shell thickness, the corner radius on the inside is not less than three times the flange thickness, and the welding meets all the requirements for circumferential joints given in Part UW.

(3) Sketch (b-2). \( C = 0.33m \) but not less than 0.20 for forged circular and noncircular heads integral with or butt welded to the vessel, where the flange thickness is not less than the shell thickness, the corner radius on the inside is not less than the following:

\[
\begin{align*}
  r_{\text{min}} &= 0.375 \text{ in. (10 mm)} \text{ for } t_s \leq 1\frac{1}{2} \text{ in. (38 mm)} \\
  r_{\text{min}} &= 0.25 t_s \text{ for } t_s > 1\frac{1}{2} \text{ in. (38 mm)} \text{ but need not be greater than } \frac{3}{4} \text{ in. (19 mm)}
\end{align*}
\]

The welding shall meet all the requirements for circumferential joints given in Part UW.

(4) Sketch (c). \( C = 0.13 \) for circular heads lap welded or brazed to the shell with corner radius not less than \( 3t \) and \( Y \) not less than required by eq. (1)(6) and the requirements of UW-13 are met.

\[ C = 0.20 \] for circular and noncircular lap welded or brazed construction as above, but with no special requirement with regard to \( Y \).

\[ C = 0.30 \] for circular flanged plates screwed over the end of the vessel, with inside corner radius not less than \( 3t \), in which the design of the threaded joint against failure by shear, tension, or compression, resulting from the end force due to pressure, is based on a factor of safety of at least four, and the threaded parts are at least as strong as the threads for standard piping of the same diameter. Seal welding may be used, if desired.

(5) Sketch (d). \( C = 0.13 \) for integral flat circular heads when the dimension \( d \) does not exceed 24 in. (600 mm), the ratio of thickness of the head to the dimension \( d \) is not less than 0.05 or greater than 0.25, the head thickness \( t_h \) is not less than the shell thickness \( t_s \), the inside corner radius is not less than 0.25\( t \), and the construction is obtained by special techniques of upsetting and spinning the end of the shell, such as employed in closing header ends.

(6) Sketches (e), (f), and (g). \( C = 0.33m \) but not less than 0.20 for circular plates, welded to the inside of the vessel, and otherwise meeting the requirements for the respective types of welded vessels. If a value of \( C \) less than 1 is used in calculating \( t \), the shell thickness \( t_s \) shall be maintained along a distance inwardly from the inside face of the head equal to at least \( 2\sqrt{d t_s} \). The throat thickness of the fillet welds in sketches (e) and (f) shall be at least 0.7\( t_s \). The size of the weld \( t_w \) in sketch (g) shall be not less than 2 times the required thickness of a seamless shell nor less than 1.25 times the nominal shell thickness but need not be greater than the head thickness; the weld shall be deposited in a welding groove with the root of the weld at the inner face of the head as shown in the sketch.

\[ C = 0.33 \] for noncircular plates, welded to the inside of a vessel and otherwise meeting the requirements for the respective types of welded vessels. The throat thickness of the fillet welds in sketches (e) and (f) shall be at least 0.7\( t_s \). The size of the weld \( t_w \) in sketch (g) shall be not less than 2 times the required thickness of a seamless shell nor less than 1.25 times the nominal shell thickness but need not be greater than the head thickness; the weld shall be deposited in a welding groove with the root of the weld at the inner face of the head as shown in the sketch.

(7) Sketch (h). \( C = 0.33 \) for circular plates welded to the end of the shell when \( t_s \) is at least 1.25\( t_c \), and the weld details conform to the requirements of UW-13(e) and Figure UW-13.2 sketches (a) to (g) inclusive. See also UG-93(d)(3).

(8) Sketch (i). \( C = 0.33m \) but not less than 0.20 for circular plates if an inside fillet weld with minimum throat thickness of 0.7\( t_s \) is used and the details of the outside weld conform to the requirements of UW-13(e) and Figure UW-13.2 sketches (a) to (g) inclusive, in which the inside weld can be considered to contribute an amount equal to \( t_s \) to the sum of the dimensions \( a \) and \( b \). See also UG-93(d)(3).

