ARTICLE NE-3000
DESIGN

NE-3100 GENERAL DESIGN

NE-3110 LOADING CRITERIA

NE-3111 Loading Conditions

The loadings that shall be taken into account in designing a vessel shall include, but are not limited to, those in (a) through (i) below
(a) internal and external pressure
(b) impact loads, including rapidly fluctuating pressure
(c) weight of the vessel and normal contents under all conditions, including additional pressure due to static and dynamic head of liquids
(d) superimposed loads such as other components, operating equipment, insulation, corrosion resistant or erosion resistant linings, and piping
(e) wind loads, snow loads, and vibration loads where specified
(f) reactions of supporting lugs, rings, saddles, or other types of supports
(g) temperature effects
(h) reactions to steam and water jet impingement
(i) earthquake loads

NE-3112 Design Loadings

The containment vessel or portions thereof may be exposed to more than one pressure, temperature, and mechanical load condition as provided in the Design Specifications. The specified design parameters for Design Loadings shall be called Design Pressure, Design Temperature, and Design Mechanical Loads. These specified design parameters shall be used in computations to show compliance with the requirements on Design Loadings in NE-3220.

NE-3112.1 Design Pressure. The design internal pressure shall not be less than 100% of the maximum containment internal pressure under conditions for which the containment function is required. The design external pressure shall not be less than 100% of the maximum containment external pressure. Stability of the vessel shell may be provided by structures bearing directly against the shell or by structural attachments.

NE-3112.2 Design Temperature.
(a) The Design Temperature shall not be less than the maximum containment temperature at the coincident maximum containment pressure. The Design Temperature shall be used in all computations involving Design Pressure and coincidental Design Mechanical Loads.

(b) Where a vessel is heated by external or internal heat generation, the effect of such heating shall be incorporated in the establishment of the vessel Design Temperature.

NE-3112.3 Design Mechanical Loads. The values of dead loads and any hydrostatic loads coincident with Design Pressure shall be designated as Design Mechanical Loads. These mechanical loads shall be used in all computations involving Design Pressure and Design Temperature.

NE-3112.4 Allowable Stress Intensity and Stress Values. The rules for allowable stress intensities and allowable stresses are given in NE-3200 for vessels designed by analysis where the allowable stress intensity \( S_{mc} \) is 1.1 times the \( S \) value given in Section II, Part D, Subpart 1, Tables 1A and 1B, except \( S_{mc} \) shall not exceed 90% of the material's yield strength at temperature, shown in Section II, Part D, Table Y-1. The allowable stress \( S \) to be used in the equations of NE-3300 shall be 1.1 times those listed in Section II, Part D, Subpart 1, Tables 1A and 1B for the materials permitted in Table NE-2121(a)-1, except \( S \) shall not exceed 90% of the material's yield strength at temperature, shown in Section II, Part D, Table Y-1. The material shall not be used at metal and design temperatures that exceed the temperature limit in the applicability column for which stress intensity values are listed. The values in the Tables may be interpolated for intermediate temperatures.

NE-3113 Service Limits

Each Service Loading to which the component may be subjected shall be categorized in accordance with the following definitions and shall be described in the Design Specifications (NCA-23250) in such detail as will provide a complete basis for construction in accordance with these rules. The Service Limit categories shall be as defined in the next four subparagraphs.

NE-3113.1 Level A Service Limits. Level A Service Limits apply to all sustained loads in combination with the plant or system design basis accident loads for which the containment function is required. Other loads, either separately or in combination with the above loads, may also be considered under this service limit.

NE-3113.2 Level B Service Limits. This service limit applies to those applicable loads subject to Level A Service Limits in combination with loads due to natural
phenomena for which the plant must remain operational. Other loads, either separately or in combination with the above loads, may also be considered under this service limit.

**NE-3113.3 Level C Service Limits.** This service limit applies to those applicable loads subject to Level A and B Service Limits in combination with loads due to natural phenomena for which safe shutdown of the plant is required. Other loads, either separately or in combination with the above loads, may also be considered under this service limit.

**NE-3113.4 Level D Service Limits.** This service limit applies to those loads subject to other service limits in combination with loadings of a local dynamic nature, such as jet impingement, pipe whip, and pipe reaction loads resulting from a postulated pipe rupture, for which the containment function is required.

**NE-3113.5 Service Limits.** The Service Limits for design by analysis are specified in NE-3200. The Service Limits for design by equation are specified in NE-3300.

**NE-3114 Testing Conditions**

Testing conditions are those tests in addition to the ten hydrostatic or pneumatic tests permitted by NE-6222 and NE-6322, including leak tests or subsequent hydrostatic or pneumatic tests.

**NE-3120 SPECIAL CONSIDERATIONS**

**NE-3121 Corrosion**

Material subject to thinning by corrosion, erosion, mechanical abrasion, or other environmental effects shall have provision made in the Design Specifications for these effects by indicating the increase in the thickness of the base metal over that determined by the design equations (NE-2160). Material added or included for these purposes need not be of the same thickness for all areas of the component if different rates of attack are indicated for the various areas. It should be noted that the tests on which the design fatigue curves (Section III Appendices, Figures I-9.1 through 1-9.7) are based did not include tests in the presence of corrosive environments which might accelerate fatigue failure.

**NE-3122 Cladding**

The rules of this paragraph apply to the design and analysis of clad components constructed of material permitted under this Subsection. When corrosion or erosion of the cladding material is expected, the cladding thickness shall be increased by an amount that in the judgment of the Owner will provide the desired service life.

**NE-3122.1 Primary Stresses.** When stress analysis is required, no structural strength shall be attributed to the cladding in satisfying NE-3221.

**NE-3122.2 Design Dimensions.** The dimensions given in (a) and (b) below shall be used in the design of the component.

(a) For components subjected to internal pressure, the inside diameter shall be taken at the nominal inner face of the cladding.

(b) For components subjected to external pressure, the outside diameter shall be taken at the outer face of the base metal.

**NE-3122.3 Bearing Stresses.** In satisfying NE-3227.1, the presence of cladding shall be included.

**NE-3122.4 Maximum Allowable Stress Values.**

(a) **Integrally Clad Plate Without Credit for Full Cladding Thickness.** Except as permitted in (b) below, design calculations shall be based on the total thickness of the clad plates less the specified nominal minimum thickness of cladding. A reasonable excess thickness either of the actual cladding or of the same thickness of corrosion resistant weld metal may be included in the design calculations as an equal thickness of base plate. The maximum allowable stress value shall be 1.1 times that given for the base plate material in Section II, Part D, Subpart 1, Tables 1A and 1B.

(b) **Integrally Clad Plate With Credit for Cladding Thickness.** When clad plate conforms to one of the specifications listed in NE-2127(a) and the joints are completed by depositing corrosion resisting weld metal over the weld in the base plate to restore the cladding, the design calculations may be based on a thickness equal to the nominal thickness of the base plate plus \( S_c/S_b \) times the nominal thickness of the cladding after any allowance provided for corrosion has been deducted, where

\[
S_b = \text{maximum allowable stress value for the base plate at the Design Temperature, psi (MPa)}
\]

\[
S_c = \text{maximum allowable stress value for the cladding at the Design Temperature, psi (MPa)}
\]

When \( S_c \) is greater than \( S_b \), the multiplier \( S_c/S_b \) shall be taken equal to unity. The maximum allowable stress value shall be 1.1 times that given for the base plate material in Section II, Part D, Subpart 1, Tables 1A and 1B.

**NE-3122.5 Maximum Allowable Temperature.**

(a) When the design calculations are based on the thickness of base plate exclusive of cladding thickness, the maximum service metal temperature of the vessel shall be that allowed for the base plate material.

(b) When the design calculations are based on the full thickness of clad plate as permitted in NE-3122.4(b), the maximum service metal temperature shall be the lower of the values allowed for the base plate material and the cladding material.
NE-3123  Welds Between Dissimilar Metals

In satisfying the requirements of this Subarticle, caution should be exercised in design and construction involving dissimilar metals having different coefficients of thermal expansion.

NE-3125  Configuration

Accessibility to permit the examinations required by the Edition and Addenda of Section XI as specified in the Design Specification for the component shall be provided in the design of the component.

NE-3130  GENERAL DESIGN RULES

NE-3131  General Requirements

(a) The containment vessel design shall be such that the rules of NE-3200 are satisfied. However, in the absence of substantial mechanical or thermal loads other than pressure, the rules of NE-3300 may be used in lieu of the rules of NE-3200 for those configurations which are explicitly treated in NE-3300.

(b) For vessels, or portions thereof, designed to the rules of NE-3200, the provisions of NE-3133 may be used, if applicable, in lieu of the rules of NE-3222. For those portions designed to the rules of NE-3300, the requirements of NE-3133 shall be met.

NE-3132  Dimensional Standards for Standard Products

Dimensions of standard products shall comply with the standards and specifications listed in Table NCA-7100-1 when the standard or specification is referenced in the specific design Subarticle. However, compliance with these standards does not replace or eliminate the requirements for stress analysis when called for by the design Subarticle for a specific component.

NE-3133  Components Under External Loading

NE-3133.1  General. Rules are given in this paragraph for determining the thickness under external pressure loading in spherical shells, cylindrical shells with or without stiffening rings, and tubular products consisting of pipes, tubes, and fittings. Charts for determining the stresses in shells, hemispherical heads, and tubular products are given in Section II, Part D, Subpart 3.

NE-3133.2  Nomenclature. The symbols used in this paragraph are defined as follows:

- $A_s$ = factor determined from Figure VII-1100-1 in Section II, Part D, Subpart 3 and used to enter the applicable material chart in Section II, Part D, Subpart 3. For the case of cylinders having $D_o/A_s$ values less than 10, see NE-3133.3(b). Also, factor determined from the applicable chart in Section II, Part D, Subpart 3 for the material used in a stiffening ring, corresponding to the factor $B$ and the design metal temperature for the shell under consideration.

- $D_o$ = cross-sectional area of a stiffening ring factor determined from the applicable chart in Section II, Part D, Subpart 3 for the material used in a shell or stiffening ring at the design metal temperature, psi (MPa).

- $E$ = modulus of elasticity of material at Design Temperature, psi (MPa). For external pressure and axial compression design in accordance with this section, the modulus of elasticity to be used shall be taken from the applicable materials chart in Section II, Part D, Subpart 3. (Interpolation may be made between lines for intermediate temperatures.) The modulus of elasticity values shown in Section II, Part D, Subpart 3 for material groups may differ from those values listed in Section II, Part D, Subpart 2, Table TM for specific materials. Section II, Part D, Subpart 3 values shall be applied only to external pressure and axial compression design.

- $I$ = available moment of inertia of the stiffening ring about its neutral axis, parallel to the axis of the shell

- $I' = available moment of inertia of the combined ring-shell cross section about its neutral axis, parallel to the shell. The width of the shell which is taken as contributing to the combined moment of inertia shall not be greater than 1.10, $D_o/A_s$, and shall be taken as lying one-half on each side of the centroid of the ring. Portions of shell plates shall not be considered as contributing area to more than one stiffening ring.

- $I_s$ = required moment of inertia of the stiffening ring about its neutral axis parallel to the axis of the shell

- $I'_s$ = required moment of inertia of the combined ring-shell section about its neutral axis parallel to the axis of the shell

- $l$ = total length of a tube between tubeshells, or the design length of a vessel section, taken as the largest of the following:

  (a) the distance between head tangent lines plus one-third of the depth of each head if there are no stiffening rings

  (b) the greatest center-to-center distance between any two adjacent stiffening rings, or

  (c) the distance from the center of the first stiffening ring to the head tangent line plus one-third of the depth of the head, all measured parallel to the axis of the vessel

- $l_s$ = one-half of the distance from the center line of the stiffening ring to the next line of support on one side, plus one-half of the center line distance to...
the next line of support on the other side of the stiffening ring, both measured parallel to the axis of the component. A line of support is:

(a) a stiffening ring that meets the requirement of this paragraph
(b) a circumferential line on a head at one-third the depth of the head from the head tangent line, or
(c) a circumferential connection to a jacket for a jacketed section of a cylindrical shell

\[ P = \text{external design pressure, psi (MPa)} \] (gage or absolute, as required)

\[ P_a = \text{allowable external pressure, psi (MPa)} \] (gage or absolute, as required)

\[ R = \text{inside radius of spherical shell} \]

\[ S = \text{the lesser of 2 times the allowable stress at design metal temperature from Section II, Part D, Subpart 1, Tables 1A and 1B or 0.9 times the tabulated yield strength at design metal temperature from Section II, Part D, Subpart 1, Table Y-1, psi (MPa)} \]

\[ T = \text{minimum required thickness of cylindrical shell or tube, or spherical shell} \]

\[ T_n = \text{nominal thickness used, less corrosion allowance, of a cylindrical shell or tube} \]

**NE-3313.3 Cylindrical Shells.** The thickness of a cylindrical shell under external pressure shall be determined by the procedure given in (a) or (b) below.

(a) Cylinders having \( \frac{D_o}{T} \) values \( \geq 10 \):

**Step 1.** Assume a value for \( T \) and determine the ratios \( L/D_o \) and \( D_o/T \).

**Step 2.** Enter Section II, Part D, Subpart 3, Figure C at the value of \( L/D_o \) determined in **Step 1.** For values of \( L/D_o \) greater than 50, enter the chart at a value of \( L/D_o = 50 \). For values of \( L/D_o \) less than 0.05, enter the chart at a value of \( L/D_o = 0.05 \).

