Offshore Equipment

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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

<table>
<thead>
<tr>
<th>Operating Pressure</th>
<th>Minimum Differential Between Operating Pressure and MAWP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Less than 50 psig</td>
<td>10 psi</td>
</tr>
<tr>
<td>51–250 psig</td>
<td>25 psi</td>
</tr>
<tr>
<td>251–500 psig</td>
<td>10% of maximum operating pressure</td>
</tr>
<tr>
<td>501–1000 psig</td>
<td>50 psi</td>
</tr>
<tr>
<td>1001 psig and higher</td>
<td>5% of maximum operating pressure</td>
</tr>
</tbody>
</table>

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**

<table>
<thead>
<tr>
<th>Class</th>
<th>MAWP, psig</th>
<th>–20°F to 100°F</th>
<th>100°F to 200°F</th>
</tr>
</thead>
<tbody>
<tr>
<td>150</td>
<td>285</td>
<td>250</td>
<td></td>
</tr>
<tr>
<td>300</td>
<td>740</td>
<td>675</td>
<td></td>
</tr>
<tr>
<td>400</td>
<td>990</td>
<td>900</td>
<td></td>
</tr>
<tr>
<td>600</td>
<td>1480</td>
<td>1350</td>
<td></td>
</tr>
<tr>
<td>900</td>
<td>2220</td>
<td>2025</td>
<td></td>
</tr>
<tr>
<td>1500</td>
<td>3705</td>
<td>3375</td>
<td></td>
</tr>
<tr>
<td>2500</td>
<td>6170</td>
<td>5625</td>
<td></td>
</tr>
</tbody>
</table>
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>
<thead>
<tr>
<th>Metal</th>
<th>Not Lower Than</th>
<th>ASME Section VIII 2007 Edition</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Div. 1</td>
</tr>
<tr>
<td>Temperature</td>
<td></td>
<td>650°F</td>
</tr>
<tr>
<td>Carbon steel plates and sheets</td>
<td>SA-516</td>
<td>15,700</td>
</tr>
<tr>
<td></td>
<td>SA-60</td>
<td>17,100</td>
</tr>
<tr>
<td></td>
<td>SA-65</td>
<td>18,600</td>
</tr>
<tr>
<td></td>
<td>SA-70</td>
<td>20,000</td>
</tr>
<tr>
<td></td>
<td>SA-285</td>
<td>12,900</td>
</tr>
<tr>
<td></td>
<td>Grade A</td>
<td>14,300</td>
</tr>
<tr>
<td></td>
<td>Grade B</td>
<td>15,700</td>
</tr>
<tr>
<td></td>
<td>Grade C</td>
<td>16,600</td>
</tr>
<tr>
<td>Low-alloy steel plates</td>
<td>SA-36</td>
<td>16,600</td>
</tr>
<tr>
<td></td>
<td>SA-387</td>
<td>15,700</td>
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<tr>
<td></td>
<td>Grade 2, cl.1</td>
<td>15,700</td>
</tr>
<tr>
<td></td>
<td>Grade 12, cl.1</td>
<td>17,100</td>
</tr>
<tr>
<td></td>
<td>Grade 11, cl.1</td>
<td>17,100</td>
</tr>
<tr>
<td></td>
<td>Grade 22, cl.1</td>
<td>17,100</td>
</tr>
<tr>
<td></td>
<td>Grade 21, cl.1</td>
<td>17,100</td>
</tr>
<tr>
<td></td>
<td>Grade 5, cl.1</td>
<td>17,100</td>
</tr>
<tr>
<td></td>
<td>Grade 2, cl.2</td>
<td>20,000</td>
</tr>
<tr>
<td></td>
<td>Grade 12, cl.2</td>
<td>18,600</td>
</tr>
<tr>
<td></td>
<td>Grade 11, cl.2</td>
<td>21,400</td>
</tr>
<tr>
<td></td>
<td>Grade 22, cl.2</td>
<td>21,400</td>
</tr>
<tr>
<td></td>
<td>Grade 21, cl.2</td>
<td>21,400</td>
</tr>
<tr>
<td></td>
<td>Grade 5, cl.2</td>
<td>21,400</td>
</tr>
<tr>
<td></td>
<td>SA-203</td>
<td>18,600</td>
</tr>
<tr>
<td></td>
<td>Grade A</td>
<td>20,000</td>
</tr>
<tr>
<td></td>
<td>Grade B</td>
<td>20,000</td>
</tr>
<tr>
<td></td>
<td>Grade D</td>
<td>18,600</td>
</tr>
<tr>
<td></td>
<td>Grade E</td>
<td>20,000</td>
</tr>
<tr>
<td>High-alloy steel plates</td>
<td>SA-240</td>
<td>20,000</td>
</tr>
<tr>
<td></td>
<td>Grade 304L</td>
<td>16,700</td>
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<tr>
<td></td>
<td>Grade 316</td>
<td>20,000</td>
</tr>
<tr>
<td></td>
<td>Grade 316L</td>
<td>16,700</td>
</tr>
</tbody>
</table>

