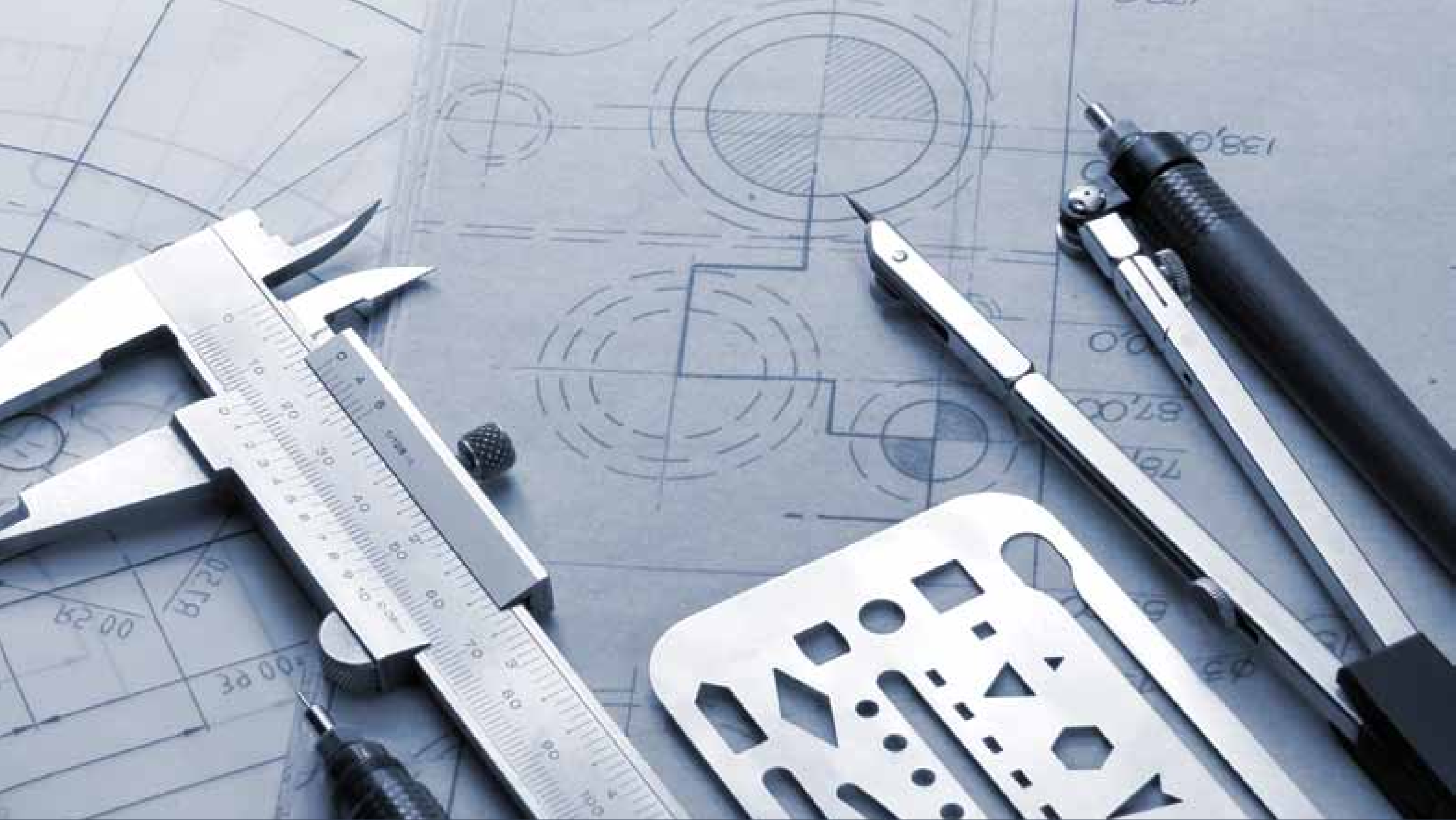

Avoid Costly **Design Mistakes**

Common errors keep plants from getting the most reliable and suitable vessels

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"JUST BUILD it to the Code." That's the most common response you hear during a design review that involves purchasing a new pressure vessel. Yet, there're as many choices within the ASME code as there are when searching for your next vehicle. Smart choices can save you money during fabrication as well as over the lifecycle of the vessel. So, here, we'll attempt to condense the 5,000+ pages (50+ lbs) of the ASME Boiler and Pressure Vessel Code, or "the Code," as it's affectionately known, into a simple guide when specifying vessels, heat exchangers and tanks. We'll focus on 10 key factors.

1. *Inside diameter versus outside diameter.* Process engineers often specify a vessel's diameter based on inside diameter (ID) to ease volumetric calculations. This also will simplify fabrication/installation of internal hardware (e.g., support rings, trays, distributors, etc.). However, sometimes specifying a vessel based on outside diameter (OD) is better. For instance, after a purchase order is issued, heads are the first things ordered — obtaining off-the-shelf heads is more likely if specified by nominal pipe sizes (NPS), which is OD from diameters 14 in. to 36 in. (For thin-wall heads, i.e., 2-in. thick or less, choosing ID or OD makes little difference, while most heads more than 36 in. are custom made.) As heads get thicker, hot forming is necessary and dies are based on ID. Thick, hot formed heads can be OD ordered but require an extra manufacturing step.
2. *Design pressure and temperature.* Required wall thickness is more sensitive to pressure than temperature. Therefore, specifying a design pressure 100 psig over the maximum operating pressure is more costly than specifying a design temperature 100°F higher than what's needed. A design pressure of 25 psig–50 psig above that at the maximum operating/upset condition and not less than 90% of maximum allowable working pressure is industry practice. Keep design temperature to no more than 50°F–100°F above that at maximum operating/upset conditions. Also, watch your design pressure and temperature so as not to cross into the next higher flange class. Check ASME B16.5 for design temperature and limitations for flanges.
3. *Vacuum rating.* Although current project needs may not require a vessel to be vacuum rated, over its life-

