(3) Design temperature is no warmer than 650°F (345°C) nor colder than -20°F (-29°C). Occasional operating temperatures colder than -20°F (-29°C) are acceptable when due to lower seasonal atmospheric temperature.

(4) The thermal or mechanical shock loadings are not a controlling design requirement. (See UG-22.)

(5) Cyclic loading is not a controlling design requirement. (See UG-22.)

**UG-21 DESIGN PRESSURE**

Each element of a pressure vessel shall be designed for at least the most severe condition of coincident pressure (including coincident static head in the operating position) and temperature expected in normal operation. For this condition, the maximum difference in pressure between the inside and outside of a vessel, or between any two chambers of a combination unit, shall be considered [see UG-98 and 3-2]. See also U-2(a).

**UG-22 LOADINGS**

The loadings to be considered in designing a vessel shall include those from:

(a) internal or external design pressure (as defined in UG-21);

(b) weight of the vessel and normal contents under operating or test conditions;

(c) superimposed static reactions from weight of attached equipment, such as motors, machinery, other vessels, piping, linings, and insulation;

(d) the attachment of:

(1) internals (see Nonmandatory Appendix D);

(2) vessel supports, such as lugs, rings, skirts, saddles, and legs (see Nonmandatory Appendix C);

(e) cyclic and dynamic reactions due to pressure or thermal variations, or from equipment mounted on a vessel, and mechanical loadings;

(f) wind, snow, and seismic reactions, where required;

(g) impact reactions such as those due to fluid shock;

(h) temperature gradients and differential thermal expansion;

(i) abnormal pressures, such as those caused by deflagration;

(j) test pressure and coincident static head acting during the test (see UG-99).

**UG-23 MAXIMUM ALLOWABLE STRESS VALUES**

(a) The maximum allowable stress value is the maximum unit stress permitted in a given material used in a vessel constructed under these rules. The maximum allowable tensile stress values permitted for different materials are given in Section II, Part D, Subpart 1. Section II, Part D is published as two separate publications. One publication contains values only in the U.S. Customary units and the other contains values only in SI units. The selection of the version to use is dependent on the set of units selected for construction. A listing of these materials is given in the following tables, which are included in Subsection C. For material identified as meeting more than one material specification and/or grade, the maximum allowable tensile stress value for either material specification and/or grade may be used provided all requirements and limitations for the material specification and grade are met for the maximum allowable tensile stress value chosen.

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</table>

(b) The maximum allowable longitudinal compressive stress to be used in the design of cylindrical shells or tubes, either seamless or butt welded, subjected to loadings that produce longitudinal compression in the shell or tube shall be the smaller of the following values:

(1) the maximum allowable tensile stress value permitted in (a) above;

(2) the value of the factor B determined by the following procedure where

\[ E = \text{modulus of elasticity of material at design temperature} \]

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.)

\[ R_o = \text{outside radius of cylindrical shell or tube} \]

\[ t = \text{the minimum required thickness of the cylindrical shell or tube} \]

The joint efficiency for butt-welded joints shall be taken as unity.

The value of B shall be determined as follows.
Step 1. Using the selected values of \( t \) and \( R \), calculate the value of factor \( A \) using the following equation:

\[
A = \frac{0.125}{(R_0/t)}
\]

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 (see UG-20). Interpolation may be made between lines for intermediate temperatures. If tabular values in Section II, Part D, Subpart 3 are used, linear interpolation or any other rational interpolation method may be used to determine a \( B \) value that lies between two adjacent tabular values for a specific temperature. Such interpolation may also be used to determine a \( B \) value at an intermediate temperature that lies between two sets of tabular values, after first determining \( B \) values for each set of tabular values.

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. If tabular values are used, the last (maximum) tabulated value shall be used. 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 \( R_0 \) 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:

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

(f) Values for yield strength, \( S_y \), as a function of temperature are provided in Section II, Part D, Subpart 1, Table Y-1. If the material being used is not listed in this table, while being listed in other tables of Section II Part D Subpart 1, or the specified temperature exceeds the highest temperature for which a value is provided, the yield strength may be determined as below for use in the design equations in this division. \( S \) is the maximum allowable stress at the temperature specified [see (a) above] and \( f \) is the factor (e.g. weld factor) used to determine the allowable stress as indicated in the notes for the stress line. If the value of \( f \) is not provided, set \( f \) equal to 1.

1. (1) If allowable stress is established based on the 66 2/3% yield criterion, then yield strength, \( S_y \), shall be taken as 1.5S/f.

2. (2) If the allowable stress is established based on yield criterion between 66 2/3% and 90%, then the yield strength, \( S_y \), shall be taken as 1.1S/f.

NOTE For temperatures where the allowable stress, \( S \), is based on time dependent properties, the yield strength obtained by these formulas may be overly conservative.

are expected to occur simultaneously during normal operation\(^{13}\) of the vessel, the induced maximum general primary membrane stress does not exceed the maximum rules, such as those for cast iron in flanged joints, the above loads shall not induce a combined maximum primary membrane stress plus primary bending stress across the thickness that exceeds \( 1\frac{1}{4} \times \) the maximum allowable stress value in tension (see UG-23). It is recognized that high localized discontinuity stresses may exist in vessels designed and fabricated in accordance with these rules. Insofar as practical, design rules for details have been written to limit such stresses to a safe level consistent with experience.

The maximum allowable stress values that are to be used in the thickness calculations are to be taken from the tables at the temperature that is expected to be maintained in the metal under the conditions of loading being considered. Maximum stress values may be interpolated for intermediate temperatures.

(d) For the combination of earthquake loading, or wind loading with other loadings in UG-22, the wall thickness of a vessel computed by these rules shall be determined such that the general primary membrane stress shall not exceed 1.2 times the maximum allowable stress permitted in (a), (b), or (c) above. This rule is applicable to stresses caused by internal pressure, external pressure, and axial compressive load on a cylinder.

Earthquake loading and wind loading need not be considered to act simultaneously.

(e) Localized discontinuity stresses [see (c) above] are calculated in Mandatory Appendix 1, 1-5(g) and 1-8(e), Part UHX, and Mandatory Appendix 5. The primary plus secondary stresses\(^{14}\) at these discontinuities shall be limited to \( S_{ps} \) where \( S_{ps} = 3S \), and \( S \) is the maximum allowable stress of the material at temperature [see (a) above].

In lieu of using \( S_{ps} = 3S \), a value of \( S_{ps} = 2S \) may be used, where \( S \) is the yield strength at temperature, provided the following are met:

1. (1) the allowable stress of material \( S \) is not governed by time-dependent properties as provided in Section II, Part D, Subpart 1, Table 1A or Table 1B;

2. (2) the room temperature ratio of the specified minimum yield strength to specified minimum tensile strength for the material does not exceed 0.7;

3. (3) the value for \( S \) at temperature can be obtained from Section II, Part D, Subpart 1, Table Y-1.

(f) Maximum shear stress in restricted shear, such as dowel bolts or similar construction in which the shearing member is so restricted that the section under consideration would fail without a reduction of area, shall be limited to 0.80 times the values in Section II, Part D, Subpart 1, Table 1A, Table 1B, or Table 3.

(g) Maximum bearing stress shall be limited to 1.60 times the values in Section II, Part D, Subpart 1, Table 1A, Table 1B, or Table 3.

UG-24 CASTINGS

(a) Quality Factors. A casting quality factor as specified below shall be applied to the allowable stress values for cast materials given in Subsection C except for castings...
(e) Configuration e: tubesheet gasketed with shell and integral with channel, extended as a flange;
(f) Configuration f: tubesheet gasketed with shell and integral with channel, not extended as a flange.

**UHX-12.2 Conditions of Applicability**

The general conditions of applicability given in UHX-10 apply.

**UHX-12.3 Nomenclature**

The symbols described below are used for the design of the tubesheet. Symbols $D_o$, $E^*$, $h'$, $\mu$, $\mu^*$, and $v^*$ are defined in UHX-11.

- $A$ = outside diameter of tubesheet, except as limited by UHX-10(b)
- $A_p$ = total area enclosed by $C_p$
- $C$ = bolt circle diameter (see Mandatory Appendix 2)
- $C_p$ = perimeter of the tube layout measured stepwise in increments of one tube pitch from the center-to-center of the outermost tubes (see Figure UHX-12.2)
- $D_c$ = inside channel diameter
- $D_s$ = inside shell diameter
- $E$ = modulus of elasticity for tubesheet material at design temperature
- $E_c$ = modulus of elasticity for channel material at design temperature
- $E_s$ = modulus of elasticity for shell material at design temperature
- $G_1$ = midpoint of contact between flange and tubesheet
- $G_c$ = diameter of channel gasket load reaction (see Mandatory Appendix 2)
- $G_s$ = diameter of shell gasket load reaction (see Mandatory Appendix 2)
- $h$ = tubesheet thickness

$\text{MAX} \{a, b, c, \ldots\}$ = greatest of $a$, $b$, $c$, ...

- $P_s$ = shell side design pressure. For shell side vacuum, use a negative value for $P_s$
- $P_{sd,\text{max}}$ = maximum shell side design pressure
- $P_{sd,\text{min}}$ = minimum shell side design pressure (negative if vacuum is specified, otherwise zero)
- $P_t$ = tube side design pressure. For tube side vacuum, use a negative value for $P_t$
- $P_{td,\text{max}}$ = maximum tube side design pressure
- $P_{td,\text{min}}$ = minimum tube side design pressure (negative if vacuum is specified, otherwise zero)

$S$ = allowable stress for tubesheet material at tubesheet design temperature (see UG-23)

- $S_c$ = allowable stress for channel material at design temperature
- $S_s$ = allowable stress for shell material at design temperature

**NOTE:** For a welded tube or pipe, use the allowable stress for the equivalent seamless product. When the allowable stress for the equivalent seamless product is not available, divide the allowable stress of the welded product by 0.85.

