Mandatory Appendices

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MANDATORY APPENDICES

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APPENDIX 1
SUPPLEMENTARY DESIGN FORMULAS

1-1 THICKNESS OF CYLINDRICAL AND SPHERICAL SHELLS

(a) The following formulas, in terms of the outside radius, are equivalent to and may be used instead of those given in UG-27(c) and (d).

(1) For cylindrical shells (circumferential stress),

$$t = \frac{PR_o}{SE + 0.4P} \quad \text{or} \quad P = \frac{SEt}{R_o - 0.4t} \tag{1}$$

where

R_o = outside radius of the shell course under consideration, in. (mm)

(2) For spherical shells,

$$t = \frac{PR_o}{2SE + 0.8P} \quad \text{or} \quad P = \frac{2SEt}{R_o - 0.8t} \tag{2}$$

Other symbols are as defined in UG-27.

1-2 THICK CYLINDRICAL SHELLS

(a)(1) Circumferential Stress (Longitudinal Joints). When the thickness of the cylindrical shell under internal design pressure exceeds one-half of the inside radius, or when P exceeds $0.385SE$, the following formulas shall apply:

When P is known and t is desired,

$$t = R(Z^{1/2} - 1) = R_o \frac{(Z^{1/2} - 1)}{Z^{1/2}} \tag{1}$$

where

$$Z = \frac{SE + P}{SE - P}$$

Where t is known and P is desired,

$$P = SE\left(\frac{Z - 1}{Z + 1}\right) \tag{2}$$

where

$$Z = \left(\frac{R + t}{R}\right)^2 = \left(\frac{R_o}{R}\right)^2 = \left(\frac{R_o}{R_o - t}\right)^2$$

(2) Longitudinal Stress (Circumferential Joints). When the thickness of the cylindrical shell under internal design pressure exceeds one-half of the inside radius, or when P exceeds $1.25SE$, the following formulas shall apply:

When P is known and t is desired,

$$t = R\,(Z^{1/2} - 1) = R_o\left(\frac{Z^{1/2} - 1}{Z^{1/2}}\right) \tag{3}$$

where

$$Z = \left(\frac{P}{SE} + 1\right)$$

When t is known and P is desired,

$$P = SE(Z - 1) \tag{4}$$

where

$$Z = \left(\frac{R + t}{R}\right)^2 = \left(\frac{R_o}{R}\right)^2 = \left(\frac{R_o}{R_o - t}\right)^2$$

Symbols are as defined in UG-27 and 1-1.

1-3 THICK SPHERICAL SHELLS

When the thickness of the shell of a wholly spherical vessel or of a hemispherical head under internal design pressure exceeds $0.356R$, or when P exceeds $0.665SE$, the following formulas shall apply:

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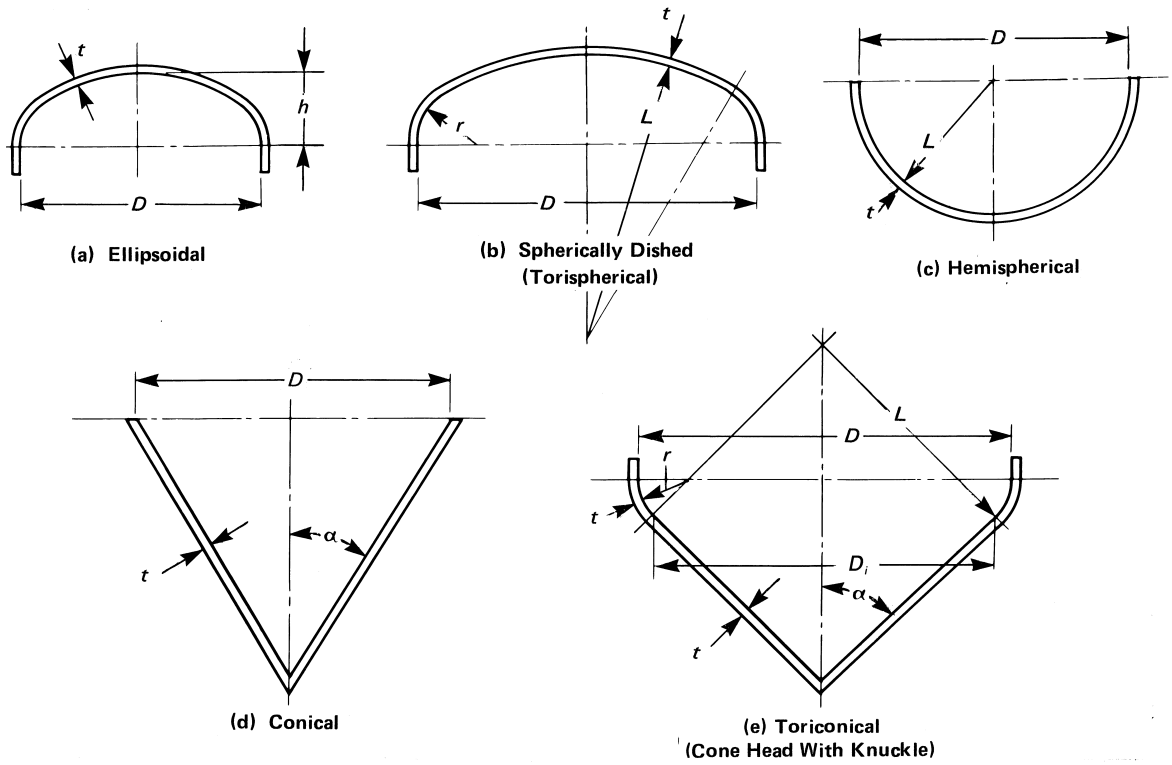


FIG. 1-4 PRINCIPAL DIMENSIONS OF TYPICAL HEADS

When P is known and t is desired,

$$t = R(Y^{1/3} - 1) = R_o\left(\frac{Y^{1/3} - 1}{Y^{1/3}}\right) \qquad (1)$$

where

$$Y = \frac{2(SE + P)}{2SE - P}$$

When t is known and P is desired,

$$P = 2SE\left(\frac{Y - 1}{Y + 2}\right) \qquad (2)$$

where

$$Y = \left(\frac{R + t}{R}\right)^3 = \left(\frac{R_o}{R_o - t}\right)^3$$

Symbols are as defined in UG-27 and 1-1.

1-4 FORMULAS FOR THE DESIGN OF FORMED HEADS UNDER INTERNAL PRESSURE

(a) The formulas of this paragraph provide for the design of formed heads of proportions other than those given in UG-32, in terms of inside and outside diameter.

(b) The symbols defined below are used in the formulas of this paragraph (see Fig. 1-4):

t = minimum required thickness of head after forming, in. (mm)

P = internal design pressure (see UG-21), psi (kPa)

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TABLE 1-4.1
VALUES OF FACTOR K
(Use Nearest Value of $D/2h$; Interpolation Unnecessary)

$D/2h$	3.0	2.9	2.8	2.7	2.6	2.5	2.4	2.3	2.2	2.1	2.0
K	1.83	1.73	1.64	1.55	1.46	1.37	1.29	1.21	1.14	1.07	1.00
$D/2h$	1.9	1.8	1.7	1.6	1.5	1.4	1.3	1.2	1.1	1.0	. . .
K	0.93	0.87	0.81	0.76	0.71	0.66	0.61	0.57	0.53	0.50	. . .