**Step 3.** Move horizontally to the line for the value of \( D_o/T \) determined in **Step 1.** Interpolation may be made for intermediate values of \( D_o/T \). From this point of intersection, move vertically downward to determine the value of factor \( A \).

**Step 4.** Using the value of \( A \) calculated in **Step 3,** enter the applicable material chart in Section II, Part D, Subpart 3 for the material under consideration. Move vertically to an intersection with the material/temperature line for the Design Temperature. Interpolation may be made between lines for intermediate temperatures. In cases where the value of \( A \) falls to the right of the end of the material/temperature line, assume an intersection with the horizontal projection of the upper end of the material/temperature line. For values of \( A \) falling to the left of the material/temperature line, see **Step 7.**

**Step 5.** From the intersection obtained in **Step 4,** move horizontally to the right and read the value of \( B \) psi (MPa).

Step 6. Using this value of \( B \), calculate the value of the maximum allowable external pressure \( P_a \) using the following equation:

\[ P_a = \frac{4B}{3(D_o/T)} \]

Step 7. For values of \( A \) falling to the left of the applicable material/temperature line, the value of \( P_a \) can be calculated by using the following equation:

\[ P_a = \frac{2AE}{3(D_o/T)} \]

Step 8. Compare \( P_a \) with \( P \). If \( P_a \) is smaller than \( P \), select a larger value for \( T \) and repeat the design procedure until a value of \( P_a \) is obtained that is equal to or greater than \( P \).

(b) Cylinders having \( D_o/T \) values \( < 10 \):

**Step 1.** Using the same procedure as given in (a) above, obtain the value of \( B \). For values of \( D_o/T \) less than 4, the value of \( A \) can be calculated using the following equation:

\[ A = \frac{1.1}{(D_o/T)^2} \]

For values of \( A \) greater than 0.10, use a value of 0.10.

**Step 2.** Using the value of \( B \) obtained in **Step 1,** calculate a value \( P_{a1} \) using the following equation:

\[ P_{a1} = \frac{2.167}{(D_o/T)} - 0.0833B \]

**Step 3.** Calculate a value \( P_{a2} \) using the following equation:

\[ P_{a2} = \frac{2S}{(D_o/T)} \left[ 1 - \frac{1}{(D_o/T)} \right] \]

**Step 4.** The smaller of the values of \( P_{a1} \) calculated in **Step 2,** or \( P_{a2} \) calculated in **Step 3,** shall be used as the maximum allowable external pressure \( P_a \). Compare \( P_a \) with \( P \). If \( P_a \) is smaller than \( P \), select a larger value for \( T \) and repeat the design procedure until a value for \( P_a \) is obtained that is equal to or greater than \( P \).

**NE-3313.4 Spherical Shells and Formed Heads.**

(a) Spherical Shells. The minimum required thickness of a spherical shell under external pressure, either seamless or of built-up construction with butt joints, shall be determined by the procedure given in **Steps 1** through **6.**

**Step 1.** Assume a value for \( T \) and calculate the value of factor \( A \) using the following equation:

\[ A = \frac{0.125}{(R/T)} \]

**Step 2.** Using the value of \( A \) calculated in **Step 1,** enter the applicable material chart in Section II, Part D, Subpart 3 for the material under consideration. Move vertically to an intersection with the material/temperature line for the
Design Temperature. Interpolation may be made between lines for intermediate temperatures. In cases where the value at \( A \) falls to the right of the end of the material/temperature line, assume an intersection with the horizontal projection of the upper end of the material/temperature line. For values at \( A \) falling to the left of the material/temperature line, see Step 5.

Step 3. From the intersection obtained in Step 2, move horizontally to the right and read the value of factor \( B \) psi (MPa).

Step 4. Using the value of \( B \) obtained in Step 3, calculate the value of the maximum allowable external pressure \( P_a \) using the following equation:

\[
P_a = \frac{B}{(R/T)}
\]

Step 5. For values of \( A \) falling to the left of the applicable material/temperature line for the Design Temperature, the value of \( P_a \) can be calculated using the following equation:

\[
P_a = \frac{0.0625E}{(R/T)^2}
\]

Step 6. Compare \( P_a \) obtained in Step 4 or Step 5 with \( P \). If \( P_a \) is smaller than \( P \), select a larger value for \( T \) and repeat the design procedure until a value for \( P_a \) is obtained that is equal to or greater than \( P \).

(b) The nomenclature defined below is used in the equations of (c) through (e) below.

- \( D_o \): outside diameter of the head skirt or the outside diameter of a cone head at the point under consideration, measured perpendicular to the longitudinal axis of the cone, in. (mm)
- \( K_1 \): a factor depending on the ellipsoidal head proportions, given in Table NE-3332.2-1
- \( R \): for hemispherical heads, the inside radius in the corroded condition, in. (mm)
- \( = \): for ellipsoidal heads, the equivalent inside spherical radius taken as \( K_1 D_o \) in the corroded condition, in. (mm)
- \( = \): for torispherical heads, the inside radius of the crown portion of the head in the corroded condition, in. (mm)
- \( T \): minimum required thickness of head after forming, exclusive of corrosion allowance, in. (mm)

(c) Hemispherical Heads. The required thickness of a hemispherical head having pressure on the convex side shall be determined in the same manner as outlined in (a) above for determining the thickness for a spherical shell.

(d) Ellipsoidal Heads. The required thickness of an ellipsoidal head having pressure on the convex side, either seamless or of built-up construction with butt joints, shall not be less than that determined by the following procedure:

**Step 1.** Assume a value for \( T \) and calculate the value of factor \( A \) using the following equation:

\[
A = \frac{0.125}{(R/T)}
\]

**Step 2.** Using the value of \( A \) calculated in Step 1, follow the same procedure as that given for spherical shells in (a) through (e) above.

(e) Torispherical Heads. The required thickness of a torispherical head having pressure on the convex side, either seamless or of built-up construction with butt joints, shall not be less than that determined by the same design procedure as is used for ellipsoidal heads given in (d) above, using the appropriate value for \( R \).

**NE-3313.5 Stiffening Rings for Cylindrical Shells.**

(a) The required moment of inertia of a circumferential stiffening ring shall not be less than that determined by one of the following two equations:

\[
l_e = \frac{D_o^2 L_i (T + \frac{A_h}{L_o}) A}{14}
\]

\[
l_e' = \frac{D_o^2 L_i (T + \frac{A_h}{L_o}) A}{10.9}
\]

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 the stiffener shall be shortened by one-half of each overlap.

(b) The available moment of inertia \( I \) or \( I' \) for a stiffening ring shall be determined by the following procedure.

**Step 1.** Assuming that the shell has been designed and \( D_o \), \( L_o \), and \( T_n \) are known, select a member to be used for the stiffening ring and determine its cross-sectional area \( A_s \). Then calculate factor \( B \) using the following equation:

\[
B = \frac{3}{4} \left( \frac{P D_o}{T_n + A_h/L_o} \right)^{1/4}
\]

**Step 2.** Enter the right side of the applicable material chart in Section II, Part D, Subpart 3 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 or Step 5 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.
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 equation:

\[
A = \frac{2B}{E}
\]

Step 6. Compute the value of required moment of inertia from the equations for \( I_2 \) or \( I'_2 \) above.

Step 7. Calculate the available moment of inertia \( I \) or \( I' \) of the stiffening ring using the section corresponding to that used in Step 6.

Step 8. If the required moment of inertia is greater than the moment of inertia for the section selected in Step 1, a new section with a larger moment of inertia must be selected and a new moment of inertia determined. If the required moment of inertia is smaller than the moment of inertia for the section selected in Step 1, that section should be satisfactory.

(c) For fabrication and installation requirements for stiffening rings, see NE-4437.

NE-3133.6 Cylinders Under Axial Compression. The maximum allowable compressive stress to be used in the design of cylindrical shells and tubular products subjected to loadings that produce longitudinal compressive stresses in the shell shall be the smaller of the following values:

(a) 1.1 times the \( S_m \) value for the applicable material at Design Temperature given in Section II, Part D, Subpart 1, Tables 1A and 1B.

(b) the value of \( B \) determined from the applicable chart in Section II, Part D, Subpart 3, using the following definitions for the symbols on the charts:

\[
R = \text{the inside radius of the cylindrical shell or tubular product, in. (mm)}
\]

\[
T = \text{minimum required thickness of the shell or tubular product, exclusive of the corrosion allowance, in. (mm)}
\]

The value of \( B \) shall be determined from the applicable chart contained in Section II, Part D, Subpart 3 as given in Steps 1 through 5.

Step 1. Using the selected values of \( T \) and \( R \), calculate the value of factor \( A \) using the following equation:

\[
A = \frac{0.125}{(R/T)}
\]

Step 2. Using the value of \( A \) calculated in Step 1, enter the applicable material chart in Section II, Part D, Subpart 3 for the material under consideration. Move vertically to an intersection with material/temperature line for the Design Temperature. Interpolation may be made between lines for intermediate temperatures. In cases where the value at \( A \) falls to the right of the end of the material/temperature line, assume an intersection with the horizontal projection of the upper end of the material/temperature line. For values of \( A \) falling to the left of the material/temperature line, see Step 4.

Step 3. From the intersection obtained in Step 2, move horizontally to the right and read the value of factor \( B \). This is the maximum allowable compressive stress for the values of \( T \) and \( R \) used in Step 1.

Step 4. For values of \( A \) falling to the left of the applicable material/temperature line, the value of \( B \) shall be calculated using the following equation:

\[
B = \frac{AE}{2}
\]

Step 5. Compare the value of \( B \) determined in Step 3 or Step 4 with the computed longitudinal compressive stress in the cylindrical shell or tube, using the selected values of \( T \) and \( R \). If the value of \( B \) is smaller than the computed compressive stress, a greater value of \( T \) must be selected and the design procedure repeated until a value of \( B \) is obtained which is greater than the compressive stress computed for the loading on the cylindrical shell or tube.

NE-3133.7 Conical Heads. The required thickness of a conical head under pressure on the convex side shall not be less than that determined by (a), (b), and (c) below.

(a) When one-half of the included apex angle of the cone is equal to or less than \( 22\frac{1}{2} \) deg, the thickness of the cone shall be the same as the required thickness of a cylindrical shell, the length of which equals the axial length of the cone or the axial distance center-to-center of stiffening rings, if used, and the outside diameter of which is equal to the outside diameter at the large end of the cone or section between stiffening rings.

(b) When one-half of the included apex angle of the cone is greater than \( 22\frac{1}{2} \) deg and not more than 60 deg, the thickness of the cone shall be the same as the required thickness of a cylindrical shell, the outside diameter of which equals the largest inside diameter of the cone measured perpendicular to the cone axis, and the length of which equals an axial length that is the lesser of either the distance center-to-center of stiffening rings, if used, or the largest inside diameter of the section of the cone considered.

(c) When one-half of the included apex angle of the cone is greater than 60 deg, the thickness of the cone shall be the same as the required thickness for a flat head under external pressure, the diameter of which equals the largest inside diameter of the cone (NE-3325).

NE-3134 Material Properties

Values for intermediate temperatures may be found by interpolation.

NE-3134.1 Yield Strength Values. The values of yield strength \( S_m \) shall be those given in Section II, Part D, Subpart 1, Table Y-1.
NE-3134.2  Ultimate Tensile Strength Values. The values of ultimate tensile strength shall be those given in Section II, Part D, Subpart 1, Table U.

NE-3134.3  Coefficients of Thermal Conductivity and Thermal Diffusivity. The values shall be those given in Section II, Part D, Subpart 2, Table TCD.

NE-3134.4  Coefficients of Thermal Expansion. The values of thermal expansion coefficients shall be those given in Section II, Part D, Subpart 2, Table TE.

NE-3134.5  Modulus of Elasticity Values. The values of modulus of elasticity shall be those given in Section II, Part D, Subpart 2, Table TM.

NE-3134.6  Allowable Stress Intensity and Stress Values. The allowable stress intensity $S_{m1}$ shall be the $S_m$ listed in Section II, Part D, Subpart 2, Tables 2A and 2B, and the allowable stress intensity $S_{mc}$ shall be 1.1 times the $S$ listed in Section II, Part D, Subpart 1, Tables 1A and 1B. The basis for establishing allowable stress intensity and stress values is given in Section III Appendices, Mandatory Appendix III.

NE-3135  Attachments

(a) Except as in (c) and (d) below, attachments and connecting welds within the jurisdictional boundary of the containment vessel as defined in NE-1130 shall meet the stress limits of NE-3200 or NE-3300.

(b) The design of the containment vessel shall include consideration of the interaction effects and loads transmitted through the attachment to and from the pressure retaining portion of the vessel. Thermal stresses, stress concentrations, and restraint of the pressure retaining portion of the vessel shall be considered.

(c) Beyond $2t$ from the pressure retaining portion of the containment vessel, where $t$ is the nominal thickness of the pressure retaining material, the appropriate design rules of NF-3000 may be used as a substitute for the design rules of NE-3000 for portions of attachments which are in the vessel support load path.

(d) Nonstructural attachments shall meet the requirements of NE-4435.

NE-3200  DESIGN BY ANALYSIS
NE-3210  DESIGN CRITERIA
NE-3211  General Requirements for Acceptability

The requirements for the acceptability of a design by analysis are given in (a) and (b) below.

