2
General Design

Procedure 2-1: General Vessel Formulas [1,2]

Notation

P = internal pressure, psi
D_i, D_o = inside/outside diameter, in.
S = allowable or calculated stress, psi
E = joint efficiency
L = crown radius, in.
R_i, R_o = inside/outside radius, in.
K, M = coefficients (See Note 3)
σ_x = longitudinal stress, psi
σ_ϕ = circumferential stress, psi
R_m = mean radius of shell, in.
t = thickness or thickness required of shell, head, or cone, in.
r = knuckle radius, in.

Notes

1. Formulas are valid for:
   a. Pressures <3,000 psi.
   b. Cylindrical shells where $t \leq 0.5\ R_i$ or $P \leq 0.385\ SE$. For thicker shells see Reference 1, Para. 1–2.
   c. Spherical shells and hemispherical heads where $t \leq 0.356\ R_i$ or $P \leq 0.665\ SE$. For thicker shells see Reference 1, Para. 1–3.
2. All ellipsoidal and torispherical heads having a minimum specified tensile strength greater than 80,000 psi shall be designed using S = 20,000 psi at ambient temperature and reduced by the ratio of the allowable stresses at design temperature and ambient temperature where required.

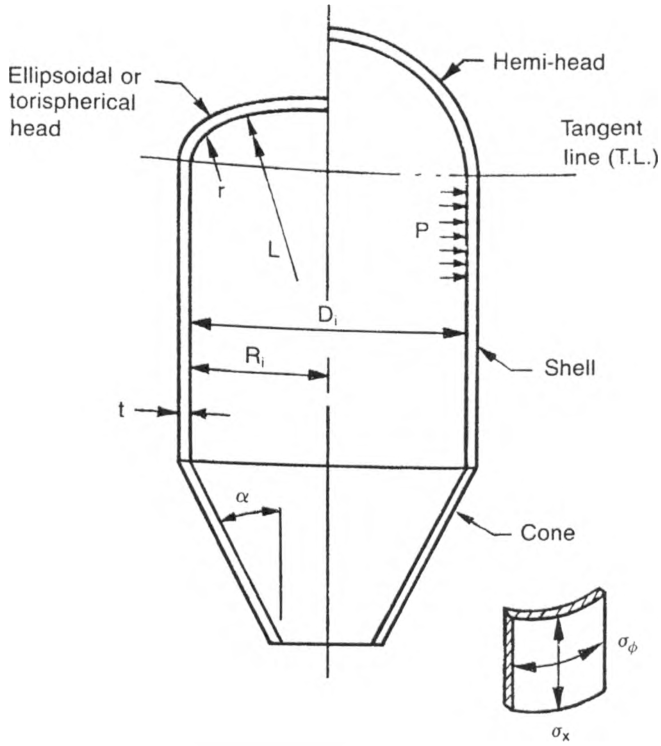


Figure 2-1. General configuration and dimensional data for vessel shells and heads.

3. Formulas for factors:

$$K = 0.167\left[2 + \left(\frac{D}{2h}\right)^2\right]$$

$$M = 0.25\left(3 + \sqrt{\frac{L}{r}}\right)$$

Table 2-1
General vessel formulas

Part	Stress Formula	Thickness, t		Pressure, P		Stress, S	
		I.D.	O.D.	I.D.	O.D.	I.D.	O.D.
Shell							
Longitudinal [1, Section UG-27(c)(2)]	$\sigma_x = \frac{PR_m}{0.2t}$	$\frac{PR_i}{2SE + 0.4P}$	$\frac{PR_o}{2SE + 1.4P}$	$\frac{2SEt}{R_i - 0.4t}$	$\frac{2SEt}{R_o - 1.4t}$	$\frac{P(R_i - 0.4t)}{2Et}$	$\frac{P(R_o - 1.4t)}{2Et}$
Circumferential [1, Section UG-27(c)(1); Section 1-1 (a)(1)]	$\sigma_\phi = \frac{PR_m}{t}$	$\frac{PR_i}{SE - 0.6P}$	$\frac{PR_o}{SE + 0.4P}$	$\frac{SEt}{R_i + 0.6t}$	$\frac{SEt}{R_o - 0.4t}$	$\frac{P(R_i + 0.6t)}{Et}$	$\frac{P(R_o - 0.4t)}{Et}$
Heads							
Hemisphere [1, Section 1-1 (a)(2); Section UG-27(d)]	$\sigma_x = \sigma_\phi = \frac{PR_m}{2t}$	$\frac{PR_i}{2SE - 0.2P}$	$\frac{PR_o}{2SE + 0.8P}$	$\frac{2SEt}{R_i + 0.2t}$	$\frac{2SEt}{R_o - 0.8t}$	$\frac{P(R_i + 0.2t)}{2Et}$	$\frac{P(R_o - 0.8t)}{2Et}$
Ellipsoidal [1, Section 1-4(c)]	See Procedure 2-7	$\frac{PD_iK}{2SE - 0.2P}$	$\frac{PD_oK}{2SE + 2P(K - 0.1)}$	$\frac{2SEt}{KD_i + 0.2t}$	$\frac{2SEt}{KD_o - 2t(K - 0.1)}$	See Procedure 2-7	
2:1 S.E. [1, Section UG-32(d)]	See Procedure 2-7	$\frac{PD_i}{2SE - 0.2P}$	$\frac{PD_o}{2SE + 1.8P}$	$\frac{2SEt}{D_i + 0.2t}$	$\frac{2SEt}{D_o - 1.8t}$	See Procedure 2-7	
100%–6% Torispherical [1, Section UG-32(e)]	See Procedure 2-7	$\frac{0.885PL_i}{SE - 0.1P}$	$\frac{0.885PL_o}{SE + 0.8P}$	$\frac{SEt}{0.885L_i + 0.1t}$	$\frac{SEt}{0.885L_o - 0.8t}$	See Procedure 2-7	
Torispherical L/r < 16.66 [1, Section 1-4(d)]	See Procedure 2-7	$\frac{PL_iM}{2SE - 0.2P}$	$\frac{PL_oM}{2SE + P(M - 0.2)}$	$\frac{2SEt}{L_iM + 0.2t}$	$\frac{2SEt}{L_oM - t(M - 0.2)}$	See Procedure 2-7	
Cone							
Longitudinal	$\sigma_x = \frac{PR_m}{2t\cos\alpha}$	$\frac{PD_i}{4\cos\alpha(SE + 0.4P)}$	$\frac{PD_o}{4\cos\alpha(SE + 1.4P)}$	$\frac{4SEt\cos\alpha}{D_i - 0.8t\cos\alpha}$	$\frac{4SEt\cos\alpha}{D_o - 2.8t\cos\alpha}$	$\frac{P(D_i - 0.8t\cos\alpha)}{4Et\cos\alpha}$	$\frac{P(D_o - 2.8t\cos\alpha)}{4Et\cos\alpha}$
Circumferential [1, Section 1-4(e); Section UG-32(g)]	$\sigma_\phi = \frac{PR_m}{t\cos\alpha}$	$\frac{PD_i}{2\cos\alpha(SE - 0.6P)}$	$\frac{PD_o}{2\cos\alpha(SE + 0.4P)}$	$\frac{2SEt\cos\alpha}{D_i + 1.2t\cos\alpha}$	$\frac{2SEt\cos\alpha}{D_o - 0.8t\cos\alpha}$	$\frac{P(D_i + 1.2t\cos\alpha)}{2Et\cos\alpha}$	$\frac{P(D_o - 0.8t\cos\alpha)}{2Et\cos\alpha}$

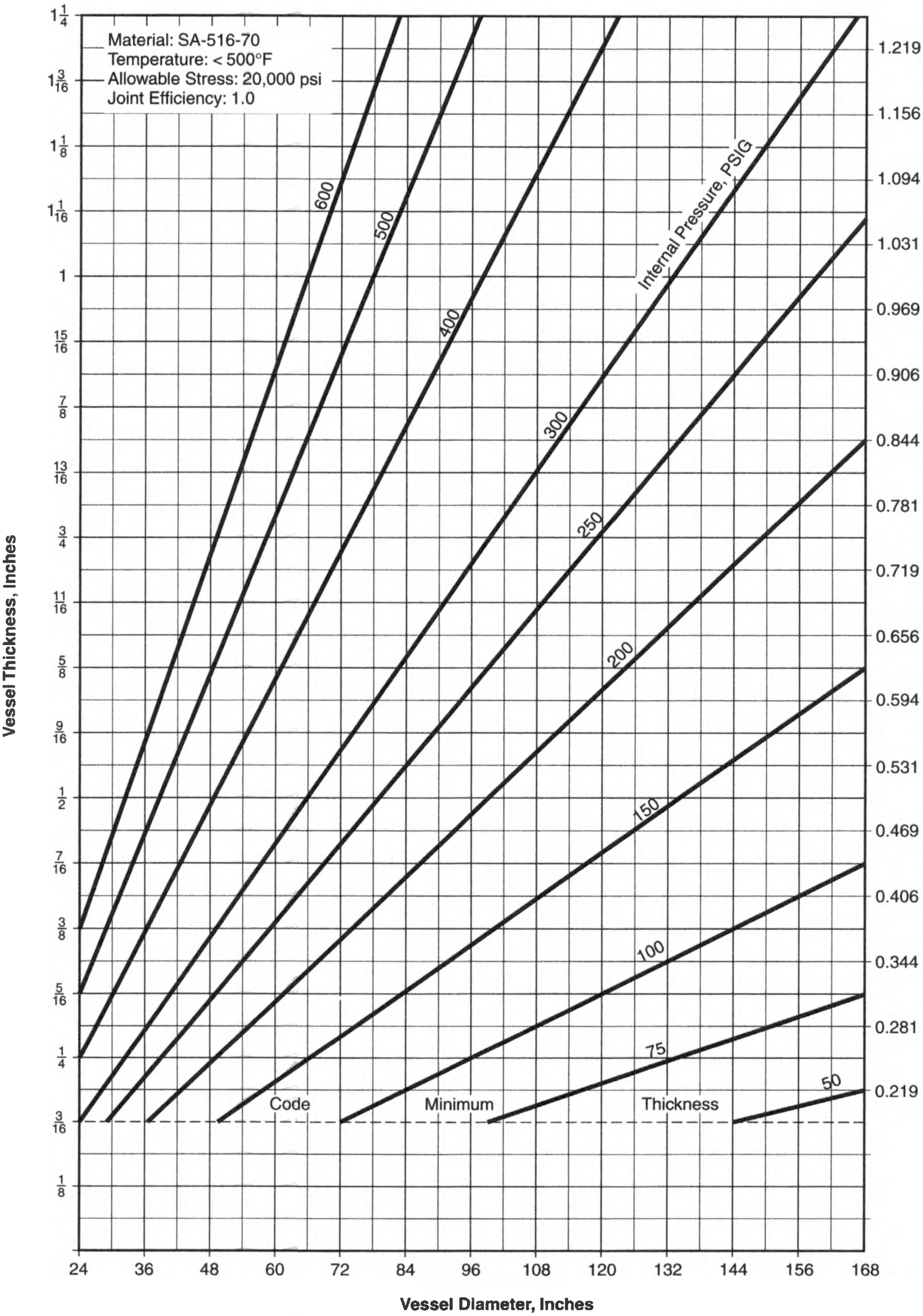


Figure 2-1a. Required shell thickness of cylindrical shell.

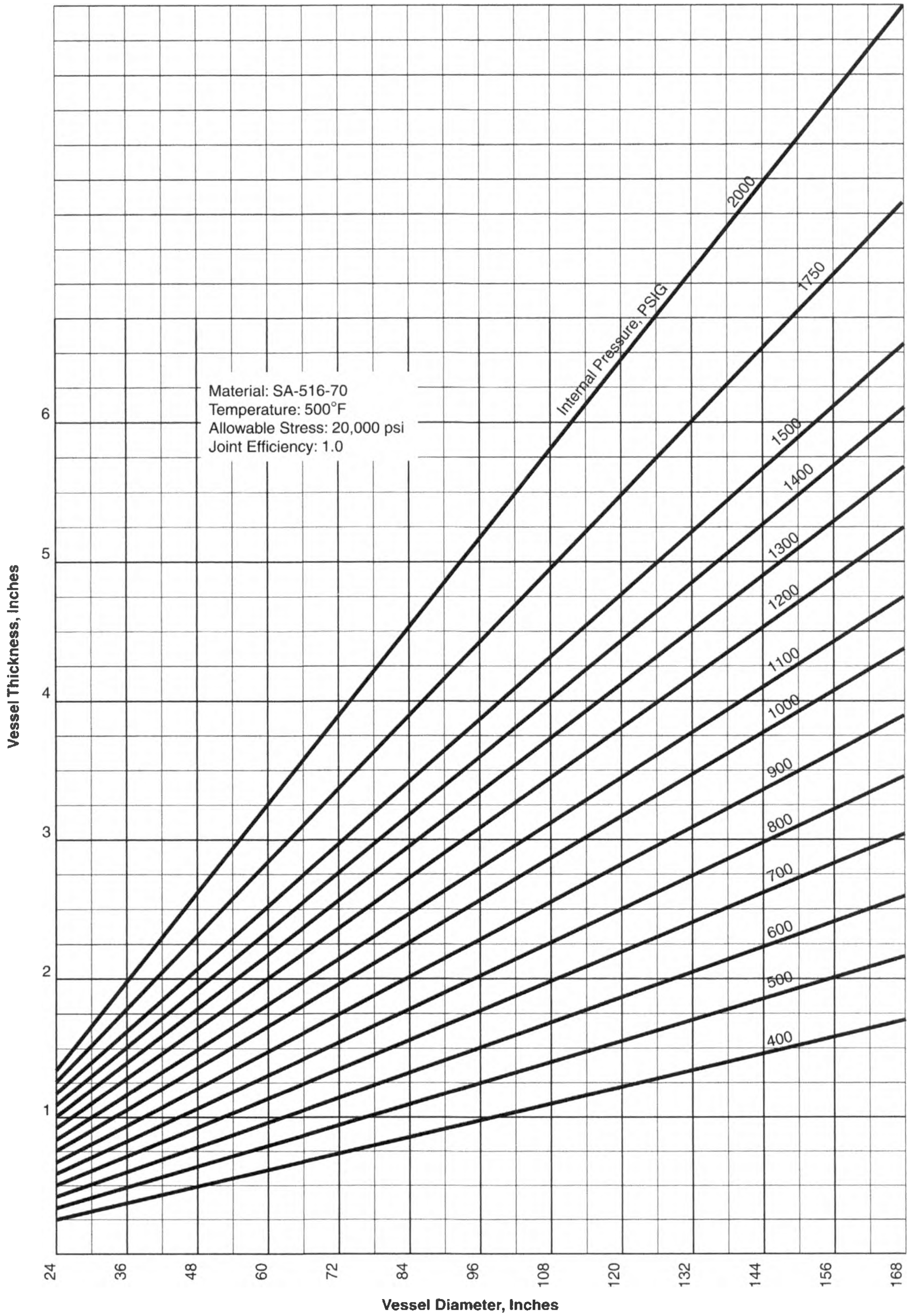


Figure 2-1a. (*continued*).

Procedure 2-2: External Pressure Design

Notation

A = factor "A," strain, from ASME Section II, Part D, Subpart 3, dimensionless
A_s = cross-sectional area of stiffener, in.^{2}
B = factor "B," allowable compressive stress, from ASME Section II, Part D, Subpart 3, psi
D = inside diameter of cylinder, in.
D_o = outside diameter of cylinder, in.
D_L = outside diameter of the large end of cone, in.
D_s = outside diameter of small end of cone, in.
E = modulus of elasticity, psi
I = actual moment of inertia of stiffener, in.^{4}
I_s = required moment of inertia of stiffener, in.^{4}
I'_s = required moment of inertia of combined shell-ring cross section, in.^{4}
L = for cylinders—the design length for external pressure, including 1/3 the depth of heads, in. For cones—the design length for external pressure (see Figures 2-1b and 2-1c), in.
L_e = equivalent length of conical section, in.
L_s = length between stiffeners, in.
$L_{T\text{-}T}$ = length of straight portion of shell, tangent to tangent, in.
P = design internal pressure, psi
P_a = allowable external pressure, psi
P_x = design external pressure, psi
R_o = outside radius of spheres and hemispheres, crown radius of torispherical heads, in.
t = thickness of cylinder, head or conical section, in.
t_e = equivalent thickness of cone, in.
α = half apex angle of cone, degrees

Unlike vessels which are designed for internal pressure alone, there is no single formula, or unique design, which fits the external pressure condition. Instead, there is a range of options available to the designer which can satisfy the solution of the design. The thickness of the cylinder is only one part of the design. Other factors which affect the design are the length of cylinder and the use, size, and spacing of stiffening rings. Designing vessels for external pressure is an iterative procedure. First, a design is selected with all of the variables included, then the design is checked to determine if it is adequate. If inadequate, the procedure is repeated until an acceptable design is reached.

Vessels subject to external pressure may fail at well below the yield strength of the material. The geometry of the part is the critical factor rather than material strength. Failures can occur suddenly, by collapse of the component.

External pressure can be caused in pressure vessels by a variety of conditions and circumstances. The design pressure may be less than atmospheric due to condensing gas or steam. Often refineries and chemical plants design all of their vessels for some amount of external pressure, regardless of the intended service, to allow for steam cleaning and the effects of the condensing steam. Other vessels are in vacuum service by nature of venturi devices or connection to a vacuum pump. Vacuums can be pulled inadvertently by failure to vent a vessel during draining, or from improperly sized vents.

External pressure can also be created when vessels are jacketed or when components are within multichambered vessels. Often these conditions can be many times greater than atmospheric pressure.

When vessels are designed for both internal and external pressure, it is common practice to first determine the shell thickness required for the internal pressure condition, then check that thickness for the maximum allowable external pressure. If the design is not adequate then a decision is made to either bump up the shell thickness to the next thickness of plate available, or add stiffening rings to reduce the "L" dimension. If the option of adding stiffening rings is selected, then the spacing can be determined to suit the vessel configuration.

Neither increasing the shell thickness to remove stiffening rings nor using the thinnest shell with the maximum number of stiffeners is economical. The optimum solution lies somewhere between these two extremes. Typically, the utilization of rings with a spacing of 2D for vessel diameters up to about eight feet in diameter and a ring spacing of approximately "D" for diameters greater than eight feet, provides an economical solution.

The design of the stiffeners themselves is also a trial and error procedure. The first trial will be quite close if the old API-ASME formula is used. The formula is as follows:

$$I_s = \frac{0.16 D_o^3 P_x L_s}{E}$$

Stiffeners should never be located over circumferential weld seams. If properly spaced they may also double as insulation support rings. Vacuum stiffeners, if combined with other stiffening rings, such as cone reinforcement rings or saddle stiffeners on horizontal vessels, must be designed for the combined condition, not each independently. If at all possible, stiffeners should always clear shell nozzles. If unavoidable, special attention should be given to the design of a boxed stiffener or connection to the nozzle neck.

Design Procedure For Cylindrical Shells

Step 1: Assume a thickness if one is not already determined.

Step 2: Calculate dimensions "L" and "D." Dimension "L" should include one-third the depth of the heads. The overall length of cylinder would be as follows for the various head types:

W/(2) hemi-heads	$L = L_{T-T} + 0.333D$
W/(2) 2:1 S.E. heads	$L = L_{T-T} + 0.1666D$
W/(2) 100% – 6% heads	$L = L_{T-T} + 0.112D$

Step 3: Calculate L/D_o and D_o/t ratios

Step 4: Determine Factor "A" from ASME Code, Section II, Part D, Subpart 3, Fig G: Geometric Chart for Components Under External or Compressive Loadings (see Figure 2-1e).

Step 5: Using Factor "A" determined in step 4, enter the applicable material chart from ASME Code, Section II, Part D, Subpart 3 at the appropriate temperature and determine Factor "B."

Step 6: If Factor "A" falls to the left of the material line, then utilize the following equation to determine the allowable external pressure:

$$P_a = \frac{2AE}{3(D_o/t)}$$

Step 7: For values of "A" falling on the material line of the applicable material chart, the allowable external pressure should be computed as follows:

$$P_a = \frac{4B}{3(D_o/t)}$$

Step 8: If the computed allowable external pressure is less than the design external pressure, then a decision must be made on how to proceed. Either (a) select a new thickness and start the procedure from the beginning or (b) elect to use stiffening rings to reduce the "L" dimension. If stiffening rings are to be utilized, then proceed with the following steps.

Step 9: Select a stiffener spacing based on the maximum length of unstiffened shell (see Table 2-1a). The stiffener spacing can vary up to the maximum value allowable for the assumed thickness. Determine the number of stiffeners necessary and the corresponding "L" dimension.

Step 10: Assume an approximate ring size based on the following equation:

$$I = \frac{0.16 D_o^3 P_x L_s}{E}$$

Step 11: Compute Factor "B" from the following equation utilizing the area of the ring selected:

$$B = \frac{0.75 P D_o}{t + A_s/L_s}$$

Step 12: Utilizing Factor "B" computed in step 11, find the corresponding "A" Factor from the applicable material curve.

Step 13: Determine the required moment of inertia from the following equation. Note that Factor "A" is the one found in step 12.

$$I_s = \frac{\left[D_o^2 L_s \left(t + A_s/L_s\right) A\right]}{14}$$

Step 14: Compare the required moment of inertia, I, with the actual moment of inertia of the selected member. If the actual exceeds that which is required, the design is acceptable but may not be optimum. The optimization process is an iterative process in which a new member is selected, and steps 11 through 13 are repeated until the required size and actual size are approximately equal.

Notes

1. For conical sections where $\alpha < 22.5$ degrees, design the cone as a cylinder where $D_o = D_L$ and length is equal to L.
2. If a vessel is designed for less than 15 psi, and the external pressure condition is not going to be stamped on the nameplate, the vessel does not have to be designed for the external pressure condition.

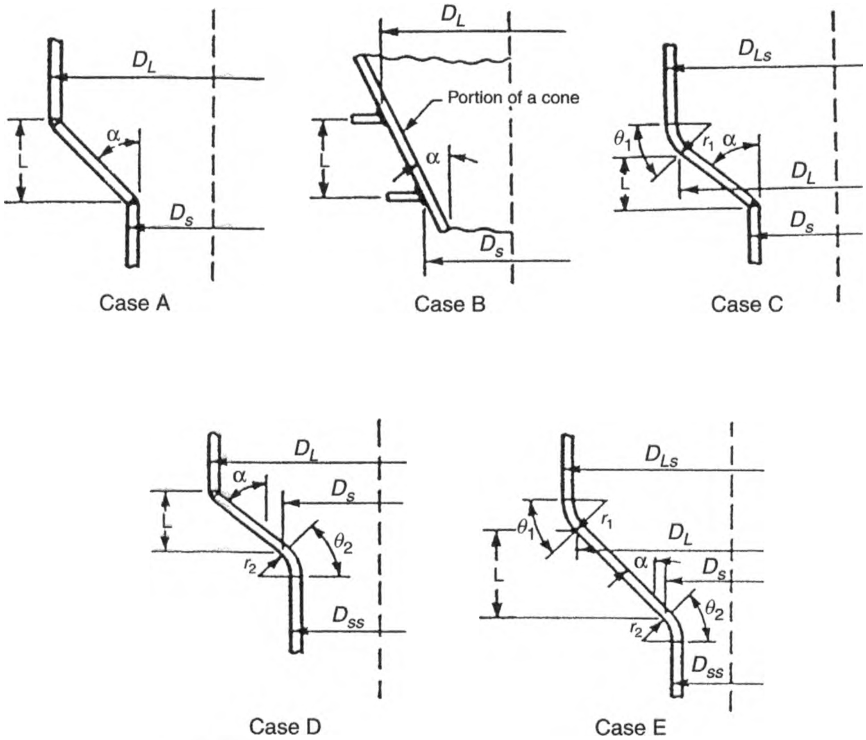


Figure 2-1b. External pressure cones 22 1/2° $< \alpha <$ 60°.

For Case B, $L_e = L$

For Cases A, C, D, E:

$$L_e = 0.5\left(1 + \frac{D_s}{D_L}\right)$$

$$t_e = t \cos \alpha$$

$$D_{L/t_e} =$$

$$L_{e/D_L} =$$

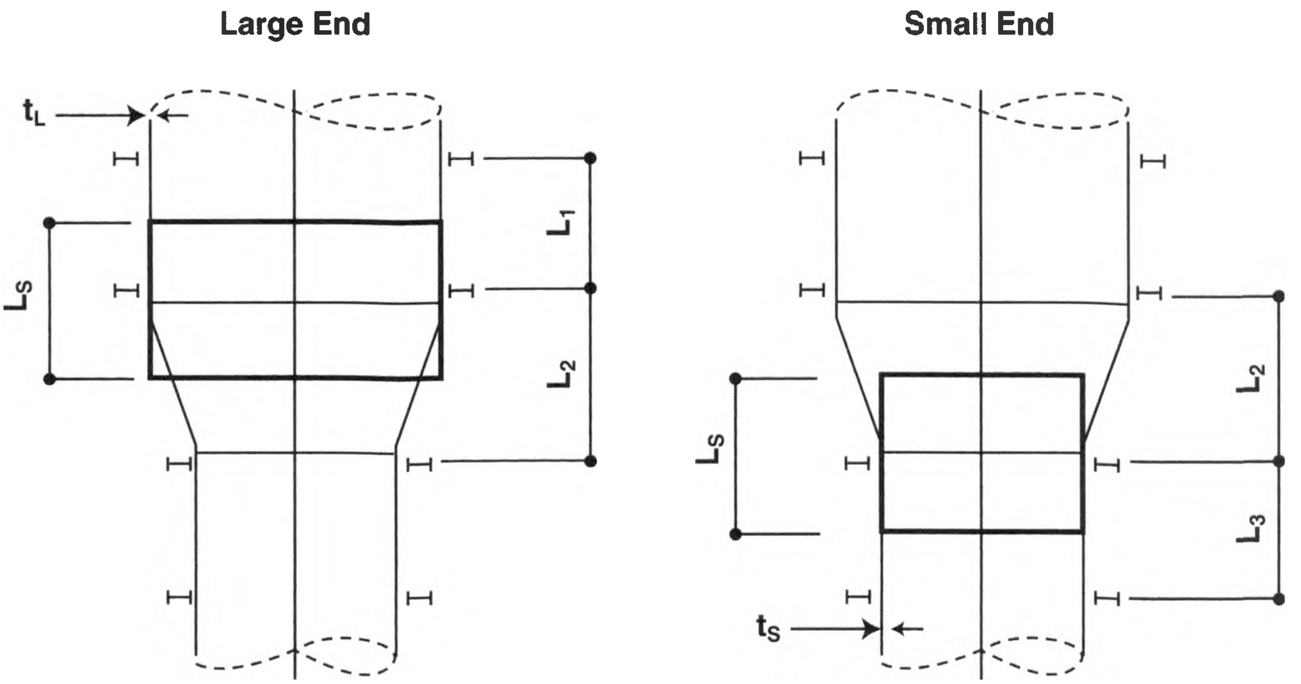


Figure 2-1c. Combined shell/cone design for stiffened shells.

Design stiffener for large end of cone as cylinder where:

$$D_o = D_L$$

$$t = t_L$$

$$L_s = \frac{L_1}{2} + \frac{L_2}{2}$$

Design stiffener for small end of cone as cylinder where:

$$D_o = D_s$$

$$t = t_s$$

$$L_s = \frac{L_3}{2} + \frac{L_2}{2}\left[1 + \frac{D_s}{D_L}\right]$$

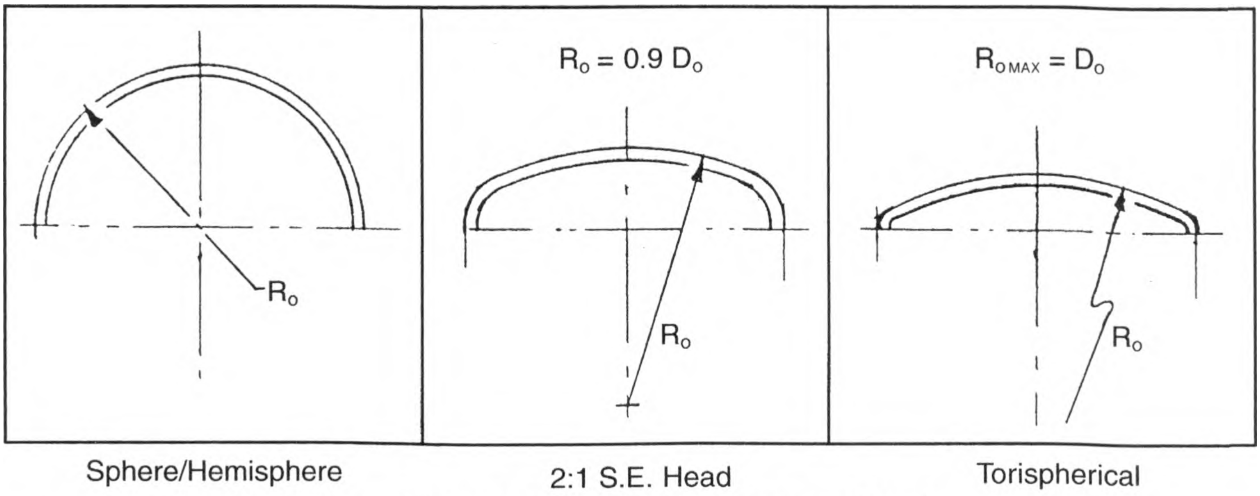


Figure 2-1d. External pressure ~ spheres and heads.

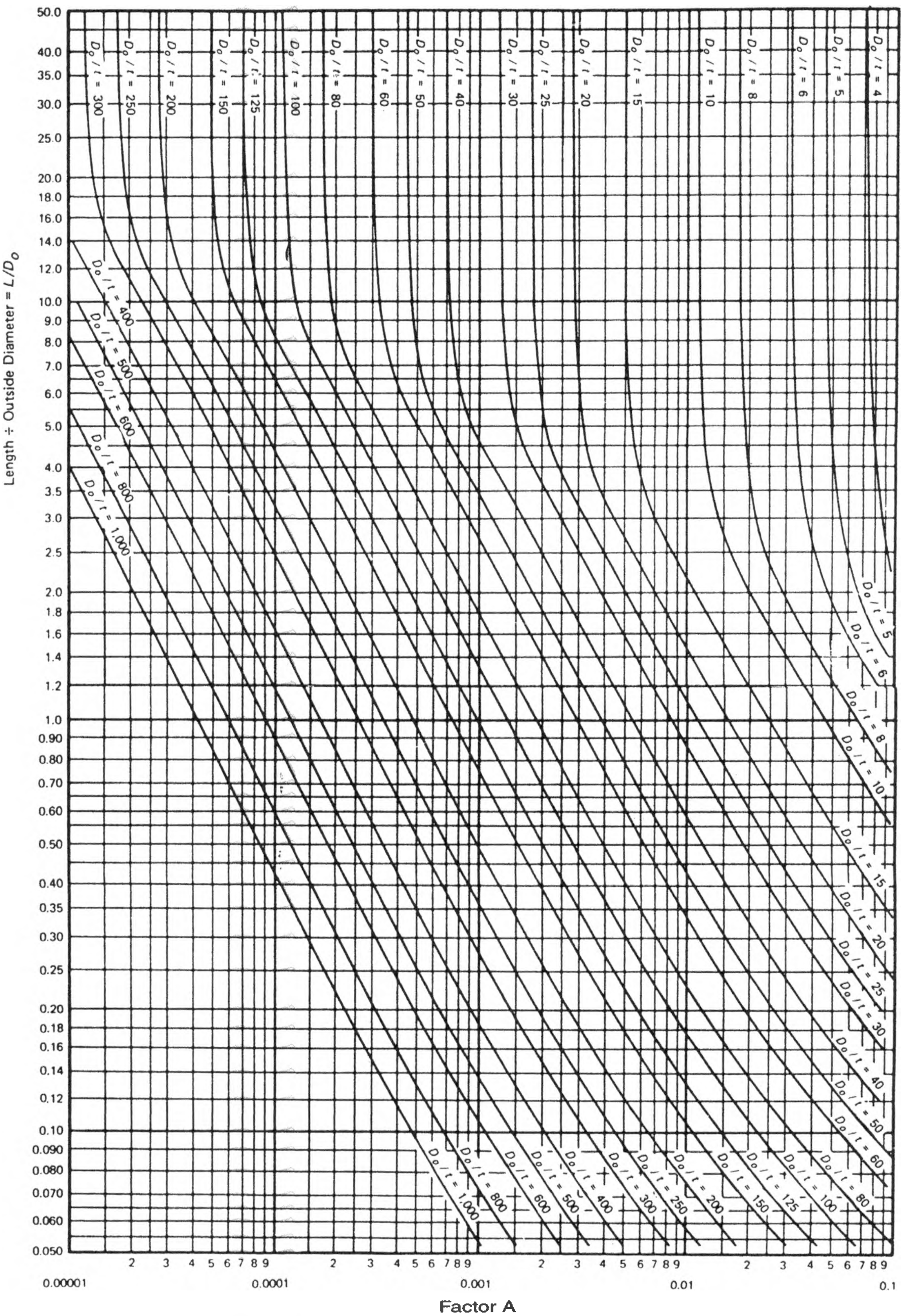


Figure 2-1e. Geometric chart for components under external or compressive loadings (for all materials). (*Reprinted by permission from the ASME Code, Section VIII, Div. 1.*)

Design Procedure For Spheres and Heads

Step 1: Assume a thickness and calculate Factor "A."

$$A = \frac{0.125t}{R_o}$$

Step 2: Find Factor "B" from applicable material chart.

B =

Step 3: Compute P_a.

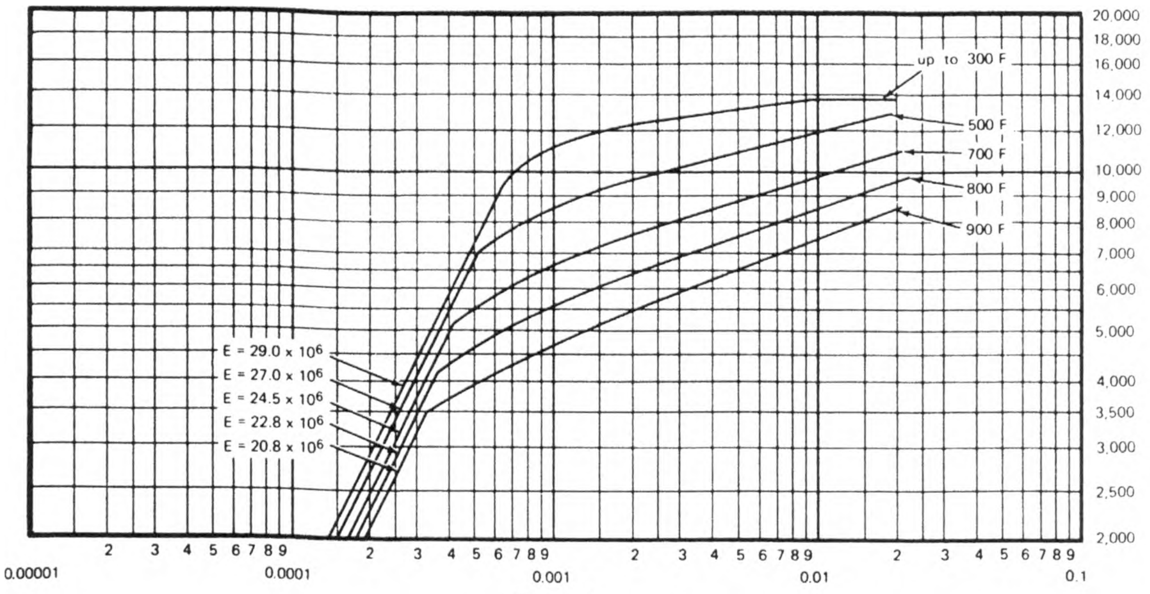


Figure 2-1f. Chart for determining shell thickness of components under external pressure when constructed of carbon or low-alloy steels (specified minimum yield strength 24,000psi to, but not including, 30,000psi). *(Reprinted by permission from the ASME Code, Section VIII, Div. 1.)*

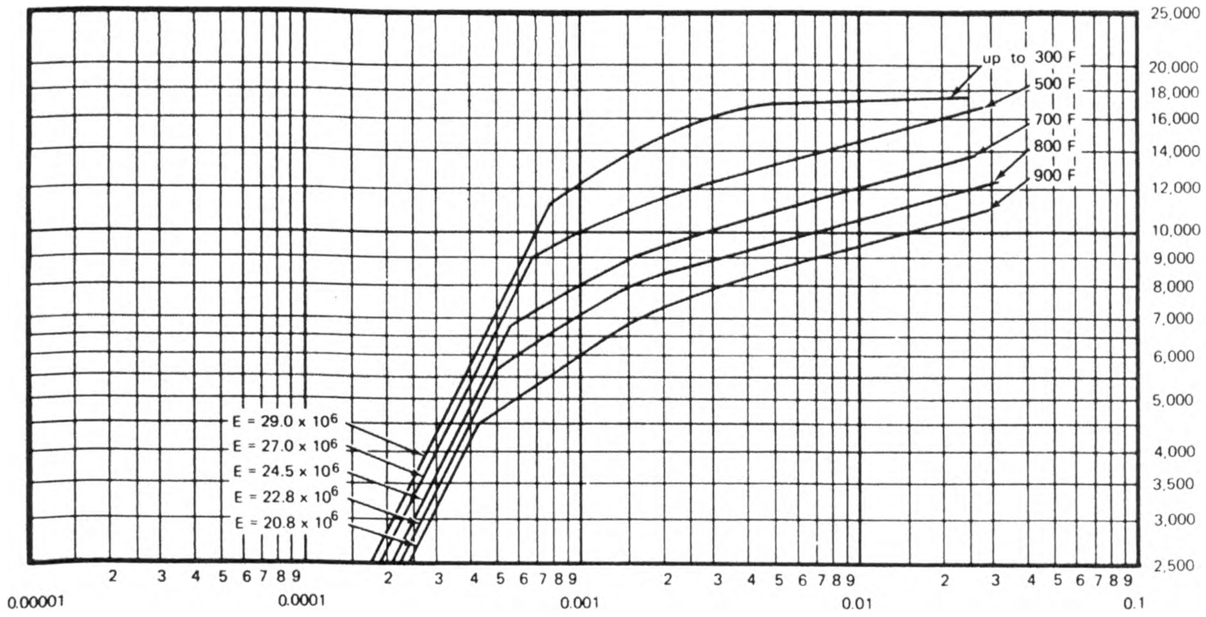


Figure 2-1g. Chart for determining shell thickness of components under external pressure when constructed of carbon or low-alloy steels (specified minimum yield strength 30,000 psi and over except materials within this range where other specific charts are referenced) and type 405 and type 410 stainless steels. (*Reprinted by permission from the ASME Code, Section VIII, Div. 1.*)

A to left of material line $P_a = \dfrac{0.0625E}{(R_o/t)^2}$

A to right of material line $P_a = \dfrac{Bt}{R_o}$

Notes

1. As an alternative, the thickness required for 2:1 S.E. heads for external pressure may be computed from the formula for internal pressure where $P = 1.67\ P_x$ and $E = 1.0$.

