and any other source of cooling.
- The metal temperature should be determined by computation using accepted heat transfer procedures or by measurement from equipment in service

Design pressure

- The design pressure for a vessel is called its "maximum allowable working pressure" (MAWP) or "working pressure".
- The MAWP determines the setting of the relief valve and must be higher than the normal pressure of the process contained in the vessel, which is called the vessel's "operating pressure."
- If the operating pressure is too close to the relief valve setting, small surges in operating pressure could cause the relief valve to activate prematurely.

Table 6-1
Setting Maximum Allowable Working Pressures

Operating Pressure	Minimum Differential Between Operating Pressure and MAWP
Less than 50 psig	10 psi
51–250 psig	25 psi
251–500 psig	10% of maximum operating pressure
501–1000 psig	50 psi
1001 psig and higher	5% of maximum operating pressure

Vessels with high-pressure safety sensors have an additional 5% or 5 psi, whichever is greater to the minimum differential.

- Often, especially for small vessels, it is advantageous to use a higher MAWP than is recommended in Table 6-1.
- It may be possible to increase the MAWP at little or no cost and thus have greater future flexibility if process changes (e.g., greater throughput) require an increase in operating pressure.
- The MAWP of the vessel cannot exceed the MAWP of the nozzles, valves, and pipe connected to the vessel, which are manufactured in accordance with industry standard pressure rating classes.
- Once a preliminary MAWP is selected, it is necessary to calculate a wall thickness for the shell and heads of the pressure vessel.

Table 6-2
Summary ANSI Pressure Ratings Material Group 1.1

	MAWP, psig	
Class	**–20°F to 100°F**	**100°F to 200°F**
150	285	250
300	740	675
400	990	900
600	1480	1350
900	2220	2025
1500	3705	3375
2500	6170	5625

Maximum allowable stress values

- The maximum allowable stress values to be used in the calculation of a vessel's wall thickness are given in the ASME code for many different materials.
- These stress values are a function of temperature. Section VIII of the ASME code, which governs the design and construction of all pressure vessels with operating pressures greater than 15 psig, is published in two divisions.
- Each sets its own maximum allowable stress values. Division 1, governing the *design by rules*, is less stringent from the standpoint of certain design details and inspection procedures, and thus incorporates a higher safety factor.

  : The 1998 edition incorporates a safety factor of 4

  : The 2001 and later editions incorporate a safety factor of 3.5.

  : The 2001 edition of the code yields higher allowable stresses and thus smaller wall thicknesses.

  : For example, using a material with a 60,000-psi tensile strength, a vessel built under the 1998 edition (safety factor = 4) yields a maximum allowable stress value of 15,000 psi while a vessel built under the 2001 edition (safety factor = 3.5) yields a maximum allowable stress value of 17,142 psi.
- On the other hand, Division 2 governs the *design by analysis* and incorporates a lower safety factor of 3.

  : Thus, the maximum allowable stress value for a 60,000-psi tensile strength material will become 20,000 psi.

- Many companies require that all their pressure vessels be constructed in accordance with Division 2 because of the more exacting standards.
- Others find that they can purchase less expensive vessels by allowing manufacturers the choice of either Division 1 or Division 2. Normally, manufacturers will choose Division 1 for low-pressure vessels and Division 2 for high-pressure vessels. (Otherwise wall may too thick for high pressure vessels)
- The maximum allowable stress values at normal temperature range for the steel plates most commonly used in the fabrication of pressure vessels are given in Table 6-3.
- For stress values at higher temperatures and for other materials, the latest edition of the ASME code should be referenced.

Table 6-3
Maximum Allowable Stress Value for Common Steels (2007 Edition)

			ASME Section VIII 2007 Edition	
			Div. 1	Div. 2
Metal		**Not Lower Than**	**–20°F**	**–20°F**
Temperature		**Not Exceeding**	**650°F**	**100°F**
Carbon steel plates and sheets	SA-516	Grade 55	15,700	18,300
		Grade 60	17,100	20,000
		Grade 65	18,600	21,700
		Grade 70	20,000	23,300
	SA-285	Grade A	12,900	15,000
		Grade B	14,300	16,700
		Grade C	15,700	18,300
	SA-36		16,600	16,900
Low-alloy steel plates	SA-387	Grade 2, cl.1	15,700	18,300
		Grade 12, cl.1	15,700	18,300
		Grade 11, cl.1	17,100	20,000
		Grade 22, cl.1	17,100	20,000
		Grade 21, cl.1	17,100	20,000
		Grade 5, cl.1	17,100	20,000
		Grade 2, cl.2	20,000	23,300
		Grade 12, cl.2	18,600	21,700
		Grade 11, cl.2	21,400	25,000
		Grade 22, cl.2	21,400	25,000
		Grade 21, cl.2	21,400	25,000
		Grade 5, cl.2	21,400	25,000
	SA-203	Grade A	18,600	21,700
		Grade B	20,000	23,300
		Grade D	18,600	21,700
		Grade E	20,000	23,300
High-alloy steel plates	SA-240	Grade 304	20,000	20,000**
		Grade 304L	16,700	16,700
		Grade 316	20,000	20,000
		Grade 316L	16,700	16,700

Austenitic Stainless set **at 2/3 Yield /** Allowable Stress NOT **3.0 or 3.5 S.F** due to low Yield Strength values relative to ultimate Tensile Strength, 304 UTS 75,000 Yield 30,000
Example: Hydrostatic Testing 1.3 × 20,000 = 26,000 (**Yield is 30,000**) for 304