D = inside diameter of the head skirt; or inside length of the major axis of an ellipsoidal head; or inside diameter of a cone head at the point under consideration measured perpendicular to the longitudinal axis, in. (mm)

D_o = outside diameter of the head skirt; or outside length of the major axis of an ellipsoidal head; or outside diameter of a cone head at the point under consideration measured perpendicular to the longitudinal axis, in. (mm)

S = maximum allowable working stress, as given in Subsection C, psi (kPa), except as limited by footnote 1 to 1-4(c) and (d), UG-24, UG-32(e), and UW-12.

E = lowest efficiency of any Category A joint in the head (for hemispherical heads this includes head-to-shell joint). For welded vessels, use the efficiency specified in UW-12.

r = inside knuckle radius, in. (mm)

L = inside spherical or crown radius for torispherical and hemispherical heads, in. (mm)

$L = K_1 D$ for ellipsoidal heads in which K_1 is obtained from Table UG-37, in. (mm)

L_o = outside spherical or crown radius, in. (mm)

L/r = ratio of the inside crown radius to the inside knuckle radius, used in Table 1-4.2

M = a factor in the formulas for torispherical heads depending on the head proportion L/r

h = one-half of the length of the minor axis of the ellipsoidal head, or the inside depth of the ellipsoidal head measured from the tangent line (head-bend line), in. (mm)

K = a factor in the formulas for ellipsoidal heads depending on the head proportion $D/2h$

$D/2h$ = ratio of the major to the minor axis of ellipsoidal heads, which equals the inside diameter of the skirt of the head divided by twice the inside height of the head, and is used in Table 1-4.1

α = one-half of the included (apex) angle of the cone at the center line of the head

(c) Ellipsoidal Heads[1]

$$t = \frac{PDK}{2SE - 0.2P} \text{ or } P = \frac{2SEt}{KD + 0.2t} \quad (1)$$

$$t = \frac{PD_oK}{2SE + 2P(K - 0.1)}$$

or

$$P = \frac{2SEt}{KD_o - 2t(K - 0.1)} \quad (2)$$

where

$$K = \frac{1}{6}\left[2 + \left(\frac{D}{2h}\right)^2\right]$$

Numerical values of the factor K are given in Table 1-4.1.

Example 1.[2] Determine the required thickness t of a seamless ellipsoidal head, exclusive of provision for corrosion for the following conditions:

$D = 40$ in; $h = 9$ in; $P = 200$ psi; $S = 13{,}750$ psi; $E = 1.00$.

---

[1] Ellipsoidal heads designed under $K > 1.0$ and all torispherical heads made of materials having a specified minimum tensile strength exceeding 70,000 psi (482 MPa) shall be designed using a value of S equal to 20,000 psi (137.2 MPa) at room temperature and reduced in proportion to the reduction in maximum allowable stress values at temperature for the material as shown in the appropriate table (see UG-23). 01

[2] This calculation is intended only to illustrate the use of the formula herein. Other paragraphs in this Division may have to be satisfied to permit use of the full tabular stress value.

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$$\frac{D}{2h} = \frac{40}{18} = 2.22$$

From Table 1-4.1, $K = 1.14$. Substituting in Eq. (1),

$$t = \frac{200 \times 40 \times 1.14}{[2 \times 13{,}750 \times (1.00) - (0.2 \times 200)]} = 0.33 \text{ in.}$$

Example 2.[2] Determine the maximum allowable working pressure P of a seamless ellipsoidal head for the following conditions:

$D = 30$ in.; $h = 7.5$ in.; total thickness = $\frac{1}{2}$ in. with no allowance for corrosion; maximum operating temperature = 800°F; $E = 1.00$.

From the appropriate table given in Subpart 1 of Section II, Part D, $S = 10{,}200$ psi.

$$\frac{D}{2h} = \frac{30}{15} = 2.0$$

From Table 1-4.1, $K = 1.0$. Substituting in Eq. (1),

$$P = \frac{2 \times 10{,}200 \times 1.0 \times 0.5}{[1 \times 30 + (0.2 \times 0.5)]} = 339 \text{ psi}$$

(d) Torispherical Heads[1]

$$t = \frac{PLM}{2SE - 0.2P} \quad \text{or} \quad P = \frac{2SEt}{LM + 0.2t} \qquad (3)$$

$$t = \frac{PL_oM}{2SE + P(M - 0.2)}$$

or

$$P = \frac{2SEt}{ML_o - t(M - 0.2)} \qquad (4)$$

where

$$M = \tfrac{1}{4}\left(3 + \sqrt{\frac{L}{r}}\right)$$

Numerical values of the factor M are given in Table 1-4.2.

Example 1.[2]Determine the required thickness t, exclusive of allowance for corrosion, of a torispherical head for the following conditions:

$D = 40$ in.; $L = 40$ in.; $r = 4$ in.; $P = 200$ psi; $S = 13{,}750$ psi; $E = 1.00$ (seamless head).

$$\frac{L}{r} = \frac{40}{4} = 10$$

and from Table 1-4.2, $M = 1.54$. Substituting in Eq. (3),

$$t = \frac{200 \times 40 \times 1.54}{[2 \times 13{,}750 \times (1.00) - (0.2 \times 200)]} = 0.45 \text{ in.}$$

Example 2.[2]Determine the maximum allowable working pressure P of a torispherical head for the following conditions:

$D = 30$ in.; $L = 24$ in.; $r = 2.00$ in.; $E = 1.00$ (seamless head); total thickness = 0.5 in. with no allowance for corrosion; material conforms to SA-515 Grade 70; maximum operating temperature = 900°F.

From the appropriate table given in Subpart 1 of Section II, Part D, $S = 6500$ psi.

$$\frac{L}{r} = \frac{24}{2.00} = 12.0$$

From Table 1-4.2, $M = 1.62$. Substituting in Eq. (3),

$$P = \frac{2 \times 6500 \times 1.0 \times 0.5}{24 \times 1.62 + 0.2 \times 0.5} = 167 \text{ psi}$$

(e) Conical Heads

$$t = \frac{PD}{2 \cos \alpha \, (SE - 0.6P)}$$

or

$$P = \frac{2SEt \cos \alpha}{D + 1.2t \cos \alpha} \qquad (5)$$

$$t = \frac{PD_o}{2 \cos \alpha \, (SE + 0.4P)}$$

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TABLE 1-4.2
VALUES OF FACTOR *M*
(Use Nearest Value of *L/r*; Interpolation Unnecessary)

L/r	1.0	1.25	1.50	1.75	2.00	2.25	2.50	2.75	3.00	3.25	3.50
M	1.00	1.03	1.06	1.08	1.10	1.13	1.15	1.17	1.18	1.20	1.22
L/r	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0
M	1.25	1.28	1.31	1.34	1.36	1.39	1.41	1.44	1.46	1.48	1.50
L/r	9.5	10.00	10.5	11.0	11.5	12.0	13.0	14.0	15.0	16.0	$16\frac{2}{3}$[1]
M	1.52	1.54	1.56	1.58	1.60	1.62	1.65	1.69	1.72	1.75	1.77

NOTE:
(1) Maximum ratio allowed by UG-32(j) when *L* equals the outside diameter of the skirt of the head.

or

$$P = \frac{2SEt \cos \alpha}{D_o - 0.8t \cos \alpha} \quad (6)$$

1-5 RULES FOR CONICAL REDUCER SECTIONS AND CONICAL HEADS UNDER INTERNAL PRESSURE

(a) The formulas of (d) and (e) below provide for the design of reinforcement, if needed, at the cone-to-cylinder junctions for conical reducer sections and conical heads where all the elements have a common axis and the half-apex angle $\alpha \leq 30$ deg. Subparagraph (g) below provides for special analysis in the design of cone-to-cylinder intersections with or without reinforcing rings where α is greater than 30 deg.