Table 2-1a
Maximum length of unstiffened shells

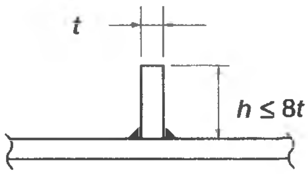
	Thickness (in.)															
Diameter (in.)	¼	5/16	⅜	7/16	½	9/16	⅝	11/16	¾	13/16	⅞	15/16	1	1 1/16	1⅛	1 3/16
36	204															
	∞															
42	168	280														
	313	∞														
48	142	235	358													
	264	437	∞													
54	122	203	306	437												
	228	377	∞													
60	104	178	268	381												
	200	330	499	∞												
66	91	157	238	336	458											
	174	293	442	626	∞											
72	79	138	213	302	408	537										
	152	263	396	56t	∞											
78	70	124	193	273	369	483	616									
	136	237	359	508	686	∞										
84	63	110	175	249	336	438	559									
	123	212	327	462	625	816	∞									
90	57	99	157	228	308	402	510	637								
	112	190	300	424	573	748	∞									
96	52	90	143	210	284	370	470	585	715							
	103	173	274	391	528	689	875	∞								
102	48	82	130	190	263	343	435	540	661	795						
	94	160	249	363	490	639	810	1,005	∞							
108	44	76	118	176	245	320	405	502	613	738	875					
	87	148	228	337	456	594	754	935	∞							
114	42	70	109	162	223	299	379	469	571	687	816					
	79	138	211	311	426	555	705	874	1,064	∞						
120	39	65	101	149	209	280	355	440	536	642	762	894				
	74	128	197	287	400	521	660	819	997	∞						
126	37	61	95	138	195	263	334	414	504	603	715	839	974			
	69	120	184	266	374	490	621	770	938	1,124	∞					
132	35	57	88	129	181	242	315	391	475	569	673	789	916	1,053		
	65	113	173	248	348	462	586	727	884	1,060	1,253	∞				
138	33	54	83	121	169	228	297	369	449	538	636	744	864	994		
	62	106	163	234	325	437	555	687	836	1,002	1,185	∞				
144	31	51	78	114	158	214	275	350	426	510	603	705	817	940	1,073	
	59	98	154	221	304	411	526	652	793	950	1,123	1,312	∞			
150		49	74	107	148	201	261	332	405	485	573	669	774	891	1,017	1,152
		92	146	209	286	385	499	619	753	902	1,066	1,246	1,442	∞		
156		46	70	101	140	189	248	309	385	462	546	637	737	846	966	1,095
		87	138	199	271	363	475	590	717	859	1,015	1,186	1,373	∞		
162		44	67	96	133	178	233	294	367	440	520	608	703	806	919	1,042
		83	131	189	258	342	448	562	684	819	968	1,131	1,309	1,509	∞	
	¼	5/16	⅜	7/16	½	9/16	⅝	11/16	¾	13/16	⅞	15/16	1	1 1/16	1⅛	1 3/16

Notes:

1. All values are in inches.
2. Values are for temperatures up to 500°F.
3. Top value is for full vacuum, lower value is half vacuum.
4. Values are for carbon or low-alloy steel (F_y > 30,000 psi) based on Figure 2-1g.

Table 2-1b
Moment of inertia of bar stiffeners

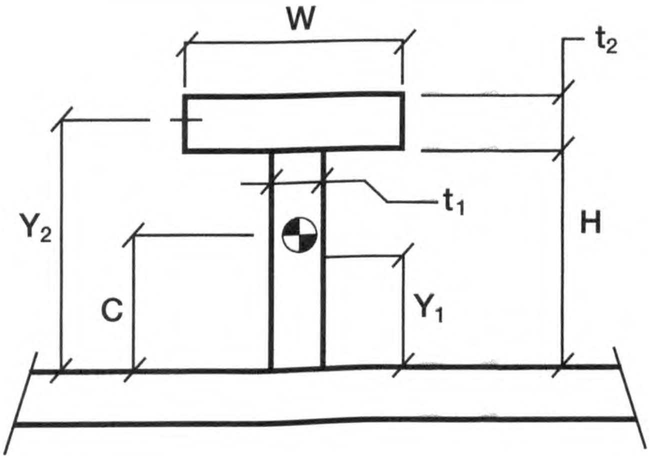
		Height, h, in.														
Thk t, in.	**Max, ht, in.**	**1**	**1½**	**2**	**2½**	**3**	**3½**	**4**	**4½**	**5**	**5½**	**6**	**6½**	**7**	**7½**	**8**
¼	2	0.020	0.070	0.167												
		0.250	0.375	0.5												
5/16	2.5	0.026	0.088	0.208	0.407											
		0.313	0.469	0.625	0.781											
⅜	3	0.031	0.105	0.25	0.488	0.844										
		0.375	0.563	0.75	0.938	1.125										
7/16	3.5		0.123	0.292	0.570	0.984	1.563									
			0.656	0.875	1.094	1.313	1.531									
½	4		0.141	0.333	0.651	1.125	1.786	2.667								
			0.75	1.00	1.25	1.50	1.75	2.00								
9/16	4.5			0.375	0.732	1.266	2.00	3.00	4.271							
				1.125	1.406	1.688	1.969	2.25	2.53							
⅝	5				0.814	1.41	2.23	3.33	4.75	6.510						
					1.563	1.875	2.188	2.50	2.813	3.125						
11/16	5.5					1.55	2.46	3.67	5.22	7.16	9.53					
						2.063	2.406	2.75	3.094	3.438	3.78					
¾	6					1.69	2.68	4.00	5.70	7.81	10.40	13.5				
						2.25	2.625	3.00	3.375	3.75	4.125	4.50				
13/16	6.5						2.90	4.33	6.17	8.46	11.26	14.63	18.59			
							2.844	3.25	3.656	4.063	4.469	4.875	5.281			
⅞	7							4.67	6.64	9.11	12.13	15.75	20.02	25.01		
								3.50	3.94	4.375	4.813	5.25	5.688	6.125		
1	8							5.33	7.59	10.42	13.86	18.00	22.89	28.58	35.16	42.67
								4.00	4.50	5.00	5.50	6.00	6.50	7.00	7.50	8.00



$$I = \frac{th^3}{12}$$

Note: Upper value in table is the moment of inertia. Lower value is the area.

Table 2-1c
Moment of inertia of composite stiffeners



$$I_1 = \frac{t_1 H^3}{12}$$

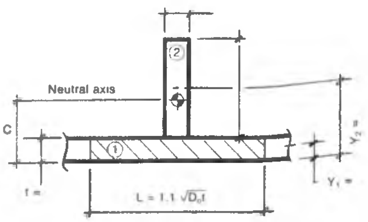
$$I_2 = \frac{w t_2^3}{12}$$

$$c = \frac{\sum A_n Y_n}{\sum A}$$

$$I = \sum A_n Y_n^2 + \sum I - C \sum A_n Y_n$$

Type	H	W	t_1	t_2	$\sum A$	$\sum I$	C	I
1	3	3	0.375	0.5	2.63	0.87	2.50	2.84
2	3	4	0.5	0.5	3.50	1.17	2.50	3.80
3	4	4	0.375	0.5	3.50	2.04	3.28	6.45
4	4	5	0.5	0.625	5.13	2.77	3.41	9.28
5	4.5	5	0.5	0.5	4.75	3.85	3.57	11.25
6	5	4	0.5	0.625	5.00	5.29	3.91	15.12
7	5.5	4	0.5	0.5	4.75	6.97	4.01	17.39
8	6	5	0.5	0.625	6.13	9.10	4.69	25.92
9	6	6	0.625	0.625	7.50	11.37	4.66	31.82
10	5.5	6	0.875	0.875	10.01	12.47	4.42	37.98
11	6.5	6	0.75	0.75	9.38	17.37	4.99	48.14
12	7	6	0.625	0.75	8.88	18.07	5.46	51.60
13	8	6	0.75	1	12.00	32.50	6.25	93.25
14	8	6	1	1	14.00	43.16	5.93	112.47

Moment of Inertia of Stiffening Rings

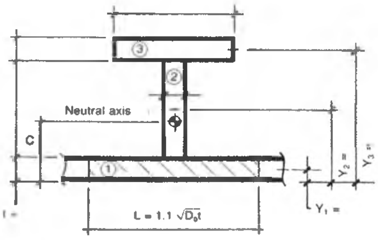


$C = \frac{\Sigma AY}{\Sigma A} =$

$I = \Sigma AY^2 + \Sigma I - C\,\Sigma AY$

Part	Area: A	Y	Y^2	AY	AY^2	I
1						
2						
Σ =						

Figure 2-1h. Case 1: Bar-type stiffening ring.



$C = \frac{\Sigma AY}{\Sigma A} =$

$I = \Sigma AY^2 + \Sigma I - C\,\Sigma AY$

Part	Area: A	Y	Y^2	AY	AY^2	I
1						
2						
3						
Σ =						

Figure 2-1i. Case 2: T-type stiffening ring.

Stiffening ring check for external pressure

Ls		$B = 0.75\dfrac{PD_o}{t + A_s/L_s}$		Moment of inertia w/o shell
t		If B ≤ 2,500 psi,		$I_s = \dfrac{D_o^2 L_s(t + A_s/L_s)A}{14}$
P		A = 2B/E		Moment of inertia w/ shell
D_o		If B > 2,500 psi, determine A from applicable material charts		$I_s = \dfrac{D_o^2 L_s(t + A_s/L_s)A}{10.9}$
A_s				
E = modulus of elasticity				

From Ref. 1, Section UG-29.

Procedure 2-3: Properties of Stiffening Rings

Notation

A_C = Cross sectional area of composite area, in^2
A_r = Cross sectional area of ring, in^2
A_S = Cross sectional area of shell, in^2
D_m = Mean diameter of shell, in
E = Modulus of elasticity at design temperature, PSI
F_y = Minimum specified yield strength at design temperature, PSI
K_8, K_9 = Zick's coefficients
I_C = Moment of inertia of composite section, in^4
P = Internal pressure, PSIG
P_X = External pressure, PSIG
Q = Load at saddle, Lbs
R_m = Mean radius of shell, in
r = Inside radius of shell, in
R = Inside radius of shell in feet
S_{13} = Circumferential stress in shell due to load Q, PSI
S_{14} = Circumferential stress in ring due to load Q, PSI
t = Thickness, shell, in
ν = Poisson's ratio
σ_S = Stress in shell due to internal pressure, PSI
σ_T = Stress in shell due to external pressure, PSI

Derivation of Formula for Influence of Shell with Stiffening Ring

Length of shell acting with ring, L

$$.5\,L = (R_m\,t)^{1/2}/(3(1 - v^2))^{1/4}$$

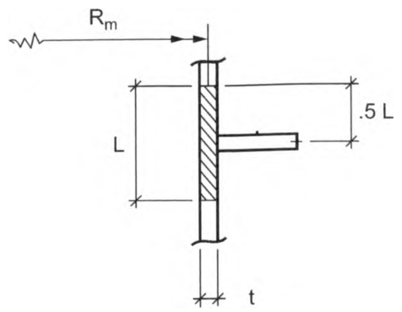


Figure 2-2. Stiffening Ring Dimensions.

For $\nu = .3$;

$$.5\,L = (R_m\,t)^{1/2}\,/\,1.285$$
$$= .78(R_m\,t)^{1/2} = .55\,(D_m\,t)^{1/2}$$

$$L = 1.56\,(R_m\,t)^{1/2} = 1.1(D_m\,t)^{1/2}$$

Lateral Buckling of Stiffening Rings per CC2286

Lateral buckling is dependent on stiffener geometry. The requirements for stiffener geometry per CC2286 are as follows;

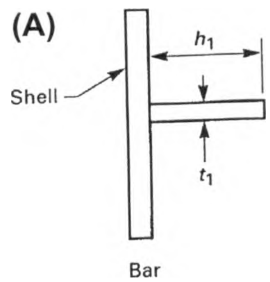
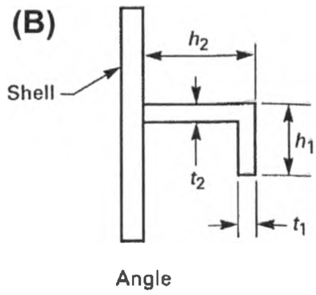


Figure 2-3. (A), (B), (C). Stiffener Geometry Per CC2286



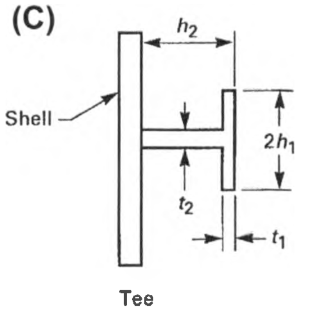


Figure 2-3. *(continued)*.

Case 1: Flat bar stiffener, flange of a tee stiffener, or outstanding leg of an angle stiffener;

$$h_1/t_1 \le .375\ (E/F_y)^{1/2}$$

Case 2: Web of tee stiffener or leg of angle stiffener attached to the shell;

$$h_2/t_2 \le (E/F_y)^{1/2}$$

Shell Stresses Due to Internal or External Pressure on the Ring Section

- Stress in shell due to external pressure, σ_S
  $\sigma_S = (P_X\, L\, R_m)/A_C$
- Stress in shell due to internal pressure, σ_T
  $\sigma_T = [(P\, R_m)/t][A_S/A_C]$

Horizontal Vessel: Shell Stresses at Internal or External Ring Section Due to Load Q

Case 1: External Ring Stiffener

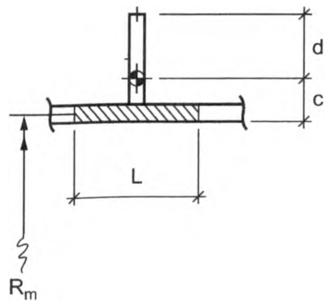


Figure 2-4. External stiffening ring dimensions.

- Stress in shell, S_{13}

  $S_{13} = (-)(K_8\, Q)/A_C + (K_9\, Q\, r\, C)/I_C$
- Stress in ring, S_{14}

  $S_{14} = (-)(K_8\, Q)/A_C(-)(K_9\, Q\, r\, d)/I_C$
- Combined stresses;

  If S_{13} is negative;

$(-)S_{13}(-)\sigma_S < .5\ F_y$

If S_{13} is positive;

$(+)S_{13}(+)\sigma_T < 1.5\ S$

Case 2: Internal Ring Stiffener

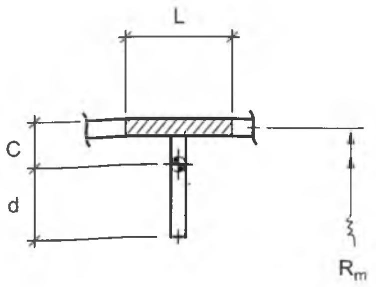


Figure 2-5. Internal stiffening ring dimensions.

- Stress in shell, S_{13}

  $S_{13} = (-)(K_8\,Q)/A_C(-)(K_9\,Q\,r\,C)/I_C$
- Stress in ring, S_{14}

  $S_{14} = (-)\,(K_8\,Q)/A_C(+)(K_9\,Q\,r\,d)/I_C$
- Combined stresses;

  $(-)\,S_{13}\,(-)\,\sigma_S < .5\,F_y$

Sample Problem Given

$Q = 267$ kips

$\theta = 150°$

$A_r = 1 \times 8 = 8\ \text{in}^2$

$A_S = L\,t = 12.93(.875) = 11.31\ \text{in}^2$

$A_C = 19.31\ \text{in}^2$

$S = 20$ KSI

$F_y = 32.6$ KSI

$I_C = 135.88\ \text{in}^4$

$C = 2.27$ in

$d = 6.61$ in

$K_8 = .3$

$K_9 = .032$

$r = 78$ in

$P = 175$ PSIG

$P_X = (-)\,15$ PSIG

$R_m = 78.5625$

$t = 0.875$ in (corr)

$L = 1.56\,(R_m\,t)^{1/2}$

$= 1.56\,(78.5625\,(875))^{1/2}$

$= 12.93$ in

Case 1: External Ring

- Stress in shell, S_{13}

  $S_{13} = (-)\,(K_8\,Q)/A_C + (K_9\,Q\,r\,C)/I_C$

  $= (-)\,.3\,(267)/19.31 + [.032\,(267)\,78\,(2.27)]$

  $/135.88$

  $= (-)\,4.15\ +\ 11.13\ =\ (+)\,6.98$ KSI
- Stress in ring, S_{14}

  $S_{14} = (-)\,(K_8\,Q)/A_C\,(-)\,(K_9\,Q\,r\,d)/I_C$

  $= (-)\,.3\,(267)/19.31\,(-)\,[.032\,(267)\,78$

  $(6.61)]/135.88$

  $= (-)\,4.15\,(-)\,32.42 = (-)\,36.57$ KSI
- Stress in shell due to external pressure, σ_S

  $\sigma_S = (P_X\,L\,R_m)/A_C$

  $= [(-)\,15\,(12.93)78.5625)]/19.31$

  $= (-)7.89$ KSI
- Stress in shell due to internal pressure, σ_T

  $\sigma_T = [(P\,R_m)/t]\,[A_S/A_C]$

  $= [175\,(78.5625)/.875]\,[11.31/19.31]$

  $= (+)\,9.2$ KSI
- Combined stresses;

  Since S_{13} is positive;

  $(+)\,S_{13}\,(+)\,\sigma_T < 1.5\,S$

  $6.98 + 9.2 = 16.18$ KSI

  $<1.5\,S = 1.5\,(20) = 30$ KSI OK

Case 2: Internal Ring (Same properties)

- Stress in shell, S_{13}

$$S_{13} = (-)\,(K_8\,Q)/A_C\,(-)\,(K_9\,Q\,r\,C)/I_C$$
$$= (-)\,4.15\,(-)\,11.13 = (-)\,15.28\ \text{KSI}$$

- Stress in ring, S_{14}

$$S_{14} = (-)\,(K_8\,Q)/A_C\,(+)\,(K_9\,Q\,r\,d)/I_C$$
$$= (-)\,4.15\,(+)\,32.42 = (+)\,28.27\ \text{KSI}$$

- Combined stresses;

$$(-)\,S_{13}\,(-)\,\sigma_S < .5\,F_y$$
$$(-)\,15.28\,(-)\,7.89 = (-)23.17\ \text{KSI}$$
$$< .5\,F_y = .5\,(32.6) = 16.3\ \text{KSI}$$

No Good!
Conclusion: Use an external ring or increase the properties of the internal ring.

Procedure 2-4: Code Case 2286 [1,8,21]

Nomenclature

- A = cross-sectional area of cylinder, $\pi(D_o - t)t$, in^2
- A_S = cross-sectional area of a ring stiffener, in^2
- A_F = cross-sectional area of a large ring stiffener, in^2
- D_i = inside diameter of cylinder, in.
- D_o = inside diameter of cylinder, in.
- D_e = outside diameter of assumed equivalent cylinder for design of cones or conical sections, in.
- D_S = outside diameter at small end of cone, or conical section between lines of support, in.
- D_L = outside diameter at large end of cone, or conical section between lines of support, in.
- E = modulus of elasticity at design temperature, ksi
- f_a = longitudinal compressive membrane stress from axial load, ksi
- f_b = longitudinal compressive membrane stress from bending moment, ksi
- f_h = circumferential compressive membrane stress from external pressure, ksi
- f_q = longitudinal compressive membrane stress from pressure load on end of cylinder, ksi
- f_v = shear stress from applied loads, ksi
- f_x = $f_a + f_q$, ksi
- F_{ba} = allowable longitudinal compressive membrane stress from bending moment, ksi
- F_{ca} = allowable longitudinal compressive membrane stress from uniform axial compression with $\lambda_c > 0.15$, ksi
- F_{aha} = allowable longitudinal compressive membrane stress from uniform axial compression with hoop compression with $0.15 < \lambda_c < 1.2$, ksi
- F_{bha} = allowable longitudinal compressive membrane stress from bending moment with hoop compression, ksi
- F_{cha} = allowable longitudinal compressive membrane stress from uniform axial compression with hoop compression for $0.15 < \lambda_c < 1.2$, ksi.
- F_{cha} = F_{aha} when $f_q = 0$.
- F_{hba} = allowable circumferential compressive membrane stress from bending moment with hoop compression, ksi
- F_{he} = elastic circumferential compressive membrane failure stress from external pressure, ksi
- F_{ha} = allowable circumferential compressive membrane stress from external pressure, ksi

F_{hva} = allowable circumferential compressive membrane stress from shear load with hoop compression, ksi

F_{hxa} = allowable circumferential compressive membrane stress from uniform axial compression with hoop compression, for $\lambda_c \leq 0.15$, ksi

F_{va} = allowable shear stress from applied shear load, ksi

F_{ve} = elastic shear failure stress from applied shear load, ksi

F_{vha} = allowable shear stress from shear load with hoop compression, ksi

F_{xa} = allowable compressive membrane stress from axial compression, for $\lambda_c \leq 0.15$, ksi

F_{xe} = elastic longitudinal compressive membrane failure (local buckling) stress from hoop compression, ksi

F_{xha} = allowable longitudinal compressive membrane stress from uniform axial compression with hoop compression for $\lambda_c \leq 0.15$, ksi.

FS = stress reduction factor or design factor (circumferential and longitudinal directions have separate values)

I = moment of inertia of cylinder cross section, $\pi(D_o - t)^3 t/8$, in^4

I_S' = moment of inertia of ring stiffener plus effective length of shell about its centroidal axis of combined section, in^4

I_S = moment of inertia of ring stiffener about its centroidal axis, in^4

K = coefficient to approximate end conditions for λ_c calculation (2.1 for free standing vessels supported at grade)

$L, L_1, L_2\cdots$ = distance of unstiffened vessel between lines of support, in.

$L_B, L_{B1}, L_{B2}\cdots$ = distance between bulkheads or large rings, in.

L_c = axial length of cone or conical section, in.

L_F = one-half of the sum of the distances, L_B, from one large ring to the next or line of support, in.

L_S = one-half of the sum of the distance, L, from one ring to the next of line of support, in.

L_t = overall length of vessel including 1/3 depth of each head, in.

L_u = laterally unsupported length for a free-standing vessel, for a skirt-supported vessel with no guide wires, the distance is from the top tangent line to the base of the skirt, in.

M = bending moment across cylinder cross section, in-kips

P = design external pressure, ksi

P_a = allowable external pressure

Q = uniform axial compression, kips

Q_p = axial compression as a result of external pressure, ksi

r = radius of gyration of a cylinder, $r = 0.25*(D_o^2 + D_i^2)^{1/2}$, in.

R = radius to centerline of shell, in.

R_c = radius of centroid of combined ring stiffener and effective length of shell, $R_c = R + Z_c$, in.

R_o = radius to outside of shell

s = section modulus of cylinder cross section, $\pi(D_o - t)^2 t/4$, in.

t = thickness of shell, less corrosion allowance, in.

t_c = thickness of cone, less corrosion allowance, in.

T/T = tangent to tangent length, in

V = shear force, kips

Z_c = radial distance from centerline of shell to centroid of combined section of ring and effective length of shell, in.

$$Z_c = \frac{A_S Z_S}{A_S + L_e t}$$

Z_S = radial distance from centerline of shell to centroid of ring stiffener (positive for outside rings), in.

λ_c = slenderness factor for column buckling

$$\lambda_c = \frac{KL_u}{\pi r}\sqrt{\frac{F_{xa}FS}{E}}$$

In 1998, Code Case 2286 *Alternative Rules for Determining Allowable External Pressure and Compressive*

Stresses for Cylinders, Cones, Spheres, and Formed Heads was approved for Section VIII, Divisions 1 and 2 of the ASME Code. Currently, Code Case 2286 may be applied only for Division 1, since the rules found in Code Case 2286 were absorbed into Part 4 of Division 2.

Code Case 2286 not only has rules for cylinders, cones, spheres, and formed heads, but also for the sizing of cone-cylinder junction rings, stiffening rings for external pressure, tolerances, and reinforcement for openings. When 2286 is used, it must be used for the entire vessel. Using 2286 will require a more rigorous analysis than the vacuum chart method in Division 1 and may provide a thinner shell, and fewer and/or smaller stiffening rings. In these cases, the overall cost of the vessel may be lowered as a result of the additional analysis and may be justified in many cases. An analogy may be used in comparing the use of Division 2 over Division 1 in that more design and analysis of the vessel is required in Division 2, however as a result, lower design margins are allowed. It should be noted that local jurisdictions may not always accept the use of 2286 for all equipment though this should be evaluated on a case by case basis.

Code Case 2286 assumes that shells are axisymmetric, that for unstiffened vessels the shells are the same thickness, and that for stiffened cylinders and cones the thickness between stiffeners is uniform. Additionally, capacity reduction factors (or knockdown factors) are provided for general use, but are incorporated in the allowable stress equations. Stress reduction factors (FS) must be found for each direction of loading so that the values of FS are determined independently for both the longitudinal and circumferential directions. The stress reduction factors cover elastic and inelastic buckling, and plastic collapse behavior for elements in compression.

There are several differences between Code Case 2286 and the older vacuum chart method. One difference is the upper limit of temperatures that each method has. Code Case 2286 has a limit of 800F for carbon steels found in UCS-23.1 and 800F for stainless steels in UHA-23, whereas the vacuum charts have higher limits for the more common carbon steels and stainless steels (except in a few cases). See Section II, Part D, Mandatory Appendix 3 for criteria and figures with temperature limits for various classes of materials under external pressure and axial compression. It should be noted that neither method has accounted for the effects of creep on buckling. A complete list of what 2286 covers that the vacuum chart method in Division 1 does not may be found in the 2286 document. A partial list of differences is as follows:

	Code Case 2286	Vacuum Chart Method
Carbon steel (UCS-23.1) temperature limits	800F	900F for most carbon steels
Stainless steel (UHA-23) temperature limits	800F	800F-1500F for most stainless steels
Do / t upper limit	2,000	1,000
Slenderness ratio (KL/r)	200	None
Combined loads	Yes	No
Stiffener geometry requirements	Yes	No

In discussing the differences between the two procedures, most vessels in the refining industry have D_o / t ratios far less than 1,000, so the lower limits of the vacuum charts are usually not a concern. The limitations on slenderness ratios of 200 are from AISC specifications and were based off of engineering judgment, economics in construction, and handling concerns. The evaluation of combined load cases (such as external pressure with uniform axial compression and bending) is also something that is not considered in the vacuum chart method. The use of Code Case 2286 requires the calculation of individual load cases first, and then determining if the combined load cases fulfill the acceptance criteria. This can become tedious since calculations are typically done for each weld plane.

Additionally, tolerances for cylindrical and conical shells are provided when subjected to external pressure, and uniform axial compression and bending. If the tolerances are exceeded, the allowable buckling stresses must be adjusted. In the case of vessels with large diameter over thickness ratios (approximately 300 or higher), it may be prudent to discuss shell tolerances with the fabricator to ensure that if tolerances cannot be met, the reduced allowable buckling stresses can be used for checking the design of the vessel.

The procedure for designing a simple cylindrical vessel to Code Case 2286 or Section VIII, Division 2 is to first establish geometry as well as determine shell thickness values to begin with. Typically, some thickness is established from, say, internal pressure and those thickness values are used for starting the procedure in Code Case 2286. External loads from wind and seismic design criteria must be established to complete the procedure.

The allowable primary membrane compressive stress shall be less than or equal to the maximum allowable tensile stress from Section II, Part D. The procedure is iterative. First, the allowable stresses are determined for each loading condition without being combined, and then each combined case is evaluated.

Step 1: Determine the allowable external pressure.
First, calculate M_x, C_h, and F_{he} using the equations supplied in the Code.
Next, calculate F_{ic}, which is the buckling stress that would be found if FS = 1 in the allowable stress equations. For external pressure, the equations are associated by the F_{he}/F_y ratios.
Then, calculate FS, where the equations are associated with the value of F_{ic} as compared to F_y.
Calculate F_{ha} with the calculated F_{ic} and the calculated FS value. This is the allowable stress value to use in the thickness calculations, assuming no other loadings are occurring.
Finally, determine the allowable pressure.
If the allowable pressure is not sufficient for design conditions, increase the thickness or add ring stiffeners.
Step 2: Determine the allowable longitudinal stress due to axial compression.
First, calculate M_x, c(bar), C_x, and F_{xe} using the equations supplied in the Code.
Next, calculate F_{ic}, which is the buckling stress that would be found if FS = 1 in the allowable stress equations. For uniform axial compression, the equations are associated by the D_o/t ratios.
Then, calculate FS, where the equations are associated with the value of F_{ic} as compared to F_y.
Calculate F_{xa} with the calculated F_{ic} and the calculated FS value.
Calculate λ_c to see if the vessel is subject to column buckling. If it is not then the allowable longitudinal stress due to axial compression is F_{xa}.
If it is, then F_{ca} must be calculated, which will be a reduced value of F_{xa}.
Finally, determine if the actual stress is less than or equal to the allowable stress due to uniform axial compression.
Step 3: Determine the allowable longitudinal stress due to bending moment.
First, using the D_o/t ratio and the $\gamma = F_y D_o/ET$ ratio, determine the appropriate of the four equations to use for F_{ba}.
Next, calculate F_{ic}, which is the buckling stress that would be found if FS = 1 in the allowable stress equations.
Then, calculate FS, where the equations are associated with the value of F_{ic} as compared to F_y.
Calculate F_{ba} with the calculated F_{ic} and the calculated FS value.
Finally, determine if the actual stress is less than or equal to the allowable stress due the bending moment.
Step 4: Determine the allowable in-plane shear stress due to a shear force:
First, calculate α_v, M_x, C_v, F_{ve}, and η_v using the equations supplied in the Code.
Next, calculate F_{ic}, which is the buckling stress that would be found if FS = 1 in the allowable stress equations.
Then, calculate FS, where the equations are associated with the value of F_{ic} as compared to F_y.
Calculate F_{va} with the calculated F_{ic} and the calculated FS value.
Finally, determine if the actual stress is less than or equal to the allowable stress due the shear force. Note that the shear stress may be calculated at various angles around the circumference.
Step 5: Determine the interaction of the stresses under combined loads.
Step 6: Determine the size of the ring stiffener if stiffeners are needed. Ring stiffeners may be sized as either small rings or large rings, as either ring is considered a line of support.

There are several examples using the incorporated Code Case 2286 in Division 2 that may be used as reference in the document ASME PTB-3 Section VIII Division 2 Example Problem Manual.

Procedure 2-5: Design of Cones

Notation

A = factor A from ASME
A_r = Excess area available in junction without a ring, in^2
A_S = Cross sectional area of ring, in^2
A_{eL} = Effective area of reinforcement, large end, in^2
A_{eS} = Effective area of reinforcement, small end, in^2
A_{rL} = Area of reinforcement required at large end, in^2
A_{rS} = Area of reinforcement required at small end, in^2
A_{TL} = Equivalent area of shell, cone, and ring at large end, in^2
A_{TS} = Equivalent area of shell, cone, and ring at small end, in^2
B = Factor B from ASME, PSI
Ca = Corrosion Allowance, in
D_e = OD of equivalent cylinder,in
E_1, E_2 = Joint efficiency in shell or cone
E_s, E_c, E_r = Modulus of elasticity of shell, cone or ring, psi
$f_{1\text{-}4}$ = Axial load per unit circumference excluding pressure, Lbs/in-circ
F_L = Longitudinal load at large end if this is a line of support, Lbs/in
F_S = Longitudinal load at small end if this is a line of support, Lbs/in
I = Available moment of inertia, ring only, in^4
I_S = Required moment of inertia, in^4
I' = Available moment of inertia of combined shell-ring section, in^4
I_S' = Required moment of inertia of combined shell-ring section, in^4
K = Ratio defined herein
L = Length of cone along axis of cone, in
L_a = Maximum distance along shell from junction that can be included as part of reinforcement, in
L_b = Maximum distance along shell from junction to centroid of ring, in
L_C = Length of cone along shell, in
L_{ce} = Length of equivalent cylinder for external pressure, in
L_L = Distance along shell at large end, from junction to first stiffening element, in
L_S = Distance along shell at small end, from junction to first stiffening element, in
M = Factor at large end if this is a LOS
M_1, M_2 = Moment at Elev 1 or 2, in-Lbs
N = Factor at small end if this is a LOS
P = Internal pressure, psig
P_x = External pressure, psig
$Q_{L1\text{-}4}$ = Axial load per unit circumference including pressure, Lbs/in-circ
t_{S1}, t_{S2} = Thickness required, shell, in
t_{rc} = Thickness required, cone, in
t_e = Thickness, equivalent cylinder for large end, in
$t_{1\text{-}2}$ = Thickness of shell, small/large end, in
t_c = thickness of cone, in
t_{rx} = Thickness required for cone external pressure, in
S_s, S_c, S_R = Allowable stress of shell, cone and ring respectively, psi
W_1, W_2 = Weights at Elev 1 or 2, Lbs
X, Y = Factors for cones where $\alpha > 30°$
X_1, Y_1 = ASME ratios defined herein
Δ = From Tables 2-2, 2-3 and 2-4, degrees
σ_X = Membrane longitudinal stress plus discontinuity longitudinal stress due to bending, PSI
σ_ϕ = Membrane hoop stress plus average discontinuity hoop stress, PSI
LOS = Line of Support

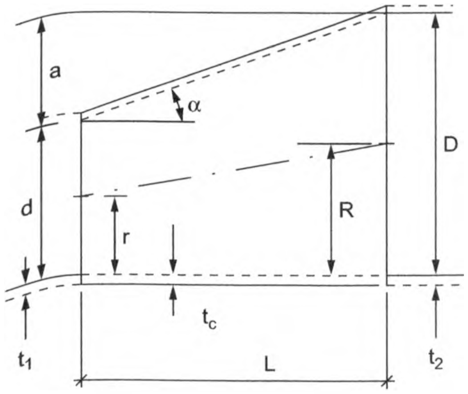


Figure 2-6. Eccentric cone.

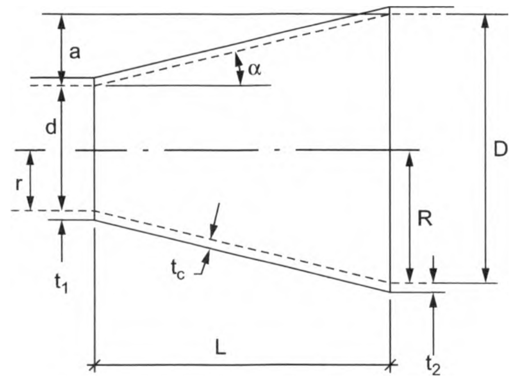


Figure 2-7. Concentric cone.

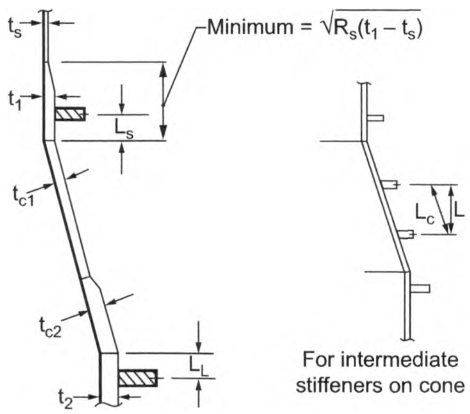


Figure 2-8. Dimensions of cone-cylinder intersections.

Geometry

FOR ECCENTRIC CONE...

$a = D - d$

$\tan \alpha = a/L$

$L = a \tan \alpha$

FOR CONCENTRIC CONE...

$a = .5\,(D - d)$

$\tan \alpha = a/L$

$L = a \tan \alpha$

$$L_C = \left[L^2 + (R - r)^2\right]^{1/2}$$

1.0. Design Cone for Internal Pressure

1.1. Thickness Required

- Required thickness of cone, Internal Pressure, $\alpha \leq 30°$;
  Small End;

$t_{C1} = (P\,r)/(\cos \alpha\,(S_C\,E_2 - .6P))$

Large End;

$t_{C2} = (P\,R)/(\cos \alpha(S_C\,E_2 - .6P))$

- Required thickness of Shell;

Small End;

$t_{S1} = (P\,r)/(S_S\,E_1 - .6P)$

Large End;

$t_{S2} = (P\,R)/(S_S E_1 - .6P)$

1.2. Reinforcement Required

- Values of X_1 and Y_1
  X_1 = Smaller of $S_S\,E_1$ or $S_C\,E_2$
  Y_1 = Greater of $S_S\,E_S$ or $S_C\,E_C$

Table 2-2
Values of Δ, degrees - large end, α ≤ 30 degrees, internal pressure

P/X_1	Δ
.001	11
.002	15
.003	18
.004	21
.005	23
.006	25
.007	27
.008	28.5
.009	30

Large End

- P/X_1 =
  Δ = From Table 2-2
  If $\Delta < \alpha$ then reinforcement is required
  If $\Delta \geq \alpha$ then no reinforcement is required

If reinforcement is required follow the following steps:

- Calculate ratio, K
  $K = Y_1/(S_R\,E_R) > 1$
  $K = 1$ if additional area of reinforcement is not required

- Area of reinforcement required at large end of cone, A_{rL}

Case 1: External Loads are Not Included

$$A_{rL} = \left[(P R^2 K)/2 X_1\right] \left[1 - (\Delta/\alpha)\right] \tan\alpha$$

Case 2: External Loads are Included

$$A_{rL} = [(K Q_L R)/X_1] [1 - (\Delta/\alpha)] \tan\alpha$$

From the worksheet, select the largest value of Q_L for the large end in tension.

Reinforcement Available

- Effective area available at large end, A_{el}

  $$A_{el} = \left(t_2 - t_{S2}\right) (R t_2)^{1/2} + \left(t_C - t_r\right) \times (R t_C/\cos\alpha)^{1/2}$$

- Area required at large end, A_r

  $$A_r = A_{rL} - A_{eL}$$

If A_r is negative, the design is adequate as is. If A_r is positive, then either a ring must be added or the area of section increased. A_r is the net area required for the ring only!

- If a ring is required, the maximum distance to centroid of ring, L_b

  $$L_b = .25 (R t_2)^{1/2}$$

- If the shell is made thicker to provide more area for reinforcement, then the maximum length of shell included in the reinforcement shall be;

  $$L_a = (R t_2)^{1/2}$$

Small End

Table 2-3
Values of Δ, degrees - small end, α ≤ 30 degrees, internal pressure

P/X_1	Δ
.002	4
.005	6
.010	9
.02	12.5
.04	17.5
.08	24
.1	27
.125	30

- $P/X_1 =$

  $\Delta =$ From Table 2-3

  If $\Delta < \alpha$ then reinforcement is required

  If $\Delta \geq \alpha$ then no reinforcement is required

If reinforcement is required follow the following steps:

Case 1: External Loads are Not Included:

$$A_{rS} = \left[(P r^2 K)/2 X_1\right] \left[1 - (\Delta/\alpha)\right] \tan\alpha$$

Case 2: External Loads are Included

$$A_{rS} = [(K Q_L r)/X_1] [1 - (\Delta/\alpha)] \tan\alpha$$

From the worksheet, select the largest value of Q_L for the small end in tension.