Determining wall thickness

- ASME code section VIII, Division 1

Wall Thickness—Cylindrical Shells

$$t = \frac{\mathrm{Pr}}{SE - 0.6P}, \tag{6-1}$$

Wall Thickness—2:1 Ellipsoidal Heads

$$t = \frac{Pd}{2SE - 0.2P}, \tag{6-2}$$

Wall Thickness—Hemispherical Heads

$$t = \frac{\mathrm{Pr}}{2SE - 0.2P}, \tag{6-3}$$

Wall Thickness—Cones

$$t = \frac{Pd}{2\cos\alpha(SE - 0.6P)}, \tag{6-4}$$

where

S = maximum allowable stress value, psi (kPa),

t = thickness, excluding corrosion allowance, in. (mm),

P = maximum allowable working pressure, psig (kPa),

r = inside radius before corrosion allowance is added, in. (mm),

d = inside diameter before corrosion allowance is added, in. (mm),

E = joint efficiency, see Table 6-4 (most vessels are fabricated in accordance with type of joint no. 1),

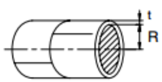
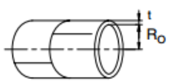
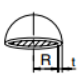
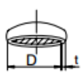
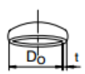
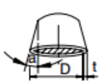
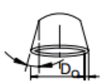
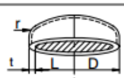
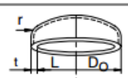
α = half the angle of the apex of the cone.

- Figure 6-1 summarizes the formulas for pressure vessels under internal pressure (ASME Section VIII, Division 1).

FORMULAS FOR VESSELS UNDER INTERNAL PRESSURE

NOTATION

α = Half Apex Angle of Cone, Deg.
D = Inside diameter, inches
D_O = Outside diameter, inches
E = Efficiency of welded joints
L = Inside crown radius, inches
L_O = Outside crown radius, inches
M = Factor, see table below
P = Design pressure or maximum allowable pressure, psig

In Terms INSIDE Radius or Diameter		In Terms OUTSIDE Radius or Diameter	
 Cylindrical Shell	$t=\frac{PR}{SE-0.6P}$ $P=\frac{SE\,t}{R+0.6t}$ Formulas for Longitudinal seam	 Cylindrical Shell	$t=\frac{PR_O}{SE+0.4P}$ $P=\frac{SE\,t}{R_O+0.4t}$ Formulas for Longitudinal seam
 Sphere Hemispherical Head	$t=\frac{PR}{2SE-0.2P}$ $P=\frac{2SE\,t}{R+0.2t}$	 Sphere Hemispherical Head	$t=\frac{PR_O}{2SE-0.8P}$ $P=\frac{2SE\,t}{R_O+0.8t}$
 2:1 Ellipsoidal Head	$t=\frac{PD}{2SE-0.2P}$ $P=\frac{2SE\,t}{D+0.2t}$	 2:1 Ellipsoidal Head	$t=\frac{PD_O}{2SE-1.8P}$ $P=\frac{2SE\,t}{D_O+1.8t}$
 Cone & Conical Section	$t=\frac{PD}{2\cos\alpha\,(SE-0.6P)}$ $P=\frac{2SE\,t\cos\alpha}{D+1.2t\cos\alpha}$ α Maximum = 30 Deg.	 Cone & Conical Section	$t=\frac{PD_O}{2\cos\alpha\,(SE-0.4P)}$ $P=\frac{2SE\,t\cos\alpha}{D_O+1.8t\cos\alpha}$ α Maximum = 30 Deg.
 Flanged & Dished Head	$t=\frac{PLM}{2SE-0.2P}$ $P=\frac{2SE\,t}{LM+0.2t}$	 Flanged & Dished Head	$t=\frac{PL_OM}{2SE+P(M-0.2)}$ $P=\frac{2SE\,t}{ML_O-t(M-0.2)}$

FACTOR																			
FACTOR M	L/t	6.5	7.5	8.0	8.5	9.0	9.5	10.00	10.5	11.0	11.5	12.00	13.0	14.0	15.0	16.0	16.67
	M	1.39	1.41	1.44	1.46	1.50	1.52	1.54	1.56	1.58	1.60	1.62	1.65	1.69	1.72	1.75	1.77

PRESSURE VESSEL HANDBOOK PUBLISHING, INC.
P.O.BOX 35365 - TULSA, OK. 74153-0365

FORMULAS FOR VESSELS UNDER INTERNAL PRESSURE

1. THE FORMULAS CONFORM TO THE ASME CODE FOR PRESSURE VESSELS. SECTION VIII, DIVISION L

2. CORROSION ALLOWANCE. If the vessel is subject to thinning by corrosion allowance shall be added to the thickness calculated by the formulas. Code UG-25.
When allowance for corrosion is provided computing the wall thickness or allowable pressure in terms of the inside dimensions, the allowance shall be added lo the length of Inside radius or diameter.

3. MAXIMUM ALLOWABLE WORKING PRESSURE for a vessel is the pressure allowable for the weakest element of the vessel. It should be limited not by minor parts but by the shell, heads of flanges.

4 HYDROSTATIC TEST PRESSURE o one and one-half times the maximum allowable working pressure or the design pressure when calculations are not made to determine the maximum allowable working pressure. Code UG-99.