time, changes in feedstock, products and technology will occur. A large number of re-rates now performed are on older vessels originally not documented for vacuum that now require it due to process changes. Many new vessels will rate for full vacuum and all for partial vacuum. So have the fabricator evaluate your proposed design for vacuum and apply it to the code stamp. With today's software, this calculation can be easily performed and at no cost. You may get full vacuum rating without any modifications — if not, consider spending a little extra now by welding on a stiffening ring and a couple of re-pads to avoid having to go through the cumbersome re-rate process and field hydro-test down the road. (See www.ChemicalProcessing.com/voices/plant_insites.html.)

4. *Head choices.* Functionality, not cost, should determine head choice; so understanding the functional differences is crucial. Dished heads for ASME vessels typically are available in three styles; elliptical (2:1), flanged and dished (F&D), and hemispherical (hemi-heads). Under 600 psig, elliptical heads are the most common and least expensive in terms of wall thickness and forming costs. Above 600 psig, hemi-heads are economically attractive due to their inherent low-stress shape; below 600 psig, they are the most expensive choice because they are constructed of welded, segmental parts not a single piece. F&D (torispherical) heads have the lowest profile (height/diameter ratio) and compete well with elliptical heads under 100 psig, although they have half the volume. The low profile of the F&D head only is advantageous when top head accessibility is required for maintaining instruments, agitator, etc., or when space is limited below or, for horizontal vessels, to the sides. For vessels 24 in. or less, off-the-shelf pipe caps (elliptical) provide the most economical design.

   Flat heads have very limited use for pressure vessels more than 24 in. in diameter. Because of their flat geometry, they offer far less resistance to pressure than elliptical and F&D heads of the same thickness. Engineers occasionally will specify a flat head, but this practice is uneconomical for pressures above 15 psig–25 psig. If a large diameter flat head is necessary for code equipment, then stiffening the head with structural I-beams is possible but requires sophisticated finite elemental analysis, a skill that not all fabricators possess.