In the design of reinforcement for a cone-to-cylinder juncture, the requirements of UG-41 shall be met.

(b) Nomenclature

A_{rL} = required area of reinforcement at large end of cone, in.^{2} (mm^2)
A_{rs} = required area of reinforcement at small end of cone, in.^{2} (mm^2)
A_{eL} = effective area of reinforcement at large end intersection, in.^{2} (mm^2)
A_{es} = effective area of reinforcement at small end intersection, in.^{2} (mm^2)
E_s = modulus of elasticity of cylinder material, psi (kPa)
E_c = modulus of elasticity of cone material, psi (kPa)
E_r = modulus of elasticity of reinforcing ring material, psi (kPa)
E_1 = efficiency of longitudinal joint in cylinder. For compression (such as at large end of cone), $E_1 = 1.0$ for butt welds.
E_2 = efficiency of longitudinal joint in cone. For compression, $E_2 = 1.0$ for butt welds.
f_1 = axial load at large end due to wind, dead load, etc., excluding pressure, lb/in. (kN/m)
f_2 = axial load at small end due to wind, dead load, etc., excluding pressure, lb/in. (kN/m)
P = internal design pressure (see UG-21), psi (kPa)
Q_L = algebraical sum of $PR_L/2$ and f_1, lb/in. (kN/m)
Q_s = algebraical sum of $PR_s/2$ and f_2, lb/in. (kN/m)
R_s = inside radius of small cylinder at small end of cone, in. (mm)
R_L = inside radius of large cylinder at large end of cone, in. (mm)
S_s = allowable stress of cylinder material at design temperature, psi (kPa)
S_c = allowable stress of cone material at design temperature, psi (kPa)
S_r = allowable stress of reinforcing ring material at design temperature, psi (kPa)
t = minimum required thickness of cylinder at cone-to-cylinder junction, in. (mm)
t_c = nominal thickness of cone at cone-to-cylinder junction, in. (mm)
t_r = minimum required thickness of cone at cone-to-cylinder junction, in. (mm)
t_s = nominal thickness of cylinder at cone-to-cylinder junction, in. (mm)
α = half-apex angle of cone or conical section, deg.
Δ = angle indicating need for reinforcement at cone-to-cylinder junction having a half-apex angle $\alpha \leq 30$ deg. When $\Delta \geq \alpha$, no reinforcement is required at the junction (see Tables 1-5.1 and 1-5.2), deg.
y = cone-to-cylinder factor
= S_sE_s for reinforcing ring on shell
= S_cE_c for reinforcing ring on cone

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TABLE 1-5.1
VALUES OF Δ FOR JUNCTIONS AT THE LARGE CYLINDER FOR α ≤ 30 deg.

P/S_sE_1	0.001	0.002	0.003	0.004	0.005
Δ, deg.	11	15	18	21	23
P/S_sE_1	0.006	0.007	0.008	0.009[1]	. . .
Δ, deg.	25	27	28.5	30	. . .

NOTE:
(1) Δ = 30 deg. for greater values of P/S_sE_1.

TABLE 1-5.2
VALUES OF Δ FOR JUNCTIONS AT THE SMALL CYLINDER FOR α ≤ 30 deg.

P/S_sE_1	0.002	0.005	0.010	0.02
Δ, deg.	4	6	9	12.5
P/S_sE_1	0.04	0.08	0.10	0.125[1]
Δ, deg.	17.5	24	27	30

NOTE:
(1) Δ = 30 deg. for greater values of P/S_sE_1.

(c) For a cone-to-cylinder junction, the following values shall be determined at large end and again at the small end in order that both the large end and the small end can be examined:

Determine P/S_sE_1 and then determine Δ at the large end and at the small end, as appropriate, from Tables 1-5.1 and 1-5.2.

Determine k:

$k = 1$ when additional area of reinforcement is not required

$= y/S_r E_r$ when a stiffening ring is required, but k is not less than 1.0

(d) Reinforcement shall be provided at the junction of the cone with the large cylinder for conical heads and reducers without knuckles when the value of Δ obtained from Table 1-5.1, using the appropriate ratio P/S_sE_1, is less than α. Interpolation may be made in the Table.

The required area of reinforcement shall be at least equal to that indicated by the following formula when Q_L is in tension:

$$A_{rL} = \frac{kQ_LR_L}{S_sE_1}\left(1 - \frac{\Delta}{\alpha}\right)\tan\alpha \qquad (1)^{**}$$

At the large end of the cone-to-cylinder juncture, the $PR_L/2$ term is in tension. When f_1 is in compression and the quantity is larger than the $PR_L/2$ term, the design shall be in accordance with U-2(g). The calculated localized stresses at the discontinuity shall not exceed the stress values specified in 1-5(g)(1) and (2).

The effective area of reinforcement can be determined in accordance with the following formula:

$$A_{eL} = (t_s - t)\sqrt{R_Lt_s} + (t_c - t_r)\sqrt{R_Lt_c/\cos\alpha} \qquad (2)$$

Any additional area of reinforcement which is required shall be situated within a distance of $\sqrt{R_Lt_s}$ from the junction of the reducer and the cylinder. The centroid of the added area shall be within a distance of $0.25 \times \sqrt{R_Lt_s}$ from the junction.

(e) Reinforcement shall be provided at the junction of the conical shell of a reducer without a flare and the small cylinder when the value of Δ obtained from Table 1-5.2, using the appropriate ratio P/S_sE_1, is less than α.

The required area of reinforcement shall be at least equal to that indicated by the following formula when Q_s is in tension:

$$A_{rs} = \frac{kQ_sR_s}{S_sE_1}\left(1 - \frac{\Delta}{\alpha}\right)\tan\alpha \qquad (3)^{**}$$

At the small end of the cone-to-cylinder juncture, the $PR_s/2$ term is in tension. When f_2 is in compression and the quantity is larger than the $PR_s/2$ term, the design shall be in accordance with U-2(g). The calculated localized stresses at the discontinuity shall not exceed the stress values specified in 1-5(g)(1) and (2).

The effective area of reinforcement can be determined in accordance with the following formula:

$$A_{es} = 0.78\sqrt{R_st_s}[(t_s - t) + (t_c - t_r)/\cos\alpha] \qquad (4)$$

Any additional area of reinforcement which is required shall be situated within a distance of $\sqrt{R_st_s}$ from the junction, and the centroid of the added area shall be within a distance of $0.25\sqrt{R_st_s}$ from the junction.

(f) Reducers not described in UG-36(e)(5), such as those made up of two or more conical frustums having different slopes, may be designed in accordance with (g).

(g) When the half-apex angle α is greater than 30 deg., cone-to-cylinder junctions without a knuckle may be used, with or without reinforcing rings, if the design is based on special analysis, such as the beam-on-elastic-foundation analysis of Timoshenko, Hetenyi, or Watts and Lang. See U-2(g). When such an analysis

** For some of the terms of the above equation(s), it may be necessary to convert millimeters to meters to obtain a rational result in SI units.