- $S_y = \text{yield strength from UG-23(f) for tubesheet material at tubesheet design temperature (T)}$
- $S_{y,c} = \text{yield strength from UG-23(f) for channel material at channel design temperature (T_0)}$
- $S_{y,s} = \text{yield strength from UG-23(f) for shell material at shell design temperature (T_0)}$

**NOTE:** The yield strength shall be taken from Section II, Part B, Subpart I, Table Y-1. When a yield strength value is not listed in Table Y-1, one may be obtained by using the procedure in UG-25(a)(2), Step 2.

- $t_c$ = channel thickness
- $t_s$ = shell thickness
- $W^*$ = tubesheet effective bolt load determined in accordance with UHX-8
- $v_c$ = Poisson’s ratio of channel material
- $v_s$ = Poisson’s ratio of shell material
Configurations e and f: This option may only be used when \( \sigma_c \leq S_{PS,c} \). In UHX-12.5.4, replace \( E_t \) with \( E_t' = E_t \sqrt{1.5S_{c}/\sigma_c} \) and recalculate \( k_E \) and \( \lambda_c \).

Configurations a, b, c, e, and f: Perform the steps in UHX-12.5.5 and UHX-12.5.7, and recalculate the tubesheet bending stress \( \sigma \) given in UHX-12.5.8.

If \( \sigma \leq 2\sigma \), the assumed tubesheet thickness \( h \) is acceptable and the design is complete. Otherwise, the design shall be reconsidered by using Option 1 or 2.

### UHX-12.6 Calculation Procedure for Simply Supported U-Tube Tubesheets

#### UHX-12.6.1 Scope
This procedure describes how to use the rules of UHX-12.5 when the effect of the stiffness of the integral channel and/or shell is not considered.

#### UHX-12.6.2 Conditions of Applicability
This calculation procedure applies only when the tubesheet is integral with the shell or channel (configurations a, b, c, e, and f).

#### UHX-12.6.3 Calculation Procedure
The calculation procedure outlined in UHX-12.5 shall be performed accounting for the following modifications:

(a) Perform the steps in UHX-12.5.1 through UHX-12.5.9.

(b) Perform the step in UHX-12.5.10 except as follows:

1. The shell (configurations a, b, and c) is not required to meet a minimum length requirement.

2. The channel (configurations a, e, and f) is not required to meet a minimum length requirement.

3. Configuration a: If \( \sigma_s \leq S_{PS,s} \) and \( \sigma_s \leq S_{PS,c} \), then the shell and channel are acceptable. Otherwise, increase the thickness of the overstressed component(s) (shell and/or channel) and return to UHX-12.5.1.

   Configurations b and c: If \( \sigma_s \leq S_{PS,s} \), then the shell is acceptable. Otherwise, increase the thickness of the shell and return to UHX-12.5.1.

   Configurations e and f: If \( \sigma_s \leq S_{PS,c} \), then the channel is acceptable. Otherwise, increase the thickness of the channel and return to UHX-12.5.1.

(c) Do not perform the step in UHX-12.5.11.

(d) Repeat the steps in UHX-12.5.1 through UHX-12.5.8 with the following changes until the tubesheet stress criteria have been met:

1. UHX-12.5.4 (Step 4):

   Configurations a, b, and c: \( \beta_s = 0 \), \( k_s = 0 \), \( \lambda_s = 0 \), \( \delta_s = 0 \).

   Configurations a, e, and f: \( \beta_c = 0 \), \( k_c = 0 \), \( \lambda_c = 0 \), \( \delta_c = 0 \).

2. UHX-12.5.7 (Step 7): \( M = |M_o| \).

### UHX-13 RULES FOR THE DESIGN OF FIXED TUBESHEETS

#### UHX-13.1 Scope
These rules cover the design of tubesheets for fixed tubsheet heat exchangers. The tubesheets may have one of the four configurations shown in Figure UHX-13.1:

(a) Configuration a: tubesheet integral with shell and channel;

(b) Configuration b: tubesheet integral with shell and gasketed with channel, extended as a flange;

(c) Configuration c: tubesheet integral with shell and gasketed with channel, not extended as a flange;

(d) Configuration d: tubesheet gasketed with shell and channel.

### UHX-13.2 Conditions of Applicability
The two tubesheets shall have the same thickness, material and edge conditions.

### UHX-13.3 Nomenclature

The symbols described below are used for the design of the tubesheets. Symbols \( D_o \), \( E_t \), \( h \), \( \mu \), \( \mu^* \) and \( \nu^* \) are defined in UHX-11.

\[
\begin{align*}
A &= \text{outside diameter of tubesheet, except as limited by UHX-10(b)} \\
q &= \text{outside diameter of tubesheet, except as limited by UHX-10(b)} \\
a_c &= \text{radial channel dimension} \\
D_a &= \text{configuration a: } a_c = D_a/2 \\
D_b &= \text{configuration b, c, and d: } a_c = G/2 \\
D_s &= \text{configuration c: } a_c = D_s/2 \\
D_t &= \text{configuration d: } a_c = G_t/2 \\
C_p &= \text{bolt circle diameter (see Mandatory Appendix 2)} \\
C_l &= \text{perimeter of the tube layout measured stepwise in increments of one tube pitch from the center-to-center of the outermost tubes (see Figure UHX-12.2)} \\
D_c &= \text{inside channel diameter} \\
D_t &= \text{inside diameter of the expansion joint at its convolution height} \\
D_s &= \text{inside shell diameter} \\
d_t &= \text{nominal outside diameter of tubes} \\
E &= \text{modulus of elasticity for tubesheet material at } T \\
E_c &= \text{modulus of elasticity for channel material at } T_c \\
E_s &= \text{modulus of elasticity for shell material at } T_s \\
E_{s,w} &= \text{joint efficiency (longitudinal stress) for shell} \\
E_{t,w} &= \text{modulus of elasticity for tube material at } T_t \\
G_l &= \text{midpoint of contact between flange and tubesheet} \\
G_c &= \text{diameter of channel gasket load reaction (see Mandatory Appendix 2)} \\
G_s &= \text{diameter of shell gasket load reaction (see Mandatory Appendix 2)} \\
h &= \text{tubesheet thickness} \\
j &= \text{ratio of expansion joint to shell axial rigidity (} J = 1.0 \text{ if no expansion joint)}
\end{align*}
\]
\( k \) = constant accounting for the method of support for the unsupported tube span under consideration
\( = 0.6 \) for unsupported spans between two tubesheets
\( = 0.8 \) for unsupported spans between a tubesheet and a tube support
\( = 1.0 \) for unsupported spans between two tube supports
\( K_f \) = axial rigidity of expansion joint, total force/elongation
\( L_t \) = tube length between inner tubesheet faces
\( = L_t - 2h \)
\( L_s \) = tube length between outer tubesheet faces
\( \text{MAX (}(a),(b),(c),...) \) = greatest of \( a, b, c, ... \)
\( N_t \) = number of tubes
\( P_s \) = effective pressure acting on tubesheet
\( P_{sd} \) = shell side design or operating pressure, as applicable. For shell side vacuum, use a negative value for \( P_s \)
\( P_{sd,\text{max}} \) = maximum shell side design pressure
\( P_{sd,\text{min}} \) = minimum shell side design pressure (negative if vacuum is specified, otherwise zero)
\( P_{sox,\text{max}} \) = \( \max(0, \text{maximum shell side operating pressure for operating condition } x) \)
\( P_{sox,\text{min}} \) = \( \min(0, \text{minimum shell side operating pressure for operating condition } x) \)
\( P_t \) = tube side design or operating pressure, as applicable. For tube side vacuum, use a negative value for \( P_t \)
\( P_{td,\text{max}} \) = maximum tube side design pressure
\( P_{td,\text{min}} \) = minimum tube side design pressure (negative if vacuum is specified, otherwise zero)
\( P_{tox,\text{max}} \) = \( \max(0, \text{maximum tube side operating pressure for operating condition } x) \)
\( P_{tox,\text{min}} \) = \( \min(0, \text{minimum tube side operating pressure for operating condition } x) \)
\( S \) = allowable stress for tubesheet material at \( T \)
\( S_e \) = allowable stress for channel material at \( T_c \)
\( S_{PS} \) = allowable primary plus secondary stress for tubesheet material at \( T \) per UG-23(e)
\( S_{PS,c} \) = allowable primary plus secondary stress for channel material at \( T_c \) per UG-23(e)
\( S_{PS,x} \) = allowable primary plus secondary stress for shell material at \( T_s \) per UG-23(e)
\( S_x \) = allowable stress for shell material at \( T_s \)
\( S_{\text{L,B}} \) = maximum allowable longitudinal compressive stress in accordance with UG-23(b) for the shell
\( S_t \) = allowable stress for tube material at \( T_t \)
NOTE: For a welded tube or pipe, use the allowable stress for the equivalent seamless product. When the allowable stress for the equivalent seamless product is not available, divide the allowable stress of the welded product by 0.85.
\( S_y \) = yield strength from UG-23(f) for tubesheet material at tubesheet temperature \( T \)
\( S_{y,c} \) = yield strength from UG-23(f) for channel material at channel temperature \( T_c \)
\( S_{y,x} \) = yield strength from UG-23(f) for shell material at shell temperature \( T_s \)
\( S_{y,t} \) = yield strength from UG-23(f) for tube material at tube temperature \( T_t \)