5. *Jacket choices.* Consider functionality, not cost. Choosing the correct jacket is paramount to achieve process needs. The three common types — conventional, half-pipe and dimple — offer advantages and disadvantages with respect to process parameters, reliability and cost [1] (Table 1).

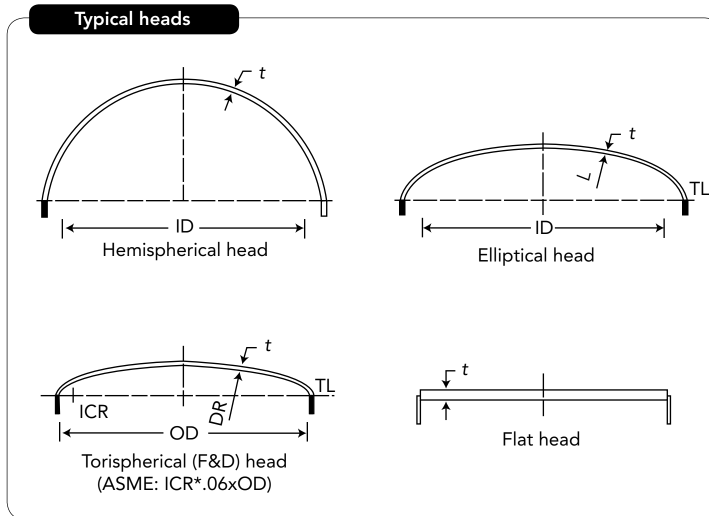


Figure 1. Curved heads predominate and avoid the pressure limitations of flat heads.

6. *Cones.* Conical sections (cones) are needed where there's a change in diameter or as a bottom head, e.g., for a bin or hopper. The rule here is keep the transition angle (referred to as the half apex angle) to 30° or less unless process conditions govern, as exceeding 30° adds costs. The ASME Code demands the piece have a rolled knuckle at both ends when the transition is greater than 30°; bending stresses complicate the calculation, putting it beyond the skill of many fabricators.

7. *Nozzles loads and projections.* The ASME Code [2] requires consideration of all loads. Designers routinely perform wind and seismic calculations but too often overlook nozzle loads due to thermal pipe stress — these can cause visible damage. If attached piping operates at more than 200°F we suggest providing the fabricator with the nozzle loads in Table 2 for a reasonable nozzle stiffening. By providing a

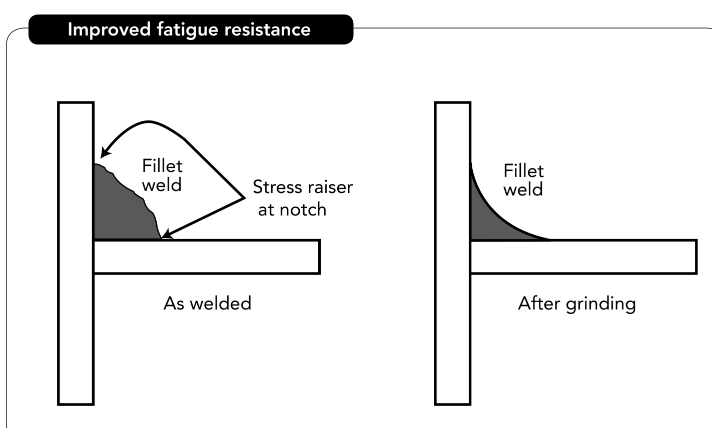


Figure 2. Blend grinding of fillet weld can prevent fatigue but the throat of the weld (the distance from face to root) must meet code requirements after grinding.