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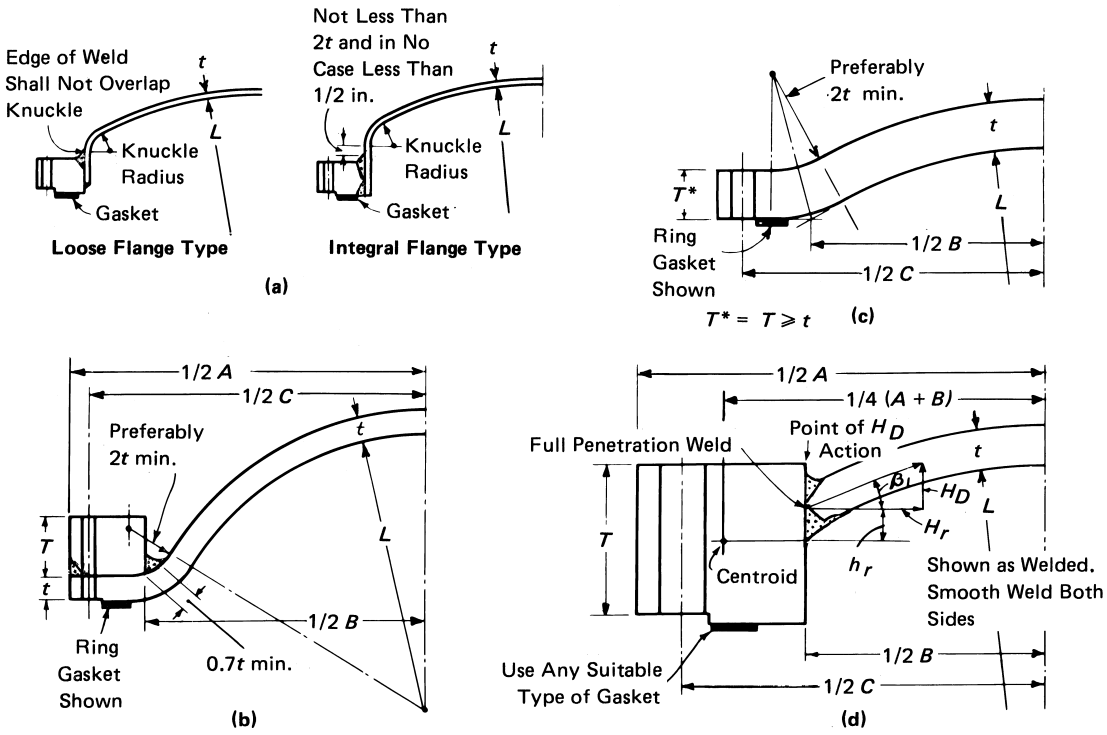


FIG. 1-6 SPHERICALLY DISHED COVERS WITH BOLTING FLANGES

is made, the calculated localized stresses at the discontinuity shall not exceed the following values.

(1) (Membrane hoop stress) + (average discontinuity hoop stress) shall not be greater than $1\frac{1}{2}$ *SE,* where the "average discontinuity hoop stress" is the average hoop stress across the wall thickness due to the discontinuity at the junction, disregarding the effect of Poisson's ratio times the longitudinal stress at the surfaces.

(2) (Membrane longitudinal stress) + (discontinuity longitudinal stress due to bending) shall not be greater than 4*SE*.

The angle joint (see 3-2) between the cone and cylinder shall be designed equivalent to a double butt-welded joint, and because of the high bending stress, there shall be no weak zones around the angle joint. The thickness of the cylinder may have to be increased to limit the difference in thickness so that the angle joint has a smooth contour.

The joint efficiencies *E* shall be in accordance with UW-12.

1-6 SPHERICALLY DISHED COVERS (BOLTED HEADS)

(a) Circular spherical dished heads with bolting flanges, both concave and convex to the pressure and conforming to the several types illustrated in Fig. 1-6, shall be designed in accordance with the formulas which follow.

(b) The symbols used in the formulas of this paragraph are defined as follows:

t = minimum required thickness of head plate after forming, in. (mm)

L = inside spherical or crown radius, in. (mm)

r = inside knuckle radius, in. (mm)

P = internal pressure (see UG-21) for the pressure on concave side, and external pressure for the pressure on convex side [see UG-28(f)], psi (kPa)

S = maximum allowable stress value, psi (kPa) (see UG-23)

T = flange thickness, in. (mm)

M_o = the total moment, in.-lb (kN•m), determined as in 2-6 for heads concave to pressure and 2-11 for heads convex to pressure; except that for heads of the type shown in Fig. 1-6 sketch (d), H_D and h_D shall be as defined below, and an additional moment $H_r h_r$ (which may add or subtract) shall be included where

H_r = radial component of the membrane load in the spherical segment, lb (kN), acting at the intersection of the inside of the

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flange ring with the center line of the dished cover thickness

$= H_D \cot \beta_1$

h_r = lever arm of force H_r about centroid of flange ring, in. (mm)

H_D = axial component of the membrane load in the spherical segment, lb (kN), acting at the inside of the flange ring

$= 0.785\ B^2P$

h_D = radial distance from the bolt circle to the inside of the flange ring, in. (mm)

β_1 = angle formed by the tangent to the center line of the dished cover thickness at its point of intersection with the flange ring, and a line perpendicular to the axis of the dished cover

$$= \text{arc sin}\left(\frac{B}{2L + t}\right)$$

NOTE: Since $H_r h_r$ in some cases will subtract from the total moment, the moment in the flange ring when the internal pressure is zero may be the determining loading for flange design.

A = outside diameter of flange, in. (mm)

B = inside diameter of flange, in. (mm)

C = bolt circle, diameter, in. (mm)

(c) It is important to note that the actual value of the total moment M_o may calculate to be either plus or minus for both the heads concave to pressure and the heads convex to pressure. However, for use in all of the formulas which follow, the absolute values for both P and M_o are used.

(d) Heads of the type shown in Fig. 1-6 sketch (a):

(1) the thickness of the head t shall be determined by the appropriate formula in UG-32 for pressure on concave side, and UG-33(a)(1) for pressure on convex side;

(2) the head radius L or the knuckle radius r shall comply with the limitations given in UG-32;

(3) the flange shall comply at least with the requirements of Fig. 2-4 and shall be designed in accordance with the provisions of 2-1 through 2-7 for pressure on concave side, and 2-11 for pressure on convex side. (Within the range of flange standards listed in Table U-3, the flange and drillings may conform to the standards, and the thickness specified therein shall be considered as a minimum requirement.)

(e) Heads of the type shown in Fig. 1-6 sketch (b) (no joint efficiency factor is required):

(1) head thickness

(a) for pressure on concave side,

$$t = \frac{5PL}{6S} \quad (1)$$

(b) for pressure on convex side, the head thickness shall be determined based on UG-33(c) using the outside radius of the spherical head segment;

(2) flange thickness for ring gasket

$$T = \sqrt{\frac{M_o}{SB}\left[\frac{A + B}{A - B}\right]} \quad (2)^{**}$$

(3) flange thickness for full face gasket

$$T = 0.6\sqrt{\frac{P}{S}\left[\frac{B(A + B)(C - B)}{A - B}\right]} \quad (3)$$

NOTE: The radial components of the membrane load in the spherical segment are assumed to be resisted by its flange.

(Within the range of flange standards listed in Table U-3, the flange and drillings may conform to the standards, and the thickness specified therein shall be considered as a minimum requirement.)

