6
Special Designs

Procedure 6-1: Design of Large-Diameter Nozzle Openings [1]

There are three methods for calculating the strength of reinforcement required for openings in pressure vessels:

1. Area replacement rules per UG-36(b).
2. Analysis per Appendix 1-7.
   a. 2/3 area replacement rule.
   b. Membrane-bending stress analysis.
3. FEA.

The Code defines when and where these methods apply. Reinforcement for large-diameter openings has been in the Code for a long time. The previous rule was simply to move the majority of the area replacement closer to the nozzle neck, also called the *2/3 rule*. Unfortunately, there were a few cases of flange leakage where the flange was located close to the shell. It was discovered that as the opening opened up, the flange was distorted. It was actually bending. In addition, the 2/3 rule did not allow for an accurate way to determine MAWP for the vessel without proof testing.

This issue was addressed in 1979 by McBride and Jacobs. The principle was to calculate stresses in two distinct areas, membrane and bending. Membrane stresses are based on pressure area times metal area. Bending is based on AISC beam formulas. The neck-and-shell section (and sometimes the flange as well) is assumed as bent on the hard axis. This is not a beam-on-elastic-foundation calculation. It is more of a brute-force approach.

This procedure was eventually adopted by the Code and incorporated. Unfortunately, it turned out that the procedure, while good for most cases, was not good for all. Yet it was still superior to what we used before this paper was published. The ASME has now revised the applicability of the procedure to the cases where it has been deemed safe.

Large openings calculated by this procedure are limited to openings less than 70% of the vessel diameter. There are four cases that can be solved for, depending on your nozzle geometry.

Reinforcement for Large-Diameter Openings

Per ASME, Section VIII, Appendix l-7(b)l(b), the rules for "radial nozzles," not oblique or tangential, must meet strength requirements in addition to area replacement rules. The following lists the parameters for which these additional calculations shall be performed:

a. Exceed the limits of UG-36(b).
b. Vessel diameter > 60 in.
c. Nozzle diameter > 40 in.
d. Nozzle diameter > $3.4\sqrt{Rt}$.
e. The ratio $R_n/R < 0.7$ (that is, the nozzle does not exceed 70% of the vessel diameter).

Table 6-1 shows the ratio of vessel diameter, D, and shell thickness, t, where the values of $3.4\sqrt{Rt}$ are greater than 40. The heavy line indicates the limits for which 40 is exceeded. For nozzles that exceed these parameters, a finite element analysis (FEA) should be performed.

Table 6-1
Parameters for large-diameter nozzles

t \ D	60	72	84	96	108	120	132	144	156	168	180
1.00											
1.25											
1.5										38.16	39.5
1.75								38.2	39.7	41.2	42.6
2.00							39.1	40.8	42.5		
2.25						39.5	41.4				
2.50					39.5	41.6					
2.75				39.1	41.4						
3.00				40.8							
3.25			39.72	42.5							
3.50			41.22								
3.75		39.5									
4.00	37.2	40.8									
4.25	38.4										
4.50	39.5										
4.75	40.5										

Use $^2/_3$ rule area replacement when $3.4\sqrt{Rt} < 40$

Use membrane-bending analysis when $3.4\sqrt{Rt} > 40$

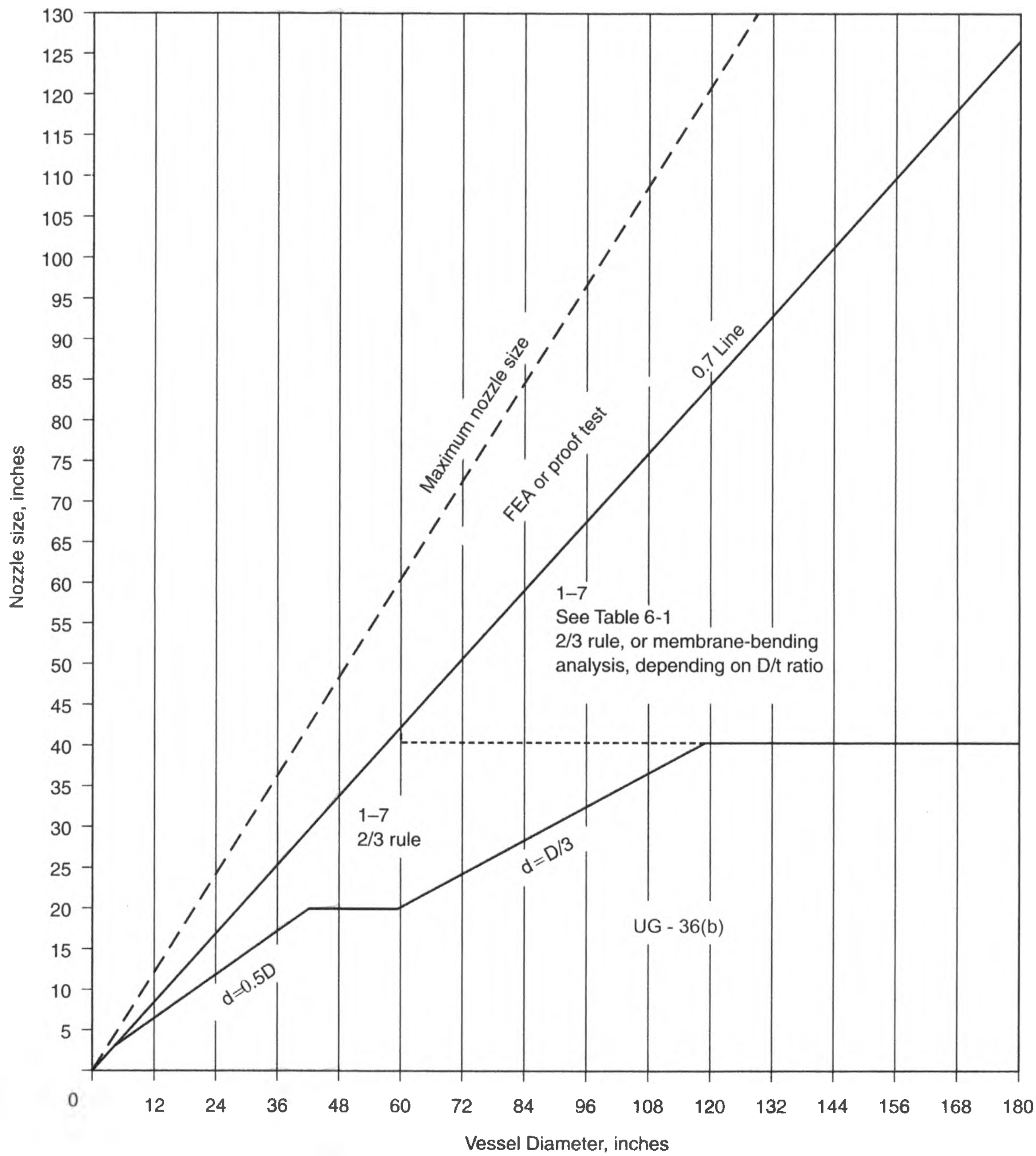


Figure 6-1. Guideline of nozzle reinforcement rules.

Large Openings—Membrane and Bending Analysis

Notation

A_s = area of steel, in.^{2}
A_p = area of pressure, in.^{2}
P = internal pressure, psi (design or test)
r_m = mean radius of nozzle, in.
R_m = mean radius of shell, in.
T = thickness of shell, in.
t = thickness of nozzle, in.
F_y = minimum specified yield strength, ksi
σ = maximum combined stress, psi
σ_b = bending stress, psi
σ_m = membrane stress, psi
I = moment of inertia, in.^{4}
M = bending moment, in.-lb

Procedure

Step 1: Compute boundary limits for bending along shell and nozzle in accordance with Note 3. Limit will be governed by whether material of construction has a yield strength, F_y, less than or greater than 40 ksi.

Along shell =
Along nozzle =

Step 2: Utilizing the appropriate case (Figure 6-3) calculate the moment of inertia, I, and the distance from centroid to the inside of the shell, C.

$I =$
$C =$

Step 3: Compute membrane and bending stresses in accordance with the equations given later.

$\sigma_m =$
$\sigma_b =$

Step 4: Combine stresses and compare with allowable.

$\sigma_m + \sigma_b =$

Calculations

- *Membrane stress, σ_m nozzles with reinforcing pads* (Cases 1 and 3).

$$\sigma_m = P\left[\frac{\left(R_i\left(r_i + t + \sqrt{R_{mT}}\right) + R_i\left(T + T_e + \sqrt{r_m t}\right)\right)}{A_S}\right]$$

- *Membrane stress, S_m nozzles without reinforcing pads* (Cases 2 and 4).

$$\sigma_m = P\left[\frac{\left(R_i\left(r_i + t + \sqrt{R_{mT}}\right) + R_i\left(T + \sqrt{r_m t}\right)\right)}{A_S}\right]$$

- *Bending stress, σ_b.*

$$M = P\left(\frac{r_i^3}{6} + R_i r_i C\right)$$

$$\sigma_b = \frac{MC}{I}$$

- *Allowable stesses.*

$$\sigma_m < S$$
$$\sigma_m + \sigma_b < 1.5S$$

Notes

1. Openings that exceed the limits of UG-36(b)(1) shall meet the requirements of the 2/3 rule.
2. This analysis combines the primary membrane stress due to pressure with the secondary bending stress resulting from the flexure of the nozzle about the hard axis.
3. Boundaries of metal along the shell and nozzle wall are as follows:

	Along Shell	Along Nozzle
Cases 1 and 2	$\sqrt{R_m T}$	$\sqrt{r_m t}$
Cases 3 and 4	16T	16t

4. This procedure applies to radial nozzles only.

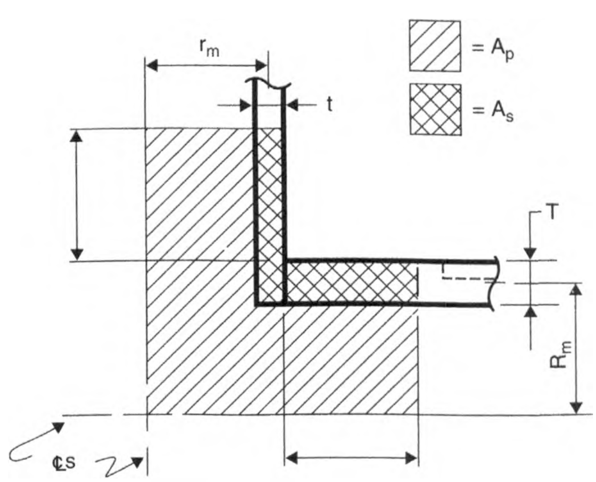
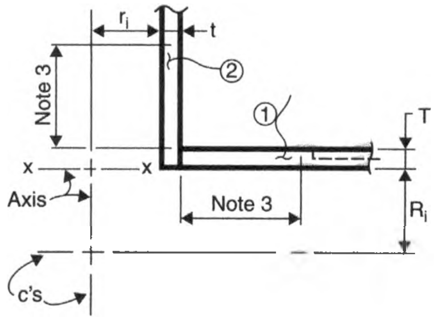


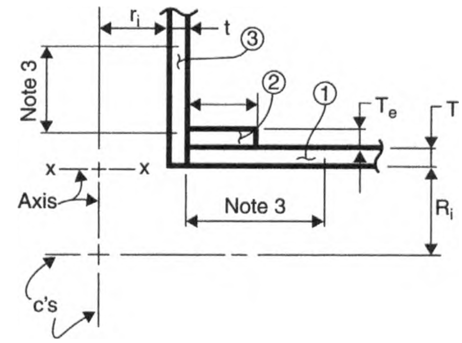
Figure 6-2. Areas of pressure and steel for nozzles.



Moment of Inertia					
Part	A	Y	AY	AY²	I
1					
2					
$\sum$					

$$C = \frac{\sum AY}{\sum A} \qquad I = \sum AY^2 + \sum I - C\sum AY$$

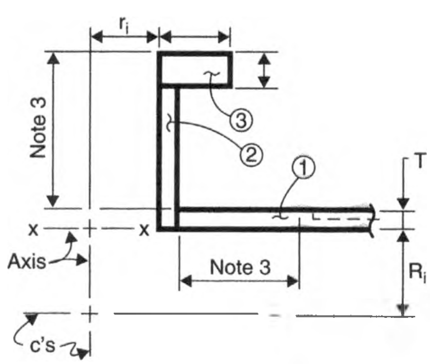
Case 1



Part	A	Y	AY	AY²	I
1					
2					
3					
$\sum$					

$$C = \frac{\sum AY}{\sum A} \qquad I = \sum AY^2 + \sum I - C\sum AY$$

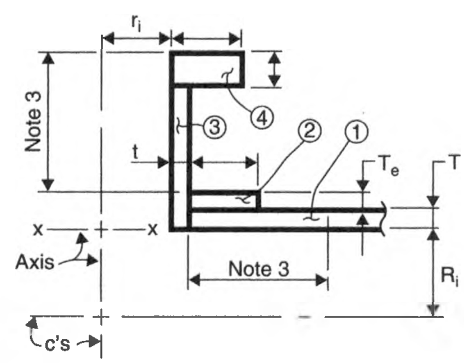
Case 2



Moment of Inertia					
Part	A	Y	AY	AY²	I
1					
2					
3					
$\sum$					

$$C = \frac{\sum AY}{\sum A} \qquad I = \sum AY^2 + \sum I - C\sum AY$$

Case 3



Part	A	Y	AY	AY²	I
1					
2					
3					
4					
$\sum$					

$$C = \frac{\sum AY}{\sum A} \qquad I = \sum AY^2 + \sum I - C\sum AY$$

Case 4

Figure 6-3. Calculation form for moment of inertia I and centroid C for various nozzle configurations. Select the case that fits the geometry of the nozzle being considered.

Procedure 6-2: Tower Deflection [3]

Notation

L = overall length of vessel, in.
L_n = length of section, in.
E_n = modulus of elasticity of section, psi
I_n = moment of inertia of section, in.^{4}
W_n = concentrated loads, lb
w = uniformly distributed load, lb/in.
W_{max} = uniformly distributed load at top of vessel, lb/in.
W_{min} = uniformly distributed load at bottom of vessel, lb/in.
X = ratio L_n/L for concentrated loads
δ = deflection, in.