HOW TYPES OF JACKETING COMPARE		
TYPE OF JACKETING	**ADVANTAGES**	**DISADVANTAGES**
Conventional (full enclosure)	High flowrates Low pressure drop More coverage than other jackets	Poor heat transfer coefficients and heat transfer due to low velocities Thick wall required to withstand jacket pressure* Bypassing and dead zones — resulting in poor heat transfer Highest cost
Half-pipe	High flowrates Low pressure drop Suitable for dirty fluids and no bypassing High jacket pressure with no adverse effect on inner vesselwall thickness Good fatigue resistance (cyclic service) if applied with full penetration welds	Difficult to fabricate around nozzles Incomplete coverage unless spacing is kept to 1 in. or less More expensive than dimple jacket
Dimple	Least expensive Ability to withstand high pressures Easy to work around nozzle	Limited to steam and low flow liquids only performing maintenance heat transfer Least resistant to fatigue failures (cyclic service) Susceptible to plugging — requires clean fluid Difficult to estimate flow/pressure drop without vendor supplied empirical procedures Bypassing and dead zones — resulting in poor heat transfer
*Note: a spiral baffle in the jacket space, welded per Code as a stiffener to the inner wall, will reduce required inner wall thickness and improve heat transfer due to increased fluid velocity.		

Table 1. Each type of jacketing has a particular combination of pluses and minuses.

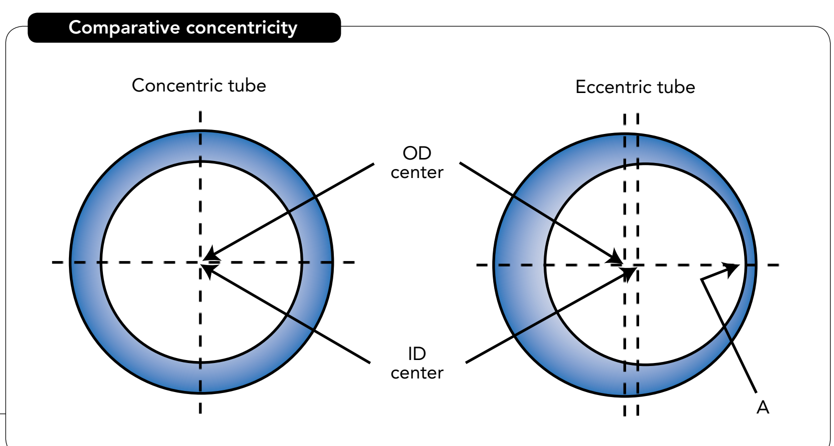


Figure 3. Welded tubing (left) is inherently more concentric than seamless tubing (right). Eccentricity exaggerated for illustration purposes; "A" is the governing dimension, minimum wall thickness.

reasonable nozzle load, the vessel fabrication and piping design can proceed in parallel and avoid pipe stress/nozzle loading issues months into fabrication.

Also, nozzle projections below the support ring or lugs shouldn't stick out further than the support bolt circle or structural steel will have to be removed when setting the vessel. This is ill-advised for heavy equipment.

8. *Rectangular tanks.* It's not cost effective to specify a rectangular vessel for pressure other than static head; therefore, only consider this configuration for atmospheric tanks. Flat surfaces are highly stressed under pressure (and vacuum) and the required thickness without adding stiffeners can be mind-boggling. An engineer needing a rectangular tank often incorrectly specifies API 650 or ASME. Neither API 650 nor any other API standard exists for rectangular tanks. Appendix 13 of the ASME Pressure Vessel Code provides a methodology but will lead to an expensive over-design. Most fabricators will apply the stress/strain formulas in Roark [3] to design a safe and economical tank that can operate under 15 psig.

9. *Cyclic service.* If a vessel will experience an unusual number of thermal or pressure cycles over its design life, this could result in premature fatigue failure (usually at a weld) unless preemptive measures are taken. Fatigue is cumulative material damage that manifests as a small crack and progressively worsens (sometimes to failure) as the material is repeatedly cycled. A 1985 survey showed that fatigue was the second most prevalent cause of failure in industry (25%), closely behind corrosion (29%) [4].