Reinforcement Available

- Effective area available at large end, A_{eS}

  $$A_{eS} = .78 \left(t_1 - t_{S1}\right)(r t_1)^{1/2} + ((t_C - t_r)/\cos\alpha)^{1/2}$$

- Area required at large end, A_r

  $$A_r = A_{rS} - A_{eS}$$

If A_r is negative, the design is adequate as is. If A_r is positive, then either a ring must be added or the area of section increased. A_r is the net area required for the ring only!

- If a ring is required, the maximum distance to centroid of ring, L_b

  $$L_b = .25 (r t_1)^{1/2}$$

- If the shell is made thicker to provide more area for reinforcement, then the maximum length of shell included in the reinforcement shall be;

  $$L_a = (r t_1)^{1/2}$$

2.0. Design Cone for External Pressure

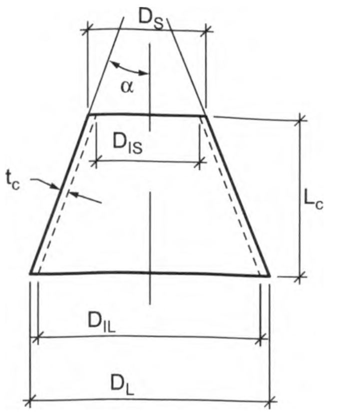


Figure 2-9. Dimensions of concentric cone for external pressure.

2.1. Thickness Required

For $\alpha \leq 22.5°$

- Design the cone as a cylinder where $D_o = D_L$ and the length is equal to L.
- t_{rx} = ________

For α > 22.5°

- Determine half apex angle, α

  $a = \arctan\,[.5(D_{iL} - D_{iS})/L]$
- Length of equivalent cylinder, L_{ce}

  $L_{ce} = L/\cos\alpha$
- Diameter of equivalent cylinder, D_e

  $D_e = .5\,[(D_L + D_S)/\cos\alpha]$
- Thickness of equivalent cylinder, t_e

  $t_e = t_C \cos\alpha$
- Calculate ratios;

  $L_{ce}/D_e =$

  $D_e/t_e =$
- Design the cone as a cylinder with the properties calculated above
- t_{rx} = ________

2.2. Reinforcement Required

LARGE END

- Values of X_1 and Y_1

  X_1 = Smaller of $S_S\,E_1$ or $S_C\,E_2$

  Y_1 = Greater of $S_S\,E_S$ or $S_C\,E_C$

Table 2-4
Values of Δ, degrees - large end, $\alpha \leq 60$ degrees, external pressure

P_X/X_1	Δ
0	0
.002	5
.005	7
.010	10
.02	15
.04	21
.08	29
.1	33
.125	37
.15	40
.2	47
.25	52
.3	57
.35	60

Note: If $P_X/X_1 > .35$ Use $\Delta = 60$

- $P_X/X_1 =$

  Δ = From Table 2-4

  If $\Delta < \alpha$ then reinforcement is required

  If $\Delta \geq \alpha$ then no reinforcement is required

If reinforcement is required follow the following steps:

- Calculate ratio, K

  $K = Y_1/(S_R\,E_R) > 1$

  K = 1 if no additional area is required

Case 1: External Loads are Not Included

$$A_{rL} = \left[(P_X\,R^2\,K)/2\,X_1\right]\left[1 - .25(P_X\,R)\;(\Delta/\alpha)\right]$$

Case 2: External Loads are Included

$$A_{rL} = [(K\,Q_L\,R\tan\alpha)/X_1]\,[1 - .25\,((P_X\,R - Q_L)/Q_L)\;(\Delta/\alpha)]$$

From the worksheet, select the largest value of Q_L for the large end in compression.

Reinforcement Available

- Effective area available at large end, A_{eL}

  $A_{eL} = .55\,(D_L\,t_2)^{1/2}\,(t_2 + t_C/\cos\alpha)$

- Area required at large end, A_r

  $A_r = A_{rL} - A_{eL}$

If A_r is negative, the design is adequate as is. If A_r is positive, then either a ring must be added or the area of section increased. A_r is the net area required for the ring only!

- If a ring is required, the maximum distance to centroid of ring, L_b

  $L_b = .25\,(R\,t_2)^{1/2}$

- If the shell is made thicker to provide more area for reinforcement, then the maximum length of shell included in the reinforcement shall be;

  $L_a = (R\,t_2)^{1/2}$

If the Large End is a Line of Support (LOS)

1. Assume size of ring;
   Size = _____
   A = ______
   I = _______
2. Determine length of cylinder acting with cone, L_L
   L_L = ______
3. Determine length of cone, L_C

   $$L_C = \left[L^2 + (R - r)^2\right]^{1/2}$$

4. Calculate equivalent area of cylinder, cone and ring, A_{TL}

   $$A_{TL} = .5\,(L_L\,t_2) + .5\,(L_C\,t_C) + A_S$$

5. Calculate Value M

   $$M = .5\,(-R\tan\alpha) + .5\,L_L + \left[(R^2 - r_2)/\times(3\,R\tan\alpha)\right]$$

6. Calculate Value F_L

   $$F_L = P_X\,M + f_n\,\mathrm{Tan}\,\alpha$$

   Choose f_n as worst value, compression
7. Calculate Factor B

   $$B = .75\,[(F_L\,D_L)/A_{TL}]$$

8. Find Factor A from applicable material curve or calculate as follows;

   $$A = 2\,B\,/E_n$$

   Where E_n lesser of E_S, E_C or E_r
9. Required moment of inertia, I_S or I_S'

   $$I_S = \left[(A\,D_L^2\,A_{TL})/14\right]$$

   $$I_S' = \left[(A\,D_L^2\,A_{TL})/10.9\right]$$

10. Compare required values of I with actual;

    $I > I_S$

    $I' > I_S'$

SMALL END

Case 1: External Loads are Not Included

$$A_{rs} = (P_X\,r^2\,K\,\mathrm{Tan}\,\alpha)/2\,X_1$$

Case 2: External Loads are Included

$$A_{rs} = (K\,Q_L\,r\,\mathrm{Tan}\,\alpha)/X_1$$

From the worksheet, select the largest value of Q_L for the small end in compression.

Reinforcement Available

- Effective area available at small end, A_{eS}

  $$A_{eS} = .55\,(t_1 - t_S)(D_S\,t_1)^{1/2} + (t_C - t_r/\cos\alpha)$$

- Area required at large end, A_r

  $A_r = A_{rS} - A_{eS}$

If A_r is negative, the design is adequate as is. If A_r is positive, then either a ring must be added or the area of section increased. A_r is the net area required for the ring only!

- If a ring is required, the maximum distance to centroid of ring, L_b

  $L_b = .25\,(r\,t_1)^{1/2}$

- If the shell is made thicker to provide more area for reinforcement, then the maximum length of shell included in the reinforcement shall be;

  $L_a = (r\,t_1)^{1/2}$

If the Small End is a Line of Support (LOS)

1. Assume size of ring;
   Size = _____
   A = ______
   I = _______
2. Determine length of cylinder acting with cone, L_S
   L_S = ______

3. Determine length of cone, L_C

$$L_C = \left[L^2 + (R - r)^2\right]^{1/2}$$

4. Calculate equivalent area of cylinder, cone and ring, A_{TS}

$$A_{TS} = .5\,(L_S\, t_1) + .5\,(L_C\, t_C) + A_S$$

5. Calculate Value N

$$N = .5\,(r \tan\alpha) + .5\,L_S + \left[(R^2 - r^2)/ \times (6\, r \tan\alpha)\right]$$

6. Calculate Value F_S

$$F_S = P_X\, N + f_n \tan\alpha$$

Choose f_n as worst value, compression

7. Calculate Factor B

$$B = .75\,[(F_S\, D_S)/A_{TS}]$$

8. Find Factor A from applicable material curve or calculate as follows;

$$A = 2\,B/E_n$$

Where E_n lesser of E_S, E_C or E_r

9. Required moment of inertia, I_S or I_S'

$$I_S = [(A\, D_S^2\, A_{TS})/14]$$

$$I_S' = [(A\, D_S^2\, A_{TS})/10.9]$$

10. Compare required values of I with actual;

$I > I_S$

$I' > I_S'$

3.0. Design of Cones with Half Apex Angle, α, Between 30° and 60°, Internal Pressure Only!

Based on CC2150

LARGE END;

- Required thickness of shell;

$$t_{S2} = (P\,R)/(S_S\, E_1 - .6P)$$

- Required thickness of cone

$$t_{C2} = (P\,R)/(\cos\alpha\,(S_C\, E_2 - .6P))$$

CHECK STRESSES

- Determine ratio;

$R/t_2 =$

- Determine values of factors, X and Y, from the appropriate figure. Use Fig 2-10 if shell and cone are the same thickness. Use Fig 2-11 if the shell and cone are not the same thickness.

X =

Y =

- Calculate longitudinal stress, σ_X

$$\sigma_X = (P\,R/t_2)\left[.5 + X\,(R/t_2)^{1/2}\right]$$

- Calculate circumferential stress, σ_ϕ

$$\sigma_\phi = (P\,R/t_2)\left[1 - Y\,(R/t_2)^{1/2}\right]$$

- Allowable stresses;

$\sigma_X < 3\,S_S$

$\sigma_\phi < 1.5\,S_S$

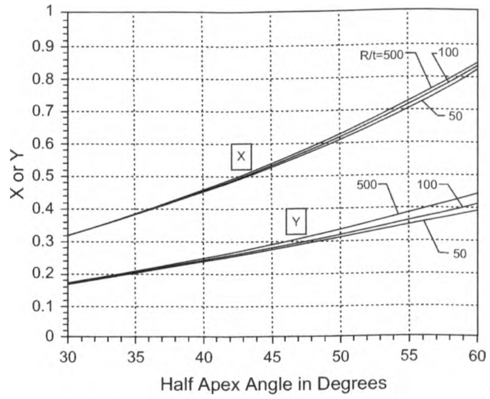


Figure 2-10. *X* and *Y* for cone thickness = t (Use when Cone and Shell are the same thickness)

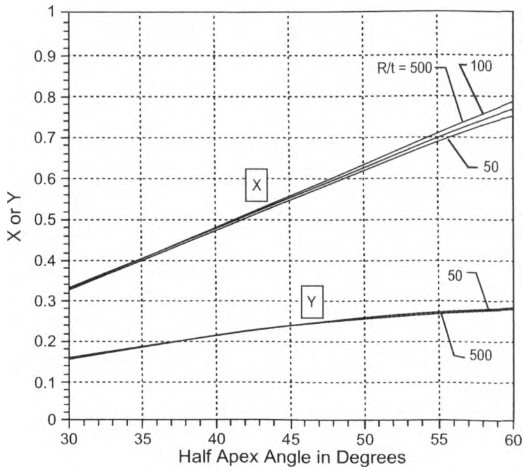


Figure 2-11. *X* and *Y* for cone thickness = t/cos α (Use when Cone and Shell are the not same thickness)

Sample Problem

$\alpha = 45°$
$P = 500$ psig
$D = 120$ in
$R = 60.125$ in Corr
$DT = 600°F$
Matl: SA-516-70
$S_S = 19{,}400$ PSI
$E = 1$
$Ca = 0.125$ in
Code: ASME VIII-1

- Required thickness of Shell;

$$t_{S2} = (P\,R)/(S_S\,E_1 - .6P)$$
$$= (500\,(60.125))/(19400 - 300)$$
$$= 1.57 + .125 = 1.699$$

Use $t_2 = 1.75$ in (1.625 in Corr)

- Required thickness of cone

$$t_{C2} = (P\,R)/[\cos\alpha\,(S_C\,E - .6P)]$$
$$= (500\,(60.125))/[\cos 45\,(19400 - 300)]$$
$$= 2.22 + .125 = 2.35$$

Use $t_C = 2.375$ in (2.25 in Corr)

CHECK STRESSES

- Determine ratio;

$$R/t_2 = 60.125/1.625 = 37$$

- Determine values of factors, X and Y, from Fig 2-11

$X = .53$
$Y = .24$

- Calculate longitudinal stress, σ_X

$$\sigma_X = \left(P\,R/t_2\right)\left[.5 + X\,(R/t_2)^{1/2}\right]$$
$$= \left[500\left(60.125\right)/1.625\right]$$
$$\left[.5 + .53\,(60.125/1.625)^{1/2}\right]$$
$$= 68{,}691 \text{ PSI}$$

Allowable stress $= 3\ S_S = 3\ (19{,}400) = 58{,}200$ PSI

Therefore design is not acceptable!

Increase shell thickness by 0.125 in and recalculate stresses

- Calculate circumferential stress, σ_ϕ

$$\sigma_\phi = \left(P\,R/t_2\right)\left[1 - Y\,(R/t_2)^{1/2}\right]$$
$$= \left[500\left(60.125\right)/1.625\right]$$
$$\left[1 - .24\,(60.125/1.625)^{1/2}\right]$$
$$= (-)\ 8{,}507 \text{ PSI (Acceptable)}$$

Notes

1. For small vessels or horizontal vessels with cones where there are not significant external loads, then the external loads may be neglected.
2. Line of Support: This indicates a formal work line or an elevation on the vessel used for external pressure calculations. It does not mean that the vessel is supported from that point.
3. The worksheet should be used to determine the loadings at the plane being investigated.
4. External Load Conditions;

INTERNAL PRESSURE:

a: Large End: $.5PR + f_n$ is in tension, use this value in the procedure. If this value is (-), in compression, then a stress analysis is required.

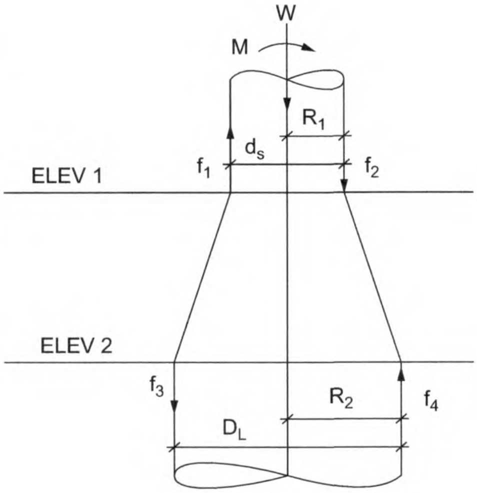
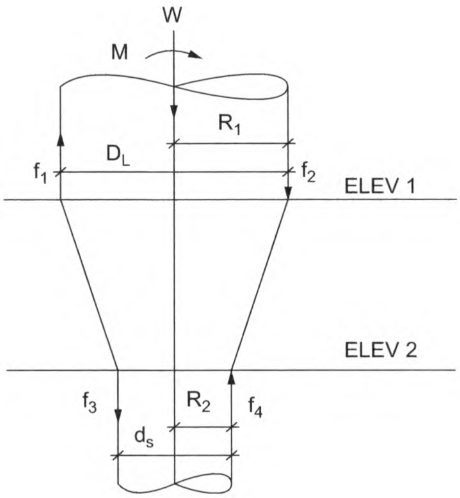
b: Small End: $.5PR + f_n$ is in tension, use this value in the procedure. If this value is (-), in compression, then a stress analysis is required.

EXTERNAL PRESSURE:

c: Large End: $.5P_x\,R + f_n$ is in compression, use this value in the procedure. If this value is (+), in tension, then a stress analysis is required.

d: Large End: $.5P_x\,R + f_n$ is in compression, use this value in the procedure. If this value is (+), in tension, then a stress analysis is required.

WORKSHEET TO DETERMINE LOADS AT CONE-CYLINDER INTERSECTIONS

CASE 1	CASE 2
	

Loads at Elevations due to External Forces, f_n (2)

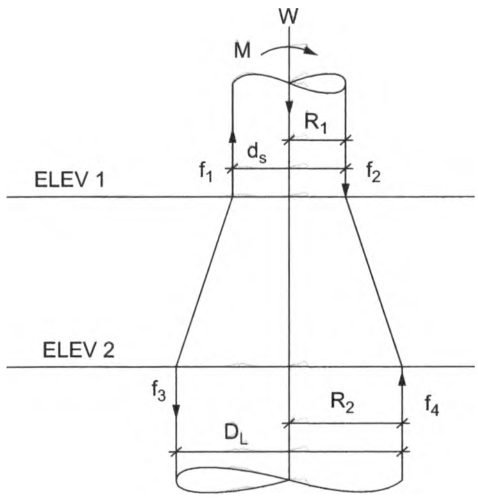
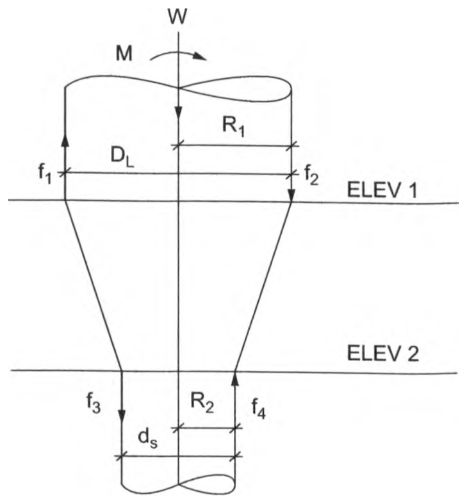
	CASE 1			CASE 2	
ELEV / SMALL END	$f_1 = \frac{-W_1}{\pi d_s} + \frac{4M_1}{\pi d_s^2}$		LARGE END	$f_1 = \frac{-W_1}{\pi D_L} + \frac{4M_1}{\pi D_L^2}$	
	$f_2 = \frac{-W_1}{\pi d_s} - \frac{4M_1}{\pi d_s^2}$			$f_2 = \frac{-W_1}{\pi D_L} - \frac{4M_1}{\pi D_L^2}$	
ELEV 2 / LARGE END	$f_3 = \frac{-W_2}{\pi D_L} + \frac{4M_2}{\pi D_L^2}$		SMALL END	$f_3 = \frac{-W_2}{\pi d_s} + \frac{4M_2}{\pi d_s^2}$	
	$f_4 = \frac{-W_2}{\pi D_L} - \frac{4M_2}{\pi D_L^2}$			$f_4 = \frac{-W_2}{\pi d_s} - \frac{4M_2}{\pi d_s^2}$	

TOTAL LOAD, Q_L (1)			DATA			
	INT	EXT	W_1		D_L	
Q_{L1} = +/- .5 $P_n R_1$ +/- f_1			W_2		d_s	
Q_{L2} = +/- .5 $P_n R_1$ +/- f_2			M_1		R_1	
Q_{L3} = +/- .5 $P_n R_2$ +/- f_3			M_2		R_2	
Q_{L4} = +/- .5 $P_n R_2$ +/- f_4			P		P_x	

NOTES:

1. The expression (.5 $P_n R_n$) is (+) for internal pressure and (-) for external pressure. The sign of f_n is + or - based on the results of the calculation above
2. f_1 and f_3 are (-) if uplift due to moment is < weight. f1 and f3 are (+) if uplift due to moment is > weight.

WORKSHEET TO DETERMINE LOADS AT CONE-CYLINDER INTERSECTIONS (EXAMPLE)

CASE 1	CASE 2
	

Loads at Elevations due to External Forces, f_n (2)

ELEV	SMALL END	$f_1 = \dfrac{-W_1}{\pi d_s} + \dfrac{4M_1}{\pi d_s^2}$	(+) 735 Lbs/In	LARGE END	$f_1 = \dfrac{-W_1}{\pi D_L} + \dfrac{4M_1}{\pi D_L^2}$	
		$f_2 = \dfrac{-W_1}{\pi d_s} - \dfrac{4M_1}{\pi d_s^2}$	(-) 1121 Lbs/In		$f_2 = \dfrac{-W_1}{\pi D_L} - \dfrac{4M_1}{\pi D_L^2}$	
ELEV 2	LARGE END	$f_3 = \dfrac{-W_2}{\pi D_L} + \dfrac{4M_2}{\pi D_L^2}$	(+) 180 Lbs/In	SMALL END	$f_3 = \dfrac{-W_2}{\pi d_s} + \dfrac{4M_2}{\pi d_s^2}$	
		$f_4 = \dfrac{-W_2}{\pi D_L} - \dfrac{4M_2}{\pi D_L^2}$	(-) 396 Lbs/In		$f_4 = \dfrac{-W_2}{\pi d_s} - \dfrac{4M_2}{\pi d_s^2}$	

TOTAL LOAD, Q_L (1)			DATA			
	INT	EXT	W_1	36,500 Lbs	D_L	120.438"
Q_{L1} = +/- .5 P_n R_1 +/- f_1	(+)1489 Lbs/ in	(+) 622 Lbs / in	W_2	41,100 Lbs	d_s	60.3125"
Q_{L2} = +/- .5 P_n R_1 +/- f_2	(-)367 Lbs/in	(-) 1234 Lbs/in	M_1	2,652,000 in-Lbs	R_1	30.156"
Q_{L3} = +/- .5 P_n R_2 +/- f_3	(+) 1685 Lbs/in	(-) 45 Lbs/in	M_2	3,288,000 in-Lbs	R_2	60.219"
Q_{L4} = +/- .5 P_n R_2 +/- f_4	(+) 1109 Lbs/in	(-) 621 Lbs/in	P	50 PSIG	P_x	7.5 PSIG

NOTES:

1. The expression (.5 P_n R_n) is (+) for internal pressure and (-) for external pressure. The sign of f_n is + or - based on the results of the calculation above
2. f_1 and f_3 are (-) if uplift due to moment is < weight. f1 and f3 are (+) if uplift due to moment is > weight.

Procedure 2-6: Design of Toriconical Transitions [1,3]

Notation

P = internal pressure, psi
S = allowable stress, psi
E = joint efficiency
P_1, P_2 = equivalent internal pressure, psi
f_1, f_2 = longitudinal unit loads, lb/in.
σ_1, σ_2 = circumferential membrane stress, psi
α = half apex angle, deg
m = code correction factor for thickness of large knuckle
P_x = external pressure, psi
M_1, M_2 = longitudinal bending moment at elevation, in.-lb
W_1, W_2 = dead weight at elevation, lb

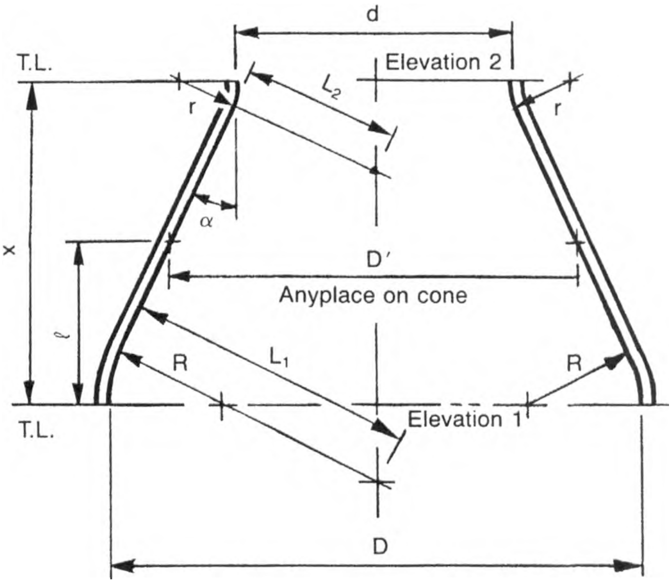
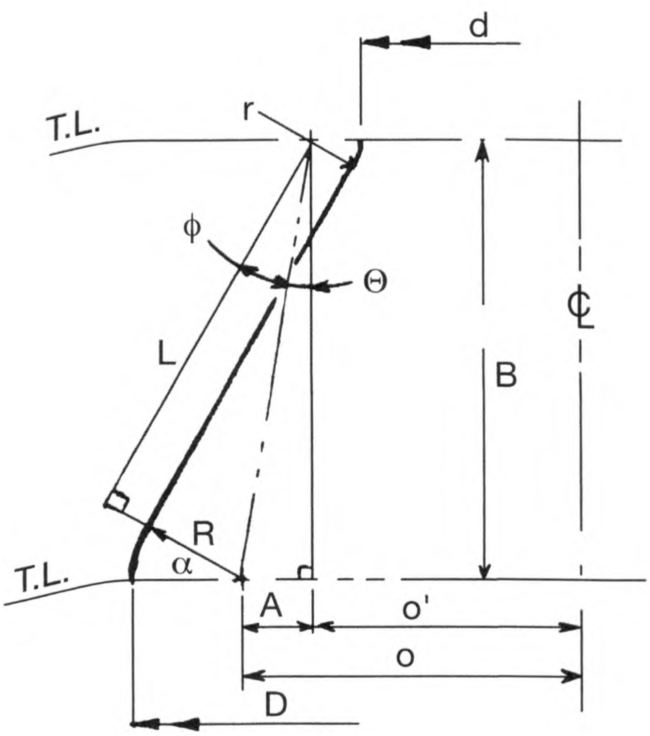


Figure 2-12. Dimensional data for a conical transition.

Calculating Angle, α



Case 1

$$o > o'$$

Step 1:

$$\sin \phi = \frac{R + r}{\sqrt{A^2 + B^2}} =$$

$$\phi = \_\_\_\_\_\_\_\_$$

Step 2:

$$\tan \Theta = \frac{A}{B}$$

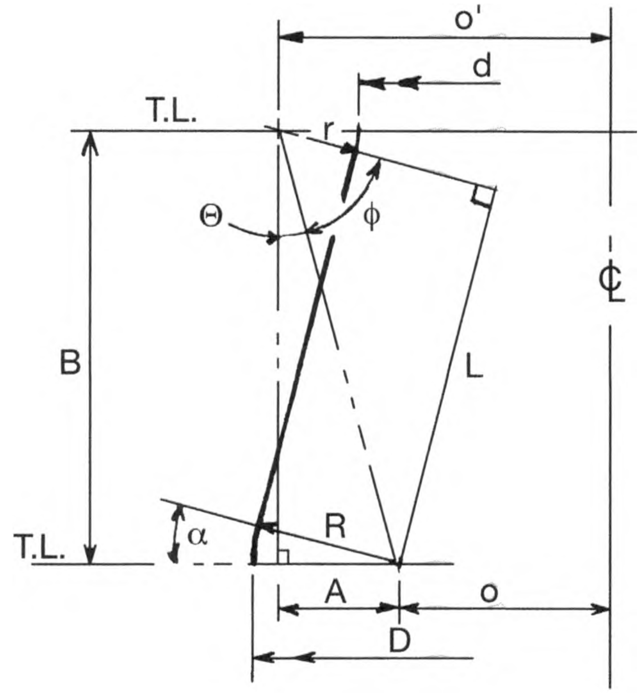
$$\Theta = \_\_\_\_\_\_\_\_$$

Step 3:

$$\alpha = \phi + \Theta$$

$$\alpha = \_\_\_\_\_\_\_\_$$

$$L = \cos \phi \sqrt{A^2 + B^2}$$



Case 2

$$o' > o$$

Step 1:

$$\cos\phi = \frac{R + r}{\sqrt{A^2 + B^2}} =$$

$$\phi = \_\_\_\_\_\_\_\_\_\_$$

Step 2:

$$\tan\Theta = \frac{A}{B}$$

$$\Theta = \_\_\_\_\_\_\_\_\_\_$$

Step 3:

$$\alpha = 90 - \Theta - \phi$$

$$\alpha = \_\_\_\_\_\_\_\_\_\_$$

$$L = \cos\phi\sqrt{A^2 + B^2}$$

Dimensional Formulas

$$D_1 = D - 2(R - R\cos\alpha)$$

$$D_2 = D + 2(R - R\cos\alpha)$$

$$D' = D - 2R\left(1 - \frac{1}{\cos\alpha}\right) - 2\ell\tan\alpha$$

$$L_1 = \frac{D_1}{2\cos\alpha}$$

$$L_2 = \frac{D_2}{2\cos\alpha}$$

$$m = 0.25\left(3 + \sqrt{\frac{L_1}{R}}\right)$$

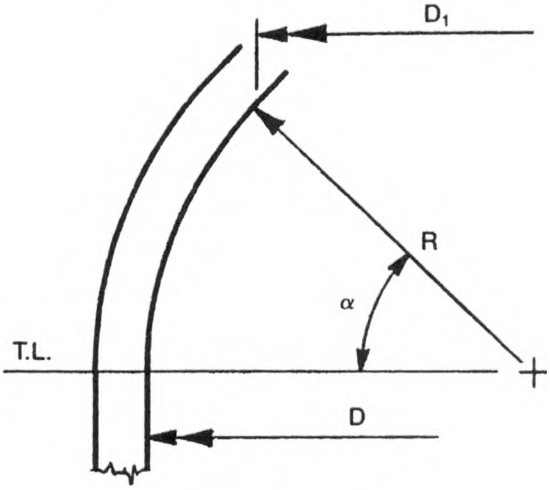


Figure 2-13. Dimensional data for the large end of a conical transition.

Large End (Figure 2-13)

- *Maximum longitudinal loads, f_1*
  (+) tension; (–) compression

$$f_1 = \frac{-W_1}{\pi D_1} \pm \frac{4M_1}{\pi D_1^2}$$

- *Determine equivalent pressure, P_1.*

$$P_1 = P + \frac{4f_1}{D_1}$$

- *Circumferential stress, D_1.*
  Compression:

$$\sigma_1 = \frac{PL_1}{t} - \frac{P_1L_1}{t}\left[\frac{L_1}{2R}\right]$$

- *Circumferential stress at D_1 without loads, σ_1.* Compression:

$$\sigma_1 = \frac{PL_1}{t}\left(1 - \frac{L_1}{2R}\right)$$

- *Thickness required knuckle, t_{rk}* [1, section 1-4(d)]. With loads:

$$t_{rk} = \frac{P_1L_1m}{2SE - 0.2P_1}$$

Without loads:

$$t_{rk} = \frac{PL_1m}{2SE - 0.2P}$$

- *Thickness required cone, t_{rc}* [1, section UG-32(g)]. With loads:

$$t_{rc} = \frac{P_1D_1}{2\cos\alpha(SE - 0.6P_1)}$$

Without loads:

$$t_{rc} = \frac{PD_1}{2\cos\alpha(SE - 0.6P)}$$

Small End (Figure 2-14)

- *Maximum longitudinal loads, f_2.*

(+) tension; (–) compression

$$f_2 = \frac{-W_2}{\pi D_2} \pm \frac{4M_2}{\pi D_2^2}$$

- *Determine equivalent pressure, P_2.*

$$P_2 = P + \frac{4f_2}{D_2}$$

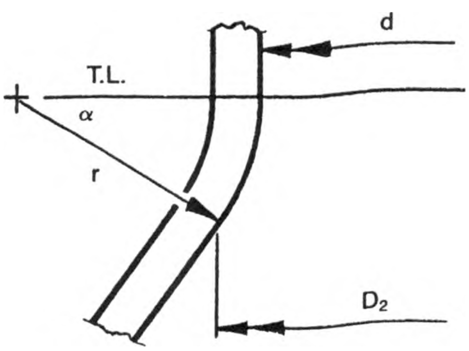


Figure 2-14. Dimensional data for the small end of a conical transition.

- *Circumferential stress at D_2.*

Compression:

$$\sigma_2 = \frac{PL_2}{t} + \frac{P_2L_2}{t}\left[\frac{L_2}{2r}\right]$$

- *Circumferential stress at D_2 without loads, σ_2.*

Compression:

$$\sigma_2 = \frac{PL_2}{t}\left(1 - \frac{L_2}{2r}\right)$$

- *Thickness required cone, at D_2, t_{rc}* [1, section UG-32(g)]. With loads:

$$t_{rc} = \frac{P_2D_2}{2\cos\alpha(SE - 0.6P_2)}$$

Without loads:

$$t_{rc} = \frac{PD_2}{2\cos\alpha(SE - 0.6P)}$$

- *Thickness required knuckle.* There is no requirement for thickness of the reverse knuckle at the small end of the cone. For convenience of fabrication it should be made the same thickness as the cone.

Additional Formulas (Figure 2-15)

- *Thickness required of cone at any diameter D', $t_{D'}$*

$$t_{D'} = \frac{PD'}{2\cos\alpha(SE - 0.6P)}$$

- *Thickness required for external pressure* [1, section UG-33(f)].

$t_e = t\cos\alpha$

$D_L = D_2 + 2t_e$

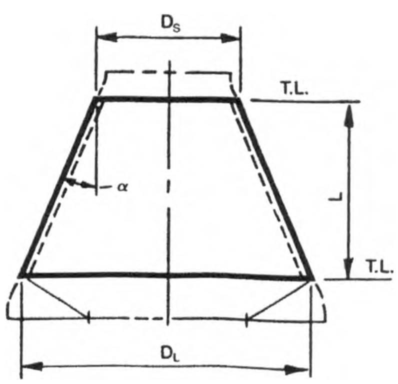


Figure 2-15. Dimensional data for cones due to external pressure.

$$D_s = D_l + 2t_e$$
$$L = X - \sin\alpha(R + t) - \sin\alpha(r - t)$$
$$L_e = \frac{L}{2}\left(1 + \frac{D_S}{D_L}\right)$$
$$\frac{L_e}{D_L} =$$
$$\frac{D_L}{t_e} =$$

Using these values, use Figure 2-1e to determine Factor A.

- *Allowable external pressure, P_a.*

$$P_a = \frac{2AEt_e}{D_L}; \quad P_a > P_x$$

where E = modulus of elasticity at design temperature.

Notes

1. *Allowable stresses.* The maximum stress is the compressive stress at the tangency of the large knuckle and the cone. Failure would occur in local yielding rather than buckling; therefore the allowable stress should be the same as required for cylinders. Thus the allowable circumferential compressive stress should be the lesser of 2SE or F_y. Using a lower allowable stress would require the knuckle radius to be made very large—well above code requirements. See Reference 3.
2. Toriconical sections are mandatory if angle α exceeds 30° unless the design complies with Para. 1-5(e) of the ASME Code [1]. This paragraph requires a discontinuity analysis of the cone-shell juncture.
3. No reinforcing rings or added reinforcement is required at the intersections of cones and cylinders, providing a knuckle radius meeting ASME Code requirements is used. The minimum knuckle radius for the large end is not less than the greater of 3t or 0.12(R + t). The knuckle radius of the small end (flare) has no minimum. (See [Reference 1, Figure UG-36]).
4. Toriconical transitions are advisable to avoid the high discontinuity stresses at the junctures for the following conditions:
   a. High pressure—greater than 300 psig.
   b. High temperature—greater than 450 or 500°F.
   c. Low temperature—less than –20°F.
   d. Cyclic service (fatigue).

Procedure 2-7: Stresses in Heads Due to Internal Pressure [2,3]

Notation

L = crown radius, in.
r = knuckle radius, in.
h = depth of head, in.
R_L = latitudinal radius of curvature, in.
R_m = meridional radius of curvature, in.
σ_ϕ = latitudinal stress, psi
σ_x = meridional stress, psi
P = internal pressure, psi

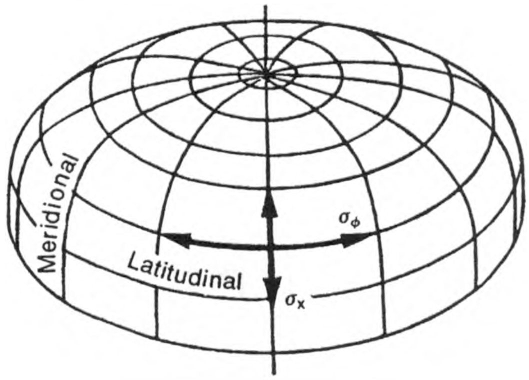


Figure 2-16. Direction of stresses in a vessel head.

Formulas

Lengths of R_L and R_m for ellipsoidal heads:

- At equator:

$$R_m = \frac{h^2}{R}$$

$$R_L = R$$

- At center of head:

$$R_m = R_L = \frac{R^2}{h}$$

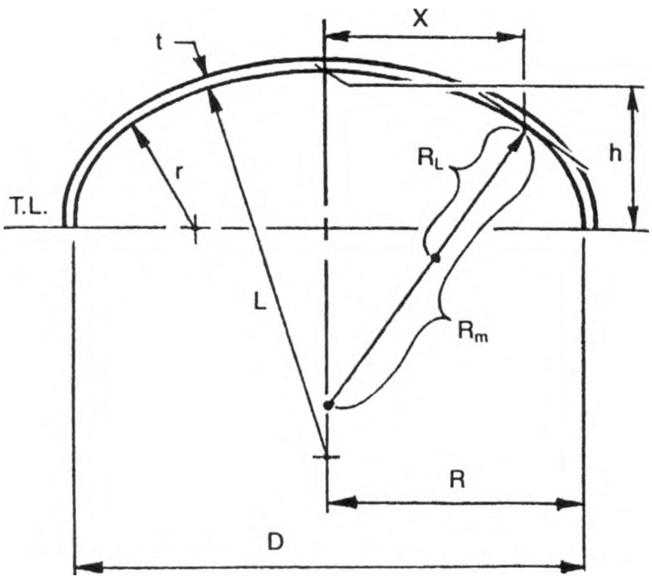


Figure 2-17. Dimensional data for a vessel head.

- At any point X:

$$R_L = \sqrt{\frac{R^4}{h^2} + X^2\left(1 - \frac{R^2}{h^2}\right)}$$

$$R_m = \frac{R_L^3 h^2}{R^4}$$

Notes

1. Latitudinal (hoop) stresses in the knuckle become compressive when the R/h ratio exceeds 1.42. These heads will fail by either elastic or plastic buckling, depending on the R/t ratio.
2. Head types fall into one of three general categories: hemispherical, torispherical, and ellipsoidal. Hemispherical heads are analyzed as spheres and were covered in the previous section. Torispherical (also known as flanged and dished heads) and ellipsoidal head formulas for stress are outlined in the following form.