*Austenitic Stainless steel 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 x 20,000 = 26,000 (Yield is 30,000) for 304
Determining wall thickness
• ASME code section VIII, Division 1

Wall Thickness—Cylindrical Shells
\[ t = \frac{Pr}{SE - 0.6P}, \]  \hspace{1cm} (6-1)

Wall Thickness—2:1 Ellipsoidal Heads
\[ t = \frac{Pd}{2SE - 0.2P}, \]  \hspace{1cm} (6-2)

Wall Thickness—Hemispherical Heads
\[ t = \frac{Pr}{2SE - 0.2P}, \]  \hspace{1cm} (6-3)

Wall Thickness—Cones
\[ t = \frac{Pd}{2\cos \alpha(SE - 0.6P)}, \]  \hspace{1cm} (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),
\( \alpha \) = 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).

<table>
<thead>
<tr>
<th>NOTATION FOR VESSELS UNDER INTERNAL PRESSURE</th>
<th>FORMULAS FOR VESSELS UNDER INTERNAL PRESSURE</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$ = Half Apex Angle of Cone, Deg.</td>
<td>1. THE FORMULAS CONFORM TO THE ASME CODE FOR</td>
</tr>
<tr>
<td>$L_0$ = Outside Crown radius, inches</td>
<td>PRESSURE VESSELS, SECTION VIII, DIVISION 1.</td>
</tr>
<tr>
<td>$D$ = Inside diameter, inches</td>
<td>2. CORROSION ALLOWANCE. If the vessel is subject</td>
</tr>
<tr>
<td>$D_0$ = Outside diameter, inches</td>
<td>to thinning by corrosion allowance shall be</td>
</tr>
<tr>
<td>$P$ = Design pressure or maximum</td>
<td>added to the thickness calculated by</td>
</tr>
<tr>
<td>$E$ = Efficiency of welded joints</td>
<td>the formulas, Code UG-35.</td>
</tr>
</tbody>
</table>

In Terms INSIDE Radius or Diameter

$$t = \frac{P}{Se-0.6P} = \frac{2Se-0.2P}{2Se-0.8P}$$

Cylindrical Shell

In Terms OUTSIDE Radius or Diameter

$$t = \frac{P_{D_0}}{Se-0.4P} = \frac{2Se-0.8P}{2Se-0.4P}$$

Cylindrical Shell

Hydrostatic Test Pressure - a one and one-half times the maximum allowable working pressure of the design pressure when calculations are not made to determine the maximum allowable working pressure, Code UG-38.

Degree of Radiographic Examination. Condition where ultrasonics are mandatory is described in the Code UV-11.

Stress Values of Materials, 6,000 psi. Exempt from Code, Table UV-49.

<table>
<thead>
<tr>
<th>Specification</th>
<th>For Metal Temperature Not Exceeding Deg. F.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number Grade</td>
<td>20 to 900</td>
</tr>
<tr>
<td>100</td>
<td>700</td>
</tr>
<tr>
<td>550</td>
<td>800</td>
</tr>
<tr>
<td>650</td>
<td>900</td>
</tr>
<tr>
<td>701</td>
<td>1000</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Grade</th>
<th>40</th>
<th>50</th>
<th>60</th>
<th>70</th>
<th>80</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.00</td>
<td>1.05</td>
<td>1.10</td>
<td>1.15</td>
<td>1.20</td>
<td></td>
</tr>
<tr>
<td>0.85</td>
<td>0.90</td>
<td>0.95</td>
<td>1.00</td>
<td>1.05</td>
<td></td>
</tr>
</tbody>
</table>

Efficiency (E) to be Used in Calculations of Seamless heads, ASME Code UG-12.20

<table>
<thead>
<tr>
<th>TYPE OF HEAD</th>
<th>DECORATION OF EXAMINATION OF HEAD TO SHELL JOINT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hemispherical</td>
<td>Full</td>
</tr>
<tr>
<td>Other</td>
<td>Full</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>DECORATION OF EXAMINATION OF HEAD TO SHELL JOINT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full</td>
</tr>
<tr>
<td>No. 1</td>
</tr>
<tr>
<td>No. 2</td>
</tr>
<tr>
<td>Others</td>
</tr>
</tbody>
</table>

1.00 1.05
0.85 0.90
1.00 1.05

14. NOTATION

- $D = \text{inside diameter of cone at the large end, inches}$
- $D_o = \text{outside diameter of ellipsoidal head, inches}$
- $D_{o1} = \text{outside diameter of ellipsoidal head, inches}$
- $E = \text{Leakage area of joint in the shell or head; for helical heads, this includes head to shell joint; for flange heads, this includes head to flange joint}$
- $L = \text{outside crown radius of flanged and dished head, inches}$
- $L = \text{inside crown radius of flanged and dished head, inches}$
- $L = \text{inside design pressure or maximum allowable working pressure}$
- $P = \text{pressure pounds per square inches}$
- $r = \text{Knuckle radius}$
- $R = \text{outside radius of shell or hemispherical head, inches}$
- $R_{o1} = \text{outside radius of shell or hemispherical head, inches}$
- $S = \text{Allowable stress value of material pounds per square inch, inches}$
- $t = \text{thickness of shell, inches}$
- 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 90° apex (α=45°) or a 60° apex (α=30°). These are referred to respectively as a “45°” or “60°” 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.