It's up to the purchaser to instruct the fabricator what design/fabrication practices to follow to avoid fatigue. Cyclic service is usually associated with batch processes and ASME [5] provides the following rules:

Design for fatigue if $N_1 + N_2 + N_3 + N_4 \geq 400$ for non-integral (fillet weld) construction and $\geq 1{,}000$ for integral construction (i.e., no load-bearing fillet welds), or 60 and 350, respectively, in the knuckle region of formed heads, where N_1 is the number of full startup/shutdown cycles; N_2 is the number of cycles where pressure swings 15% (non-integral) or 20% (integral); N_3 is the number of thermal cycles with a temperature differential (ΔT) exceeding 50°F between two adjacent points no more than $2.5\,(Rt)^{1/2}$ apart (where R is inside radius of vessel and t is thickness of the vessel being considered) — apply a two-times factor if ΔT exceeds 100°F, a four times factor if more than 150°F, and see Div. 2 for more than 250°F; and N_4 is the number of thermal cycles for welds attaching dissimilar materials in which $(\alpha_1 - \alpha_2)\Delta T$ (where α is the thermal expansion coefficient) exceeds 0.00034, or for carbon steel welded to stainless steel, the number of cycles where $2\Delta T$ exceeds 340.

Equipment and piping in continuous processes also can experience fatigue due to the relentless mechanical loading/unloading of reciprocating compressors, piston pumps, bin vibrators or from vibration, etc., from any type of mis-aligned rotating equipment.

Fatigue failures in welded equipment most commonly occur in fillet welds where there's an abrupt change in equipment geometry. Division 2 of the ASME Code designs around fatigue cracking in nozzles by limiting the use of fillet welds. However, fillet welds and sharp corners are ubiquitous in Div. 1 designs and can't be avoided without cost.

Crack initiation usually begins at the surface due to small microcracks. Therefore, surface smoothness is a good defense. Polished surfaces have four times the fatigue resistance [6] but polishing generally can't be justified

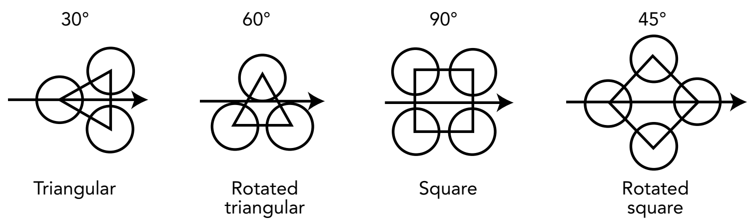
VESSEL AND HEAT EXCHANGER NOZZLE LOADS

FORCE	LOAD*
Lateral force in any direction	F × 450D lbs
Bending or torsional moment	F × 110 D^2 ft-lbs

*Note: D is pipe diameter in inches and F is flange rating factor: for 150-lb flange rating, F = 0.6; for 300, 0.7; for 600, 0.8; for 900, 1.0; for 1,500, 1.1; and for 2,500, 1.2. For API-specified equipment, refer to the respective standard for nozzle loads.

Table 2. Use these loads if attached piping operates at ≥200°F and formal stress analysis hasn't been done. Contact OEM for glass lined equipment.

Tube patterns



Note: Flow arrows are perpendicular to the baffle cut edge.

Figure 4. More tubes can fit with 30° and 60° configurations but mechanical cleaning may be harder.

for fatigue life alone. Shot peening imparts compressive stresses into the metal surface that impede crack initiation but, again, only high-end applications can economically justify peening. For mid- to small-size process vessels, good weld quality often is the most economical defense against fatigue; so, state requirements in the equipment specifications. Because fatigue cracks often initiate at the toe or root of fillet welds, grinding the face to gently blend the weld into the base metal with a generous radius remarkably reduces stress risers (Figure 2). Another method to reduce stress risers is to TIG (tungsten-inert-gas) wash a weld toe to improve smoothness and remove microcracks. Initially target welds where cyclic loading is occurring.