TORISPHERICAL HEADS			
σ_x		σ_ϕ	
At Junction of Crown and Knuckle			
$\sigma_x = \frac{PL}{2t}$		$\sigma_\phi = \frac{PL}{4t}\left(3 - \frac{L}{R}\right)$	
In Crown			
$\sigma_x = \frac{PL}{2t}$		$\sigma_\phi = \sigma_x$	
In Knuckle			
See following section		See following section	
At Tangent Line			
$\sigma_x = \frac{PL}{2t}$		$\sigma_\phi = \frac{PR}{t}$	

ELLIPSOIDAL HEADS			
σ_x		σ_ϕ	
At Any Point X			
$\sigma_x = \frac{PR_L}{2t}$		$\sigma_\phi = \frac{PR_t}{t}\left(1 - \frac{R_L}{2R_m}\right)$	
At Center of Head			
$\sigma_x = \frac{PR^2}{2th}$		$\sigma_\phi = \sigma_x$	
At Tangent Line			
$\sigma_x = \frac{PR}{2t}$		$\sigma_\phi = \frac{PR}{t}\left(1 - \frac{R^2}{2h^2}\right)$	

CIRCUMFERENTIAL COMPRESSION STRESS IN KNUCKLE REGION OF TORISPHERICAL HEAD DUE TO INTERNAL PRESSURE

DATA
B = ASME Code allowable compressive stress, PSI
C.a. = Corrosion allowance, in
D = Inside diameter, in
DT = Design temperature, °F
F_C = Allowable compressive stress, PSI
K = ASME Code coefficient
L = Crown radius, use .9 D for 2:1 S.E. heads, in
M = ASME Code coefficient
P = Design pressure, PSIG
r = Knuckle radius, in
S = Allowable stress, tension, PSI
t = Head thickness, new, in
t_C = Head thickness, corroded, in
σ_ϕ = Circumferential stress, PSI

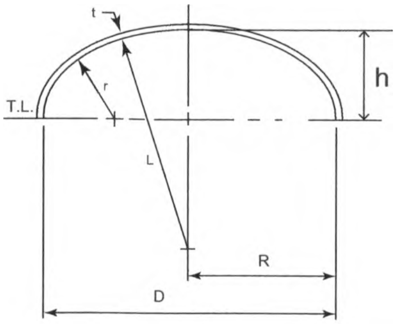
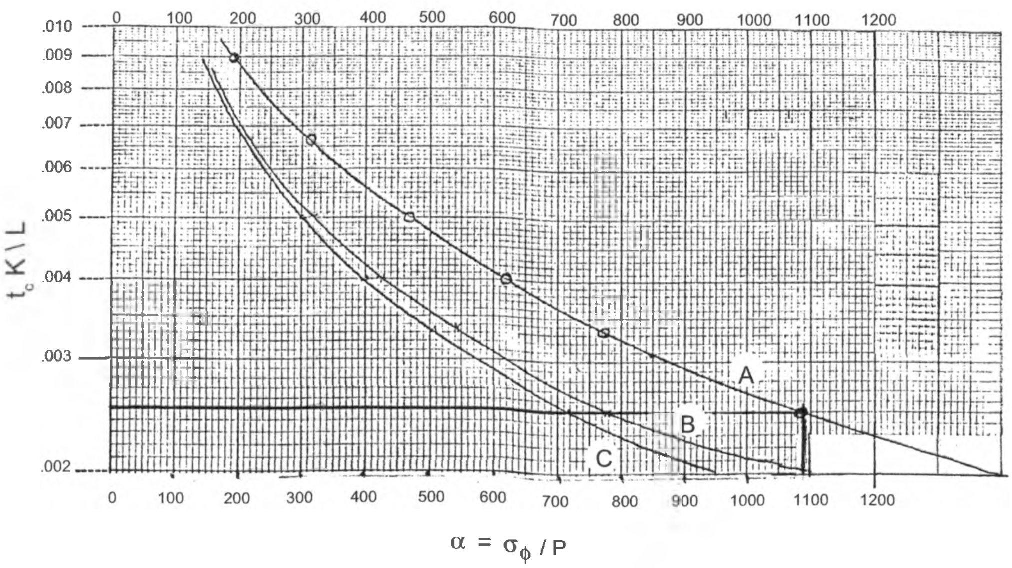


Figure 2.18 Dimensions of heads



Figures 2.19 Plot of head parameters

This procedure allows the designer to either check the compressive stress in an existing head, or to determine the thickness required for a new head. It should be noted that, the calculated thickness for internal pressure may not be adequate for the buckling stress in the knuckle. The example shown clearly illustrates this point.

Table 2-5
Head values for Figure 2.19

Ratio or Factor	Curve A	Curve B	Curve C
L / D	1.0	.9	.8
R / D	.06	.1	.1
M	1.77	1.5	1.46
K	1.13	1.33	1.37

Calculations

- Calculate coefficients K & M;

$$K = .167\left[2\,(D/2\,h)^2\right]$$

$$M = .25\left[3 + (L/r)^{1/2}\right]$$

- Determine ratios,
  L/D = ____
  r/D = _____
  L/r = ______
  L/K = ______
  t_C K / L = ______
- From Figure 2.19, using the appropriate curve, determine α
  α = _______
- Using value α, calculate σ_ϕ

$$\sigma_\phi = P\,\alpha$$

ALLOWABLE STRESS;

F_C = Lesser of 1.5 S or 1.5 B

To find factor B...
$A = .125\ t_C/r$
B = From Chart
Use ______

EXAMPLE;

Material: SA-516-70
Head Proportions: 100% - 6%
D = 114 in
P = 25 PSIG
DT = 500°F
S = 20,000 PSI
E = 1.0
C.a. = .125 in
L = 114 in
R = 6.875 in
L/r = 16.58

- Thickness required for internal pressure, t_r

$$M = .25\left[3 + (L/r)^{1/2}\right] = 1.768$$

$$t_r = (PLM)/(2SE - .2P)$$
$$= (25\,(114)\,1.768)/(2\,(20{,}000)1.0 - 5)$$
$$= .126 + .125 = .251 \text{ Use } .375 \text{ in}$$
$$t_C = .25 \text{ in}$$

- Determine circumferential stress, σ_ϕ

$t_C\,K/L = .25\,(1.13)/114 = .00248$
From curve A; $\alpha = \sigma_\phi$ / P = 1090

$$\sigma_\phi = 1090\,P = 1090\,(25) = 27{,}250 \text{ PSI}$$

- Allowable stress,
  Lesser of...

$1.5\,S = 1.5 \times 20{,}000 = 30{,}000$ PSI
$A = .125\,t_C/r = .125\,(.25)/6.875$
$= .00454$
B = 13,000 PSI
1.5 B = 19,500 PSI

Therefore design is not adequate. Increase thickness and recalculate;

TRIAL 2;

- Determine required σ_ϕ / P ratio;

$$\sigma_\phi/P = 19{,}500/25 = 780$$

- Find the t_C K / L ratio from graph for

$\sigma_\phi/P = 780$
$t_C\,K/L = .0033$

- Therefore t_r = .0033 (114) / 1.13 = .332
  The new thickness would be .332 + .125 = .458, use .5 in
- Check;

$t_C = .5 - .125 = .375$
$t_C\,K/L = .00372$
$\sigma_\phi/P = 690$
$\sigma_\phi = 690\,P = 690\,(25) = 17{,}250$ PSI

Therefore design is acceptable.

Procedure 2-8: Design of Intermediate Heads [1,3]

Notation

A = factor A for external pressure
A_s = shear area, in.^{2}
B = allowable compressive stress, psi
F = load on weld(s), lb/in.
τ = shear stress, psi
E = joint efficiency
E_1 = modulus of elasticity at temperature, psi
S = code allowable stress, psi
H_D = hydrostatic end force, lb
P_i = maximum differential pressure on concave side of head, psi
P_e = maximum differential pressure on convex side of head, psi
K = spherical radius factor (see Table 2-6)
L = inside radius of hemi-head, in
= 0.9D for 2:1 S.E. heads, in
= KD for ellipsoidal heads, in
= crown radius of F & D heads, in

Table 2-6
Spherical radius factor, K

$\frac{D}{2h}$	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6	2.8	3.0
K	.5	.57	.65	.73	.81	.9	.99	1.08	1.18	1.27	1.36

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Required Head Thickness, t_r

- *Internal pressure, P_i.* Select appropriate head formula based on head geometry. For dished only heads as in Figure 2-21, Case 3:

$$t_r = \frac{5P_iL}{6S}$$

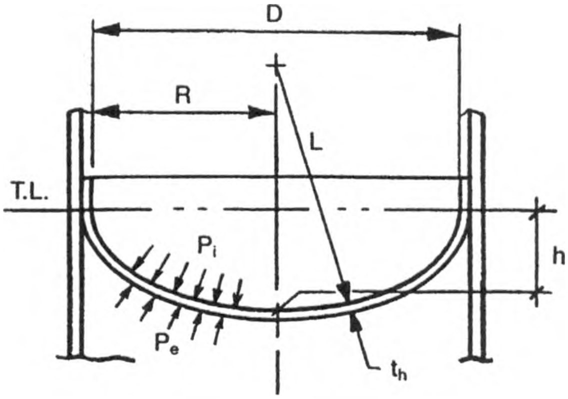


Figure 2-20. Dimensional data for an intermediate head.

- *External pressure, P_e.* Assume corroded head thickness, t_h

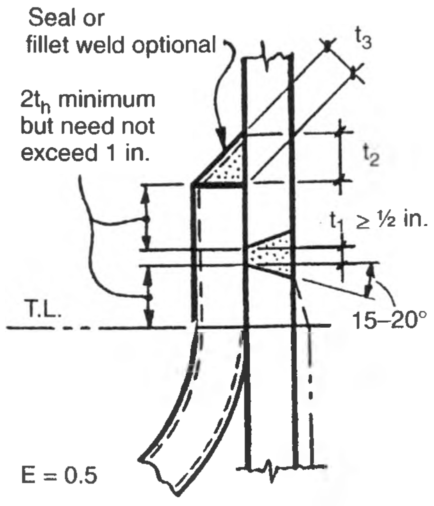
$$\text{Factor A} = \frac{0.125t_h}{L}$$

Factor B can be taken from applicable material charts in Section II, Part D, Subpart 3 of Reference 1. Alternatively (or if Factor A lies to the left of the material/temperature line):

$$B = \frac{AE_1}{2}$$

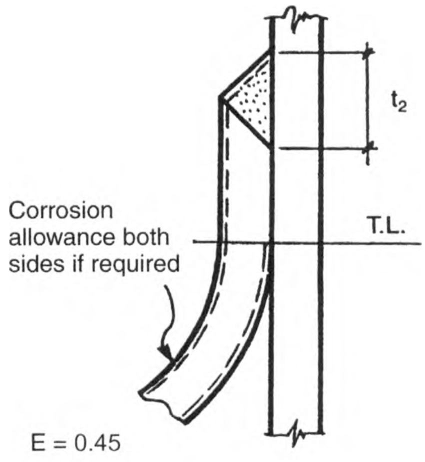
$$t_r = \frac{P_eL}{B}$$

The required head thickness shall be the greater of that required for external pressure or that required for an internal pressure equal to 1.67 × P_e. See Reference 1, Para. UG-33(a).



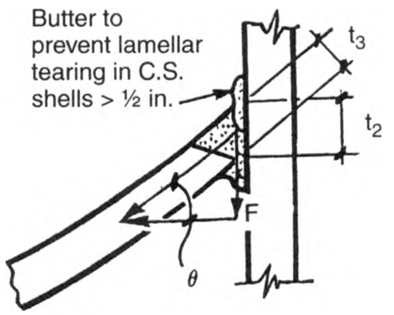
$A_s = t_1$ + lesser of t_2 or t_3

Case 1



$A_s = t_2$

Case 2

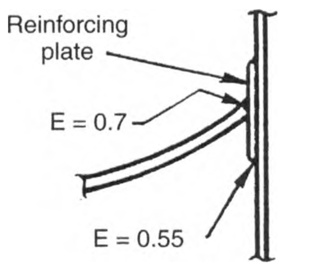


$\sin\theta = \dfrac{D}{2L + t}$

$A_s =$ lesser of t_2 or t_3

E = 0.7 (butt weld)

Case 3



Design the weld attaching the head as in Case 3 and the welds attaching the reinforcing plate to share full load

Case 3 Alternate

Figure 2-21. Methods of attachment of intermediate heads.

Shear Stress

- *Hydrostatic end force, H_D.*

$$H_D = \frac{P\pi D^2}{4}$$

where P = 1.5 ×greater of P_i or P_e. (See Reference 1, Figure UW-13.1.)

- *Shear loads on welds, F.*

$$F = \frac{H_D}{\pi D \sin\theta}$$

Note: sin θ applies to Figure 2-21, Case 3 head attachments only!

- *Shear stress, τ.*

$$\tau = \frac{F}{A_s}$$

- *Allowable shear stress, SE.*

Procedure 2-9: Design of Flat Heads [1,2,4,5,6]

Notation

C = attachment factor
D = long span of noncircular heads, in.
d = diameter of circular heads or short span of non-circular heads, in.
E = joint efficiency (Cat. A seam only)
ℓ = length of straight flange measured from tangent line, in.
P = internal pressure psi
r = inside corner radius of head, in.
S = code allowable stress, tension, psi
t = minimum required thickness of head, in.
t_f = thickness of flange of forged head, in.
t_h = thickness of head, in.
t_r = minimum required thickness of seamless shell, in.
t_s = thickness of shell, in.
t_w = thickness of weld joint, in.
t_p = minimum distance from outside of head to edge of weld prep, in.
Z = factor, dependent on d/D ratio
Q_o = shear force per unit length, lb/in.
N_o = axial tensile force per unit length, lb/in.
M_o = radial bending moment, in.-lb/in.
ν = Poisson's ratio, 0.3 for steel

$a_{1,2,3}$, $b_{1,2,3}$ = Influence coefficients for head
$a_{4,5,6}$, $b_{4,5,6}$ = Influence coefficients for shell

Formulas

- *Circular heads.*

$$t = d\sqrt{\frac{CP}{SE}}$$

- *Noncircular heads.*

$$t = d\sqrt{\frac{ZCP}{SE}}$$

where $Z = 3.4 - \frac{2.4d}{D}; < 2.5$

- *Dimensionless factors.*

$$m = \frac{t_r}{t_s}$$

$$\beta = \sqrt[4]{\frac{12(1-\nu^2)}{d^2 t_s^2}}$$

$$a_1 = (-)3(1-\nu)\frac{d}{t_h}$$

$$a_2 = 2(1-\nu)$$

$$a_3 = \frac{3d(1-\nu)}{32t_h}$$

$$a_4 = (-)\frac{t_h}{t_s}\left[\frac{(\beta d)^2}{2}\right]$$

$$a_5 = (-)\frac{t_h}{t_s}\left(\frac{\beta d}{2}\right)$$

$$a_6 = (-)\frac{t_h}{t_s}\left(\frac{2-\nu}{8}\right)$$

$$b_1 = \frac{6(1-\nu)d^2}{(\beta d)^2 t_s t_h}$$

$$b_2 = (-)\frac{3(1-\nu)d}{(\beta d)^2 t_s}$$

$$b_3 = (-)\frac{3(1-\nu)d^2}{16(\beta d)^2 t_s t_h}$$

$$b_4 = (-)(\beta d)\left(\frac{t_h}{t_s}\right)^2$$

$$b_5 = (-)0.5\left(\frac{t_h}{t_s}\right)^2$$

$$b_6 = 0$$

Cases

Case 1 (Figure 2-22)

1. C = 0.17 for forged circular or noncircular heads.
2. $r \geq 3t_h$
3. C = 0.1 for circular heads if

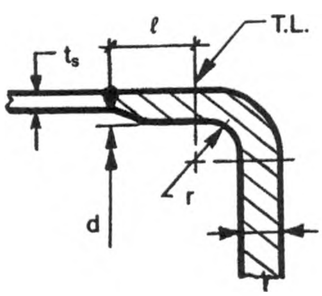


Figure 2-22. Case 1: Flanged head [1, Figure UG-34 (a)].

$$\ell \geq \left(1.1 - \frac{0.8t_s^2}{t_h^2}\right)\sqrt{dt_h}$$

or if the requirement of the flange length is not met and if

$$t_s \geq 1.12t_h\sqrt{1.1 - \frac{\ell}{\sqrt{dt_h}}}$$

for length $2\sqrt{dt_s}$ and taper is 4:1 minimum.

Case 2 (Figure 2-23)

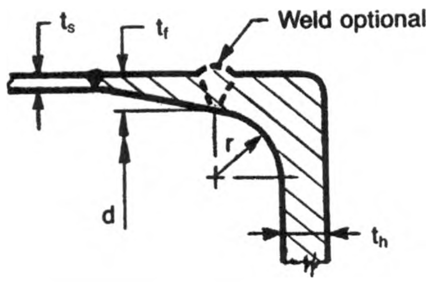


Figure 2-23. Case 2: Forged head [1, Figure UG-34 (b-1)].

1. $C = 0.17$
2. $t_f \geq 2t_s$
3. $r \geq 3t_f$
4. For forged circular or noncircular heads.

Case 3 (Figure 2-24)

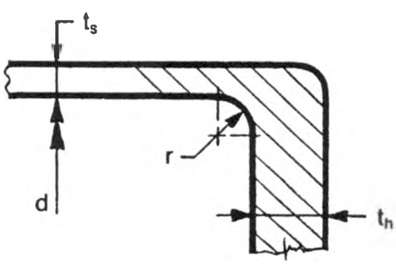


Figure 2-24. Case 3: Integrally forged head [1, Figure UG-34 (b-2)].

1. $C = 0.33$ m but ≥ 0.2
2. $r \geq 0.375$ in. if $t_s \leq 1.5$ in.
3. $r \geq 0.25t_s$ if t_s is greater than 1.5 in. but need not be greater than 0.75 in.

Case 4 (Figure 2-25)

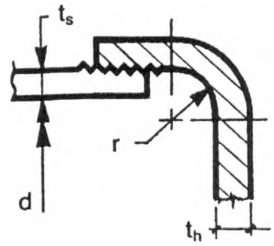


Figure 2-25. Case 4: Screwed flat head [1, Figure UG-34(c)].

1. $C = 0.3$
2. $r \geq 3t_h$
3. Design threads with 4:1 safety factor against failure by shear, tension, or compression due to hydrostatic end force.
4. Seal welding optional.
5. Threads must be as strong as standard pipe threads.

Case 5 (Figure 2-26)

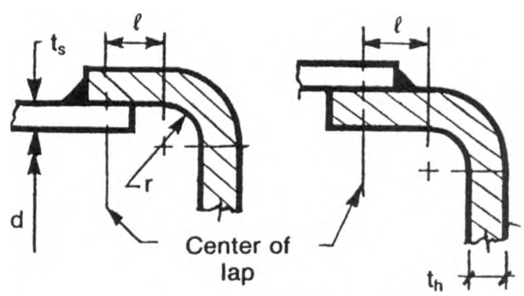


Figure 2-26. Case 5: Lap welded head [1, Figure UG-34(c)].

1. Circular heads: $C = 0.13$ if

$$\ell \geq \left(1.1 - \frac{0.8t_s^2}{t_h^2}\right)\sqrt{dt_h}$$

2. Noncircular heads and circular heads regardless of ℓ: $C = 0.2$.
3. $r \geq 3t_h$

Case 6 (Figure 2-27)

1. $C = 0.13$
2. $d \leq 24$ in.

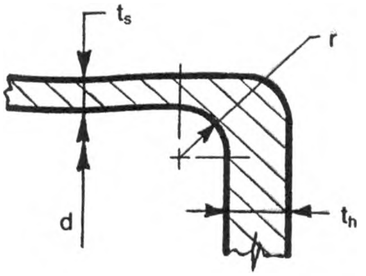


Figure 2-27. Case 6: Integrally forged head [1, Figure UG-34(d)].

3. $0.05 \leq t_h/d \leq 0.25$
4. $t_h \geq t_s$
5. $r \geq 0.25t_h$
6. Head integral with shell by upsetting, forging, or spinning.
7. Circular heads only.

Case 7 (Figure 2-28)

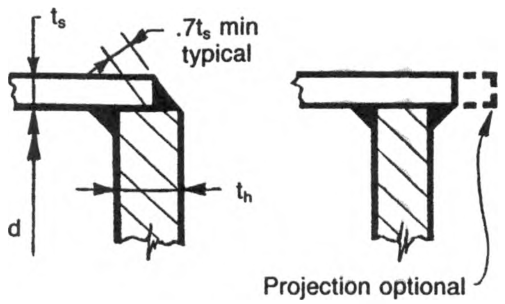


Figure 2-28. Case 7: Welded flat heads [1, Figure UG-34(e)(f)].

1. Circular heads: $C = 0.33$ m but ≥ 0.2. If $m < 1$, then shell cannot be tapered within $2\sqrt{dt_s}$ from inside of head.
2. Noncircular heads: $C = 0.33$
3. Liquid penetrant (L.P.) or magnetic particle test (M.T.) end of shell and O.D. of head if t_s or t_h is greater than 1/2 in. thick (before and after welding).

Case 8 (Figure 2-29)

1. Circular heads: $C = 0.33$ m but ≥ 0.2.

   $t_w \geq 2t_r$ and $\geq 1.25t_s$ but $\leq t_h$

If $m < 1$, then shell cannot be tapered within $2\sqrt{dt_s}$ from inside of head.

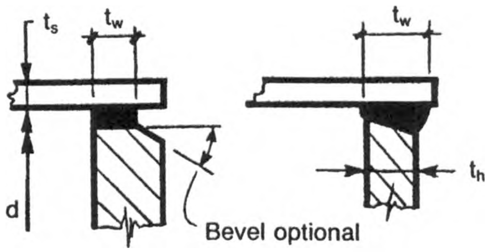


Figure 2-29. Case 8: Welded flat heads (Full penetration welds required) [1, Figure UG-34(g)].

2. Noncircular heads: $C = 0.33$
3. See Note 3 in Case 7.

Case 9 (Figure 2-30)

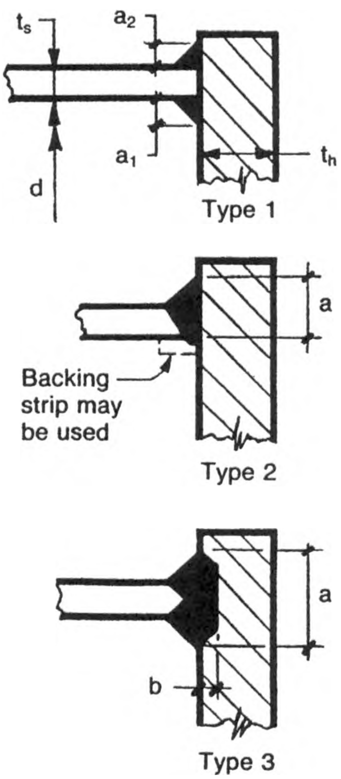


Figure 2-30. Case 9: Welded flat heads [1, Figure UG-34(h), Figure UW-13.2(g) (e-1)(f)].

1. Circular heads only.
2. $C = 0.33$
3. $t_s \geq 1.25t_r$
4. L.P./M.T. end of shell and O.D. of head if t_s or t_h is greater than ½ in. thick (before and after welding).
5. *Type 1*: $a_1 + a_2 \geq 2t_s$

   $0.5a_2 \leq a_1 \leq 2a_2$

Type 2: $a \geq 2t_s$
Type 3: $a + b \geq 2t_s$
$b = 0$ is permissible

Case 10 (Figure 2-31)

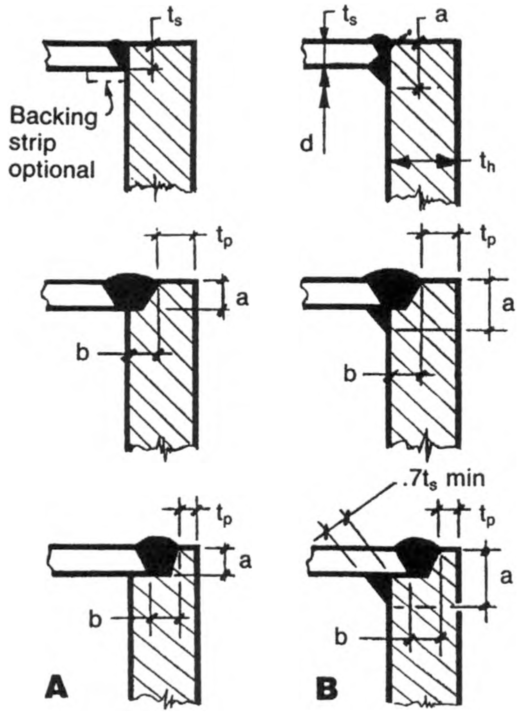


Figure 2-31. Case 10: Welded flat heads [1, Figure UG-34(h)(i), Figure UW-13.2(a)(b)(c)(d)].

1. For Figure 2-31A: $C = 0.33$ and $t_s \geq 1.25t_r$
2. For Figure 2-31B: $C = 0.33$ m but ≥ 0.2
3. $t_p \geq (t_s$ or 0.25 in.)
4. $t_w \geq t_s$
5. $a + b \geq 2t_s$
6. $a \geq t_s$
7. L.P./M.T end of shell and O.D. of head if t_s or t_h is greater than 1/2 in. thick (before and after welding).

Case 11 (Figure 2-32)

1. $C = 0.3$
2. All possible means of failure (by shear, tension, compression, or radial deformation, including flaring, resulting from pressure and differential thermal expansion) are resisted by factor of safety of 4:1.
3. Seal welding may be used.
4. Circular heads only.

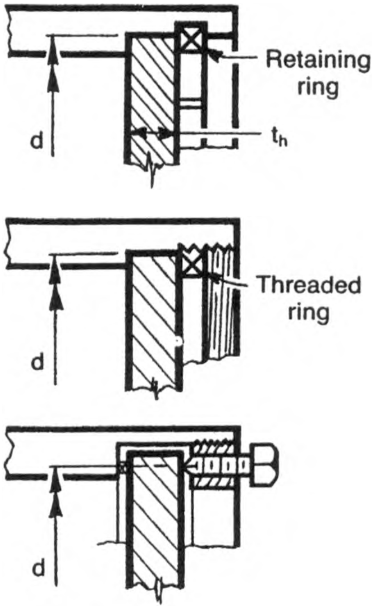


Figure 2-32. Case 11: Heads attached by mechanical lock devices [1, Figure UG-34(m)(n)(o)].

Case 12 (Figure 2-33)

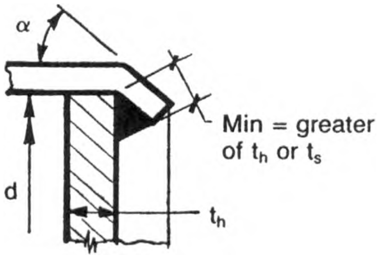


Figure 2-33. Case 12: Crimped head [1, Figure UG-34(r)].

1. $C = 0.33$
2. Circular plates only.
3. $d = 18$-in. maximum.
4. $\alpha = 30°$ minimum, 45° maximum.

Case 13 (Figure 2-34)

1. $C = 0.33$
2. Circular plates only.
3. $d = 18$-in. maximum.
4. $\alpha = 30°$ minimum, 45° maximum.
5. $t_s/d > P/S$ and $t_s/d > 0.05$

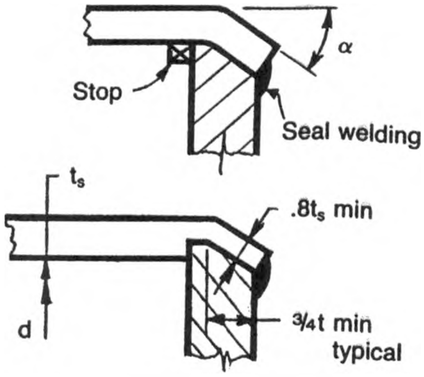


Figure 2-34. Case 13: Crimped heads [1, Section UG-34(s)].

6. Maximum allowable working pressure ≤ S/5d.
7. Crimping must be done at the proper forging temperature.

Stresses in Flat Heads

Maximum stress occurs at the junction, is axial in direction, and may be in either the head or the shell. When $t_h/t_s \le 1$, the maximum stress is in the head at the junction. When $t_h/t_s > 1$, the maximum stress is in the shell at the junction. The bending moment M_o is a result of internal forces N_o and Q_o.

- *Internal force, Q_o.*

$$Q_o = Pd_m\left[\frac{(a_4 - a_1)b_3 - (a_3 - a_6)(b_4 - b_1)}{(a_4 - a_1)(b_5 - b_2) - (a_5 - a_2)(b_4 - b_1)}\right]$$

- *Bending moment, M_o.*

$$M_o = Pd_m^2\left[\frac{(a_3 - a_6)(b_5 - b_2) - (a_5 - a_2)b_3}{(a_4 - a_1)(b_5 - b_2) - (a_5 - a_2)(b_4 - b_1)}\right]$$

- *Axial stress in shell at junction, σ_s* [4, Equation 6.122].

$$\sigma_s = \frac{Pd_m}{4t_s} + \left|\frac{6M_o}{t_s^2}\right|$$

- *Axial stress in shell at junction, σ_h* [4, Equation 6.132].

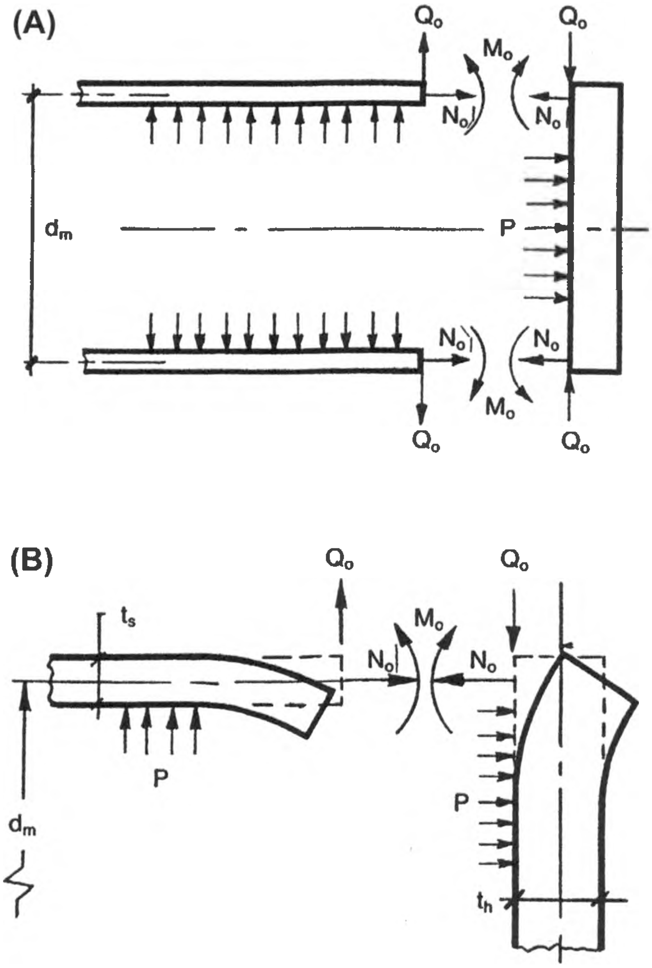


Figure 2-35. Discontinuity at flat head and cylindrical shell juncture.

$$\sigma_h = \left|\frac{Q_o}{t_h}\right| + \left|\frac{6M_o}{t_h^2} - \frac{3Q_o}{t_h}\right|$$

- *Primary bending stress in head, σ_b. Note:* Primary bending stress is maximum at the center of the head.

$$\sigma_b = (\pm)\frac{3(3+\nu)}{8}\left[\frac{Pd^2}{4t_h^2}\right]$$

(−) Inside head, compression
(+) Outside head, tension

Procedure 2-10: Design of Large Openings in Flat Heads [1]

Notation

P = internal pressure, psi
M_o = bending moment in head, in.-lb
M_h = moment acting on end of hub or shell at juncture, in.-lb
M_D = component of moment M_o due to H_D, in.-lb
M_T = component of moment M_o due to H_T, in.-lb
H = hydrostatic end force, lb
H_D = hydrostatic end force on area of central opening, lb
H_T = $H - H_D$, lb
S_H = longitudinal hub stress, psi
S_R = radial stress in head, psi
S_T = tangential stress in head, psi
S_{HS} = longitudinal hub stress, shell, psi
S_{RS} = radial stress, head, at O.D., psi
S_{TS} = tangential stress, head, at O.D., psi
S_{HO} = longitudinal hub stress at central opening, psi
S_{RO} = radial stress, head, at central opening, psi
S_{TO} = tangential stress, head, at central opening, psi
Z, Z_1, Y, T, U, F, V, f, e, d, L, X_1, and θ are all factors.

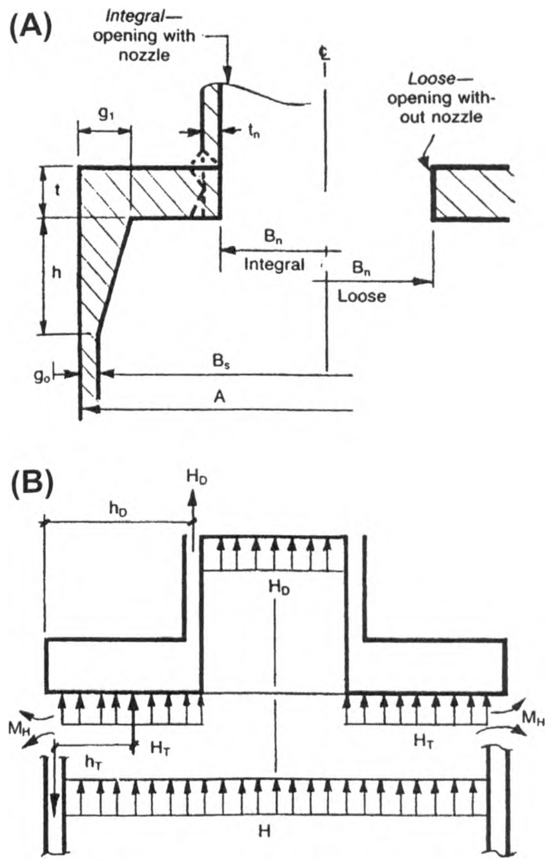


Figure 2-36. Dimensions (A) and loading diagram (B) for a flat integral head with opening.

Factor Formulas

1. Calculate geometry factors:

$$\frac{g_1}{g_o} =$$

$$K = \frac{A}{B_n} =$$

$$h_o = \sqrt{B_n g_o} =$$

$$\frac{h}{h_o} =$$

2. Using the factors calculated in Step 1, find the following factors in Procedure 3-1.

Z =
Y =
T =
U =
F =
V =
f =

3. Using the values found in the preceding steps, compute the following factors:

$$e = \frac{F}{h_o} =$$

$$d = \frac{U h_o g_o^2}{V} =$$

$$L = \frac{te + 1}{T} + \frac{t^3}{d} =$$

$$Z_1 = \frac{2K^2}{K^2 - 1} =$$

Stress and Moment Calculations

1. *Hydrostatic end forces, H, H_D, H_T.*

$$H = \frac{\pi B_s^2 P}{4}$$

$$H_D = \frac{\pi B_n^2 P}{4}$$

$$H_T = H - H_D$$

2. *Moment arms, h_D and h_T.*
   - Integral:

$$h_D = \frac{A - B_n - t_n}{2}$$

   - Loose:

$$h_D = \frac{A - B_n}{2}$$

   - Integral or loose:

$$h_T = \frac{B_s - B_n}{4} + \frac{g_o}{2}$$

3. *Moments.*

$$M_D = h_D H_D$$

$$M_T = h_T H_T$$

$$M_o = M_D + M_T$$

4. *Stresses in head and hub.*

$$S_H = \frac{f M_o}{L g_1^2 B_n}$$

$$S_R = \frac{(1.33te + 1)M_o}{L t^2 B_n}$$

   - Integral:

$$S_T = \frac{Y M_o}{t^2 B_n} - Z S_R$$

   - Loose:

$$S_T = \frac{Y M_o}{t^2 B_n}$$

5. *Factor, θ.*
   - Integral:

$$B_1 = B_n + g_o$$

If $f \geq 1$,

$$\theta = \frac{0.91(g_1/g_o)^2 B_1 V S_H}{f h_o}$$

   - Loose:

$$\theta = \frac{B_n S_T}{t}$$

6. *Moment at juncture of shell and head, M_H.*

$$M_H = \frac{\theta}{\dfrac{1.74 h_o V}{g_o^3 B_1} + \dfrac{\theta}{M_o}\left(1 + \dfrac{Ft}{h_o}\right)}$$

where h_o, g_o, V, B_1, and F refer to shell.

7. *Factor X_1.*

$$X_1 = \frac{M_o - M_H\left(1 + \dfrac{Ft}{h_o}\right)}{M_o}$$

where F and h_o refer to shell.

8. *Stress at head-shell juncture.*

$$S_{HS} = \frac{1.1 X_1 \theta h_o f}{(g_1/g_o)^2 B_s V}$$

$$S_{RS} = \frac{1.91 M_H\left(1 + \dfrac{Ft}{h_o}\right)}{B_s t^2} + \frac{0.64 F M_H}{B_s h_o t}$$

$$S_{TS} = \frac{X_1 \theta t}{B_s} - \frac{0.57 M_H\left(1 + \dfrac{Ft}{h_o}\right)}{B_s t^2} + \frac{0.64 F Z M_H}{B_s h_o t}$$

where B_s, F, h_o, Z, f, g_o, g_1, and V refer to shell.

9. *Calculate stresses at head-nozzle juncture.*

$$S_{HO} = X_1 S_H$$

$$S_{RO} = X_1 S_R$$

$$S_{TO} = X_1 S_T + \frac{0.64 F Z_1 M_H}{B_s h_o t}$$

where F, B_s, and h_o refer to shell.

Notes

1. This procedure is only applicable for integrally attached flat heads with centrally located openings which exceed one-half the head diameter. For applicable configurations see sketches in ASME Code, Figures UG-34(a), (b-1), (b-2), (d), or (g).
2. For details where inside corner of shell-head juncture is machined to a radius: $g_1 = g_o$ and $f = 1$.

3. The method employed in this procedure is to disregard the shell attached to the outside diameter of the flat head and then analyze the flat head with a central opening.
4. This procedure is based on Appendix 14 of ASME Section VIII, Division 1.

Procedure 2-11: Calculate MAP, MAWP, and Test Pressures

Notation

S_a = allowable stress at ambient temperature, psi
S_{DT} = allowable stress at design temperature, psi
S_{CA} = allowable stress of clad material at ambient temperature, psi
S_{CD} = allowable stress of clad material at design temperature, psi
S_{BA} = allowable stress of base material at ambient temperature, psi
S_{BD} = allowable stress of base material at design temperature, psi
C.a. = corrosion allowance, in.
t_{sc} = thickness of shell, corroded, in.
t_{sn} = thickness of shell, new, in.
t_{hc} = thickness of head, corroded, in.
t_{hn} = thickness of head, new, in.
t_b = thickness of base portion of clad material, in.
t_c = thickness of cladding, in.
R_n = inside radius, new, in.
R_c = inside radius, corroded, in.
R_o = outside radius, in.
D_n = inside diameter, new, in.
D_c = inside diameter, corroded, in.
D_o = outside diameter, in.
P_M = MAP, psi
P_W = MAWP, psi
P = design pressure, psi
P_S = shop hydro pressure (new and cold), psi
P_F = field hydro pressure (hot and corroded), psi
E = joint efficiency, see Procedure 2-1 and Appendix C

Definitions

Maximum Allowable Working Pressure (MAWP): The MAWP for a vessel is the maximum permissible pressure at the top of the vessel in its normal operating position at a specific temperature, usually the design temperature. When calculated, the MAWP should be stamped on the nameplate. The MAWP is the maximum pressure allowable in the "hot and corroded" condition. It is the least of the values calculated for the MAWP of any of the essential parts of the vessel, and adjusted for any difference in static head that may exist between the part considered and the top of the vessel. This pressure is based on calculations for every element of the vessel using nominal thicknesses exclusive of corrosion allowance. It is the basis for establishing the set pressures of any pressure-relieving devices protecting the vessel. The design pressure may be substituted if the MAWP is not calculated.