(a) The design shall be such that stress intensities do not exceed the applicable limits at temperature described in this Subarticle.

(b) In addition to the requirement in (a) above, the buckling stress shall be considered in accordance with NE-3222.

NE-3212  Basis for Determining Stresses

The theory of failure used in the rules of this Subsection for combining stresses is the maximum shear stress theory. The maximum shear stress at a point is equal to one-half the difference between the algebraically largest and the algebraically smallest of the three principal stresses at the point.

NE-3213  Terms Relating to Stress Analysis

Terms used in this Subsection relating to stress analysis are defined in the following subparagraphs.

NE-3213.1  Stress Intensity. Stress intensity is defined as twice the maximum shear stress, which is the difference between the algebraically largest principal stress and the algebraically smallest principal stress at a given point. Tensile stresses are considered positive and compressive stresses are considered negative.

NE-3213.2  Gross Structural Discontinuity. Gross structural discontinuity is a geometric or material discontinuity that affects the stress or strain distribution through the entire wall thickness of the pressure retaining member. Gross-discontinuity type stresses are those portions of the actual stress distributions that produce net bending and membrane force resultants when integrated through the wall thickness. Examples of a gross structural discontinuity are head-to-shell junctions, flange-to-shell junctions, nozzles, and junctions between shells of different diameters or thicknesses.

NE-3213.3  Local Structural Discontinuity. Local structural discontinuity is a geometric or material discontinuity that affects the stress or strain distribution through a fractional part of the wall thickness. The stress distribution associated with a local discontinuity causes only very localized deformation or strain and has no significant effect on the shell-type discontinuity deformations. Examples are small fillet radii, small attachments, and partial penetration welds.

NE-3213.4  Normal Stress. Normal stress is the component of stress normal to the plane of reference. This is also referred to as direct stress. Usually the distribution of normal stress is not uniform through the thickness of a part, so this stress is considered to have two components, one uniformly distributed and equal to the average stress across the thickness under consideration, and the other varying from this average value across the thickness.

NE-3213.5  Shear Stress. Shear stress is the component of stress tangent to the plane of reference.

NE-3213.6  Membrane Stress. Membrane stress is the component of normal stress that is uniformly distributed and equal to the average stress across the thickness of the section under consideration.

NE-3213.7  Bending Stress. Bending stress is the component of normal stress that varies across the thickness. The variation may or may not be linear.
NE-3213.8 Primary Stress. Primary stress is any normal stress or shear stress developed by an imposed loading that is necessary to satisfy the laws of equilibrium of external and internal forces and moments. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses that considerably exceed the yield strength will result in failure or, at least, in gross distortion. Primary membrane stress is divided into general and local categories. A general primary membrane stress is one that is so distributed in the structure that no redistribution of load occurs as a result of yielding. Examples of primary stress are:

(a) general membrane stress in a circular cylindrical shell or a spherical shell due to internal pressure or to distributed loads;

(b) bending stress in the central portion of a flat head due to pressure.

Refer to Table NE-3217-1 for examples of primary stress.

NE-3213.9 Secondary Stress. Secondary stress is a normal stress or a shear stress developed by the constraint of adjacent material or by self-constraint of the structure. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the conditions that cause the stress to occur, and failure from one application of the stress is not to be expected. Examples of secondary stress are:

(a) general thermal stress [NE-3213.13(a)];

(b) bending stress at a gross structural discontinuity.

Refer to Table NE-3217-1 for examples of secondary stress.

NE-3213.10 Local Primary Membrane Stress. Cases arise in which a membrane stress produced by pressure or other mechanical loading and associated with a discontinuity would, if not limited, produce excessive distortion in the transfer of load to other portions of the structure. Conservatism requires that such a stress be classified as a local primary membrane stress even though it has some characteristics of a secondary stress.

A stressed region may be considered local if the distance over which the membrane stress intensity exceeds $1.15 n_{mc}$ does not extend in the meridional direction more than $1.0 \sqrt{R t}$, where $R$ is the minimum mid-surface radius of curvature and $t$ is the minimum thickness in the region considered. Regions of local primary stress intensity involving axisymmetric membrane stress distributions that exceed $1.15 n_{mc}$ shall not be closer in the meridional direction than $2.5 \sqrt{R (t_1 + t_2)} / 2$ and $t_2$ is defined as $(R_1 + R_2) / 2$ (where $t_1$ and $t_2$ are the minimum thicknesses at each of the regions considered, and $R_1$ and $R_2$ are the minimum mid-surface radii of curvature in these regions where the membrane stress intensity exceeds $1.15 n_{mc}$). Discrete regions of local primary membrane stress intensity, such as those resulting from concentrated loads acting on brackets, where the membrane stress intensity exceeds $1.15 n_{mc}$, shall be spaced so that there is no overlapping of the areas in which the membrane stress intensity exceeds $1.15 n_{mc}$.

NE-3213.11 Peak Stress. Peak stress is that increment of stress that is additive to the primary plus secondary stresses by reason of local discontinuities or local thermal stress [NE-3213.13(b)] including the effects, if any, of stress concentrations. The basic characteristic of a peak stress is that it does not cause any noticeable distortion and is objectionable only as a possible source of a fatigue crack or a brittle fracture. A stress that is not highly localized falls into this category if it is of a type that cannot cause noticeable distortion. Examples of peak stress are:

(a) the thermal stress in the austenitic steel cladding of a carbon steel part;

(b) certain thermal stresses that may cause fatigue but not distortion;

(c) the stress at a local structural discontinuity;

(d) surface stresses produced by thermal shock.

NE-3213.12 Load-Controlled Stress. Load-controlled stress is the stress resulting from application of a loading, such as internal pressure, inertial loads, or gravity, whose magnitude is not reduced as a result of displacement.

NE-3213.13 Thermal Stress. Thermal stress is a self-balancing stress produced by a nonuniform distribution of temperature or by differing thermal coefficients of expansion. Thermal stress is developed in a solid body whenever a volume of material is prevented from assuming the size and shape that it normally would under a change in temperature. For the purpose of establishing allowable stresses, two types of thermal stress are recognized, depending on the volume or area in which distortion takes place, as described in (a) and (b) below.

(a) General thermal stress is associated with distortion of the structure in which it occurs. If a stress of this type, neglecting stress concentrations, exceeds twice the yield strength of the material, the elastic analysis may be invalid, and successive thermal cycles may produce incremental distortion. Therefore, this type is classified as secondary stress in Table NE-3217-1. Examples of general thermal stress are:

(1) stress produced by an axial temperature distribution in a cylindrical shell;

(2) stress produced by the temperature difference between a nozzle and the shell to which it is attached;

(3) the equivalent linear stress produced by the radial temperature distribution in a cylindrical shell.

(b) Local thermal stress is associated with almost complete suppression of the differential expansion and thus produces no significant distortion. Such stresses shall be considered only from the fatigue standpoint and are therefore classified as peak stresses in Table NE-3217-1. In
### Table NE-3221-1
Summary of Stress Intensity Limits

<table>
<thead>
<tr>
<th>Loading Condition</th>
<th>Level C Service Stress Intensity Limit Where the Structure is Integral and Continuous</th>
<th>Level D Service Stress Intensity Limit Where the Structure is Not Integral and Continuous, and at Partial Penetration Welds</th>
<th>Level D Service Stress Intensity Limit Where the Structure is Not Integral and Continuous, and at Elastic Analysis</th>
<th>Level D Service Stress Intensity Limit Where the Structure is Not Integral and Continuous, and at Inelastic Analysis</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_m$</td>
<td>$1.0 S_{m_{eq}}$</td>
<td>$1.0 S_{m_{eq}}$</td>
<td>$1.2 S_{m_{eq}}$ or $1.0 S_{N}$ [Note [1]]</td>
<td>$S_{f}$ [Note [2]]</td>
</tr>
<tr>
<td>$P_L$</td>
<td>$1.5 S_{m_{eq}}$</td>
<td>$1.5 S_{m_{eq}}$</td>
<td>$1.8 S_{m_{eq}}$ or $1.5 S_{N}$ [Note [2]]</td>
<td>$1.5 S_{f}$ [Note [2]]</td>
</tr>
<tr>
<td>$P_1 + P_2$</td>
<td>$1.5 S_{m_{eq}}$</td>
<td>$1.5 S_{m_{eq}}$</td>
<td>$1.8 S_{m_{eq}}$ or $1.5 S_{N}$ [Note [2]]</td>
<td>$1.5 S_{f}$ [Note [2]]</td>
</tr>
<tr>
<td>[Note (3)]</td>
<td>$P_1 + P_2 + Q$ or $P_1 + P_2 + Q + F$</td>
<td>$3.0 S_{m_{eq}}$</td>
<td>$3.0 S_{m_{eq}}$ [Note [5]]</td>
<td>N/A [Note [4]] [Note [5]]</td>
</tr>
<tr>
<td>[Note (4)]</td>
<td>$P_1 + P_2 + Q + F$ or $P_1 + P_2 + Q + F$</td>
<td>$5.0 S_{m_{eq}}$ [Note [5]]</td>
<td>$5.0 S_{m_{eq}}$ [Note [5]]</td>
<td>N/A [Note [4]] [Note [5]]</td>
</tr>
</tbody>
</table>

**NOTES:**

1. Limits identified by \( \ast \) indicate a choice of the larger of two limits.
2. $S_f$ is 85% of the general primary membrane allowable permitted in Section III Appendices, Appendix F. In the application of the rules of Section III Appendices, Appendix F, $S_{m_{eq}}$, if applicable, shall be as specified in Section II, Part D, Subpart 1, Tables 2A and 2B.
3. Values shown are for a solid rectangular section. See NE-3221-3(c) for other than a solid rectangular section.
4. N/A — No evaluation required.
5. Evaluation not required for Level C Service.

### NE-3221.1 General Primary Membrane Stress Intensity.
(Derived from $P_m$ in Figures NE-3221-1, NE-3221-2, NE-3221-3, and NE-3221-4.) This stress intensity is derived from the average value across the thickness of a section of the general primary stresses (NE-3213.8), produced by pressure and other specified mechanical loads but excluding all secondary and peak stresses. Averaging is to be applied to the stress components prior to determination of the stress intensity values. The allowable Design Limit and the allowable for each of the Service Limits is as given below.

- **(a)** Design Limit and Levels A and B Service Limits: $P_m$ shall not exceed 1.0 $S_{m_{eq}}$.
- **(b)** Level C Service Limits
  1. $P_m$ shall not exceed the greater of 1.2 $S_{m_{eq}}$ or 1.0 $S_f$ for regions of the vessel which are integral and continuous.
  2. $P_m$ shall not exceed 1.0 $S_{m_{eq}}$ for regions of the vessel which are not integral and continuous, such as bolted flanges and mechanical joints.
- **(c)** Level D Service Limits
  1. $P_m$ shall not exceed the greater of the following two values for regions of the containment which are integral and continuous.

- **(a)** 85% of the value permitted in Section III Appendices, Nonmandatory Appendix F. In the application of the rules of Section III Appendices, Nonmandatory Appendix F, $S_{m_{eq}}$, if applicable, shall be as specified in Section II, Part D, Subpart 1, Tables 2A and 2B.
- **(b)** The value established in (b)(1) for Service Level C limits.
  1. $P_m$ shall not exceed the greater of 1.25 $m_{eq}$ or 1.0 $S_f$ for regions of the vessel which are not integral and continuous, and at partial penetration welds.

### NE-3221.2 Local Membrane Stress Intensity.
(Derived from $P_L$ in Figures NE-3221-1, NE-3221-2, NE-3221-3, and NE-3221-4.) This stress intensity is derived from the average value across the thickness of a section of the local primary stresses (NE-3213.10) produced by pressure and specified mechanical loads, but excluding all secondary and peak stresses. Averaging is to be applied to the stress components prior to determination of the stress intensity values. The allowable value of this stress intensity is 1.5 times the values given in NE-3221.1, except that the 1.5 factor is not permitted for Level D Service Limits when inelastic component analysis is used as permitted in Section III Appendices, Nonmandatory Appendix F.

### NE-3221.3 Primary General or Local Membrane Plus Primary Bending Stress Intensity.
(Derived from $P_1 + P_2$ in Figures NE-3221-1, NE-3221-2, NE-3221-3, and NE-3221-4.) This stress intensity is derived from the highest value, across the thickness of a section, of the general
Figure NE-3221-3
Stress Categories and Limits of Stress Intensity for Level C Service Limits Where the Structure Is Integral and Continuous; and for Level D Service Limits Where the Structure Is Not Integral and Continuous, and at Partial Penetration Welds

<table>
<thead>
<tr>
<th>Stress Category</th>
<th>General Membrane</th>
<th>Local Membrane</th>
<th>Bending</th>
</tr>
</thead>
<tbody>
<tr>
<td>Description</td>
<td>Average primary stress across solid section. Excludes effects of discontinuities and concentrations.</td>
<td>Average stress across any solid section. Considers effects of discontinuities but not concentrations.</td>
<td>Component of primary stress proportional to distance from centroid of solid section. Excludes effects of discontinuities and concentrations.</td>
</tr>
<tr>
<td>Symbol (Note 1)</td>
<td>$P_m$</td>
<td>$P_L$</td>
<td>$P_b$</td>
</tr>
</tbody>
</table>

Combination of stress components and allowable limits of stress intensities.