Cases

Case 1: Uniform Vessel, Uniform Load

$$\delta = \frac{wL^4}{8EI}$$

Case 2: Nonuniform Vessel, Uniform Load

- *If E is constant*

$$\delta = \frac{w}{8E}\left[\left(\frac{L_1^4}{I_1} + \frac{L_2^4}{I_2} + \cdots + \frac{L_n^4}{I_n}\right) - \left(\frac{L_2^4}{I_1} + \frac{L_3^4}{I_2} + \cdots + \frac{L_n^4}{I_{n-1}}\right)\right]$$

- *If E is not constant*

$$\delta = \frac{w}{8}\left[\left(\frac{L_1^4}{I_1E_1} + \frac{L_2^4}{I_2E_2} + \cdots + \frac{L_n^4}{I_nE_n}\right) - \left(\frac{L_2^4}{I_1E_1} + \frac{L_3^4}{I_2E_2} + \cdots + \frac{L_n^4}{I_{n-1}E_{n-1}}\right)\right]$$

Section n	L_n	L_n^4	I_n	$\frac{L_n^4}{I_n}$	$\frac{L_n^4}{I_{n-1}}$
			$\Sigma=$		

$$\delta = \frac{w}{8E}\left[\sum\frac{L_n^4}{I_n} - \sum\frac{L_n^4}{I_{n-1}}\right]$$

Case 3: Nonuniform Vessel, Nonuniform Load

$$\delta = \left[\sum\frac{L_n^4}{I_n} - \sum\frac{L_n^4}{I_{n-1}}\right]\left[\frac{w_{min}}{8E} + \frac{5.5(w_{max} - w_{min})}{60E}\right]$$

Section n	L_n	L_n^4	I_n	$\frac{L_n^4}{I_n}$	$\frac{L_n^4}{I_{n-1}}$
			$\Sigma=$		

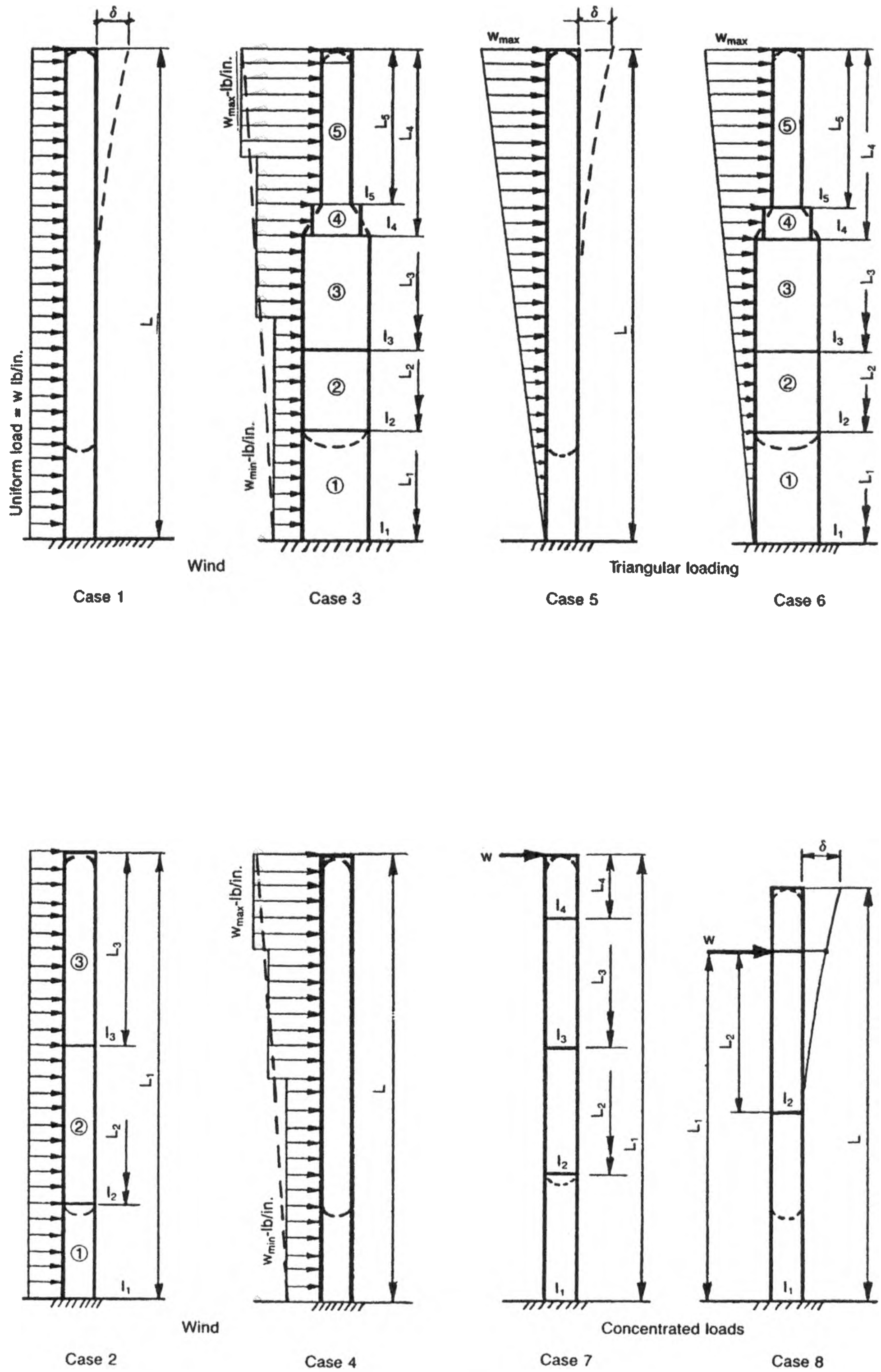


Figure 6-4. Dimension and various loadings for vertical, skirt-supported vessels.

Case 4: Uniform Vessel, Nonuniform Load

$$\delta = \frac{L^4}{I}\left[\frac{w_{min}}{8E} + \frac{5.5(w_{max} - w_{min})}{60E}\right]$$

Case 5: Uniform Vessel, Triangular Load

Load at base = 0
Load at top = w, lb/in.

$$\delta = \frac{5.5wL^4}{60EI}$$

Case 6: Nonuniform Vessel, Triangular Load

$$\delta = \frac{5.5w}{60E}\left[\sum\frac{L_n^4}{I_n} - \sum\frac{L_n^4}{I_{n-1}}\right]$$

Section n	L_n	L_n^4	I_n	$\frac{L_n^4}{I_n}$	$\frac{L_n^4}{I_{n-1}}$
			$\Sigma=$		

Case 7: Concentrated Load at Top of Vessel

- *Uniform vessel*

$$\delta = \frac{WL^3}{3EI}$$

Section n	L_n	L_n^3	I_n	$\frac{L_n^3}{I_n}$	$\frac{L_n^3}{I_{n-1}}$
			$\Sigma=$		

- *Nonuniform vessel*

$$\delta = \frac{W}{3E}\left[\sum\frac{L_n^3}{I_n} - \sum\frac{L_n^3}{I_{n-1}}\right]$$

Case 8: Concentrated Lateral Load at Any Elevation

$$X = \frac{L_1}{L}$$

- *For uniform vessel.*

$$\delta = \frac{WL_1^3}{3EI}\left(\frac{3 - X}{2X}\right)$$

- *For nonuniform vessel.*

$$\delta = \left[\frac{W}{3E}\left(\sum\frac{L_n^3}{I_n} - \sum\frac{L_n^3}{I_{n-1}}\right)\right]\left[\frac{3 - X}{2X}\right]$$

Section n	L_n	L_n^3	I_n	$\frac{L_n^3}{I_n}$	$\frac{L_n^3}{I_{n-1}}$
			$\Sigma=$		

Notes

1. This procedure calculates the static deflection of tall towers due to various loadings and accounts for the following:
   a. Different wind pressures at different elevations.
   b. Various thicknesses, diameters, and moments of inertia at different elevations.
   c. Different moduli of elasticity at different elevations due to a change in material or temperatures.
2. This procedure is not valid for vessels that are subject to wind-induced oscillations or that must be designed dynamically.
3. Deflection should be limited to 6 in. per 100 ft.
4. Deflections due to combinations of various loadings should be added to find the overall deflection.

Procedure 6-3: Design of Ring Girders [4–8]

The circular girder supports the weight of the tank, vessel, or bin; its contents; and any installed plant equipment. The ring beam will take the load from the vessel uniformly distributed over its full circumference, and in turn will be supported on a structural steel framework in at least four places.

The shell of a column-supported tank, vessel, or bin can be considered as a ring beam whether or not there is a special built-up beam structure for that purpose.

Horizontal seismic force is transferred from the shell or short support skirt to the ring beam by tangential shear. The girder performs the function of transmitting the horizontal shear from the tank shell to the rods and posts of the supporting structure.

The girder is analyzed as a closed horizontal ring acted upon by the horizontal shear stresses in the tank shell and by the horizontal components of the stresses in the rods and posts in the top panel of the supporting steel framework.

Maximum girder stresses generally occur when the direction of the earthquake force is parallel to a diameter passing through a pair of opposite posts.

The ring beam (girder) is subjected to compression, bending, and torsion due to the weight of the tank, contents, and horizontal wind or seismic forces. The maximum bending moment will occur at the supports. The torsional moment will be zero at the supports and maximum at an angular distance β away from support points.

This procedure assumes that the rods are tension-only members and connect every adjacent post. It is not valid for designs where the rods skip a post or two!

For cases where the ring beam has additional moment, tangential and/or radial loads (such as sloping columns) these additional horizontal loads may be calculated using ring redundants. See procedure on stresses in circular rings.

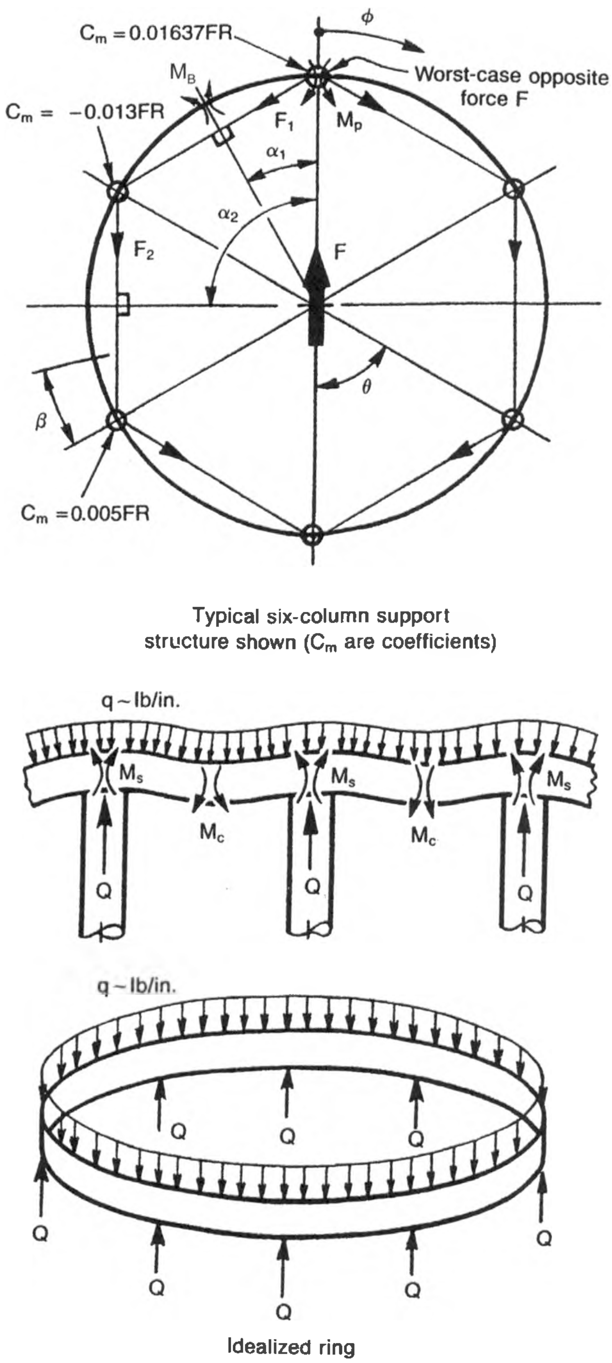


Figure 6-5. Dimension, forces, and loading at a ring girder.

Notation

D = diameter of column circle, in.
F = horizontal wind or earthquake force at plane of girder, lb

Table 6-2
Internal bending moments

No. of Posts	Due to Force Q M_s	M_c	M_T	β	Due to Force F M_P	M_B
4	−0.1366 QR	+0.0705 QR	+0.0211 QR	19°-12′	+0.0683 FR	−0.049 FR
6	−0.0889 QR	+0.0451 QR	+0.0090 QR	12°-44′	+0.0164 FR	−0.013 FR
8	−0.0661 QR	+0.0333 QR	+0.0050 QR	9°-33′	+0.0061 FR	−0.0058 FR
10	−0.0527 QR	+0.0265 QR	+0.0032 QR	7°-37′	+0.0030 FR	−0.0029 FR
12	−0.0438 QR	+0.0220 QR	+0.0022 QR	6°-21′	+0.0016 FR	−0.0016 FR
16	−0.0328 QR	+0.0164 QR	+0.0090 QR	4°-46′	+0.0007 FR	−0.0007 FR

1. Values in table due to force Q are based on Walls, Bins, and Grain Elevators *by M.S. Ketchum, McGraw-Hill Book Co., 1929. Coefficients have been modified for force Q rather than weight W.*
2. Values in table due to force F are based on "Stress Analysis of the Balcony Girder of Elevated Water Tanks Under Earthquake Loads" by W.E. Black; Chicago Bridge and Iron Co., 1941.

$F_{1,2}$ = resisting force in tie rod, panel force, lb
f_b = bending stress, psi
R = radius of column circle, in.
R_t = torsional resistance factor
Q = equivalent vertical force at each support due to dead weight and overturning moment, lb
q = uniform vertical load on ring beam, lb/in.
q_t = tangential shear, lb/in.
W = operating weight, lb
β = location of maximum torsional moment from column, degrees
I_x, I_y = moment of inertia, in.^{4}
τ = torsional shear stress, psi
B_p = bearing pressure, psi
J = polar moment of inertia, in.^{4}
M = bending moment in base plate due to bearing pressure, in.-lb
M_B = horizontal bending moment between posts due to force F, in.-lb
M_c = vertical bending moment between posts due to force Q, in.-lb
M_o = overturning moment of vessel at base of ring beam, in.-lb
M_p = horizontal bending moment at posts due to force F, in.-lb
M_s = vertical bending moment at posts due to force Q, in.-lb
M_T = torsional moment at distance β from post, in.-lb

Formulas

$$M_s = \frac{WR}{N}\left[\frac{1}{\theta} - \frac{0.5}{\tan\theta/2}\right]$$

$$M_c = M_s\cos\frac{\theta}{2} + \frac{WR}{2N}\left[\sin\frac{\theta}{2} - \frac{2\sin^2\theta/4}{\theta/2}\right]$$

$$M_T = (-)M_s\sin\beta - \frac{WR}{2N}(1-\cos\beta)$$

$$+\frac{WR\beta}{2\pi}\left(1 - \frac{\sin\beta}{\beta}\right)$$

$$q_t = \frac{F\sin\phi}{\pi R}$$

$$F_{1,2\ldots} = \frac{2F\sin\alpha_n}{N}$$

F_n is maximum where $\alpha = 90°$ since $\sin 90° = 1$.

$$q = (-)\frac{W}{\pi D} \pm \frac{4M_o}{\pi D^2}$$

$$Q = \frac{\pi D q}{N}$$

Load Diagrams

Vertical Forces on Ring Beam

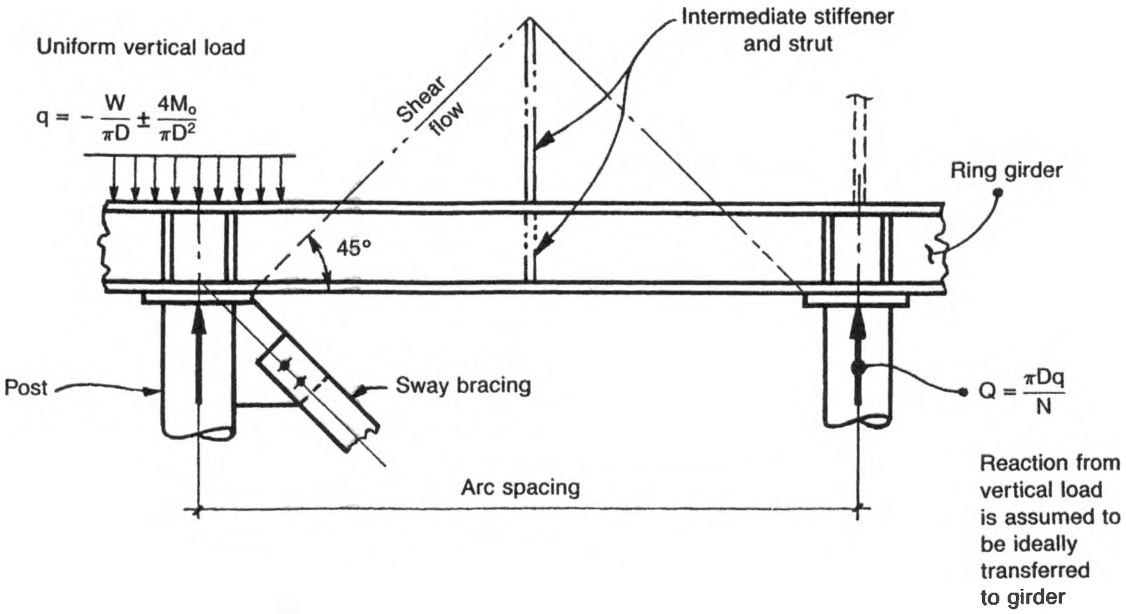


Figure 6-6. Loading diagram for a ring girder: vertical forces on a ring beam.

