(a) The provisions of 3.11.2 are met when using the reduced (colder) operating temperature as the MDMT, but in no case shall the operating temperature be colder than –104°C (–155°F); or

(b) For vessels or components whose thicknesses are based on pressure loading only, the coincident operating temperature may be as cold as the MDMT stamped on the nameplate less the allowable temperature reduction as determined from 3.11.2.5. The ratio used in 3.11.2.5(a), Step 4, of the procedure in 3.11.2.5 shall be the ratio of maximum pressure at the coincident operating temperature to the design pressure of the vessel at the stamped MDMT, but in no case shall the operating temperature be colder than –104°C (–155°F).

3.11.2.8 Establishment of the MDMT Using a Fracture Mechanics Methodology.

(a) In lieu of the procedures in 3.11.2.1 through 3.11.2.7, the MDMT may be established using a fracture mechanics approach. The fracture mechanics procedures shall be in accordance with API 579-1/ASME FFS, Part 9, Level 2 or Level 3.

(b) The assessment used to determine the MDMT shall include a systematic evaluation of all factors that control the susceptibility to brittle fracture, e.g., stresses from the applied loadings including thermal stresses, flaw size, fracture toughness of the base metal and welded joints, heat treatment, and the loading rate.

(c) The reference flaw size used in the fracture mechanics evaluation shall be a surface flaw with a depth of
\[ a = \min\{t/4, 25 \text{ mm (1 in.)}\} \text{ and a length of } 2c = 6a \text{ where } t \text{ is the thickness of the plate containing the reference flaw}. \]

If approved by the user, an alternative reference flaw size may be used based on the weld joint geometry and the NDE that will be used and demonstrated for qualification of the vessel (see Part 7).

(d) The material fracture toughness shall be established using the exemption curve for the material (see Notes to Figures 3.7 and 3.8) and MPC Charpy impact energy correlation described in API 579-1/ASME FFS-1, Appendix F, F.4. If approved by the user, an alternative material fracture toughness may be used based on fracture toughness test results.

(e) The MDMT established using a fracture mechanics approach shall not be colder than that given in 3.11.2.3(e).

3.11.2.9 Postweld Heat Treatment Requirements for Materials in Low Temperature Service.

(a) If the MDMT is colder than –48°C (–55°F) and the stress ratio defined in 3.11.2.5(a), Step 4 is greater than or equal to 0.3 for Class 1, or 0.24 for Class 2, then welded joints shall be subject to PWHT in accordance with the requirements of 6.4.2.

(b) The requirement in (a) above does not apply to the welded joints listed in (1) and (2) below in vessel or vessel parts fabricated of P-No. 1 materials that are impact tested at the MDMT or colder in accordance with 3.11.2.1. The minimum average energy requirement for base metal, weld metal, and heat-affected zones shall be 41 J (30 ft-lb) instead of the values shown in Figure 3.3 for parts not subject to PWHT or Figure 3.4 for parts subject to PWHT or for nonwelded parts.

(1) Type 1 Category A and B joints, not including cone-to-cylinder junctions, that have been 100% radiographed. Category A and B joints attaching sections of unequal thickness shall have a transition with a slope not exceeding 3:1.

(2) Fillet welds having leg dimensions not exceeding 10 mm (3/8 in.) attaching lightly loaded attachments, provided the attachment material and the attachment weld meet the requirements of 3.11.2 and 3.11.8. Lightly loaded attachments, for this application, are defined as attachments in which the stress in the attachment weld does not exceed 25% of the allowable stress. All such welds shall be examined by liquid penetrant or magnetic particle examination in accordance with Part 7 of this Division.

3.11.2.10 Impact Tests of Welding Procedures.

(a) For welded construction, the welding procedure qualification shall include impact testing of weld metals in accordance with 3.11.2.1 when required by (b) and (c).

(b) Welds made with filler metal shall be deposited using welding procedures qualified with impact testing when

(1) either base metal is required to be impact tested by the rules of this Division; or

(2) any individual weld pass exceeds 13 mm (1/2 in.) in thickness and the MDMT is colder than 21°C (70°F); or

(3) joining base metals exempt from impact testing by 3.11.2.3, 3.11.2.4, and 3.11.2.5 when the MDMT is colder than –48°C (–55°F); or

(4) joining base metals from Figure 3.7 or Figure 3.8, Curves C or D, or metals exempted from impact testing by 3.11.2.4(b), and the MDMT is colder than –29°C (–20°F) but not colder than –48°C (–55°F). Qualification of the welding procedure with impact testing is not required when no individual weld pass in the fabrication weld exceeds 6 mm (1/4 in.) in thickness, and each heat and/or lot of filler metal or combination of heat and/or lot of filler metal and batch of flux has been classified by their manufacturer through impact testing per the applicable SFA specification at a temperature not warmer than the MDMT. Additional testing beyond the scope of the SFA specification may be performed by the filler metal and/or flux manufacturer to expand their classification for a broader range of temperatures.

