Volume 2012, Issue 9 September 15, 2012

PV Newsletter

Monthly Publication from CoDesign Engineering Skills Academy

ASME Section VIII, Div. 2: Design by Rule Requirements

The User is responsible for defining all applicable loads and conditions acting on the pressure vessel that affect its design. These loads and conditions are specified in the **User's Design Specification**. The manufacturer cannot include any loadings or conditions in the design that are not defined in the UDS.

General Requirements

Part 4 of the ASME VIII-2 provides rules for commonly used pressure vessel shapes under pressure loading; however, it does not provide rules to cover all loadings, geometries and details. When design rules are not provided, a stress analysis in accordance with Part 5 is mandatory considering all of the loadings specified in the UDS.

A screening criterion in accordance with paragraph 5.5.2 shall be applied to all pressure parts to determine if fatigue analysis is required. If fatigue analysis is required as a result of this screening, then the analysis shall be performed in accordance with paragraph 5.5.2.

Thickness Requirements

The minimum thickness permitted for shells and heads after forming shall be 1.6 mm (0.0625 in) exclusive of any corrosion allowance. Exceptions are:

a) Heat transfer plates of plate-type heat exchangers
b) Inner pipe of double pipe heat exchangers, or pipe and tubes that are enclosed and protected by a shell, casing or ducting, where such pipes or tubes are DN 150 (NPS 6) and less
c) Shells and heads used in compressed air service, steam service or water service, made from carbon or low alloy steel materials. **These shall be at least 2.4 mm (0.0938 in) exclusive of any corrosion allowance**
d) Air cooled heat exchangers and cooling tower heat exchangers, provided all of the following provisions are met:
    1) Tubes shall be protected by fins or other means
    2) Tube OD shall be a minimum of 10 mm (0.375 in) and a maximum of 38 mm (1.5 in)
    3) The minimum thickness used shall not be less than that calculated by equations in this Part, and in no case less than 0.5 mm (0.022 in)

The selected thickness of material shall be such that forming, heat treatment and other fabrication processes will not reduce the thickness of material at any point below the minimum required thickness. Plate material shall be ordered not thinner than the minimum required thickness. Vessels made of plate furnished with an under-tolerance of not more than the smaller value of 0.3 mm (0.01 in) or 6% of the ordered thickness may be used at the full maximum allowable working pressure for the thickness ordered. If a pipe or tube is ordered by its nominal wall thickness, the manufacturing under-tolerance on wall thickness shall be taken into account. After the minimum wall thickness is determined, it shall be increased by an amount sufficient to provide the under-tolerance allowed in the specification.

Design Basis

The design thickness of the vessel part is determined using the methods of this Part with load and load combinations specified in Tables 1 and 2.

Table 1: Design Loads

Design Load Parameter	Description
P	Internal or External Specified Design Pressure (see paragraph 4.1.5.2.a)
P_s	Static head from liquid or bulk materials (e.g. catalyst)
D	Dead weight of the vessel, contents, and appurtenances at the location of interest, including the following: • Weight of vessel including internals, supports (e.g. skirts, lugs, saddles, and legs), and appurtenances (e.g. platforms, ladders, etc.) • Weight of vessel contents under operating and test conditions • Refractory linings, insulation • Static reactions from the weight of attached equipment, such as motors, machinery, other vessels, and piping
L	• Appurtenance Live loading • Effects of fluid flow, steady state or transient • Loads resulting from wave action
E	Earthquake loads (see ASCE 7 for the specific definition of the earthquake load, as applicable)
W	Wind Loads
S	Snow Loads
F	Loads due to Deflagration

Table 2: Design Load Combinations

Design Load Combination (1)	General Primary Membrane Allowable Stress (2)
$P+P_s+D$	S
$P+P_s+D+L$	S
$P+P_s+D+S$	S
$0.9P+P_s+D+0.75L+0.75S$	S
$0.9P+P_s+D+(W\,or\,0.7E)$	S
$0.9P+P_S+D+0.75(W\,or\,0.7E)+0.75L+0.75S$	S
$0.6D+(W\,or\,0.7E)$ (3)	S
P_s+D+F	See Annex 4.D

Notes
1) The parameters used in the Design Load Combination column are defined in Table 4.1.1.
2) S is the allowable stress for the load case combination (see paragraph 4.1.5.3.c)
3) This load combination addresses an overturning condition. If anchorage is included in the design, consideration of this load combination is not required.