(f) Heads of the type shown in Fig. 1-6 sketch (c) (no joint efficiency factor is required):

(1) head thickness

(a) for pressure on concave side,

$$t = \frac{5PL}{6S} \quad (4)$$

(b) for pressure on convex side, the head thickness shall be determined based on UG-33(c) using the outside radius of the spherical head segment;

(2) flange thickness for ring gasket for heads with round bolting holes

$$T = Q + \sqrt{\frac{1.875M_o(C + B)}{SB(7C - 5B)}} \quad (5)^{**}$$

where

$$Q = \frac{PL}{4S}\left(\frac{C + B}{7C - 5B}\right) \quad **$$

** For some of the terms of the above equation(s), it may be necessary to convert millimeters to meters to obtain a rational result in SI units.

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(3) flange thickness for ring gasket for heads with bolting holes slotted through the edge of the head

$$T = Q + \sqrt{\frac{1.875M_o(C + B)}{SB(3C - B)}} \qquad (6)^{**}$$

where

$$Q = \frac{PL}{4S}\left(\frac{C + B}{3C - B}\right)$$

(4) flange thickness for full-face gasket for heads with round bolting holes

$$T = Q + \sqrt{Q^2 + \frac{3BQ(C - B)}{L}} \qquad (7)$$

where

$$Q = \frac{PL}{4S}\left(\frac{C + B}{7C - 5B}\right)$$

(5) flange thickness for full-face gasket for heads with bolting holes slotted through the edge of the head

$$T = Q + \sqrt{Q^2 + \frac{3BQ(C - B)}{L}} \qquad (8)$$

where

$$Q = \frac{PL}{4S}\left(\frac{C + B}{3C - B}\right)$$

(6) the required flange thickness shall be T as calculated in (2), (3), (4), or (5) above, but in no case less than the value of t calculated in (1) above.

(g) Heads of the type shown in Fig. 1-6 sketch (d) (no joint efficiency factor is required):

(1) head thickness

(a) for pressure on concave side,

$$t = \frac{5PL}{6S} \qquad (9)$$

(b) for pressure on convex side, the head thickness shall be determined based on UG-33(c) using the outside radius of the spherical head segment;

(2) flange thickness

$$T = F + \sqrt{F^2 + J} \qquad (10)$$

where

$$F = \frac{PB\sqrt{4L^2 - B^2}}{8S(A - B)}$$

and

$$J = \left(\frac{M_o}{SB}\right)\left(\frac{A + B}{A - B}\right)$$

(h) These formulas are approximate in that they do not take into account continuity between the flange ring and the dished head. A more exact method of analysis which takes this into account may be used if it meets the requirements of U-2.

1-7 LARGE OPENINGS IN CYLINDRICAL SHELLS

1-7(a) Openings exceeding the dimensional limits given in UG-36(b)(1) shall be provided with reinforcement that complies with the following rules. Two-thirds of the required reinforcement shall be within the following limits:

1-7(a)(1) parallel to vessel wall: the larger of three-fourths times the limit in UG-40(b)(1), or equal to the limit in UG-40(b)(2);

1-7(a)(2) normal to vessel wall: the smaller of the limit in UG-40(c)(1), or in UG-40(c)(2).

1-7(b) Openings for radial nozzles that exceed the limits in UG-36(b)(1)

1-7(b)(1) and which also are within the range defined by the following limits shall meet the requirements in (b)(2), (3), and (4) below:

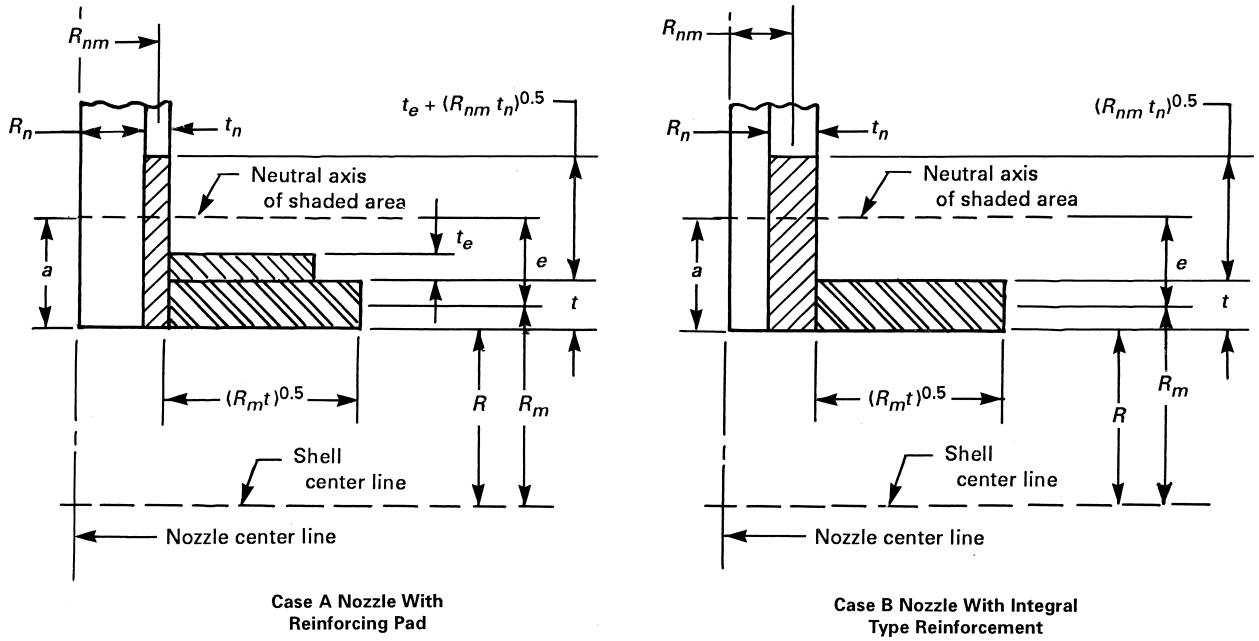
(a) vessel diameters greater than 60 in. (mm) I.D.;

(b) nozzle diameters which exceed 40 in. (mm) I.D. and also exceed $3.4\sqrt{Rt}$; the terms R and t are defined in Figs. 1-7-1 and 1-7-2;

(c) the ratio R_n/R does *not* exceed 0.7; for nozzle openings with R_n/R exceeding 0.7, refer to (c) below and/or U-2(g).

** For some of the terms of the above equation(s), it may be necessary to convert millimeters to meters to obtain a rational result in SI units.

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GENERAL NOTE:
When any part of a flange is located within the greater of the $\sqrt{R_{nm}\,t_n} + t_e$ or $16t_n + t_e$ limit as indicated in Fig. 1-7-1 or Fig. 1-7-2 Case A, or the greater of $\sqrt{R_{nm}\,t_n}$ or $16t_n$ for Fig. 1-7-1 or Fig. 1-7-2 Case B, the flange may be included as part of the section which resists bending moment.

FIG. 1-7-1

The rules are limited to radial nozzles in cylindrical shells that do not have internal projections, and do not include any analysis for stresses resulting from externally applied mechanical loads. For such cases U-2(g) shall apply.

1-7(b)(2) The membrane stress S_m as calculated by Eq. (1) or (2) below shall not exceed S, as defined in UG-37 for the applicable materials at design conditions. The maximum combined membrane stress S_m and bending stress S_b shall not exceed $1.5S$ at design conditions. S_b shall be calculated by Eq. (5) below.