The MAWP for any vessel part is the maximum internal or external pressure, including any static head, together with the effect of any combination of loadings listed in UG-22 which are likely to occur, exclusive of corrosion allowance at the designated coincident operating temperature. The MAWP for the vessel will be governed by the MAWP of the weakest part.

The MAWP may be determined for more than one designated operating temperature. The applicable allowable stress value at each temperature would be used. When more than one set of conditions is specified for a given vessel, the vessel designer and user should decide which set of conditions will govern for the setting of the relief valve.

Maximum Allowable Pressure (MAP): The term MAP is no longer used by the Code but is used here as a matter of convenience. It refers to the maximum permissible pressure based on the weakest part in the new (uncorroded) and cold condition, and all other loadings are not taken into consideration.

Design Pressure: The pressure used in the design of a vessel component for the most severe condition of coincident pressure and temperature expected in normal operation. For this condition, and test condition, the maximum difference in pressure between the inside and outside of a vessel, or between any two chambers of a combination unit, shall be considered. Any thickness required for static head or other loadings shall be additional to that required for the design pressure.

Design Temperature: For most vessels, it is the temperature that corresponds to the design pressure. However, there is a maximum design temperature and a minimum design temperature for any given vessel. The minimum design temperature would be the MDMT (see Procedure 2-14). The MDMT shall be the lowest temperature expected in service or the lowest allowable temperature as calculated for the individual parts. Design temperature for vessels under external pressure shall not exceed the maximum temperatures given on the external pressure charts.

Operating Pressure: The pressure at the top of the vessel at which it normally operates. It shall be lower than the MAWP, design pressure, or the set pressure of any pressure relieving device.

Operating Temperature: The temperature that will be maintained in the metal of the part of the vessel being considered for the specified operation of the vessel.

Calculations

- *MAWP, corroded at Design Temperature P_ω. Shell:*

$$P_W = \frac{S_{DT}Et_{sc}}{R_c + 0.6t_{sc}} \text{ or } \frac{S_{DT}Et_{sc}}{R_o - 0.4t_{sc}}$$

2:1 S.E. Head:

$$P_W = \frac{2S_{DT}Et_{hc}}{D_c + 0.2t_{hc}} \text{ or } \frac{2S_{DT}Et_{hc}}{D_o - 1.8t_{hc}}$$

- *MAP, new and cold, P_M*

Shell:

$$P_M = \frac{S_aEt_{sn}}{R_n + 0.6t_{sn}} \text{ or } \frac{S_aEt_{sn}}{R_o - 0.4t_{sn}}$$

2:1 S.E. Head:

$$P_M = \frac{2S_aEt_{hn}}{D_n + 0.2t_{hn}} \text{ or } \frac{2S_aEt_{hn}}{D_o - 1.8t_{hn}}$$

- *Shop test pressure, P_s.*

$$P_s = 1.3P_M \text{ or } 1.3P_W\left[\frac{S_a}{S_{DT}}\right]$$

- *Field test pressure, P_F*

$$P_F = 1.3P$$

- *For clad vessels where credit is taken for the clad material, the following thicknesses may he substituted into the equations for MAP and MAWP:*

$$t_{sc}, t_{hc} = t_b + \left[\frac{S_{CD}}{S_{BD}}(t_c - C.a.)\right]$$

$$t_{sn}, t_{hn} = t_b + \left[\frac{S_{CA}t_c}{S_{BA}}\right]$$

Notes

1. Also check the pressure-temperature rating of the flanges for MAWP and MAP.
2. All nozzles should be reinforced for MAWP.
3. The MAP and MAWP for other components, i.e., cones, flat heads, hemi-heads, torispherical heads, etc., may be checked in the same manner by using the formula for pressure found in Procedure 2-1 and substituting the appropriate terms into the equations.
4. It is not necessary to take credit for the cladding thickness. If it is elected not to take credit for the cladding thickness, then base all calculations on the full base metal thickness. This assumes the cladding is equivalent to a corrosion allowance, and no credit is taken for the strength of the cladding.

Procedure 2-12: Nozzle Reinforcement

The following are only guidelines based on Section VIII, Division 1 of the ASME Code [1]. This is not an attempt to cover every possibility nor is it to become a substitute for using the Code.

1. *Limits.*
   a. No reinforcement other than that inherent in the construction is required for nozzles [1, Section UG-36(c)(3)]:
      - 3-in. pipe size and smaller in vessel walls 3/8–in. and less.
      - 2-in. pipe size and smaller in vessel walls greater than 3/8 in.

   b. Normal reinforcement methods apply to [1, Section UG-36(b)(1)]:
      - *Vessels 60-in. diameter and less*—1/2 the vessel diameter but not to exceed 20 in.
      - *Vessels greater than 60-in. diameter*—1/3 the vessel diameter but not to exceed 40 in.

   c. For nozzle openings greater than the limits of guideline 1.b., reinforcement shall be in accordance with Appendix 1-7 of the ASME Code.
2. *Strength.*
   It is advisable but not mandatory for reinforcing pad material to be the same as the vessel material [1, Section UG-41]:
   a. If a *higher strength material* is used, either in the pad or in the nozzle neck, no additional credit may be taken for the higher strength.
   b. If a *lower strength material* is used, either in the pad or in the nozzle, then the area taken as reinforcement must be decreased proportionately by the ratio of the allowable stress values of the two materials. Weld material taken as reinforcement must also be decreased as a proportion, assuming the weld material is the same strength as the weaker of the two materials joined.
3. *Thickness.*
   While minimum thicknesses are given in Reference 1, Section UG-16(b), it is recommended that pads be not less than 75% nor more than 150% of the part to which they are attached.
4. *Width.*
   While no minimum is stated, it is recommended that re-pads be at least 2 in. wide.
5. *Forming.*
   Reinforcing pads should be formed as closely to the contour of the vessel as possible. While normally put on the outside of the vessel, re-pads can also be put inside providing they do not interfere with the vessel's operation [1, Section UG-82].
6. *Tell-tale holes.*
   Reinforcing pads should be provided with a 1/4-in. tapped hole located at least 45° off the longitudinal center line and given an air-soap suds test [1, Section UW-15(d)].
7. *Elliptical or obround openings.*
   When reinforcement is required for elliptical or obround openings and the long dimension exceeds twice the short dimension, the reinforcement across the short dimension shall be increased to guard against excessive distortion due to twisting moment [1, Section UG-36(a)(1)].
8. *Openings in flat heads.*
   Reinforcement for openings in flat heads and blind flanges shall be as follows [1, Section UG-39]:
   a. *Openings < 1/2 head diameter*—area to be replaced equals 0.5d (t_r), or thickness of head or flange may be increased by:
      - Doubling C value.
      - Using C = 0.75.
      - Increasing head thickness by 1.414.

   b. *Openings > 1/2 head diameter*—shall be designed as a bolted flange connection. See Procedure 2-15.
9. *Openings in torispherical heads.*
   When a nozzle opening *and* all its reinforcement fall within the dished portion, the required thickness of head for reinforcement purposes shall be computed using M = 1 [1, Section UG-37(a)].
10. *Openings in elliptical heads.*
    When a nozzle opening *and* all its reinforcement fall within 0.8D of an elliptical head, the required thickness of the head for reinforcement purposes shall be equal to the thickness required for a seamless sphere of radius K(D) [1, Section UG-37(a)].
11. *General.*
    Reinforcement should be calculated in the corroded condition assuming maximum tolerance (minimum t). For non x-rayed vessels, t_r must be

computed using a stress value of 0.8S [1, Section UG-37(a)].

12. *Openings through seams.* [1, Section UW-14].
    a. Openings that have been reinforced may be located in a welded joint. E = joint efficiency of seam for reinforcement calculations. The ASME Code, Section VIII, Division 1, does not allow a welded joint to have two different weld joint efficiencies. Credit may not be taken for a localized x-rayed portion of a spot or non x-rayed seam.
    b. Small nozzles that are not required to be checked per the Code can be located in circumferential joints providing the seam is x-rayed for a distance three times the diameter of the opening with the center of the hole at midlength.
13. *Re-pads over seams.*
    If at all possible, pads should not cover weld seams. When unavoidable, the seam should be ground flush before attaching the pad [1, Section UG-82].
14. *Openings near seams.*
    Small nozzles (for which the Code does not require the reinforcement to be checked) shall not be located closer than 1/2 in. to the edge of a main seam. When unavoidable, the seam shall be x-rayed, per ASME Code, Section UW-51, a distance of one and a half times the diameter of the opening either side of the closest point [1, Section UW-14].
15. *External pressure.*
    Reinforcement required for openings subject to external pressure only *or* where longitudinal compression governs shall only be 50% of that required for internal pressure and t_r is thickness required for external pressure [1, Section UG-37(d)].
16. *Ligaments.*
    When there is a series of closely spaced openings in a vessel shell and it is impractical to reinforce each opening, the construction is acceptable, provided the efficiency of the ligaments between the holes is acceptable [1, Section UG-53].
17. *Multiple openings.* [1, Section UG-42].
    a. For two openings closer than 2 times the average diameters and where limits of reinforcement overlap, the area between the openings shall meet the following:
       - *Must have a combined area equal to the sum of the two areas.*
       - *No portion of the cross-section shall apply to more than one opening.*
       - *Any overlap area shall be proportioned between the two openings by the ratio of the diameters.*
       - *If the area between the openings is less than 50% of that required for the two openings, the supplemental rules of Appendix 1-7(a) and (c) shall apply.*
    b. When more than two openings are to be provided with combined reinforcement:
       - *The minimum distance between the centers is* 1⅓ the average diameters.
       - *The area of reinforcement between the two nozzles shall be at least 50% of the area required for the two openings.*
    c. For openings less than 1⅓ times the average diameters:
       - *No credit may be taken for the area between the openings.*
       - *These openings shall be reinforced as in (d).*
    d. Multiple openings may be reinforced as an opening equal in diameter to that of a circle circumscribing the multiple openings.
18. *Plane of reinforcement.*
    A correction factor f may be used for "integrally reinforced" nozzles to compensate for differences in stress from longitudinal to circumferential axis of the vessel. Values of f vary from 1.0 for the longitudinal axis to 0.5 for circumferential axis [1, Section UG-37].

VESSEL DESCRIPTION:						ITEM NO:			SIZE:		
	1	2	3	4	5	Nozzle	1	2	3	4	5
Nozzle						t_n					
Location						t_{nc}					
Size and schedule						t_m					
P at elevation						Limit h					
I.D. New						A_2					
d (corroded)						A_3					
Shell/head t_{corr}						A_4					
Shell/head t_r						Pad size $t_e \times D_p$					
A						O.D. Nozzle					
Limit L						A_5					
A_1						A_T					

THICKNESS REQUIRED

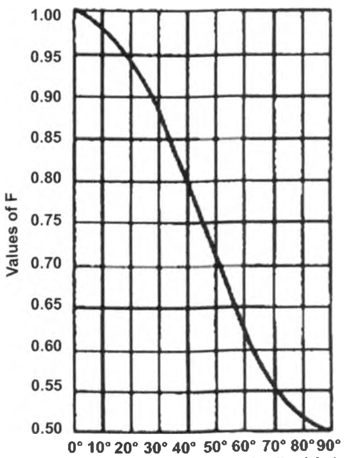
Shell		Head		Nozzles					
$t_r = \dfrac{PR}{S - 0.6P}$		$t_{rh} = \dfrac{PD}{2S - 0.2P}$		$t_{rn} = \dfrac{PR_n}{S - 0.6P}$					
S, shell		*D		Nozzle					
		S, head		P					
*Note: D = R for heml-heads D = 0.9D if nozzle and reinforcement lie within 0.8D of 2:1 head D = L if nozzle and reinforcement lie within dished portion of a flanged and dished head.				R_n					
				S					
				t_{rn}					

FORMULAS

$A = dt_rF + 2t_nt_rF(1 - f_{r1})$		h = lesser of 2.5 t or 2.5 $t_{nc} + t_e$ h_1 = lesser of 2.5 t or 2.5 (t_n − 2c.a.)	
$A_1 = (2L - d)(t - Ft_r) - 2t_n(t - Ft_r)(1 - f_{r1})$			
$A_2 = 2h(t_n - t_m)f_{r1}$ $A_3 = 2h_1(t_n - 2\text{ c.a})f_{r1}$		$f_{r1} = \dfrac{S_{noz}}{S_{shell}} < 1$	< 1
A_4 = (WELDS) t_e $(A_{41} + A_{43})f_{r1} + A_{42}f_{r4}$ $(A_5 = (D_p - d - 2t_n)t_e f_{r4}$		$f_{r4} = \dfrac{S_{pad}}{S_{shell}} < 1$	< 1
$A_T = A_1 + A_2 + A_3 + A_4 + A_5$			
L = greater of d or $R_n + t + t_{nc}$			

DESIGN DATA

Corrosion allowance. c.a.		Specific gravity	
Design liquid level		Thinning allowance	



Angle of Plane With Longitudinal Axis Chart for determining the value of F [1, Figure UG-37].

Notes: Assumes E = 1 & f_{r1} = 1.0 for nozzle abutting vessel wall.

Figure 2-37. Worksheet for nozzle reinforcement calculations.

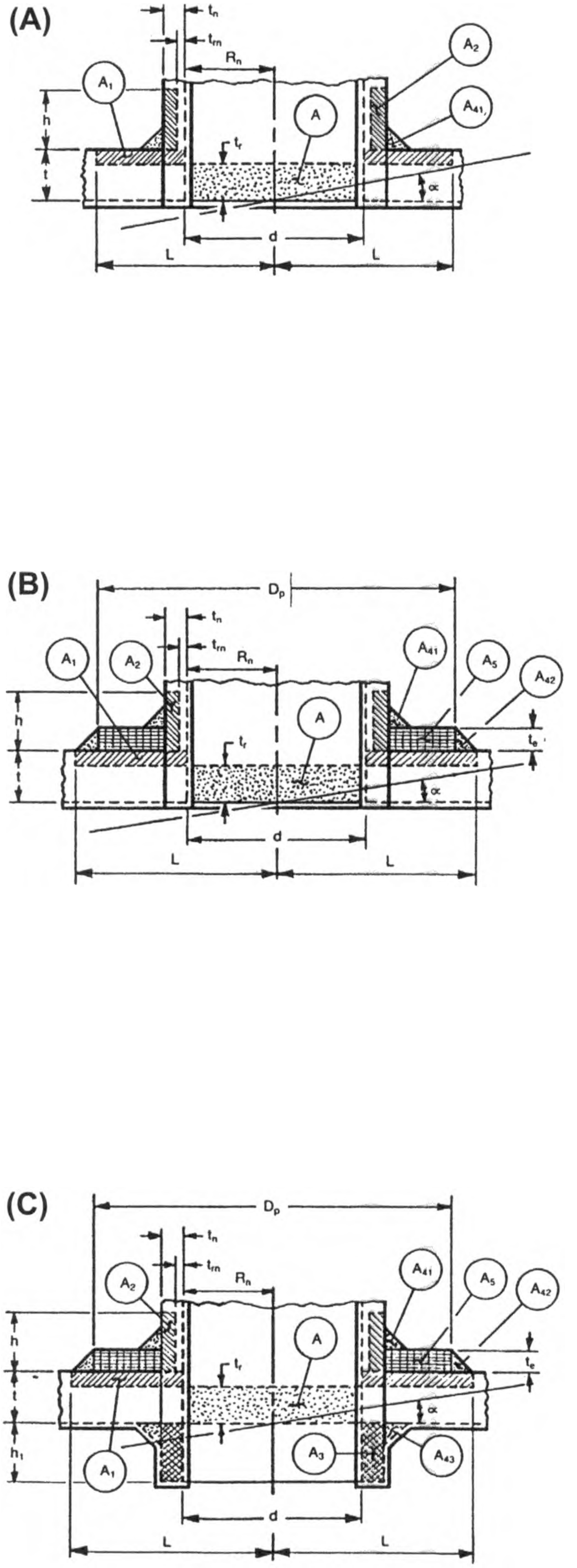


Figure 2-38. Typical nozzle connections.

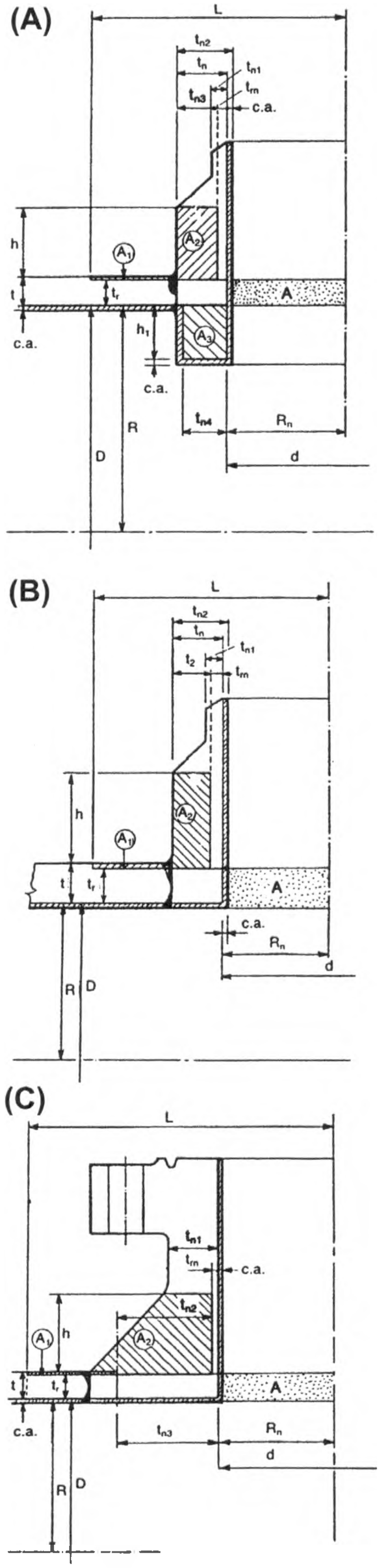
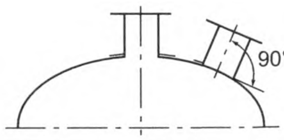
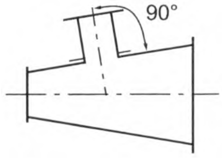
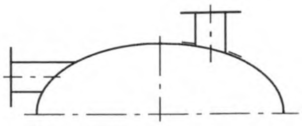
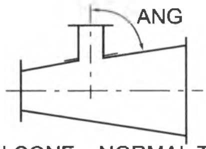
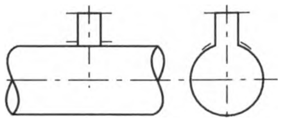
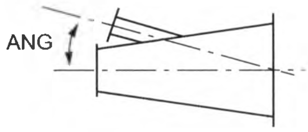
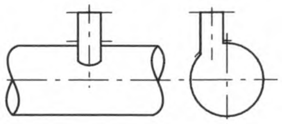
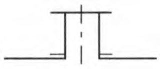
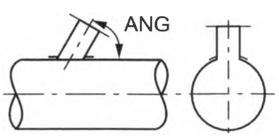
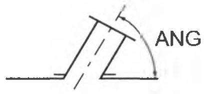
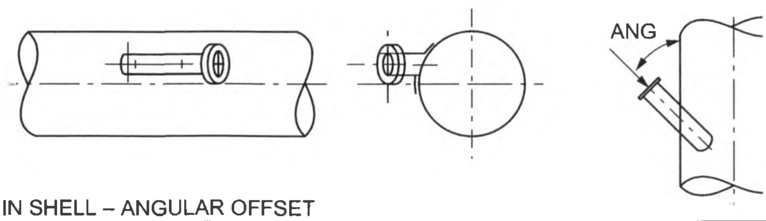


Figure 2-39. Typical self-reinforced nozzles.

NOZZLE POSITION	
 NOZZLE IN HEAD – NORMAL TO WALL	 NOZZLE IN CONE – NORMAL TO WALL
 NOZZLE IN HEAD – NORMAL OR PARALLEL TO VESSEL CENTERLINE	 NOZZLE IN CONE – NORMAL TO CENTERLINE
 NOZZLE IN SHELL – NORMAL TO WALL	 NOZZLE IN CONE – INCLINED TO LONG AXIS
 NOZZLE IN SHELL – OFFSET AND NORMAL TO CENTERLINE	 NOZZLE IN FLAT HEAD – NORMAL TO HEAD
 NOZZLE IN SHELL – ANGLED	 NOZZLE IN FLAT HEAD – ANGLED
 NOZZLE IN SHELL – ANGULAR OFFSET	

Procedure 2-13 Find or Revise the Center of Gravity of a Vessel

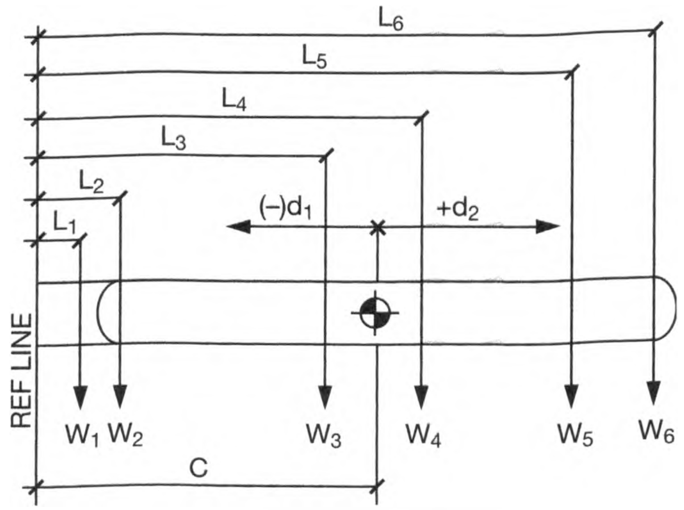


Figure 2-40. Load diagram for a typical vertical vessel.

Notation

C = distance to center of gravity, ft or in.
D′ = revised distance to C.G., ft or in.
d_n = distance from original C.G. to weights to add or remove, (+) or (−) as shown, ft or in.
L_n = distance from REF line to C.G. of a component weight, ft or in.
W_n = weight of vessel component, contents or attachments, lb
W′ = new overall weight, lb W + or − $\sum W_n$
W = overall weight, lb, $\sum W_n$
ω_n = revised unit weights, lb (+) to add weight (−) to remove weight

To find the C.G:

$$C = \frac{\sum L_n W_n}{W}$$

To revise C.G:

$$D' = C \pm \frac{\sum d_n \omega_n}{W'}$$

Procedure 2-14: Minimum Design Metal Temperature (MDMT)

Notation

R = use the lesser of R_1 or R_2
R_1 = ratio of thickness required at MDMT to the corroded thickness
R_2 = ratio of the actual stress to the allowable stress
t_{MT} = thickness required of the part at MDMT, in.
t_{DT} = thickness required of the part at design temperature, in.
t_n = thickness of the part, new, in (exclusive of thinning allowance for heads and undertolerance for pipe)
t_c = thickness of the part, corroded, in.
C.a. = corrosion allowance, in.
E = joint efficiency
S_{MT} = allowable stress at MDMT, psi
S_{DT} = allowable stress at design temperature, psi
S_a = actual tension stress in part due to pressure and all loadings, psi
T_1 = lowest allowable temperature for a given part based on the appropriate material curve of Figure 2-43, degrees F
T_2 = reduction in MDMT without impact testing per Figure 2-42, degrees F

This MDMT procedure is used to determine the lowest permissible temperature for which charpy impact testing is or is not required. The ASME Code requires this be determined for every pressure vessel and the MDMT be stamped on the nameplate. While every pressure vessel has its own unique MDMT, this may or may not be the MDMT that is stamped on the nameplate. Not only does every pressure vessel have its own unique MDMT, but every component of that pressure vessel has an MDMT. The vessel MDMT is the highest temperature of all the component MDMT's. On occasion, the MDMT is specified by the end user as an arbitrary value. The vessel fabricator is then responsible to verify that the actual MDMT of every component used in that pressure vessel is lower than the arbitrary value requested for the nameplate stamping. Considering this, there are various definitions for MDMT depending on how it is used. The definitions follow:

1. *Arbitrary MDMT:* A discretionary, arbitrary temperature, specified by a user or client, or determined in accordance with the provisions of UG-20. Some users have a standard value that has been chosen as the lowest mean temperature of the site conditions, such as 15°F.
2. *Exemption MDMT*: The lowest temperature at which the pressure vessel may be operated at full design pressure without impact testing of the component parts.
3. *Test MDMT:* The temperature at which the vessel is charpy impact tested.

The ASME Code rules for MDMT are built around a set of material exemption curves as shown in Figure 2-43. These curves account for the different toughness characteristics of carbon and low alloy steel and determine at what temperature and corresponding thickness impact testing will become mandatory.

There is an additional exemption curve (see Figure 2-42), which allows a decrease in the MDMT of every component, and thus the vessel, depending on one of several ratios specified. This curve would permit carbon steel, without impact testing, to be used at a temperature of (−)150°F, provided the combined stresses are less than 40% of the allowable stress for that material. Granted, the vessel would be more than twice as thick as it needed to be for the pressure condition alone, but if the goal was to exempt the vessel from impact testing, it could be accomplished.

Since impact testing is a major expense to the manufacturer of a pressure vessel, the designer should do everything to avoid it. Impact testing can always be avoided but may not be the most economical alternative. Following these steps will help eliminate the need for impact testing and, at the same time, will provide the lowest MDMT.

1. Upgrade the material to a higher group.
2. Increase the thickness of the component to reduce the stress in the part.
3. Decrease the pressure at MDMT. This is a process change and may or may not be possible. Sometimes a vessel does not operate at full design pressure at the low temperature condition but has alternate conditions, such as shutdown or depressurization. These alternate low temperature conditions can also be stamped on the nameplate.

Formulas

$$R_1 = \frac{t_r E}{t_c}$$

Table 2-7
Determination of MDMT (Example)

Part	Material	Material Group	S_{MT} ksi	S_{DT} ksi	t_n	t_{DT} (3)	t_{MT} (3)	t_c	S_a ksi	R_1	R_2	T_1	T_2	MDMT °F
l	SA-516-70	B	17.5	16.6	1.00	0.869	0.823	0.875	13.97	0.799	0.798	31°	20.1°	+11
d(1)	SA-516-70	B	17.5	16.6	0.857	0.653	0.620	0.732	14.89	0.847	0.851	21.8	15°	+7
Noz(2)	SA-53-B	B	12.8	12.2	0.519	0.174	0.166	0.394	5.26	0.421	0.410	−5.18	59°	
300# Flg. (4)	SA-105	B	17.5	16.6	0.519	0.128	0.121	0.394	5.26	0.307	0.300	−5.18	70°	
Blind (5)	SA-266-2	B	17.5	16.6	6.06	—	1.48	5.94	—	—	—	51°	105°	−54
Body Flg.	SA-266-2	B	17.5	16.6	Same as Shell									+11
r PL	SA-516-70	B	17.5	16.6	1.00	—	—	1.00	—	—	—	—	—	+11(6)
ng	SA-193-B7	—	—	—	—	—	—	—	—	—	—	—	—	−40

:

The governing thickness for heads is based on that portion of the head which is in tension. For a 2:1 S.E. head this is the crown position where R = 0.90.
Includes pipe 12½% under tolerance.
Thickness exclusive of C.a.
Thickness at the hub (weld attachment) governs.
The governing thickness of flat heads and blind flanges is 1/4 of actual thickness.
Since the tension stress in the wear plate is less than the tension stress in the shell, the MDMT for the shell will govern.

$$R_2 = \frac{S_a}{S_{MT}}$$

$$t_c = t_n - C.a.$$

$$T_2 = (1 - R)100$$

$$MDMT = T_1 - T_2$$

Procedure

Step 1: Determine the lowest anticipated temperature to which the vessel will be subjected.

Step 2: Compare the lowest combined pressure-temperature case with the MDMT for each component.

Step 3: Determine if any components must be impact tested in their proposed material grade and thickness. This would establish the MDMT.

Step 4: Establish the overall MDMT as the highest value of MDMT for each of the component parts.

Notes

1. For flat heads, tubesheets, and blind flanges, the thickness used for each of the respective thickness' is that thickness divided by 4.
2. For corner, fillet, or lap-welded joints, the thickness used shall be the thinner of the two parts being joined.
3. For butt joints, the thickness used shall be the thickest joint.
4. For any Code construction, if the vessel is stress relieved and that stress relieving was not a Code requirement, the MDMT for that vessel may be reduced by 30° without impact testing.

Design Conditions (for example)

D.T. = 700°F
P = 400 PSIG
C.a. = 0.125
R_i = 30 in
E (Shell) = 0.85
E (Head) = 1.00
MDMT for vessel = + 11°F

General Notes on Assignment of Materials to Curves (Reprinted with permission from ASME Code, Section VIII, Div. 1.)

a. Curve A—all carbon and all low alloy steel plates, structural shapes, and bars not listed in Curves B, C, and D below.
b. Curve B

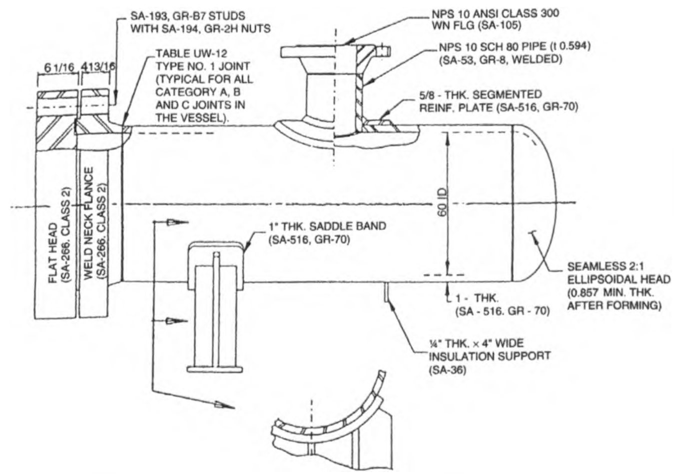
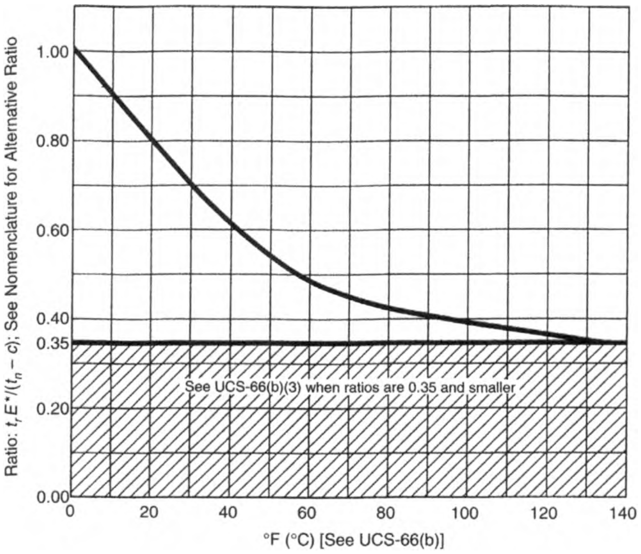


Figure 2-41. Dimensions of vessel used for MDMT example.

1. SA-285 Grades A and B
   SA-414 Grade A
   SA-515 Grade 60
   SA-516 Grades 65 and 70 if not normalized
   SA-612 if not normalized
   SA-662 Grade B if not normalized
2. all materials of Curve A if produced to fine grain practice and normalized which are not listed for Curves C and D below.
3. except for bolting (see (e) below), plates, structural shapes, and bars, all other product forms (such as pipe, fittings, forgings, castings, and tubing) not listed for Curves C and D below.
4. parts permitted under UG-11 shall be included in Curve B even when fabricated from plate that otherwise would be assigned to a different curve.

c. Curve C

1. SA-182 Grades F21 and F22 if normalized and tempered
   SA-302 Grades C and D
   SA-336 Grades F21 and F22 if normalized and tempered
   SA-387 Grades 21 and 22 if normalized and tempered
   SA-516 Grades 55 and 60 if not normalized
   SA-533 Grades B and C
   SA-662 Grade A
2. all materials of Curve B if produced to fine grain practice and normalized and not listed for Curve D below.



Nomanclature (Note reference to General Notes of Fig. UCS-66-2.)

t_r = required thickness of the component under consideration in the corroded condition for all applicable loadings (General Note (2), based on the applicable joint efficiency E (General Note (3), in. (mm)

t_n = nominal thickness of the component under consideration before corrosion allowance is deducted, in. (mm)

c = corrosion allowance, in. (mm)

E^* = as defined in General Note (3)

Alternative Ratio = S^*E^* divided by the product of the maximum allowable stress value from Table UCS-23 times E, where S^* is the applied general primary membrane tensile stress and E and E^* are as defined in General Note (3)

Figure 2-42. Reduction in minimum design metal temperature without impact testing.

d. Curve D
   SA-203
   SA-508 Class 1
   SA-516 if normalized
   SA-524 Classes 1 and 2
   SA-537 Classes 1, 2, and 3
   SA-612 if normalized
   SA-622 if normalized
e. For bolting the following impact test exemption temperature shall apply:

Spec. No.	Grade	Impact Test Exemption Temperature, °F
SA-193	B5	−20
SA-193	B7	−40
SA-193	B7M	−55
SA-193	B16	−20
SA-307	B	−20
SA-320	B L7, L7A, L7M, L43	Impact tested
SA-325	1, 2	−20
SA-354	BC	0
SA-354	BD	+20
SA-449		−20
SA-540	B23/24	+10

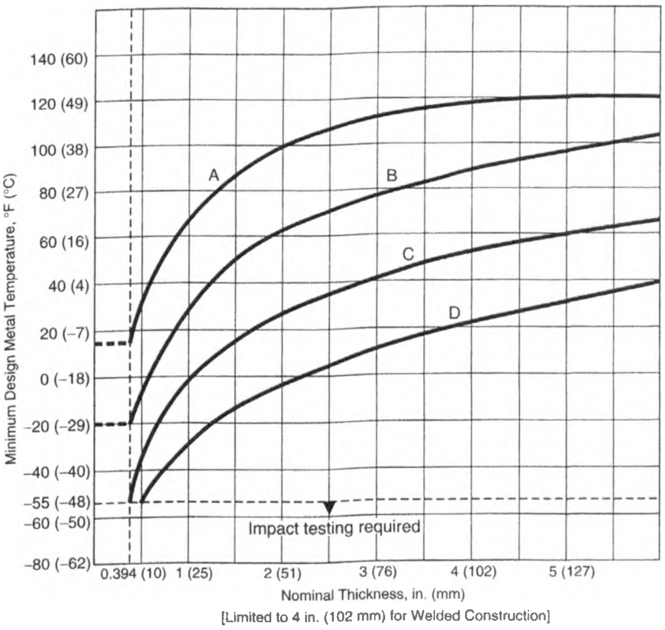


Figure 2-43. Impact test exemption curves.

f. When no class or grade is shown, all classes or grades are included.
g. The following shall apply to all material assignment notes:
   1. Cooling rates faster than those obtained by cooling in air, followed by tempering, as permitted by the material specification, are considered to be equivalent to normalizing or normalizing and tempering heat treatments.
   2. Fine grain practice is defined as the procedures necessary to obtain a fine austenitic grain size as described in SA-20.

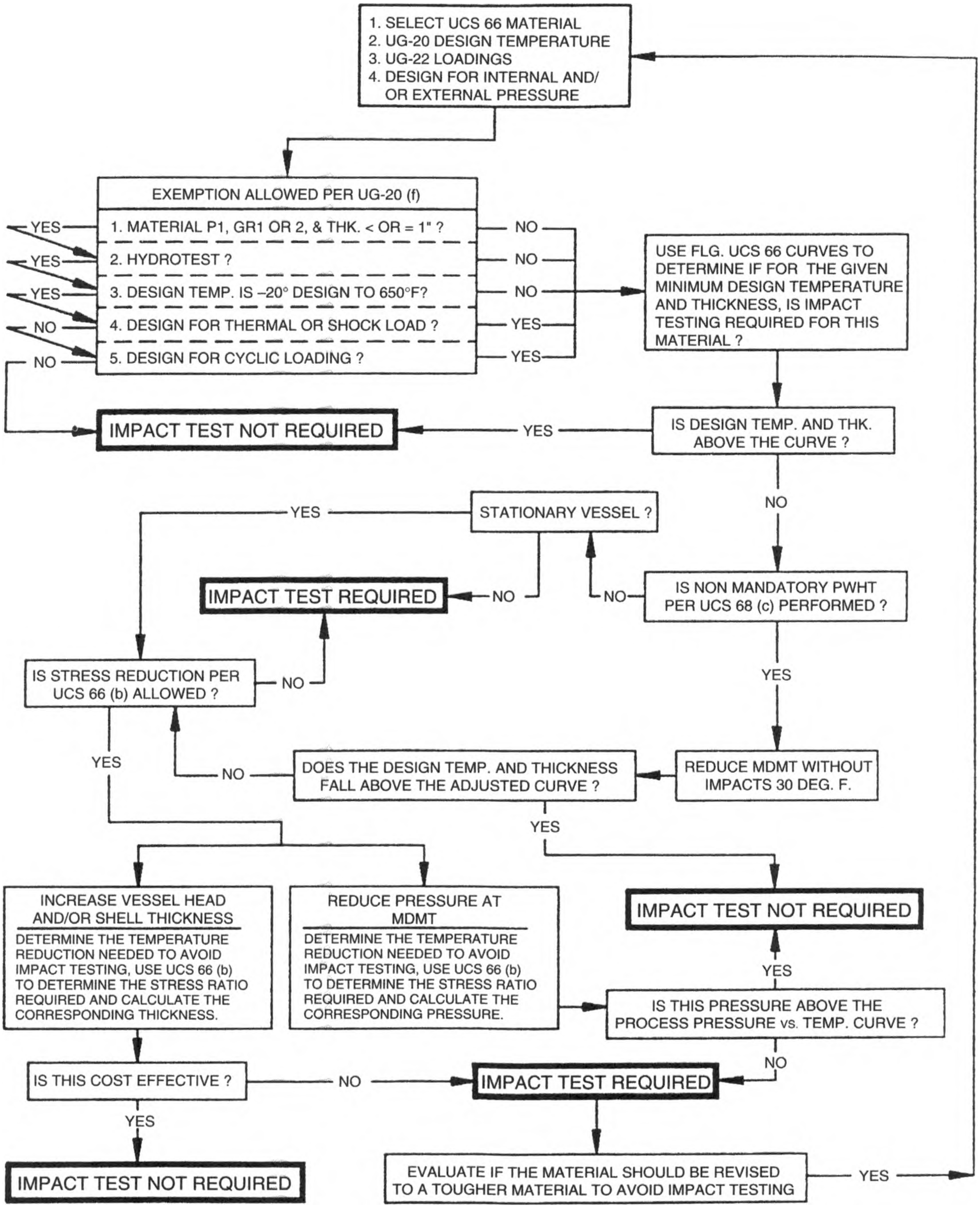


Figure 2-44. Flow chart showing decision-making process to determine MDMT and impact-testing requirements.