Combinations of loads that result in a maximum thickness shall be evaluated. If the vessel or a part is subjected to cyclic operation and a fatigue analysis is required, then a pressure cycle histogram and corresponding thermal cycle histogram shall be provided in the UDS.

Design Allowable Stress

Allowable stresses for all permissible materials of construction are provided in Annex 3.A.The wall thickness of a vessel for any combination of loads listed in Table 2 that induces primary stress and are expected to occur simultaneously during operation shall satisfy the following equations:

$P_m \le S$
$P_m + P_b \le 1.5S$

When hydrostatic test is performed, the test pressure shall not exceed that value which results in the following equivalent stress limits:

$P_m \le 0.95S_y$

$P_m + P_b \le 1.43S_y$ for $P_m \le 0.67S_y$

$P_m + P_b \le (2.43S_y - 1.5P_m)$ for $0.67S_y < P_m \le 0.95S_y$

When pneumatic test is performed, the test pressure shall not exceed that value which results in the following equivalent stress limits:

$P_m \le 0.80S_y$

$P_m + P_b \le 1.20S_y$ for $P_m \le 0.67S_y$

$P_m + P_b \le (2.20S_y - 1.5P_m)$ for $0.67S_y < P_m \le 0.8S_y$

Combination Units

Sometimes a pressure vessel may consist of more than one independent pressure chamber, operating at the same or different pressure and temperatures. The parts separating each independent pressure chamber are called common elements. Each common element shall be designed for the most severe condition of coincident pressure and temperature expected in normal operation.

The common elements may be designed for a differential pressure less than the maximum of the design pressures of its adjacent chambers (differential pressure design) or a mean metal temperature less than the maximum of the design temperatures of its adjacent chambers (mean metal temperature design) or both. Such a design is only permitted when the vessel is installed in a system that controls the common element operating conditions. The common element and its corresponding differential pressure and/or corresponding design temperature shall be indicated in the "Remarks" section of the Manufacturer's Data Report and marked on the vessel.

Welded Joints

The term weld category defines the location of a joint in a vessel, but not the weld joint type. The weld categories are defined in Table 4.2.1 of the Code and reproduced below as Table 3.

Weld Category	Description
A	• Longitudinal and spiral welded joints within the main shell, communicating chambers (1), transitions in diameter, or nozzles • Any welded joint within a sphere, within a formed or flat head, or within the side plates (2) of a flat-sided vessel • Circumferential welded joints connecting hemispherical heads to main shells, to transitions in diameter, to nozzles, or to communicating chambers.
B	• Circumferential welded joints within the main shell, communicating chambers (1), nozzles or transitions in diameter including joints between the transition and a cylinder at either the large or small end • Circumferential welded joints connecting formed heads other than hemispherical to main shells, to transitions in diameter, to nozzles, or to communicating chambers.
C	• Welded joints connecting flanges, Van Stone laps, tubesheets or flat heads to main shell, to formed heads, to transitions in diameter, to nozzles, or to communicating chambers (1) • Any welded joint connecting one side plate (2) to another side plate of a flat-sided vessel.
D	• Welded joints connecting communicating chambers (1) or nozzles to main shells, to spheres, to transitions in diameter, to heads, or to flat-sided vessels • Welded joints connecting nozzles to communicating chambers (1) (for nozzles at the small end of a transition in diameter see Category B).
E	• Welded joints attaching nonpressure parts and stiffeners

Notes:

1. Communicating chambers are defined as appurtenances to the vessel that intersect the shell or heads of a vessel and form an integral part of the pressure containing enclosure, e.g., sumps.
2. Side plates of a flat-sided vessel are defined as any of the flat plates forming an integral part of the pressure containing enclosure.

Table 3:

Definitions of Welded Categories

The weld joint type defines the type of weld between components. The definitions for weld joint types are shown in Table 4.2.2 of the Code and reproduced below as Table 4.