\( T \) = tubesheet design temperature for the design condition or operating metal temperature for operating condition \( x \), as applicable [see UHX-13.4(b)]
\( T_c \) = channel design temperature for the design condition or operating metal temperature for operating condition \( x \), as applicable [see UHX-13.4(b)]
\( t_c \) = channel thickness
\( T_s \) = shell design temperature for the design condition or operating metal temperature for operating condition \( x \), as applicable [see UHX-13.4(b)]
\( t_s \) = shell thickness
\( T_{s,m} \) = mean shell metal temperature along shell length
\( T_{s,m,x} \) = shell axial mean metal temperature for operating condition \( x \), as applicable
\( T_t \) = tube design temperature for the design condition or operating metal temperature for operating condition \( x \), as applicable [see UHX-13.4(b)]
\( t_t \) = nominal tube wall thickness
\( T_{t,m} \) = mean tube metal temperature along tube length
\( T_{t,m,x} \) = tube axial mean metal temperature for operating condition \( x \), as applicable
\( W_t \) = tube-to-tubesheet joint load
\( W^* \) = tubesheet effective bolt load determined in accordance with UHX-8
\( x = 1, 2, 3, ..., n \), integer denoting applicable operating condition under consideration (e.g., normal operating, start-up, shutdown, cleaning, upset)
\( \ell \) = unsupported tube span under consideration
\( \alpha_{s,m} \) = mean coefficient of thermal expansion of shell material at \( T_{s,m} \)
\( \alpha_{t,m} \) = mean coefficient of thermal expansion of tube material at \( T_{t,m} \)
\( y \) = axial differential thermal expansion between tubes and shell
\( \Delta_t \) = axial displacement over the length of the thin-walled bellows element (see UHX-16)
\( \Delta_{s,x} \) = shell axial displacement over the length between the inner tubesheet faces, \( L \) [see UHX-17(c)]
\( \nu \) = Poisson’s ratio of tubesheet material
\( \nu_c \) = Poisson’s ratio of channel material
\( \nu_s \) = Poisson’s ratio of shell material
UHX-13.5.12.1 Option 1. Increase the assumed tubesheet thickness \( h \) and return to the step in UHX-13.5.1.

UHX-13.5.12.2 Option 2. Increase the integral shell and/or channel thickness as follows:

- Configurations a, b, and c: If \( \sigma_c > 1.5 \sigma_s \), increase the shell thickness \( t_s \) and return to UHX-13.5.1 (Step 1). It is permitted to increase the shell thickness adjacent to the tubesheet only. (See UHX-13.6.)
- Configuration a: If \( \sigma_c > 1.5 \sigma_s \), increase the channel thickness \( t_c \) and return to the step in UHX-13.5.1.

UHX-13.5.12.3 Option 3. Perform the elastic-plastic calculation procedure as defined in UHX-13.7 only when the conditions of applicability stated in UHX-13.7.2 are satisfied.

UHX-13.6 Calculation Procedure for Effect of Different Shell Material and Thickness Adjacent to the Tubesheet

UHX-13.6.1 Scope.

(a) This procedure describes how to use the rules of UHX-13.5 when the shell has a different thickness and/or a different material adjacent to the tubesheet (see Figure UHX-13.4).

(b) Use of this procedure may result in a smaller tubesheet thickness and should be considered when optimization of the tubesheet thickness or shell stress is desired.

UHX-13.6.2 Conditions of Applicability. This calculation procedure applies only when the shell is integral with the tubesheet (Configurations a, b, and c).

UHX-13.6.3 Additional Nomenclature.

- \( E_{s,1} \) = modulus of elasticity for shell material adjacent to tubesheets at \( T_s \)
- \( \ell_{s,1} \) = lengths of shell of thickness \( t_{s,1} \) adjacent to tubesheets
- \( S_{PS,s,1} \) = allowable primary plus secondary stress for shell material at \( T_s \) per UG-23(e)
- \( S_{s,1} \) = allowable stress for shell material adjacent to tubesheets at \( T_s \)
- \( S_{s,b,1} \) = maximum allowable longitudinal compressive stress in accordance with UG-23(b) for the shell adjacent to the tubesheets
- \( S_{y,s,1} = \) yield strength from UG-23(f) for shell material adjacent to the tubesheets at shell temperature \( T_s \)
- \( t_{s,1} \) = shell thickness adjacent to tubesheets
- \( \alpha_{s,m} \) = mean coefficient of thermal expansion of shell material adjacent to tubesheets at \( T_{s,m} \)

UHX-13.6.4 Calculation Procedure. The calculation procedure outlined in UHX-13.5 shall be performed, accounting for the following modifications:

(a) The shell shall have a thickness of \( t_{s,1} \) for a minimum length of \( 1.8 \sqrt{D_{f,s,1}} \) adjacent to the tubesheets.

(b) In the step in UHX-13.5.2, replace the formula for \( K_s \) with:

\[
K_s^2 = \frac{\pi (D_s + t_c)}{L - \ell_{s,1} - \ell_c} \left[ \frac{\ell_{s,1}}{t_{s,1}} + \frac{\ell_c}{t_c} \right]
\]

Calculate \( K_{s'} \) and \( J \), replacing \( K_s \) with \( K_s^* \). Calculate \( \beta_s \), \( k_s \), and \( \delta_s \), replacing \( t_s \) with \( t_{s,1} \) and \( E_s \) with \( E_{s,1} \).

(c) In the step in UHX-13.5.5, replace the formula for \( \gamma \) with:

\[
\gamma^* = \frac{\left[ T_{s,m} - T_0 \right] \alpha_{s,m} L - \left[ T_{s,1} - T_0 \right]}{\alpha_{s,m} \left( L - \ell_{s,1} - \ell_c \right) + \alpha_{s,m} \ell (1 + \ell_c)}
\]

(d) In the step in UHX-13.5.6, calculate \( P_y \), replacing \( \gamma \) with \( \gamma^* \).

(e) In the step in UHX-13.5.10, calculate \( \sigma_{s,m} \), replacing \( t_s \) with \( t_{s,1} \). Replace \( S_s \) with \( S_{s,1} \) and \( S_{s,b} \) with \( S_{s,b,1} \).

(f) In the step in UHX-13.5.11, calculate \( \sigma_{s,m} \) and \( \sigma_{s,b} \), replacing \( t_s \) with \( t_{s,1} \) and \( E_s \) with \( E_{s,1} \). Replace \( S_s \) with \( S_{s,1} \) and \( S_{PS,s} \) with \( S_{PS,s,1} \).

If the elastic-plastic calculation procedure of UHX-13.7 is being performed, replace \( S_{y,s} \) with \( S_{y,s,1} \), \( S_{PS,s} \) with \( S_{PS,s,1} \), and \( E_s \) with \( E_{s,1} \) in UHX-13.7.

If the radial thermal expansion procedure of UHX-13.8 is being performed, replace \( t_s \) with \( t_{s,1} \) and \( E_s \) with \( E_{s,1} \) in UHX-13.8.

UHX-13.7 Calculation Procedure for Effect of Plasticity at Tubesheet/Channel or Shell Joint

UHX-13.7.1 Scope. This procedure describes how to use the rules of UHX-13.5 when the effect of plasticity at the shell-tubesheet and/or channel-tubesheet joint is to be considered.

When the calculated tubesheet stresses are within the allowable stress limits, but either or both of the calculated shell or channel total stresses exceed their allowable stress limits, an additional "elastic-plastic solution" calculation may be performed.

This calculation permits a reduction of the shell and/or channel modulus of elasticity, where it affects the rotation of the joint, to reflect the anticipated load shift resulting from plastic action at the joint. The reduced effective modulus has the effect of reducing the shell and/or channel stresses in the elastic-plastic calculation; however, due to load shifting this usually leads to an increase in the tubesheet stress. In most cases, an elastic-plastic calculation using the appropriate reduced shell or channel
(5) Acceptance Criteria

(-a) Design loading case acceptance criteria:
\[ |\sigma_{s,m}| \leq S_{\delta} E_{s,w} \] and \[ |\sigma_{ecc,s,m}| \leq S_{ecc} E_{ecc,w} \] and \[ |\sigma_{ecc,l,m}| \leq S_{ecc} E_{ecc,w} \] and \[ |\sigma_{s,l,m}| \leq S_{\delta} E_{s,l,w} \]

(-b) Operating loading case acceptance criteria:
\[ |\sigma_{s,m}| \leq S_{PS,S} \] and \[ |\sigma_{ecc,s,m}| \leq S_{PS,ecc} \] and \[ |\sigma_{ecc,l,m}| \leq S_{PS,ecc} \] and \[ |\sigma_{s,l,m}| \leq S_{PS,S,L} \]

(-c) If axial membrane stress is negative (design and operating): \[ |\sigma_{s,m}| \leq S_{s,b} \] and \[ |\sigma_{ecc,s,m}| \leq S_{ecc,b} \] and \[ |\sigma_{ecc,l,m}| \leq S_{ecc,b} \] and \[ |\sigma_{s,l,m}| \leq S_{s,b} \]

If any of these acceptance criteria are not satisfied, reconsider the design of the failing components and return to (a).

UHX-14 RULES FOR THE DESIGN OF FLOATING TUBESHEETS

UHX-14.1 Scope

(a) These rules cover the design of tubesheets for floating tubesheet heat exchangers that have one stationary tubesheet and one floating tubesheet. Three types of floating tubesheet heat exchangers are covered as shown in Figure UHX-14.1:

(1) Sketch (a), immersed floating head;
(2) Sketch (b), externally sealed floating head;
(3) Sketch (c), internally sealed floating tubesheet.

(b) Stationary tubesheets may have one of the six configurations shown in Figure UHX-14.2:

(1) Configuration a: tubesheet integral with shell and channel;
(2) Configuration b: tubesheet integral with shell and gasketed with channel, extended as a flange;
(3) Configuration c: tubesheet integral with shell and gasketed with channel, not extended as a flange;
(4) Configuration d: tubesheet gasketed with shell and channel;
(5) Configuration e: tubesheet gasketed with shell and integral with channel, extended as a flange;
(6) Configuration f: tubesheet gasketed with shell and integral with channel, not extended as a flange.

(c) Floating tubesheets may have one of the four configurations shown in Figure UHX-14.3:

(1) Configuration A: tubesheet integral;
(2) Configuration B: tubesheet gasketed, extended as a flange;
(3) Configuration C: tubesheet gasketed, not extended as a flange;
(4) Configuration D: tubesheet internally sealed.