![Diagram of internal cone vessel](image-url)
• 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>
<thead>
<tr>
<th>No.</th>
<th>Type of Joint Description</th>
<th>Limitation</th>
<th>(a) Fully Radiographed</th>
<th>(b) Spot Examined</th>
<th>(c) Not Spot Examined</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>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.</td>
<td>None</td>
<td>1.00</td>
<td>0.85</td>
<td>0.70</td>
</tr>
<tr>
<td>2</td>
<td>Single-welded butt joint with backing strip other than those included under (1).</td>
<td>(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)</td>
<td>0.90</td>
<td>0.80</td>
<td>0.65</td>
</tr>
<tr>
<td>3</td>
<td>Single-welded butt joint without using backing strip</td>
<td>Circumferential joints only, not over 5/8-in. thick and not over 24-in. outside diameter</td>
<td>—</td>
<td>—</td>
<td>0.60</td>
</tr>
<tr>
<td>4</td>
<td>Double full fillet lap joint</td>
<td>Longitudinal joints only, not over 3/8-in. thick</td>
<td>—</td>
<td>—</td>
<td>0.55</td>
</tr>
</tbody>
</table>
|   | Single full fillet lap joints with plug welds conforming to UW-17. | 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. | 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 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>
<thead>
<tr>
<th>Table 6-5</th>
<th>Materials Typically Specified</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Low Pressure</td>
</tr>
<tr>
<td></td>
<td>Common Steel</td>
</tr>
<tr>
<td></td>
<td>T &gt; -20°F</td>
</tr>
<tr>
<td>Plate</td>
<td>SA-36</td>
</tr>
<tr>
<td></td>
<td>SA-285-C</td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td>Flanges and fittings</td>
<td>SA-105</td>
</tr>
<tr>
<td></td>
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<td></td>
</tr>
</tbody>
</table>
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.](image-url)
• Shell weight can be estimated from

Field Units

\[ W = 11dtL, \]  
\[ (6-5a) \]

SI Units

\[ W = 0.0254dtL, \]  
\[ (6-5b) \]

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**

\[ W \approx 0.34td^2 + 1.9td. \]  \hspace{1cm} (6-6a)

The weight of a cone is

\[ W = \frac{0.23td^2}{\sin \alpha}. \]  \hspace{1cm} (6-7a)

**SI Units**

\[ W \approx 9.42 \times 10^{-6}td^2 + 1.34 \times 10^{-3}td \]  \hspace{1cm} (6-6b)

The weight of a cone is

\[ W \approx 6.37 \times 10^{-6} \frac{td^2}{\sin \alpha}, \]  \hspace{1cm} (6-7b)

where \( \alpha = \) 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.

\[ L = \frac{0.25d}{12} + 2, \quad (6-8a) \]

\[ L = \frac{0.5d}{12 \tan \alpha} + 2. \quad (6-9a) \]

\[ L = 2.5 \times 10^{-4} d + 0.61, \quad (6-8b) \]

\[ L = 2.54 \times 10^{-4} \frac{d}{\tan \alpha} + 0.61, \quad (6-9b) \]

where \( L \) = skirt length in ft (m).
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: _______ Barrel/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 (psig): _______ Max  Min
   - F. H₂S Content: _______ Moi%  CO₂ Content: _______ Moi%
   - G. Geographical Location:

2. **Design Requirements:**
   - A. Type: Vertical  Horizontal  Spherical
   - 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

3. **API RP 14C Safety Systems Required:**
   - No  Yes

4. **Coatings:**
   - A. External: Mfr. Std: _______ Other
     Specify if Other:
   - B. Internal: (Specify)
     C. Cathodic Protection: (Specify)

5. **Special Instructions:**
   - A. Radiographic Inspection: ASME Code: _______ Other
     Specify if Other:

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**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.

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

\[
V_{in} \leq (3.500/\rho_f)^{1/2}, \quad (6-10a)
\]

**Field Units**

\[
V_{in} \leq (5.217.7/\rho_f)^{1/2}, \quad (6-10b)
\]

**SI Units**

where

- \( V_{in} \) = maximum inlet nozzle fluid velocity, ft/s (m/s),
- \( \rho_f \) = 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.
• 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?