Horizontal Forces on Ring Beam In the analysis for in-plane bending moment and thrust, the wind or seismic force is assumed to be transferred to the girder by a sine-distributed tangential shear. (See Figure 6-7.) These loads are resisted by the horizontal reaction components of the sway bracing as shown in Figure 6-8.

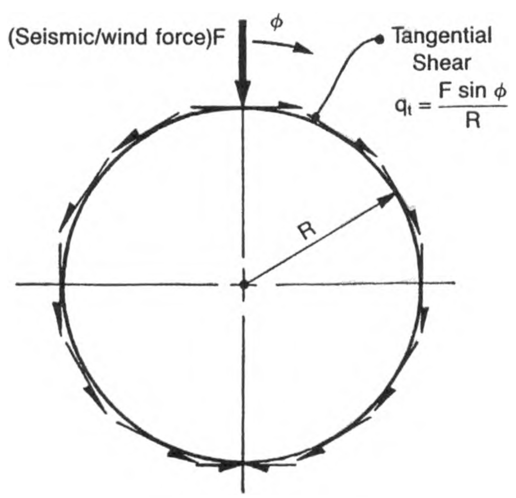


Figure 6-7. Loading diagram for a ring girder: shell to beam.

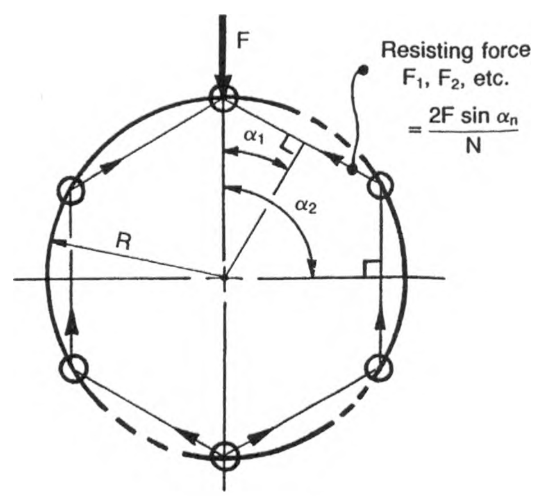


Figure 6-8. Loading diagram for a ring girder: support structure to beam.

Procedure

- *Determine loads q and Q.*

$$q = (-)\frac{W}{D} \pm \frac{4M_o}{\pi D^2}$$

$$Q = \frac{\pi D q}{N}$$

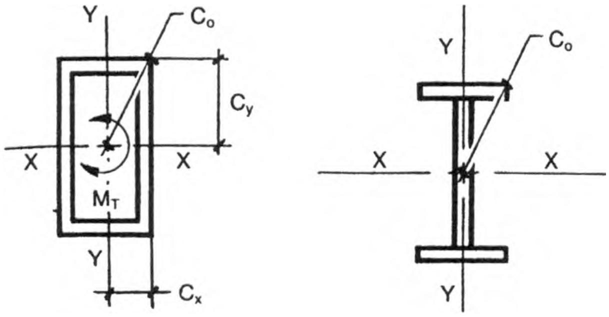


Figure 6-9. Axis and distance of extreme fibers of typical beam sections.

- *Determine bending moments in ring.*
  Note: All coefficients are from Table 6-2.

M_S = coefficient × QR
M_C = coefficient × QR
M_T = coefficient × QR
M_P = coefficient × FR
M_B = coefficient × FR

- *Determine properties of ring.*

For torsion the formula for shear stress, τ, is

$$\tau = \frac{M_T C_o}{J}$$

where J = Polar moment of inertia, in.^{4}
$= I_x + I_y$
C_o = Distance to extreme fiber, in.

Note: Box sections are best for resisting torsion.

An alternate procedure is suggested by Blodgett in *Design of Welded Structures* for substituting a torsional resistance factor, R_t, for the polar moment of inertia in the equation for stress. The torsional resistance factor, R_t, is determined by dividing up the composite section into its component parts, finding the properties of these components, and adding the individual properties to obtain the sum. An example is shown in Figure 6-10.

R_t for any rectangular section $= \gamma b d^3$. See Table 6-3 for γ.

- *Stresses in beam.*

Note: Bending is maximum at the posts. Torsion is maximum at β.

$$f_{bx} = \frac{M_s C_y}{I_x}$$

$$f_{by} = \frac{M_P C_x}{I_y}$$

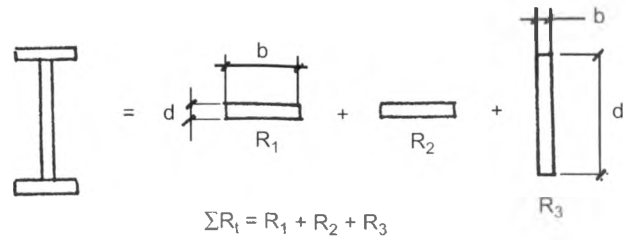


Figure 6-10. Determination of value R_t for typical section.

Table 6-3
Values of coefficient γ

b/d	γ
1.0	0.141
1.5	0.196
1.75	0.214
2.0	0.229
2.5	0.249
3.0	0.263
4.0	0.281
6.0	0.299
8.0	0.307
10.8	0.313
∞	0.333

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$$\tau = \frac{M_T C_o}{\sum R_t}$$

- *Additional bending in base plate.*

Additional bending occurs in base plate due to localized bearing of post on ring.

Bearing pressure, B_p, psi

$$B_p = \frac{Q}{A} \pm$$

where A = assumed contact area, area of cap plate or cross-sectional area of post. See Figure 6-11. Assume reaction is evenly distributed over the contact area.

ℓ = Cantilever, in.
L = Semifixed span, in.

Note: Maximum bending is at center of base plate.

- *Moment for cantilever portion.*

$$M = \frac{B_p \ell^2}{2}$$

- *Moment for semifixed span.*

$$M = \frac{B_p L^2}{10}$$

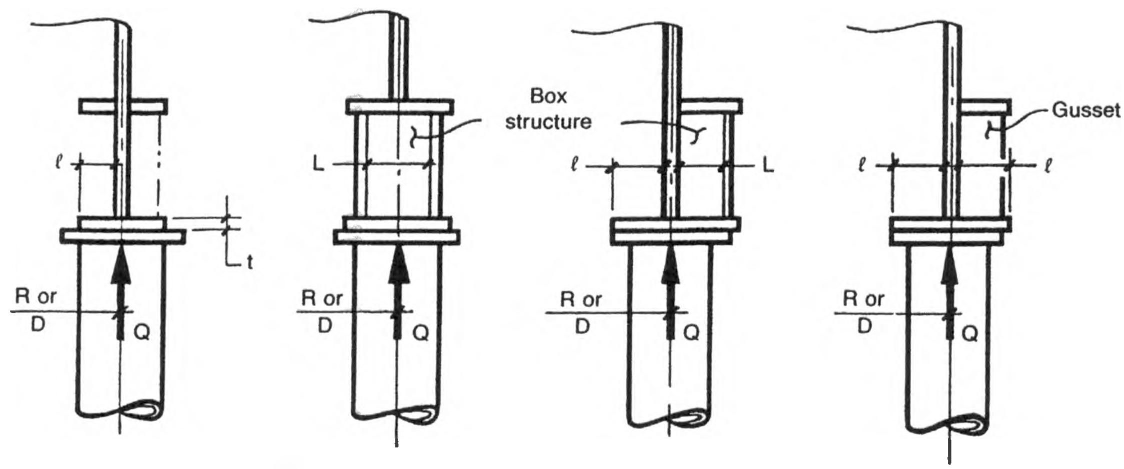


Figure 6-11. Dimensions and loadings for various ring girders.

- *Bending stress, f_b.*

$$f_b = \frac{6M}{t^2}$$

Notes

1. The shell of a column-supported tank, vessel, or bin is considered to be a "circular girder" or "ring beam" uniformly loaded over its periphery and supported by columns equally spaced on the ring circumference.
2. The ring beam (girder) is subjected to compression, bending, and torsion due to the weight of the tank and contents and horizontal wind or seismic force.
3. The maximum bending moment occurs at the supports.
4. The torsional moment M_T will be 0 at the supports and maximum at angular distance β away from supports.

Procedure 6-4: Design of Vessels with Refractory Linings [9–12]

Vessel engineers must eventually become familiar with refractory materials, techniques, strategies and installation in order to properly specify, analyze and check vendor designs for refractory lined vessels, equipment and components. Refractory lining is utilized as a heat barrier, insulator, or as an abrasion resistant lining, or both. Dual component linings are utilized for insulating and abrasion resistance.

Refractory lined equipment and vessels are some of the most interesting applications encountered in the engineering field. During the past 80 years engineers have been perfecting systems where they can connect a layer of ceramic material on steel to protect the steel from effects of high temperature and abrasion. This unique application allows both materials to do what they do best. The alternative, in the case of high temperature, is to specify an exotic alloy. Not only is this more expensive, but even the most exotic alloys available for these applications are only good to about 1500°F. While there are metals that are used up to 2800°F, such as moly used for heat shields, these alloys are not suitable for building vessels, heaters and gasifiers. Above this temperature, you must use some type of refractory barrier to resist the heat.

Refractory materials can be attached to the skin by a variety of anchorage methods. The refractory itself may be put into place by casting, vib-casting, gunniting or ram packed. Anchorage methods, the metal parts that hold the refractory to the metal skin, can be steer bar, wavy V, hex mesh, S bar and various types of punch tabs.

The combination of refractory attached to a steel skin is an unholy alliance, but have some synergies that make the combination work. For example, refractory is good in compression and steel is good in tension. Refractory and metal have drastically different coefficients of thermal expansion. Since the refractory must be connected in some fashion to the steel, the two materials grow at different rates. The result is that the refractory always cracks. This too works to the advantage of the composite system. The cracks in the refractory initiated during cooling, will "seal up" once heated again. The hot face is so much hotter than the cold face that the cracks seal up during operation and prevent the heat from propagating to the skin.

Every refractory has a limited service temperature, or maximum temperature. Above these temperatures brick linings are used. The cutoff of refractory in lieu of brick is about 1800°F. Bricks have the capacity to seal and return to shape after heating without spalling. Solid monolithic linings do not. In addition, bricks can be made in thicker sections than can be cast from refractory.

The cracks formed in monolithic linings above these temperatures are too severe for the material to recover. In the early day of FCC units, the regenerators and reactors were brick lined, like furnaces, because the technology of gunnited and cast linings was not capable of withstanding the service. Brick linings are still utilized in a variety of applications. Shapes are available to conform to cylindrical shapes of containers most commonly used in these vessels and equipment.

In addition to heat resistance and insulating, refractory can provide some excellent abrasion resistance as well. This situation led to the development of the dual component lining used for the past 50 to 60 years. This concept was born because of the properties of the refractory materials themselves. In general, the lighter the refractory, the better its insulating properties but the worse its erosion resistance. The reverse is true for denser, heavier refractories. The denser the refractory, the higher the thermal conductivity, the poorer its insulating capacity but the better its erosion resistance. Early designers quickly arrived at a solution to this problem where insulating and abrasion resistance were both required. The solution was to have a layer of light insulating refractory with a top layer of dense, abrasion resistance refractory.

This combination was usually accomplished by attaching pins and anchors to the skin. The anchors supported the refractory, while the pins extended to the surface where hex mesh was attached to the pins by welding or other mechanical means. The dense, hard abrasion resisting material could be much thinner. The typical application was 3-4" of soft insulating refractory, say 60 PCF, covered with 1" thick of very dense, 165 PCF material in hex mesh. Each material doing what it was best suited for.

But dual component linings had their drawbacks. They were susceptible to degradation due to vibration. Biscuits would break out of the hex mesh and allow the soft layer underneath to be eroded. In short time the steel skin could also be eroded and lead to repairs and/or shut down.

Today, there is a compromise material used for these applications. They use denser refractories, though not as dense as the 165 PCF material through the entire thickness. It has good abrasion resistance properties, and fair insulation refractories. It weighs about 135 PCF and is vib-cast into position. A similar product is made for casting. The material is mixed with stainless steel needles, approximately 2-4 lbs per 60 lbs of refractory, to give the material added strength. This material has a higher thermal conductivity than the insulating castables utilized in the dual component system and therefore you have higher shell temperatures, and greater growth. This additional growth must be designed for in the overall supports and guidance of the system. Shell temperatures are in the range of 100 to 200°F higher than dual component systems.

Vib-cast materials must be cast with an internal form that must be removed later. This is costly but necessary. The materials are cast with very little water so must be vibrated during casting to remove air bubbles to ensure a monolithic lining without voids. This material has been perfected as an alternative to dual component systems without the inherent weaknesses of the dual component system, that is, the loss of biscuits due to vibration and wear.

Lining equipment and vessels with refractory is an economic alternative that will undoubtedly be around for a long time. Thus it behooves the engineers in our industry to familiarize themselves with the properties and techniques used in the industry today, to better serve the needs of industry tomorrow.

The circular cross section of vessels and stacks provide ideal shapes for the supporting and sustaining refractory linings from a stress standpoint. There are a variety of stresses developed in the lining itself as well as stresses induced in the steel containment shell. Compressive stresses are developed in the lining and are a natural result of the temperature gradient. These compressive stresses help to keep the lining in position during operation. This compressive condition is desirable, but must not be so high as to damage the lining.

Several idealized assumptions have been made to simplify the calculation procedure. These are;

1. Assumes steady state conditions
2. Assumes that stress strain relationships are purely elastic
3. Assumes shrinkage varies linearly with temperature
4. Assumes that thermal conductivity and elastic moduli are uniform throughout the lining
5. Circumferential stresses are greater than longitudinal stresses in cylindrical vessels and therefore are the only ones calculated here.

The hot face is in compression during operation and heat-up cycles and in tension during cool-down cycles. The tension and compressive loads vary across the cross-section of the lining during heating and cooling phases. The material at the mean temperature is not necessarily in compression during operation but may be tension or neutral. The hot face stress should always be compressive and is the maximum compressive stress in the lining. If it is not compressive, it can be made to be so either by increasing the thickness of the lining or choosing a refractory with a higher thermal conductivity. Excessive compressive stresses will cause spalling.

The cold face is under tensile stress. This stress often exceeds the allowable tensile stress of the material and cracks must develop to compensate for the excessive tensile stress. The tensile stress is always maximum at the cold face.

Upon cooling of the vessel, the irreversible shrinkage will cause cracks to propogate through the lining. The shrinkage of the hot face amounts to about .001 in/in and crack width at the surface would vary from .01 in to .03 in. These cracks will close early in the reheat cycle and will remain closed under compression at operating temperatures.

Monolithic refractories creep under compressive stress. At stresses much less than the crush strength, the creep rate diminishes with time and approaches zero. Creep occurs under nominally constant stress. When strain instead of stress is held constant, the stress relaxes by the same mechanism that causes creep. Creep rate decreases at lower temperatures and drops off with temperature.

Allowable Refractory Stresses

There is no code or standard that dictates the allowable stresses for refractory materials. Refractory suppliers do not have established criteria for acceptable stress levels. In addition there is very limited experimental information on the behavior of refractory materials under multiaxial stress states.

One criteria that has been used is a factor of safety of 2 based on the minimum specified crush strength of the material at temperature for the allowable compressive stress. The corresponding allowable tensile stress is 40% of the modulus of rupture at 1000°F.

Spalling

Spalling is the technical term used to describe cracking of ceramic materials.

Most materials will crack if rapidly heated and cooled. Metals do not spall because of high ductility and high thermal conductivity. Graphite bricks and crucibles will not spall because of high thermal conductivity. Glass and silica brick spall easily due to high thermal expansion.