UHX-14.2 Conditions of Applicability

The two tubesheets shall have the same thickness and material.

UHX-14.3 Nomenclature

The symbols described below are used for the design of the stationary and floating tubesheets. Symbols \( D_{o}, E^{*}, h^{*}, \mu, \mu^{*} \), and \( v^{*} \) are defined in UHX-11.

\[ A = \text{outside diameter of tubesheet, except as limited by UHX-10(b)} \]
\[ a_{c} = \text{radial channel dimension} \]
\[ \text{Configurations a, e, f, and A: } a_{c} = D_{c}/2 \]
\[ \text{Configurations b, c, d, B, and C: } a_{c} = G_{c}/2 \]
\[ \text{Configuration D: } a_{c} = A/2 \]
\[ a_{o} = \text{equivalent radius of outer tube limit circle} \]
\[ A_{p} = \text{total area enclosed by } C_{p} \]
\[ a_{s} = \text{radial shell dimension} \]
\[ \text{Configurations a, b, and c: } a_{s} = D_{s}/2 \]
\[ \text{Configurations d, e, and f: } a_{s} = G_{s}/2 \]
\[ \text{Configurations A, B, C, and D: } a_{s} = a_{c} \]
\[ C = \text{bolt circle diameter (see Mandatory Appendix 2)} \]
\[ C_{p} = \text{perimeter of the tube layout measured stepwise in increments of one tube pitch from the center-to-center of the outermost tubes (see Figure UHX-12.2)} \]
\[ D_{c} = \text{inside channel diameter} \]
\[ D_{s} = \text{inside shell diameter} \]
\[ d_{t} = \text{nominal outside diameter of tube} \]
\[ E = \text{modulus of elasticity for tubesheet material at } T \]
\[ E_{c} = \text{modulus of elasticity for channel material at } T_{c} \]
\[ E_{s} = \text{modulus of elasticity for shell material at } T_{s} \]
\[ E_{t} = \text{modulus of elasticity for tube material at } T_{t} \]
\[ G_{1} = \text{midpoint of contact between flange and tubesheet} \]
\[ G_{c} = \text{diameter of channel gasket load reaction (see Mandatory Appendix 2)} \]
\[ G_{s} = \text{diameter of shell gasket load reaction (see Mandatory Appendix 2)} \]
\[ h = \text{tubesheet thickness} \]
\[ k = \text{constant accounting for the method of support for the unsupported tube span under consideration} \]
\[ = 0.6 \text{ for unsupported spans between two tubesheets} \]
\[ = 0.8 \text{ for unsupported spans between a tubesheet and a tube support} \]
\[ = 1.0 \text{ for unsupported spans between two tube supports} \]
\[ L = \text{tube length between inner tubesheet faces} \]
\[ = L_{t} - 2h \]
\[ l = \text{unsupported tube span under consideration} \]
\[ L_{t} = \text{tube length between outer tubesheet faces} \]
\[ \text{MAX [(a), (b), (c), ...]} = \text{greatest of } a, b, c, ... \]
\[ N_{t} = \text{number of tubes} \]
\[ P_{e} = \text{effective pressure acting on tubesheet} \]
\[ P_{s} = \text{shell side design or operating pressure, as applicable. For shell side vacuum, use a negative value for } P_{s}. \]
\[ P_{sd,\text{max}} = \text{maximum shell side design pressure} \]
\[ P_{sd,\text{min}} = \text{minimum shell side design pressure (negative if vacuum is specified, otherwise zero)} \]
Figure UHX-14.1
Floating Tubesheet Heat Exchangers

(a) Typical Floating Tubesheet Exchanger With an Immersed Floating Head

(b) Typical Floating Tubesheet Exchanger With an Externally Sealed Floating Head

(c) Typical Floating Tubesheet Exchanger With an Internally Sealed Floating Tubesheet
Figure UHX-4.2
Stationary Tubing Configurations

(a) Configuration a:
Tubing Integral With Shell and Channel

(b) Configuration b:
Tubing Integral With Channel, Extended as a Flange

(c) Configuration c:
Tubing Integral With Shell and Gasketed Channel, Not Extended as a Flange

(d) Configuration d:
Tubing Gasketed With Shell and Channel

(e) Configuration e:
Tubing Gasketed With Shell and Integral Channel

(f) Configuration f:
Tubing Gasketed With Shell and Integral Channel, Not Extended as a Flange

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\[ P_{sox,\text{max}} = \max(0, \text{maximum shell side operating pressure for operating condition } x) \]

\[ P_{sox,\text{min}} = \min(0, \text{minimum shell side operating pressure for operating condition } x) \]

\[ P_t = \text{tube side design or operating pressure, as applicable. For tube side vacuum, use a negative value for } P_t. \]

\[ P_{td,\text{max}} = \text{maximum tube side design pressure} \]

\[ P_{td,\text{min}} = \text{minimum tube side design pressure (negative if vacuum is specified, otherwise zero)} \]

\[ P_{tox,\text{max}} = \max(0, \text{maximum tube side operating pressure for operating condition } x) \]

\[ P_{tox,\text{min}} = \min(0, \text{minimum tube side operating pressure for operating condition } x) \]

\[ S = \text{allowable stress for tubesheet material at } T \]

\[ S_{y,c} = \text{allowable stress for channel material at } T_c \]

\[ S_{ps,\text{c}} = \text{allowable primary plus secondary stress for tubesheet material at } T_{ps,\text{c}} \]

\[ S_{ps,c} = \text{allowable primary plus secondary stress for channel material at } T_{ps,c} \]

\[ S_1 = \text{allowable stress for shell material at } T_1 \]

\[ S_t = \text{allowable stress for tube material at } T_t \]

**NOTE:** For a welded tube or pipe, use the allowable stress for the equivalent seamless product. When the allowable stress for the equivalent seamless product is not available, divide the allowable stress of the welded product by 0.85.

\[ S_{p} = \text{yield strength from UG-23(l) for tubesheet material at tubesheet temperature } (T) \]

\[ S_{y,c} = \text{yield strength from UG-23(l) for channel material at channel temperature } (T_c) \]

\[ S_{y,\text{c}} = \text{yield strength from UG-23(l) for shell material at shell temperature } (T_s) \]

\[ S_{p,\text{c}} = \text{yield strength from UG-23(l) for tube material at tube temperature } (T_t) \]

**NOTE:** The yield strength shall be taken from Section II, Part D, Subpart 1, Table Y-1. When a yield strength value is not listed in Table Y-1, one may be obtained by using the procedure in UG-28(c)(2)-Step 3.

\[ T = \text{tubesheet design temperature for the design condition or operating metal temperature for operating condition } x, \text{ as applicable [see UH-14.4(c)]} \]

\[ T_a = \text{ambient temperature, } 70^\circ \text{F (20°C)} \]

\[ T_c = \text{channel design temperature for the design condition or operating metal temperature for operating condition } x, \text{ as applicable [see UH-14.4(c)]} \]

\[ t_c = \text{channel thickness} \]

\[ T_s = \text{shell design temperature for the design condition or operating metal temperature for operating condition } x, \text{ as applicable [see UH-14.4(c)]} \]

\[ t_s = \text{shell thickness} \]

\[ T_t = \text{tube design temperature for the design condition or operating metal temperature for operating condition } x, \text{ as applicable [see UH-14.4(c)]} \]

\[ t_t = \text{nominal tube wall thickness} \]

\[ W_e = \text{tube-to-tubesheet joint load} \]

\[ W^* = \text{tubesheet effective bolt load determined in accordance with UHX-8} \]

\[ x = 1, 2, 3, \ldots, n, \text{integer denoting applicable operating condition under consideration (e.g., normal operating, start-up, shutdown, cleaning, upset)} \]

\[ \nu = \text{Poisson’s ratio of tubesheet material} \]

\[ \nu_c = \text{Poisson’s ratio of channel material} \]

\[ \nu_s = \text{Poisson’s ratio of shell material} \]

\[ \nu_t = \text{Poisson’s ratio of tube material} \]

**UHX-14.4 Design Considerations**

(a) The calculation shall be performed for the stationary end and for the floating end of the exchanger. Since the edge configurations of the stationary and floating tubesheets are different, the data may be different for each set of calculations. However, the conditions of applicability given in UHX-14.2 must be maintained. For the stationary end, diameters \( A, C, D_o, D_o, G_o, G_t, \) and thickness \( t_c \) shall be taken from Figure UHX-14.2. For the floating end, diameters \( A, C, D_o, G_o, G_t, \) and thickness \( t_c \) shall be taken from Figure UHX-14.3, and the radial shell dimension \( a_s \) shall be taken equal to \( a_c. \)

(b) It is generally not possible to determine, by observation, the most severe condition of coincident pressure, temperature, and radial differential thermal expansion. Thus, it is necessary to evaluate all the anticipated loading conditions to ensure that the worst load combination has been accounted for.

\( P_{td,\text{min}} \) and \( P_{tox,\text{min}} \) shall specify all the design conditions that govern the design of the exchanger (i.e., tubesheet, tube, shell, channel, tube-to-tubesheet joint). These shall include, but not be limited to, normal operating, start-up, shutdown, cleaning, and upset conditions.

For each of these conditions, the following loading cases shall be considered to determine the effective pressure \( P_e \) to be used in the design equations:

(1) **Design Loading Cases.** Table UHX-14.4-1 provides the load combinations required to evaluate the heat exchanger for the design condition. When \( P_{td,\text{min}} \) and \( P_{tox,\text{min}} \) are both zero, design loading case 4 does not need to be considered.

(2) **Operating Loading Cases.** The operating loading cases are required only when the effect of radial differential thermal expansion is to be considered [see (e)].