(c) Except for welds made as part of the material specification, welds made without the use of filler metal shall be completed using welding procedures qualified with impact testing when
\[ P_{sox,\text{max}} = \max(0, \text{maximum shell side operating pressure for operating condition } x) \]
\[ P_{sox,\text{min}} = \min(0, \text{minimum shell side operating pressure for operating condition } x) \]
\[ P_t = \text{tube side design or operating pressure, as applicable. For tube side vacuum, use a negative value for } P_t. \]
\[ P_{td,\text{max}} = \text{maximum tube side design pressure} \]
\[ P_{td,\text{min}} = \text{minimum tube side design pressure (negative if vacuum is specified, otherwise zero)} \]
\[ P_{tox,\text{max}} = \max(0, \text{maximum tube side operating pressure for operating condition } x) \]
\[ P_{tox,\text{min}} = \min(0, \text{minimum tube side operating pressure for operating condition } x) \]

\[ W_z = \text{tube-to-tubesheet joint load} \]
\[ W^* = \text{tubesheet effective bolt load determined in accordance with UHX-8} \]
\[ x = 1, 2, 3, \ldots, n, \text{integer denoting applicable operating condition under consideration (e.g., normal operating, start-up, shutdown, cleaning, upset)} \]
\[ ν = \text{Poisson’s ratio of tubesheet material} \]
\[ ν_c = \text{Poisson’s ratio of channel material} \]
\[ ν_s = \text{Poisson’s ratio of shell material} \]
\[ ν_t = \text{Poisson’s ratio of tube material} \]

**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 \( A, C, D_a, D_o, G_a, G_o, G_t \), and thickness \( t_s \) shall be taken from Figure UHX-14.2. For the floating end, diameters \( A, C, D_a, D_o, G_a, G_o, G_t \), and thickness \( t_s \) 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 considered in the design.

The user or his designated agent shall specify all the design and operating conditions that govern the design of the main components of the heat exchanger (i.e., tubesheets, tubes, 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 \( P_s \) 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 \( P_{sd,\text{min}} \) and \( P_{td,\text{min}} \) 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 (3)].

(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, \( P_s \) shall be the tube side design pressure, and \( P_t \) shall be \( P_s \) 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 conditions given in UHX-14.2 must be maintained.
The designer shall consider the effect of deflections in the tube sheet design, especially when the tube sheet thickness \( h \) is less than the tube diameter. 

**UHX-14.4(d)** The designer shall consider the effect of radial differential thermal expansion adjacent to the tube sheet in accordance with UHX-14.6, if required by UHX-14.6.1.

**UHX-14.4(e)** The designer may consider the tube sheet as simply supported in accordance with UHX-14.7.

### UHX-14.5 Calculation Procedure

The procedure for the design of tube sheets for a floating tube sheet heat exchanger is as follows. Calculations shall be performed for both the stationary tube sheet and the floating tube sheet.

**UHX-14.5.1 Step 1.** Determine \( D_w, \mu, \mu^*, \) and \( h'_c \) from UHX-11.5.1.

- **Loading cases 4, 5, 6, and 7:** \( h'_c = 0 \)
- **Calculate** \( a_w, \rho_w, \rho_c, x_w, \) and \( x_c \)
  
  \[ a_w = \frac{D_w}{2} \]
  
  \[ \rho_w = \frac{a_w}{a_c} \]
  
  \[ \rho_c = \frac{a_c}{a_w} \]
  
  \[ x_w = 1 - N \left( \frac{d_w}{2a_w} \right)^2 \]
  
  \[ x_c = 1 - N \left( \frac{d_c - 2a_c}{2a_c} \right)^2 \]

**UHX-14.5.2 Step 2.** Calculate shell coefficients \( \beta_n, k_n, \lambda_n, \) and \( \delta_n. \)

- Configurations a, b, and c:
  
  \[ \beta_n = \frac{\sqrt{12(1 - \nu_c^2)}}{\sqrt{(D_n + t_n) t_n}} \]
  
  \[ k_n = \beta_n \frac{E_c t_n^2}{6(1 - \nu_c^2)} \]
  
  \[ \lambda_n = \frac{6D_n}{h^3} k_n \left( 1 + h \beta_n + \frac{h^2 \beta_n^2}{2} \right) \]
  
  \[ \delta_n = \frac{D_n^2}{4E_c t_n} \left( 1 - \nu_c^2 \right) \]