However while all refractory materials crack, we hope that they will not spall. Spalling of refractory materials most commonly refers to the fracture of the surface material, usually due to trapped water inside the refractory that converts to steam before the water vapor can escape. This is why strict precautions are taken to dry-out the refractory by baking out any trapped water at a very slow rate. Since refractories are mixed with water to make them viscous enough to cast or gunnite, some amount of water remains trapped in the solidified structure. Like concrete, a certain amount of water is consumed in the chemical reaction of the refractory as it solidifies.

The loose water will become destructive and may cause spalling if not removed. The excess water is removed by slowly baking out the material and allowing the water vapor to slowly dissipate out of the material. This bake out procedure is called the "dry-out" procedure. The material is considered to be safe, and water free once the steel containment reaches 212°F. In effect if the cold face is 212°F, then all of the refractory is at least this temperature as well. This is proof that no free water remains in the material.

Spalling Resistance, R, is given by the following formula;

$R = k\,S / \alpha\,E\,C_p\,p$

k = Thermal conductivity, BTU-in/hr/sq. ft./°F
S = Tensile strength, PSI
α = Coefficient of thermal expansion, in/in/°F $\times 10^{-6}$ from 70°F
E = Modulus of elasticity, PSI
C_p = Specific heat, BTU/Lb/°F
ρ = Density, Lbs/in^3

Refractory Selection

Economy in the use of refractories is governed largely by the selection of the types best suited for a given application. A careful study of equipment design and operating conditions will greatly aid in this selection process.

The type and class of refractories most suitable for a particular case will be determined by such factors as the following:

1. Reason for using refractory;
   a. Thermal protection
   b. Abrasion protection
   c. Both of above
2. Factors related to operation;
   a. Function of the equipment
   b. Properties of contents
   c. Rate of operation
   d. Continuity of operation
   e. Temperature
   f. Rate of change of temperature
   g. Source of heat
   h. Heat release per volume
   i. Rate of heat dissipation
   j. Chemical attack
   k. Velocity of gas stream
   l. Abrasion from dust or particles in moving gas
   m. Impingement of flame for hot spots
   n. Machinery vibration
3. Economic factors;
   a. Delivered cost
   b. Cost of installation
   c. Standard sizes vs special shapes
   d. Service life
   e. Shelf life of product
4. Types of refractory;
   a. Fireclay and high alumina brick
   b. Insulating firebrick
   c. Castable refractory
   d. High alumina plastic refractory
5. Classification of refractory;
   a. Acid
   b. Basic
   c. Neutral
   d. Silica
   e. Special (carbon, Si-C, Zr)
6. Installation;
   a. Brick
   b. Castable
   c. Ramming (plastic)
   d. Vib-cast
   e. Gunnited
7. Anchor type;
   a. Wavy V
   b. Steer bar
   c. "S" bar
   d. Hex mesh
   e. Tyne
   f. Wire
   g. Studs
8. Anchor parameters;
   a. Anchor spacing (1.5 to 3 $\times$ t_L)
   b. Anchor height (2/3 $\times$ t_L)
   c. Anchor pattern (diamond, square, staggered)
   d. Metallurgy
   e. Ceramic
9. Properties of refractory;
   a. Thermal rating
   b. Chemical properties
   c. Abrasion properties
   d. Strength properties
   e. Corrosion/Chemical resistance
   f. Thermal expansion
   g. Thermal shock
   h. Temperature of vitrification
   i. Reversible thermal expansion
   j. Resistance to mechanical stress and impact
   k. Thermal conductivity
   l. Heat capacity
   m. Electrical resistivity
10. Factors relating to design & construction;
   a. Type of equipment
   b. Design and dimensions of walls and arches
   c. Volume for maximum fuel input
   d. Loads imposed upon the lining materials
   e. Conditions of heating
   f. Degree of insulation
   g. Air or water cooling
   h. Type of construction
   i. Methods of bonding and support
   j. Type of bond
   k. Thickness of joints
   l. Nature of bonding material
   m. Provision for thermal expansion

Calculating Heat Loss and Cold Face Temperatures

In any refractory lined system it is required to determine the heat loss and skin temperatures in order to do a proper design of the metal containment (shell) as well as to determine the selection and effectiveness of the refractory itself.

For either case, the designer must first determine the outer skin temperature of the metal containment. This is accomplished by first knowing the following;

1. Refractory material
2. Refractory properties

3. Refractory thickness
4. Internal process temperature
5. Heat transfer coefficients, inside and out
6. Ambient temperature

The heat loss is necessary to validate the selection of the refractory material and thickness. It also will validate the process calculations for heat and material balance.

In any refractory lined system, it is required to determine the thermal expansion of the system in order to determine the stress, support points and restraint points. In order to do this, the designer must accurately determine the skin temperature of the various components. From this data, the thermal expansion of the components is computed, and from this, the stress, depending on support and restraint points.

The structural engineer will need this load data to properly design their foundations and structures to accommodate these loads. The vessel engineer will need this data to determine the local stresses imposed by the installed transfer lines. The piping engineer will need the loads, forces and expansion data to determine if expansion joints are needed in the system. The skin temperatures are needed to determine the allowable stresses of the steel jackets.

The stress in the refractory itself should also be determined based on this analysis to ensure that the refractory material is not overstressed. In any refractory system, the hot face of the refractory expands at a greater rate then the cold wall steel. This keeps the refractory in compression when hot, and seals the cracks. When cooled, cracks will develop in the refractory due to the different temperatures and properties of the steel and refractory.

Before this process can begin, the thermal conductivity (k value) of the selected refractory material must be known. This normally comes from the manufacture of the material and typically published in their literature. However there is considerable variance in test results, depending on how the test is done. Actual results in the field may also vary significantly depending on the installation parameters. Different manufacturers use different testing methods. Testing methods include the following techniques;

1. Hot Wire method
   (ASTM C1113)
2. Water Calorimeter
   (ASTM C-201)
3. Water Calorimeter
   (ASTM C 417)
4. Panel test
5. Dynatech Test

Thermal Conductivity: k

- Units are BTU-in/hr/sq. ft./°F
- A measurement of a materials ability to transmit heat (conduction) or to resist transmitting heat (insulating)
- The lower the value of thermal conductivity, the greater the insulating value. The higher the value, the better conductor.
- Insulators range from .2 to 100. Metals range from 100 to 2700.
- Thermal conductivity for any material increases with temperature. However, the rate of increase, decreases with temperature.
- Thermal conductivity is typically noted as a "k" value.
- Another common term for defining insulating efficiency is the "R" Value.
  $R = t/k$ where t is the thickness of the refractory.

Specific Heat: C_p

- Units are in BTU/Lb/°F
- Definition: The amount of heat energy required to raise 1 Lb of material, 1°F. It is a measure of the amount of heat that can be stored in a material.
- The specific heat of water is 1. Air is .24. Metals range from .03 to .12. Refractory materials are in the range of .2 to .27
- Therefore refractory materials can store much more heat per pound than metals and are about like air in terms of thermal storage.

Refractory Failures and Potential Causes of Hot Spots

The following list are some potential causes of refractory failure, cracking and subsequent hot spots;

1. Refractory spalling; Spalling can be caused by excessive moisture in the material during heating. Too rapid of heat-up or cool down cycles. Too high of thermal gradient across the lining due to improper design, either too thick a lining or too low a thermal conductivity. This case leads to excessive hot face compression.

2. Poor refractory installation
3. Poor refractory material
4. Excessive deflection or flexing of the steel shell due to pressure, surge or thermal stresses
5. Differential expansion
6. Excessive thermal gradient
7. Upsets or excursions leading to rapid heating or cooling rates. These should be limited to about 100°F/hour
8. Upsets or excursions leading to temperatures near or exceeding the maximum service temperature.
9. Poor design details
10. Poor refractory selection
11. Improper curing or dry out rates
12. Poor field joints
13. Temperature differential
14. Incorrect anchorage system
15. Vibration
16. Anchor failure

Refractory Lined Equipment

1. Furnaces
2. Boilers
3. Fired Heaters
4. Incinerators
5. FCC Transfer Lines
6. FCC Flue Gas Lines
7. Expansion Joints
8. Stacks
9. Stills
10. Retorts
11. Dutch Ovens
12. Kilns
13. Sulfur Plants
    a. Converters
    b. Reaction Furnace
    c. Waste Heat Boilers
    d. Hydrogenation Reactor

Refractory Lined Vessels

1. FCC Vessels
2. RFCC Vessels
3. Flexicoker Vessels
4. Fluid Coker Vessels
5. Flexicracking
6. Reactors
7. Secondary Reformers
8. Gasifiers
9. Rheniformer Reactors
10. Third Stage Separators
11. Adiabatic Reformer
12. Orifice Chambers

General Refractory Notes

- Once the hot spots have occurred there is obviously a heat leak path to the vessel wall. The subsequent heating of the shell locally also affects the anchors. Since the anchors are made of stainless steel, they grow more then the shell and therefore relax their grip on the refractory. This in turn allows the gap between the shell and refractory to grow.
- Refractory failures are categorized as either tension or compression failures. These failures can result from bending or pure tension/compression loads. In a tension failure the crack is initiated and grows. A "cold joint" is the preferred fix for a tension failure.
- A compression failure will tend to pull the lining away from the wall. A flexible joint with ceramic fiber is the good solution for this type of failure.
- During operation the hot face is in compression varying through the thickness to tension against the steel shell. This is caused by thermal expansion of the material and thermal gradient forces developed internally.
- During the cooling cycle the hot face will be in tension. If the cooling cycle is too rapid or the anchoring too rigid, then the tensile stress of the material becomes critical in resisting cracking.
- Due to low tensile strength, cracking occurs at early stages of load cycles, which ultimately results in load redistribution.
- Temperature loading such as heat-up, cool-down and holding periods at lower temperatures results in stress cycling.
- Refractory properties are non-linear.
- Compressive strength is practically independent of temperature while tensile strength is very dependent on temperature.
- Refractory material undergoes a permanent change in volume due to both loss of moisture during the dryout cycle as well as a change in the chemical structure. The effects of moisture loss as well as chemical metamorphosis are irreversible.

- During initial heating, the steel shell has a tendency to want to pull away from the refractory. The cooler the shell the less the impact on the refractory. The cooler shell tends to hold the refractory in compression longer.
- The use of holding periods during the heat-up and cool-down cycles results in relaxation of compressive stresses due to creep. However this same creep may introduce cracks once the lining is cooled off.
- The two most important effects on refractory linings are creep and shrinkage.
- Optimum anchor spacing is 1.5 – 3 times the thickness of the lining
- Optimum anchor depth is approximately 2/3 of the lining thickness.

Calculate Skin Temperature for Refractory Lined Components

Notation

h = Combined convection and radiation coefficient, outside BTU/Hr/Ft^2/ °F

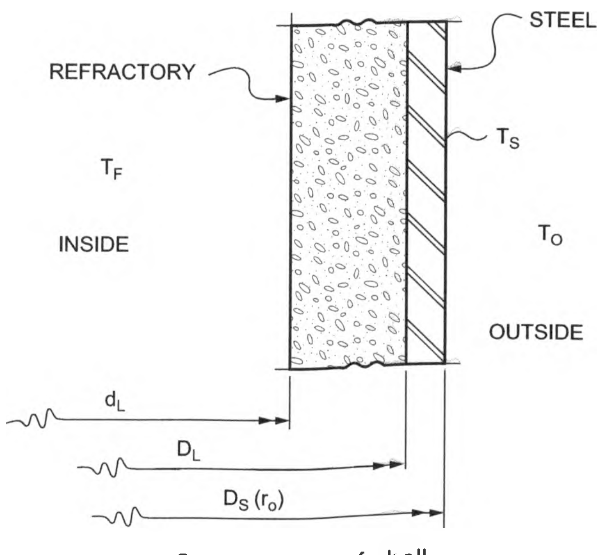
K_L = Thermal conductivity of refractory at average temperature of layer BTU-in/Hr/Ft^2/°F

L_n = Log base n

T_s = Skin temperature, outside, °F

T_F = Internal process temperature, °F

T_O = Outside ambient temperature, °F



Cross section of shell

Calculation. Solve the following equation for T_S;

$$(T_S - T_O)hr_o = \frac{(T_F - T_S)K_L}{L_n(D_L/d_L)}$$

Table 6-4
Values of K_L, BTU-in/Hr/Ft^2/°F

Material	1000°F	1500°F	2000°F
RS-3	1.65	1.80	
RS-7	2.35	2.56	
RS-17EC	6.30	6.90	7.70

Sample Problem

T_O = 20°F
T_F = 1425°F
h = 4.0 BTU/Hr/Ft^2/°F
d_L = 312 inches
D_L = 320 inches
D_S = 323.5 inches
r_o = 161.75 inches
K_L = 2.56 (RS-7 @ 1500°F) BTU-in/Hr/Ft^2/°F
D_L/d_L = 1.0256

Solve for T_S ...

$$(T_S - 20)4.0(161.75) = ((1425 - T_S)2.56)/L_n\ 1.0256$$

$$16.35(T_S - 20) = 3648 - 2.56\ T_S$$

$$16.35\ T_S - 327.09 = 3648 - 2.56\ T_S$$

$$18.91\ T_S = 3975.09$$

$$T_S = 210°F$$

Notes

1. To determine worst case, assume outside temperature as summer, no wind
2. The value for h, temperature coefficient, is temperature dependent. Therefore an initial temperature must be assumed
3. For multiple layers of refractory, calculate each layer separately.

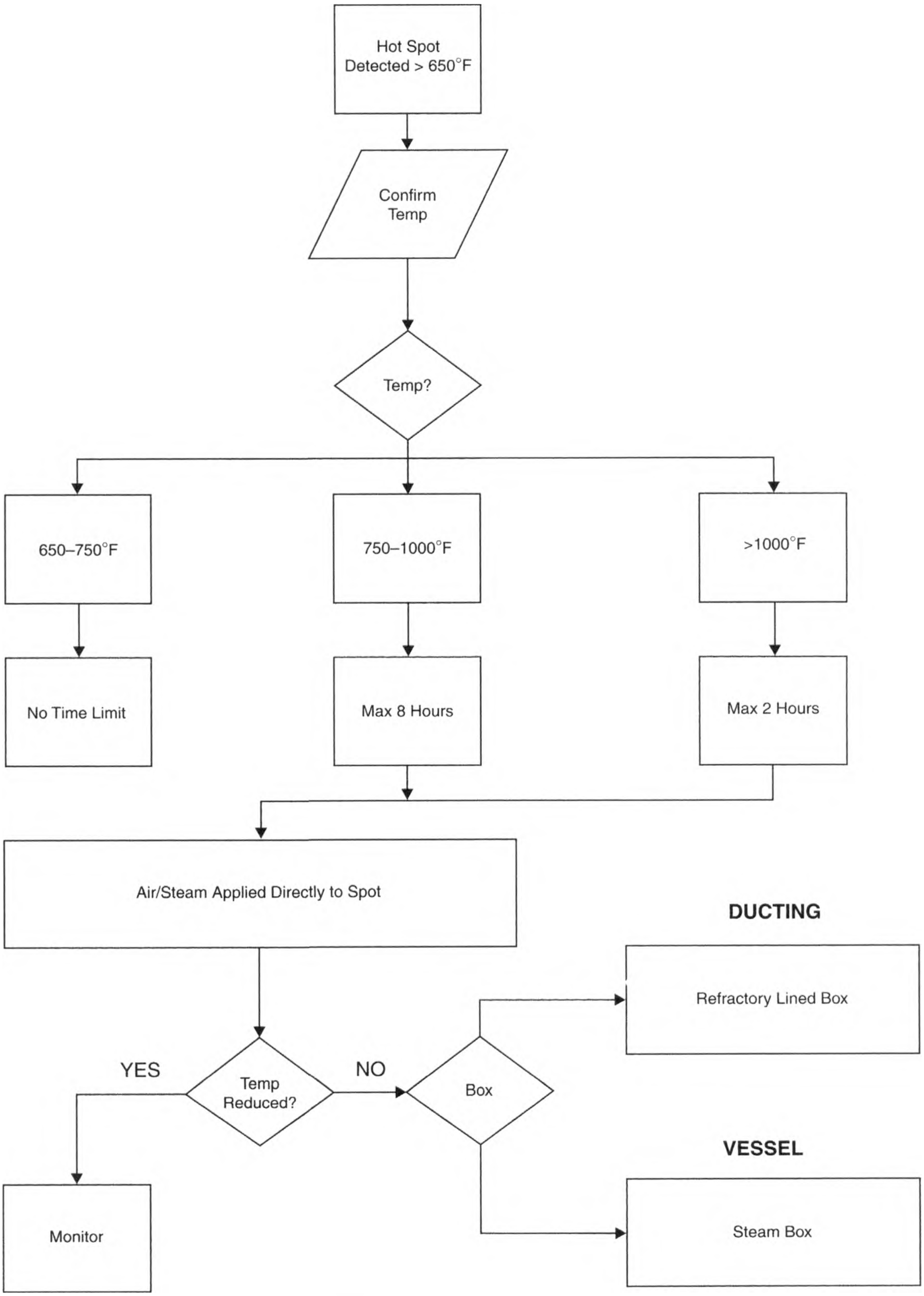


Figure 6-12. Hot spot decision tree.