Weld Joint Type	Description
1	Butt joints and angle joints where the cone half-apex angle is less than or equal to 30 degrees produced by double welding or by other means which produce the same quality of deposited weld metal on both inside and outside weld surfaces. Welds using backing strips which remain in place do not qualify as Type No.1 butt joints.
2	Butt joints produced by welding from one side with a backing strip that remains in place.
3	Butt joints produced by welding from one side without a backing strip.
7	Corner joints made with full penetration welds with or without cover fillet welds
8	Angle joints made with a full penetration weld where the cone half-apex angle is greater than 30 degrees
9	Corner joints made with partial penetration welds with or without cover fillet welds
10	Fillet welds

Table 4:

Definition of Weld Joint Types

Various types of joints that are permitted are as follows:

a) Butt joint
b) Corner joint
c) Angle joint
d) Spiral weld
e) Fillet weld

The definitions for these are listed in paragraph 4.2.5.1. Not all joints are permitted for all categories.

All Category A weld joints shall be Type No. 1 butt joints. The acceptable Category A welds are shown in Tables 4.2.4 and 4.2.5 of the Code. Other requirements for welds in Category A locations are given in paragraph 4.2.5.2.

All Category B weld joints shall be either 1) Type No. 1 butt joints, or 2) Type No. 2 butt joints except as limited in paragraph 4.2.5.7 for quenched and tempered high strength steels, or 3) Type No. 3 butt joints. Type No 3 butt joints may only be used for shells having a thickness of 16 mm (0.625 in) or less and diameter of 610 mm (24 in) and less. Acceptable Category B welds are shown in Tables 4.2.4 and 4.2.5. Other requirements for welds in Category B locations are given in paragraph 4.2.5.3.

All Category C weld joints shall be either 1) Type No. 1 butt joints, or 2) full penetration corner joints except as limited in paragraph 4.2.5.7 for quenched and tempered high strength steels, or 3) fillet welded joints for the attachment of loose type flanges shown in Table 4.2.9 of the Code with some limitations. The acceptable Category C welds are shown in Tables 4.2.4, 4.2.6, 4.2.7, 4.2.8 and 4.2.9. The limitations for fillet welded joints and other requirements for welds in Category C locations are given in paragraph 4.2.5.4.

All Category D weld joints shall be either 1) Type No. 1 butt joints, or 2) full penetration corner joints except as limited in paragraph 4.2.5.7 for quenched and tempered high strength steels, or 3) full penetration corner joints at the nozzle neck or fillet welds or both, or 4) partial penetration corner joint at the nozzle neck. The acceptable Category D welds are shown in Tables 4.2.4, 4.2.10, 4.2.11, 4.2.12, 4.2.13 and 4.2.14. Other requirements for welds in Category D locations are given in paragraph 4.2.5.5.

Category E welds are for attaching non-pressure parts and stiffeners. The requirements for welds in Category E locations are given in paragraph 4.2.5.6.

Shells under Internal Pressure

Paragraph 4.3 provides rules for determining the required wall thickness of cylindrical, conical, spherical, torispherical, and ellipsoidal shells and heads subject to internal pressure. Tolerances for shells and formed heads are given in paragraph 4.3.2; shells that do not meet these tolerance requirements may be evaluated using paragraph 4.14.

Cylindrical Shells

$$t = \frac{D}{2}\left(exp\left[\frac{P}{SE}\right] - 1\right)$$

where,
D = Inside diameter of shell
P = Internal design pressure
S = Allowable stress value from Annex 3.A
E = Weld joint factor

Conical Shells

$$t = \frac{D}{2cos[\propto]}\left(exp\left[\frac{P}{SE}\right] - 1\right)$$

where,
D = Inside diameter of shell
P = Internal design pressure
S = Allowable stress value from Annex 3.A
E = Weld joint factor
α = One-half of the apex angle

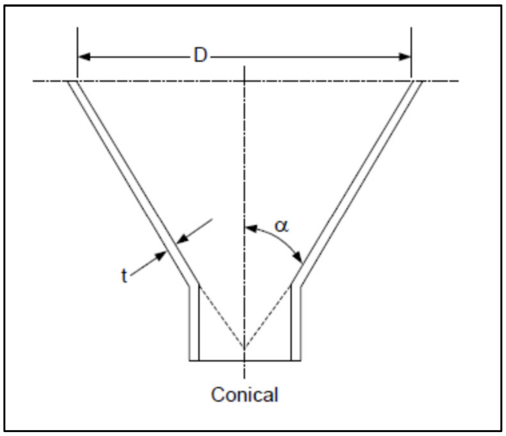


Figure 1: Conical Shell

The rules regarding the cylindrical-to-conical shell transition junctions are given in paragraphs 4.3.11 and 4.3.12.