Experience has shown that most fatigue problems occur due to inadequately supported attachments or where saddles/supports lacked wear pads or rounded corners.

10. *Tubing.* This can be a significant cost element when ordering large heat exchangers. Cost can vary appreciably depending on the fabrication requirements specified. It's not our intent to steer you away from the highest quality tube but merely to point out subtleties that can noticeably affect price.

• *Diameter.* Tubing is specified based on OD. For quickest delivery, stick to commonly stocked sizes, typically ¾-in. and 1-in. tubes for the chemical industry. Specifying smaller tubes (e.g., ½ in.) will increase the exchanger's tube count and cost; this will improve duty but will cause higher pressure drop and may make mechanical cleaning more difficult. Therefore, only consider tubes smaller than ¾ in. for cleaner services or when increasing the shell diameter/length isn't an option. Larger tubes (>1 in.) have the opposite effect but may be necessary to satisfy process conditions. Another option to increase effective surface area without changing tube diameter is to specify finned tubes or twisted tubes — but those are limited to clean applications.

• *Length.* Tubes are stocked in 20-ft lengths. Seamless tubes are made from individual billets or hollows and so can vary in length by one to two feet. The length of welded tubes is more exact because they're produced from a continuous strip coil. The most wasteful and costly option for stocked tubes is ordering units just over 10 ft in length because nearly 50% of the tube is discarded. As tube count increases, direct mill orders become economically attractive; in such cases, any length tube can be supplied, if your schedule allows. Mills have minimum orders (i.e., 2,000 lb.–2,500 lb.), though "mini-mills" will take orders at half these quantities.

• *Gauge.* Tubes come in different wall thicknesses (or gauge). Industry standards [7] detail the appropriate wall thickness based on material type and service. Table 3 provides guidance for a ¾-in. tube where no prior service history is available.

• *Corrosion allowance.* This typically isn't added because tubes are considered a replaceable feature of the exchanger. If designing for a corrosive service, specifying the next-heavier-gauge wall thickness or choosing a higher alloyed tube material is more appropriate.

• *Seamless versus welded tube.* There's a perception that seamless tubes are more reliable than welded tubes. This is currently less valid as some manufacturers have developed specialized techniques for making welded tubing that give products that show no preferential weld corrosion and have properties equal to those of seamless tubing [8,9]. Seamless tubing will cost more and usually has longer delivery. Welded tubing requires a greater amount of non-destructive examination (NDE), but this typically only adds pennies per foot of tubing if done at the mill [8,9].

Eccentricity is inherent in producing seamless tubes [10]. They typically are made by piercing, extrusion or pilgering. The inner mandrel/die can't stay perfectly centered during the tube forming process. Welded tubes on the other hand begin with strip material that is very consistent in wall thickness. So, welded tubes tend to be more concentric (Figure 3). Seamless tube standards permit larger wall-thickness tolerances than those allowed by welded tube standards [11].

• *Minimum versus average wall*

TUBE MATERIAL SELECTION

MATERIAL	SERVICE	GAUGE	WALL THICKNESS, IN.
Carbon steel	Harsh	12	0.109*
	Mild	14	0.083*
Aluminum	Mild	14	0.083*
Stainless steel and high alloy	Harsh	14	0.083†
	Mild	16	0.065†
Copper	Mild	16	0.065*
Titanium and zirconium	Harsh	18	0.049†
	Mild	20	0.035†
Niobium and tantalum	Harsh	N/A	0.020†
	Mild	N/A	0.015†
* minimum			
† average			

Table 3. When using ¾-in. tube and not having a history of materials in the particular service, follow these guidelines.

thickness. Minimum wall tubes cost a bit more than average wall tubing. When it's unnecessary to use minimum wall tubing, such as for high pressure or corrosive service where metal loss is anticipated, it may be more economical to permit the use of average wall welded tubing and specify additional NDE or corrosion evaluation of the tube seam.