- Configurations d, e, f, A, B, C, and D: \( \beta_n = 0, k_n = 0, \lambda_n = 0, \delta_n = 0. \)

  - **Calculate channel coefficients** \( \beta_c, k_c, \lambda_c, \) and \( \delta_c. \)
  - **Configurations a, e, f, and A:**
    
    \[ \beta_c = \frac{\sqrt{12(1 - \nu_c^2)}}{\sqrt{(D_c + t_c) t_c}} \]
    
    \[ k_c = \beta_c \frac{E_c t_c^2}{6(1 - \nu_c^2)} \]
    
    \[ \lambda_c = \frac{6D_c}{h^3} k_c \left( 1 + h \beta_c + \frac{h^2 \beta_c^2}{2} \right) \]
    
    \[ \delta_c = \frac{D_c^2}{4E_c t_c} \left( 1 - \nu_c^2 \right) \]

For a cylinder:

\[ \delta_c = \frac{D_c^2}{4E_c t_c} \left( 1 - \nu_c^2 \right) \]

For a hemispherical head:

\[ \delta_c = \frac{D_c^2}{4E_c t_c} \left( 1 - \nu_c^2 \right) \]

- Configurations b, c, d, B, C, and D: \( \beta_c = 0, k_c = 0, \lambda_c = 0, \delta_c = 0. \)

**UHX-14.5.3 Step 3.** Calculate \( h/p. \) If \( \rho \) changes, recalculate \( d^* \) and \( \mu^* \) from UHX-11.5.1.

- **Determine** \( E^*/E \) and \( v^* \) relative to \( h/p \) from UHX-11.5.2.

  **Calculate** \( X_n. \)

  \[ X_n = \left[ 24 (1 - v^*)^{3/2} \frac{E_n t_n (d_n - t_n) a_n^3}{E^* L h^3} \right]^{1/2} \]

Using the calculated value of \( X_n, \) enter either Table UHX-13.1 or Fig. UHX-13.2 to determine \( Z_n, Z_v, \) and \( Z_m. \)

**UHX-14.5.4 Step 4.** Calculate diameter ratio \( K \) and coefficient \( F. \)

\[ K = \frac{A}{D_w} \]

\[ F = \frac{1 - v^*}{E^*} (\lambda_c + \lambda_c + E \ln K) \]

- **Calculate** \( \Phi \) and \( Q_1: \)
  
  \[ \Phi = (1 + v^*) F \]
  
  \[ Q_1 = \frac{\rho_c - 1 - \Phi Z_v}{1 + \Phi Z_m} \]
(a) when \( b_o \leq 1/8 \) in. (6 mm), \( G \) = mean diameter of gasket or joint contact face;

(b) when \( b_o > 1/4 \) in. (6 mm), \( G \) = outside diameter of gasket contact face less \( 2b \)

\[ g_1 = \text{thickness of hub neck at intersection with hub shoulder} \]

\[ g_2 = \text{height of hub shoulder (} g_2 \text{ shall not be larger than} T. \) \]

\[ g_o = \text{thickness of hub neck at small end} \]

\[ \bar{g} = \text{radial distance from the hub inside diameter} \ B \text{ to the hub shoulder ring centroid} \]

\[ H = \text{total hydrostatic end force} = 0.785G^2P \]

\[ h = \text{hub taper length} \]

\[ h_2 = \text{average thickness of hub shoulder} = T - (g_2 \tan \phi)/2 \]

\[ H_D = \text{hydrostatic end force on bore area} = 0.785B^2P \]

\[ h_D = \text{radial distance from effective clamp-hub reaction circle to the circle on which} H_D \text{ acts} = \frac{(C - (B + g_1))/2}{2} \]

\[ H_G = \text{difference between total effective axial clamping preload and the sum of total hydrostatic end force and total joint contact surface compression} = 1.571W/\tan (\phi + \mu) - (H + H_P) \]

\[ h_G = \text{radial distance from effective clamp-hub reaction circle to the circle on which} H_G \text{ acts in. (mm) (for full face contact geometries, } h_G = 0 \) \]

\[ h_m = \text{total axial gasket seating requirements for Make-up (3.14b\bar{g}y) or the axial seating load for self-energygizing gaskets, if significant} \]

\[ h_n = \text{hub neck length [minimum length of } h_n \text{ is 0.5} g_1 \text{ or } 1/4 \text{ in. (6 mm), whichever is larger} \]

\[ h_o = \sqrt{B\bar{g}_o} \]

\[ H_P = \text{total joint contact surface compression load} \]

\[ H_P = 2b \times 3.14GmP \]