(9) Sketches (j) and (k). \( C = 0.3 \) for circular and noncircular heads and covers bolted to the vessel as indicated in the figures. Note that eq. (c)(2)(2) or (c)(3)(5) shall be used because of the extra moment applied to the cover by the bolting.

When the cover plate is grooved for a peripheral gasket, as shown in sketch (k), the net cover plate thickness under the groove or between the groove and the outer edge of the cover plate shall be not less than

\[ \sqrt{1.9Wh_G/Sd^3} \]

for circular heads and covers, or not less than

\[ \sqrt{6Wh_G/Sld^3} \]

for noncircular heads and covers.

(10) Sketches (m), (n), and (o). \( C = 0.3 \) for a circular plate, being the end of a vessel, having the inside diameter exceeding 12 in. (300 mm); or for heads having an integral flange screwed over the end of a vessel having an inside diameter not exceeding 12 in. (300 mm); and when the design of the threaded joint, against failure by shear,
where,

\[ f_{\text{welds}} = \min \left[ f_{w1} S_n \left( A_2 + A_3 \right) \frac{\pi}{4} PR_n^2 k_p^2 \right] \]  

(4.5.184)

(b) If the nozzle is pad reinforced, and the computed shear stresses in the welds given by Equations (4.5.185) through (4.5.187) satisfy Equation (4.5.188), then the design is complete. If the shear stress in the weld does not satisfy Equation (4.5.188), increase the weld size and return to Step 3.

\[ \tau_1 = \frac{f_{wp}}{L_4 \left( 0.6X_1 + 0.49L_{437} \right)} \]  

(4.5.185)

\[ \tau_2 = \frac{f_{wp}}{L_4 \left( 0.6X_2 + 0.49L_{417} \right)} \]  

(4.5.186)

\[ \tau_3 = \frac{f_{wp}}{t_{\text{wp}} \left( 0.49L_{427} \right)} \]  

(4.5.187)

\[ \max \left[ \tau_1, \tau_2, \tau_3 \right] \leq S \]  

(4.5.188)

where,

\[ f_{ws} = \frac{f_{\text{welds}} - S}{t - S + t_{wp} S_p} \]  

(4.5.189)

\[ f_{wp} = \frac{f_{\text{welds}} + S}{t - S + t_{wp} S_p} \]  

(4.5.190)

4.5.14.3 The procedure to evaluate attachment welds of a nozzle in a flat head subject to pressure loading is shown below.

Step 1. Compute the weld throat dimensions, as applicable.

\[ L_{41T} = 0.7071L_{41} \]  

(4.5.191)

\[ L_{42T} = 0.7071L_{42} \]  

(4.5.192)

\[ L_{43T} = 0.7071L_{43} \]  

(4.5.193)

Step 2. Determine if the weld sizes are acceptable.

(a) If the nozzle is integrally reinforced and set-in the flat head, and the computed shear stress in the welds given by Equations (4.5.194) through (4.5.196) satisfy Equation (4.5.197), then the design is complete. If the shear stress in the welds does not satisfy Equation (4.5.197), increase the weld size and return to Step 1.

\[ \tau_1 = \frac{V_g}{0.6X_1 + 0.49L_{437T}} \]  

(4.5.195)

\[ \tau_2 = \frac{V_g}{0.6X_2 + 0.49L_{417T}} \]  

(4.5.196)

\[ \tau_3 = \frac{P \left( R_n + t_0 \right)}{2 \left( 0.49L_{417T} + 0.6X_1 + 0.49L_{437T} \right)} \]  

(4.5.197)

\[ \max \left[ \tau_1, \tau_2, \tau_3 \right] \leq S \]  

(4.5.197)
\[ H = 0.785G^2P \]
\[ H_T = H - H_D \]
\[ H_G = W_0 - H \]