Spherical Shells and Heads

$$t = \frac{D}{2}\left(exp\left[\frac{0.5P}{SE}\right] - 1\right)$$

where,
D = Inside diameter of shell
P = Internal design pressure
S = Allowable stress value from Annex 3.A
E = Weld joint factor

Torispherical Heads

Part 4 has different procedures for calculating minimum thickness of torispherical heads under internal pressure when crown and knuckle thicknesses are the same, and when crown and knuckle thicknesses are different. We will only discuss the procedure where the crown and knuckle thicknesses are the same. Refer to Figure 2.

STEP 1 - Determine the inside diameter D, and assume values for the crown radius L, the knuckle radius r, and the wall thickness t.

STEP 2 - Compute the ratios L/D, r/D and L/t, and determine if the following equations are satisfied. If the equations are satisfied, then proceed to STEP 3; otherwise the head shall be designed in accordance with Part 5.

$0.7 \leq L/D \leq 1.0$
$r/D \geq 0.06$
$20 \leq L/t \leq 2000$

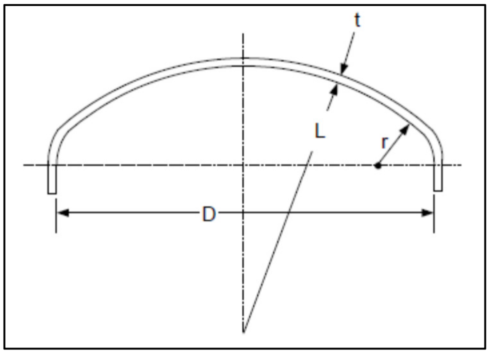


Figure 2: Torispherical Head of Uniform Thickness

STEP 3 - Calculate the following geometric constants.

$$\beta_{th} = arccos\left[\frac{0.5D-r}{L-r}\right], radians$$

$$\emptyset_{th} = \frac{\sqrt{Lt}}{r}, radians$$

$$R_{th} = \frac{0.5D-r}{cos[\beta_{th}-\emptyset_{th}]} + r \quad \text{for} \quad \emptyset_{th} < \beta_{th}$$

$$R_{th} = 0.5D \quad \text{for} \quad \emptyset_{th} \geq \beta_{th}$$

STEP 4 - Compute the coefficients C_1 and C_2 using the following equations

$$C_1 = 9.31\left(\frac{r}{D}\right) - 0.086 \quad \text{for} \quad \frac{r}{D} \leq 0.08$$

$$C_1 = 0.692\left(\frac{r}{D}\right) + 0.605 \quad \text{for} \quad \frac{r}{D} > 0.08$$

$$C_2 = 1.25 \quad \text{for} \quad \frac{r}{D} \leq 0.08$$

$$C_2 = 1.46 - 2.6\left(\frac{r}{D}\right) \quad \text{for} \quad \frac{r}{D} > 0.08$$

STEP 5 - Calculate the value of internal pressure expected to produce elastic buckling of the knuckle.

$$P_{eth} = \frac{C_1 E_t t^2}{C_2 R_{th}\left[\frac{R_{th}}{2} - r\right]}$$

STEP 6 - Calculate the value of internal pressure that will result in a maximum stress in the knuckle equal to the material yield strength.

$$P_y = \frac{C_3 t}{C_2 R_{th}\left[\frac{R_{th}}{2r} - 1\right]}$$

If the allowable stress at the design temperature is governed by time-independent properties, then C_3 is the material yield strength at the design temperature, or $C_3 = S_y$. If the allowable stress at the design temperature is governed by time-dependent properties, then C_3 is determined as follows:

1) If the allowable stress is established based on 90% yield criterion, then C_3 is the material allowable stress at the design temperature multiplied by 1.1, or $C_3 = 1.1S$.
2) If the allowable stress is established based on 67% yield criterion, then C_3 is the material allowable stress at the design temperature multiplied by 1.5, or $C_3 = 1.5S$.