(For self-energized gaskets, use \( H_P = 0 \) or actual retarding load if significant.)

\[ H_T = \text{difference between total hydrostatic end force and hydrostatic end force on bore area} = H - H_D \]

\[ h_T = \text{radial distance from effective clamp-hub reaction circle to the circle on which} H_T \text{ acts} = \frac{(C - (B + G))/2}{2} \]

\[ \bar{h} = \text{axial distance from the hub face to the hub shoulder ring centroid} \]

\[ = \frac{T^2g_1 + h_2^2g_2}{2(Tg_1 + h_2g_2)} \]

\[ I_c = \text{moment of inertia of clamp relative to neutral axis of entire section} \]

\[ = \left( \frac{A_1}{3} + \frac{A_2}{4} + \frac{A_3}{3} - A_cX^2 \right) \]

\[ I_h = \text{moment of inertia of hub shoulder relative to its neutral axis} \]

\[ = \frac{g_1}{3} + \frac{g_2}{3} - \left( \frac{g_2h_2 + g_1\bar{h}}{2} \right)^2 \]

\[ L_a = \text{distance from } W \text{ to the point where the clamp lug joins the clamp body [see Figure 24-1 sketch (e)]} \]

\[ I_c = \text{effective clamp lip length} \]

\[ L_h = \text{clamping length [see Figure 24-1 sketch (e)]} \]

\[ l_m = \text{effective clamp lip moment arm} = L_c - (C - T)/2 \]

\[ L_w = \text{clamp lug width [see Figure 24-2]} \]

\[ m = \text{gasket factor from Table 2-5.1} \]

\[ M_D = \text{moment due to} H_D = H_Dh_D \]

\[ M_F = \text{offset moment} = H_D(g_1 - g_o)/2 \]

\[ M_G = \text{moment due to} H_G = H_G\bar{h} \]

\[ M_H = \text{reaction moment at hub neck} \]

\[ M_P = \text{pressure moment} \]

\[ = 3.14 \times PBT \frac{T}{2 - \bar{h}} \]

\[ M_R = \text{radial clamp equilibrating moment} \]

\[ = 1.571W \left( \frac{\bar{h} - T + [(C - N) \tan \phi]/2}{2} \right) \]

\[ M_T = \text{moment due to} H_T \]

\[ = H_TH_T \]

\[ N = \text{outside diameter of hub neck} \]

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

\[ Q = \text{reaction shear force at hub neck} \]

\[ = 1.818M_H / \sqrt{B\bar{g}_1} \]

\[ r = \text{clamp or hub cross section corner radius} \]

\[ = 1/4 \text{ in. (6 mm) min., } C_t \text{ max.} \]

\[ S_1 = \text{hub longitudinal stress on outside at hub neck} \]

\[ S_2 = \text{maximum Lamé hoop stress at bore of hub} \]

\[ S_3 = \text{maximum hub shear stress at shoulder} \]

\[ S_4 = \text{maximum radial hub shear stress in neck} \]

\[ S_5 = \text{clamp longitudinal stress at clamp body inner diameter} \]

\[ S_6 = \text{clamp tangential stress at clamp body outer diameter} \]

\[ S_7 = \text{maximum shear stress in clamp lips} \]

\[ S_B = \text{clamp lug bending stress} \]

\[ S_g = \text{effective bearing stress between clamp and hub} \]

\[ S_a = \text{allowable bolt stress at room temperature} \]

\[ S_{AC} = \text{allowable design stress for clamp material at (assembly condition) room temperature} \]

\[ S_{AH} = \text{allowable design stress for hub material at (assembly condition) room temperature} \]

\[ S_b = \text{allowable bolt stress at design temperature} \]
g₁ = thickness of hub neck at intersection with hub shoulder

\( g₂ = \) height of hub shoulder (\( g₂ \) shall not be larger than \( T \))

\( \bar{g} = \) radial distance from the hub inside diameter \( B \) to the hub shoulder ring centroid

\[ \frac{Tg₁^2 + h₂g₂(2g₁ + g₂)}{2(Tg₁ + h₂g₂)} \]

\( G = \) diameter at location of gasket load reaction. Except as noted in Fig. 24-1, \( G \) is defined as follows (see Table 2-5.2):

(a) when \( bₙ \leq \frac{3}{4} \) in. (6 mm), \( G = \) mean diameter of gasket or joint contact face;

(b) when \( bₙ > \frac{3}{4} \) in. (6 mm), \( G = \) outside diameter of gasket contact face less 2b