5 DEGREE OF RADIOGRAPHIC EXAMINATION. Condition when full radiography is mandatory are described in the Code UW-11.

6 STRESS VALUES OF MATERIALS, S 1000 psi. Excerpt from Code, table UCS-73

Specification		For Metal Temperature Not Exceeding Deg. F.							
Number	Grade	-20 to 650	700	750	800	850	900	950	1010
SA-283	C	12.6	-	-	-	-		-	-
SA-285	C	13.7	13.2	12.0	10.2	8.3	6.5	-	-
SA-515	55	13.7	13.2	12.0	10.2	8.3	6.5	4.5	2.5
SA-515	60	15.0	14.3	12.9	10.8	8.6	6.5	4.5	2.5
SA-5I5	65	16.2	15.5	13.8	11.4	8.9	6 5	4 5	2.5
SA-515	70	17.5	16.6	14.7	12.0	9.2	6.5	4.2	2.5
SA-516	55	13.7	13.2	12.0	10.2	8.3	6.5	4.5	2.5
SA-516	60	15.0	14.3	12.9	10.8	8.6	6.5	4.5	2.5
SA-516	65	16.2	15.5	13.8	11.4	8.9	6 5	4.5	2.5
SA-516	70	17.5	16.6	14.7	12.0	9.2	6.5	4.5	2.5

7 CYLINDRICAL SHELL- In the formulas of the form the stress in the longitudinal joint are considered, since usually this governs.

Stress in the girth seam will govern only when the circumferential joint efficiency is less than half the longitudinal joint efficiency, or when besides the internal pressure additional loadings (wind load, reaction of saddles etc.) arc outing longitudinal bending or tension.

The formula considering the stress in the girth seam

$$t=\frac{PR}{2SE+0.4P} \qquad P=\frac{2SE\,t}{R-0.4t}$$

When the thickness of the shell exceed one half of the inside radius, or P exceeds 0.385 SE. The formulas given in the Code UA-2(c) shall be applied.

8 SPHERE AND HEMISPHERICAL HEAD. When the wall thickness exceeds 0.356 R, or P exceeds 0.665 SE, the formula given in the Code UA-3 shall be applied.

9 ELLIPSOIDAL HEAD. When the ratio of the major and minor axis is other than 2:1. Formulas and factors given in the Code UA-4(c) shall be applied.

10 CONE AND CONICAL SECTION. The formulas given in the form are applicable only when the half apex angle of cone does not exceed 30 degree.

11 ASME FLANGED AND DISHED HEAD. The formulas are applicable when the inside crown radius is not greater than the outside diameter of the head and the inside knuckle radius is not less than 6% of the outside diameter or 3 times the thickness of the head.

12 EFFICIENCY OF WELDED JOINTS (E)
Excerpt from ASME Code Table UW-12.

No.	TYPE OF JOINT	DEGREE OF EXAMINATION		
		FULL	SPOT	NO
1	Double-welded bolt joint or single welded bolt joint with backing strip which does not remain in place.	1.00	0.85	0.70
2	Single-welded bolt joint with backing strip which remains in place.	0.90	0.80	0.65
3	Single-welded bolt joint without use of backing strip.	—	—	0.60

13. EFFICIENCY (E) TO BE USED IN CALCULATIONS OF SEAMLESS HEADS ASME Code UW-12 (b)

TYPE OF HEAD	TYPE OF JOINT	DEGREE OF EXAMINATION OF HEAD TO SHELL JOINT		
		FULL	SPOT	NO
Hemi Spherical	No 1	1.00	0.85	0.60
	No 2	0.90	0.80	
Others	ANY	1.00	0.85	

14. NOTATION

a = Half-apex angle of cone, deg.
D = Inside diameter of cone at the large end, inches
Inside diameter of ellipsoidal head, inches
D_o = Outside diameter of cone at the large end, inches
Outside diameter of ellipsoidal head, inches
E = Lowest efficiency of any joint in the shell or head; for hemispherical head this includes head to shell joint
L = Inside crown radius of flanged and dished head, inches
L_o = Outside crown radius of flanged and dished head, inches
P = Internal design pressure or maximum allowable working pressure pounds per square inches
r = Knuckle radius
R = Inside radius of shell or hemispherical head, inches
R_o = Outside radius of shell or hemispherical head, inches
S = Allowable stress value of material pounds per square, inches
t = Thickness of shell or head, inches

Figure 6-1. Formulas for vessels under internal pressure (ASME Section VIII, Division 1). (Reprinted with permission from *Pressure Vessel Handbook*, Gulf Publishing, Inc., Tulsa, Oklahoma.)

- Figure 6-2 defines the various types of heads.
- Most production facility vessels use 2:1 ellipsoidal heads because they are readily available, are normally less expensive, and take up less room than hemispherical heads.
- Cone-bottom vertical vessels are sometimes used where solids are anticipated to be a problem.
- Most cones have either a 90º apex (α=45º) or a 60º apex (α=30º). These are referred to respectively as a “45º” or “60º” cone because of the angle each makes with the horizontal.
- Equation (6-4) is for the thickness of a conical head that contains pressure.