Notation

Shell Properties

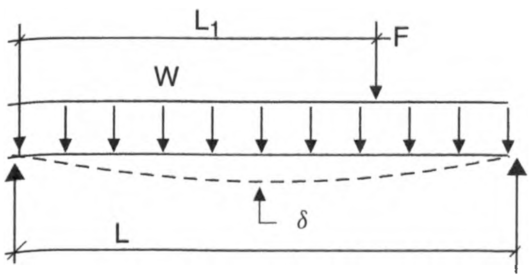
D = shell ID, in.
D_s = shell OD, in.
E_s = modulus of elasticity, shell, psi
I_s = moment of inertia, shell, $in.^4$
K_s = thermal conductivity, shell, BTU-in/hr/ft^2/°F
t_s = thickness, shell, in.
W_s = specific density, steel, pcf
α_s = thermal coefficient of expansion, shell, in./in./°F

Refractory Properties

D_L = refractory OD, in.
d_L = refractory ID, in.
E_L = modulus of elasticity, refractory, psi
F_u = allowable compressive stress, refractory, psi
I_L = moment of inertia, refractory, $in.^4$
K_L = thermal conductivity, refractory, BTU-in/hr/ft^2/°F
S_{TS}, S_{TL} = irreversible shrinkage of lining @ temperatures T_S, T_L
t_L = thickness, refractory, in.
W_L = specific density of refractory, pcf
α_L = thermal coefficient of expansion, refractory, in./in./°F
μ_L = Poisson's ratio, refractory

General

E_{eq} = modulus of elasticity of composite section, psi
h_i, h_o = film coefficients, inside or outside, BTU/hr/ft^2/°F
P = internal pressure, psig
Q = heat loss through wall, BTU/hr/ft^2
T_a = temperature, outside ambient, °F
T_c = temperature, outside ambient during construction, °F
T_L = temperature, refractory, mean, °F
T_{L1} = temperature, lining, inside, °F
T_o = temperature, internal operating, °F
T_s = temperature, shell, mean, °F
T_{s1} = temperature, shell, inside, °F
T_{s2} = temperature, shell, outside, °F
W = overall weight, lb
W_{eq} = equivalent specific density, pcf
δ = deflection, in.
ε_ϕ = circumferential strain due to internal pressure, in./in.
L1 = thermal expansion, shell, in./in.
L2 = thermal expansion, shell, without lining stress, in./in.
L3 = mean thermal expansion, in./in.
L4 = mean shrinkage, in./in.
L5 = net mean unrestrained expansion, in./in.
L6 = net differential circumferential expansion, in./in.
σ_{L1} = mean compressive stress, refractory, due to restraint of shell, psi
σ_{L2} = stress differential from mean, refractory, due to thermal expansion gradient, psi
σ_{L3} = stress differential from mean, refractory, at hot face due to shrinkage, psi
σ_{L4} = circumferential stress in refractory, at hot face, psi
σ_{L5} = circumferential stress in refractory, at cold face, psi
σ_{sc} = circumferential stress in shell caused by the lining, psi
σ_ϕ = circumferential stress due to internal pressure, psi



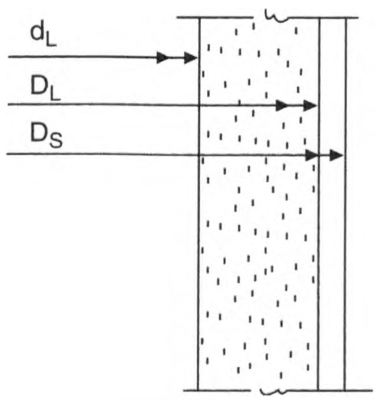


Figure 6-13. Lining dimensions.

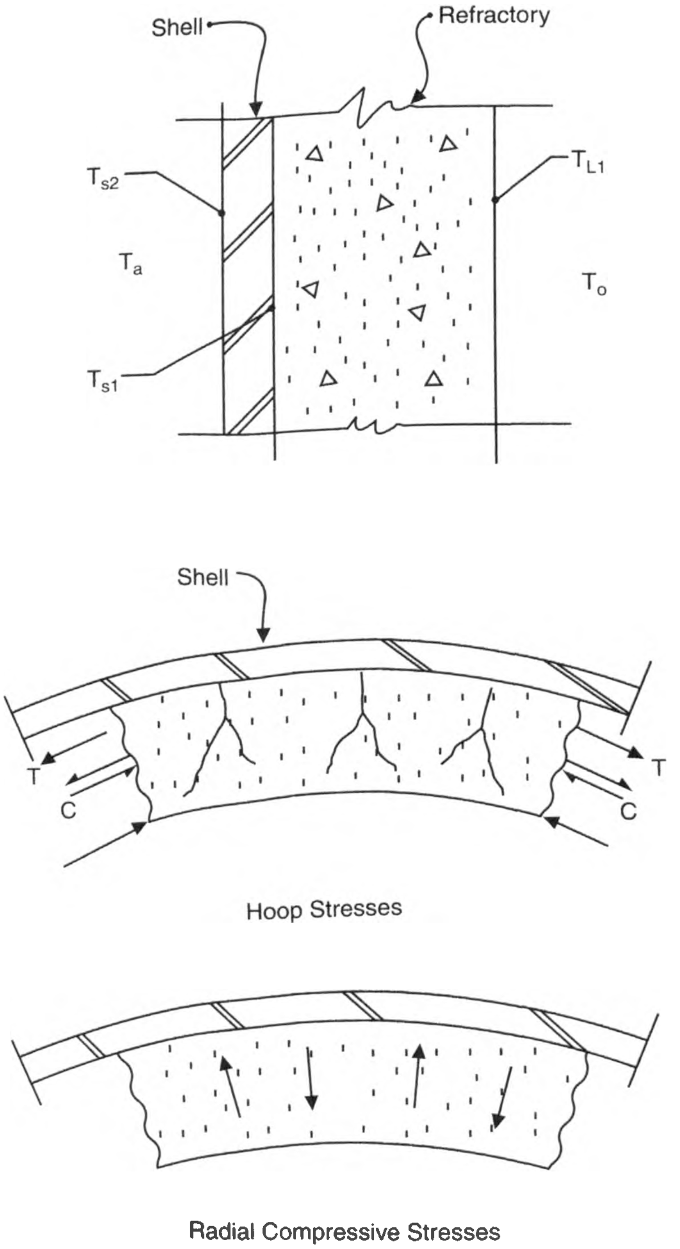


Figure 6-14. Stress/temperatures in wall.

Calculations

Properties of Vessel or Pipe

- *Equivalent specific density, w_{eq}.*

$$w_{eq} = w_s + w_L\left(\frac{D_L^2 - d_L^2}{D_s^2 - D_L^2}\right)$$

- *Moment of inertia.*

$$\text{Steel:} \quad I_s = \frac{\pi}{64}\left(D_s^4 - D_L^4\right)$$

$$\text{Refractory:} \quad I_L = \frac{\pi}{64}\left(D_L^4 - d_L^4\right)$$

$$\text{Composite:} \quad I = I_s + I_L$$

- *Equivalent modulus of elasticity, E_{eq}.*

$$E_{eq} = E_s + \frac{E_L I_L}{I_s}$$

Temperatures

- *Heat loss through wall, Q.*

$$Q = \frac{T_o - T_a}{\frac{1}{h_i} + \frac{t_L}{K_L} + \frac{t_s}{K_s} + \frac{1}{h_o}} \tag{1}$$

- *Outside shell temperature, T_{s1}.*

$$T_{s1} = T_a + Q\left(\frac{1}{h_o}\right) \tag{2}$$

- *Inside shell temperatures, T_{s2}.*

$$T_{s2} = T_{s1} + Q\left(\frac{t_s}{K_s}\right) \tag{3}$$

- *Inside lining temperature, T_{L1}.*

$$T_{L1} = T_{s2} + Q\left(\frac{t_L}{K_L}\right) \tag{4}$$

- *Verification of temperature gradient.*

$$T_o = T_{L1} + Q\left(\frac{1}{h_i}\right) \tag{5}$$

- *Mean shell temperature, T_s.*

$$T_s = 0.5(T_{s1} + T_{s2}) \tag{6}$$

- *Mean lining temperature, T_L.*

$$T_L = 0.5(T_{s1} + T_{L1}) \tag{7}$$

Stresses and Strain

- *Circumferential pressure stress, σ_ϕ.*

$$\sigma_\phi = \frac{PD}{2t_s} \tag{8}$$

- *Circumferential pressure strain, ε_ϕ.*

$$\varepsilon_\phi = \frac{0.85\sigma_\phi}{E_s} \tag{9}$$

Thermal Expansions

- *Thermal expansion of shell, $\Delta L1$.*

$$\Delta L1 = \alpha_s(T_s - T_c) \tag{10}$$

- *Total circumferential expansion without lining stress, $\Delta L2$.*

$$\Delta L2 = \varepsilon_\phi + \Delta L1 \tag{11}$$

- *Mean thermal expansion, $\Delta L3$.*

$$\Delta L3 = \alpha_L(T_L - T_c) \tag{12}$$

- *Mean shrinkage, $\Delta L4A$.*

$$\Delta L4 = 0.5(S_{TS} + S_{TL}) \tag{13}$$

- *Net mean unrestrained expansion, $\Delta L5$.*

$$\Delta L5 = \Delta L3 - \Delta L4 \tag{14}$$

- *Net differential circumferential expansion, $\Delta L6$.*

$$\Delta L6 = \Delta L2 - \Delta L5 \tag{15}$$

Stresses

- *Mean compressive stress in lining due to restraint of shell, σ_{L1}.*

$$\sigma_{L1} = E_L \Delta L6 \left(\frac{E_s t_s}{E_L t_L + E_s t_s}\right) \tag{16}$$

- *Differential stress from mean at hot face and cold face of lining due to thermal expansion, σ_{L2}.*

$$\sigma_{L2} = \frac{(E_L \alpha_L)(T_{L1} - T_{s2})}{2(1 - \mu_L)} \tag{17}$$

- *Differential stress from mean at hot and cold faces of lining due to shrinkage, σ_{L3}.*

$$\sigma_{L3} = \frac{E_L(S_{TL} - S_{TS})}{2(1 - \mu_L)} \tag{18}$$

- *Circumferential stress in lining at hot face, σ_{L4}.*

$$\sigma_{L4} = \sigma_{L1} - \sigma_{L2} + \sigma_{L3} \tag{19}$$

- *Circumferential stress in lining at cold face, σ_{L5}.*

$$\sigma_{L5} = \sigma_{L1} + \sigma_{L2} - \sigma_{L3} \tag{20}$$

- *Circumferential stress in shell caused by lining, σ_{sc}.*

$$\sigma_{sc} = -\sigma_{L1}\left(\frac{t_L}{t_s}\right) \tag{21}$$

Stress and Deflection Due to External loads

- *Uniform load, w.*

$$w = \frac{W}{L}$$

- *Deflection due to dead weight alone, δ.*

$$\delta = \frac{5wL^4}{384E_{eq}I}$$

- *Deflection due to concentrated load, δ.*

$$X = \frac{L_1}{L}$$

$$\delta = \frac{FL_1^3}{3E_{eq}I}\left(\frac{3 - X}{2X}\right)$$

Table 6-5
Properties of refractory materials

Properties	At Temperature (°F)	Material				
		AA-22S	RS-3	RS-6	RS-7	RS-17EC
Modulus of elasticity, E (10^6 psi)	230	47	2.1	4.1	3.7	18.9
	500	35	1.5	2.94	2.7	16.8
	1000	16.5	0.84	1.62	1.5	15.5
	1500	7.9	0.5	0.93	0.8	14.1
Density, d (pcf)		170	60	75–85	85–95	130–135
Thermal conductivity, K (BTU-in/hr/ft^2/°F)	500		1.7	2.7	2.5	10
	1000	10.3	1.65	2.85	2.8	6.3
	1500	10.4	1.8	3	3.2	6.9
	2000	10.6		3.2	2	7.7
Coefficient of thermal expansion (10^{-6}in./in./°F)		4.7	4.4		4.7	3.5
Poisson's ratio						0.16
Specific heat (BTU/lb/°F)						0.24
% Permanent linear change	1500	−0.1 TO −0.5	−0.3 TO −0.7	−0.1 TO −0.3	−0.2 TO −0.4	−0.1 TO −0.3
	2000	−0.4 TO −1.1	−0.5 TO −1.1	−0.8 TO −1.2	−0.4 TO −0.6	−0.1 TO −0.3
Modulus of rupture (psi)	1000	1400	100	200	200–300	1500–1900
	1500	1400–2200	100–200	200–500	300–700	1400–1800
	2000		150–250	200–500	200–500	
Cold crush strength (psi)	1000	8000–12000	300	1500	600–1000	9000–12000
	1500	7500–10000	300–600	1500–1800	700–1100	8000–11000
	2000	7000–10000	500–800	1200–1600	600–1000	9000–12000
Allowable compressive stress (psi)	1000	4000	150	750	400	5000
Allowable tensile stress (psi)	1000	560	40	80	100	680

Table 6-6
Given input for sample problems

Shell Properties			Refractory Properties		
Item	Case 1	Case 2	Item	Case 1	Case 2
D	360 in.	374 in.	t_L	4 in.	4 in.
t_s	0.5 in.	1.125 in.	E_L	0.6×10^6	0.8×10^6
E_s	28.5×10^6	27.7×10^6	α_L	4.0×10^{-6}	4.7×10^{-6}
α_s	6.8×10^{-6}	7.07×10^{-6}	K_L	4.4	3.2
K_s	300	331.2	μ_L	0.25	0.2
μ_s	0.3	0.3	σ_{ult}	2000 psi	100 psi
T_a	80°F	−20°F	S_{TS}	0.00028	0.002
T_c	60°F	50°F	S_{TL}	0.00108	−0.00025
T_o	1100°F	1400°F	h_i	40	40
P	12 PSIG	25 PSIG	h_o	4	3.5

Table 6-7
Summary of results for sample problems

Equation	Variable	Case 1	Case 2	Equation	Variable	Case 1	Case 2
1	Q	860	908	12	ΔL_3	2.512×10^{-3}	3.53×10^{-3}
2	T_{s1}	295	239	13	ΔL_4	6.8×10^{-4}	1.75×10^{-3}
3	T_{s2}	296	242	14	ΔL_5	1.832×10^{-3}	-4.7×10^{-4}
4	T_{L1}	1079	1377	15	ΔL_6	-1.02×10^{-4}	1.88×10^{-3}
5	T_o	1100	1400	16	σ_{L1}	−52.4 psi	148.9 psi
6	T_s	296	241	17	σ_{L2}	+/−1251 psi	+/−266 psi
7	T_L	688	810	18	σ_{L3}	+/−320 psi	−112.5 psi
8	σ_ϕ	4320	4155	19	σ_{L4}	−983 psi	−229.6 psi
9	ε_ϕ	1.29×10^{-4}	1.275×10^{-4}	20	σ_{L5}	879 psi	427.5 psi
10	ΔL_1	1.6×10^{-3}	1.28×10^{-3}	21	σ_{sc}	419 psi	−530 psi
11	ΔL_2	1.73×10^{-3}	1.41×10^{-3}				

Procedure 6-5: Vibration of Tall Towers and Stacks [13–23]

Tall cylindrical stacks and towers may be susceptible to wind-induced oscillations as a result of vortex shedding. This phenomenon, often referred to as *dynamic instability*, has resulted in severe oscillations, excessive deflections, structural damage, and even failure. Once it has been determined that a vessel is dynamically unstable, either the vessel must be redesigned to withstand the effects of wind-induced oscillations or external spoilers must be added to ensure that vortex shedding does not occur.