(3) When differential pressure design is specified by the user or his designated agent, the design shall be based only on design loading case 3 and operating loading cases 3 and 4 for each specified operating condition. If the tube side is the higher-pressure side, \( P_t \) shall be the tube side design pressure, and \( P_s \) shall be \( P_t \) less the differential design pressure. If the shell side is the higher-pressure side, \( P_s \) shall be the shell side design pressure, and \( P_t \) shall be \( P_s \) less the differential design pressure. For the operating
MANDATORY APPENDIX 1
SUPPLEMENTARY DESIGN FORMULAS

1-1 THICKNESS OF CYLINDRICAL AND SPHERICAL SHELLS

The following equations, in terms of the outside radius, are equivalent to and may be used instead of those given in UG-27(d):
(a) For cylindrical shells (circumferential stress),

\[ t = \frac{PR_o}{SE + 0.4P} \quad \text{or} \quad P = \frac{SEt}{R_o - 0.4t} \quad (1) \]

where

\[ R_o = \text{outside radius of the shell course under consideration} \]

(b) For spherical shells,

\[ t = \frac{PR_o}{2SE + 0.8P} \quad \text{or} \quad P = \frac{2SEt}{R_o - 0.8t} \quad (2) \]

Other symbols are as defined in UG-27.

1-2 CYLINDRICAL SHELLS

(a) Circumferential Stress (Longitudinal Joints). When the thickness of the cylindrical shell under internal design pressure exceeds one-half of the inside radius, or when \( P \) exceeds \( 0.385SE \), the following equations shall apply. The following equations may be used in lieu of those given in UG-27(c):

When \( P \) is known and \( t \) is desired,

\[ t = R \left( \exp \left[ \frac{P}{SE} \right] - 1 \right) = R_o \left( 1 - \exp \left[ -\frac{P}{SE} \right] \right) \quad (1) \]

Where \( t \) is known and \( P \) is desired,

\[ P = SE \log_e \left( \frac{R + t}{R} \right) = SE \log_e \left( \frac{R_o}{R_o - t} \right) \quad (2) \]

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

When \( P \) is known and \( t \) is desired,

\[ t = R \left[ \exp \left( \frac{0.50 \cdot P}{SE} \right) - 1 \right] = R_o \left( 1 - \exp \left[ -\frac{0.50 \cdot P}{SE} \right] \right) \quad (1) \]

When \( t \) is known and \( P \) is desired,

\[ P = 2.0 \cdot SE \log_e \left( \frac{R + t}{R} \right) = 2.0 \cdot SE \log_e \left( \frac{R_o}{R_o - t} \right) \quad (2) \]

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

1-3 SPHERICAL SHELLS

When the thickness of the shell of a wholly spherical vessel or of a hemispherical head under internal design pressure exceeds \( 0.356R \), or when \( P \) exceeds \( 0.665SE \), the following equations shall apply. The following equations may be used in lieu of those given in UG-27(d).

When \( P \) is known and \( t \) is desired,

\[ t = R \left[ \exp \left( \frac{0.50 \cdot P}{SE} \right) - 1 \right] = R_o \left( 1 - \exp \left[ -\frac{0.50 \cdot P}{SE} \right] \right) \quad (1) \]

When \( t \) is known and \( P \) is desired,

\[ P = 2.0 \cdot SE \log_e \left( \frac{R + t}{R} \right) = 2.0 \cdot SE \log_e \left( \frac{R_o}{R_o - t} \right) \quad (2) \]

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

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

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

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The equations in (c) and (d) given below shall be used for \( t/L \geq 0.002 \). For \( t/L < 0.002 \), the rules of (f) shall also be met.

(b) The symbols defined below are used in the equations of this paragraph (see Figure 1-4):

\[ D = \text{inside diameter of the head skirt; or inside length of the major axis of an ellipsoidal head; or inside diameter of a cone head at the point under consideration measured perpendicular to the longitudinal axis} \]

\[ D_o = \text{outside diameter of the head skirt; or outside length of the major axis of an ellipsoidal head; or outside diameter of a cone head at the point under consideration measured perpendicular to the longitudinal axis} \]

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

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

\[ E_T = \text{modulus of elasticity at maximum design temperature, psi. The value of } E_T \text{ shall be taken from the applicable Section II, Part D, Subpart 2, Table TM} \]

\[ h = \text{one-half of the length of the minor axis of the ellipsoidal head, or the inside depth of the ellipsoidal head measured from the tangent line (head-bend line)} \]

\[ K = \text{a factor in the equations for ellipsoidal heads depending on the head proportion } D/2h \]

\[ L = \text{inside spherical or crown radius for torispherical and hemispherical heads} \]

\[ L_o = \text{outside spherical or crown radius} \]

\[ L/r = \text{ratio of the inside crown radius to the inside knuckle radius, used in Table 1-4.2} \]

\[ M = \text{a factor in the equations for torispherical heads depending on the head proportion } L/r \]

\[ P = \text{internal design pressure (see UG-21)} \]

\[ r = \text{inside knuckle radius} \]

\[ S = \text{maximum allowable working stress, as given in Subsection C except as limited by endnote 88, UG-24, UG-32(d), and UW-12} \]

\[ S_y = \text{yield strength from UG-23(f) at maximum design temperature} \]

\[ t = \text{minimum required thickness of head after forming} \]

\[ t_s = \text{minimum specified thickness of head after forming, in. (mm). } t_s \text{ shall be } \geq t \]

\( \alpha = \text{one-half of the included (apex) angle of the cone at the centerline of the head} \)

\[ \text{(c) Ellipsoidal Heads} \]

\[ t = \frac{P D K}{2SE - 0.2P} \text{ or } P = \frac{2SE t}{KD + 0.2t} \]  

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

or

\[ P = \frac{2SE t}{KD - 2t(K - 0.1)} \]

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

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

\[ \text{(d) Torispherical Heads} \]

\[ t = \frac{PLM}{2SE - 0.2P} \text{ or } P = \frac{2SE t}{LM + 0.2t} \]

\[ t = \frac{PLM}{2SE + P(M - 0.2)} \]

or

\[ P = \frac{2SE t}{ML - t(M - 0.2)} \]

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

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

\[ \text{(e) Conical Heads} \]

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

or

\[ P = \frac{2SE t \cos \alpha}{D + 1.2t \cos \alpha} \]

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

or

\[ P = \frac{2SE \cos \alpha}{D + 1.2t \cos \alpha} \]
(2) flange thickness for ring gasket

$$T = \frac{M_0}{SB} \sqrt{A + B} \quad (2)$$

(3) flange thickness for full face gasket

$$T = 0.6 \sqrt{\frac{PL}{S} \frac{B(A + B)(C - B)}{A - B}} \quad (3)$$

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

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

(1) head thickness

(-a) for pressure on concave side,

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

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

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

$$T = Q + \sqrt{\frac{1.875M_0(C + B)}{SB(7C - 5B)}} \quad (5)$$

where

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

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

$$T = Q + \sqrt{\frac{1.875M_0(C + B)}{SB(3C - B)}} \quad (6)$$

where

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

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

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

where

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

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

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

where

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

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

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

(1) head thickness

(-a) for pressure on concave side,

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

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

(2) flange thickness

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

where

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

and

$$J = \frac{M_0}{SB} \left( \frac{A + B}{A - B} \right)$$

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

1-7 LARGE OPENINGS IN CYLINDRICAL AND CONICAL SHELLS

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

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

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

(b) In addition to meeting the requirements of (a),
(1) openings for radial nozzles that exceed the limits in UG-36(b)(1) and that also are within the range defined by the following limits shall meet the requirements in (2), (3), and (4) below:

- (a) vessel diameters greater than 60 in. (1520 mm) I.D.;
- (b) nozzle diameters that exceed 40 in. (1020 mm) I.D. and also exceed 3.44R/t; the terms R and t are defined in Figures 1-7-1 and 1-7-2;
- (c) the ratio R/n does not exceed 0.7; for nozzle openings with R/n exceeding 0.7, refer to U-2(g).

The rules are limited to radial nozzles in cylindrical and conical shells (with the half-apex angle equal to or less than 30 deg) that do not have internal projections, and do not include any analysis for stresses resulting from externally applied mechanical loads. For such cases, U-2(g) shall apply.

(2) The membrane stress SM as calculated by eq. (4)(1) or (4)(2) below shall not exceed S, as defined in UG-37 for the applicable materials at design conditions. The maximum combined membrane stress SM and bending stress Sb shall not exceed 1.5S at design conditions. Sb shall be calculated by eq. (4)(5) below.

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

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

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

**Case A** (see Figure 1-7-1)

\[
S_m = p \left( \frac{R(R_n + t_n + \sqrt{R_m t_n}) + R_n(t_e + \sqrt{R_m t_n})}{A_5} \right)
\]  

**Case B** (see Figure 1-7-1)

\[
S_m = p \left( \frac{R(R_n + t_n + \sqrt{R_m t_n}) + R_n(t_e + \sqrt{R_m t_n})}{A_5} \right)
\]

**Cases A and B** (See Figure 1-7-1 or Figure 1-7-2)

\[
M = \left( \frac{R_n^3}{6} + RR_n^2 \right) P
\]

\[
a = e + t/2
\]

\[
S_b = \frac{Ma}{I}
\]

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

- \(A_s\) = shaded (cross-hatched) area in Figure 1-7-1, Case A or Case B
- \(a\) = distance between neutral axis of the shaded area in Figure 1-7-1 or Figure 1-7-2 and the inside of vessel wall
- \(e\) = distance between neutral axis of the shaded area and midwall of the shell
- \(I\) = moment of inertia of the larger of the shaded areas in Figure 1-7-1 or Figure 1-7-2 about neutral axis
- \(P\) = internal or external pressure
- \(R_m\) = mean radius of shell
- \(R_{nm}\) = mean radius of nozzle neck
- \(S_b\) = bending stress at the intersection of inside of the nozzle neck and inside of the vessel shell along the vessel shell longitudinal axis
- \(S_m\) = membrane stress calculated by eq. (4)(1) or eq. (4)(2)
- \(S_T\) = yield strength from UG-23(f) for the material at test temperature

(c) In the design and fabrication of large openings, the Manufacturer should consider details that may be appropriate to minimize distortion and localized stresses around the opening. For example, reinforcement often may be advantageously obtained by use of heavier shell plate for a vessel course or inserted locally around the opening; weld may be ground to concave contour and the inside corners of the opening rounded to a generous radius to reduce stress concentrations. The user and the Manufacturer should agree on the extent and type of nondestructive examination of welds that may be appropriate for the intended service conditions and the materials of construction. Proof testing may be appropriate in extreme cases of large openings approaching full vessel diameter, openings of unusual shape, etc.
(d) Particular attention shall be given to the effects of local internal and external loads and expansion differentials at design temperature, including reactions at supporting lugs, piping, and other types of attachments, as specified in UG-22.