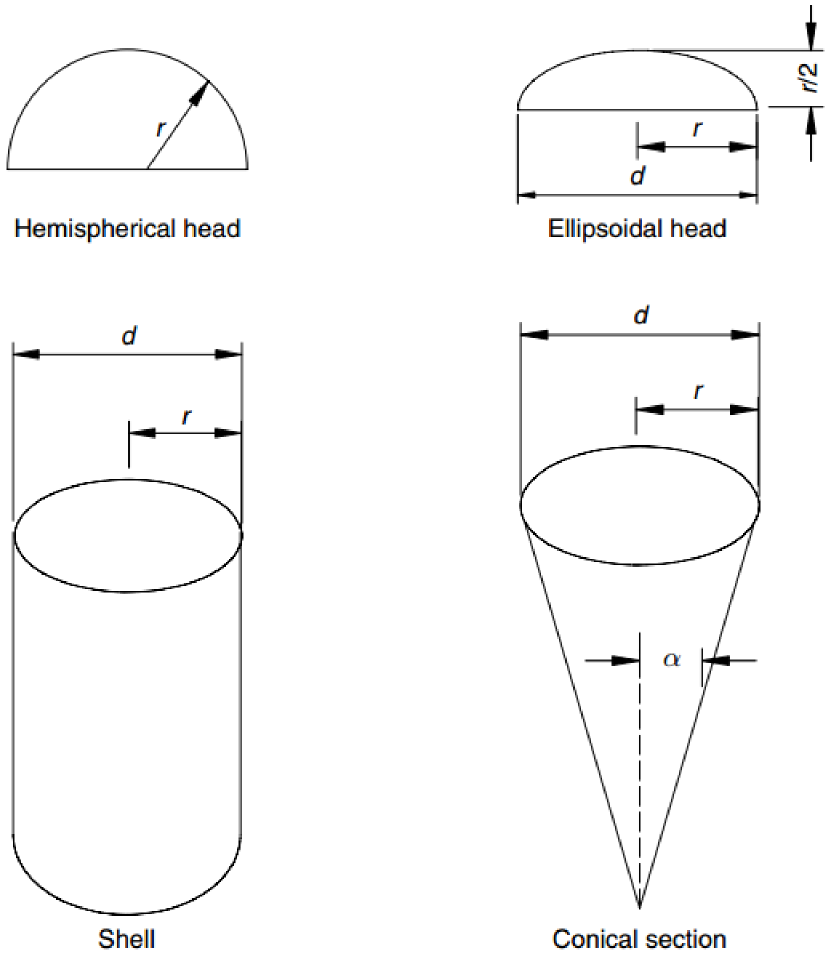


Figure 6-2. Pressure vessel shapes.

- Some operators use internal cones within vertical vessels with standard ellipsoidal heads as shown in Figure 6-3. The ellipsoidal heads contain the pressure, and thus the internal cone can be made of very thin steel.

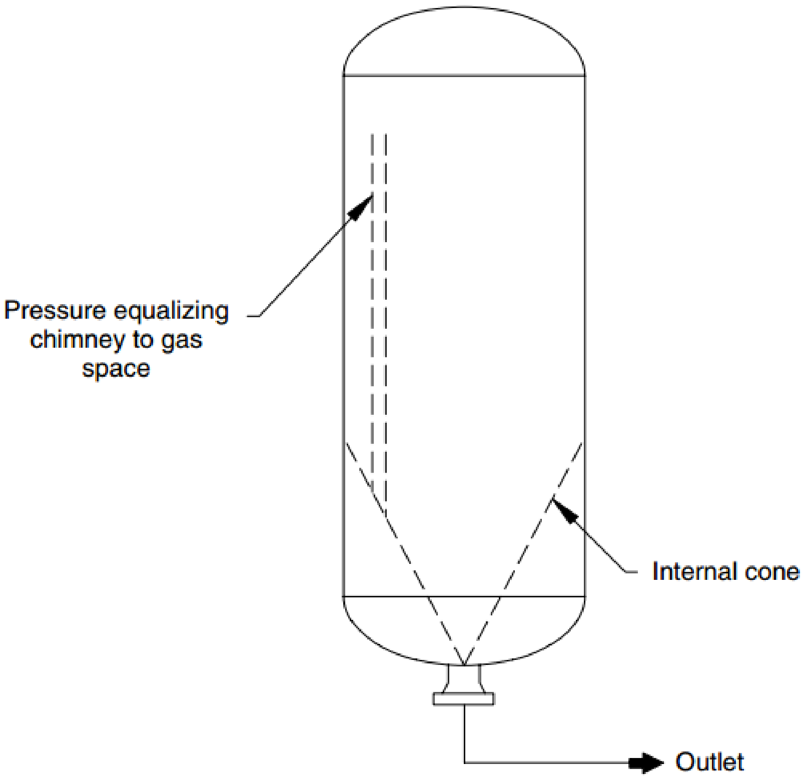


Figure 6-3. Internal cone vessel.

- Table 6-4 lists joint efficiencies that should be used in Eqs. (6-1) to (6-4).
- This is Table UW-12 in the ASME code.