Legend
- ALLOWABLE VALUE
- CALCULATED VALUE

![Diagram of stress categories and limits]

NOTES:
1. The symbols $P_m$, $P_L$, and $P_b$ do not represent single quantities, but sets of six representing the six stress components $a_x$, $a_y$, $T_x$, $T_y$, $T_{xy}$, and $r_{xy}$.
2. Values shown are for a solid rectangular section. See NE-3221.3(d) for other than a solid rectangular section.
3. Use greater of values specified.
Figure NE-3221-A
Stress Categories and Limits of Stress Intensity for Level D Service Limits Where the Structure Is Integral and Continuous

<table>
<thead>
<tr>
<th>Stress Category</th>
<th>General Membrane</th>
<th>Primary</th>
<th>Bending</th>
</tr>
</thead>
<tbody>
<tr>
<td>Description</td>
<td>Average primary stress across solid section. Excludes effects of discontinuities and concentrations.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Symbol [Note (3)]</td>
<td>$P_m$</td>
<td>$P_c$</td>
<td>$P_e$</td>
</tr>
<tr>
<td>Combination of stress components and allowable limits of stress intensities.</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Legend**
- Allowable value
- Calculated value

**Notes:**
1. The symbols $P_m$, $P_c$, and $P_e$ do not represent single quantities, but sets of six quantities representing the six stress components $\sigma_x, \sigma_y, \sigma_z, \tau_{xy}, \tau_{xz}, \tau_{yz}$.
2. $S_f$ is 85% of the general membrane allowable permitted in Section III Appendices, Appendix F. In the application of the rules of Section III Appendices, Appendix F, $S_{m1}$, if applicable, shall be as specified in Section II, Part D, Subpart 1, Tables 2A and 2B.
3. Values shown are for a solid rectangular section. See NE-3221.3(d) for other than a solid rectangular section.
4. Elastic analysis
5. Inelastic analysis

**Insert: Nonmandatory**

**Delete lines (merge cells)**
for Level D Service Limits when inelastic component analysis is used as permitted in Section III Appendices, Non-mandatory Appendix F.

**NE-3221.4 Primary Plus Secondary Stress Intensity.** This stress intensity is derived from the highest value, at any point across the thickness of a section, of the general or local primary membrane stresses, plus primary bending stresses plus secondary stresses, produced by the specified conditions. The allowable value of the maximum range of this stress intensity is $3.0S_{m1}$ for Level A and B Service Limits. Secondary stresses need not be evaluated for Design Loadings or for Level C and D Service Limits.

**NE-3221.5 Analysis for Cyclic Operation.**

(a) **Suitability for Cyclic Condition.** The suitability of a vessel or part for specified Service Loadings involving cyclic application of loads and thermal conditions shall be determined by the methods described herein, except that the suitability of high strength bolts shall be determined by the methods of NE-3232.3[b] and the possibility of thermal stress ratcheting shall be investigated in accordance with NE-3221.6. If the specified Service Loadings for the component meet all of the requirements of (d), no analysis for cyclic service is required, and it may be assumed that the limits on peak stress intensities as governed by fatigue have been satisfied by compliance with the applicable requirements for materials, design, fabrication, examination, and testing of this Subsection. If the specified Service Loadings do not meet all of the conditions of (d), a fatigue analysis shall be made in accordance with (e) or a fatigue test shall be made in accordance with Section III Appendices, Mandatory Appendix II, II-1200.

(b) **Peak Stress Intensity.** This stress intensity is derived from the highest value at any point across the thickness of a section of the combination of all primary, secondary, and peak stresses produced by specified service pressures and other mechanical loads, and by general and local thermal effects associated with Service Conditions, and including the effects of gross and local structural discontinuities.

(c) **Conditions and Procedures.** The conditions and procedures of (e) are based on a comparison of peak stress intensities with strain cycling fatigue data. The strain cycling fatigue data are represented by design fatigue strength curves of Section III Appendices, Mandatory Appendix I, Figures I-9.1 through I-9.7. These curves show the allowable amplitude $S_a$ of the alternating stress intensity component (one-half of the alternating stress intensity range) plotted against the number of cycles. This stress intensity amplitude is calculated on the assumption of elastic behavior and, hence, has the dimensions of stress, but it does not represent a real stress when the elastic range is exceeded. The fatigue curves are obtained from uniaxial strain cycling data in which the imposed strains have been multiplied by the elastic modulus and a design margin has been provided so as to make the calculated stress intensity amplitude and the allowable stress intensity amplitude directly comparable. The curves have been adjusted, where necessary, to include the maximum effects of mean stress, which is the condition where the stress fluctuates about a mean value which is different from zero. As a consequence of this procedure, it is essential that the requirements of NE-3221.4 be satisfied at all times with transient stresses included, and that the calculated value of the alternating stress intensity be proportional to the actual strain amplitude. To evaluate the effect of alternating stresses of varying amplitudes, a linear damage relation is assumed in NE-2221(e)(5).

(d) **Vessels Not Requiring Analysis for Cyclic Service.** An analysis for cyclic service is not required, and it may be assumed that the limits on peak stress intensities as governed by fatigue have been satisfied for a vessel by compliance with the applicable requirements for construction in this Subsection, provided the specified Service Loadings of the vessel or portion thereof meet all the conditions stipulated in (1) through (6) below.

(1) **Atmospheric-to-Service Pressure Cycle.** The specified number of times (including startup and shutdown) that the pressure will be cycled from atmospheric pressure to service pressure and back to atmospheric pressure during service does not exceed the number of cycles on the applicable fatigue curve of Section III Appendices, Mandatory Appendix I, Figures I-9.1 through I-9.7 corresponding to an $S_a$ value of 3 times the $S_{m1}$ value for the material at service temperature.

(2) **Normal Service Pressure Fluctuation.** The specified full range of pressure fluctuations during service does not exceed the quantity $\left(\frac{S_a}{3}\times \text{Design Pressure} \times \frac{S_a}{S_{m1}}\right)$ where $S_a$ is the value obtained from the applicable design fatigue curve for the total specified number of significant pressure fluctuations and $S_{m1}$ is the allowable stress intensity for the material at service temperature. If the total specified number of significant pressure fluctuations exceeds the maximum number of cycles defined on the applicable design fatigue curve, the $S_a$ value corresponding to the maximum number of cycles defined on the curve may be used. Significant pressure fluctuations are those for which the total excursion exceeds the quantity: 

$$\text{Design Pressure} \times \frac{1}{3} \times \frac{S_a}{S_{m1}}$$

where $S$ is defined as follows:

- (a) if the total specified number of service cycles is $10^6$ cycles or less, $S$ is the value of $S_a$ obtained from the applicable design fatigue curve for $10^6$ cycles;

- (b) if the total specified number of service cycles exceeds $10^6$ cycles, $S$ is the value of $S_a$ obtained from the applicable design fatigue curve for the maximum number of cycles defined on the curve.

(3) **Temperature Difference — Startup and Shutdown.** The temperature difference in °F (°C) between any two adjacent points of the vessel during service does not exceed $S_a(2Ea)$ where $S_a$ is the value obtained from the applicable design fatigue curves in psi (MPa) for the
(4) Effect of Elastic Modulus. Multiply $S_{alt}$ (as determined in NE-3216.1 or NE-3216.2) by the ratio of the modulus of elasticity given on the design fatigue curve to the value of the modulus of elasticity used in the analysis. Enter the applicable design fatigue curve of Section III Appendices, Mandatory Appendix I, Figures I-9.1 through I-9.7 at this value on the ordinate axis and find the corresponding number of cycles on the abscissa. If the service cycle being considered is the only one which produces significant fluctuating stresses, this is the allowable number of cycles.

(5) Cumulative Damage. If there are two or more types of stress cycles which produce significant stresses, their cumulative effect shall be evaluated as stipulated in Steps 1 through 6 below.

Step 1. Designate the specified number of times each type of stress cycle of types 1, 2, 3, ..., n will be repeated during the life of the component as $n_1, n_2, n_3, ..., n_n$, respectively.

NOTE: In determining $n_1, n_2, n_3, ..., n_n$ consideration shall be given to the superposition of cycles of various origins which produce a total stress difference range greater than the stress difference ranges of the individual cycles. For example, if one type of stress cycle produces 10,000 cycles of a stress difference variation from zero to +20,000 psi and another type of stress cycle produces 10,000 cycles of a stress difference variation from zero to -50,000 psi, the two types of cycle to be considered are defined by the following parameters:

(a) Type 1 cycle: $n_1 = 10,000$
   \[ S_{alt1} = \frac{(60,000 + 30,000)}{2} = 45,000 \text{ psi} \]
(b) Type 2 cycle: $n_2 = 10,000$
   \[ S_{alt2} = \frac{(50,000 + 0)}{2} = 25,000 \text{ psi} \]

Step 2. For each type of stress cycle, determine the alternating stress intensity $S_{alt}$ by the procedures of NE-3216.1 or NE-3216.2 above. Call these quantities $S_{alt1}, S_{alt2}, S_{alt3}, ..., S_{altn}$.

Step 3. For each value $S_{alt1}, S_{alt2}, S_{alt3}, ..., S_{altm}$, use the applicable design fatigue curve to determine the maximum number of repetitions which would be allowable if this type of cycle were the only one acting. Call these values $N_{alt1}, N_{alt2}, N_{alt3}, ..., N_{altm}$.

Step 4. For each type of stress cycle, calculate the usage factors $U_1, U_2, U_3, ..., U_n$ from $U_1 = n_1/N_{alt1}, U_2 = n_2/N_{alt2}, U_3 = n_3/N_{alt3}, ..., U_n = n_n/N_{altm}$.

Step 5. Calculate the cumulative usage factor $U$ from $U = U_1 + U_2 + U_3 + ... + U_n$.

Step 6. The cumulative usage factor $U$ shall not exceed 1.0.

NE-3221.6 Thermal Stress Ratcheting. It should be noted that under certain combinations of steady state and cyclic loading there is a possibility of large distortions developing as the result of ratcheting action; that is, the deformation increases by a nearly equal amount for each cycle. Examples of this phenomenon are treated in this subparagraph and in NE-3227.3.

(a) The limiting value of the maximum cyclic thermal stress permitted in a portion of an axisymmetric shell loaded by steady state internal pressure in order to prevent cyclic growth in diameter is as follows:

$$ x = \text{maximum general membrane stress due to pressure divided by the yield strength}^{16} S_y $$

$$ y' = \text{maximum allowable range of thermal stress computed on an elastic basis divided by the yield strength}^{16} S_y $$

(1) Case 1: Linear variation of temperature through the wall:

$$ y' = \frac{1}{x} \text{ for } 0 < x < 0.5 $$

$$ y' = 4(1 - x) \text{ for } 0.5 < x < 1.0 $$

(2) Case 2: Parabolic constantly increasing or constantly decreasing variation of temperature through the wall:

$$ y' = \frac{5.2(1 - x)}{x} \text{ for } 0.615 < x < 1.0 $$

and approximately for $x < 0.615$ as follows: $y' = 4.65, 3.55, 2.70$ for $x = 0.3, 0.4, 0.5$, respectively.

(b) Use of yield strength $S_y$ in the above relations in stead of the proportional limit allows a small amount of growth during each cycle until strain hardening raises the proportional limit to $S_y$. If the yield strength of the material is higher than two times the $S_y$ value for the maximum number of cycles on the applicable fatigue curve of Section III Appendices, Mandatory Appendix I, Figures I-9.1 through I-9.7 for the material, the latter value shall be used if there is to be a large number of cycles because strain softening may occur.

NE-3221.7 Deformation Limits. Any deformation limits prescribed by the Design Specifications shall be satisfied.

NE-3222 Buckling Stress Values

NE-3222.1 Basic Compressive Allowable Stress. The maximum buckling stress values to be used for the evaluation of instability shall be either of the following:

(a) one-third the value of critical buckling stress determined by one of the methods given below:

(1) rigorous analysis which considers the effects of gross and local buckling, geometric imperfections, nonlinearities, large deformations, and inertial forces (dynamic loads only);

(2) classical (linear) analysis reduced by margins which reflect the difference between theoretical and actual load capacities;

(3) tests of physical models under conditions of restraint and loading the same as those to which the configuration is expected to be subjected;
the elastic equations shall be used, except that the numerical value substituted for Poisson’s ratio shall be determined from the expression:

\[ v = 0.5 - 0.2 \frac{S}{S_0} \text{but not less than 0.3} \]

where:

\( S_0 \) = alternating stress intensity determined in NE-3221.5(e) prior to the elastic modulus adjustment in NE-3221.5(e)(4)
\( S_y \) = the yield strength of the material at the mean value of the temperature of the cycle

**NE-3228 Applications of Plastic Analysis**

The following subparagraphs provide guidance in the application of plastic analysis and some relaxation of the stress limits of NE-3221 which are allowed if plastic analysis is used.

**NE-3228.1 Plastic Analysis.** The limits on local membrane stress intensity (NE-3221.2), primary plus secondary stress intensity (NE-3221.4), thermal stress ratchet in shell (NE-3221.6), and progressive distortion of nonintegral connections (NE-3227.3) need not be satisfied at a specific location if, at the location, the procedures of (a) through (c) below are used.

(a) In evaluating stresses for comparison with the remaining stress limits, the stresses are calculated on an elastic basis.