• *Tube pattern*. Shell and tube heat exchangers typically are fabricated with one of four types of tube patterns — 30°, 60°, 45° and 90° (Figure 4). Duty, pressure drop, cleanability, cost and vibration all depend on which pattern is chosen. Consider process needs, not cost, when making the selection.

A 30° or 60° pattern is laid out in a triangle configuration. The main benefit is that about 10% more tubes can fit in the same area as a 45° or 90° pattern. There's very little difference between the 30° and 60° patterns. Often a thermal designer will run analyses of both and select the one that provides the best pressure drop and vibration results. The disadvantage of a 30° or 60° pattern is that it's difficult to mechanically clean on the shell side. Therefore, such a pattern is chosen for cleaner services; frequently the bundle isn't removable.

The 45° or 90° pattern is selected if shell-side mechanical cleaning is required. Such a pattern also requires a removable bundle. The 45° is more common than the 90° because it provides more shell-side flow disturbance, which improves heat transfer. A 90° pattern is used to reduce pressure drop at the expense of duty and often is employed in boiling service to enable better vapor disengagement.

MAKE THE RIGHT CHOICES

In today's chemical industry, too many engineers given the task of specifying welded equipment such as vessels, heat exchangers and tanks aren't well versed in what's necessary to develop an economic design that provides suitable safety and performance. Myriad choices must be made — and each will incrementally add to the final cost and schedule.

PREVIOUS ARTICLES

This article is the final part of a series on pointers for welded equipment (e.g., pressure vessels, heat exchangers and tanks). The earlier articles are:

"Avoid Costly Materials Mistakes,"
www.ChemicalProcessing.com/articles/2008/003.html

"Avoid Costly Fabrication Mistakes,"
www.ChemicalProcessing.com/articles/2008/065.html

When looking for savings, cutting the wrong corners may turn out to be very costly over the equipment's service life.

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REFERENCES

1. "Vessel Design Manual," ITT Industries, White Plains, N.Y. (2000).
2. "Loadings," Paragraph UG-22, ASME Code, Sect. VIII, Div. 1, ASME, New York City (2007).
3. Young, W.C. and R. Budynas, "Roark's Formulas for Stress and Strain," 7th ed., McGraw-Hill, New York City (2002).
4. Brooks, C.R. and A. Choudhury, "Metallurgical Failure Analysis," McGraw-Hill, New York City (1993).
5. "Fatigue Screening Criteria," Table 5.9, ASME Code, Sect. VIII, Div. 2, ASME, New York City (2007).
6. Deutschman, A.D., W.J. Michels and C.E. Wilson, "Machine Design: Theory and Practice," Macmillian, New York City (1975).
7. "Shell and Tube Heat Exchanger Specification," PIP VESST001, Process Industry Practices, Austin, Texas (2007).
8. James, M.M. and D. O'Donnell, "When to Specify Welded, Welded and Drawn, or Seamless Tubing," *Welding Journal* (June 2008). Available online at www.rathgibson.com/downloads/listing.aspx?did=59.
9. James, M.M., "Development of High Quality Welded Heat Exchanger Tubing in Lieu of Seamless," presented at ATI Corrosion Solutions Conference, Sunriver, Ore. (2007).
10. Kearns, W.H., "Minimizing Wall Thickness Variation in Seamless Tubing," *The Fabricator*, (Aug. 28, 2003), www.TheFabricator.com/TubePipeProduction/TubePipeProduction_Article.cfm?ID=678
11. "Specifications for General Requirements for Carbon, Ferrite Alloy, and Austenitic Alloy Steel Tubes," ASME/ASTM SA/A-450, ASME, New York City (2004).