Table 6-4
Maximum Allowable Joint Efficiencies for Arc and Gas Welded Joints

No.	Type of Joint Description	Limitation	(a) Fully Radiographed[1]	(b) Spot Examined	(c) Not Spot Examined[3]
1	Butt joints as attained by double welding or by other means that will obtain the same quality of deposited weld metal on the inside and outside weld surfaces of UW-35. Welds using metal backing strips that remain in the place are excluded.	None	1.00	0.85	0.70
2	Singled-welded butt joint with backing strip other than those included under (1).	(a) None except as in (b) below (b) Butt weld with one plate offset—for circumferential joints only, see UW-13(c) and Fig. UW-13.1(k)	0.90	0.80	0.65
3	Single-welded butt joint without using backing strip	Circumferential joints only, not over 5/8-in. thick and not over 24-in. outside diameter	—	—	0.60
4	Double full filet lap joint	Longitudinal joints only, not over 3/8-in. thick	—	—	0.55

5	Single full fillet lap joints with plug welds conforming to UW-17.	(a) Circumferential joints[4] for attachment of heads not over 24-in. outside diameter to shells not over 1/2 in. thick (b) Circumferential joints for the attachment to shells of jackets not over 5/8 in. in nominal thickness where the distance from the center of the plug weld to the edge of the plate is not less than $1\frac{1}{2}$ times the diameter of the hole for the plug.	—	—	0.50
6	Single full fillet lap joints without plug welds.	(a) For the attachment of heads convex to pressure to shells not over 5/8-in. required thickness, only with use of fillet weld on inside of shell; or (b) For attachment of heads having pressure on either side to shells not over $\frac{1}{4}$-in. required thickness with fillet weld on outside of head flange only.	—	—	0.45

[1]See UW-12(a) and UW-51.
[2]See UW-12(b) and UW-52.
[3]The maximum allowable joint efficiencies shown in this column are the weld joint efficiencies multiplied by 0.80 (and rounded off to the nearest 0.05) to effect the basic reduction in allowable stress required by the division for welded vessels that are not spot examined. See UW-12(c).
[4]Joints attaching hemispherical heads to shells are executed.

- Table 6-5 lists some of the common material types used to construct pressure vessels.
- Individual operating companies have their own standards, which differ from those listed in this table.

Table 6-5
Materials Typically Specified

	Low Pressure	Common Steel T>−20°F	NACE MR-01-75	Low Temp −50°F< T <0°	Low Temp FT<−50°F	High CO_2 Service
Plate	SA-36 SA-285-C	SA-516-70	SA-516-70	SA-516-70	SA-240-304	SA-240-16L
Pipe	SA-53-B	SA-106-B	SA-106B	SA-106-B	SA-333-6 TP-304	SA-312 TP-316L
Flanges and fittings	SA-105	SA-105 SA-181-1	SA-105 SA-181-1	SA-350-LF1	SA-182 F-304	SA-182 F-316L
Stud B8M bolts	SA-193-B7	SA-193-B7	SA-193-B7M	SA-320-L7	SA-193-B-8	SA-193-8M
Nuts 8MA	SA-192-2H	SA-194-2H	SA-194-2M	SA-194-4	SA-194-8A	SA-194-MA

Corrosion allowance

- Typically, a corrosion allowance of 0.125 in. for non-corrosive service and 0.250 in. for corrosive service is added to the wall thickness calculated in Eqs. (6-1) to (6-4).

Inspection Procedures

- All ASME code vessels are inspected by an approved code inspector.
- The manufacturer will supply code papers signed by the inspector. The nameplate on the vessel will be stamped to signify it has met the requirements of the code.
- One of these requirements is that the vessel was pressure tested (1998 edition, 1.5 times the MAWP; 2001 and later editions, 1.3 times the MAWP). However, this is only one of the requirements.
- The mere fact that a vessel is pressure tested 1.3 or 1.5 times the MAWP does not signify that it has met all the design and quality assurance safety aspects of the code.
- It must be pointed out that a code stamp does not necessarily mean that the vessel is fabricated in accordance with critical nozzle dimensions or internal devices as required by the process. The code inspector is only interested in those aspects that relate to the pressure handling integrity of the vessel.
- The owner must do his own inspection to assure that nozzle locations are within tolerance, vessel internals are installed as designed, coatings are applied properly, etc.

Estimating vessel weights

- It is important to be able to estimate vessel weights, since most cost estimating procedures start with the weight of the vessel.
- The vessel weight, both empty and full with water, may be necessary to adequately design a foundation or to assure that the vessel can be lifted or erected once it gets to the construction site.
- The weight of a vessel is made up of the weight of the shell, the weight of the heads, and the weight of internals, nozzles, pedestals, and skirts.

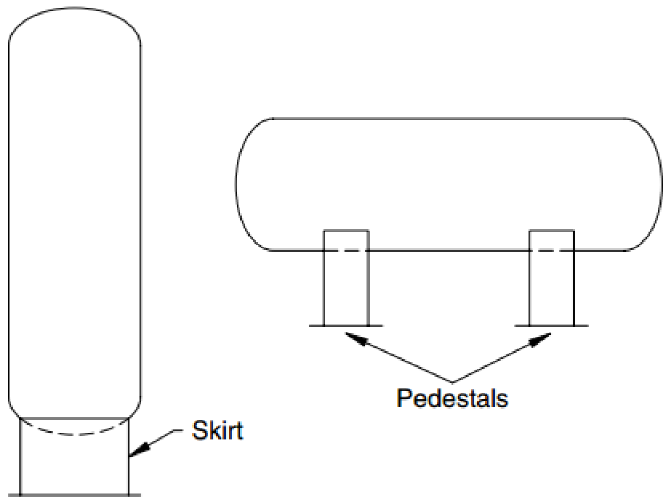


Figure 6-4. Vessel support devices.