The deflections resulting from vortex shedding are perpendicular to the direction of wind flow and occur at relatively low wind velocities. When the natural period of vibration of a stack or column coincides with the frequency of vortex shedding, the amplitude of vibration is greatly magnified. The frequency of vortex shedding is related to wind velocity and vessel diameter. The wind velocity at which the frequency of vortex shedding matches the natural period of vibration is called the *critical wind velocity.*

Wind-induced oscillations occur at steady, moderate wind velocities of 20–25 miles per hour. These oscillations commence as the frequency of vortex shedding approaches the natural period of the stack or column and are perpendicular to the prevailing wind. Larger wind velocities contain high-velocity random gusts that reduce the tendency for vortex shedding in a regular periodic manner.

A convenient method of relating to the phenomenon of wind excitation is to equate it to fluid flow around a cylinder. In fact this is the exact case of early discoveries related to submarine periscopes vibrating wildly at certain speeds. At low flow rates, the flow around the cylinder is laminar. As the stream velocity increases, two symmetrical eddies are formed on either side of the cylinder. At higher velocities vortices begin to break off from the main stream, resulting in an imbalance in forces exerted from the split stream. The discharging vortex imparts a fluctuating force that can cause movement in the vessel perpendicular to the direction of the stream.

Historically, vessels have tended to have many fewer incidents of wind-induced vibration than stacks. There is a variety of reasons for this:

1. Relatively thicker walls.
2. Higher first frequency.
3. External attachments, such as ladders, platforms, and piping, that disrupt the wind flow around the vessel.
4. Significantly higher damping due to:
   a. Internal attachments, trays, baffles, etc.
   b. External attachments, ladders, platforms, and piping.
   c. Liquid holdup and sloshing.
   d. Soil.
   e. Foundation.
   f. Shell material.
   g. External insulation.

Damping Mechanisms

Internal linings are also significant for damping vibration; however, most tall, slender columns are not lined, whereas many stacks are. The lining referred to here would be the refractory type of linings, not paint, cladding, or some protective metal coating. It is the damping effect of the concrete that is significant.

Damping is the rate at which material absorbs energy under a cyclical load. The energy is dissipated as heat from internal damping within the system. These energy losses are due to the combined resistances from all of the design features mentioned, i.e., the vessel, contents, foundation, internals, and externals. The combined resistances are known as the *damping factor*.

The total damping factor is a sum of all the individual damping factors. The damping factor is also known by other terms and expressions in the various literature and equations and expressed as a coefficient. Other common terms for the damping factor are *damping coefficient, structural damping coefficient, percent critical damping*, and *material damping ratio.* In this procedure this term is always referred to either as factor D_F or as β.

There are eight potential types of damping that affect a structure's response to vibration. They are divided into three major groups:

Resistance:

Damping from internal attachments, such as trays.
Damping from external attachments, such as ladders, platforms, and installed piping.
Sloshing of internal liquid.

Base support:

Soil.
Foundation.

Energy absorbed by the shell (hysteretic):

Material of shell.
Insulation.
Internal lining.

Karamchandani, Gupta, and Pattabiraman give a detailed account of each of these damping mechanisms for process towers (trayed columns). They estimate the "percent critical damping" at 3% for empty vessels and 5% for operating conditions. The value actually used by most codes is only a fraction of this value.

Design Criteria

Once a vessel has been designed statically, it is necessary to determine if the vessel is susceptible to wind-induced vibration. Historically, the rule of thumb was to do a dynamic wind check only if the vessel L/D ratio exceeded 15 and the POV was greater than 0.4 seconds. This criterion has proven to be unconservative for a number of applications. In addition, if the critical wind velocity, V_c, is greater than 50 mph, then no further investigation is required. Wind speeds in excess of 50 mph always contain gusts that will disrupt uniform vortex shedding.

This criterion was amplified by Zorrilla, who gave additional sets of criteria. Criterion 1 determines if an analysis should be performed. Criterion 2 determines if the vessel is to be considered stable or unstable. Criterion 3 involves parameters for the first two criteria.

Criterion 1

- If $W/LD_r^2 \leq 20$, a vibration analysis must be performed.
- If $20 < W/LD_r^2 \leq 25$, a vibration analysis should be performed.
- If $W/LD_r^2 > 25$, a vibration analysis need not be performed.

Criterion 2

- If $W\delta/LD_r^2 \leq 0.75$, the vessel is unstable.
- If $0.75 < W\delta/LD_r^2 \leq 0.95$, the vessel is probably unstable.
- If $W\delta/LD_r^2 > 0.95$, the structure is stable.

Criterion 3

This criterion must be met for Criteria 1 and 2 to be valid.

- $L_C/L < 0.5$
- $10{,}000\ D_r < 8$
- $W/W_s < 6$
- $V_c > 50$ mph; vessel is stable and further analysis need not be performed.

Criterion 4

An alternative criterion is given in ASME STS-1-2011, "Steel Stacks". This standard is written specifically for stacks. The criterion listed in this standard calculates a "critical vortex shedding velocity", V_{zcrit}. This value is then compared to the critical wind speed, V_c, and a decision made.

- If $V_c < V_{zcrit}$, vortex shedding loads shall be calculated.
- If $V_{zcrit} < V_c < 1.2\ V_{zcrit}$, vortex shedding loads shall be calculated; however, the loads may be reduced by a factor of $(V_{zcrit}/V_c)^2$.
- If $V_c > 1.2\ V_{zcrit}$, vortex shedding may be ignored.

Equations are given for calculating all of the associated loads and forces for the analysis. This procedure utilizes the combination of two components of β, one β for aerodynamic damping, β_a, and one for steel damping, β_s. The two values are combined to determine the overall β.

Criterion 5

An alternative criterion is also given in the Canadian Building Code, NBC. The procedure for evaluating effects of vortex shedding can be approximated by a static force acting over the top third of the vessel or stack. An equation is given for this value, F_L, and shown in this procedure.

Dynamic Analysis

If the vessel is determined by this criterion to be unstable, then there are two options:

a. The vessel must be redesigned to withstand the effects of wind-induced vibration such that dynamic deflection is less than 6 in./100 ft of height.
b. Design modifications must be implemented such that wind-induced oscillations do not occur.

Design Modifications

The following design modifications may be made to the vessel to eliminate vortex shedding:

a. Add thickness to bottom shell courses and skirt to increase stiffness and raise the natural frequency.
b. Modify the top diameter where possible.
c. For stacks, add helical strakes to the top third of the stack only as a last resort. Spoilers or strakes should protrude beyond the stack diameter by a distance of d/12 but not less than 2 in.
d. Cross-brace vessels together.
e. Add guy cables or wires to grade.
f. Add internal linings.
g. Reduce vessel below dynamic criteria.

Precautions

The following precautions should be taken.

a. Include ladders, platforms, and piping in your calculations to more accurately determine the natural frequency.
b. Grout the vessel base as soon as possible after erection while it is most susceptible to wind vibration.
c. Add external attachments as soon as possible after erection to break up vortices.
d. Ensure that tower anchor bolts are tightened as soon as possible after erection.

Definitions

Critical wind velocity: The velocity at which the frequency of vortex shedding matches one of the normal modes of vibration.

Logarithmic decrement: A measure of the ability of the overall structure (vessel, foundation, insulation, contents, soil, lining, and internal and external attachments) to dissipate energy during vibration. The logarithmic ratio of two successive amplitudes of a damped, freely vibrating structure or the percentage decay per cycle.

Static deflection: Deflection due to wind or earthquake in the direction of load.

Dynamic deflection: Deflection due to vortex shedding perpendicular to the direction of the wind.

Notes

1. See procedure 3-3 to determine a vessel's fundamental period of vibration (POV).
2. See procedure 3-4 to determine static deflection.
3. Vessel should be checked in the empty and operating conditions with the vessel fully corroded.

4. Concentrated eccentric loads can be converted to an additional equivalent uniform wind load.
5. L/D ratios for multidiameter columns can be determined as shown in Note 8.
6. A fatigue evaluation should be performed for any vessel susceptible to vortex shedding. A vessel with a POV of 1 second and subjected to 3 hours per day for 30 years would experience 120 million cycles.
7. This procedure is for cylindrical stacks or vessels only, mounted at grade. It is not appropriate for tapered stacks or vessels. There is a detailed procedure in ASME STS-1 for tapered stacks. Multidiameter columns and stacks can be evaluated by the methods shown. This procedure also does not account for multiple vessels or stacks in a row.
8. L/D ratios can be approximated as follows:

$$\frac{L_1D_1 + L_2D_2 + \ldots + L_xD_x + L_{sk}D_{sk}}{D_x^2}$$

where quantities L_xD_x are calculated from the top down.

Table 6-8
Summary of critical damping

		Case 1: Empty			Case 2: Operating		
			β			β	
Item	Description	δ		%	δ		%
1	Material	0.07	0.011	1.1	0.07	0.011	1.1
2	Insulation	0.0063	0.001	0.1	0.0063	0.001	0.1
3	Soil	0.125	0.02	2	0.125	0.02	2
4	Attachments	0.0063	0.001	0.1	0.0063	0.001	0.1
5	Liquid				0.094	0.015	1.5
	Total	0.208	0.033	3.34	0.302	0.048	4.84

Sources: Ref. 13.

Table 6-9
Logarithmic decrement, δ

		Soil Type		
Type	Description	Soft (1)	Medium (2)	Rock/Piles (3)
1	Steel vessel	0.1	0.05	0.03
2	Tower with internals	0.13	0.08	0.035–0.05
3	Tower internals and operations	0.1	0.05	0.035
4	Tower, refractory lined	0.3	0.1	0.04–0.05
5	Tower, full of water	0.3	0.1	0.07
6	Unlined stack	0.1	0.05	0.035
7	Lined stack	0.3	0.1	0.07

Notes:

1. Soft soils $B_p < 1500$ psi, $\beta_F = 0.07$.
2. Medium soils, 1500 psi $< B_p <$ 3000 psi, $\beta_F = 0.03$.
3. Pile foundation, rock, or stiff soils, $\beta_F = 0.005$.

Table 6-10
Values of β

	Standard						
Soil Type	ASCE 7-95	Major Oil Co	ASME STS-1	NBC	Misc. Papers	Gupta	Compress
Soft Medium Rock/Piles	0.005 0.01 0.005	See Table 6-11; 0.004-0.0127	See Note 1 and Table 6-12	Unlined = 0.0016–0.008 Lined = 0.0048–0.0095 See Note 2	See Note 3	See Table 6-8; 0.03–0.05	Default = 2% (0.02)

Table 6-11
β Values per a major oil company

Equipment Description	β
Vessels:	
1. Empty without internals	0.0048
2. Empty with tray spacing > 5 ft.	0.0051
3. Empty with tray spacing 3–5 ft	0.0056
4. Empty with tray spacing < 3 ft	0.0064
5. Operating with tray spacing 5–8 ft	0.0116
6. Operating with tray spacing < 5 ft	0.0127
7. Vessel full of liquid	0.018
Stacks mounted at grade	0.004–0.008

Table 6-12
Values of β_s per ASME STS-1

	Damping Value	
Type of Stack	Rigid Support	Elastic Support
Unlined	0.002	0.004
Lined	0.003	0.006

Notes

1.

$$\beta = \beta_a + \beta_s$$

$$\beta_a = \frac{C_f \cdot \rho \cdot D_r \cdot V_z}{4 \cdot \pi \cdot w_r \cdot f_1}$$

$$\beta_s = \text{from table}$$

2. For lined and unlined stacks only!

$$\beta = \frac{\delta}{2\pi}$$

3.

$$\beta = \frac{w\delta}{D^2} \text{ or } \frac{W\delta}{LD^2}$$

Table 6-13
Coefficient C_f per ASME STS-1

		L/D		
	Surface Texture	1	7	25
$D(q_z)^{0.5} > 2.5$	Smooth	0.5	0.6	0.7
	Rough	0.7	0.8	0.9
	Very rough	0.8	1	1.2
$D(q_z)^{0.5} > 2.5$	All	0.7	0.8	1.2

Table 6-14
Topographic factors per ASME STS-1

Exposure Category	b	α
A	0.64	0.333
B	0.84	0.25
C	1	0.15
D	1.07	0.111

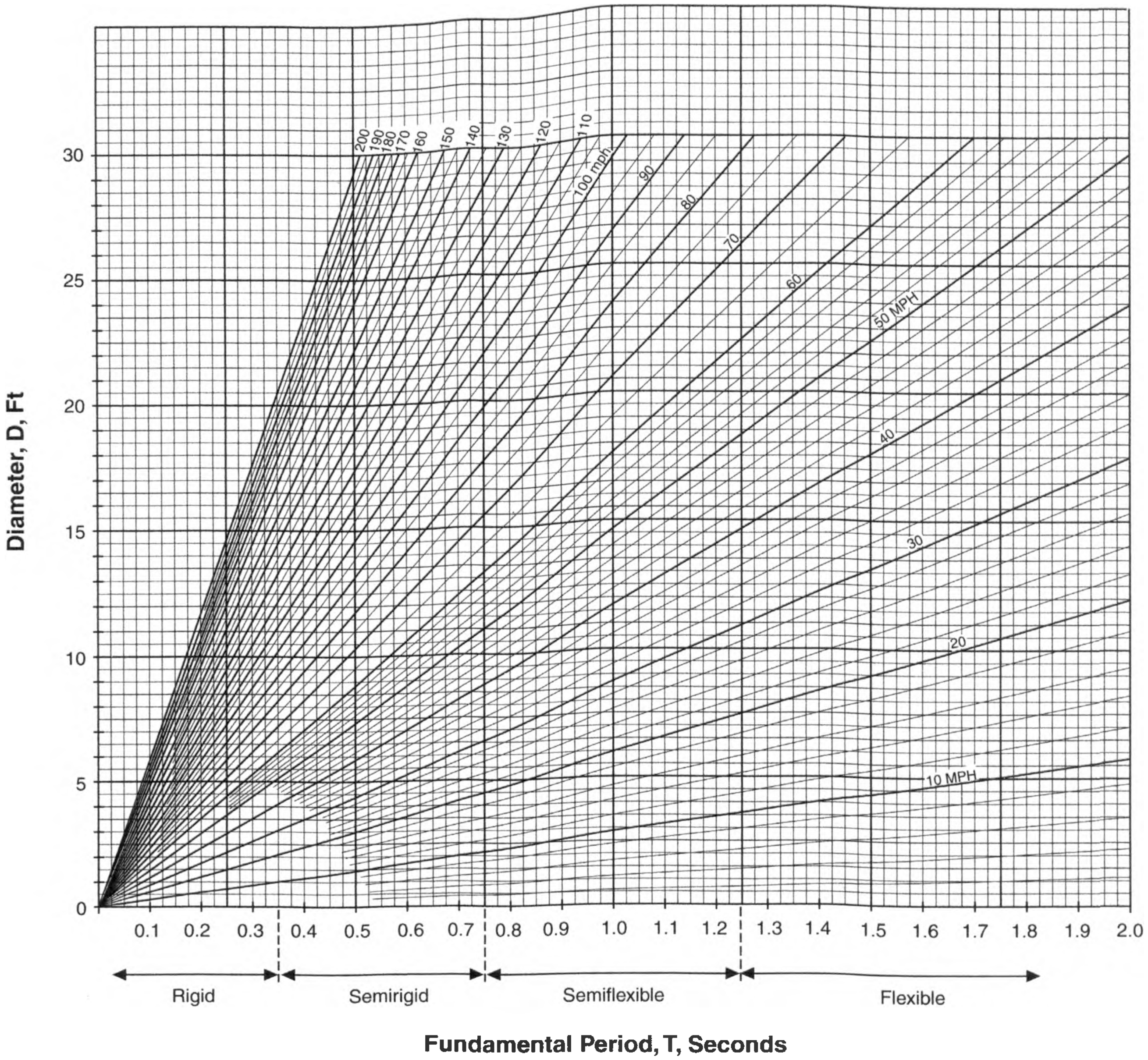


Figure 6-15. Graph of critical wind velocity, V_c.