Procedure 2-15: Buckling of Thin Wall Cylindrical Shells [21]

This section provides commentary on the buckling of cylinders subject to external pressure, uniform axial compression, a bending moment across the cross-section, and in-plane shear stresses. Cylinders may also be subject to any combination of these loads. Unstiffened or large ring-stiffened cylinders may fail by local buckling or column buckling, and small ring-stiffened cylinders may fail by local buckling, column buckling, general instability, or local buckling of ring stiffeners. The rings described herein are assumed to be circumferential ring stiffeners. The terms 'large' and 'small' are consistent with Code Case 2286 (and the incorporated Code Case 2286 methodology in Section VIII, Division 2, Part 4), where the term 'large' refers to a bulkhead where the number of circumferential lobes is assumed to be two ($n = 2$) and the term 'small' refers to any larger number of circumferential lobes.

Local buckling describes failure by buckling of the cylinder in a radial direction. Column buckling for a cylinder describes failure by out-of-plane buckling while the shape of the cross section prior to buckling is circular. General instability describes failure where one or more ring stiffeners buckles along with the cylinder in a circumferential pattern with at least two waves. Local buckling of ring stiffeners refers to buckling of elements of the ring stiffener.

Note that an element in compression subject to 1) elastic buckling, will buckle before the element can develop the yield stress, 2) inelastic buckling, will buckle after some of the element develops the yield stress, and 3) plastic collapse, will collapse after all of the element develops the yield stress.

External Pressure

Buckling of the cylinder due to external pressure may occur in the elastic, inelastic, or plastic range. It is a function of L/r_g and D/t ratios as well as the physical properties of the material. Decreasing the effective length below the critical length has a positive effect the strength of the cylinder under external pressure. The critical length is the length at which the unstiffened cylinder

Decreasing the effective length may be done with ring stiffeners. Where cylinders with 'large' stiffening rings are used, the large rings will act as bulkheads whereby the shell portion between the rings acts similar to an unstiffened cylinder. In this case the large rings are assumed not to buckle with the shell. In general, where cylinders with 'small' stiffening rings are used the failure mode of general instability is introduced. Specifically within the Code (Code Case 2286 and Section VIII, Division 2, Part 4), the general instability failure mode is precluded by increasing the required moment of inertia of a small ring by a value of 20%. The difference between the large and small rings in Code Case 2286 is seen by using the more general equation representing the small ring, and using a value of $n = 2$ which yields the equation for the large ring. Figure 2-45 illustrates the meaning of circumferential buckling waves, where the circle represents the original shape.

Local buckling of ring stiffeners may be accomplished by using compact shapes for ring elements. Code Case 2286 and Section VIII, Division 2, Part 4 include geometry requirements (used from AISC) to ensure local buckling of ring stiffeners is avoided.

Axial Compression

Cylinders subject to uniform axial compression can fail by global or local buckling in the elastic, inelastic, and plastic range. Global buckling is usually determined by the length to radius of gyration ratio (L/r_g), and local buckling is determined by the diameter to thickness ratio (D/t). Each of these can be either elastic or inelastic. In some cases, circumferential ring stiffeners can be used on cylinders, but these are used mostly when external pressure is a concern. They may have a positive impact on axial compression if they are close enough to each other. Code Case 2286 and Section VIII, Division 2, Part 4 indicate that if the rings are less than $15(R_o t)^{1/2}$ from each other, then the circumferential rings are permitted to increase the calculated local buckling strength due to axial compression and bending. Longitudinal, or stringer, stiffeners may be used to increase the allowable axial capacity of the cylinder; however these are not typically used in process plants and are more common in offshore platform supports.

Cylinders under axial compression are more sensitive to geometric imperfections in the elastic range than in the

Figure 2-45. Circumferential buckling waves for $n = 2$ (left) and $n = 6$ (right)

inelastic range. For Code Case 2286, the reduction factor for shape imperfections is 0.207 for D_0/t values of greater than or equal to 1,247. The factor becomes higher for lower D_0/t ratios. The capacity reduction factors are already built into the allowable stress equations in Code Case 2286.

Bending Moment

Cylinders subject to a bending moment across the cross-section have similar characteristics as the case of a cylinder under uniform axial compression. Ovaling of the cross-section occurs during bending however as outlined in Code Case 2286 the allowable stress for a cylinder in bending is greater than a cylinder under uniform axial compression for the same geometry.

Shear

Cylinders subject to in-plane shear stresses can also fail in the elastic, inelastic, and plastic range. Though shear buckling is rarely a controlling factor in the design of process vessels, the interaction may have an unfavorable effect.

Interaction

The effect of combining internal/external pressure with axial tension/compression may be represented by an ellipse created using a yield criterion. In the four quadrants created by the yield criterion, four combinations of the internal/external pressure with axial tension/compression may be represented. In the case of external pressure, quadrants three and four (external pressure and axial compression, and external pressure and axial tension) are representative of what is discussed here. Furthermore, only quadrant three is evaluated in Code Case 2286 and Section VIII, Division 2, Part 4 since the topic of internal pressure is not addressed.

Procedure 2-16: Optimum Vessel Proportions [16-20]

This procedure specifically addresses drums but can be made applicable to any kind of vessel. The basic question is: What vessel proportions, usually expressed as L/D ratio, will give the lowest weight for a given volume? The maximum volume for the least surface area, and weight, is of course a sphere. Unfortunately, spheres are generally more expensive to build. Thus, spheres are not the most economical option until you get to very large volumes and for some process applications where that shape is required.

For vessels without pressure, atmospheric storage vessels, for example, the optimum L/D ratio is 1, again using the criteria for the maximum volume for the minimum surface area. This optimum L/D ratio varies with the following parameters:

Pressure.
Allowable stress.
Corrosion allowance.
Joint efficiency.

In *Process Equipment Design*, Brownell and Young suggest that for vessels less than 2 in. in thickness, the optimum L/D ratio is 6 and for greater thicknesses is 8. However, this does not account for the parameters just shown. Others have suggested a further breakdown by pressure categories:

Pressure (PSIG)	L/D Ratio
0–250	3
250–500	4
>500	5

Although this refinement is an improvement, it still does not factor in all of the variables. But before describing the actual procedure, a brief description of the sizing of drums in general is warranted. Here are some typical types of drums:

Knock-out drums.
Accumulator drums.
Suction drums.
Liquid–vapor separators.
Liquid–liquid separators.
Storage vessels.
Surge drums.

Typically the sizing of drums is related to a process consideration such as liquid holdup (surge), storage volume, or velocity considerations for separation. Surge volume in process units relates to the response time required for the alarms and operators to respond to upstream or downstream conditions.

For small liquid holdup, vessels tend to be vertical, while for large surge volumes they tend to be horizontal. For small volumes of liquid it may be necessary to increase the L/D ratio beyond the optimum proportions to allow for adequate surge control. Thus there may be an economic L/D ratio for determining the least amount of metal for the given process conditions as well as a practical operating L/D ratio.

For liquid–vapor separators the diameter of the vessel is determined by the velocity of the product and the time it takes for the separation to occur. Baffles and demister pads can speed up the process. In addition, liquid–vapor separators must provide for minimum vapor spaces. The sizing of vessels is of course beyond this discussion and is the subject of numerous articles.

An economic L/D ratio is between 1 and 10. L/D ratios greater than 10 may produce the lowest surface-area-to-volume ratio but should be considered impractical for most applications. Obviously plot space is also a consideration in ultimate cost. In general, the higher the pressure the larger the ratio, and the lower the pressure the lower the ratio. As previously stated, the optimum L/D ratio for an atmospheric drum is 1. Average pressure vessels will range between 3 and 5.

Two procedures are included here and are called Method 1 and Method 2. The two procedures, though similar in execution, yield different results. Both methods take into account pressure, corrosion, joint efficiency, and allowable stress. Even with this much detail, it is impossible to determine exactly what proportions will yield the lowest overall cost, since there are many more variables that enter into the ultimate cost of a vessel. However, determining the lowest weight is probably the best parameter in achieving the lowest cost.

The procedure for determining the optimum L/D ratios for the two methods is as follows:

Given

V, volume
P, pressure
C, corrosion allowance
S, allowable stress
E, joint efficiency

Method 1

1. Calculate F_1.
2. From Fig. 2-46, using F_1 and vessel volume, V, determine the vessel diameter, D.
3. Use D and V to calculate the required length, L.

Method 2

1. Calculate F_2.
2. From Fig. 2-47 determine L/D ratio.
3. From the L/D ratio, calculate the diameter, D.
4. Use D and V to calculate the required length, L.

Table 2-8
Optimum vessel proportions—comparison of two methods

V (cu. ft.)	P (PSIG)	Method[1]	D (ft)	L (ft)	t (in.)	W (lb)	L/D
1500	150	1	7.5	34	0.5625	20,365	4.5
		2	8.5	23.6	0.625	20,086	2.8
	300	1	6	53	0.8125	35,703	8.8
		2	7.5	31.5	0.8125	28,668	4.2
2000	150	1	7	52	0.5	25,507	7.4
		2	9	28.4	0.625	24,980	3.2
	300	1	6.5	61	0.875	51,179	9.4
		2	8.5	32.4	1.125	39,747	3.8
3000	150	1	8.5	53	0.625	40,106	6.3
		2	10.5	31.1	0.6875	35,537	3
	300	1	7.5	68	0.9375	65,975	9.1
		2	9.5	39.2	1.25	69,717	4.1
5000	150	1	10	64	0.6875	62,513	6.4
		2	11.5	44.3	1.125	86,781	3.9
	300	1	8.5	88	1.125	107,861	10.4
		2	11.5	44.3	1.375	106,067	3.9

[1]*Methods are as follows, based on graphs: Method 1: K. Abakians,* Hydrocarbon Processing, *June 1963. Method 2: S.P. Jawadekar,* Chemical Engineering, *Dec. 15,1980.*

Optimum Vessel Proportions for Vessels with (2) 2:1 S.E. Heads

Notation

- V = vessel volume, cu ft
- P = internal pressure, PSIG
- L = length, T–T, ft
- T = shell thickness, in.
- W = vessel weight, lb
- D = diameter, ft
- C = corrosion allowance, in.
- A = surface area, sq ft
- F_n = vessel ratios
- S = allowable stress, psi
- E = joint efficiency
- w = unit weight of plate, PSF
- L_e = equivalent length of cylinder equal to the volume of a vessel with (2) 2:1 S.E. heads
- h = height of cone, ft
- R = radius, ft
- C_1, K_1 = constant for ellipsoidal heads

Diameter for Different L/D Ratios

L/D	D
3	$\sqrt[3]{\frac{6V}{5\pi}}$
4	$\sqrt[3]{\frac{12V}{13\pi}}$
5	$\sqrt[3]{\frac{3V}{4\pi}}$
6	$\sqrt[3]{\frac{12V}{19\pi}}$
7	$\sqrt[3]{\frac{6V}{11\pi}}$
8	$\sqrt[3]{\frac{12V}{25\pi}}$

Equations

$$L_e = L + 0.332D$$

$$V = \frac{\pi D^3}{12} + \frac{\pi D^2 L}{4}$$

$$D = \sqrt[3]{\frac{4V}{\pi\left(0.3333 + \frac{L}{D}\right)}}$$

$$W = Aw$$

$$A = 2.18\,D^2 + \pi DL$$

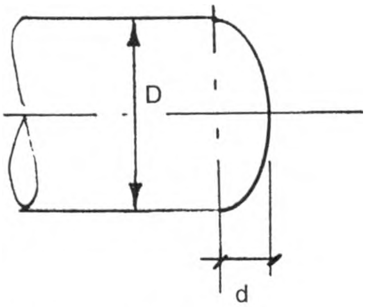
$$t = \frac{PR}{SE - 0.6P} + C$$

$$L = \frac{4V}{\pi D^2} - \frac{D}{3}$$

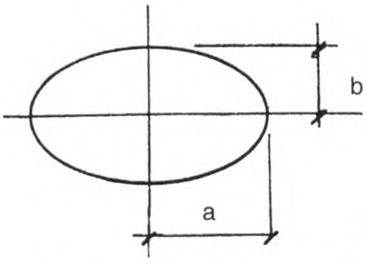
$$F_1 = \frac{P}{CSE}$$

$$F_2 = C\left(\frac{SE}{P} - 0.6\right)$$

Atmospheric Tank Proportions



Ellipsoidal Heads



Flat Elliptical Ends

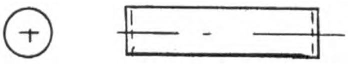
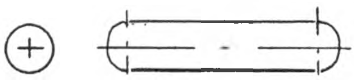
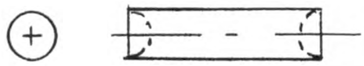
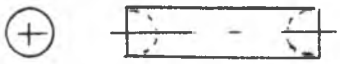
$$K_1 = \frac{2d}{R}$$

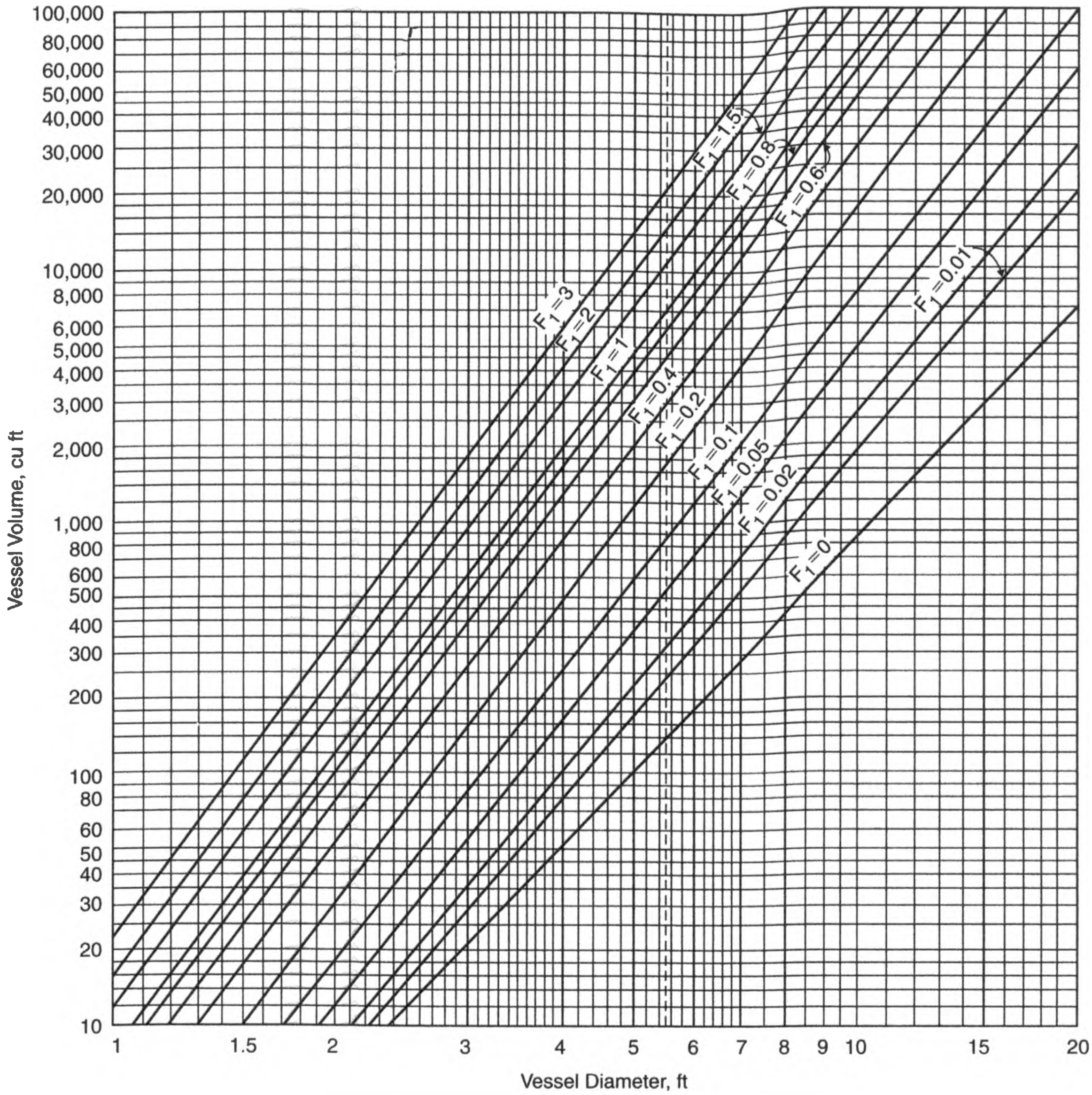
$$K_2 = \frac{b}{a}$$

$$C_1 = 2 + \frac{K_1^2}{\sqrt{1 - K_1^2}} \ln\left(\frac{1 + \sqrt{1 - K_1^2}}{1 - \sqrt{1 - K_1^2}}\right)$$

Note: For 2:1 S.E. Heads, $C_1 = 2.76$ and $K_1 = 0.5$.

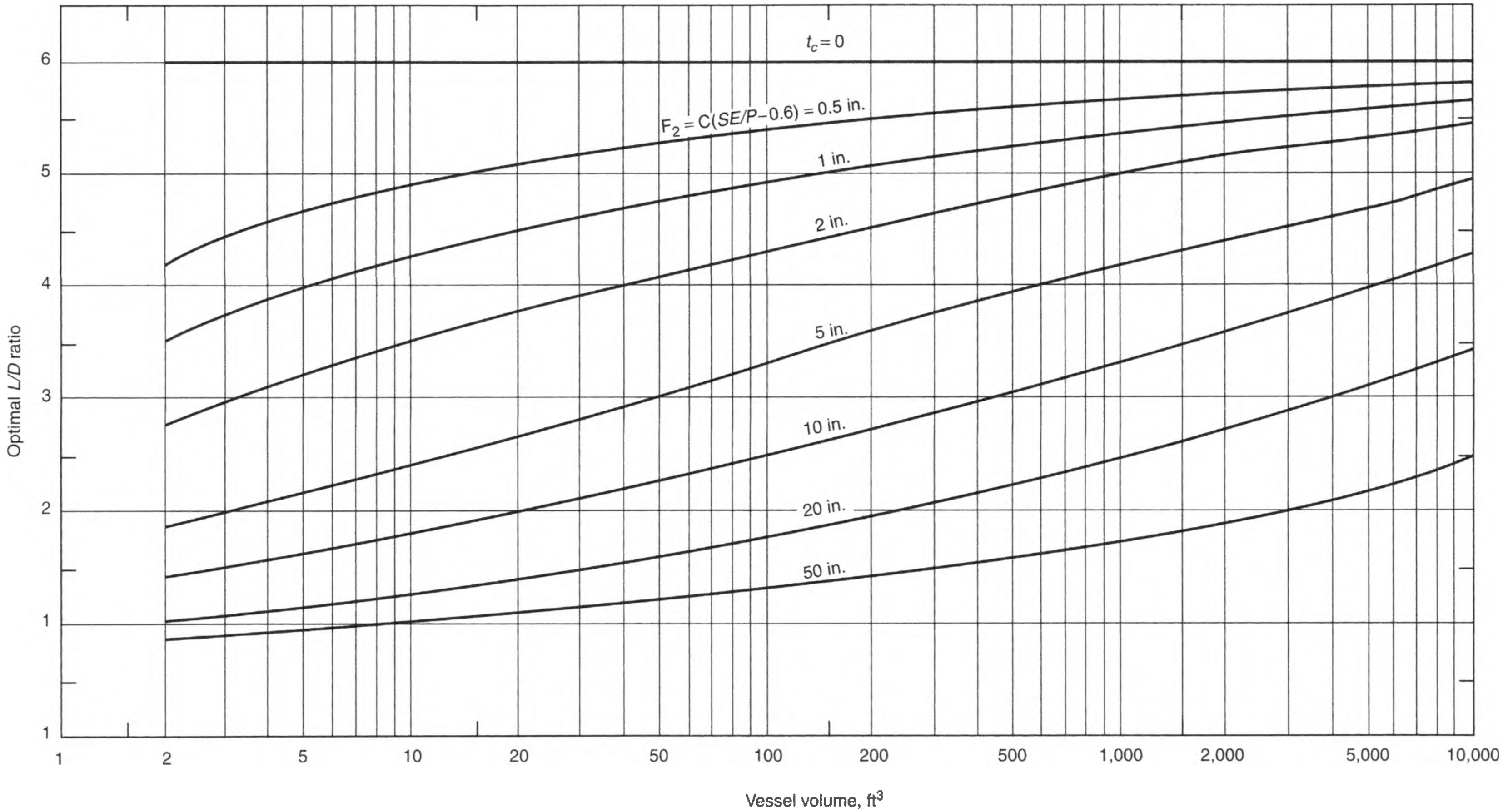
Table 2-9
Optimum tank proportions

Case	Optimum Proportions	Volume
Cylinder with flat ends 	$L = D$	$2\pi R^3$
Cylinder with ellipsoidal heads 	$L = R(C_1 + 4K_1)$	$\pi R^3\left(\frac{3C_1 - 8K_1}{3}\right)$
Cylinder with internal ellipsoidal heads 	$L = R(C_1 + 4K_1)$	$\pi R^3\left(\frac{3C_1 + 8K_1}{3}\right)$
Cylinder with internal hemi-heads 	$L = 8R$	$6.66\pi R^3$
Cylinder with conical ends 	$h = 0.9R$ $L = 0.9R$	$1.5\pi R^3$
Cylinder with internal conical ends 	$h = 0.9R$ $L = 3.28R$	$2.68\pi R^3$
Elliptical tank with flat ends 	$L = 2K_2 a\sqrt{\frac{2}{1 + K_2^2}}$	$2K_2^2\pi a^3\sqrt{\frac{2}{1 + K_2^2}}$



(From K. Abakians, *Hydrocarbon Processing and Petroleum Refiner*, June 1963.)

Figure 2-46. Method 1: Chart for determining optimum diameter.



(From S.P. Jawadekar, *Chemical Engineering*, Dec. 15, 1980.)

Figure 2-47. Method 2: Chart for determining the optimum L/D ratio.

Procedure 2-17: Estimating Weights of Vessels and Vessel Components

Estimating of weights of vessels is an important aspect of vessel engineering. In the conceptual phase of projects, weights are estimated in order to determine costs and budgets for equipment, foundations, erection, and transportation. Estimated weights also help to get more accurate bids from suppliers. Accurate weights are necessary for the design of the vessel itself to determine forces and moments.

There are a number of different types of weights that are calculated. Each weight is used for different purposes.

1. ***Fabricated weight***: Total weight as fabricated in the shop.
2. ***Shipping weight***: Fabrication weight plus any weight added for shipping purposes, such as shipping saddles.
3. ***Erection weight***: Fabrication weight plus any weight installed for the erection of the equipment, such as any insulation, fireproofing, piping, ladders, platforms.
4. ***Empty weight***: The overall weight of the vessel sitting on the foundation, fully dressed, waiting for operating liquid.
5. ***Operating weight***: Empty weight plus any operating liquid weight.
6. ***Test weight***: This weight can be either shop or field test weight, that is, the vessel full of water.

There are a number of ways to estimate the weights of vessels, depending on the accuracy required. Vessel weights can be estimated based on computer design programs. These programs typically calculate the volume of metal for the vessel shell and head and add weights for supports, nozzles, trays, and other components. Another fast and easy way to get the volume of metal in the shell and heads is to use the surface area in square feet and multiply this by the unit weight for the required thickness in pounds per square foot.

In addition to the base weight of metal in the shell and heads, the designer must include an allowance for plate overages per Table 2-11. The mill never rolls the plates the exact specified thickness since there would be the possibility of being below thickness. The safety margin added by the mill is referred to as *plate overage* or *overweight percentage*. The plate overage varies by the thickness of the material.

In addition to the plate overage, the fabricator (or head manufacturer) also adds a *thinning allowance* to the head to ensure that the head meets the minimum thickness in all areas. Depending on the type of head, the diameter, and the thickness required, a thinning allowance can be determined. This can be as much as 1.5 in. for large-diameter hemi-heads over 4 in. thick! The metal does not disappear during the forming process but may "flow" to the areas of most work.

On a typical spun 2:1 S.E. head, the straight flange will get thicker and the knuckle will get thinner due to forming. The crown of the head should remain about the same. Therefore the completed head has a thickness averaging the initial thickness of the material being formed.

After the weights of all the components are added for a total weight, an additional percentage is typically added to allow for other components and welding. The typical percentages are as follows:

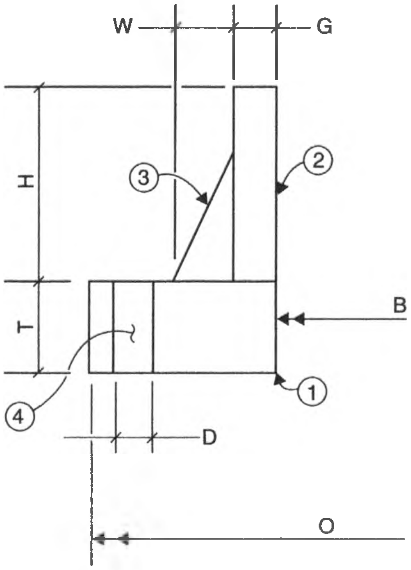
<50,000 lb	Add 10%
50,000-75,000 lb	Add 8%
75,000-100,000 lb	Add 6%
>100,000 lb	Add 5%

The weight of any individual component can easily be calculated based on the volume of the material times the unit density weight given in Table 2-11. Any shape can be determined by calculating the surface area times the thickness times the density. The designer need only remember the density of steel for most vessels of 0.2833 lb/in.^{3} to determine any weight. For vessels or components of other materials, either the density of that material or the factor for that material relative to carbon steel can be used. These values are also listed in the following tables.

Table 2.10
General formulas for computing weights of vessel components

ITEM		WEIGHT FORMULAS	THICKNESS FORMULAS	DATA
SHELL	Per Ft	$10.68\ D_m\ t$	ASME Section VIII, Div 1	D = ID of shell or cone, in
	C.S.	$.89\ D_m\ L\ t$	SHELL;	d = ID, small end of cone, in
	Other	$\pi\ D_m\ L\ t\ \delta$	$t = (P\ R_i) / (S\ E - .6\ P)$	D_m = Mean diameter of shell, in
SPHERE	C.S.	$.89\ D_m^2\ t$		t = Thickness, in
	Other	$\pi\ D_m^2\ t\ \delta$	2:1 S.E. HEAD;	A_c = Area of cone, in^2
HEMI	C.S.	$.445\ D_m^2\ t$	$t = (P\ R_i) / (S\ E - .2\ P)$	L = Length of shell or cone, in
	Other	$1.57\ D_m^2\ t\ \delta$		δ = Density of material, PCI
2:1 SE HEAD	C.S.	$.307\ D_m^2\ t$	HEMI HEAD;	R = Estimated radius of hemi head, in
	Other	$1.084\ D_m^2\ t\ \delta$	$t = (P\ R_i) / (2\ S\ E - .2\ P)$	R_i = Inside radius, in
CONE	C.S.	$.2833\ A_c\ t$		S = Allowable stress, PSI
	Other	$A_c\ t\ \delta$	CONE;	P = Internal pressure, PSIG
			$t = (P\ R_i) / [\cos \alpha\ (S\ E - .6\ P)]$	E = Joint efficiency
			ASME Section VIII, Div 2	α = Half apex angle of cone, degrees
			SHELL;	**FORMULAS**
			$t = R_i\ (e^{P/S} - 1)$	$D_m = D + t$ Use shell thickness in formula
NOTES			HEMI HEAD;	$R = .5\ D_m - .25\ t$
1. Add thinning allowance to all head thickness			$t = R_i\ (e^{5P/S} - 1)$	For 30° cone, $A_c = 1.57\ (D^2 - d^2)$
				For all other cones; $A_c = \pi\ [.5\ (D + d)]\ [L^2 + (.5\ (D - d)^2)]^{1/2}$

Calculation of Weight of Weld Neck Flange



Data

T = thickness of flange
O = flange OD
D = bolt hole diameter
H = hub height
G = hub thickness at small end
W = width of hub
B = ID of flange
V = volume, in.^{3}
d = density of material, lb/in.^{3}
N = number of bolts/holes

Formulas

1. $\left[\frac{O^2\pi}{4} - \frac{B^2\pi}{4}\right]T = (+)$
2. $\{B + G\}\pi GH = (+)$
3. $0.5\{(B + 2G + W)\pi WH\} = (+)$
4. $\left[\frac{D^2\pi}{4}\right]TN = (-)$
5. $V = 1 + 2 + 3 - 4 =$
6. $\text{weight} = V \times d =$

Table 2-11
Weights of carbon steel plate and stainless steel sheet, PSF

Thickness (in.)	Raw Weight	Weight Including % Overweight	% Overweight	Thickness (in.)	Raw Weight	Weight Including % Overweight	% Overweight
0.125	5.1	5.65	10.75	0.875	35.7	36.91	3.38
0.1875	7.66	8.34	9	0.9375	38.28	39.54	3.38
0.25	10.2	10.97	7.5	1	40.8	42.02	3
0.3125	12.76	13.61	6.75	1.0625	43.38	44.65	3
0.375	15.3	16.22	6	1.125	45.94	47.28	3
0.4375	17.86	18.79	5.25	1.25	51	52.53	3
0.5	20.4	21.32	4.5	1.375	56.15	57.78	3
0.5625	22.97	23.98	4.5	1.5	61.2	63.04	3
0.625	25.6	26.46	3.75	1.625	66.35	68.29	3
0.6875	28.07	29.1	3.75	1.75	71.4	73.54	3
0.75	30.6	31.63	3.38	1.875	76.56	78.8	3
0.8125	33.17	34.27	3.38	2	81.6	84.05	3
Stainless Steel Sheet							
Thickness Gauge	Weight	Thickness Gauge	Weight				
10 GA	5.91	20 GA	1.58				
11 GA	5.25	24 GA	1.05				
12 GA	4.59	26 GA	0.788				
14 GA	3.28	28 GA	0.656				
16 GA	2.63	30 GA	0.525				
18 GA	2.1						

Note: *% Overweight is based on standard mill tolerance added to the thickness of plate to guarantee minimum thickness.*

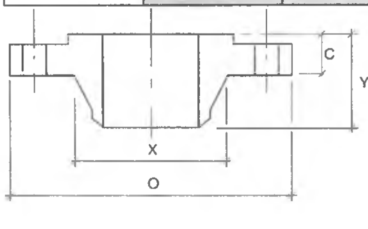
Table 2-12
Weights of flanges, 2 in. to 24 in. (lb)

Size (in.)	Rating					
	150	300	600	900	1500	2500
2	9	10	12	25	25	42
	4	*8*	*10*	*25*	*25*	*39*
3	14	16	20	32	48	94
	14	*16*	*20*	*32*	*48*	*86*
4	16	26	41	51	73	145
	19	*27*	*41*	*54*	*73*	*135*
6	25	45	77	110	164	380
	28	*50*	*86*	*113*	*159*	*345*
8	40	70	111	187	273	580
	48	*80*	*140*	*197*	*301*	*530*
10	56	94	180	268	454	1075
	70	*120*	*230*	*290*	*507*	*1025*
12	86	140	226	372	670	1525
	105	*184*	*295*	*413*	*775*	*1300*
14	111	190	334	562	940	
	135	*249*	*378*	*494*	*975*	
16	141	250	462	685	1250	
	176	*324*	*527*	*619*	*1300*	
18	153	305	531	924	1625	
	214	*416*	*665*	*880*	*1750*	
20	188	380	678	1164	2050	
	284	*516*	*855*	*1107*	*2225*	
24	270	540	959	2107	3325	
	398	*763*	*1175*	*2099*	*3625*	

Notes:
1. Top value in block is the weight of a weld neck flange.
2. Bottom value in block is the weight of a blind flange.

Table 2-13
Dimensions and weights of large-diameter flanges, 26 in. to 60 in., ASME B16.47, Series B

Size (in.)		Dimensions						Weight	
		O	C	Y	X	N	d	RFWN	Blind
26	150	30.94	1.62	3.5	29.64	36	0.75	120	340
	300	34.12	3.5	5.69	27.62	32	1.25	400	860
28	150	32.94	1.75	3.75	28.94	40	0.75	140	415
	300	36.25	3.5	5.88	29.75	36	1.25	450	970
30	150	34.94	1.75	3.94	31	44	0.75	150	470
	300	39	3.69	6.22	32	36	1.375	550	1250
32	150	37.06	1.81	4.25	33.06	48	0.75	170	550
	300	41.5	4.06	6.62	34	32	1.5	685	1550
34	150	39.56	1.94	4.34	35.12	40	0.875	210	660
	300	43.62	4.06	6.81	36.12	36	1.5	750	1635
36	150	41.62	2.06	4.62	37.19	44	0.875	240	780
	300	46.12	4.06	7.12	38	32	1.625	840	1835
38	150	44.25	2.12	4.88	39.12	40	1	290	905
	300	48.12	4.38	7.56	40	36	1.625	915	2150
40	150	46.25	2.19	5.06	41.31	44	1	310	1025
	300	50.12	4.56	7.81	42	40	1.625	990	2425
42	150	48.25	2.31	5.25	43.38	48	1	345	1175
	300	52.5	4.69	8.06	44	36	1.75	1135	2745
44	150	50.25	2.38	5.38	45.38	52	1	370	1310
	300	54.5	5	8.44	46.19	40	1.75	12635	3150
46	150	52.81	2.44	5.69	47.44	40	1.125	435	1490
	300	57.5	5.06	8.75	43.38	36	1.875	1470	3560
48	150	54.81	2.56	5.88	49.5	44	1.125	480	1680
	300	59.5	5.06	8.81	50.31	40	2	1575	3850
50	150	56.81	2.69	6.06	51.5	48	1.125	520	1900
	300	61.5	5.44	9.25	52.38	44	2	1710	4365
52	150	58.81	2.75	6.19	53.56	52	1.125	550	2080
	300	63.5	5.62	9.56	54.44	48	2	1840	4800
54	150	61	2.81	6.38	55.62	56	1.125	620	2290
	300	65.88	5.38	9.44	56.5	48	2	1980	4965
56	150	63	2.88	6.56	57.69	60	1.125	680	2500
	300	69.5	6.06	10.56	58.81	36	2.375	2600	6240
58	150	65.94	2.94	6.88	59.69	48	1.25	830	2800
	300	71.94	6.06	10.81	60.94	40	2.375	2770	6675
60	150	67.94	3	7.06	61.81	52	1.25	1075	3030
	300	73.94	5.94	10.69	62.94	40	2.375	2870	6930



d = bolt hole diameter, in.
N = number of bolt holes

Table 2-14
Weights of nozzles and manways, 1 in. to 60 in.

Rating Size	150	300	600	900	1500	2500	Rating Size	150	300
1″	4	5	6	11	13	18	26″	230	490
1.5″	6	9	11	16	21	34	28″	260	550
2″	8	10	13	27	42	47	30″	280 ***1126***	665 ***2880***
3″	15	21	22	34	78	110	32″	305	805
4″	21	31	42	60	110	160	34″	345	880
6″	37	55	81	127	215	360	36″	385 ***1726***	980 ***3685***
8″	54	81	132	207	335	520	38″	440	1045
10″	72	116	215	310	650	1000	40″	465	1125
12″	107	158	261	418	940	1350	42″	510 ***2387***	1275 ***4600***
14″	132	232	407	613	950		44″	540	1365
16″	163 ***400***	289 ***705***	549 ***1260***	751 ***1610***	1175 ***3250***		46″	610	1620
18″	200 ***479***	340 ***875***	639 ***1530***	1042 ***2270***	1475 ***5200***		48″	660 ***2970***	1775 ***5515***
20″	235 ***593***	421 ***1065***	783 ***1925***	1283 ***2800***	1725 ***5430***		50″	705	1915
24″	310 ***825***	587 ***1600***	1100 ***2685***	2287 ***5455***	2650 ***9000***		52″	740	2025
							54″	800	2170
							56″	835	2790
							58″	970	2970
							60″	1050 ***5760***	3080 ***8675***

Notes:

1. Weights include pipe and WN flg.
2. Lower weight in box is weight of manway and includes nozzle, blind, and bolts.
3. Class 1500 manways are based on LWN.

Table 2-15
Weights of valve trays, PSF

Dia.	One Pass		Two Pass		Four Pass	
	C.S.	Alloy	C.S.	Alloy	C.S.	Alloy
<84″	13	11	14	12		
84″ to 180″	12	10	13	11	15	13
>180″	11.5	9.5	12.5	10.5	14.5	12.5

Notes:

1. Compute area on total cross-sectional area of vessel. The downcomer areas compensate for the weight of downcomers themselves.
2. Tray weights include weights of trays and downcomers.