STEP 7 - Calculate the value of internal pressure expected to result in a buckling failure of the knuckle.

$$P_{ck} = 0.6P_{eth} \quad \text{for} \quad G \leq 1.0$$

$$P_{ck} = \left(\frac{0.77508G - 0.20354G^2 + 0.019274G^3}{1 + 0.19014G + 0.089534G^2 + 0.0093965G^3}\right)P_y \quad \text{for} \quad G > 1.0$$

where, $G = \frac{P_{eth}}{P_y}$

STEP 8 - Calculate the allowable pressure based on a buckling failure of the knuckle.

$$P_{ak} = \frac{P_{ck}}{1.5}$$

STEP 9 - Calculate the allowable pressure based on rupture of the crown.

$$P_{ac} = \frac{2SE}{\frac{L}{t}+0.5}$$

STEP 10 - Calculate the maximum allowable internal pressure.

$$P_a = min[P_{ak}, P_{ac}]$$

STEP 11 - If the allowable internal pressure from STEP 10 is greater than or equal to the design pressure, then the design is complete. If not, then increase the head thickness, and repeat STEPS 2 through 10.

Ellipsoidal Heads

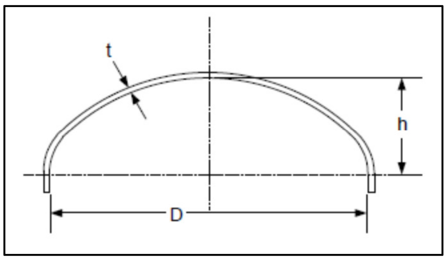


Figure 3: Ellipsoidal Head

The minimum required thickness of an ellipsoidal head subjected to internal pressure shall be calculated using the procedures for the torispherical head having uniform thickness for crown and knuckle sections. The following substitutions for r and L should be made:

$$r = D\left(\frac{0.5}{k} - 0.08\right)$$

$$L = D(0.44k + 0.02)$$

where, $k = \frac{D}{2h}$

The rules in this paragraph are applicable for elliptical heads that satisfy the following equation:

$$1.7 \leq k \leq 2.2$$

Elliptical heads that do not satisfy this equation shall be designed using Part 5.

Shells under External Pressure and Allowable Compressive Stresses

Paragraph 4.4 gives rules for determining the required wall thickness of cylindrical, conical, spherical, torispherical and ellipsoidal shells and heads subject to external pressure. The equations in this paragraph are applicable for $D_o/t \leq 2000$. If $D_o/t > 2000$, the design shall be in accordance with Part 5.

The allowable stresses are determined by applying a design factor, FS, to the predicted buckling stresses. The required values of FS are 2.0 when the buckling stress is elastic and 1.67 when the predicted buckling stress equals the minimum specified yield strength at the design temperature. A linear variation shall be used between these limits. In the equations below for FS, F_{ic} is the predicted buckling stress that is determined by setting FS = 1.0 in the allowable stress equations.

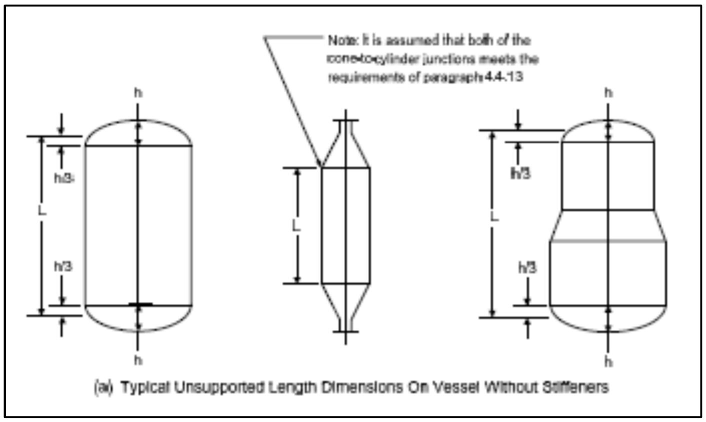
FS = 2.0	for	$F_{ic} \leq 0.55\ S_y$
FS = 2.407 - 0.741(F_{ic}/S_y)	for	$0.55\ S_y < F_{ic} < S_y$
FS = 1.667	for	$F_{ic} = S_y$