1-7(b)(3) Evaluation of combined stresses from internal pressure and external loads shall be made in accordance with U-2(g).

1-7(b)(4) For membrane stress calculations, use the limits defined in Fig. 1-7-1, and comply with the strength of reinforcement requirements of UG-41. For bending stress calculation, the greater of the limits defined in Fig. 1-7-1 or Fig. 1-7-2 may be used. The strength reduction ratio requirements of UG-41 need not be applied, provided that the allowable stress ratio of the material in the nozzle neck, nozzle forging, reinforcing plate, and/or nozzle flange divided by the shell material allowable stress is at least 0.80.

NOTE: The bending stress S_b calculated by Eq. (5) is valid and applicable only at the nozzle neck-shell junction. It is a primary bending stress because it is a measure of the stiffness required to maintain equilibrium at the longitudinal axis junction of the nozzle-shell intersection due to the bending moment calculated by Eq. (3).

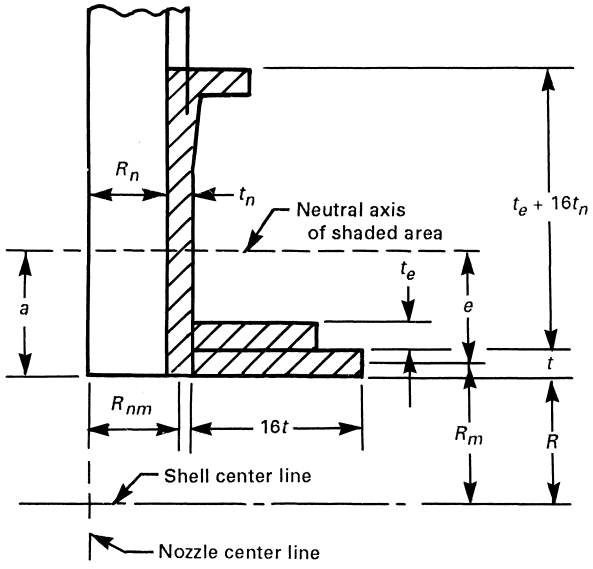
Case A (See Fig. 1-7-1)

$$S_m = P\left(\frac{R(R_n + t_n + \sqrt{R_m t}) + R_n(t + t_e + \sqrt{R_{nm}t_n})}{A_s}\right) \quad (1)$$

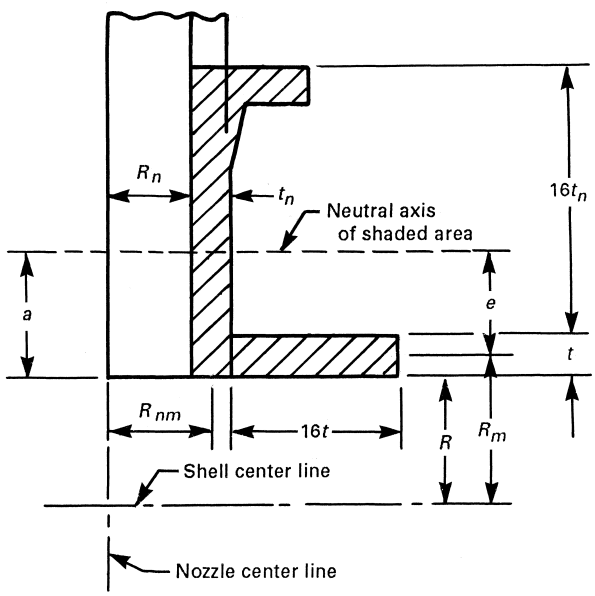
Case B (See Fig. 1-7-1)

$$S_m = P\left(\frac{R(R_n + t_n + \sqrt{R_m t}) + R_n(t + \sqrt{R_{nm}t_n})}{A_s}\right) \quad (2)$$

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Case A Nozzle With Reinforcing Pad



Case B Nozzle With Integral Type Reinforcement

GENERAL NOTE:
When any part of a flange is located within the greater of the $\sqrt{R_{nm}\,t_n} + t_e$ or $16t_n + t_e$ limit as indicated in Fig. 1-7-1 or Fig. 1-7-2 Case A, or the greater of $\sqrt{R_{nm}\,t_n}$ or $16t_n$ for Fig. 1-7-1 or Fig. 1-7-2 Case B, the flange may be included as part of the section which resists bending moment.

FIG. 1-7-2

Cases A and B (See Fig. 1-7-1 or Fig. 1-7-2)

$$M = \left(\frac{R_n^3}{6} + R\,R_n e\right) P \qquad (3)^{**}$$

$$a = e + t/2 \qquad (4)$$

$$S_b = \frac{Ma}{I} \qquad (5)^{**}$$

1-7(b)(5) Nomenclature. Symbols used in Figs. 1-7-1 and 1-7-2 are as defined in UG-37(a) and as follows:

A_s = shaded (cross-hatched) area in Fig. 1-7-1, Case A or Case B, in.^{2} (mm^2)

I = moment of inertia of the larger of the shaded areas in Fig. 1-7-1 or Fig. 1-7-2 about neutral axis, in.^{4} (mm^4)

a = distance between neutral axis of the shaded area in Fig. 1-7-1 or Fig. 1-7-2 and the inside of vessel wall, in. (mm)

R_m = mean radius of shell, in. (mm)

R_{nm} = mean radius of nozzle neck, in. (mm)

e = distance between neutral axis of the shaded area and midwall of the shell, in. (mm)

S_m = membrane stress calculated by Eq. (1) or (2), psi (kPa)

S_b = bending stress at the intersection of inside of the nozzle neck and inside of the vessel shell along the vessel shell longitudinal axis, psi (kPa)

S_y = yield strength of the material at test temperature, see Table Y-1 in Subpart 1 of Section II, Part D, psi (kPa)

1-7(c) It is recommended that special consideration be given to the fabrication details used and inspection

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employed on large openings; reinforcement often may be advantageously obtained by use of heavier shell plate for a vessel course or inserted locally around the opening; welds may be ground to concave contour and the inside corners of the opening rounded to a generous radius to reduce stress concentrations. When radiographic examination of welds is not practicable, liquid penetrant examination may be used with nonmagnetic materials and either liquid penetrant or magnetic particle inspection with ferromagnetic materials. If magnetic particle inspection is employed, the prod method is preferred. The degree to which such measures should be used depends on the particular application and the severity of the intended service. Appropriate proof testing may be advisable in extreme cases of large openings approaching full vessel diameter, openings of unusual shape, etc.

1-8 RULES FOR REINFORCEMENT OF CONE-TO-CYLINDER JUNCTION UNDER EXTERNAL PRESSURE

(a) The formulas of (b) and (c) below provide for the design of reinforcement, if needed, at the cone-to-cylinder junctions for reducer sections and conical heads where all the elements have a common axis and the half-apex angle $\alpha \leq 60$ deg. Subparagraph (e) below provides for special analysis in the design of cone-to-cylinder intersections with or without reinforcing rings where α is greater than 60 deg.

In the design of reinforcement for a cone-to-cylinder juncture, the requirements of UG-41 shall be met.