(b) In lieu of satisfying the specific requirements of NE-3221.2, NE-3221.4, NE-3221.6, and NE-3227.3 at a specific location, the structural action is calculated on a plastic basis and the design shall be considered to be acceptable if shakedown occurs [as opposed to continuing deformation] and if the deformations which occur prior to shakedown do not exceed specified limits.

(c) In evaluating stresses for comparison with fatigue allowables, the numerically maximum principal total strain range which occurs after shakedown shall be multiplied by one-half of the modulus of elasticity of the material (Section II, Part D, Subpart 2, Table TM) at the mean value of the temperature of the cycle.

**NE-3228.2 Limit Analysis.** The limits on local membrane stress intensity (NE-3221.2) and primary membrane plus primary bending stress intensity (NE-3221.3) need not be satisfied at a specific location if it can be shown by means of limit analysis or by tests that the specified loadings do not exceed two-thirds of the lower bound collapse load except for those materials of Section II, Part D, Subpart 1, Table 2A to which Note G7 is applicable and Table 2B to which Note G1 is applicable. For these latter materials the specified loading shall not exceed the product of the applicable permanent strain limiting factor of Section II, Part D, Subpart 1, Table 2-2 times the lower bound collapse load.

**NE-3228.3 Simplified Elastic-Plastic Analysis.** The 35\( m_1 \) limit on the range of primary plus secondary stress intensity (NE-3221.4) may be exceeded provided the requirements of (a) through (f) below are met.

(a) The range of primary plus secondary membrane plus bending stress intensity, excluding thermal bending stresses, shall be \( \leq 35 \text{ m}_1 \).

(b) The value of \( S_0 \) used for entering the design fatigue curve is multiplied by the factor \( K_0 \) where:

\[ K_0 = 1.0, \quad S_0 \leq 35 \text{ m}_1 \]

\[ = 1.0 + [(1 - n) m \text{ m}_1 ] [(S_0 / 35 \text{ m}_1 ) - 1], \quad \text{for} \]

\( 35 \text{ m}_1 < S_0 < 3 m \text{ m}_1 \)

\[ = 1, \quad \text{for} \ S_0 \geq 3 m \text{ m}_1 \]

\( S_0 \) = range of primary plus secondary stress intensity

The values of the material parameters \( m \) and \( n \) are given for the various classes of materials in Table NE-3228.3(b)-1.

(c) The rest of the fatigue evaluation stays the same as required in NE-3221.5, except that the procedure of NE-3227.6 need not be used.

(d) The component meets the thermal ratcheting requirement of NE-3221.6.

(e) The temperature does not exceed those listed in Table NE-3228.3(b)-1 for the various classes of materials.

(f) The material shall have a specified minimum yield strength to specified minimum tensile strength ratio of less than 0.80.

**NE-3228.4 Impulse Loads.** A plastic analysis or test may be used to justify relaxation of the Levels A, B, C, and D Service Limits given in NE-3221.1, NE-3221.2, NE-3221.3, and NE-3221.4 if the applied loading is impulsive in nature. The plastic analysis or test shall demonstrate that the factor against failure under the applied impulsive loading is not less than the factor against failure provided by Level A Service Limits for sustained loads. This demonstration shall be included in the Design Report for review by the Owner or the Owner’s designee for acceptability to the regulatory authority having jurisdiction at the nuclear power plant site.

<table>
<thead>
<tr>
<th>Table NE-3228.3(b)-1</th>
<th>Values of ( m ), ( n ), and ( T_{max} ) for Various Classes of Permitted Materials</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Materials</strong></td>
<td><strong>( m )</strong></td>
</tr>
<tr>
<td>Low alloy steel</td>
<td>2.0</td>
</tr>
<tr>
<td>Martensitic stainless steel</td>
<td>2.0</td>
</tr>
<tr>
<td>Carbon steel</td>
<td>3.0</td>
</tr>
<tr>
<td>Austenitic stainless steel</td>
<td>1.7</td>
</tr>
<tr>
<td>Nickel-chromium-iron</td>
<td>1.7</td>
</tr>
<tr>
<td>Nickel-copper</td>
<td>1.7</td>
</tr>
</tbody>
</table>
NE-3230  STRESS LIMITS FOR BOLTS

NE-3231  Design Conditions

(a) The number and cross-sectional area of bolts required to resist the Design Pressure shall be determined in accordance with the procedures of Appendix XI, using the larger of the bolt loads given by the equations of Appendix XI as a Design Mechanical Load. The allowable bolt design stresses shall be 1.1 times the values given in Section II, Part D, Subpart 1, Table 3 for bolting materials.

(b) When sealing is effected by a seal weld instead of a gasket, the gasket factor \( m \) and the minimum design seating stress \( y \) may be taken as zero.

(c) When gaskets are used for preservice testing only, the design is satisfactory if the above requirements are satisfied for \( m = y = 0 \).

NE-3232  Combined Loads

Actual service stresses in bolts, such as those produced by the combination of preload, pressure, and differential thermal expansion, may be higher than the allowable stresses given in NE-3231(a).

NE-3232.1  Average Stress. The maximum value of service stress, averaged across the bolt cross section and neglecting stress concentrations, shall not exceed 2.2 times the stress values of Section II, Part D, Subpart 1, Table 3.

NE-3232.2  Maximum Stress. The maximum value of service stress, except as restricted by NE-3232.3, at the periphery of the bolt cross section resulting from direct tension plus bending and neglecting stress concentrations shall not exceed 3.3 times the stress values of Section II, Part D, Subpart 1, Table 3. Stress intensity, rather than maximum stress, shall be limited to this value when the bolts are tightened by methods other than heaters, stretchers, or other means which minimize residual torsion.

NE-3232.3  Fatigue Analysis of Bolts. Unless the components on which they are installed meet all the conditions of NE-3221.5(d) and thus require no fatigue analysis, the suitability of bolts for cyclic service shall be determined in accordance with the procedures of (a) through (e) below.

(a) Bolting Having Less Than 100.0 ksi (700 MPa) Tensile Strength. Bolts made of materials which have specified minimum tensile strengths of less than 100.0 ksi (700 MPa) shall be evaluated for cyclic service by the methods of NE-3221.5(e) using the applicable design fatigue curve of Section III Appendices, Mandatory Appendix I, Figure 1-9.4 and an appropriate fatigue strength reduction factor \( [c] \).

(b) High Strength Alloy Steel Bolting. High strength alloy steel bolts and studs may be evaluated for cyclic service by the methods of NE-3221.5(e) using the design fatigue curve of Section III Appendices, Mandatory Appendix I, Figure 1-9.4, provided the requirements of (1), (2), and (3) below are met.

(1) The maximum value of the service stress (NE-3223.2) at the periphery of the bolt cross section (resulting from direct tension plus bending) and neglecting stress concentration shall not exceed \( 2.7S_{m1} \), if the higher of the two fatigue design curves given in Section III Appendices, Mandatory Appendix I, Figure 1-9.4 is used. The \( 2S_{m1} \) limit for direct tension is unchanged.

(2) Threads shall be of a V-type having a minimum thread root radius no smaller than 0.003 in. (0.08 mm).

(3) Fillet radii at the end of the shank shall be such that the ratio of fillet radius to shank diameter is not less than 0.060.

(c) Fatigue Strength Reduction Factor (NE-3213.17). Unless it can be shown by analysis or tests that a lower value is appropriate, the fatigue strength reduction factor used in the fatigue evaluation of threaded members shall not be less than 4.0. However, when applying the rules of (b) for high strength alloy steel bolts, the value used shall not be less than 4.

(d) Effect of Elastic Modulus. Multiply \( S_{a1t} \) (as determined in NE-3216.1 or NE-3216.2) by the ratio of the modulus of elasticity given on the design fatigue curve to the value of the modulus of elasticity used in the analysis. Enter the applicable design fatigue curve at this value on the ordinate axis and find the corresponding number of cycles on the abscissa. If the service cycle being considered is the only one which produces significant fluctuating stresses, this is the allowable number of cycles.

(e) Cumulative Damage. The bolts shall be acceptable for the specified cyclic application of loads and thermal stresses provided the cumulative usage factor \( U \) as determined in NE-3221.5(e)(5) does not exceed 1.0.

NE-3236  Design Stress Values

The design stress intensity values \( S_{oe} \) and allowable stress values \( S \) are 1.1 times the values given in Section II, Part D, Subpart 1, Table 3 for boltin. Values for intermediate temperatures may be found by interpolation. The basis for establishing stress values is given in Section III Appendices, Mandatory Appendix III.

NE-3300  DESIGN BY FORMULA

NE-3310  DESIGN CRITERIA

NE-3311  Requirements for Acceptability

Rules are provided in this Subarticle for Design Loadings and Levels A and B Service Loadings which do not include substantial mechanical or thermal loads other than pressure. The design shall be such that the rules of this Subarticle are satisfied for any configurations and loadings explicitly treated.
When $t$ is known and $P$ is desired:

$$P = S(Z - 1)$$

where

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

**NE-3324.4 Spherical Shells.**

(a) When the thickness of the shell of a spherical vessel does not exceed 0.356$R$, or $P$ does not exceed 0.665$S$, the following equations shall apply. Any reduction in thickness within a shell course of a spherical shell shall be in accordance with **NE-3361**.

$$t = \frac{PR}{2S - 0.2P}$$

or

$$P = \frac{25t}{R + 0.2t}$$

(b) The following equations, in terms of the outside radius, are equivalent to and may be used instead of those given in (a) above:

$$t = \frac{P_R}{2S + 0.8P}$$

(c) When the thickness of the shell of a spherical vessel or of a hemispherical head under internal pressure exceeds 0.356$R$, or when $P$ exceeds 0.665$S$, the following equations shall apply.

When $P$ is known and $t$ is desired:

$$t = R\left(1 - \frac{1}{\sqrt[3]{\frac{P}{S}}} - 1\right) = R_0\left(\frac{\sqrt[3]{\frac{P}{S}} - 1}{\sqrt[3]{\frac{P}{S}}}\right)$$

where

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

When $t$ is known and $P$ is desired:

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

where

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

**NE-3324.5 Formed Heads, General Requirements.**

Formed heads shall meet the requirements of (a) through (f) below.

(a) All formed heads, thicker than the shell and concave to pressure, for butt welded attachment, shall have a skirt length sufficient to meet the requirements of Figure NE-3358.1(a)-1 when a tapered transition is required.

(b) Any taper at a welded joint within a formed head shall be in accordance with **NE-3361**. The taper at a circumferential welded joint connecting a formed head to a main shell shall meet the requirements of **NE-3358** for the respective type of joint shown therein.

(c) All formed heads concave to pressure and for butt welded attachment need not have an integral skirt when the thickness of the head is equal to or less than the thickness of the shell. When a skirt is provided, its thickness shall be at least that required for a seamless shell of the same diameter.

(d) The inside crown radius to which a head is dished shall not be greater than the outside diameter of the skirt of the head. The inside knuckle radius of a torispherical head shall not be less than 6% of the outside diameter of the skirt of the head but in no case less than three times the head thickness.

(e) If a torispherical, ellipsoidal, or hemispherical head is formed with a flattened spot or surface, the diameter of the flat spot shall not exceed that permitted for flat heads as given by eq. NE-3325.2(b)(1) or eq. NE-3325.2(b)(2) using $C = 0.25$.

(f) Openings in formed heads under internal pressure shall comply with the requirements of **NE-3330**.

**NE-3324.6 Ellipsoidal Heads.**

(a) Ellipsoidal Heads. The required thickness of a dished head of semiellipsoidal form, in which one-half of the minor axis, inside depth of the head minus the skirt, equals one-fourth the inside diameter of the head skirt, shall be determined by:

$$t = \frac{PD}{2S - 0.2P}$$

or

Equation for "t" and "p" (on the following page) should be on the same line. See example from 2011 Addenda provided on page 18 of this file.
(b) Ellipsoidal Heads of Other Ratios. The minimum required thickness of an ellipsoidal head of other than a 2:1 ratio shall be determined by:

\[ t = \frac{PD_K}{2S - 0.2P} \]

or

\[ p = \frac{2St}{KD + 0.2t} \]

or Delete

\[ t = \frac{PD_K}{2S + 2P(K - 0.1)} \]

or

\[ p = \frac{2St}{KD_0 - 2t(K - 0.1)} \]

where

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

Numerical values of the factor \( K \) are given in Table NE-3324.2-1

**NE-3324.7 Hemispherical Heads.**

(a) When the thickness of a hemispherical head does not exceed 0.356\( L \), or \( P \) does not exceed 0.665\( S \), the following equations shall apply:

\[ t = \frac{PL}{2S - 0.2P} \]

or

\[ P = \frac{2St}{L + 0.2t} \]

(b) When the thickness of the hemispherical head under internal pressure exceeds 0.356\( L \), or when \( P \) exceeds 0.665\( S \), the following equations shall apply:

\[ t = L\left(\frac{L/3}{s/3} - 1\right) = L_0\left(\frac{L_0/3}{s/3} - 1\right) \]

where

\[ Y = \frac{2(S + P)}{2S - P} \]

or

\[ p = \frac{2S(Y - 1)}{Y + 2} \]

where

\[ Y = \left(\frac{L + t}{L} \right)^3 = \left(\frac{L_0}{L_0 - t} \right)^3 \]

**NE-3324.8 Torispherical Heads.**

(a) Torispherical Heads With a 6% Knuckle Radius. The required thickness of a torispherical head in which the knuckle radius is 6% of the inside crown radius shall be determined by:

\[ t = \frac{0.885PL}{S - 0.1P} \]

or

\[ p = \frac{St}{0.885L + 0.1t} \]

(b) Torispherical Heads of Other Proportions. The required thickness of a torispherical head in which the knuckle radius is other than 6% of the inside crown radius shall be determined by:

\[ t = \frac{PLM}{2S - 0.2P} \]

or

\[ P = \frac{2St}{LM + 0.2t} \]

\[ t = \frac{PL_0M}{2S + P(M - 0.2)} \]
the flat spot shall not exceed that permitted for flat heads as given by eq. (1) or (2) of NE-3325.2 using $C = 0.25$.