- Shell weight can be estimated from

Field Units

$$W = 11dtL, \tag{6-5a}$$

SI Units

$$W = 0.0254dtL, \tag{6-5b}$$

where

W = weight, lb (kg),
d = internal diameter, in. (mm),
t = wall thickness, in. (mm),
L = shell length, ft (m).

- The weight of one 2:1 ellipsoidal head is approximately

Field Units

$$W \approx 0.34td^2 + 1.9td. \tag{6-6a}$$

The weight of a cone is

$$W = \frac{0.23td^2}{\sin\alpha}. \tag{6-7a}$$

SI Units

$$W \approx 9.42 \times 10^{-6} td^2 + 1.34 \times 10^{-3} td \tag{6-6b}$$

The weight of a cone is

$$W \approx 6.37 \times 10^{-6} \frac{td^2}{\sin\alpha}, \tag{6-7b}$$

where α = one-half the cone apex angle.

- The weight of nozzles and internals can be estimated at 5 to 10% of the sum of the shell and head weights.
- As a first approximation, the weight of a skirt can be estimated as the same thickness as the shell (neglecting the corrosion allowance) with a length given by Eq. (6.8) for an ellipsoidal head and Eq. (6.9) for a conical head.
- For very tall vessels the skirt will have to be checked to assure it is sufficient to support both the weight of the vessel and its appentorances and the overturning moment generated by wind forces.
- The weight of pedestals for a horizontal vessel can be estimated as 10% of the total weight of the vessel.

Field Units

$$L = \frac{0.25d}{12} + 2, \qquad (6\text{-}8a)$$

$$L = \frac{0.5d}{12 \tan \alpha} + 2. \qquad (6\text{-}9a)$$

SI Units

$$L = 2.5 \times 10^{-4} d + 0.61, \qquad (6\text{-}8b)$$

$$L = 2.54 \times 10^{-4} \frac{d}{\tan \alpha} + 0.61, \qquad (6\text{-}9b)$$

where L = skirt length in ft (m).

Pressure vessel specifications

- Some companies summarize their pressure vessel requirements on a pressure vessel design information sheet such as the one shown in Figure 6-5.

SEPARATOR DESIGN INFORMATION

1. Operating Conditions:
   A. Liquid Volumes
      1. Oil/Condensate: ________ Barrel/Day Gravity: ________ °API
      2. Water: ________ Barrels/Day Sp. Gr.: ________ (Water = 1.0)
   B. Oil/Condensate Characteristics
      1. Roaming: Nil ______ Moderate ______ Severe ______
      2. Paraffin Problem: No _____ Yes _____
      3. Slug Flow: No _____ Yes: ____ (If Yes, give details such as maximum liquid rate, slug volume, etc.) ________
   C. Gas: ________ MMscfd Sp. Gr.: ________ (Air - 1.0)
   D. Operating Temperature (°F): ______ Max ______ Min
   E. Operating Pressure ipsig): ______ Max ______ Min
   F. H.S Content: ______ Mole% CO_2. Content: ______. Mole%
   G. Geographical Location: ________

II Design Requirements:
   A. Type: ________ Vertical ______ Horizontal ______ Spherical.
      Manufacturer's Recommendation: ________
      ______ Two-Phase ______ Three-Phase
   B. Design Pressure: ________ psig at Temperature ________ °F
   C. Type Mist Extractor: (Specify) ________
   D. Corrosion Allowance: ________ (inches)
   E. Corrosion Allowance for Non-Pressure Internal Parts: ______ (inches)
   F. NACE MR-01-75 Required: No _____ Yes _____
   G. Special Stress Relieving: No _____ Yes _____
      Specify if Yes: ________
   H. API RP 14C Safety Systems Required: No _____ Yes _____

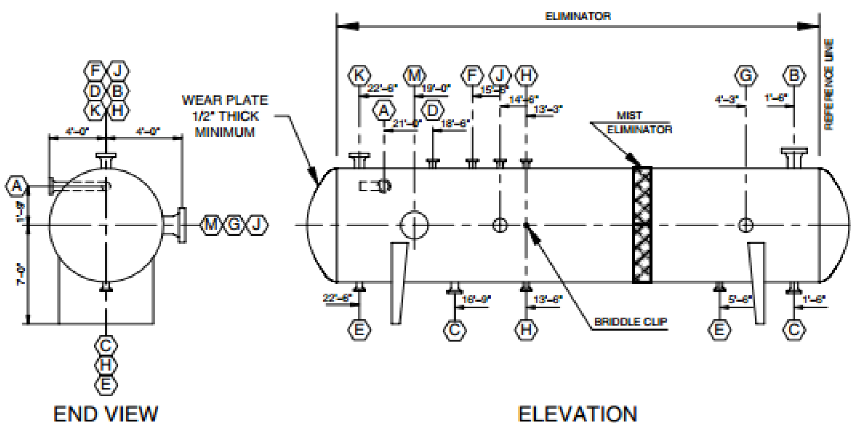
III. Coatings:
   A. External: Mfgr. Std. ______ Other ______
      Specify if Other: ________
   B. Internal: (Specify) ________
   C. Cathodic Protection: (Specify) ________

IV. Special Instructions:
   A. Radiographic Inspection: ASME Code ______ Other ______
      Specify if Other: ________
   B. Hydrostatic Test Pressure: ASME Code ______ Other ______
      Specify if Other: ________
   C. Hardness Testing Requirements: (Specify) ________
   D. Lifting Lugs: (Specify) ________
   E. Skid Mounting: (Specify) ________
   F. Welding Requirement: ASME Code ______ Other ________
      Specify if Other: ________
   G. Sand Removal System: (Specify) ________
   H. Other: ________

Figure 6-5. Example of separator design information sheet.