The nomenclature given below is used in the formulas of the following subparagraphs:

A = factor determined from Fig. G and used to enter the applicable material chart in Subpart 3 of Section II, Part D

A_{eL} = effective area of reinforcement at large end intersection, in.^{2} (mm^2)

A_{es} = effective area of reinforcement at small end intersection, in.^{2} (mm^2)

A_{rL} = required area of reinforcement at large end of cone, in.^{2} (mm^2)

A_{rs} = required area of reinforcement at small end of cone, in.^{2} (mm^2)

A_s = cross-sectional area of the stiffening ring, sq in. (mm^2)

A_T = equivalent area of cylinder, cone, and stiffening ring, sq in. (mm^2), where

$$A_{TL} = \frac{L_L t_s}{2} + \frac{L_c t_c}{2} + A_s \text{ for large end}$$

$$A_{TS} = \frac{L_s t_s}{2} + \frac{L_c t_c}{2} + A_s \text{ for small end}$$

B = factor determined from the applicable material chart in Subpart 3 of Section II, Part D for maximum design metal temperature, psi (kPa) [see UG-20(c)]

D_L = outside diameter of large end of conical section under consideration, in. (mm)

D_o = outside diameter of cylindrical shell, in. (mm). (In conical shell calculations, the value of D_s and D_L should be used in calculations in place of D_o depending on whether the small end D_s, or large end D_L, is being examined.)

D_s = outside diameter at small end of conical section under consideration, in. (mm)

E_1 = efficiency of longitudinal joint in cylinder. For compression (such as at small end of cone), $E_1 = 1.0$ for butt welds.

E_2 = efficiency of longitudinal joint in cone. For compression, $E_2 = 1.0$ for butt welds.

E_c = modulus of elasticity of cone material, psi (kPa)

E_r = modulus of elasticity of stiffening ring material, psi (kPa)

E_s = modulus of elasticity of shell material, psi (kPa)

$E_x = E_c$, E_r, or E_s

f_1 = axial load at large end due to wind, dead load, etc., excluding pressure, lb/in. (kN/m)

f_2 = axial load at small end due to wind, dead load, etc., excluding pressure, lb/in. (kN/m)

I = available moment of inertia of the stiffening ring cross section about its neutral axis parallel to the axis of the shell, in.^{4} (mm^4)

I' = available moment of inertia of combined shell-cone or ring-shell-cone cross section about its neutral axis parallel to the axis of the shell, in.^{4} (mm^4). The nominal shell thickness t_s shall be used, and the width of the shell which is taken as contributing to the moment of inertia of the combined section shall not be greater than $1.10\sqrt{Dt_s}$ and shall be taken as lying one-half on each side of the cone-to-cylinder junction or of the centroid of the ring. Portions of the shell plate shall not be considered as contributing area to more than one stiffening ring.

CAUTIONARY NOTE: Stiffening rings may be subject to lateral buckling. This should be considered in addition to the requirements for I_s and I'_s [see U-2(g)].

I_s = required moment of inertia of the stiffening ring cross section about its neutral axis parallel to the axis of the shell, in.^{4} (mm^4)

I'_s = required moment of inertia of the combined

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shell-cone or ring-shell-cone cross section about its neutral axis parallel to the axis of the shell, in.^{4} (mm^4)

If the stiffeners should be so located that the maximum permissible effective shell sections overlap on either or both sides of a stiffener, the effective shell section for that stiffener shall be shortened by one-half of each overlap.

k = 1 when additional area of reinforcement is not required
= y/S_rE_r when a stiffening ring is required, but k is not less than 1.0

L = axial length of cone, in. (mm)

L_c = length of cone between stiffening rings measured along surface of cone, in. (mm). For cones without intermediate stiffeners,

$$L_c = \sqrt{L^2 + (R_L - R_s)^2}$$

L_L = design length of a vessel section, in. (mm), taken as the largest of the following:
(1) the center-to-center distance between the cone-to-large-shell junction and an adjacent stiffening ring on the large shell;
(2) the distance between the cone-to-large-shell junction and one-third the depth of head on the other end of the large shell if no other stiffening rings are used.

L_s = design length of a vessel section, in. (mm), taken as the largest of the following:
(1) the center-to-center distance between the cone-to-small-shell junction and adjacent stiffening ring on the small shell;
(2) the distance between the cone-to-small-shell junction and one-third the depth of head on the other end of the small shell if no other stiffening rings are used.

P = external design pressure, psi (kPa)

Q_L = algebraical sum of $PR_L/2$ and f_1, lb/in. (kN/m)

Q_s = algebraical sum of $PR_s/2$ and f_2, lb/in. (kN/m)

R_L = outside radius of large cylinder, in. (mm)

R_s = outside radius of small cylinder, in. (mm)

S_c = allowable stress of cone material at design temperature, psi (kPa)

S_r = allowable stress of stiffening ring material at design temperature, psi (kPa)

S_s = allowable stress of cylinder material at design temperature, psi (kPa)

t = minimum required thickness of cylinder at cone-to-cylinder junction [see UG-28(c)], in. (mm)

TABLE 1-8.1
VALUES OF Δ FOR JUNCTIONS AT THE LARGE CYLINDER FOR $\alpha \le 60$ deg.

P/S_sE_1	0	0.002	0.005	0.010	0.02
Δ, deg.	0	5	7	10	15
P/S_sE_1	0.04	0.08	0.10	0.125	0.15
Δ, deg.	21	29	33	37	40
P/S_sE_1	0.20	0.25	0.30	0.35	Note (1)
Δ, deg.	47	52	57	60	

NOTE:
(1) Δ = 60 deg. for greater values of P/SE.

t_c = nominal thickness of cone at cone-to-cylinder junction, in. (mm)

t_r = minimum required thickness of cone at cone-to-cylinder junction, in. (mm)

t_s = nominal thickness of cylinder at cone-to-cylinder junction, in. (mm)

y = cone-to-cylinder factor
= S_sE_s for stiffening ring on shell
= S_cE_c for stiffening ring on cone

α = one-half the included (apex) angle of the cone at the center line of the head

Δ = value to indicate need for reinforcement at cone-to-cylinder intersection having a half-apex angle $\alpha \le 60$ deg. When $\Delta \ge \alpha$, no reinforcement is required at the junction (see Table 1-8.1).

(b) Reinforcement shall be provided at the junction of the cone with the large cylinder for conical heads and reducers without knuckles when the value of Δ obtained from Table 1-8.1 using the appropriate ratio P/S_sE_1 is less than α. Interpolation may be made in the Table.

The required area of reinforcement shall be at least equal to that indicated by the following formula when Q_L is in compression:

$$A_{rL} = \frac{kQ_LR_L \tan\alpha}{S_sE_1}\left[1 - \tfrac{1}{4}\left(\frac{PR_L - Q_L}{Q_L}\right)\frac{\Delta}{\alpha}\right] \quad (1)^{**}$$

At the large end of the cone-to-cylinder juncture, the $PR_L/2$ term is in compression. When f_1 is in tension and the quantity is larger than the $PR_L/2$ term, the design shall be in accordance with U-2(g). The calculated localized stresses at the discontinuity shall not exceed the stress values specified in 1-5(g)(1) and (2).

** For some of the terms of the above equation(s), it may be necessary to convert millimeters to meters to obtain a rational result in SI units.

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The effective area of reinforcement can be determined in accordance with the following formula:

$$A_{eL} = 0.55\sqrt{D_L t_s}(t_s + t_c / \cos \alpha) \quad (2)$$

Any additional area of stiffening which is required shall be situated within a distance of $\sqrt{R_L t_s}$ from the junction of the reducer and the cylinder. The centroid of the added area shall be within a distance of $0.25 \times \sqrt{R_L t_s}$ from the junction.