Notation

B_f = allowable soil bearing pressure, psf
C_f = wind force coefficient, from table
C_1, C_2 = NBC coefficients
D = mean vessel diameter, in.
D_r = average diameter of top third of vessel, ft
E = modulus of elasticity, psi
F_F = fictitious lateral load applied at top tangent line, lb
F_L = equivalent static force acting on top third of vessel or stack, lb
f = fundamental frequency of vibration, Hz (cycles per second)
f_n = frequency of mode n, Hz
f_o = frequency of ovalling of unlined stack, Hz
g = acceleration due to gravity, 386 in./sec^2 or 32 ft/sec^2
I = moment of inertia, shell, in.^{4}
I_F = importance factor, 1.0–1.5
L = overall length of vessel, ft
L_c = Length of cone, ft
M_L = overturning moment due to force F_L, ft-lb
M_S = overturning moment due to seismic, ft-lb
M_R = resultant moment, ft-lb
M_w = overturning moment due to wind, ft-lb
M_{wD} = modified wind moment, ft-lb

q_H = wind velocity pressure, psf, per NBC
q_z = external wind pressure, psf per ASME STS-1
S = Strouhal number, use 0.2
T = period of vibration, sec
t = shell thickness, in.
V = basic wind speed, mph
V_c = critical wind velocity, mph
V_{c1}, V_{c2} = critical wind speeds for modes 1 and 2, mph or fps
V_{co} = critical wind speed for ovalling of stacks, ft/sec
V_r = reference design wind speed, mph, per ASME STS-1
V_z = mean hourly wind speed, ft/sec
V_{zcrit} = mean hourly wind speed at ⅚ L, ft/sec
W = overall weight of vessel, lbs
w = uniform weight of vessel, lb/ft
w_r = uniform weight of top third of vessel, lb/ft
W_s = Weight of base vessel, lbs
α, b = topographic factors per ASME STS-1
β = percent critical damping, damping factor
β_a = aerodynamic damping value
β_f = foundation damping value
β_s = structural damping value
δ = logarithmic decrement
Δ_d = dynamic deflection, perpendicular to direction of wind, in.
Δ_s = static deflection, parallel to direction of wind, in.
ρ = density of air, lb/ft^3 (0.0803) or kg/m^3 (1.2)
λ = aspect ratio, L/D

Miscellaneous Equations

- *Frequency for first three modes, f_n.*

Mode 1: $$f_1 = 0.56\sqrt{\frac{gEI}{wL^4}}$$

Mode 2: $$f_2 = 3.51\sqrt{\frac{gEI}{wL^4}}$$

Mode 3: $$f_3 = 9.82\sqrt{\frac{gEI}{wL^4}}$$

Note: I is in ft^4.

$$I = 0.032D^3t$$

$$f_n = \frac{1}{T}$$

- *Frequency for ovalling, f_o.*

$$f_o = \frac{680t}{D^2}$$

- *Critical wind velocities*:

$$V_c = V_{c1} = \frac{f_1 D}{S} = \frac{D}{ST} = \frac{D}{0.2T}(\text{fps})$$

$$V_c = \frac{3.4D}{T}(\text{mph})$$

$$V_{c2} = 6.25V_{c1}$$

$$V_{co} = \frac{f_o D}{2S}$$

- *Period of vibration, T, for tall columns and stacks.*

$$T = 1.79\sqrt{\frac{wL^4}{EIg}}$$

where L, D, and t are in feet.

Procedures

Procedure 1: Zorilla Method

Step 1: Calculate structural damping coefficient, β.

$$\beta = \frac{W\delta}{LD_r^2} \quad \text{or} \quad \beta = \frac{w\delta}{D_r^2}$$

Step 2: Evaluate:

- If $W/LD_r^2 \le 20$, a vibration analysis must be performed.
- If $20 < W/LD_r^2 \le 25$, a vibration analysis should be performed.
- If $W/LD_r^2 > 25$, a vibration analysis need not be performed.
- If $W\delta/LD_r^2 \le 0.75$, the vessel is unstable.
- If $0.75 < W\delta/LD_r^2 \le 0.95$, the vessel is probably unstable.

- If $W\delta/LD_r^2 > 0.95$, the structure is stable.

Step 3: If $\beta < 0.95$, check critical wind velocity, V_c.

$$V_c = \frac{D_r}{TS} = \text{fps}$$

$$V_c = \frac{3.41D_r}{T} = \text{mph}$$

If $V_c > V$, then instability is expected.

Step 4: Calculate dynamic deflection, Δ_d.

$$\Delta_d = \frac{(2.43)(10^{-9})L^5V_c^2}{W\delta D_r}$$

If $\Delta_d < 6$ in./100 ft, then the design is acceptable as is. If $\Delta_d > 6$ in./100 ft, then a "design modification" is required.

Procedure 2: ASME STS-1 Method

Step 1: Calculate damping factor, β.

$$\beta = \beta_a + \beta_S$$

Step 2: Calculate critical wind speed, V_c.

Step 3: Calculate critical vortex shedding velocity, V_{zcrit}.

$$V_{zcrit} = b\left(\frac{Z_{cr}}{33}\right)^{\alpha} \frac{22}{15}(V_r)$$

where

$$Z_{cr} = \frac{5L}{6}$$

$$V_r = \frac{V}{I_f}$$

b and α are from table.

Step 4: Evaluate:

- If $V_c < V_{zcrit}$, then vortex shedding loads shall be calculated.
- If $V_{zcrit} < V < 1.2\ V_{zcrit}$, then vortex shedding loads shall be calculated; however, loads may be reduced by a factor of $(V_{zcrit}/V_c)^2$.
- If $V_c > 1.2\ V_{zcrit}$, then vortex shedding may be ignored.

Step 5: To evaluate vortex shedding loads, refer to ASME STS-1, Appendix E.

Procedure 3: NBC

Step 1: Calculate critical wind velocity, V_c. No analysis need be performed if $V_c > V$.

Step 2: Calculate coefficients C_1 and C_2.

- If $\lambda > 16$, then

$$C_1 = 3 \text{ and } C_2 = 0.6$$

- If $\lambda < 16$, then

$$C_1 = \frac{3\sqrt{\lambda}}{4}$$

- If $V_c < 22.37$ mph and $\lambda > 12$, then

$$C_1 = 6 \text{ and } C_2 = 1.2$$

Step 3: If

$$\beta > \frac{C_2\rho D_r^2}{w_r}$$ then no dynamic analysis need be performed.

If

$$\beta < \frac{C_2\rho D_r^2}{w_r}$$ then dynamic analysis should be performed.

Step 4: If a dynamic analysis is required, calculate an equivalent static force to be applied over the top third of the column, F_L.

$$F_L = \frac{C_1 q_H D_r}{\sqrt{\lambda}\sqrt{\dfrac{C_2\rho D_r^2}{w_r}}}$$

Step 5: Determine moment due to force, F_L.

$$M_L = \frac{5F_L L^2}{18}$$

Step 6: Calculate modified wind moment, M_{WD}.

$$M_{WD} = M_W\left(\frac{V_C}{V_W}\right)^2$$

Step 7: Calculate resultant moment, M_R.

$$M_R = \sqrt{M_L^2 + M_{WD}^2}$$

Step 8: If $M_R > M_S$ or M_W, then compute fictitious force, F_F.

$$F_F = \frac{M_R}{L}$$

Step 9: Check vessel with lateral load, F_F, applied at the top tangent line of the vessel. If the stresses are acceptable, the vessel is OK. If the stresses are not acceptable, then the thicknesses must be revised until the stresses are acceptable.

Example No. 1

Given

$w = 146.5$ kips

$T = 0.952$ sec

$S = 0.2$

$\delta = 0.08$

$$D_r = \frac{10 + 6.5}{2} = 8.25 \text{ ft}$$

Soil type: medium.

- *Average weight of top third of column.*

$$\frac{L}{3} = \frac{198}{3} = 66 \text{ ft}$$

$$\frac{W_t}{66} = \frac{35,000}{66} = 530 \text{ lb/ft}$$

- *Dynamic check.*

$$\frac{W}{LD_r^2} = \frac{146,500}{198(8.25^2)} = 10.87 < 20$$

Therefore an analysis must be performed.

$$\beta = \frac{W\delta}{LD_r^2} = 0.08(10.87) = 0.87$$

Probably stable, proceed.

- *Critical wind speed, V_c.*

$$V_c = \frac{D_r}{TS} = \frac{8.25}{0.952(0.2)} = 43.33 \text{ fps}$$

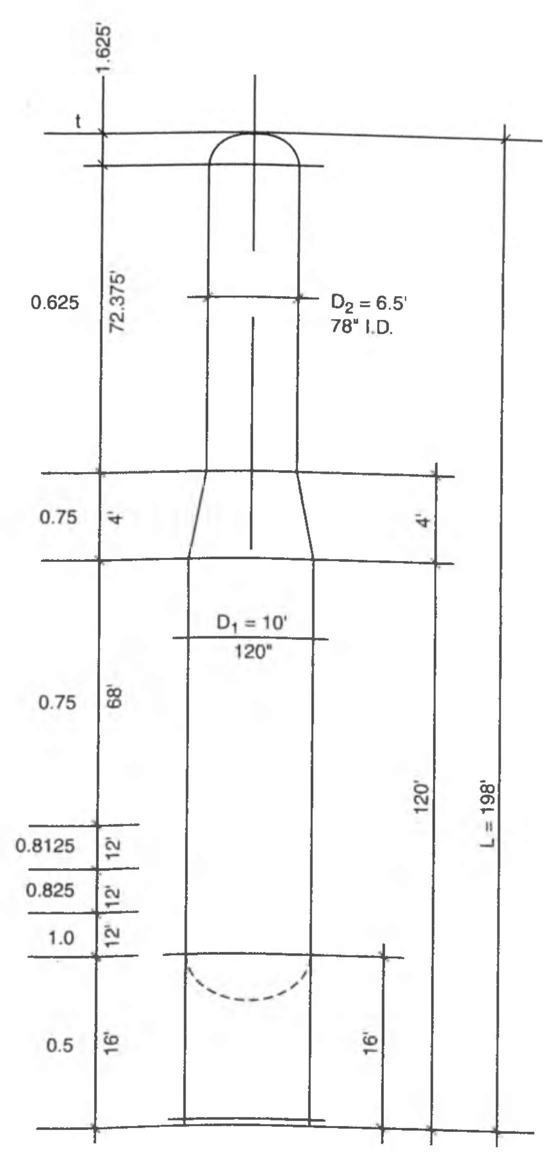
43.33 fps(0.682) = 29.55 mph

- *Dynamic deflection, Δ_d.*

$$\Delta_d = \frac{(2.43)\,(10^{-9})\,L^5 V_c^2}{W\delta D_r}$$

$$\Delta_d = \frac{(2.43)\,(10^{-9})\,198^5\,(29.55^2)}{146,500\,(0.08)\,8.25} = 6.68 \text{ in.}$$

Dimensions



Example No. 1: Wind Design, Static Deflection, 100 MPH Zone

F_n	A_f	P	H
17,405	395.2	44	160
17,534	416	42.15	120
11,012	280	39.33	100
10,550	280	37.68	80
10,088	280	36.03	60
9430	280	33.68	40
4319	140	30.85	30
2028	70	28.97	25
1962	70	28.03	20
1835	70	26.21	15
4495	180	24.97	0

VESSEL

	H	R_m	t	I
I_7	124	39.313	0.625	119,295
I_6	120	60.375	0.75	518,540
I_5	52	60.375	0.75	518,540
I_4	40	60.4	0.8125	562,450
I_3	28	60.438	0.875	608,844
I_2	16	60.5	1	695,690
I_1	0	60.25	0.5	348,551

Example No. 1: Values for computation of static deflection

Section n	L_n (ft)	L_n (in.)	I_n	L_n^4/I_n	L_n^4/I_{n-1}
1	198	2376	343,551	92,767,217	
2	182	2184	695,690	32,703,540	66,224,596
3	170	2040	606,844	28,539,319	24,894,586
4	158	1896	562,450	22,975,735	21,294,932
5	146	1752	518,540	18,169,967	16,751,453
6	78	936	518,540	1,480,202	1,480,202
7	74	888	119,295	5,212,302	1,199,139
Σ				201,848,283	131,844,908

- Static deflection due to wind, Δ_s.

$$\Delta_s = \left(\sum \frac{L_n^4}{I_n} - \sum \frac{L_n^4}{I_n - 1}\right)\left(\frac{w_{min}}{8E} + 5.5\frac{w_{max} - w_{min}}{60E}\right)$$

$$\Delta_s = (70{,}003{,}375)\left[1.143\left(10^{-7}\right) + 4.44\left(10^{-8}\right)\right] = 11.11 \text{ in.} < 6 \text{ in.}/100 \text{ ft}$$

$$w_{min} = \frac{F_n}{L_n} = \frac{4495}{15} = 300 \text{ lb/ft} = 24.97 \text{lb/in.} \qquad E = 27.3\left(10^6\right)\text{psi}$$

$$w_{max} = \frac{17{,}405}{38} = 458 \text{ lb/ft} = 38.2 \text{ lb/in.}$$

Procedure 6-6: Underground Tanks & Vessels

Underground storage tanks or vessels can be buried, or mounted in below ground pits, usually called "sumps". In either case, the design principles are the same. The vessel is subject to flotation if the pit fills with water or the groundwater encroaches on a buried tank. The designer must be sure that the anchorage is strong enough and heavy enough to resist the flotation forces.

The buoyancy of the tank is produced when groundwater encroaches beneath the tank and causes the tank to float. The pressure of flotation (buoyancy) is determined by the displacement of the tank multiplied times the equivalent weight of water of the displacement. This is the same mechanism that allows ships and boats to float. The weight of displacement is greater than the weight of the ship or boat.

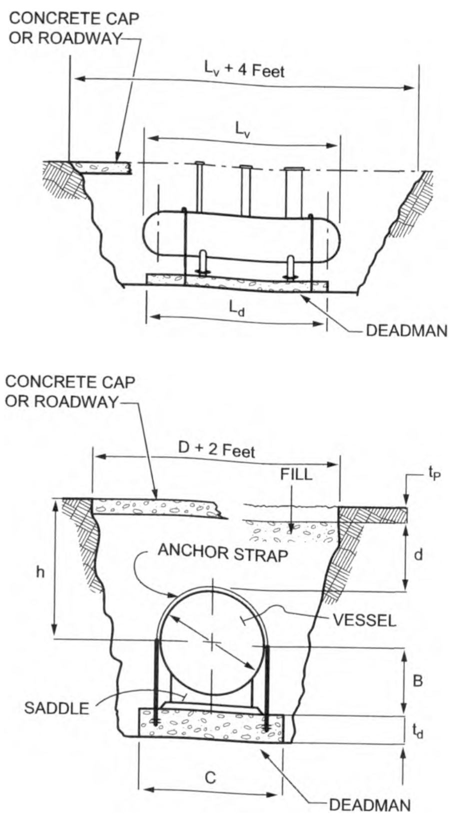
The force of flotation is resisted by the weight of the vessel, the overburden (fill), the weight of a deadman (if used) and the weight of any paving over the top of the tank. The vessel itself must be designed to resist the external pressure created by the weight of the overburden. ASME Code, Section VIII, Division 1 method for external pressure should be used for analyzing the shell for the external pressure condition.