Table 2-16
Weights of tray supports and downcomer bars (lb)

ID (in.)	C.S.	Alloy	ID (in.)	C.S.	Alloy	ID (in.)	C.S.	Alloy
30	25	17	102	113	72	174	287	174
36	28	19	108	119	75	180	294	178
42	34	23	114	123	77	186	344	207
48	37	25	120	176	108	192	354	212
54	44	35	126	183	112	198	362	218
60	47	38	132	188	116	204	374	226
66	50	40	138	195	119	210	385	231
72	53	44	144	202	122	216	396	239
78	55	46	150	244	149	222	407	245
84	99	62	156	251	152	228	418	252
90	103	65	162	271	162	234	428	259
96	109	68	168	278	167	240	440	265

Notes:

1. Tray support weights include downcomer bolting bars as well.
2. Tray support ring sizes are as follows:

   C.S.: 1/2″ × 2 1/2″
   Alloy: 5/16″ × 2 1/2″

Table 2-17
Thinning allowance for heads

Thickness	Diameter	
	<150″	>150″
0.125″ to 1″	0.0625	None
1″ to 2″	0.125	0.25
2″ to 3″	0.25	0.25
3″ to 3.75″	0.375	0.375
3.75″ to 4″	0.5	0.5
over 4.25″	0.75	0.75

Table 2-18
Weights of pipe (PLF)

Size (in.)	Schedule												
	10	20	30	STD	40	60	XS	80	100	120	140	160	XXS
0.75	0.8572			1.131	1.131		1.474	1.474				1.937	2.441
1	1.404			1.679	1.679		2.172	2.172				2.844	3.659
1.25	1.806			2.273	2.273		2.997	2.997				3.765	5.214
1.5	2.085			2.718	2.718		3.631	3.631				4.859	6.408
2	2.638			3.653	3.653		5.022	5.022				7.444	9.029
2.5	3.531			5.793	5.793		7.661	7.661				10.01	13.69
3	4.332			7.576	7.576		10.25	10.25				14.32	18.58
3.5	4.973			9.109	9.109		12.5	12.5				17.69	22.85
4	5.613			10.79	10.79	12.66	14.98	14.98		19		22.51	27.54
5	7.77			14.62	14.62		20.78	20.78		27.04		32.96	38.55
6	9.289		17.02	18.97	18.97		28.57	28.57		36.39		45.3	53.16
8	13.4	22.36	24.7	28.55	28.55	35.64	43.39	43.39	50.87	60.63	67.76	74.69	72.42
10	18.2	28.04	34.24	40.48	40.48	54.74	54.74	64.33	76.93	89.2	104.1	115.6	
12	24.2	33.38	43.77	49.56	53.52	73.16	65.42	88.51	107.2	125.5	139.7	160.3	
14	36.71	45.68	54.57	54.57	63.37	84.91	72.09	106.1	130.7	150.7	170.2	189.1	
16	42.05	52.36	62.58	62.58	82.77	107.5	82.77	136.5	164.8	192.3	223.5	245.1	
18	47.39	59.03	82.06	70.59	104.6	138.2	93.45	170.8	208	244.1	274.2	308.5	
20	52.73	78.6	104.1	78.6	122.9	166.4	104.1	208.9	256.1	296.4	341.1	379	
22	58.1			86.6			114.8						
24	63.41	94.62	140.8	94.62	171.2	238.1	125.5	296.4	367.4	429.4	483.1	541.9	
26				102.6			136.2						
28				110.7			146.8						
30	98.9	157.6	196.1	118.7			157.6						
32				126.7			168.2						
34				134.7			178.9						
36				142.7			189.6						
42				166.7			221.6						

Table 2-19
Weights of alloy stud bolts + (2) nuts per 100 pieces

Length (in.)	Stud Diameter, in.												
	0.5	0.625	0.75	0.875	1	1.125	1.25	1.375	1.5	1.625	1.75	1.875	2
3	29	49	76										
3.25	30	51	79										
3.5	31	53	82	120									
3.75	32	55	85	124									
4	34	57	88	128	188								
4.25	35	59	91	132	194								
4.5	36	61	94	136	199	246							
4.75	37	63	97	140	205	253							
5	39	65	100	144	210	259	330						
5.25	40	67	103	148	216	266	338						
5.5	41	69	106	152	221	272	347						
5.75		71	109	156	227	279	355						
6		73	112	160	232	285	363	460	568	700			
6.25			115	164	238	292	371	470	580	714			
6.5			118	168	243	298	380	480	592	728			
6.75				172	249	305	388	490	604	742			
7				176	254	311	396	500	616	756	900	1062	1227
7.25					260	318	404	510	628	770	916	1080	1248
7.5					265	324	413	520	640	784	932	1098	1270
7.75					271	331	421	530	652	798	948	1116	1291
8					276	337	429	540	664	812	964	1134	1312
8.25						344	437	550	676	826	980	1152	1334
8.5						350	446	560	688	840	996	1170	1355
8.75						357	454	570	700	854	1012	1188	1376
9						363	462	580	712	868	1028	1206	1398
9.25						370	470	590	724	882	1044	1224	1419
9.5						376	479	600	736	896	1060	1242	1440
9.75						383	487	610	748	910	1076	1260	1462
10						389	495	620	760	924	1092	1278	1483
10.25								630	772	938	1108	1296	1508
10.5								640	784	952	1124	1314	1526
10.75								650	796	966	1140	1332	1547
11								660	808	980	1156	1350	1569
11.25								670	820	994	1172	1368	1590
11.5								680	832	1008	1188	1385	1611
11.75								690	844	1022	1204	1404	1633
12								700	856	1036	1220	1422	1654
Add per additional 1/4″ length	1.5	2	3	4	5.5	6.5	8.5	10	12	14	16	18	21.5

Table 2-20
Weights of saddles and baseplates (lb)

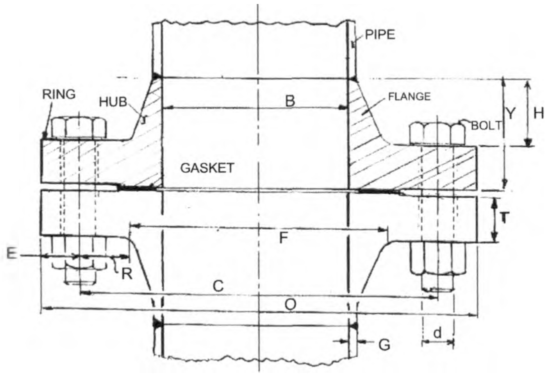
ID (in.)	Two CS Saddles	1/2 × 6 Baseplate	3/4 × 8 Baseplate	ID (in.)	Two CS Saddles	1/2 × 6 Baseplate	3/4 × 8 Baseplate
24	100	70	150	138	3060	390	790
30	150	90	190	144	3400	410	820
36	260	105	215	150	3700	430	855
42	330	125	250	156	4000	450	885
48	380	140	285	162	4250	460	920
54	440	160	320	168	4500	480	950
60	510	170	350	174	4750	490	985
66	590	190	385	180	5000	510	1020
72	680	200	420	186	5250	530	1050
78	910	220	450	192	5500	540	1080
84	1050	240	485	198	5750	560	1120
90	1160	260	520	204	6000	580	1150
96	1230	280	550	210	6250	590	1190
102	1730	290	585	216	6500	610	1220
108	1870	310	615	222	6750	630	1250
114	2330	330	650	228	7000	650	1290
120	2440	340	690	238	7250	660	1320
126	2700	360	720	240	7500	680	1360
132	2880	380	755				

Table 2-21
Density of various materials

Material	d (lb/in.^{3})	PCF	Weight Relative to C.S.
Steel	0.2833	490	1.00
300 SST	0.286	494	1.02
400 SST	0.283	489	0.99
Nickel 200	0.321	555	1.13
Permanickel 300	0.316	546	1.12
Monel 400	0.319	551	1.13
Monel 500	0.306	529	1.08
Inconel 600	0.304	525	1.07
Inconel 625	0.305	527	1.08
Incoloy 800	0.287	496	1.01
Incoloy 825	0.294	508	1.04
Hastelloy C4	0.312	539	1.10
Hastelloy G30	0.297	513	1.05
Aluminum	0.098	165	0.35
Brass	0.297	513	1.05
Cast iron	0.258	446	0.91
Ductile iron	0.278	480	0.98
Copper	0.322	556	1.14
Bronze	0.319	552	1.13

SPECIAL DESIGN RFWN FLANGES FOR CLASS 2500#

Dimensions for Weld Neck Flange



Notes:

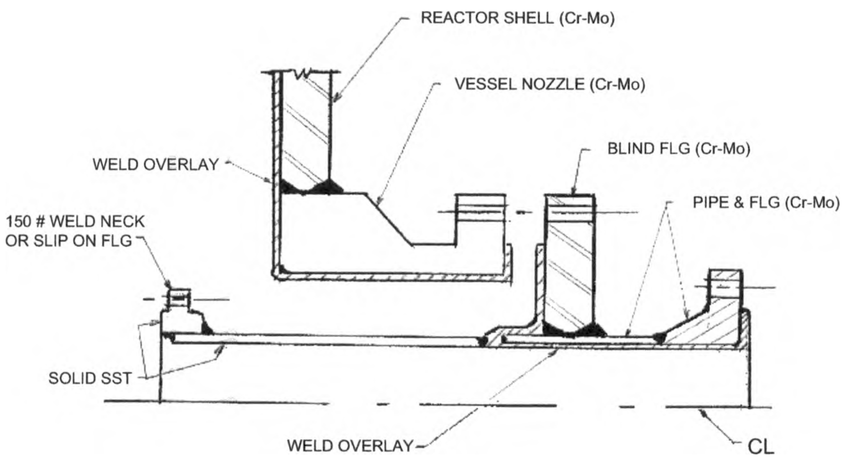
1. These dimensions and weights should be used for estimating purposes only.

2. ASME B16.5 dimensions for 2500# flanges stop at 12" NPS. These dimensions should be considered an extension of the B16.5 size range.

3. These dimensions can be used as a starting place for special designed flanges.

SIZE B	O	T	Y	C	N	d	G	E	R	F	H	WT
14	34	7.5	18.75	27	16	2.75	1.406	2.63	3.38	20.25	11	1750
16	38	7.75	19.5	30	16	3	1.593	2.88	3.63	22.75	11.5	2250
18	41	8.25	20.5	34	20	3	1.781	2.88	3.63	26.75	12	2850
20	45	8.5	22	39.25	24	3	1.93	2.88	3.63	32	13.25	3800
24	52	9.5	24	45	20	3.5	2.34	3.5	4	37	14.25	5500

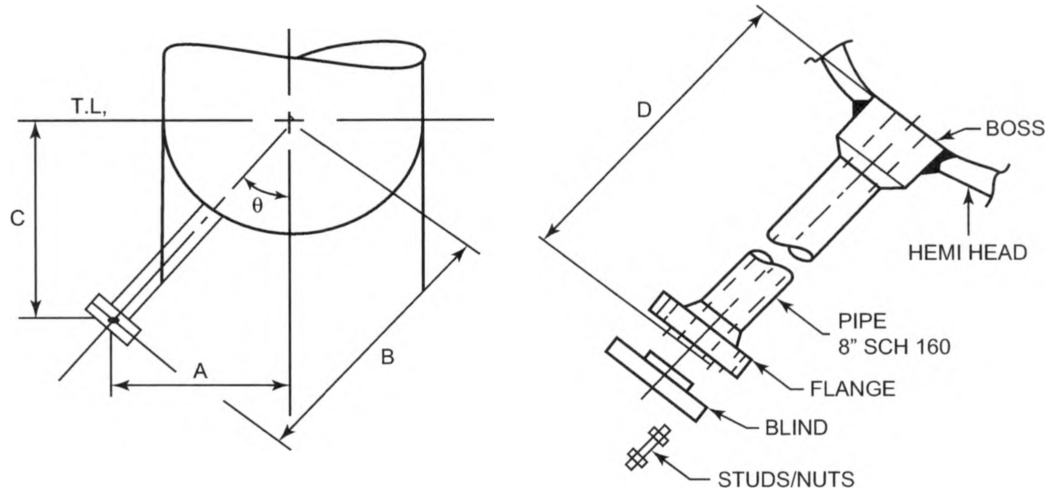
WEIGHTS OF QUENCH NOZZLES



SIZE	RATING	B	O	W_N	W_b	W_S	W_f	W_p	W_T
10" X 4"	600	10	9	433	230	70	41	39	815
	900			558	290	93	51		1030
	1500			888	507	181	73		1690
	2500			1625	1025	236	145		3070
12" X 6"	600	12	11	533	295	89	77	70	1065
	900			715	413	122	110		1425
	1500			1683	775	302	164		3000
	2500			1995	1300	548	380		4300
14" X 8"	600	14	13.5	631	378	116	111	115	1350
	900			817	494	157	187		1770
	1500			1929	975	383	273		3675
	2500			2400	1600	780	580		7475

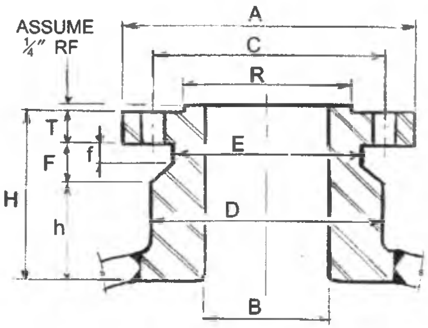
DATA	NOTES
W_N = Weight, Nozzle, Lbs	1) 600# & 900# assumes Type 'HB' Conn
W_b = Weight, Blind, Lbs	2) 1500# & 2500# assumes Type 'F' Conn
W_s = Weight, Studs & Nuts for Blind, Lbs	3) W_p assumes weight of 12" of Sch 160 Pipe and the 150# internal flange
W_f = Weight, Flange, Lbs	
W_p = Weight, Internal Pipe and Flange, Lbs	
W_T = Weight, Total, Lbs	
B = Bore of Vessel Nozzle, In	
O = OD of Internal Flange, In	

DIMENSIONS/WEIGHTS OF BOTTOM CATALYST DUMP NOZZLES



DIA	RATING	A	B	C	D	θ	WEIGHT	DIA	RATING	A	B	C	D	θ	WEIGHT
8	600	69	139	120	91	30	875	13	600	96	169	138	91	35	875
	900						1030		900						1030
	1500						1260		1500						1260
	2500						1825		2500						1825
9	600	76	152	132	98	30	925	14	600	101	176	144	92	35	885
	900						1080		900						1040
	1500						1310		1500						1270
	2500						1875		2500						1830
10	600	84	146	120	87	35	850	15	600	111	172	132	82	40	825
	900						1000		900						980
	1500						1235		1500						1210
	2500						1800		2500						1775
11	600	88	139	126	73	35	770	16	600	121	188	144	92	40	840
	900						920		900						1040
	1500						1155		1500						1270
	2500						1715		2500						1835
12	600	92	161	132	89	35	870	NOTES:							
	900						1075	1. Weights & dimensions are for estimating purposes only							
	1500						1250	2. Dimensions are based on 8" nozzle							
	2500						1815	3. Dimensions are in Inches, weights in Lbs							

Table of Dimensions and Weights for Large Diameter, Self-Reinforced Manways

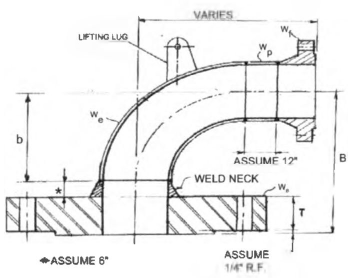


Formulas	Data
f = d + .25	N = Qty Studs d = Dia, Studs, in
C = A - 2 d ; Min = $(B_S N) / \pi$	L = Length of Studs, in C = Bolt Circle, in
E = C - X	W_N = Weight, Nozzle, Lbs W_S = Weight, Studs & Nuts, Lbs
R = B + 2a	X = Width across Flats of Nuts, in
F = f + (D - E)/2	a = Width of Raised Face, in = Gasket Width + .5" Min B_S = Min Bolt Spacing = 2.1 d
h = H - T - F -.25	
L = 2 T + 2 d + 1.5	
A = C + 2 d	

Nom Size	A	B	C	D	E	F	H	T	Studs			R	W_N	W_S
									N	d	L			
24" 600#	37	24	33	30	28.25	2.13	18	4	24	1.875	13.25	27.25	1500	370
24" 900#	41	24	35.5	31.62	29.5	3.188	20	5.5	20	2.5	17.5	27.25	2160	715
24" 1500#	46	24	39	33.62	30	5.563	22	8	16	3.5	24.5	27.25	3400	1550
24" 2500#	52	24	45	45	39.5	6.5	24	9.5	20	3.5	27.5	27.25	6000	2110
30" 600#	48	30	40.25	43	33.94	9	24	4.63	28	2	14.75	36.13	1600	535
30" 900#	50	30	43	44.5	35.75	10.44	26	6.56	20	2.5	19.62	36.5	3400	770
30" 1500#	52	30	44.75	46	37.5	12.63	28	8	24	2.5	22.5	39.25	5500	1025
30" 2500#	58	30	52	50	45.88	5.31	30	9.5	24	3	26.5	39.75	12000	1730
36" 600#	52	36	47.5	47	44	4	29	4.88	28	2.25	15.75	40.25	2500	730
36" 900#	54	36	51	51	45.63	6.44	31	6.75	20	3.5	22	40.25	3800	1820
36" 1500#	56	36	50	53	46	6.25	33	8.25	28	2.5	23	44.25	10,660	1115
36" 2500#	62	36	56	54.38	51.38	4.75	36	9.5	28	3	26.5	44.25	15,000	2020

Notes:
1) These dimensions and weights should be used for estimating purposes only. Detail designs have not been calculated.
2) All flange designs shown are non-standard except for the 24" 600# thru 1500# only.
3) All flange designs based on 2-1/4 Cr - 1 Mo material at 850 deg F.

REACTOR INLET ESTIMATED WEIGHTS



DATA
W_B = Weight, Blind, Lbs
W_f = Weight, Outlet Flange, Lbs
W_e = Weight, Elbow, Lbs
W_p = Weight, Pipe, Lbs
W_T = Weight, Total
$W_T = W_B + W_f + W_e + W_p$

BLIND

SIZE	RATING	T	W_B
24	600	4	1175
	900	5.5	2080
	1500	7.5	3500
	2500	9.5	5580
30	600	4.63	2080
	900	6.56	3725
	1500	8	5000
	2500	10.5	7140
36	600	4.88	3300
	900	6.75	5700
	1500	8.25	5400
	2500	12.25	9700

NOZZLE FLANGE

SIZE	RATING	W_f
10	600	180
	900	268
	1500	454
	2500	1075

NOZZLE FLANGE

SIZE	RATING	W_f
12	600	226
	900	372
	1500	670
	2500	800
14	600	334
	900	562
	1500	940
	2500	1750
16	600	462
	900	685
	1500	940
	2500	2260
18	600	531
	900	924
	1500	1625
	2500	2850
20	600	678
	900	1164
	1500	2050
	2500	3800

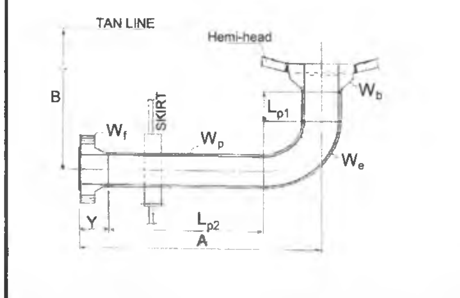
PIPE & ELBOW

SIZE	b	W_p	W_e
8	12	75	117
10	15	160	226
12	18	200	300
14	21	250	410
16	24	300	530
18	27	350	675
20	30	400	840

NOTES

1. Dim "B" = T + b + 6"
2. Pipe & elbow assumed as Sch 160
3. Weld neck is included in blind flg weight
4. Studs and nuts are not included

REACTOR BOTTOM OUTLET ESTIMATED WEIGHTS



DIA Ft	d in	A in	B in	Y in	L_{p1} in	L_{p2} in	WEIGHTS (Lbs) W_e	W_f	W_b	W_p	W_T
8	6	62	69	6.75	6	46.25	53	164	90	198	505
	8		72	8.38		41.63	117	273	150	300	840
	10		75	10		37	226	454	232	416	1330
9	6	68	75	6.75	6	52.25	53	164	90	220	527
	8		78	8.38		47.63	117	273	150	335	875
	10		81	10		43	226	454	232	475	1390
10	10	74	83	8.38	8	49	226	454	232	550	1470
	12		86	10		44.88	375	670	320	705	2070
	14		89	11.13		41.25	570	940	380	780	2670
11	10	80	92	8.38	8	55	226	454	232	610	1520
	12		95	10		50.88	375	670	320	785	2150
	14		98	11.13		47.25	570	940	380	875	2765
12	12	88	100	8.38	10	58.88	375	670	320	920	2285
	14		103	10		55.25	570	940	380	1035	2925
	16		106	11.13		51.75	800	1250	490	1260	3800
13	12	94	109	10	10	64.88	375	670	320	1000	2365
	14		112	11.13		61.25	570	940	380	1130	3020
	16		115	11.75		57.75	800	1250	490	1385	3925
14	14	100	120	11.13	12	67.25	570	940	380	1255	3145
	16		123	11.75		63.75	800	1250	490	1547	4087
	18		126	12.25		60.12	1200	1625	620	1860	5300
15	14	106	129	11.75	12	73.25	570	940	380	1350	3240
	16		132	12.25		69.75	800	1250	490	1670	4210
	18		135	12.88		66.13	1200	1625	620	2010	5455
16	16	112	140	12.25	14	75.75	800	1250	490	1830	4370
	18		143	12.88		72.13	1200	1625	620	2220	5665
	20		146	14		68	1650	2050	760	2590	7050

DATA

d = Nominal pipe Size, in

W_b = Weight, Boss, Lbs

W_e = Weight, Elbow, Lbs

W_p = Weight, Pipe, Lbs

W_f = Weight, Flange, Lbs

W_T = Weight, Total

$W_T = W_b + W_e + W_p + W_f$

EQUATIONS

$B_{min} = .5\,D + 1.5\,d + L_{p1} + 6"$

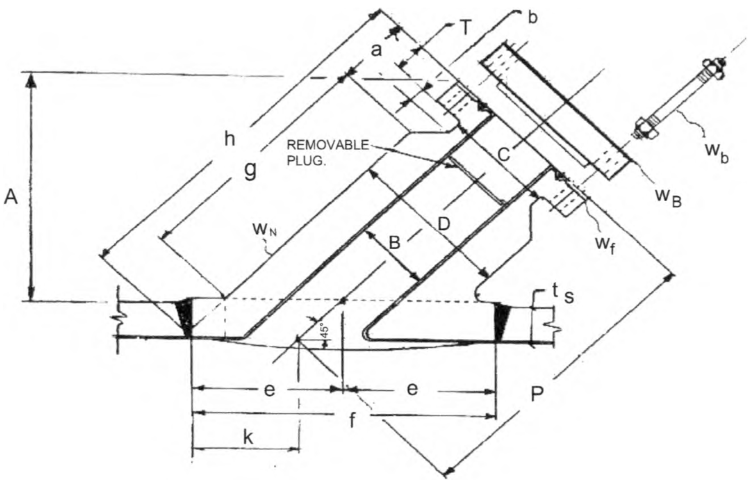
A_{min} = 8' to 11.5' Dia : .5D + 14"
12' to 16' Dia : .5D + 16"

$L_{p2} = A - 1.5\,d - Y$

NOTES

1. Pipe & Elbow are assumed as Sch 160
2. Flange is assumed as 1500#

DIMENSIONS & WEIGHTS OF SIDE CATALYST DUMP NOZZLES



NOTES:

1. Weights & dimensions are for estimating only
2. Dimensions are in inches
3. Weights are in Lbs
4. Shell thickness assumed as follows; 600# = 5", 900# = 6", 1500# = 7", 2500# = 8"
5. There is no weight allowance for the shell portion of the nozzle

FORMULAS:

$A = .707\,P + .354\,D$

$a = T + b + .5\,(D - C)$

$h = g + a + 1.414\,t_s$

$f = 2\,e$

$e = .707\,D + t_s$

SIZE	RATING	A	B	C	D	e	f	g	h	b	a	P	T	WEIGHT
6"	600	22.6	6	9.88	14	14.9	29.8	20.5	32.68	1.25	5.18	25	2.125	850
	900	25.1		10	15	16.6	33.2	22.5	37.13	1.44	6.13	28	2.438	1045
	1500	28.2		10.31	15.5	18	36	25	42.38	1.62	7.48	32	3.5	1300
	2500	32.2		11.38	19	21.4	42.9	27	48.62	2.25	10.31	36	4.5	2320
8"	600	35.5	8	11.94	16.5	16.7	33.4	36	48.84	1.38	5.84	42	2.438	1800
	900	38.4		12.44	18.5	19	38	38	53.66	1.63	7.16	45	2.75	2530
	1500	40.6		12.94	19	20.4	40.8	40	58.43	1.88	8.53	48	3.88	2960
	2500	44.5		14.12	21.75	23.4	46.8	42	64.38	2.25	11.07	52	5.25	4425
10"	600	45.6	10	15	20	19	38	48	61.5	1.5	6.5	55	2.75	3400
	900	48.6		15.5	21.5	21.2	42.4	50	66	1.8	7.56	58	3	4300
	1500	52		16.06	23	23.4	46.8	52	71.5	2.2	9.83	62	4.5	5470
	2500	57.5		17.38	26.5	26.7	53.4	54	79	2.75	13.81	68	6.75	8030

Table 2-22
Weight of (1) hemispherical head, based on inside diameter, LBS

Specified Min Thk, t_s (in)	Thk w/ thinning allowance, T_a	Diameter, in															
		24	36	48	60	72	84	96	108	120	132	144	156	168	180	192	204
0.380	0.560	151	334	590	918	1318	1790	2334	2950	3638							
0.500	0.690	186	412	725	1126	1616	2194	2860	3614	4456							
0.630	0.810	223	490	861	1337	1917	2600	3388	4281	5277	6377	7582	8890				
0.750	0.940	260	570	1000	1550	2220	3010	3920	4950	6100	7373	8763	10275				
0.880	1.060	297	650	1138	1763	2524	3421	4454	5624	6930	8371	9950	11664	13514			
1.000	1.190	335	730	1279	1978	2830	3835	4991	6300	7760	9374	11140	13056	15126			
1.130	1.500	434	938	1635	2524	3606	4880	6345	8000	9854	11900	14131	16558	19177	21989	24992	28189
1.250	1.630	475	1024	1780	2746	3920	5300	6892	8690	10697	12912	15335	17966	20806	23855	27110	30575
1.380	1.750	516	1110	1927	2970	4236	5726	7441	9380	11543	13931	16543	19380	22440	25725	29233	32967
1.500	1.880	559	1197	2075	3194	4558	6153	7992	10073	12393	14954	17755	20796	24078	27600	31362	35365
1.630	2.000	602	1285	2225	3421	4874	6582	8547	10769	13247	15980	18971	22218	25721	29480	33500	37770
1.750	2.130	645	1375	2375	3650	5195	7014	9105	11468	14104	17011	20191	23644	27370	31365	35635	40177
1.880	2.250	690	1465	2528	3880	5520	7450	9665	12170	14964	18045	21415	25075	29021	33256	37780	42592
2.000	2.380	735	1556	2682	4112	5846	7885	10228	12875	15827	19083	22645	26510	30679	35152	39930	45013
2.130	2.750			3150	4818	6838	9209	11933	15010	18440	22220	26354	30840	35679	40870	46413	52310
2.250	2.880			3311	5057	7172	9655	12508	15728	19316	23273	27600	32293	37355	42786	48586	53242
2.380	3.000				5300	7510	10105	13084	16450	20650	24330	28850	33750	39914	44708	50763	57203
2.500	3.130				5540	7850	10555	13664	17173	21082	25391	30100	35210	40722	46634	52946	59660
2.630	3.250					8190	11010	14246	17900	21969	26455	31358	36678	42414	48566	55135	62120
2.750	3.380					8532	11465	14831	18630	22860	27524	32620	38148	44109	50500	57330	64587
2.880	3.500						11925	15240	19365	23755	28600	33885	39625	45810	52445	59530	67060
3.000	3.750						12850	16605	20840	25555	30750	36430	42590	49225	56345	63945	72025
3.250	4.000							17800	22330	27370	32925	38990	45570	52660	60265	68380	77010
3.500	4.250							19000	23830	29200	35110	41570	48570	56115	64200	72840	82020
3.750	4.500							20225	25345	31040	37310	44160	51585	59590	68165	77320	87050
4.000	5.000							22700	28400	34765	41760	49400	57675	66600	76150	86350	97190
4.250	5.250								29960	36650	44000	52040	60750	70125	80175	90900	102300
4.500	6.000								34700	42400	50850	60075	70000	80840	92370	104675	117750
5.000	6.500									46286	55485	65516	76380	88077	100607	113971	128167
5.250	7.000									50242	60185	71025	82762	95400	108928	123357	138683
5.500	7.250									52241	62558	73805	85982	99086	113121	128084	143976
5.750	7.750										67355	79418	92475	106525	121570	137605	154635
6.000	8.000										69775	82250	95750	110275	125825	142400	160000

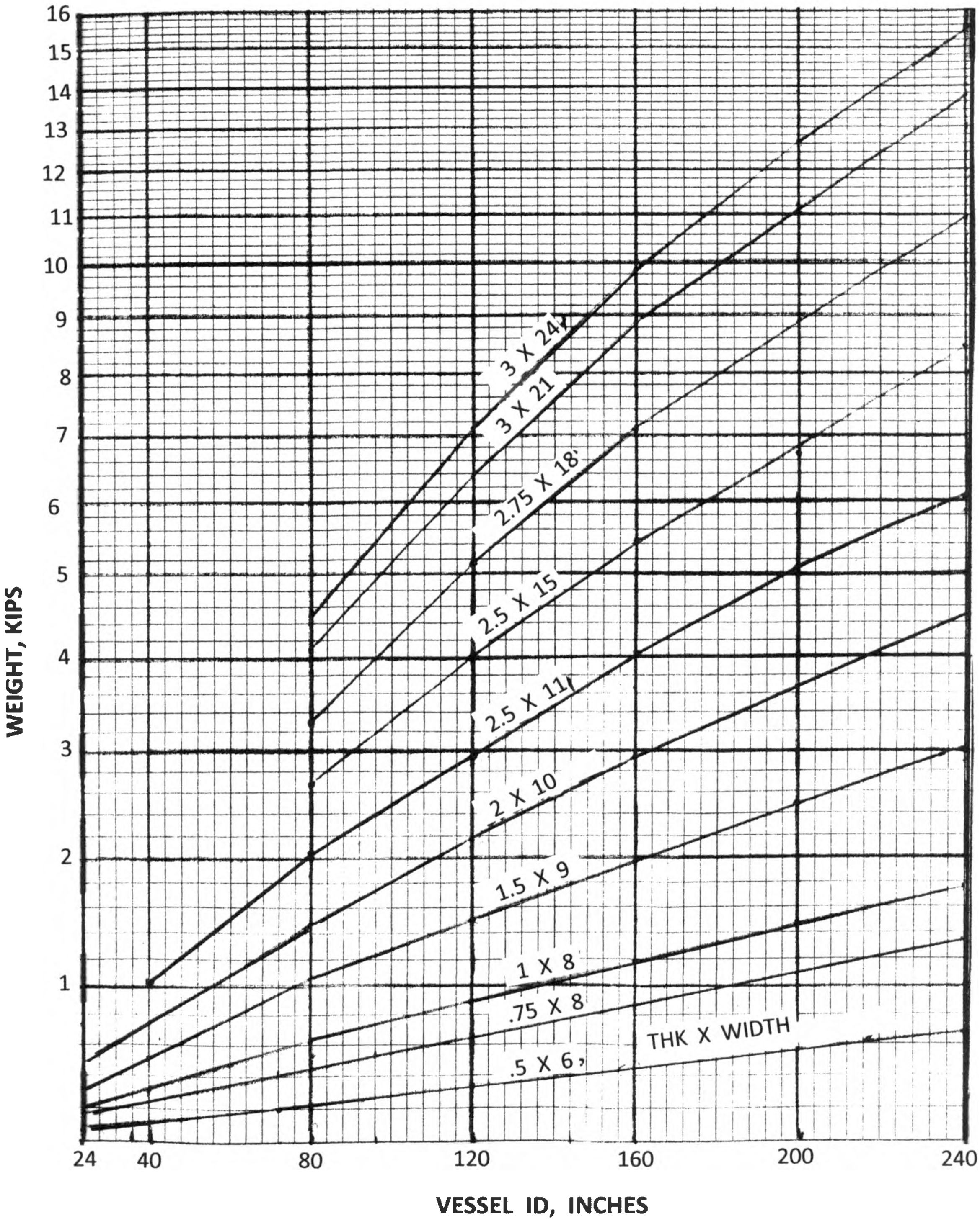
Notes:

1. Assumes carbon steel material (unit wt = .2833 PCI)
2. Thinning allowance, T_a, per Table
3. Formulas are as follows: $W = .445\, D_m^2\, t$; $D_m = D + t$ and $t = t_s + T_a$

Table 2-23
Thinning allowance for hemi-heads

Thickness (in)	Thinning Allowance, T_a
0.125 to 1	0.19
1 to 2	0.38
2 to 3	0.63
3 to 3.75	0.75
4 to 4.5	1
4.5 to 5	1.5
5 to 5.5	1.75
5.75 to 6	2

Weights of base plates



Ladder and Platform (L&P) Estimating

The following is a listing of average breakdowns, both cost and weight, for ladders and platforms (L&Ps) for refinery-type projects. Note that L&Ps include pipe supports, guides, and davits as well as ladders and platforms. Because this data is "average," it is meant to be averaged over an entire project and not to find the cost or weight of any individual item or vessel.

1. **Estimated Price Breakdown**:

• Platforms		30 PSF @ \$2.50/lb	= \$75/sq ft
• Ladders:	Caged	24 lb/ft @ \$3.00/lb	= \$72/ft
	Uncaged	10 lb/ft @ \$2.35/lb	= \$23/ft
• Misc.		\$2.50/lb	
• Handrail:	Straight	\$32/ft	
	Circular	\$42/ft	

2. **Estimated Weight Breakdown** (as a breakdown of the total quantity):

Item		Percentage (%)	Cost (\$/lb)
Platforms:	Circular	30–35%	\$2.50
	Rectangular	50–55%	\$2.00
Ladders:	Caged	7–9%	\$3.00
	Uncaged	2–3%	\$2.25
Misc.		5–10%	\$2.50
Total		100%	

3. **Average Cost of L&Ps** (assuming 100 tons):

Item	Weight (tons)	Cost (\$1000)	% (cost)
Platf Circ	31	155	34
Platf Rect	51	204	45
Ladder Caged	8	48	11
Ladder (uncaged)	2.5	11.25	2
Misc.	7.5	37.5	8
	100 T	\$455.75	100%

Average \$/lb $= 455.75/100 \times 2 =$ \$2.28/lb

4. **Average % Detailed Weight Breakdown for Trayed Columns:**

Item	Large	Medium
Ladders	13.1%	9.3%
Framing	33.3	44.2
Grating	25.3	23.5
Handrailing	18.2	9.7
Pipe supports	3.0	1.6
Bolting	2.5	2.5
Davits	4.1	7.4
Misc.	0.5	1.8
	100%	100%

5. If no estimate of L&Ps is available, an ROM weight estimate can be determined by taking 5% of the overall vessel weights for the project as a total L&P weight. A percentage breakdown may be made of this overall value as noted.

Notes:

1. Miscellaneous weights:

a. Concrete	144 PCF
b. Water	62.4 PCF
c. Gunnite	125 PCF
d. Refractory	65–135 PCF
e. Calcium silicate insulation	13.8 PCF

2. Estimate weight of liquid holdup in random packed columns as 13% of volume.

3. Weights of demister pads and support grids is as follows:

Type	Density (PCF)
931	5
326	7.2
431	9
421	10.8 (multipiece)
	12 (single piece)
Grid	3 PSF

4. Estimate weights of platforming as follows:

Type	Weight
Circular platform	30 PSF
Rectangular platform	20 PSF
Ladder with cage	24 PLF
Ladder without cage	10 PLF

5. Weight of anchor chairs per anchor bolt (wt each, lb):

Anchor Bolt Dia (in.)	Weight (lb)
1	11
1.25	12
1.50	15
1.75	20
2.0	38
2.25	48
2.5	63

Procedure 2-18: Design of Jacketed Vessels

External jackets are used to heat or cool the contents of a vessel. In effect, this turns the pressure vessel into a heat exchanger. Jacketing is an optimum means to accomplish this in terms of control, efficiency and product quality. The advantages are as follows;

1. All liquids can be used. Steam is ideal for heating. Water or Glycol are ideal for cooling.
2. Circulation, temperature and velocity of the heating/cooling media can be carefully controlled.
3. Jackets may be fabricated from a much less expensive metal than the shell.
4. Cleaning and maintenance can be minimized providing a "clean" media is utilized.

Jackets are frequently used in combination with internal heating coils or agitators. Jacketed vessels are much more common in the food, beverage and chemical industry then in the refining industry. The applications are limited in the refining industry because the services in refining application tend to be fouling and not clean. Most refinery applications for heat exchange are dirty services and therefore must be capable of being taken apart for cleaning. Since a jacketed vessel can only be cleaned with steam or chemicals, it is not considered practical for most refinery services.

The types of jackets used on vessels are as follows;

1. Conventional (AKA "Plain")
2. Jacket with a spiral baffle
3. Spiral pipe coil welded to the shell
4. Spiral half pipe coil welded to the shell
5. Dimpled jacket

Conventional Jacket

Used with steam or cooling. Liquid flow velocities are low and the flow is poorly distributed. Natural convection equations are suitable and cooling coefficients have low values. Conventional jackets are best applied to small vessels or high pressure applications, where the vessel internal pressure is twice the jacket pressure as a minimum. The conventional jacket is the most common type. An often used variation of this configuration is made by dividing the straight side into two or more separate jackets.

Jacket with a Spiral baffle

This design is a variation of the conventional jacket. The internal, spiral baffle allows for high flow velocities to be reached. Some clearance must be allowed for between the baffles and the inside of the jacket, to allow for assembly, fabrication and tolerances. Although this clearance may be small, the total leakage area per baffle turn may be substantial when compared with the cross sectional flow area of the baffle passage. Thus, the actual velocity may only be a fraction of the calculated value. To compensate for this leakage around the baffles it is recommended that a 10% allowance be applied to either the total area required or total heat required.

Spiral Pipe Coil Welded to the Shell

This design is not used frequently due to the minimal contact area between the shell and coil. It does however alleviate one of the problems of constructing a half-pipe coil. That is the cutting of the coils in half and the subsequent wastage. Some techniques have been developed to form flat strips into half pipe coils to save wastage and labor. There is just as much welding required for attaching a full pipe coil as there is with the half pipe.

Spiral Half Pipe Coil welded to the shell

This design provides high velocity and turbulence within the jacket. This in turn will result in an unusually high film coefficient. The half pipe coil is recommended for high temperature and all liquid applications. It is better than conventional jackets because the pressure drop can be carefully controlled and calculated. It is not however practical for small vessels, less than 500 gallons. Because there are no limitations to the number of inlet and outlet connections, this type of jacketing can be divided into multiple zones for maximum flexibility and efficiency.

For maximum heat transfer, the coils should be spaced about ¾" (19mm) apart, but other spacing is aceptable. Standard sizes are 2", 3" or 4" NPS.

Dimpled Jackets

There are two main types of dimpled jackets. These are;

1. Integral construction
2. Bolted-on or clamp-on type.

The bolted-on types are less expensive but do not have completely reliable heat transfer characteristics due to the "fit" of the clamp-on sections. Sometimes a heat transfer mastic is used between the two surfaces. Advantages are as follows;

1. They are cheap.
2. They are completely replaceable.
3. The service can be easily altered.
4. The shape and configuration can be easily modified.
5. They do not exert any external pressure on the vessel shell.

The integral type utilizes the inner surface of the dimple jacket for the vessel shell. Advantages of the integral type are as follows;

1. They are cheaper than a conventional jacket.
2. The savings increase with higher pressures and larger vessels.
3. They are cheaper than the half pipe jacket for low internal pressures.

General

External Pressure

The inner vessel shall be designed to resist the maximum differential pressure. Typically this would be the design pressure of the jacket. The differential pressure should include accidental vacuum in the inner vessel. Particular attention should be given to the effects of local loads and differential expansion.

The spiral baffles may not be considered as contributing stiffness to the shell for the case of external pressure since they are not closed circumferential rings. If stiffening rings are required, they must be placed inside the vessel. This may be a problem from a process standpoint. The alternatives are to increase the shell thickness, or shorten the jacket into smaller sections to reduce the L dimension.

Per ASME Code, if the internal pressure is greater than 1.67 times the external pressure, the external pressure will not govern.

The half pipe coil causes an external pressure on the vessel shell but only inside the coil area, and then not in a complete 360° circumferential zone. Therefore the effects on the vessel shell are negligible.

Design

There are two distinct aspects of the design of jackets for heat transfer. These are the thermal design and the physical design. The thermal design falls into three parts:

1. Determine the proper design basis:
   a. Vessel proportions
   b. Maximum depth of liquid
   c. Time required to heat/cool
   d. Agitated or non-agitated
   e. Type of operation; batch or continuous
2. Calculating the required heat load
3. Computing the required surface area

Physical design includes the following:

1. Selecting the type of jacket
2. Determine the areas of the vessel shell to be heated or cooled
3. Determine whether external connections are:
   a. Separate
   b. Series
   c. Parallel
4. Determine pressure drop

The type of operation is characterized in the following cases:

1. Batch operation: Heating
2. Batch operation: Cooling
3. Continuous operation: Heating
4. Continuous operation: Cooling

Pressure Drop

It is important that pressure drop be considered in the design of an external jacket. This will establish the practical limits on the length of passageway inside the jacket.