(f) Openings in formed heads under internal pressure shall comply with the requirements of NE-3330.

**NE-3324.6 Ellipsoidal Heads**

(a) Ellipsoidal Heads. The required thickness of a dished head of semiellipsoidal form, in which one-half the minor axis, inside depth of the head minus the skirt, equals one-fourth the inside diameter of the head skirt, shall be determined by:

$$ t = \frac{PD}{2S - 0.2P} \quad \text{or} \quad P = \frac{2St}{D + 0.2t} $$

(b) Ellipsoidal Heads of Other Ratios. The minimum required thickness of an ellipsoidal head of other than a 2:1 ratio shall be determined by:

$$ t = \frac{PD_0K}{2S + 2P(K - 0.1)} \quad \text{or} \quad P = \frac{2St}{KD_0 - 2t(K - 0.1)} $$

where

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

Numerical values of the factor $K$ are given in Table NE-3324.2-1.

**NE-3324.7 Hemispherical Heads**

(a) When the thickness of a hemispherical head does not exceed 0.356$L_o$ or $P$ does not exceed 0.665$S$, the following equations shall apply:

$$ t = \frac{PL}{2S - 0.2P} \quad \text{or} \quad P = \frac{2St}{L + 0.2t} $$

(b) When the thickness of the hemispherical head under internal pressure exceeds 0.356$L_o$, or when $P$ exceeds 0.665$S$, the following equations shall apply:

$$ t = L(Y/2 - 1) = L_o \left( \frac{Y/2 - 1}{Y/2} \right) $$

where

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

or

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

where

$$ Y = \left( \frac{L + 1/2}{L} \right)^3 = \left( \frac{L_o}{L_o - t} \right)^3 $$

**NE-3324.8 Torispherical Heads**

(a) Torispherical Heads With a 6% Knuckle Radius. The required thickness of a torispherical head in which the knuckle radius is 6% of the inside crown radius shall be determined by:

$$ t = \frac{0.885PL}{S - 0.1P} \quad \text{or} \quad P = \frac{St}{0.885L + 0.1t} $$

(b) Torispherical Heads of Other Proportions. The required thickness of a torispherical head in which the knuckle radius is other than 6% of the crown radius shall be determined by:

$$ t = \frac{PLM}{2S - 0.2P} \quad \text{or} \quad P = \frac{2St}{LM + 0.2t} $$

$$ t = \frac{PL_0M}{2S + P(M - 0.2)} $$

or

$$ P = \frac{2St}{ML_0 - t(M - 0.2)} $$

where

$$ M = \frac{1}{4} \left( 3 + \sqrt[3]{\frac{L}{t}} \right) $$

Numerical values of the factor $M$ are given in Table NE-3324.8(b)-1.

(c) Torispherical heads made of materials having a specified minimum tensile strength exceeding 80 ksi (550 MPa) shall be designed using a value of $S$ equal to 22 ksi (150 MPa) at room temperature and reduced in proportion to the ratio of the reduction in maximum allowable stress values from room temperature to design temperature for the material as shown in Section II, Part D, Subpart 1, Tables 1A and 1B.

**NE-3324.9 Conical Heads Without Transition Knuckle.** The required thickness of conical heads or conical shell sections that have a half apex angle $\alpha$ not greater than 30 deg shall be determined by:

$$ t = \frac{PD}{2 \cos \alpha(S - 0.6P)} \quad \text{or} \quad P = \frac{2St \cos \alpha}{D + 1.2t \cos \alpha} $$

For $\alpha$ greater than 30 deg, see NE-3324.11(b)(5). A compression ring shall be provided when required by the rule in NE-3324.11(b).

**NE-3324.10 Toriconical Heads.** Toriconical heads in which the inside knuckle radius is neither less than 6% of the outside diameter of the head skirt nor less than three times the knuckle thickness shall be used when the angle $\alpha$ exceeds 30 deg except when the design complies with NE-3324.11. The required thickness of the knuckle shall...
figure the dimensions of the component parts and the dimensions of the welds are exclusive of extra metal required for corrosion allowance.

**NE-3325.1 Nomenclature.** The symbols used are defined as follows:

- \( C \) = a factor depending upon the method of attachment of head, shell dimensions, and other items as listed in NE-3325.3, dimensionless
- \( d \) = diameter, measured as indicated in Figure NE-3325-1
- \( h_o \) = gasket moment arm, equal to the radial distance from the center line of the bolts to the line of the gasket reaction, as shown in Section III Appendices, Mandatory Appendix XI, Table XI-3221.2-2
- \( l \) = length of flange of flanged heads, measured from the tangent line of knuckle, as indicated in Figure NE-3325-1 sketches (a) and (c)
- \( m \) = the ratio \( t_{h}/t_s \)
- \( P \) = Design Pressure, psi (MPa)
- \( r \) = inside corner radius on a head formed by flanging or forging
- \( S \) = maximum allowable stress, psi (MPa), from Section II, Part D, Subpart 1, Tables 1A and 1B, times 1.1
- \( t \) = minimum required thickness of flat head or cover, exclusive of corrosion allowance
- \( t_f \) = actual thickness of the flange on a forged head, at the large end, exclusive of corrosion allowance, as indicated in Figure NE-3325-1 sketches (b-1) and (b-2)
- \( t_h \) = actual thickness of flat head or cover, exclusive of corrosion allowance
- \( t_w \) = the smallest dimension of the face of the head to the edge of the weld preparation
- \( t_{r} \) = required thickness of shell, for pressure
- \( t_s \) = actual thickness of shell, exclusive of corrosion allowance
- \( t_w \) = thickness through the weld joining the edge of the head to the inside of a vessel, as indicated in Figure NE-3325-1 sketch (f)
- \( t_l \) = throat dimension of the closure weld, as indicated in Figure NE-3325-1 sketch (p)
- \( W \) = total bolt load, lb (kN), given for circular heads for Section III Appendices, Mandatory Appendix XI, Article XI-3000, eqs. XI-3223(3) and XI-3223(4)

**NE-3325.2 Thickness.** The thickness of flat unstayed heads, covers, and blind flanges shall conform to one of the following two requirements:

(a) Circular blind flanges of ferrous materials conforming to ASME B16.5 shall be acceptable for the diameters and pressure temperature ratings in Tables 2 to 8 of that Standard, when of the types shown in Figure NE-3325-1 sketches (j) and (k).

(b) The minimum required thickness of flat unstayed circular heads, covers, and blind flanges shall be calculated by the equation:

\[
 t = \frac{d}{2} \sqrt[3]{P/S} 
\]

except when the head, cover, or blind flange is attached by bolts causing an edge moment [Figure NE-3325-1 sketch (n)], in which case the thickness shall be calculated by:

\[
 t = \frac{d}{2} \sqrt[3]{CP/S} + \frac{1.27 WH_o}{Sd^3} 
\]

When using eq. (2), the thickness \( t \) shall be calculated for both Service Loadings and gasket seating, and the greater of the two values shall be used. For Service Loadings, the value of \( P \) shall be the Design Pressure, and the value of \( S \) at the Design Temperature and \( W \) from Section III Appendices, Mandatory Appendix XI, eq. XI-3223(3) shall be used. For gasket seating, \( P \) equals zero, and the value of \( S \) at atmospheric temperature and \( W \) from Section III Appendices, Mandatory Appendix XI, eq. XI-3223(3) shall be used.

**NE-3325.3 Values of C.** For the types of construction shown in Figure NE-3325-1 and Figure NE-4243.1-1, the minimum values of \( C \) to be used in eqs. NE-3325.2(b)(1) and NE-3325.2(b)(2) shall be as given in (a) through (l) below for Figure NE-3325-1 and in (m) below for Figure NE-4243.1-1.

(a) In sketch (a), \( C = 0.17 \) for flanged circular heads forged integral with or butt welded to the vessel with an inside corner radius not less than three times the required head thickness, with no special requirement with regard to length of flange.

\[
 (1) \ C = 0.10 \text{ for circular heads, when the flange length for heads of the above design is not less than:} 
\]

\[
 l = \frac{1.1}{1.1 - 0.616 t_{h}^{0.5}} \frac{d t_{h}}{d} 
\]

(b) In sketch (b-1), \( C = 0.17 \) for forged circular heads integral with or butt welded to the vessel, where the flange thickness is not less than two times the shell thickness, the corner radius on the inside is not less than three times the flange thickness and the weld depth is all the requirements of NE-4000.

\[
 (3) \text{When } C = 0.10 \text{ is used, the taper shall be } 1:4. 
\]

(c) In sketch (b-2), \( C = 0.33 m \) but not less than 0.3 for forged circular heads integral with or butt welded to the vessel, where the flange thickness is not less than the shell thickness and the corner radius on the inside is not less than...
than 1.5 times the flange thickness. [See Figure NE-4243.1-1 sketches (a) and (b) for the special case where \( t_s = t_e \).]

(d) In sketch (c), \( C = 0.13 \) for integral flat circular heads when the dimension \( d \) does not exceed 24 in. (600 mm), the ratio of thickness of the head to the dimension \( d \) is not less than 0.05 nor greater than 0.25, the head thickness \( t_e \) is not less than the shell thickness \( t_s \), the inside corner radius is not less than 0.25\( t_s \), and the construction is obtained by special techniques of upsetting and spinning the end of the shell, such as employed in closing header ends.

(e) In sketches (d), (e), and (f), \( C = 0.33m \) but not less than 0.3 for circular plates welded to the inside of a vessel and otherwise meeting the requirements for the respective types of welded vessels. If a value of \( m \) less than 1 is used in calculating \( t_e \), the shell thickness \( t_s \) shall be maintained along a distance inwardly from the inside face of the head equal to at least \( 2d \sqrt{Ct} \). The throat thickness of the fillet welds in sketch (e) shall be at least 0.7\( t_s \). The size of the weld \( t_w \) in sketch (f) shall not be less than 2 times the required thickness of a seamless shell nor less than 1.25 times the nominal shell thickness but need not be greater than the head thickness; the weld shall be deposited in a welding groove with the root of the weld at the inner face of the head as shown in the figure.

(f) In sketch (g), \( C = 0.33 \) for circular plates welded to the end of the shell when \( t_s \) is at least 1.25\( t_e \), and the weld details conform to the requirements of NE-3358.3(e) and Figure NE-4243-1 sketches (a) through (g).

(g) In sketch (h), \( C = 0.33m \) but not less than 0.3 for circular plates if an inside fillet weld with minimum throat thickness of 0.7\( t_e \) is used and the details of the outside weld conform to the requirements of NE-3358.3(c) and Figure NE-4243-1 sketches (a) through (g), in which the inside weld can be considered to contribute an amount equal to \( t_e \) to the sum of the dimensions \( a \) and \( b \).

(h) In sketches (i) and (j), \( C = 0.02 \) for circular heads and covers bolted to the vessel as indicated in the figures. Note that eq. NE-3325.2(b)(2) shall be used because of the extra moment applied to the cover by the bolting. When the cover plate is grooved for a peripheral gasket as shown in sketch (i), the net cover plate thickness under the groove or between the groove and the outer edge of the cover plate shall not be less than \( d \sqrt{1.27W h_c / S d^3} \) for circular heads and covers.

(i) In sketches (k), (l), and (m), \( C = 0.2 \) for a circular plate inserted into the end of a vessel and held in place by a positive mechanical locking arrangement, and when all possible means of failure (either by shear, tension, compression, or radial deformation, including flaring, resulting from pressure and differential thermal expansion) are resisted using stresses consistent with this Article. See welding may be used, if desired.

(j) In sketch (n), \( C = 0.17 \) for circular covers, bolted with a full-face gasket, to shells, flanges, or side plates.

(k) In sketch (o), \( C = 0.50 \) for circular plates screwed into the end of a vessel having an inside diameter \( d \) not exceeding 12 in. (300 mm), or for heads having an integral flange screwed over the end of a vessel having an inside diameter \( d \) not exceeding 12 in. (300 mm), and when the design of the threaded joint against failure by shear, tension, compression, or radial deformation, including flaring resulting from pressure and differential thermal expansion, is based on stresses consistent with this Article. Seal welding may be used.

(l) In sketch (p), \( C = 0.33 \) for circular plates having a dimension \( d \) not exceeding 18 in. (450 mm) inserted into the vessel as shown and otherwise meeting the requirements for the respective types of welded vessels. The end of the vessel shall be crimped over at least 30 deg but not more than 45 deg. The throat of the weld shall not be less than the thickness of the flat head or shell, whichever is greater.

(m) In Figure NE-4243-1-1, \( C = 0.33m \) but not less than 0.3 when the dimensional requirements of NE-3358.4 are met.