- Some companies have a detailed general specification for the construction of pressure vessels, which defines the overall quality of fabrication required and addresses specific items such as
  - Code compliance
  - Design conditions and materials
  - Design details
    - Vessel design and tolerances
    - Vessel connections (nozzle schedules)
    - Vessel internals
    - Ladders, cages, platforms, and stairs
    - Vessel supports and lifting lugs
    - Insulation supports
    - Shop drawings
  - Fabrication
    - General
    - Welding
    - Painting
    - Inspection and testing
    - Identification stamping
    - Drawings, final reports, and data sheets
    - Preparation for shipment

- A copy of this specification is normally attached to a bid request form, which includes a pressure vessel specification sheet such as the one shown in Figure 6-6.
- This sheet contains schematic vessel drawings and pertinent specifications and thus defines the vessel in enough detail so the manufacturer can quote a price and so the operator can be sure that all quotes represent comparable quality.



ITEM. GAS SCRUBBER ITEM NO. MBD-1020
NO. REQ'D. JOB NO.
PURCHASE ORDER NO. DATE.

DESIGN AND FABRICATION DATA

CONSTRUCTION TO BE IN ACCORDANCE WITH THE LATEST EDITION OF THE ASME CODE & ADDENDA. SECTION VIII, DIVISION 2
CODE SYMBOL. REQUIRED/NOT REQUIRED
DESIGN PRESSURE. 1800 PSIG. AT. –20/100 °F
OPERATING PRESSURE. 1000–1250 PSIG. AT. 60 °F
STRESS RELIEVE. YES/NO/PER CODE RADIOGRAPH. NP/SPOT/100%
JOINT E F F - SHELL. 1.0 HEADS. 1.0
CORROSION ALLOWANCE ALLOWANCE - SHELL 0.125" HEADS. 0.125"
MATERIAL: SHELL. HEADS. SA - 516 - 70 2:1 ELLPT
FLANGES. SA - 516 - 70N PIPE. SA - 106 - B
STUDS. NOTE 3
GASKETS. SA - 193 - B7 NUTS. SA - 194 - 2H
SADDLES. SOFT IRON TYPE R.I.D. MARK "D" CADMIUM PLATED
HINGES. YES LUGS. YES (2)
INSULATION THICKNESS. YES DAVITS REQUIRED FOR MANHOLES. NO
LADDER CLIPS. NONE INSULATION RINGS. NONE
POINT PER SPEC. NO PLATFORM CLIPS. REQUIRED

DEMISTER. NOTE 2

NOTES:

1. DESIGN, FABRICATIONS, TESTING AND DOCUMENTATION SHALL BE IN ACCORDANCE WITH PARAGON SPECIFICATION
2. THE VANE TYPE MIST ELIMINATOR SHALL BE MANUFACTURED BY ACS INDUSTRIES, INC. (OR APPROVED EQUAL) AND SHALL REMOVE 99% OF ALL DROPLETS 10 MICRONS AND LARGER
3. WELD NECK FLANGES SHALL BE ASTM SA105 INTEGRALLY REINFORCED LONG WELD NECKS ARE ACCEPTABLE

NOZZLE SCHEDULE

MK	NO	SIZE	RATING	TYPE	SERVICE	PROJ.
A	1	12"	900#	RTJ	GAS/CONDENSATE INLET	–
B	1	12"	900#	RTJ	GAS OUTLET	12"
C	2	6"	900#	RTJ	CONDENSATE OUTLET	10"
D	1	2"	900#	RTJ	RELIEF/BLOWDOWN	8'
E	2	3"	900#	RTJ	DRAIN	8'
F	1	2"	900#	RTJ	PRESSURE CONNECTION	8'
G	1	2"	900#	RTJ	TEMPERATURE CONNECTION	8'
H	2	3"	900#	RTJ	LEVEL BRIDDLE	8'
J	2	2"	900#	RTJ	LEVEL BRIDDLE	8'
K	1	8"	900#	RTJ	INSPECTION W/BLIND	10"
M	1	18"	900#	RTJ	MANWAY 18" I.D.	–

PROCESS CONDITIONS

GAS FLOW RATE 200 MMSCFD
GAS SPECIFIC GRAVITY: 0.67 (AIR = 1.0)
HYDROCARBON LIQUID FLOW RATE: 2.5 BBL/MMSCF NORMAL
HYDROCARBON LIQUID SPECIFIC GRAVITY: 0.56 @ OPERATING CONDITIONS (WATER = 1.0)
OPERATING PRESSURE: 1000 PSIG MINIMUM, 1250 MAXIMUM
OPERATING TEMPERATURE: 55°F MINIMUM, 70°F MAXIMUM

NOTES
1. INTERNAL INLET PIPING SHALL BE DESIGNED TO WITHSTAND LIQUID SLUGS ARRIVING AT VELOCITIES AS HIGH AS 45 FT/SEC.
2. VESSEL ORNADINARILY OPERATES EMPTY, BUT LIQUID LEVEL DURING SLUGGING CAN BE AS HIGH AS 42" ABOVE OUTSIDE BOTTOM OF VESSEL

NO. REVISION DATE DRAWN CHECK APP'D
ISSUED FOR CLIENT APPROVAL ENGINEER. DATE. DRAWN. DATE. CHECKED. DATE. APPROVED. DATE.
BIDDING SCALE. SHEET. OF JOB NO. CLIENT.
CONSTRUCTION CLIENT JOB NO.
PARAGON ENGINEERING SERVICES HOUSTON, TEXAS
PARAGON ENGINEERING SERVICES
MBD - 1020
LP PRODUCTION SEPARATOR
DRAWING NO. MBD - 1020 REV.