This procedure can be used to determine either the depth of burial or the weight of the deadman required to prevent flotation. Underground storage tanks (UST) may be either anchored, or unanchored. If anchored, the anchorage must be sized to resist the buoyancy force. If unanchored, the vessel must be buried deep enough such that the weight of the overburden is greater than the buoyancy force.

Underground storage tanks for gas stations are typically not anchored and do not have foundations. They are simply buried in the ground, resting on a layer of sand. The depth of the overburden over the tank constitutes the greatest force for counteracting flotation. The burial depth of the tank must also allow piping to be sloped to the tank at least 1/8 inch per foot.

Nomenclature

A_r = Area required, in^2
A_S = Area of anchor strap, in^2
A_1 = Area of concrete slab or paving, Ft^2
A_2 = Projected vessel area, Ft^2
A_3 = Area of deadman, Ft^2
A_S = Cross sectional area of strap, in^2
D = Vessel diameter, Ft or in
F = Buoyancy force, Lbs
FS = Factor of Safety
F_T = Allowable tensile stress, PSI
f_T = Stress, tension, PSI
Q = Operating load on one saddle, Lbs
t = Thickness of vessel shell, in
t_d = Thickness of deadman, Ft
t_P = Thickness of concrete pad, Ft
T = Tension force in anchor strap, Lbs
U = Net uplift, Lbs
V_n = Void area in fill for nozzles, Ft^3
V_f = Volume of fill for design, Ft^3
V_{fr} = Volume of fill required, Ft^3
V_V = Volume of vessel, Ft^3
w_C = Uniform weight of concrete, PCF
w_f = Uniform weight of fill, PCF
w_w = Uniform weight of water, PCF
W_{cp} = Weight of concrete slab. Lbs
W_d = Weight of deadman, lbs
W_f = Weight of fill, Lbs
W_{fr} = Weight of fill required, Lbs
W_O = Weight, operating, vessel, Lbs
W_V = Weight, empty, vessel, Lbs
ΣW = Total restraining force, Lbs
$\Sigma W'$ = Total resisting weight, Lbs



Note: Dimentsion "d" is to grade if no concrete pad is used on top of pit

Figure 6-16. Dimensions for buried vessel.

Table 6-15
Weight of materials

Material	Weight (PCF)
Sand & Pea Gravel	60
Reinforced Concrete	90-125
Water	62.4
Clay	40-100
Rock	60-120
Soil	75-85
Steel	497

Buried Vessels

Case 1: With Deadman

Determine the size and weight of the deadman required to counteract the buoyancy forces if required. Determine the size of anchor straps and bolts necessary to resist flotation.

1. Given;

   Volume, $V_V =$
   Weight, $W_V =$
   Uniform weight of fill, $w_f =$
   Diameter, $D =$
   Depth of overburden, $d =$
   Size of concrete slab =
   Thickness of concrete slab, $t_P =$
   Area of slab, $A_1 =$
   Projected tank area, $A_2 = D \times L_V$
   Void in overburden, $V_n =$
   $h = .5\,D + d =$

2. Calculate volume of fill, V_f

$$V_f = \left[(0.33\text{ h})\left(A_1 + A_2 + (A_1A_2)^{1/2}\right)\right] - \left[0.5V_V + V_n\right]$$

3. Calculate weights of fill, W_f

   $W_f = V_f\, w_f$

   and of concrete slab, if applicable

   $W_{cp} = w_c\, A_1\, t_p$

4. Sum of restraining force, $\sum W$

$$\sum W = W_f + W_V + W_{cp}$$

5. Total buoyant force, F

$$F = 62.4(V_V + V_n)$$

6. Determine if deadman is needed;

   If $\sum W > F$ No deadman is needed

   If $\sum W < F$ A deadman is needed

   Weight of deadman required;
   Note: Deadman should be at least 8" thk (.66Ft)

$$W_d = C(t_p)w_C(L_d)$$

7. Total resisting weight

$$\sum W' = \sum W + W_d > F$$

Case 2: No Deadman

Determine the depth of burial of a vessel or tank, from the top of the vessel to finished grade, necessary to prevent the tank from floating when empty. Unanchored tank.

- Assume a factor of safety, FS

Recommend a minimum FS of 1.25

Use: _______

- Find weight of fill required, W_{fr}

$$W_{fr} = FS\left(\sum W - W_V - W_{cp}\right)$$

- Find volume of fill required, V_f

$$V_{fr} = W_{fr}/w_f$$

- Find depth of fill required. Solve for h_r:

$$h_r = \left[V_{fr} + \left(0.5\ V_V + V_n\right)\right] \Big/ \left[0.33\left(A_1 + A_2 + (A_1A_2)^{1/2}\right)\right]$$

- Minimum acceptable burial depth, d

$$d = h_r - .5\ D - t_p$$

Anchor Strap

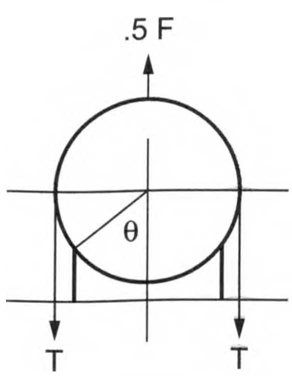


Figure 6-17. Loads on Anchor Strap (Assumes two straps are used.)

- Tension force in anchor strap, T

$$2T = 0.5\ F$$

- The strap size can be determined by either area or stress;

$$\text{Area; } A_r = T/F_T$$

$$\text{Stress; } f_T = T/A_S$$

Use: _______

External Pressure

- External pressure on tank due to fill (overburden), P_X

$$P_X = \left(W_f + W_{cp}\right)/144\ A_2$$

Note: Add P_X to any design external pressure condition and design per ASME Code requirements for the combined condition.

Saddle Design

The saddles must be designed for the operating weight of the vessel plus the weight of fill and concrete pad. Use Zick's analysis to determine the shell stresses.

- Operating load on one saddle, Q

$$Q = .5\left(W_O + W_f + W_{cp}\right)$$

Vessels in Pit

The vessel must be anchored with either anchor bolts in the saddle or anchor straps. Determine the size of anchor straps or bolts necessary to resist flotation.

- Buoyancy force, F

$$F = (V_V + V_n)w_w$$

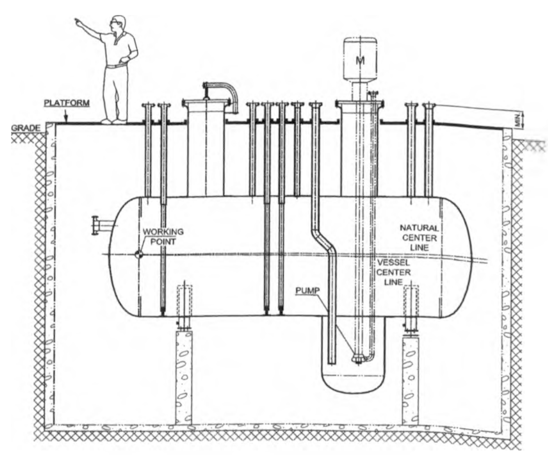


Figure 6-18. Vessel in sump pit

- Net uplift, U

  $U = F\text{-}W_V$

- If $F > U$ then no additional anchorage is required. If $U > F$, then the difference should be added to anchorage required.

Example # 1

Given:

Vessel with 2:1 SE heads, 0.25 in. wall thickness and 24 ft Tan-tan length

$$D = 8 \text{ ft}$$
$$L_V = 28 \text{ ft}$$
$$V_V = 0.262\ D^3 + .25\ \pi\ D^2\ L_{T\text{-}T} = 134 + 1206 = 1340.4 \text{ Ft}^3$$
$$W_V = 9560 \text{ Lbs}$$
$$V_n = (1)30 \text{ in dia} = 12.27 \text{ ft}^3$$
$$(1)12 \text{ in dia} = 1.96 \text{ ft}^3$$
$$(1)6 \text{ in dia} = .98 \text{ ft}^3$$
$$\text{Total} = 15.21 \text{ ft}^3$$

CASE 1: Buried Vessel. Check if a deadman and saddles are required;

$$d = 2.5 \text{ ft}$$
$$h = .5D + d = .5(8) + 2.5 = 6.5 \text{ ft}$$
$$A_1 = (D + 2)(L_V + 4) = 10 \times 32 = 320 \text{ ft}^2$$
$$A_2 = D(L_V) = 8 \times 28 = 224 \text{ ft}^2$$

- Volume of fill, V_f

$$V_f = \left[\left(0.33\ h\right)\left(A_1 + A_2 + (A_1 A_2)^{1/2}\right)\right] - \left[0.5V_V + V_n\right]$$

$$V_f = \left[(0.33(6.5))(320 + 224 + 267)\right] - \left[0.5(1340) + 1521\right] = 1072 \text{ ft}^3$$

- Weight of fill, W_f

  $W_f = w_f\ V_f = 60 \times 1072 = 64,317 \text{ Lbs}$

- Weight of concrete pad, W_{cp}

  $W_{cp} = w_c\ A_1\ t_p = 125(320).5 = 20,000 \text{ Lbs}$

- Total weight, $\sum W$

$$\sum W = W_V + W_f + W_{cp} = 9560 + 64,317 + 20,000 = 93,877 \text{ Lbs}$$

- Buoyancy force, F

$$F = 62.4(V_V + V_n) = 62.4(1340.4 + 15.21) = 84,590 \text{ Lbs}$$

Since the weight of the vessel, fill and concrete pad are heavier than the buoyancy force, no additional restraint is required.

- Factor of Safety, FS

$$FS = \sum W/F = 93,877/84,590 = 1.11$$

If a higher FS is required, than a deadman would have to be added to increase the overall weight of components holding the vessel down.

Example # 2: Vessel in Pit, No Overburden

$$W_V = 7870 \text{ Kg}$$
$$V_V = 21 \text{ m}^3$$
$$w_w = 1001 \text{ Kg/m}^3$$
$$F = V_V\ w_w = 21(1001) = 21,032 \text{ Kg}$$
$$U = F - W_V = 21,032 - 7870 = 13,162 \text{ Kg}$$

Add value of "U" to uplift calculation for design of straps and/or anchor bolts.

Notes

1. Saddle supports are required if the backfill is compacted by tamping, rolling or surface vibration. If saddles are used then a concrete anchorage pad will be required for the saddle to rest on.
2. A saddle is not required if compaction is accomplished by saturation and internal vibration.
3. A full length concrete pad may be used as support for tanks and vessels without saddles or anchorage. In this case the tank should never be placed directly on the concrete pad. At least 6 in and preferably

12 in of clean sand, pea gravel or crushed stone should be placed before the installation of the tank.
4. Underground storage tanks should be retested with air pressure before being covered.
5. Do not use chock blocks under tank to hold in place. These may interfere with the transfer of the load to the backfill.
6. Tanks should be covered with a minimum of 3 feet of earth.
7. When saddles are used, the vessel and saddle must be designed for the weight of the vessel operating as well as the overburden.
8. Nozzles create a void in the overburden and must be accounted for in calculations.
9. Underground storage tanks for storage of flammable products shall comply with the requirements of the following as appropriate;
   a. UL58 (ANSI B137-1971) for steel tanks
   b. NFPA 31 (ANSI Z95.1 – 1969)
   c. NFPA 30
   d. OSHA, Part 1910, Section 106(b)(3)
   e. ASTM D4021-86 for FRP tanks
   f. API RP 1615
10. A tank/vessel should not be anchored unless saddles are used.
11. If the entire area above tank is paved, assume the area as exerting force on the tank as $L_V + 4$ ft $\times$ D + 2 ft
12. Include any installed equipment in the restraining force calculation, for example Submersible pump, piping weights, etc
13. Anchor straps should be made of flat bar in the area with contact with the shell. However the anchor strap must be isolated from contact with the tank shell by use of a gasket.

Procedure 6-7: Local Thin Area (LTA) [2]

Occasionally a vessel shell will sustain damage or be overground in a local area such that the thickness in the damaged area is below the minimum wall thickness. This is known as a "Local Thin Area" or "LTA". ASME VIII-1 has allowance for such a case as long as certain proportions and guidelines are followed. These guidelines are taken from ASME Section VIII, Division 1, Mandatory Appendix 32 and are presented here.

Nomenclature

C = Projected circumferential length of LTA, in
d = Inside diameter of nozzle, in
L, L_1, L_2 = Projected axial length of LTA, in
R = Inside radius of vessel, in
S = ASME Code allowable stress, PSI
t = Full metal thickness, in
t_L = Minimum thickness of LTA, in
θ = Per Figure 6-19

Case 1: Single LTA

A single LTA is acceptable providing all of the following conditions are met;

1. $t_L / t \geq 0.9$
2. $L \leq \sqrt{Rt}$
3. $C \leq 2\sqrt{Rt}$
4. $t - t_L \leq 3/16$ in.
5. Axial distance at edge of LTA is at least;
   a. $2.5(R\ t)^{1/2}$ from gross structural discontinuity
   b. $d + (R\ t)^{1/2}$ from edge of unreinforced nozzle

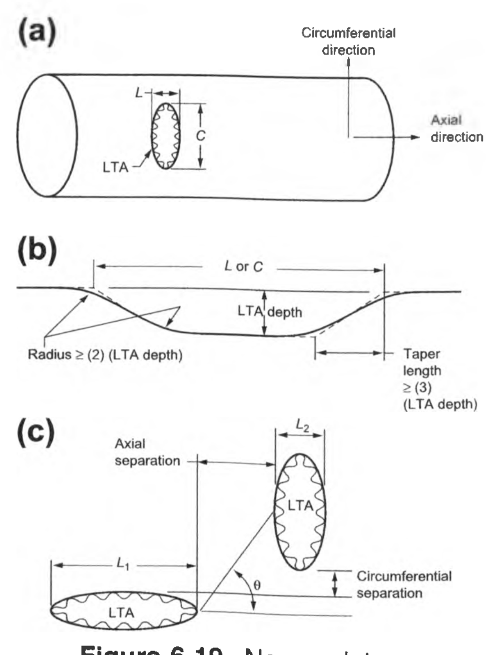


Figure 6-19. Nomenclature.

Table 6-16
Maximum metal temperature

TABLE	TEMPERATURE, °F
UCS-23	700
UNF-23.1	300
UNF-23.2	150
UNF-23.3	900
UNF-23.4	600
UNF-23.5	600
UHA-23	800
UHT-23	700

6. Taper length ≥ 3
7. Longitudinal stress on LTA from mechanical loads is $\leq 0.3S$
8. Maximum design temperature is less than shown in Table 6-16 for a given material.
9. These rules do not apply to corrosion resistant lining or weld overlay.

Case 2: Multiple LTA

A pair of LTA with finished axial length, L_1 and L_2, providing all of the conditions for a single LTA are met and all of the following:

1. When $\theta \leq 45^\circ$, the minimum axial separation shall be the greater of the following:
   a. 2 t
   b. $[(1 + 1.5 \text{ Cos } \theta) (L_1 + L_2)] / 2$
2. When $\theta > 45^\circ$, the minimum axial separation shall be;
   $[(2.91 \text{ Cos } \theta) (L_1 + L_2)]/2$

   And the minimum circumferential separation shall be ≤ 2 t
3. Multiple pairs are acceptable providing all pairs meet the conditions above.
4. Multiple LTA's may be combined as a single LTA.

Notes

1. The condition shall be noted on the Manufacturer's Data Report if the LTA occurs prior to ASME Code stamping.

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