When the cone-to-cylinder or knuckle-to-cylinder juncture is a line of support, the moment of inertia for a stiffening ring at the large end shall be determined by the following procedure.

Step 1. Assuming that the shell has been designed and D_L, L_L, and t are known, select a member to be used for the stiffening ring and determine cross-sectional area A_{TL}. Then calculate factor B using the following formula. If F_L is a negative number, the design shall be in accordance with U-2(g):

$$B = \tfrac{3}{4}\left(\frac{F_L D_L}{A_{TL}}\right) \quad **$$

where

$$F_L = PM + f_1 \tan \alpha \quad **$$

$$M = \frac{-R_L \tan \alpha}{2} + \frac{L_L}{2} + \frac{R_L^2 - R_s^2}{3R_L \tan \alpha} \quad **$$

Step 2. Enter the right-hand side of the applicable material chart in Subpart 3 of Section II, Part D for the material under consideration at the value of B determined by Step 1. If different materials are used for the shell and stiffening ring, use the material chart resulting in the larger value of A in Step 4 below.

Step 3. Move horizontally to the left to the material/temperature line for the design metal temperature. For values of B falling below the left end of the material/temperature line, see Step 5 below.

Step 4. Move vertically to the bottom of the chart and read the value of A.

Step 5. For value of B falling below the left end of the material/temperature line for the design temperature, the value of A can be calculated using the formula $A = 2B/E_x$. For value of B above the material/temperature line for the design temperature, the design shall be either per U-2(g) or by changing the cone or cylinder configuration, stiffening ring location on the shell, and/or reducing the axial compressive force to reduce the B value to below or at the material/temperature line for the design temperature. For values of B having multiple values of A, such as when B falls on a horizontal portion of the curve, the smallest value of A shall be used.

Step 6. Compute the value of the required moment of inertia from the formulas for I_s or I'_s. For the circumferential stiffening ring only,

$$I_s = \frac{AD_L^2 A_{TL}}{14.0}$$

For the shell-cone or ring-shell-cone section,

$$I'_s = \frac{AD_L^2 A_{TL}}{10.9}$$

Step 7. Determine the available moment of inertia of the ring only I or the shell-cone or ring-shell-cone I'.

Step 8. When the ring only is used,

$$I \geq I_s$$

and when the shell-cone or ring-shell-cone is used,

$$I' \geq I'_s$$

If the equation is not satisfied, a new section with a larger moment of inertia must be selected, and the calculation shall be done again until the equation is met.

The requirements of UG-29(b), (c), (d), (e), and (f) and UG-30 are to be met in attaching stiffening rings to the shell.

(c) Reinforcement shall be provided at the junction of the conical shell of a reducer without a flare and the small cylinder. The required area of reinforcement shall be at least equal to that indicated by the following formula when Q_s is in compression:

$$A_{rs} = \frac{kQ_s R_s \tan \alpha}{S_s E_1} \quad (3)**$$

At the small end of the cone-to-cylinder juncture, the $PR_s/2$ term is in compression. When f_2 is in tension and the quantity is larger than the $PR_s/2$ term, the design shall be in accordance with U-2(g). The calcu-

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lated localized stresses at the discontinuity shall not exceed the stress values specified in 1-5(g)(1) and (2).

The effective area of reinforcement can determined in accordance with the following formula:

$$A_{es} = 0.55\sqrt{D_s t_s}[(t_s - t) + (t_c - t_r)/\cos\alpha] \qquad (4)$$

Any additional area of stiffenerwhich is required shall be situated within a distance of $\sqrt{R_s t_s}$ from the junction, and the centroid of the added area shall be within a distance of $0.25\sqrt{R_s t_s}$ from the junction.

When the cone-to-cylinder or knuckle-to-cylinder juncture is a line of support, the moment of inertia for a stiffening ring at the small end shall be determined by the following procedure.

Step 1. Assuming that the shell has been designed and D_s, L_s, and t are known, select a member to be used for the stiffening ring and determine cross-sectional area A_{TS}. Then calculate factor B using the following formula. If F_s is a negative number, the design shall be in accordance with U-2(g):

$$B = \tfrac{3}{4}\left(\frac{F_s D_s}{A_{TS}}\right)$$

where

$$F_s = PN + f_2 \tan\alpha$$

$$N = \frac{R_s \tan\alpha}{2} + \frac{L_s}{2} + \frac{R_L^2 - R_s^2}{6R_s \tan\alpha} \qquad **$$

Step 2. Enter the right-hand side of theapplicable material chart in Subpart 3 of Section II, Part D for the material under consideration at the value of B determined by Step 1. If different materials are used for the shell and stiffening ring, use the material chart resulting in the larger value of A in Step 4 below.

Step 3. Move horizontally to the left to the material/temperature line for the design metal temperature. For values of B falling below the left end of the material/temperature line, see Step 5 below.

Step 4. Move vertically to the bottom of the chart and read the value of A.

Step 5. For values of B falling below the left end of the material/temperature line for the design temperature, the value of A can be calculated using the formula $A = 2B/E_x$. For value of B above the material/temperature line for the design temperature, the design shall be either per U-2(g) or by changing the cone or cylinder configuration, stiffening ring location on the shell, and/or reducing the axial compressive force to reduce the B value to below or at the material/temperature line for the design temperature. For values of B having multiple values of A, such as when B falls on a horizontal portion of the curve, the smallest value of A shall be used.

Step 6. Compute the value of the required moment of inertia from the formulas for I_s or I'_s.

For the circumferential stiffening ring only,

$$I_s = \frac{AD_s^2 A_{TS}}{14.0}$$

For the shell-cone or ring-shell-cone section,

$$I'_s = \frac{AD_s^2 A_{TS}}{10.9}$$

Step 7. Determine the available moment of inertia of the ring only I or the shell-cone or ring-shell-cone I'.

Step 8. When the ring only is used,

$$I \geq I_s$$

and when the shell-cone or ring-shell-cone is used:

$$I' \geq I'_s$$

If the equation is not satisfied, a new section with a larger moment of inertia must be selected, and the calculation shall be done again until the equation is met.

The requirements of UG-29(b), (c), (d), (e), and (f) and UG-30 are to be met in attaching stiffening rings to the shell.

(d) Reducers not described in UG-36(e)(5), such as those made up of two or more conical frustums having different slopes, may be designed in accordance with (e).

(e) When the half-apex angle α is greater than 60 deg., cone-to-cylinder junctions without a knuckle may be used, with or without reinforcing rings, if the design is based on special analysis, such as the beam-on-elastic-foundation analysis of Timoshenko, Hetenyi, or Watts and Lang. See U-2(g). The effect of shell and cone buckling on the required area and moment of inertia at the joint is to be taken into consideration in the analysis. When such an analysis is made, the

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calculated localized stresses at the discontinuity shall not exceed the following values.

(1) (Membrane hoop stress) + (average discontinuity hoop stress) shall not be greater than $1\frac{1}{2}SE$.

(2) (Membrane longitudinal stress) + (discontinuity longitudinal stress due to bending) shall not be greater than $4SE$, where the "average discontinuity hoop stress" is the average hoop stress across the wall thickness due to the discontinuity at the junction, disregarding the effect of Poisson's ratio times the longitudinal stress at the surfaces.

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