### NE-3326 Spherically Dished Covers With Bolting Flanges

#### NE-3326.1 Nomenclature.

The symbols used are defined as follows:

- \( A \) = outside diameter of flange
- \( B \) = inside diameter of flange
- \( C \) = bolt circle diameter
- \( H_o \) = axial component of the membrane load in the spherical segment, lb (N), acting at the inside of the flange ring = \( 0.785B^2P \)
- \( h_o \) = radial distance from the bolt circle to the inside of the flange ring
- \( h_r \) = radial component of the membrane load in the spherical segment = \( H_o \cot \beta_1 \), lb (N), acting at the intersection of the inside of the flange ring with the center line of the dished cover thickness
- \( h_r \) = lever arm of \( H_o \) about centroid of flange ring
- \( L \) = inside spherical or crown radius
- \( M_o \) = the total moment, in-lb (kNm), determined as in Section III Appendices, Mandatory Appendix XI, XI-3230 for heads concave to pressure, and Section III Appendices, Mandatory Appendix XI, XI-3260 for heads convex to pressure; except that for heads of the type shown in Figure NE-3326.1-1 sketch (d), \( H_o \) and \( h_o \) shall be as defined below and an additional moment \( H_h \) shall be included
- \( P \) = Design Pressure, psi (MPa)
- \( r \) = inside knuckle radius
- \( S \) = maximum allowable stress value, psi (MPa)
- \( T \) = flange thickness
- \( t \) = minimum required thickness of head plate after forming
(1) The opening is for a circular nozzle whose axis is normal to the vessel wall. If the axis of the nozzle makes an angle \( \phi \) with the normal to the vessel wall and if \( d/D \leq 0.15 \), an estimate of the \( \sigma_n \) index on the inside may be obtained from one of the following equations.

For hillside connections in spheres or cylinders:

\[
K_1 = K_1 \left(1 + 2 \sin^2 \phi \right)
\]

For lateral connections in cylinders:

\[
K_2 = K_1 \left[1 + \left( \tan \phi \right)^{2/3} \right]
\]

where

- \( K_1 = \) the \( \sigma_n \) inside stress index of Table NE-3338.2(c)-1 for a radial connection
- \( K_2 = \) the estimated \( \sigma_n \) inside stress index for the nonradial connection

(2) The arc distance measured between the center lines of adjacent nozzles along the inside surface of the shell is not less than three times the sum of their inside radii for openings in a head or along the longitudinal axis of a shell and is not less than two times the sum of their radii for openings along the circumference of a cylindrical shell.

(3) The following dimensional ratios are met:

<table>
<thead>
<tr>
<th>Ratio</th>
<th>Cylinder</th>
<th>Sphere</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inside shell diameter ( = \frac{D}{t} )</td>
<td>10 to 100</td>
<td>10 to 100</td>
</tr>
<tr>
<td>Shell thickness ( = \frac{d}{t} )</td>
<td>0.5, max.</td>
<td>0.5, max.</td>
</tr>
<tr>
<td>Inside nozzle diameter ( = \frac{D}{t} )</td>
<td>0.8, max.</td>
<td>0.8, max.</td>
</tr>
<tr>
<td>Inside shell diameter ( = \frac{d}{D} )</td>
<td>1.5, max.</td>
<td>...</td>
</tr>
</tbody>
</table>

In the case of cylindrical shells, the total nozzle reinforcement area on the transverse axis of the connections, including any outside of the reinforcement limits, shall not exceed 200% of that required for the longitudinal axis (compared to 50% permitted by Figure NE-3332.2-1), unless a tapered transition section is incorporated into the reinforcement and the shell, meeting the requirements of NE-3361.

(4) The inside corner radius \( r_1 \) (Figure NE-3334.2-1) is between 10% and 100% of the shell thickness \( t \).

(5) The outer corner radius \( r_2 \) (Figure NE-3334.2-1) is large enough to provide a smooth transition between the nozzles and the shell. In addition, for opening diameters greater than \( 1\frac{1}{2} \) times the shell thickness in cylindrical shells and 2:1 ellipsoidal heads and greater than three shell thicknesses in spherical shells, the value of \( r_2 \) shall not be less than one-half the thickness of the shell or nozzle wall, whichever is greater.

(6) The radius \( r_3 \) (Figure NE-3334.2-1) is not less than the greater of the following:

- (a) \( 0.0025 d_n \) where \( d_n \) is the outside diameter of the nozzle and is as shown in Figure NE-3334.2-1 and the angle \( \theta \) is expressed in degrees.
- (b) \( 2 \sin \theta \) times offset for the configuration shown in Figure NE-3334.2-1 sketches (a) and (b).

NE-3350 DESIGN OF WELDED CONNECTIONS

NE-3351 Welded Joint Categories

The term Category as used herein defines the location of a joint in a vessel but not the type of joint. The categories established by this paragraph are for specifying special requirements regarding joint type and degree of examination for certain welded pressure joints. Since these special requirements, which are based on service, material, and thickness, do not apply to every welded joint, only those joints to which such requirements apply are included in the categories. The special requirements will apply to joints of a given category only when specifically stated. The joints included in each category are designated as joints of categories A, B, C, and D. Figure NE-3351-1 illustrates typical joint locations included in each category.

<table>
<thead>
<tr>
<th>Category</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Longitudinal welds within the main shell or communicating chambers, transitions in diameter or nozzles; welded joints within a sphere, within a formed or flat head, or within the side plates of a flat-sided vessel; circumferential welded joints connecting hemispherical heads to main shells, to transitions in diameters, to nozzles, or to communicating chambers.</td>
</tr>
<tr>
<td>B</td>
<td>Longitudinal welds within the main shell or communicating chambers, nozzle transitions in diameter, nozzles; or transitions in diameter including joints between the transition and a cylinder at either the large or small end; or circumferential welded joints connecting formed heads other than hemispherical to main shells, transitions in diameter, nozzles, or communicating chambers.</td>
</tr>
<tr>
<td>C</td>
<td>Longitudinal welds connecting flanges, Van Stone laps, tube sheets, or flat heads to the main shell, formed heads, transitions in diameter, nozzles, or communicating chambers, any welded joint connecting one side plate to another side plate of a flat-sided vessel.</td>
</tr>
<tr>
<td>D</td>
<td>Longitudinal welds connecting communicating chambers, or nozzles to main shells, spheres, transitions in diameter, heads, or flat-sided vessels, and those joints connecting nozzles to communicating chambers. For nozzles at the small end of a transition in diameter, see Category B.</td>
</tr>
</tbody>
</table>

77
$t = \text{nominal thickness of part penetrated, in. (mm)}$

t_{min} = \text{the smaller of } \frac{\gamma}{4} \text{ in. (19 mm) or the thickness of the thinner of the parts joined}$

t_0 = 0.7t, \text{ or } \frac{\gamma}{4} \text{ in. (6 mm), whichever is less}$

$t_n = \text{nominal thickness of penetrating part, in. (mm)}$

$t_u = 0.7t_n$

$t_1 + t_2 = \frac{1}{4}t_{min}$

$t_2$ or $t_3$ = not less than the smaller of $\frac{\gamma}{4}$ in. (6 mm) or $0.7t_{min}$

(2) **Welded From One Side**

(a) Partial penetration welds used to connect nozzles from one side are allowed only for attachments on which there are no piping reactions. They shall meet the fabrication requirements of NE-424(d) and shall be capable of being examined in accordance with the requirements of NE-5242.

(b) The minimum dimensions of Figure NE-4244(d)-2 shall be met where:

$c = \text{diametral clearance between nozzle and vessel penetration, in. (mm)}$

$= 0.010 \text{ in. (0.25 mm) for } d \leq 1 \text{ in. (25 mm)}$

$= 0.020 \text{ in. (0.50 mm) for } 1 < d \leq 4 \text{ in. (100 mm)}$ and

$= 0.030 \text{ in. (0.75 mm) for } d > 4 \text{ in. (100 mm)}$ max, except that the above limits on maximum clearance need not be met for the full length of the opening, provided there is a region at the weld preparation and a region near the end of the opening opposite the weld which does meet the above limits on maximum clearance and the latter region is extensive enough (not necessarily continuous) to provide a positive stop for nozzle deflection

$d = \text{outside diameter of nozzle or of the inner cylinder as shown in Figure NE-4244(d)-2, in. (mm)}$

$r_1 = \frac{\gamma}{4}t_n$ or $\frac{\gamma}{4} \text{ in. (19 mm), whichever is less}$

$r_2 = \frac{\gamma}{8} \text{ in. (1.5 mm) min.}$

$r_3 = r_2$ or equivalent chamfer, min.

$r_4 = \frac{\gamma}{4}t_n$ or $\frac{\gamma}{4} \text{ in. (19 mm), whichever is smaller}$

$t = \text{nominal thickness of part penetrated, in. (mm)}$

$t_0 = 0.7t, \text{ or } \frac{\gamma}{4} \text{ in. (6 mm), whichever is less}$

$t_n = \text{nominal thickness of penetrating part [or the lesser of } t_0 \text{ or } t_2 \text{ in Figure NE-4244(d)-2, in. (mm)}$

$\lambda = \frac{\gamma}{8} \text{ in. (1.5 mm) min.}$

$\lambda = t_n \text{ max.}$

(3) The corners to the end of each nozzle extending less than $\sqrt{d_0}$ beyond the inner surface of the part penetrated shall be rounded to a radius of one-half of the thickness $t_n$ of the penetrating part or $\frac{\gamma}{4} \text{ in. (19 mm), whichever is smaller.}$

(4) Weld groove design for partial penetration joints attaching nozzles may require special consideration to achieve the $1.25t_n$ minimum depth of weld and adequate access for welding examination. The welds shown in the sketches of Figure NE-4244(d)-2 and Figure NE-4244(d)-3 may be on either the inside or the outside of the shell. Weld preparation may be j-groove as shown in the figures or straight bevel.

(e) **Attachment of Fittings with Internal Threads.** The attachment of internally threaded fittings shall meet the requirements of (1) through (3) below.

(1) Except as provided for in (2) and (3) below, the provisions of NE-4244(e) shall be met. The minimum weld dimensions shall be as shown in Figure NE-4244(e)-1 where:

$t_n = \frac{\gamma}{4} \text{ in. (6 mm), min.}$

$t_{min} = \text{lesser of } \frac{\gamma}{4} \text{ in. (19 mm) or the thickness of the parts joined}$

$t_1 + t_2 = \frac{1}{4}t_{min}$

Details in sketches (a) and (b):

$\lambda_1 = \text{thickness of Sch. 160 pipe (ASME B36.10), in. (mm)}$

$t_1 + t_2 = \text{not less than smaller of } \frac{\gamma}{4} \text{ in. (6 mm) or } 0.7t_{min}$

Details in sketch (c):

$\lambda = t_{min}$

Details in sketch (d):

$c = \frac{\gamma}{4}t_{min}$

$t_u = 0.7t_{min}$

(2) Fittings shown in Figure NE-4244(e)-1 sketches (a-2), (b-2), (c-2), and (d) not exceeding NPS 2 (DN 50) may be attached by welds that are exempt from size requirements other than those specified in NE-3359.

(3) **Openings**

(a) When internally threaded fittings and bolting pads not exceeding NPS 3 (DN 80) are attached to vessels having a wall thickness not greater than $\frac{\gamma}{4} \text{ in. (10 mm) by a fillet weld depoted from the outside only, the welds shall comply with the dimensions shown in Figure NE-4244(e)-2. These openings do not require reinforcement other than that inherent in the construction as permitted in NE-3332.1.}$

(b) If the opening exceeds NPS 5 (DN 125), it shall be reinforced in accordance with NE-3332 with the nozzle or other connections attached, using a suitable detail in Figure NE-4244(e)-1.

(f) **Attachment of Tubed Connections.** Tubes recessed into thick walled vessels or headers, welded from only one side, shall have a welding groove in the vessel wall not deeper than $t_n$ on the longitudinal axis of the opening. A recess $\frac{\gamma}{4} \text{ in. (1.5 mm) deep shall be provided at the bottom of the groove in which to center the nozzle. The dimension } t_u \text{ of the attachment weld shall not be less than } t_n \text{ nor less than } \frac{\gamma}{4} \text{ in. (6 mm). The minimum dimension for } t_2 \text{ shall be } \frac{\gamma}{4} \text{ in. (6 mm) [Figure NE-4244(f)-1 sketches (a) and (b)].}$

**NE-3355 Welding Grooves**

The dimensions and shape of the edges to be joined shall be such as to permit complete fusion and penetration, except as otherwise permitted in NE-3352.4.
(e) Attachment of Fittings with Internal Threads. The attachment of internally threaded fittings shall meet the requirements of (1) through (3) below.

(1) Except as provided for in (2) and (3) below, the provisions of NE-4244(e) shall be met. The minimum weld dimensions shall be as shown in Fig. NE-4244(e)-1 where:

\[ t_w = \frac{1}{4} \text{ in. (6 mm), min.} \]
\[ t_{\text{min}} = \text{lesser of } \frac{3}{4} \text{ in. (19 mm) or the thickness of the part joined} \]

Details in sketches (a) and (b):

\[ t_1 + t_2 = 1\frac{1}{4} t_{\text{min}} \]

Detail in sketch (c):

\[ t_{\text{w}} = \text{thickness of Sch. 160 pipe (ASME B36.10), in. (mm)} \]
\[ t_1 + t_2 = \text{not less than smaller of } \frac{1}{4} \text{ in. (6 mm)} \text{ or } 0.7t_{\text{min}} \]

Detail in sketch (d):

\[ c = \frac{1}{2}t_{\text{min}} \]
\[ t_{\text{w}} = 0.7t_{\text{min}} \]

(2) Fittings shown in Fig. NE-4244(e)-1 sketches (a-2), (b-2), (c-2), and (d) not exceeding NPS 2 (DN 50) may be attached by welds that are exempt from size requirements other than those specified in NE-3359.