Step 6. Determine the flange moment for the operating condition using Equation (4.16.14) or Equation (4.16.15), as applicable. When specified by the user or his designated agent, the maximum bolt spacing \( B_{max} \) and the bolt spacing correction factor \( B_{corr} \) shall be applied in calculating the flange moment for internal pressure using the equations in Table 4.16.11. The flange moment \( M_n \) for the operating condition and flange moment \( M_{sa} \) for the gasket seating condition without correction for bolt spacing \( B_{sc} = 1 \) is used for the calculation of the rigidity index in Step 10. In these equations, \( h_D, h_T, \) and \( h_G \) are determined from Table 4.16.6. For integral and loose type flanges, the moment \( M_{sa} \) is calculated using Equation (4.16.16) where \( I \) and \( I_p \) in this equation are determined from Table 4.16.7. For reverse type flanges, the procedure to determine \( M_{sa} \) shall be agreed upon between the Designer and the Owner.

\[ M_0 = \text{abs} \left[ \left( H_D h_D + H_T h_T + H_G h_G \right) B_{sc} + M_{sa} \right] F_s \]  
for internal pressure

\[ M_0 = \text{abs} \left[ \left( H_D (h_D - h_G) + H_T (h_T - h_G) \right) B_{sc} + M_{sa} \right] F_s \]  
for external pressure
Step 5. Calculate the following quantities.

(a) $\gamma$ for the operating loading cases. For the design loading cases, $\gamma = \mathcal{C}$.

\[
\gamma = \left[ q_{t,m}(r_{t,m} - r_\sigma) - q_{s,m}(r_{s,m} - r_\sigma) \right] \mathcal{L}
\]  

(b) $\omega_s$, $\omega_s^*$, $\omega_c$, and $\omega_c^*$.

\[
\omega_s = \rho_s k_s \beta_s \left[ 1 + h \beta_s \right]
\]  
\[
\omega_s^* = \frac{a_s^2 (\rho_s^2 - 1) [\rho_s - 1]}{4} - \omega_s
\]  
\[
\omega_c = \rho_s k_s \beta_c \left[ 1 + h \beta_c \right]
\]  
\[
\omega_c^* = a_c^2 \left[ \frac{(\rho_c^2 + 1) [\rho_c - 1]}{4} - \frac{(\rho_c - 1)}{2} \right] - \omega_c
\]

(c) $\gamma_b$

\[
y_b = 0 \quad \text{Configuration – } \alpha
\]  
\[
y_b = \frac{c_c - c_1}{d_b} \quad \text{Configuration – } \beta
\]  
\[
y_b = \frac{c_c - c_2}{d_b} \quad \text{Configuration – } \gamma
\]  
\[
y_b = \frac{c_c - c_2}{d_b} \quad \text{Configuration – } \delta
\]

Step 6. For each loading case, calculate the effective pressure.

\[
P_e = \frac{J\mathcal{C}_{s,t} \left[ P'_s + P_s + P_{\text{prim}} \right]}{1 + J \mathcal{C}_{z,s} \mathcal{Q}_{z_1} + (\rho_s - 1) \mathcal{Q}_{z_2}}
\]  

where

\[
P'_s = \left( x_s + 2(1 - x_s) \right) \mathcal{Q}_s + \frac{2}{\mathcal{K}_{s,t}} \left( \frac{\mathcal{Q}_s}{\mathcal{D}_s^2} \right)^2 \mathcal{Q}_s - \frac{\rho_s^2 - 1}{J \mathcal{K}_{s,t}} \left[ \frac{D_s^2 - D_t^2}{2 \mathcal{D}_s^2} \right] P_s
\]

\[
P'_t = \left( x_t + 2(1 - x_t) \right) \mathcal{Q}_t + \frac{1}{J \mathcal{K}_{s,t}} P_t
\]

\[
P = \frac{N_k K_{D_s}}{\pi a_d^2}
\]