\( h = \) hub taper length

\( h_D = \) radial distance from effective clamp-hub reaction circle to the circle on which \( H_D \) acts

\[ = \left[ C - (B + g₁) \right] / 2 \]

\( h_G = \) radial distance from effective clamp-hub reaction circle to the circle on which \( H_G \) acts in. (mm) (for full face contact geometries, \( h_G = 0 \))

\( h_n = \) hub neck length [minimum length of \( h_n \) is 0.5\( g₁ \) or \( \frac{3}{8} \) in. (6 mm), whichever is larger]

\( h_a = \sqrt{Bg₁} \)

\( h_r = \) radial distance from effective clamp-hub reaction circle to the circle on which \( H_r \) acts

\[ = \left[ C - (B + G) \right] / 2 \]

\( h₂ = \) average thickness of hub shoulder

\[ = T - (g₂ \tan θ) / 2 \]

\( \bar{h} = \) axial distance from the hub face to the hub shoulder ring centroid

\[ \frac{Tg₁ + h₂g₂}{2(Tg₁ + h₂g₂)} \]

\( H = \) total hydrostatic end force

\[ = 0.785 G²P \]

\( H_D = \) hydrostatic end force on bore area

\[ = 0.785 B²P \]

\( H_c = \) difference between total effective axial clamping preload and the sum of total hydrostatic end force and total joint contact surface force

\[ = \left[ 1.571 W / \tan (θ + μ) \right] - (H + H_p) \]

\( H_m = \) total axial gasket seating requirements for makeup (3.14\( b₂Gy \) or the axial seating load for self-energizing gaskets, if significant)

\( H_p = \) total joint contact surface compression load

\[ = 2b \times 3.14GmP \]

\( H_r = \) difference between total hydrostatic end force and hydrostatic end force on bore area

\[ = H - H_D \]

\( l_c = \) moment of inertia of clamp relative to neutral axis of entire section

\[ = \left( \frac{A₁}{3} + \frac{A₂}{4} \right) C_r² + \frac{A₅l₅²}{3} - A_cX² \]

\( I_b = \) moment of inertia of hub shoulder relative to its neutral axis

\[ = \frac{g₁T³}{3} + \frac{g₂h₂³}{3} - (g₂h₂ + g₁)T \bar{h}² \]

\( L_w = \) distance from \( W \) to the point where the clamp lug joins the clamp body [see Fig. 24-1 sketch (e)]

\( L_a = \) clamp lug height [see Fig. 24-1 sketch (o)]

\( L_n = \) clamp lug width [see Fig. 24-2]

\( l_c = \) effective clamp lip moment arm

\[ = l_c - (C - C₁)/2 \]

\( m = \) gasket lip moment arm

\( h = \sqrt{Bg₁} \)

\( M_D = \) moment due to \( H_D \)

\[ = H_Dh_D \]

\( M_r = \) offset moment

\[ = H_D(g₁ - g₂)/2 \]

\( M_G = \) moment due to \( H_G \)

\[ = H_Gh_G \]

\( M_R = \) reaction moment at hub neck

\[ = M_o \left[ 1 + \frac{1.818}{\sqrt{Bg₁}} \times \left[ T - \frac{g₂}{g₁(2B + g₁)} \right] \right] \]

\( M_r = \) total rotational moment on hub (see 24-5)

\( M_p = \) pressure moment

\[ = 3.14 \times PBT (T/2 - \bar{h}) \]

\( M_k = \) radial clamp equilibrating moment

\[ = 1.571 W \left( \bar{h} - T + [(C - N) \tan θ] / 2 \right) \]

\( M_T = \) moment due to \( H_T \)

\[ = H_T h_T \]

\( N = \) outside diameter of hub neck

\( P = \) internal design pressure (see UG-21)

\( Q = \) reaction shear force at hub neck

\[ = 1.818 M_h / \sqrt{Bg₁} \]

\( r = \) clamp or hub cross section corner radius

\[ = \frac{3}{8} \) in. (6 mm) min., \( C_r \) max.

\( Sₐ = \) allowable bolt stress at room temperature

\( Sₐ = \) allowable bolt stress at design temperature

\( S_{OH} = \) allowable design stress for hub material at (operating condition) design temperature

\( S_{AH} = \) allowable design stress for hub material at (assembly condition) room temperature

\( S_{OC} = \) allowable design stress for clamp material at (operating condition) design temperature

\( S_{AC} = \) allowable design stress for clamp material at (assembly condition) room temperature

\( S_l = \) hub longitudinal stress on outside at hub neck

\( S₂ = \) maximum Lambie hoop stress at bore of hub