(3) Openings

(a) When internally threaded fittings and bolting pads not exceeding NPS 3 (DN 80) are attached to vessels having a wall thickness not greater than \( \frac{3}{8} \) in. (10 mm) by a fillet weld deposited from the outside only, the welds shall comply with the dimensions shown in Fig. NE-4244(e)-2. These openings do not require reinforcement other than that inherent in the construction as permitted in NE-3332.1.

(b) If the opening exceeds NPS 5 (DN 125), it shall be reinforced in accordance with NE-3332 with the nozzle or other connections attached, using a suitable detail in Fig. NE-4244(e)-1.

(f) Attachment of Tubed Connections. Tubes recessed into thick walled vessels or headers, welded from only one side, shall have a welding groove in the vessel wall not deeper than \( t_4 \) on the longitudinal axis of the opening. A recess \( \frac{1}{16} \) in. (1.5 mm) deep shall be provided at the bottom of the groove in which to center the nozzle. The dimension \( t_4 \) of the attachment weld shall not be less than \( t_4 \) nor less than \( \frac{1}{4} \) in. (6 mm). The minimum dimension for \( t_4 \) shall be \( \frac{1}{4} \) in. (6 mm) [Fig. NE-4244(f)-1 sketches (a) and (b)].

NE-3355 Welding Grooves

The dimensions and shape of the edges to be joined shall be such as to permit complete fusion and penetration, except as otherwise permitted in NE-3352.4.

NE-3356 Fillet Welds

(a) Corner or T-joints for Category C welds may be made with fillet welds, provided the attachment is properly supported independently of such welds. Closures which meet the requirements of NE-3367 may be attached by fillet welds.

(b) The allowable load on fillet welds shall equal the product of the weld area based on minimum leg dimension, the allowable stress value in tension of the material being welded, and the factor 0.55.

(c) Except where specific details are permitted in other paragraphs, welded joints subject to bending stresses shall have fillet welds added where necessary to reduce stress concentration. Corner joints, with fillet welds only, shall not be used unless the plates forming the corner are properly supported independently of such welds.

NE-3358 Design Requirements for Head Attachments

NE-3358.1 Skirt Length of Formed Heads

(a) Ellipsoidal and other types of formed heads, concave or convex to the pressure, shall have a skirt length not less than that shown in Fig. NE-3358.1(a)-1. Heads that are fitted inside or over a shell shall have a driving fit before welding.

(b) A tapered transition, having a length not less than three times the offset between the adjacent surfaces of abutting sections as shown in Fig. NE-3358.1(a)-1, shall be provided at joints between formed heads and shells that differ in thickness by more than one-fourth the thickness of the thinner section or by more than \( \frac{1}{8} \) in. (3 mm), whichever is less. When a taper is required on any formed head thicker than the shell and intended for butt welded attachment [Fig. NE-3358.1(a)-1], the skirt shall be long enough so that the required length of taper does not extend beyond the tangent line.

NE-3358.2 Unstayed Flat Heads Welded to Shells

The requirements for the attachment of unstayed flat heads welded to shells are given in NE-3325, NE-3358.3, and NE-3358.4.

NE-3358.3 Head Attachments Using Corner Joints

When shells, heads, or other pressure parts are welded to a forged or rolled plate to form a corner joint, as in Figs. NE-4243-1 and NE-4243-2, the joint shall meet the requirements of (a) through (e) below.

(a) On the cross section through the welded joint, the line of fusion between the weld metal and the forged or
NE-3356 Fillet Welds

(a) Corner or T-joints for Category C welds may be made with fillet welds, provided the attachment is properly supported independently of such welds. Closures which meet the requirements of NE-3357 may be attached by fillet welds.

(b) The allowable load on fillet welds shall equal the product of the weld area based on minimum leg dimension, the allowable stress value in tension of the material being welded, and the factor 0.55.

(c) Except where specific details are permitted in other paragraphs, welded joints subject to bending stresses shall have fillet welds added where necessary to reduce stress concentration. Corner joints, with fillet welds only, shall not be used unless the plates forming the corner are properly supported independently of such welds.

NE-3358 Design Requirements for Head Attachments

NE-3358.1 Skirt Length of Formed Heads.

(a) Ellipsoidal and other types of formed heads, concave or convex to the pressure, shall have a skirt length not less than that shown in Figure NE-3358.1(a)-1. Heads that are fitted inside or over a shell shall have a driving fit before welding.

(b) A tapered transition, having a length not less than three times the offset between the adjacent surfaces of abutting sections as shown in Figure NE-3358.1(a)-1, shall be provided at joints between formed heads and shells that differ in thickness by more than one-fourth the thickness of the thinner section or by more than \( \frac{3}{16} \) in. (3 mm), whichever is less. When a taper is required on any formed head thicker than the shell and intended for butt welded attachment [Figure NE-3358.1(a)-1], the skirt shall be long enough so that the required length of taper does not extend beyond the tangent line.

NE-3358.2 Unstayed Flat Heads Welded to Shells.
The requirements for the attachment of unstayed flat heads welded to shells are given in NE-3325, NE-3358.3, and NE-3358.4.

NE-3358.3 Head Attachments Using Corner Joints.

When shells, heads, or other pressure parts are welded to a forged or rolled plate to form a corner joint, as in Figures NE-4243-1 and NE-4243-2, the joint shall meet the requirements of (a) through (e) below.

(a) On the cross section through the welded joint, the line of fusion between the weld metal and the forged or rolled plate being attached shall be projected on planes both parallel to and perpendicular to the surface of the plate being attached, in order to determine the dimensions \( a \) and \( b \), respectively.

(b) For flange rings of bolted flanged connections, and for flat heads with a projection having holes for a bolted connection, the sum of \( a \) and \( b \) shall not be less than three times the nominal wall thickness of the abutting pressure part.

(c) For other components, the sum of \( a \) and \( b \) shall not be less than two times the nominal wall thickness of the abutting pressure part. Examples of such components are flat heads without a projection having holes for a bolted connection and the side plates of a rectangular vessel.

(d) Other dimensions at the joint shall be in accordance with details as shown in Figures NE-4243-1 and NE-4243-2 where:

In Figure NE-4243-1:

\[
\begin{align*}
\text{sketch (a)} & \quad a + b \text{ not less than } 2t_c \\
& \quad b = 0 \\
& \quad t_w \text{ not less than } t_c \\
& \quad t_c = \text{actual thickness of shell, in. (mm)} \\
\text{sketch (b)} & \quad a + b \text{ not less than } 2t_c \\
& \quad t_w \text{ not less than } t_c \\
& \quad t_c \text{ not less than } t_c \\
& \quad t_c = \text{actual thickness of shell} \\
\text{sketch (c)} & \quad a + b \text{ not less than } 2t_c \\
& \quad a \text{ not less than } t_c \\
& \quad t_w \text{ not less than } t_c \\
& \quad t_c = \text{actual thickness of shell} \\
\text{sketch (d)} & \quad a + b \text{ not less than } 2t_c \\
& \quad a \text{ not less than } t_c \\
& \quad t_w \text{ not less than } t_c \\
& \quad t_c = \text{actual thickness of shell} \\
\text{sketch (e)} & \quad a + b \text{ not less than } 2t_c \\
& \quad b = 0 \\
& \quad t_c = \text{actual thickness of shell} \\
\text{sketch (f)} & \quad a + b \text{ not less than } 2t_c \\
& \quad a \text{ not less than } 0.5t_c \text{ nor greater than } 2t_c \\
& \quad a + a_c = a \\
& \quad t_c = \text{actual thickness of shell} \\
\text{sketch (g)} & \quad a + b \text{ not less than } 2t_c \\
& \quad a \text{ not less than } 0.5t_c \text{ nor greater than } 2t_c \\
& \quad a + a_c = a \\
& \quad t_c = \text{actual thickness of shell} \\
\text{sketch (h)} & \quad a \text{ not less than } t_c \\
& \quad (b = 0) \\
& \quad c \text{ not less than } t_e \text{ or } t_{o,23} \text{ whichever is less} \\
& \quad a \text{ not less than } 0.75t_c \\
& \quad c \text{ not less than } t_e \text{ or } t_{o,23} \text{ whichever is less}
\end{align*}
\]

In Figure NE-4243-2:

For forged tubeshells, forged flat heads, and forged flanges with the weld preparation bevel angle not greater than 45 deg measured from the face:

\[
t_e = \text{nominal thickness of welded parts, in. (mm)} \\
t_e = 0.7t_e \text{ or } \frac{1}{4} \text{ in. (6 mm), whichever is less} \\
t_c = \text{the lesser of } t_{e,2}/2 \text{ or } t/4
\]
For all other material forms and for forged tubesheets, forged flat heads, and forged flanges with the weld preparation bevel angle greater than 45 deg measured from the face:

\[ t_s = \text{nominal thicknesses of welded parts, in. (mm)} \]
\[ t_n = 0.7 t_s \text{, or } \frac{1}{4} \text{ in. (6 mm), whichever is less} \]
\[ t_e = \text{the lesser of } t_s \text{ or } t/2 \]

(a) Joint details that have a dimension through the joint less than the thickness of the shell, head, or other pressure part, or that provide attachment eccentric thereto, are not permissible [Figure NE-4243-1 sketches (1), (k), and (l)].

**NE-3358.4 Flat Heads With Hubs.** Hubs for butt welding as shown in Figure NE-4243-1-1 shall have the following minimum dimensions:

<table>
<thead>
<tr>
<th>Sketch</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a)</td>
<td>r not less than 1.5t</td>
</tr>
<tr>
<td>(b)</td>
<td>e not less than t</td>
</tr>
<tr>
<td>(c)</td>
<td>h not less than 1.5t</td>
</tr>
<tr>
<td>(d)</td>
<td>t, not less than 2t</td>
</tr>
<tr>
<td>(e)</td>
<td>r not less than 3t</td>
</tr>
</tbody>
</table>

**NE-3359 Design Requirements for Nozzle Attachment Welds**

In addition to the requirements of NE-3352.4, the minimum design requirements for nozzle attachment welds shall be as stipulated in (a) and (b) below.

(a) **Required Weld Strength.** Sufficient welding shall be provided on either side of the line through the center of the opening parallel to the longitudinal axis of the shell to develop the strength of the reinforcing parts as prescribed in NE-3356, through shear or tension in the weld, whichever is applicable. The strength of groove welds shall be based on the area subjected to shear or to tension. The strength of fillet welds shall be based on the area subjected to shear computed on the minimum leg dimension. The inside diameter of a fillet weld shall be used in figuring its length. Calculations are not required for full penetration welds.

(b) **Allowable Stress Values for Welds.** The allowable stress values for groove and fillet welds and for shear in nozzles, in percentage of stress values for the vessel material, are as follows:

1. Nozzle wall shear, 70%
2. Groove weld tension, 74%
3. Groove weld shear, 60%
4. Fillet weld shear, 49%

**NE-3360 SPECIAL VESSEL REQUIREMENTS**

**NE-3361 Tapered Transitions**

A tapered transition having a length not less than three times the offset between the adjacent surfaces of abutting sections (Figure NE-3361-1) shall be provided at joints between sections that differ in thickness by more than one-fourth the thickness of the thinner section or by more than \( \frac{1}{6} \) in. (3 mm), whichever is less. The transition may be formed by any process that will provide a uniform taper. The weld may be partly or entirely in the tapered section or adjacent to it.

**NE-3362 Bolted Flange and Studded Connections**

(a) It is recommended that the dimensional requirements of bolted flange connections to external piping conform to ASME B16.5, Steel Pipe Flanges and Flanged Fittings; ASME B16.24, Cast Copper Alloy Pipe Flanges and Flanged Fittings; or to ASME B16.47, Large Diameter Steel Flanges. Such flanges and flanged fittings may be used for the pressure-temperature ratings given in the appropriate standard. Flanges and flanged fittings in accordance with other standards are acceptable, provided they have been designed in accordance with the rules of Section III Appendices, Mandatory Appendix XI for the vessel Design Loads and are used within the pressure-temperature ratings so determined.

(b) Where tapped holes are provided for studs, the threads shall be full and clean, and shall engage the stud for a length not less than the larger of \( d_s \) or:

\[ \frac{0.75d_s}{2} \times \frac{\text{maximum allowable stress value}}{\text{of stud material at Design Temperature}} \]

\[ \frac{\text{maximum allowable stress value}}{\text{of tapped material}} \]

\[ \text{at Design Temperature} \]

in which \( d_s \) is the diameter of the stud. The thread engagement need not exceed \( 1\frac{1}{2}d_s \).

**NE-3363 Access Openings**

Access openings, where provided, shall consist of handhole or manhole openings having removable covers. These may be located on either the inside or outside of the shell or head openings and may be attached by studs or bolts in combination with gaskets or welded membrane seals or strength welds. Plugs using pipe threads are not permitted.

**NE-3364 Attachments**

Attachments used to transmit support loads shall meet the requirements of NE-3135.

**NE-3365 Supports**

All vessels shall be so supported and the supporting members shall be so arranged and attached to the vessel wall as to provide for the maximum imposed loads.