(e) Except as otherwise specified in this Appendix, vessel parts of noncircular cross section subject to external pressure shall be designed in accordance with U-2(g).

(f) The end closures for vessels of this type shall be designed in accordance with the provisions of U-2(g) and/or UG-101 except in cases where the ends are flat plates subject to rating under the rules of UG-34. Unstayed flat heads used as welded end plates for vessels described in this Appendix shall conform to the rules of UG-34 except that a C factor of 0.20 shall be used in all cases.

(g) The requirements for ligaments prescribed in UG-53 shall apply except as modified in 13-6 for the case of multidiameter holes in plates. [See 13-18(b).]

The ligament efficiencies \( e_m \) and \( e_b \) shall only be applied to the calculated stresses for the plates containing the ligaments.

(1) When \( e_m \) and \( e_b \) are less than the joint efficiency \( E \) (see 13-5 and UW-12), which would be used if there were no ligaments in the plate, the membrane and bending stresses calculated based on the gross area of the section shall be divided by \( e_m \) and \( e_b \), respectively, to obtain the stresses based on the net area for the section. The allowable design stresses for membrane and membrane plus bending shall be calculated as described in (b) using \( E = 1.0 \).

(2) When \( e_m \) and \( e_b \) are greater than the joint efficiency \( E \), which would be used if there were no ligaments in the plate, the stresses shall be calculated as if there were no ligaments in the plate. The allowable design stresses for membrane and membrane plus bending shall be calculated as described in (b) using the appropriate \( E \) factor required by UW-12.

(h) The design equations in this Appendix are based on vessels in which the length \( L \), to side dimension \( H \) or \( h \) ratio (aspect ratio) is greater than 4. These equations are conservatively applicable to vessels of aspect ratio less than 4 and may thus be used as specified in this Appendix. Vessel sideplates with aspect ratios less than 4 are strengthened by the interaction of the end closures and may be designed in accordance with the provisions of U-2(g) by using established techniques of structural analysis. Membrane and bending stresses shall be determined throughout the structure and shall not exceed the allowable values established in this Appendix. Short unreinforced or unstayed vessels of rectangular cross section having an aspect ratio not greater than 2.0 may be designed in accordance with 13-18(b) and 13-18(c).

(i) Bolted full-side or end plates and flanges may be provided for vessels of rectangular cross section. Many acceptable configurations are possible. Therefore, rules for specific designs are not provided, and these parts shall be designed in accordance with the provisions of UG-34 for unstayed flat plates and U-2(g) for the flange assembly. Analysis of the components must consider gasket reactions, bolting forces, and resulting moments, as well as pressure and other mechanical loading.

(j) Openings may be provided in vessels of noncircular cross section as follows:

(1) Openings in noncircular vessels do not require reinforcement other than that inherent in the construction, provided they meet the conditions given in UG-36(c)(3).

(2) As a minimum, the reinforcement of other openings in noncircular vessels shall comply with UG-39, except the required thickness to be used in the reinforcement calculations shall be the thickness required to satisfy the stress criteria in (b). Compensation for openings in noncircular vessels must account for the bending strength as well as the membrane strength of the side with the opening. In addition, openings may significantly affect the stresses in adjacent sides. Because many acceptable configurations are possible, rules for specific designs are not provided [see U-2(g)].

(k) For vessels without reinforcements and for vessels with stay plates and stay rods (13-7, 13-9, 13-10, 13-12, and 13-13), the moments of inertia are calculated on a per-unit-width basis. That is, \( I = bh^2/12 \), where \( b = 1.0 \). For vessels with reinforcements that do not extend around the corners of the vessel (13-8 and 13-11), the moments of inertia are calculated using the traditional definition, \( I = p(t^2 + a^2)/12 \). For width of cross section for vessels with reinforcements, see 13-8(d). For unreinforced vessels of rectangular cross section (13-7), the given moments are defined on a per-unit-width basis. That is, \( M_A \) and \( M_r \) have dimensions (length \( \times \) force/length) = force.

13-5 NOMENCLATURE

Symbols used in this Appendix are as follows:

\[ A = R(2y + \pi d_2) \]
\[ A_1 = \text{cross-sectional area of reinforcing member} \]
\[ A_2 = \text{attached to plate of thickness } t_1 \]
\[ A_3 = \text{attached to plate of thickness } t_2 \]
\[ A_4 = r(2y_1 + \pi) \]
\[ B = R^2(y^2 + \pi y d_2 + 2a_2) \]
\[ b_1 = p - d_1 \] (Figure 13-6)
\[ b_2 = p - d_2 \] (Figure 13-6)
\[ b_m = p - d_m \] (Figure 13-6)
\[ b_n = p - d_n \] (Figure 13-6)
\[ C = \text{plate coefficient, UG-47} \]
\[ c = \text{distance from neutral axis of cross section to extreme fibers} \] (see \( c_1 \) and \( c_2 \)). The appropriate \( c_1 \) or \( c_2 \) value shall be substituted for the \( c \) term in the stress equations.
\[ C_1 = R^2(2y^2 + 3\pi a_2 + 12a_2) \]
\[ C_2 = r^2(2y_1^2 + 3\pi y_1 + 12) \]

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\( c_i \) = distance from neutral axis of cross section of plate, composite section, or section with multi-diameter holes (see 2-12) to the inside surface of the vessel. Sign is always positive (+).

\( c_o \) = distance from neutral axis of cross section of plate, composite section, or section with multi-diameter holes (see 2-12) to the extreme outside surface of the section. Sign is always negative (-).

\( \pm c_x \) = distance from neutral axis of cross section to any intermediate point. Sign is positive (+) when inward and sign is negative (-) when outward.

\( D_1 = R^2(y^2 + 2\pi \gamma a_2 + 12\gamma a_2 + 2\pi a_2) \)

\( d_i \) = diameter of hole of length \( T_i \) (pitch diameter for threaded hole) (Figure 13-6)

\( d_2 \) = diameter of hole of length \( T_2 \) (pitch diameter for threaded hole) (Figure 13-6)

\( D_E \) = equivalent uniform diameter of multi-diameter hole

\( d_j \) = distance from midlength of plate to weld joint or centerline of row of holes in the straight segment of the plate

\( d_n \) = diameter of hole of length \( T_n \) (pitch diameter for threaded hole) (Figure 13-6)

\( d_o \) = diameter of hole of length \( T_o \) (pitch diameter for threaded hole) (Figure 13-6)

\( E \) = joint efficiency factor as required by UW-12 for all Category A butt joints (see UW-3) and to any Category C or D butt joints. The joint efficiency factor is used as described in 13-4(b) and 13-4(g) to calculate the allowable design membrane and membrane plus bending stresses.

\( E_1 = R^2(4\gamma^2 + 6\pi \gamma a_2 + 24\gamma a_2 + 3\pi a_2) \)

\( E_2 \) = modulus of elasticity at design temperature

\( E_3 \) = modulus of elasticity at ambient temperature

\( e_b \) = bending ligament efficiency [see 13-4(g), 2-12, and 13-18(b)]

\( e_m \) = membrane ligament efficiency [see 13-4(g), 2-12, and 13-18(b)]

\( F = (3AD_1 - 2BC_1)(14E_i - 6b^2) \)

\( H \) = inside length of short side of rectangular vessel

\( = 2(L_1 + L_{11}) \) for equations in 13-8(d) for Figure 13-2(a), sketches (5) and (6)

\( h \) = inside length of long side of unstayed rectangular vessel; or dimension perpendicular to the \( H \) dimension in stayed vessels as shown in Figure 13-2(a), sketches (7), (8), (9), and (10), in which case \( h \) may be greater than, equal to, or less than \( H \).

\( = 2(L_2 + L_{21}) \) for equations in 13-8(d) for Figure 13-2(a), sketches (5) and (6)

\( = 2L_2 \) for equations in 13-8(d) for Figure 13-2(b), sketch (2)

\( H_0 \) = outside length of short side of rectangular vessel

\( H_1 \) = centroidal length of reinforcing member on short side of rectangular vessel

\( h_1 \) = centroidal length of reinforcing member on long side of rectangular vessel

\( h_o \) = outside length of long side of rectangular vessel

\( I \) = moment of inertia

\( I_1 \) = moment of inertia of strip of thickness \( \frac{t_1}{3} \)

\( I_2 \) = moment of inertia of strip of thickness \( \frac{t_2}{3} \)

\( I_3 \) = moment of inertia of strip of thickness \( \frac{t_3}{3} \)

\( I_{11} \) = moment of inertia of combined reinforcing member and effective width of plate \( w \) of thickness \( t_1 \)

\( I_{21} \) = moment of inertia of combined reinforcing member and effective width of plate \( w \) of thickness \( t_2 \)

\( I_{22} \) = moment of inertia of strip of thickness \( \frac{t_2}{3} \)

\( I_e \) = moment of inertia about axis parallel to long side of rectangular vessel and passing through centroid of cross-sectional area

\( J \) = plate parameter, Table 13-8(d)

\( J_2 \) = plate parameter, Table 13-13(c)

\( K \) = vessel parameter \( (L_2/L_1)a_1 \)

\( k \) = reinforcement member parameter

\( = (L_{21}/L_1)a_1 \)

\( K_1 = 2k_2 + 3 \)

\( K_2 = 3k_1 + 2k_2 \)

\( K_3 = factor \ for \ unreinforced \ rectangular \ vessel \) [Figure 13-2(a), sketch (3)]

\( K_4 \) = factor for reinforced rectangular vessel [Figure 13-2(a), sketch (5)]

\( L_1 \) = half-length of short side of rounded or chamfered corner vessel without reinforcements; half-length of reinforcement on short side of reinforced vessel

\( L_2 \) = half-length of long side plate of obround and rounded or chamfered corner rectangular vessels without reinforcements; half-length of reinforcement on long side of reinforced vessel

\( L_{3}, L_{4} \) = dimensions of rectangular vessel [Figure 13-2(a), sketches (5) and (6)]

\( L_{21}, L_{11} \) = dimensions of rectangular vessel [Figure 13-2(a), sketches (5) and (6)]

\( L_v \) = length of vessel

\( M \) = bending moment

\( M_{A}, M_{M} \) = bending moment at midpoint of long side.