Pressure drop in conventional jackets without spiral baffles and dimple jackets are deliberately excluded from this procedure. Other sources should be consulted for these applications.

For design purposes, the half pipe coil and conventional jackets with spiral baffles are treated as coils. This procedure converts the shape of the passageway into an equivalent diameter pipe coil.

Large pressure drops may mean the passageway is not capable of transmitting the required quantity of liquid at the available pressure. In addition the fluid velocities inside the passageway should be kept as high as possible to reduce film buildup.

There are no set rules or parameters for maximum allowable pressure drop. Rather, an acceptable pressure drop is related to the velocity required to effect the heat transfer. For liquids a minimum velocity of 1 to 3 feet per second should be considered as minimum. For gases "rho V squared" should be maintained around 4000.

Pressure drop in spiral passageways is dependent on whether the flow is laminar or turbulent. Typically flows are laminar at low fluid velocities and turbulent at high fluid velocities. In spiral passageways a secondary circulation takes place called the "Double Eddy" or "Dean Effect". While this circulation increases the friction loss, it also tends to stabilize laminar flow, thus increasing the "critical" Reynolds number.

In general flows are laminar at Reynolds numbers less than 2000 and turbulent when greater than 4000. At Reynolds numbers between 2000 and 4000 intermittent conditions exist that are called the "critical zone".

For steam flow the pressure drop will be high near the inlet and decrease approximately as the square of the velocity. From this relationship, combined with the effects of increased specific volume of the steam due to pressure drop, it can be shown that the average velocity of the steam in the coil is ¾ of the maximum inlet velocity. For the purposes of calculating pressure drop this ratio may be used to determine the average quantity of steam flowing within the spiral passageway.

In cases where a high velocity is required in order to utilize a high heat transfer coefficient, the pressure drop becomes important. If the pressure drop exceeds the pressure output of a positive displacement pump, then there is a real possibility that the pressure of the jacket will exceed the design conditions.

In the case of a centrifugal pump, the velocity of the fluid will adjust to the pump pressure available. This may result in a lower heat transfer coefficient then required.

For either a dimple jacket design or a half pipe jacket, the pressure drop will be higher than that of an equivalent conventional jacket, due to the increased turbulence.

The preferred method to reduce pressure drop is to shorten the path of the fluid and manifold the inlets and outlets by "parallel" routing. The "series" type external piping connections will increase the overall pressure drop. See Figure 2-50.

Heat Transfer Coefficient, U

The Heat Transfer Coefficient, U, is dependent on the following variables;

1. Thermal conductivity of metal, medium and product
2. Thickness of metal in vessel shell
3. Fluid velocity
4. Specific heat
5. Density and viscosity
6. Fouling factor (oxidation, scaling)
7. Temperature differences (driving force)
8. Trapped gasses in liquid flow
9. Type of flow regime (laminar versus turbulent, turbulent being better)

Agitation

In order to effect the best heat transfer possible, it is preferable to have the contents circulated with an agitator or mixer. This is desirable but not mandatory.

In many cases agitators are provided as part of the process for blending or mixing, not purely to enhance heat transfer.

However, there is a "natural circulation" set up by the heating of a product inside a vessel that will occur. This circulation is driven by the warmer product rising in the tank.

Agitation will also help to prevent buildup or fouling of the vessel wall. This provides for the use of a higher heat transfer coefficient.

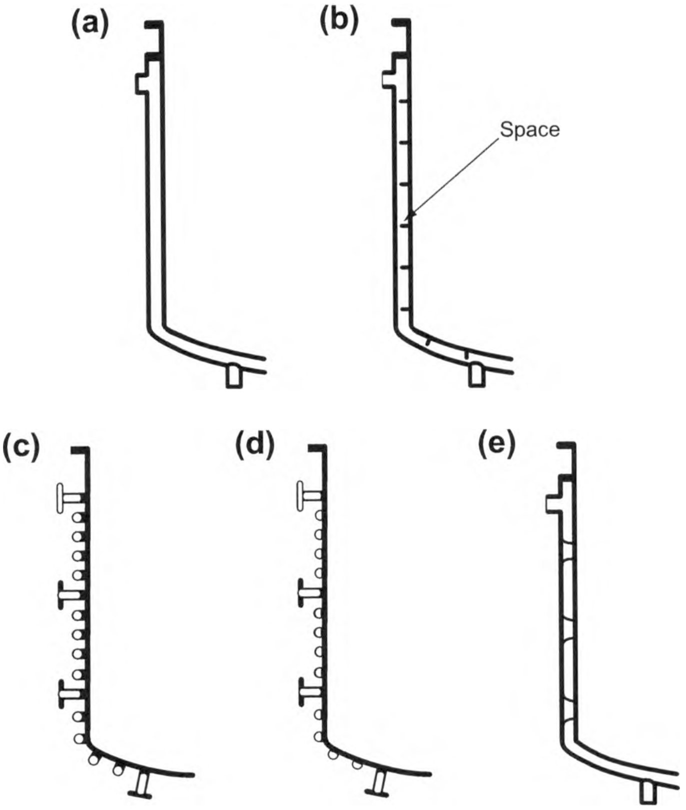


Figure 2-48. Types of Jacket Construction (a) Plain Jacket (b) Jacket with Spiral Baffle (c) Pipe Coil Welded to Shell (d) Half-Pipe Welded to Shell (e) Dimpled Jacket.

Notation

A_C = Cross sectional area of jacket flow area, in^2
A_r = Cross sectional area of jacket flow area, in^2
D = Vessel OD, Ft
D_e, d_e = Equivalent inside diameter of passageway, D_e is in Ft, d_e is inches
f = Friction factor
F_{LF} = Laminar flow factor
g = Acceleration due to gravity=4.17×10^8 Ft/Hr^2
j = Jacket width, in
L = Design length of jacket section, in
L_P = Length of path of travel, Ft
L_{BP} = Baffle pitch, in
M = Mass flow rate, Lbs/Hr
N = Number of turns of spiral baffle or half pipe
P = Internal design pressure in jacket, PSI
p = Pitch of half pipe coil, in
Q = Total heat required, BTU/Hr
Q_L = Heat loss from the exterior of the vessel shell and jacket, BTU/Hr
r = Radius of toroidal section of closure, in
R_e = Reynolds number
R_{ec} = Critical Reynolds number
R_i = Inside radius of vessel, in
R_j = Inside radius of jacket, in
R_S = Outside radius of vessel, in
S = ASME Code allowable stress, tension, PSI
t_C = Thickness of closure bar, in
t_j = Thickness of jacket, in
t_{rJ} = Thickness required, jacket, in
t_{rC} = Thickness required, closure bar, in
t_S = Thickness of vessel shell, in
U = Heat transfer coefficient, $BTU/Hr/Ft^2/°F$
V = Velocity of media, FPS or FPH
W = Rate of flow in jacket, Lbs/Hr
Y, Z = Weld sizes, in
ρ = Density of fluid, PCF
ΔP = Pressure drop, PSI
ΔP_L = Straight line pressure drop, PSI
μ = Dynamic viscosity, cP

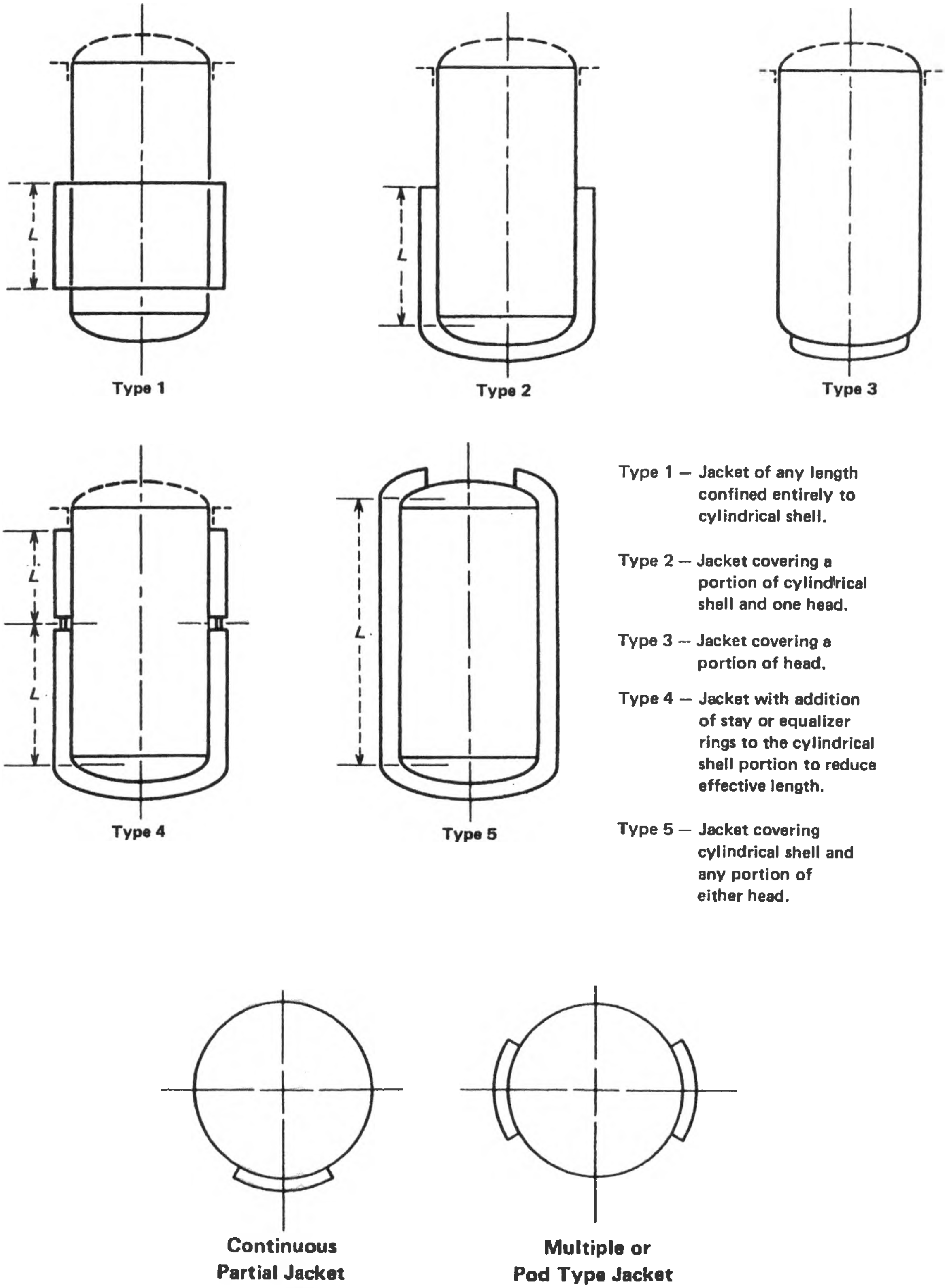
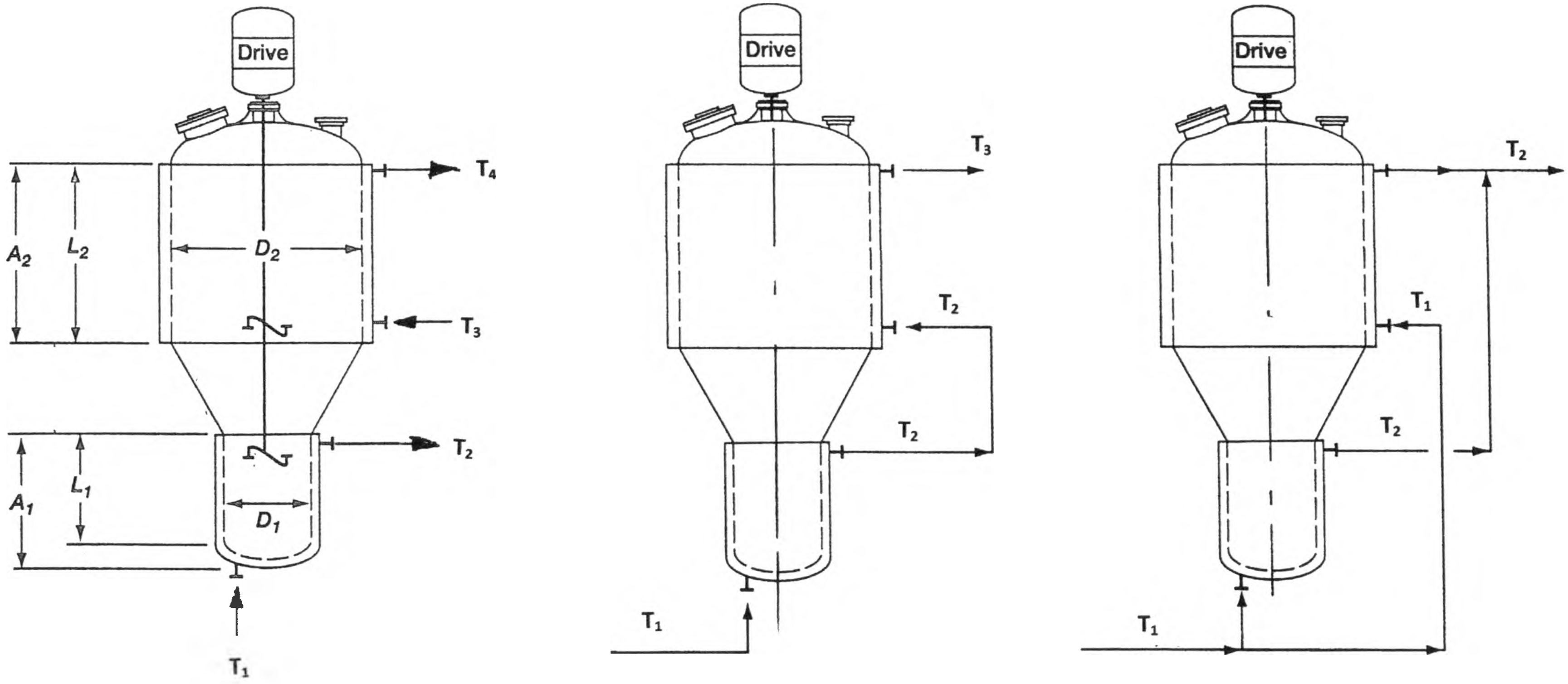


Figure 2-49. Some acceptable types of jacketed vessels.



TYPE 1
JACKETS SEPARATE

TYPE 2
JACKETS IN SERIES

TYPE 3
JACKETS IN PARALLEL

Figure 2-50. External piping configurations for multi-jacketed vessles.

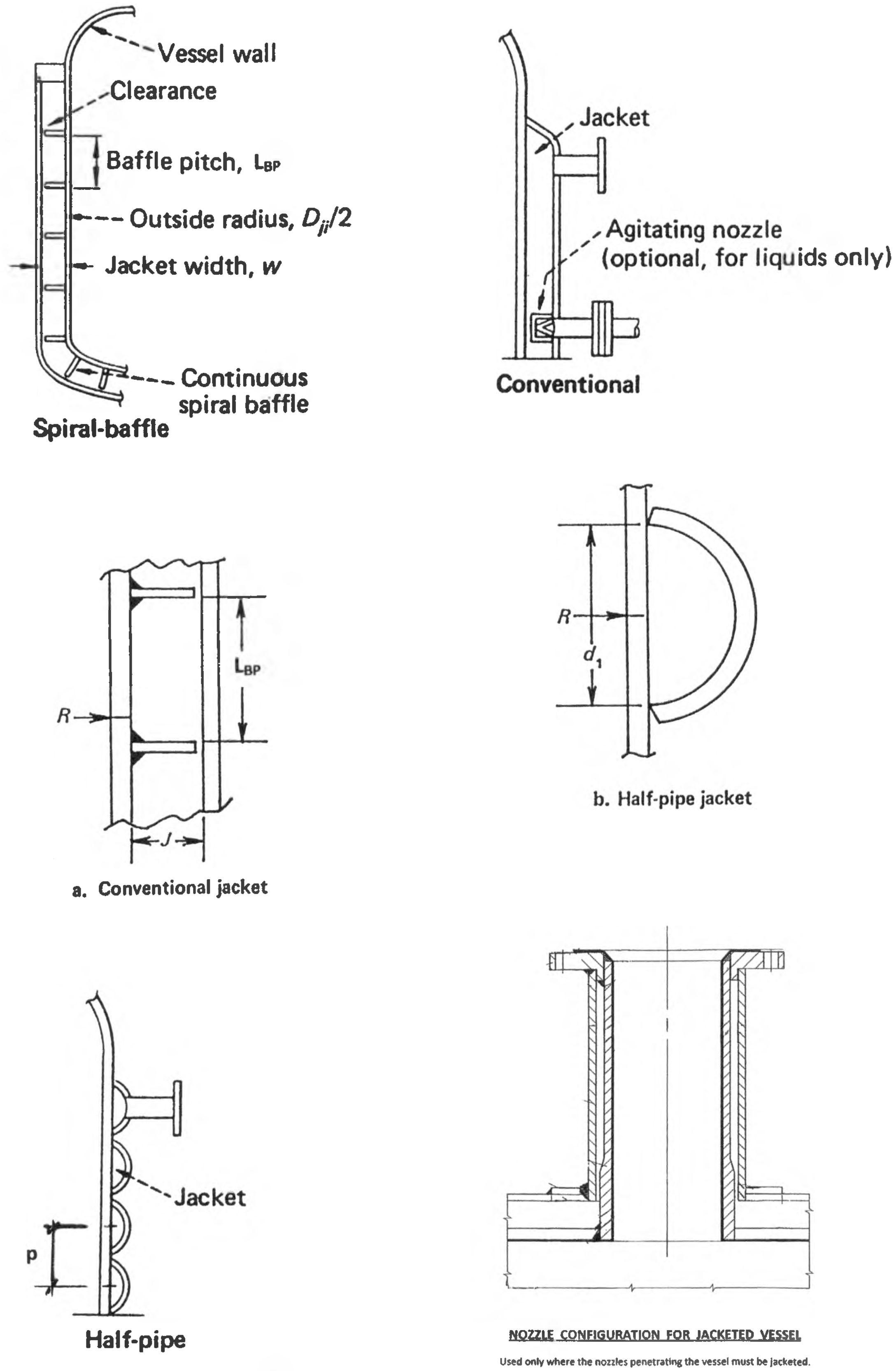


Figure 2-51. Miscellaneous jacket details.

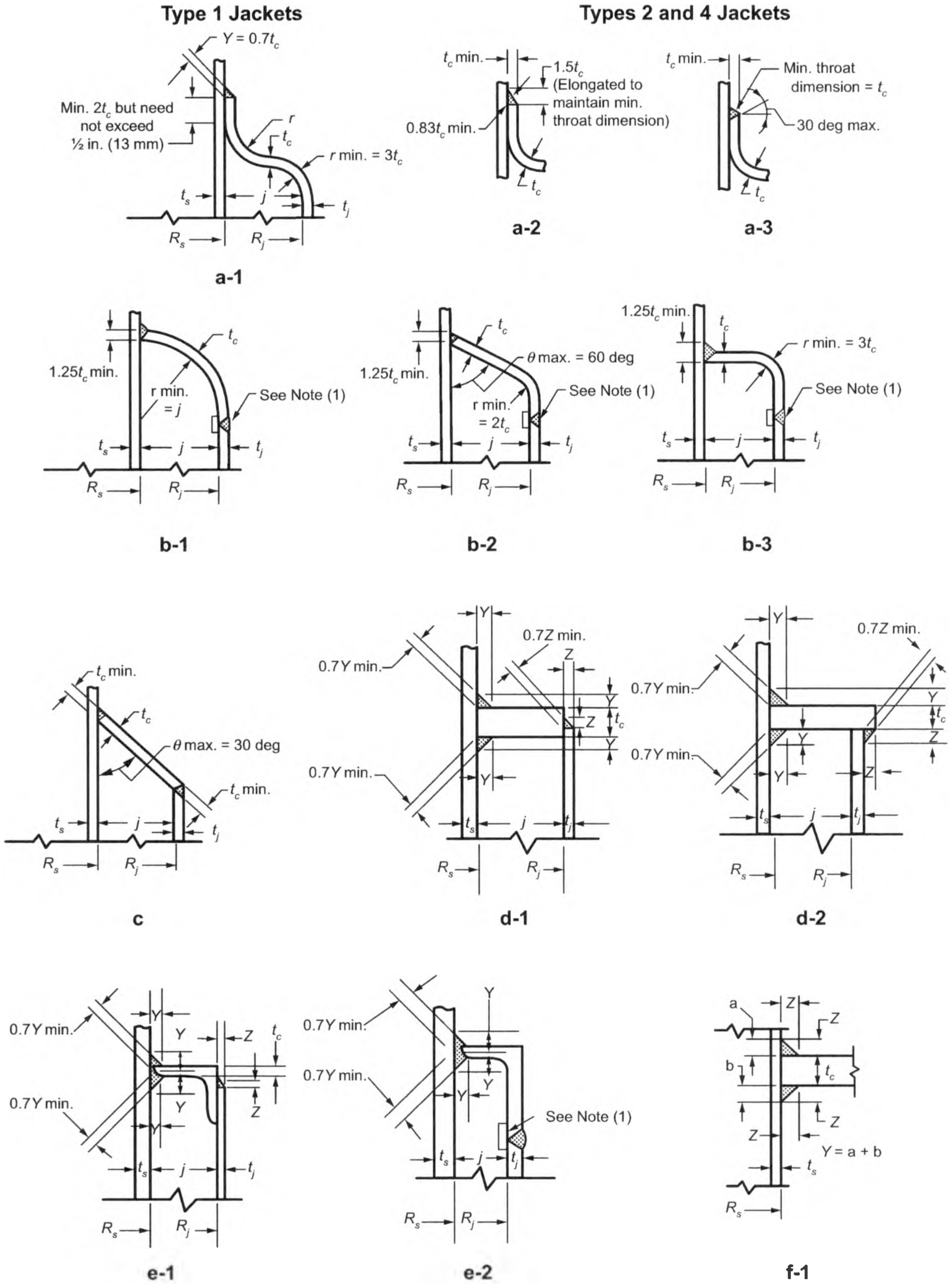


Figure 2-52a. Some acceptable types of jacket closures.

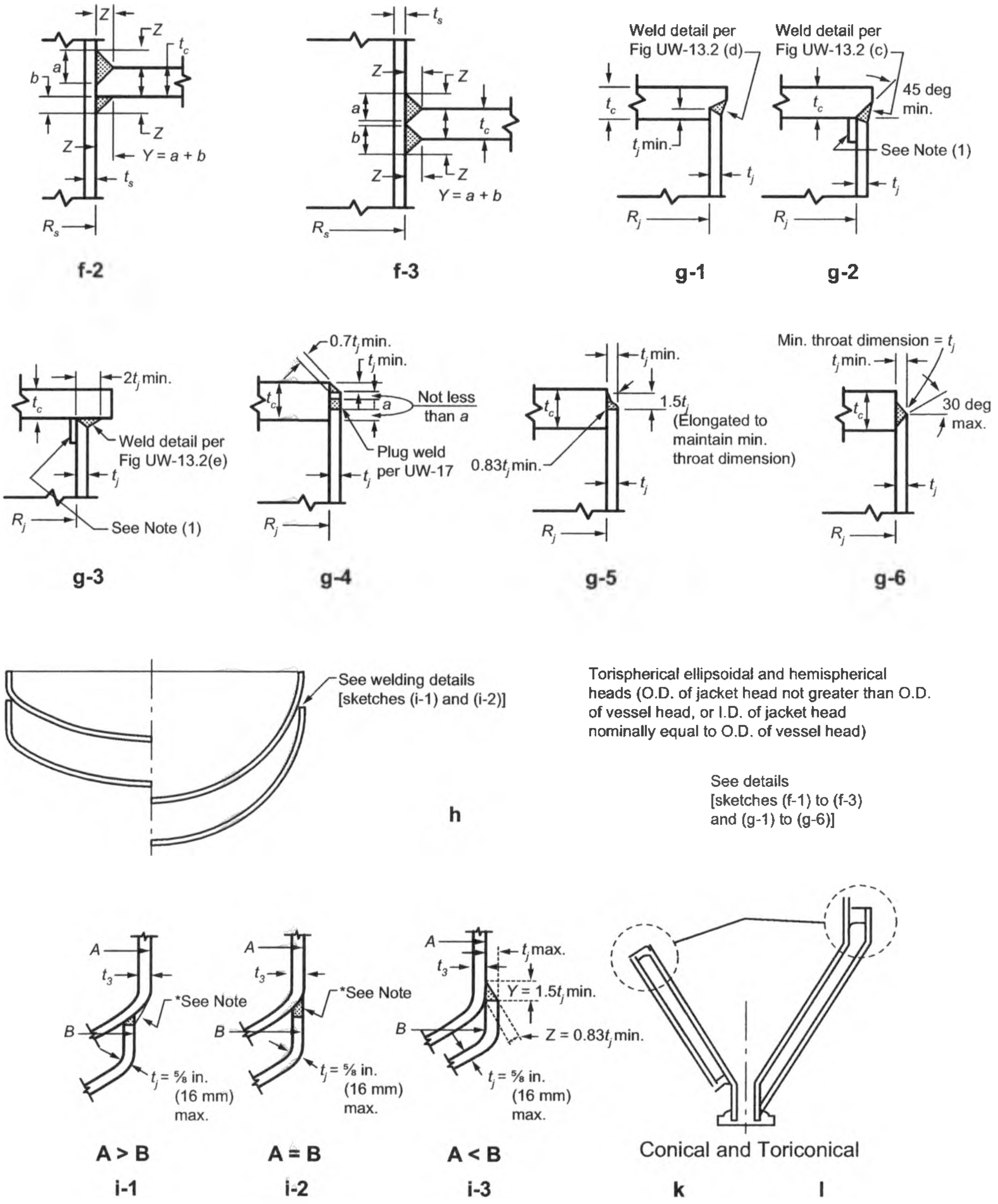


Figure 2-52b. Some acceptable types of jacket closures (Continued)

Table 2-24
Notes To Figure 2-52

Fig	Notes/Remarks
a-1	a. Type 1 jacket only! b. $t_{rc} \geq t_{rj}$ c. $r \geq 3\ t_c$ d. $t_{rc} = .625"$ (16 mm) max e. $Y \geq .7\ t_c$
a-2 & a-3	a. Type 2 or 4 jackets only! b. $t_{rc} \geq t_{rj}$ c. $r \geq 3t_c$ d. $t_{rc} = .625"$ (16 mm) max e. $Y \geq .83\ t_c$
b-1 & b-2	a $t_{rc} \geq t_{rj}$
b-3	a. $t_{rc} \geq t_{rj}$ b. $t_{rc} \geq .707\ J\ (P / S)^{1/2}$
c	a. Type 1 jacket only! b. $\theta = 30^\circ$ max c. $t_{rc} = (P\ r) / [\cos\theta\ (SE - .6\ P)]$ d. $t_{rc} \geq t_{rj}$
d-1, d-2, e-1, & e-2	a. Type 1 jacket only! b. $t_{rj} = .625"$ (16 mm) max c. Min thk of closure bar shall be the greater of the following; 1. $t_{rc} = 2\ t_{rj}$ 2. $t_{rc} = .707\ J\ (P / S)^{1/2}$ d. $Y \geq$ Smaller of $.75\ t_c$ or $.75\ t_s$ e. $Z \geq tj$
f-1, f-2 & f-3	a. May be used for any type jacket b. For Type 1 jacket, the closure bar thk shall be the greater of the following; 1. $t_{rc} = 2\ t_{rj}$ 2. $t_{rc} = .707\ J\ (P / S)^{1/2}$ c. For all other jacket types; $t_{rc} = 1.414\ [(P\ R_sJ) / S)^{1/2}]$ $J = [(2\ S\ t_s^2)/(P\ Rj)] - .5\ (t_s + tj)$ d. Weld sizes; $Y \geq$ Smaller of $1.5\ t_c$ or $1.5\ t_s$ where Y = Sum of a + b Z = Min fillet weld size used to maintain the min Y dimension
g-1, g-2 & g-3	a. May be used with any type jacket
g-4, g-5 & g-6	a. May be used with any type jacket b. $t_{rc} = .625"$ (16 mm) max
h	a. Type 3 jacket only! b. Attachment welds shall be in accordance with Figures i-1, i-2 or i-3 c. $t_{rj} = .625"$ (16 mm) max
i-1, i-2 & i-3	a. Weld details for jackets attahed to heads
k or l	a. Closures for conical or toriconical closures only b. Type 2 jacket only!

Note 1:
For sketches b-1, b-2, b-3, e-2, g-2 and g-3, a backing strip may be used with a full penetration weld.

Procedure 2-19: Forming Strains/Fiber Elongation

Nomenclature

d = Original ID of pipe or tube, in
d_f = Final OD of pipe or tube, in
d_o = OD of pipe or tube, in
D = Finished ID of pipe or tube, in
D_b = Diameter of blank plate or intermediate product, in
D_f = Final OD of pipe or tube, in
D_o = Original OD of head, in
FE = Fiber elongation, %
L_1 = Developed length of two knuckles of head or cone, in
L_2 = Developed length of crown of head or hemisphere, in
L = Initial length of swaged or flared section of pipe or tube, in
L_f = Final length of swaged or flared section of pipe or tube, in
r_o = Outside radius of pipe or tube, in
R_c = Centerline bend radius of pipe or tube, in
R_f = Final mean radius, in
R_o = Original mean radius of plate for head. Assume ∞ if forming begins with a flat plate
t = Thickness of plate, in
t_A = Wall thickness of pipe or tube before bending or forming, in
t_B = Wall thickness of pipe or tube after forming, in
ε_f = Forming strain or fiber elongation, %

Expected Wall Thinning of Pipe Bends

$$t_B = [R_c/(R_c + .5\, d_o)]\,(t_A)$$

$$R_f = R_c + r_o - .5\, t_B$$

Blank Diameter for Formed Heads (Developed Length)

Dimensions...

$$a = .5(D_m - 2\, r)$$

$$\text{Sin}\,\alpha = a/(L - r)$$

$$\alpha =$$

$$\beta = 90 - \alpha$$

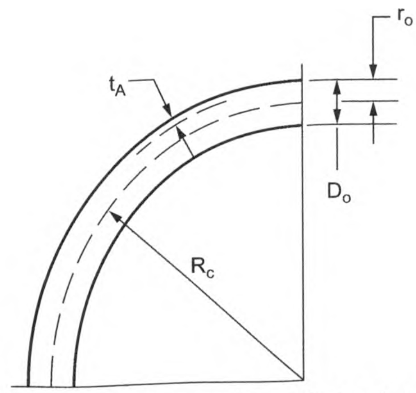


Figure 2-53. Dimensions of pipe bend.

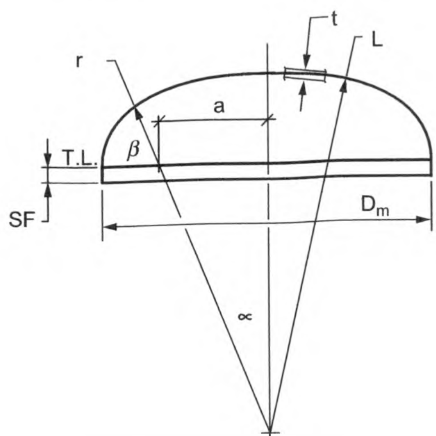


Figure 2-54. Dimensions of head.

- Length of knuckles, L_1

  $L_1 = 2r\beta\,(\pi/180)$
- Length of crown, L_2

  $L_2 = 2L\alpha\,(\pi/180)$
- Diameter of blank necessary, D_b (developed length)

  $D_b = L_1 + L_2 + 2\,SF + 1$ in

  Assumes 1 in for machining.

Example # 1: Formed Head

Given: 100 in ID, 2:1 S.E. Head, 1 in thick

$D_m = 101$ in

$L = .9\, D_m = 90.9$ in

$r = .1727\, D_m = 17.44$ in

$a = 0.5(D_m - 2r) = .5(101 - 34.88) = 33.06$ in

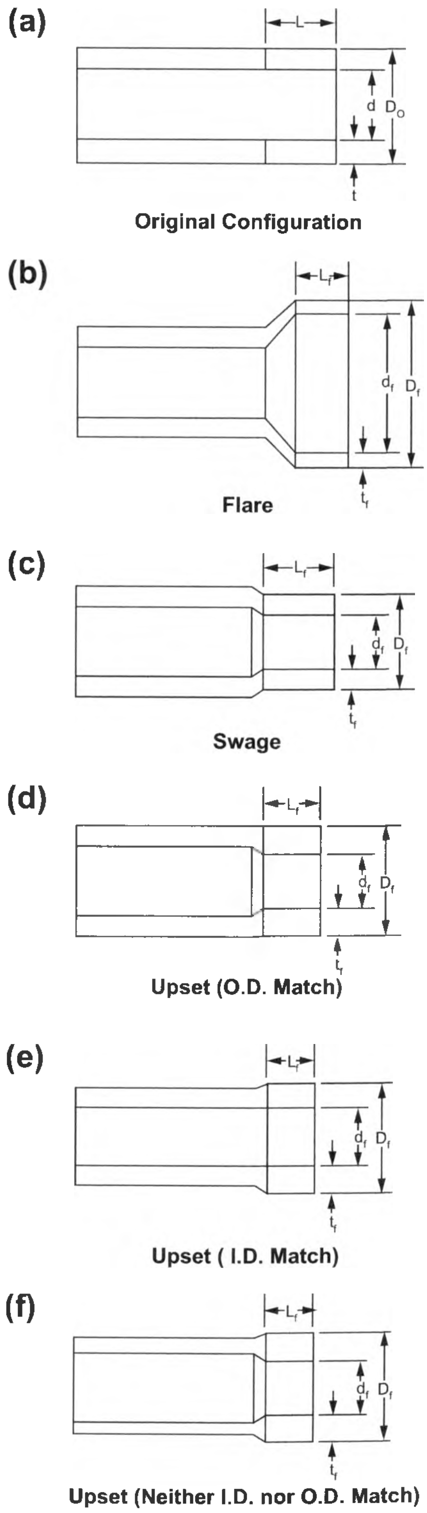


Figure 2-55. Cold forming operations for flaring, swaping, and upsetting of tubing.

$\sin\alpha = a/(L\text{-}r) = 33.06/(90.9 - 17.44) = .452$

$\alpha = 26.75^\circ$

$\beta = 90 - \alpha = 63.25$

- Length of knuckles, L_1

  $L_1 = 2r\beta(\pi/180)$

  $2(17.44)(63.25)(\pi/180) = 38.50$ in

- Length of crown, L_2

  $L_2 = 2L\alpha(\pi/180) =$

  $2(90.9)(26.75)(\pi/180) = 84.88$ in

- Diameter of blank necessary, D_b (Developed length)

  $D_b = L_1 + L_2 + 2\ SF + 1$ in

  $= 38.50 + 84.88 + 3 + 1 = 127.38$

- Per ASME Section VIII-1

  $FE = 75\ t/R_f(1 - R_f/R_o)$

  $= (75)(1)/(17.44) = 4.30\%$

  Since $R_o = \infty$, i.e. starting with flat plate, the term $(1 - R_f / R_o)$ drops out.
  Since $FE \leq 5\%$, no heat treatment required!

- Per ASME Section VIII-2

  $\varepsilon_f = 100\ L_n(D_b/D_i)$

  $= 100\ L_n(127/100) = 23.9\%$

  Since $FE > 5\%$, heat treatment is required

Example #2: Pipe Bend

Pipe Size: 4" Sch 40

$d_o = 4.5$ in

$t_A = 0.154$ in

$R_c = 6$ in

- Wall thinning after forming, t_B

  $t_B = [R_c/(R_c + .5\ d_o)](t_A)$

  $= [6/(6 + .5(4.5)](.154) = 0.112$ in

- Forming strain, ε_f (per ASME VIII-2)

  $\varepsilon_f = \max\left[(r/R_f), ((t_A\text{-}t_B)/t_A)\right].\ 100$

  Where

$$R_f = R_c + r_o - .5\,t_B$$
$$= 6 + 2.25 - .056 = 8.194\text{ in}$$
$$\varepsilon_f = \max\left[(2.25/8.194), ((0.154 - 0.112)/0.154)\right]$$
$$\cdot 100 = 27.5\%$$

Notes

1. This procedure is for carbon steel and low alloy material only.
2. As far as ASME Code is concerned, there is no difference between fiber elongation, FE, and forming strain, ε_f. In ASME VIII-1, it is called fiber elongation. In ASME VIII-2, it is called forming strain. In reality, the only way to determine actual fiber elongation is to remove a sample section of material, macroetch the sample, and view under a microscope.
3. For formed vessel heads, the fiber elongation in the knuckle will always govern. However the FE should be calculated for the crown radius as well.
4. For pipe bends, the inside of the pipe bend gets thicker, while the outside of the bend tends to stretch the material and thinning occurs. The maximum FE is at the far outside of the bend, also known as the extrados.
5. Requirements for heat treatment;

For P No. 1, Groups 1 & 2;

a. All material with a FE > 5% should be heat treated except following...
b. FE can be as high as 40% providing none of the following conditions exist;
   - Vessel is in lethal service
   - Impact testing is not required
   - t > 0.625 in
   - Forming temperature is between 250°F to 900°F

For all other material groups;

6. For all other material, see applicable material sections of ASME Code.
7. When heat treatment is required, it shall be stress relieved.
8. Normalizing is only required when called out by customer or material specifications.
9. If the section is hot formed above the normalizing temperature, and allowed to air cool, no further heat treatment is required.
10. Requirements per ASME Section VIII, Division 1 shall be per UCS-79.
11. Requirements per ASME Section VIII, Division 2 shall be per Table 6.1.

Table 2-25
Formulas for calculating forming strains

Type of Part Being Formed	Forming Strain
Heads and conical sections – formed by spinning or dishing	$\varepsilon_f = 100\ln\left(\dfrac{D_h}{D_o - 2t}\right)$
Shell – cylinders formed from plate	$\varepsilon_f = \dfrac{50t}{R_f}\left(1 - \dfrac{R_f}{R_o}\right)$
Heads and conical sections – formed by pressing	$\varepsilon_f = \dfrac{75t}{R_f}\left(1 - \dfrac{R_f}{R_o}\right)$
Tubing and pipe-bends	$\varepsilon_f = \max\left[\left(\dfrac{r_o}{R_f}\right), \left(\dfrac{t_A - t_B}{t_A}\right)\right] \cdot 100$
Tubing and pipe-flares, swages and upsets	*outside diameter hoop strain* $\varepsilon_f = \left(\dfrac{D - D_f}{D}\right) \cdot 100$ *inside diameter hoop strain* $\varepsilon_f = \left(\dfrac{d - d_f}{d}\right) \cdot 100$ *axial strain* $\varepsilon_f = \left(\dfrac{L - L_f}{L}\right) \cdot 100$ *radial strain* $\varepsilon_f = \left(\dfrac{t - t_f}{t}\right) \cdot 100$

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