Figure 6-6. Example of pressure vessel specification sheet.

Shop drawings

- Before the vessel fabrication can proceed, the fabricator will develop complete drawings and have these drawings approved by the representative of the engineering firm and/or the operating company.
- These drawings are called shop drawings.

Nozzles

- Nozzles should be sized according to pipe sizing criteria, such as those provided in API RP 14E.
- The outlet nozzle is generally the same size as the inlet nozzle.
- To prevent baffle destruction due to impingement, the entering fluid velocity is to be limited as

Field Units

$$V_{in} \leq (3{,}500/\rho_f)^{1/2}, \tag{6-10a}$$

SI Units

$$V_{in} \leq (5{,}217.7/\rho_f)^{1/2}, \tag{6-10b}$$

where

V_{in} = maximum inlet nozzle fluid velocity, ft/s (m/s),

ρ_f = density of the entering fluid, lb/ft^3 (kg/m^3).

Vortex Breaker

- As liquid flows out of the exit nozzle, it will swirl and create a vortex.
- Vortexing would carry the gas out with the liquid. Therefore, all liquid outlet nozzles should be equipped with a vortex breaker.
- Figure 6-11 shows several vortex breaker designs. Additional designs can be found in the *Pressure Vessel Handbook*. Most designs depend on baffles around or above the outlet to prevent swirling.

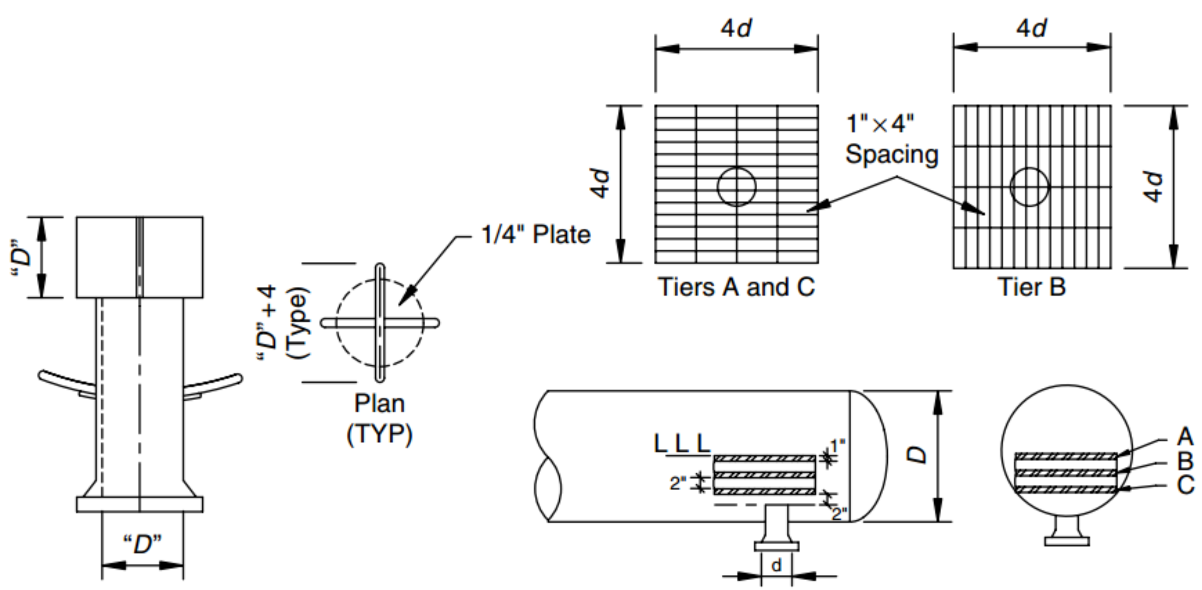


Figure 6-11. Examples of vortex breaker details.

- Manways

  : large openings that allow personnel access to the vessel internals for their maintenance and/or replacement.

- Vessel supports

  : Small vertical vessels may be supported by angle support legs, while larger vertical vessels are generally supported by a skirt support.

- Ladder and platform

  : A ladder and platform should be provided if operators are required to climb up to the top of the vessel regularly.

- Pressure relief devices

  : All pressure vessels should be equipped with one or more pressure safety valves (PSVs) to prevent overpressure. (requirement of both the ASME code and API RP 14C)

  : The PSV should be located upstream of the mist extractor

- Corrosion protection

  : Common corrosion protection methods include internal coatings with synthetic polymeric materials and galvanic (sacrificial) anodes.

  : All pressure vessels that handle corrosive fluids should be monitored periodically. Ultrasonic surveys can locate discontinuities in the metal structure, which will indicate corrosion damages.

Thank you, Question?