Positive sign results in a compression stress in the outermost fibers in the cross section.

\( M_i \) = bending moment at weld joint

\( N = K_1K_2 - k_2^2 \)

\( P \) = internal design pressure (see UG-21)
\( p = \) pitch distance; distance between reinforcing members; plate width between edges of reinforcing members

\( P_1, P_2 = \) internal design pressures in two-compartment vessel \([\text{Figure 13-2}(c)]\) where \( P_1 > P_2 \)

\( P_m = \) external design pressure

\( R = \) inside radius

\( r = \) radius to centroidal axis of reinforcement member on ovoid vessel

\( R_1 = \) least radius of gyration of noncircular cross-sectional vessel

\( S = \) allowable tensile stress values \((\text{see UG-23})\)

\( S_0 = \) bending stress \((+ = \text{tension}, - = \text{compression})\)

\( S_m = \) membrane stress

\( S_p = \) total stress \((S_m + S_b)\)

\( S_y = \) yield strength from UG-23\((f)\) for material at design temperature

\( t = \) plate thickness

\( T_1 = \) length of hole of diameter \( d_1 \)

\( t_1 = \) thickness of short-side plates of vessel

\( T_2 = \) length of hole of diameter \( d_2 \)

\( t_2 = \) thickness of long-side plates of vessel

\( t_3 = \) thickness or diameter of staying member

\( t_4 = \) thickness or diameter of staying member

\( t_5 = \) thickness of end closure plate or head of vessel

\( t_{22} = \) thickness of long-side plates of vessel

\( T_n = \) length of hole of diameter \( d_n \)

\( T_o = \) length of hole of diameter \( d_o \)

\( w = \) width of plate included in moment of inertia calculation of reinforced section

\( \bar{X} = \) distance from base of plate to neutral axis

\( \bar{y} = \) distance from geometric center of end plate to centroid of cross-sectional area of a rectangular vessel. If both long-side plates are of equal thickness \( t_n \) then \( \bar{y} = 0 \).

\( Y_1 = \) distance between centroid of reinforced cross section with \( I_{11} \) and centerline of shell plate with \( t_1 \) \([\text{Figure 13-2}(a), \text{sketch (6)})\]

\( Y_2 = \) distance between centroid of reinforced cross section with \( I_{21} \) and centerline of shell plate with \( t_2 \) \([\text{Figure 13-2}(a), \text{sketch (6)})\]

\( Z = \) plate parameter, UG-34

\( \Delta = \) material parameter associated with \( w \) \([\text{Table 13-8}(c)]\)

\( \alpha = \) rectangular vessel parameter

\( \alpha_1 = \) rectangular vessel reinforcement parameter

\( \alpha_2 = \frac{l_2}{l_1} \)

\( \alpha_3 = \frac{L_2}{L_1} \)

\( \beta = \frac{h}{p}, \frac{H}{p}, \text{or} \frac{2R}{p} \)

\( \gamma = \frac{L_2}{R} \)

\( \gamma_1 = \frac{L_2}{r} \)

\( \theta = \) angle

\( v = \) Poisson’s ratio

\( \pi = 3.1415 \)

\( \phi = R/L_1 \)

13-6 LIGAMENT EFFICIENCY OF MULTIDIAMETER HOLES IN PLATES

In calculations made according to this Appendix for the case of a plate with uniform diameter holes, the ligament efficiency factors \( e_m \) and \( e_b \) for membrane and bending stresses, respectively, are considered to be the same. See 13-4(g) and 13-18(b) for application of ligament efficiency factors. In the case of multidiameter holes, the neutral axis of the ligament may no longer be at midthickness of the plate; in this case, for bending loads, the stress is higher at one of the plate surfaces than at the other surface.

(a) Ligament Efficiency of Plate With Multidiameter Holes Subject to Membrane Stress. Figure 13-6 shows a plate with multidiameter holes. In the case of membrane stresses, the ligament efficiency is as follows:

\[
e_{m} = \frac{(p - D_B)}{p}
\]

where

\[
D_B = \frac{t}{L} (d_0 T_0 + d_1 T_1 + d_2 T_2 + \ldots + d_n T_n)
\]

(b) Ligament Efficiency of Plate With Multidiameter Holes Subject to Bending Stress. Figure 13-6 shows a plate with multidiameter holes. In the case of bending loads, the ligament efficiency is given by

\[
e_{b} = \frac{(p - D_B)}{p}
\]

where

\[
D_B = p - \frac{6l}{t^2}
\]

\[
l = \frac{1}{12} \left[ b_0 T_0^3 + b_1 T_1^3 + b_2 T_2^3 + \ldots + b_n T_n^3 \right]
\]

\[
+ \frac{b_0 T_0^2}{2} \left( T_1 + T_2 + \ldots + T_n - \bar{X} \right)^2
\]

\[
+ \frac{b_1 T_1^2}{2} \left( T_2 + \ldots + T_n - \bar{X} \right)^2
\]

\[
+ \frac{b_2 T_2^2}{2} \left( \ldots + T_n - \bar{X} \right)^2
\]

\[
+ \ldots + \frac{b_n T_n^2}{2} \left( \bar{X} - \frac{T_n}{2} \right)^2
\]
\[ S_1' = \frac{1}{2n} \left( \frac{D_t^2 T_t E_t k}{D_h + nt} \right) L_t E_t + r_c L_c E_c k P \]

shall comply with \( S_1' \leq C_w S_c \).

(b) For internally attached bellows, the circumferential membrane stress in the shell due to pressure

\[ S_1'' = \frac{1}{2n} \left( \frac{D_h + nt}{L_t E_t + r_c} \right) L_t E_t + r_c \left( D_h + nt \right) L_c E_c k P \]

shall comply with \( S_1'' \leq C_w S_s \).

26-6.3.3 Bellows Convolutions.

(a) The circumferential membrane stress due to pressure

(1) for end convolutions of externally attached bellows when \( k \) is less than 1.0

\[ S_{2,E} = \frac{1}{2} \left[ \frac{qD_m + L_t (D_h + nt)}{A + nt L_t E_t} \right] E_t P \]

shall comply with \( S_{2,E} \leq S \);

(2) for intermediate convolutions

\[ S_{2,I} = \frac{1}{2} \frac{qD_m P}{A} \]

shall comply with \( S_{2,I} \leq S \).

(b) The meridional membrane stress due to pressure is given by

\[ S_3 = \frac{w}{2nt_p} P \]

(c) The meridional bending stress due to pressure is given by

\[ S_4 = \frac{1}{2n} \left( \frac{w}{t_p} \right)^2 L_p P \]

(d) The meridional membrane and bending stresses shall comply with

\[ S_3 + S_4 \leq K_m S \]

where

\[ K_m = 1.5Y_{sm} \text{ for as-formed bellows} \]
\[ = 1.5 \text{ for annealed bellows} \]

26-6.4 Instability Due to Internal Pressure

26-6.4.1 Column Instability. The allowable internal design pressure to avoid column instability is given by:

\[ P_{sc} = 0.34 \frac{K_b}{N_q} \]

The internal pressure shall not exceed \( P_{sc} \) : \( P \leq P_{sc} \).

26-6.4.2 In-Plane Instability. The allowable internal design pressure based on in-plane instability is given by

\[ P_{pl} = (\pi - 2) \frac{A S_y^*}{D_m q \sqrt{\alpha}} \]

where

\[ \alpha = 1 + 2\delta^2 + \sqrt{1 - 2\delta^2 + 4\delta^4} \]
\[ \delta = \frac{1}{3} \frac{S_4}{S_{2,I}} \]

and \( S_y^* \) is the effective yield strength at design temperature (unless otherwise specified) of bellows material in the as-formed or annealed conditions.

In the absence of values for \( S_y^* \) in material standards, the following values shall be used:

\[ S_y^* = 2.3S_y \text{ for as-formed bellows} \]
\[ = 0.75S_y \text{ for annealed bellows} \]

where \( S_y \) is the yield strength from UG-23(f) for the bellows material at design temperature.

Higher values of \( S_y^* \) may be used if justified by representative tests.

The internal pressure shall not exceed \( P_{sc} \) : \( P \leq P_{sc} \).

26-6.5 External Pressure Strength

26-6.5.1 External Pressure Capacity. The rules of 26-6.3 shall be applied taking \( P \) as the absolute value of the external pressure.

NOTE: When the expansion bellows is submitted to vacuum, the design shall be performed assuming that only the internal ply resists the pressure. The pressure stress equations of 26-6.3 shall be applied with \( n = 1 \).

26-6.5.2 Instability Due to External Pressure. This design shall be performed according to the rules of UG-28 by replacing the bellows with an equivalent cylinder, using:

(a) an equivalent outside diameter \( D_{eq} \) given by
expanding the tube at fabrication or the interface pressure due to differential thermal expansion may be determined analytically or experimentally.

(-c) Due to differential thermal expansion, the tube may expand less than the tubesheet. For this condition, the interfacial pressure, \( P_r \), is a negative number.

(-d) When the maximum temperature is not determined by (-b) above, or the tube expands less than or equal to the tubesheet, joint acceptability shall be determined by shear load tests described in 3-3. Two sets of specimens shall be tested. The first set shall be tested at the proposed operating temperature. The second set shall be tested at room temperature after heat soaking at the proposed operating temperature for 24 hr. The proposed operating temperature is acceptable if the provisions of 3-5 are satisfied.

(f) The Manufacturer shall prepare written procedures for joints that are expanded (whether welded and expanded or expanded only) for joint strength (see Non-mandatory Appendix HH). The Manufacturer shall establish the variables that affect joint repeatability in these procedures. The procedures shall provide detailed descriptions or sketches of enhancements, such as grooves, serrations, threads, and coarse machining profiles. The Manufacturer shall make these written procedures available to the Authorized Inspector.

(19) A-2 MAXIMUM AXIAL LOADINGS

The maximum allowable axial load in either direction on tube-to-tubesheet joints shall be determined in accordance with the following:

For joint types a, b, b-1, c, d, e,

\[
L_{\text{max}} = A_t S_{df_r}
\]  

(1)

For joint types f, g, h,

\[
L_{\text{max}} = \text{MIN}(A_t S_{df_{rf}} A_t S_o)
\]  

(2)

For joint types i, j, k,

\[
L_{\text{max}} = \text{MIN}(A_t S_{df_{rf}} f_r f_{P} A_t S_o)
\]  

(3)

where

\[
A_t = \text{tube cross-sectional area}
\]

\[
d_i = \text{nominal tube inside diameter}
\]

\[
d_o = \text{nominal tube outside diameter}
\]

\[
E = \text{modulus of elasticity for tubesheet material at } T
\]

\[
E_t = \text{modulus of elasticity for tube material at } T
\]

\[
f_r = \text{factor for the length of the expanded portion of the tube. An expanded joint is a joint between tube and tubesheet produced by applying expanding force inside the portion of the tube to be engaged in the tubesheet. Expanding force shall be set to values necessary to effect sufficient residual interface pressure between the tube and hole for joint strength.}
\]

\[
= \text{MIN}[\{f_{df_r} (1.0)\} \text{ for expanded tube joints without enhancements}]
\]

\[
= 1.0 \text{ for expanded tube joints with enhancements}
\]

\[
f_r = \text{tube joint efficiency, which is set equal to the value of } f_r \text{ (test) or } f_r \text{ (no test)}
\]

\[
f_r \text{ (test)} = \text{tube joint efficiency calculated from results of tests in accordance with A-4 or taken from Table A-2 for tube joints qualified by test, whichever is less, except as permitted in A-3(k)}
\]

\[
f_r \text{ (no test)} = \text{tube joint efficiency taken from Table A-2 for tube joints not qualified by test}
\]

\[
f_{re} = \text{factor for the overall efficiency of welded and expanded joints. This is the maximum of the efficiency of the weld alone, } f_r (b), \text{ and the net efficiency of the welded and expanded joint.}
\]

\[
= \text{MAX}[f_{df_{rf}} (b)]
\]

\[
f_r = \text{factor to account for the increase or decrease of tube joint strength due to radial differential thermal expansion at the tube-to-tubesheet joint}
\]

\[
= (P_o + P_r)/P_o. \text{ Acceptable values of } f_r \text{ may range from 0 to greater than 1. When the } f_r \text{ value is negative, it shall be set to 0.}
\]

\[
f_p = \text{factor for differences in the mechanical properties of tubesheet and tube materials}
\]

\[
= \text{MIN}[\{S_p/S_{yt}\}, (1.0)] \text{ for expanded joints. When } f_p \text{ is less than 0.60, qualification tests in accordance with A-3 and A-4 are required.}
\]

\[
k = 1.0 \text{ for loads due to pressure-induced axial forces}
\]

\[
= 1.0 \text{ for loads due to thermally induced or pressure plus thermally induced axial forces on welded-only joints where the thickness through the weld throat is less than the nominal tube wall thickness } t
\]

\[
= 2.0 \text{ for loads due to thermally induced or pressure plus thermally induced axial forces on all other tube-to-tubesheet joints}
\]

\[
\ell = \text{length of the expanded portion of the tube}
\]

\[
L_{\text{max}} = \text{maximum allowable axial load in either direction on tube-to-tubesheet joint}
\]

\[
P_e = \text{tube expanding pressure. The following equation may be used:}
\]

\[
P_e = S_{yt} (t + \frac{t_o}{S_{yt}}) \left(1.945 - 1.384 \frac{d_i}{d_o}\right)
\]

\[
\text{where } t_o = \text{thickness of the tubesheet, } d_i = \text{inside diameter of the tube, } d_o = \text{outside diameter of the tube, } S_{yt} = \text{yield stress of the tubesheet material at } T
\]
\( P_o \) = interface pressure between the tube and tubesheet that remains after expanding the tube at fabrication. This pressure may be established analytically or experimentally, but shall consider the effect of change in material strength at operating temperature. The following equation may be used:

\[
P_o = P_d \left[ 1 - \left( \frac{d_t}{d_o} \right)^2 \right] - \frac{2}{\sqrt{3}} S_y t \ln \left( \frac{d_o}{d_t} \right)
\]

\( P_T \) = interface pressure between the tube and tubesheet due to differential thermal growth. This pressure may be established analytically or experimentally. The following equation may be used:

\[
P_T = \frac{R_m E_T \left( \frac{d_t}{d_o} \right)^2 \left( T - T_o \right)}{d_t^2 \left( \frac{d_t}{d_o} - R_m \right) + R_m \left( 2.9 \frac{E_T}{E} \right)}
\]

\( R_m \) = mean tube radius
\( r_a \) = tube outside radius
\( S \) = maximum allowable stress value as given in the applicable part of Section II, Part D. For a welded tube or pipe, use the allowable stress for the equivalent seamless product. When the allowable stress for the equivalent seamless product is not available, divide the allowable stress of the welded product by 0.85.

\( S_o \) = allowable stress for tube material
\( = k S \)
\( S_y \) = yield strength from UG-23(l) for tubesheet material at tubesheet design temperature \((T)\)
\( S_{yT} \) = yield strength from UG-23(l) for tube material at tubesheet design temperature \((T)\)

\( T \) = tubesheet design temperature
\( t \) = nominal tube wall thickness
\( T_o \) = ambient temperature
\( \alpha \) = mean coefficient of thermal expansion of tubesheet material at \(T\)
\( \alpha_t \) = mean coefficient of thermal expansion of tube material at \(T\)

A-3 SHEAR LOAD TEST

(a) Flaws in the specimen may affect results. If any test specimen develops flaws, the retest provisions of (k) below shall govern.

(b) If any test specimen fails because of mechanical reasons, such as failure of testing equipment or improper specimen preparation, it may be discarded and another specimen taken from the same heat.

(c) The shear load test subjects a full-size specimen of the tube joint under examination to a measured load sufficient to cause failure. In general, the testing equipment and methods are given in the Methods of Tension Testing of Metallic Materials (ASTM E8). Additional fixtures for shear load testing of tube-to-tubesheet joints are shown in Figure A-3.

(d) The test block simulating the tubesheet may be circular, square or rectangular in shape, essentially in general conformity with the tube pitch geometry. The test assembly shall consist of an array of tubes such that the tube to be tested is in the geometric center of the array and completely surrounded by at least one row of adjacent tubes. The test block shall extend a distance of at least one tubesheet ligament beyond the edge of the peripheral tubes in the assembly.

(e) All tubes in the test block array shall be from the same heat and shall be installed using identical procedures.

(1) The finished thickness of the test block may be less but not greater than the tubesheet it represents. For expanded joints, made with or without welding, the expanded area of the tubes in the test block may be less but not greater than that for the production joint to be qualified.

(2) The length of the tube used for testing the tube joint need only be sufficient to suit the test apparatus. The length of the tubes adjacent to the tube joint to be tested shall not be less than the thickness of the test block to be qualified.

(f) The procedure used to prepare the tube-to-tubesheet joints in the test specimens shall be the same as used for production.

(g) The tube-to-tubesheet joint specimens shall be loaded until mechanical failure of the joint or tube occurs. The essential requirement is that the load be transmitted axially.

(h) Any speed of testing may be used, provided load readings can be determined accurately.

(i) The reading from the testing device shall be such that the applied load required to produce mechanical failure of the tube-to-tubesheet joint can be determined.

(j) For determining \( f_c \) (test) for joint types listed in Table A-2, a minimum of three specimens shall constitute a test. The value of \( f_c \) (test) shall be calculated in accordance with A-4(a) using the lowest value of \( L \) (test). In no case shall the value of \( f_c \) (test) using a three specimen test exceed the value of \( f_c \) (test) given in Table A-2. If the
3-D.1 Yield Strength

Values for the yield strength as a function of temperature are provided in Section II, Part D, Subpart 1, Table Y-1. If the material being used is not listed in this table, while being listed in other tables of Section II Part D Subpart 1, or the specified temperature exceeds the highest temperature for which a value is provided, the yield strength may be determined as below for use in the design equations in Part 4. $S$ is the maximum allowable stress of the material at the temperature specified [see Annex 3-A] and $f$ is the factor (e.g. weld factor) used to determine the allowable stress as indicated in the notes for the stress line. If the value of $f$ is not provided, set $f$ equal to 1.

If the allowable design stress is established based on the 66 2/3% yield criterion, then the yield strength, $S_Y$, shall be taken as $1.5S/f$.

If the allowable design stress is established based on yield criterion between 66 2/3% and 90%, then the yield strength, $S_Y$, shall be taken as $1.1S/f$.

NOTE For temperatures where the allowable stress, $S$, is based on time dependent properties, the yield strength obtained by these formulas may be overly conservative.