The equations for the allowable compressive stress are based on carbon and low alloy steel plate materials as given in part 3. For other materials, a modification to the allowable stress is required. The procedure for modification of the allowable stress is given in paragraph 4.4.3.1.

Cylindrical Shell

The required thickness of a cylindrical shell subject to external pressure shall be determined using the following procedure:

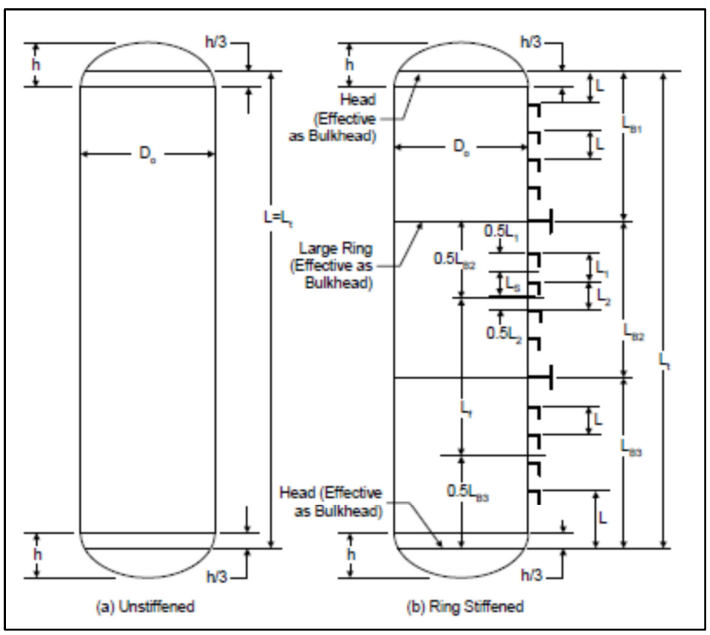
STEP 1- Assume an initial thickness, t, and unsupported length, L



(a) Typical Unsupported Length Dimensions On Vessel Without Stiffeners

Figure 4:

Lines of Supports for Typical Vessel Configurations



(a) Unstiffened (b) Ring Stiffened

Figure 5:

Lines of Supports for Unstiffened and Stiffened Cylindrical Shells

STEP 2 - Calculate the predicted elastic buckling stress, F_{he}

$$F_{he} = \frac{1.6 C_h E_y t}{D_o}$$

$$M_x = \frac{L}{\sqrt{R_o t}}$$

$$C_h = 0.55\left(\frac{t}{D_o}\right) \quad \text{for} \quad M_x \geq 2\left(\frac{D_o}{t}\right)^{0.94}$$

$C_h = 1.12 M_x^{-1.058}$ for $13 < M_x < 2\left(\frac{D_o}{t}\right)^{0.94}$

$C_h = \frac{0.92}{M_x - 0.579}$ for $1.5 < M_x \leq 13$

$C_h = 1.0$ for $M_x \leq 1.5$

STEP 3 - Calculate the predicted buckling stress, F_{ic}

$F_{ic} = S_y$ for $\frac{F_{he}}{S_y} \geq 2.439$

$F_{ic} = 0.7 S_y \left(\frac{F_{he}}{S_y}\right)^{0.4}$ for $0.552 < \frac{F_{he}}{S_y} < 2.439$

$F_{ic} = F_{he}$ for $\frac{F_{he}}{S_y} \leq 0.552$

STEP 4 - Calculate the value of design factor, FS, per paragraph 4.4.2

STEP 5 - Calculate the allowable external pressure, P_a

$P_a = 2F_{ha}\left(\frac{t}{D_o}\right)$ where, $F_{ha} = \frac{F_{ic}}{FS}$

STEP 6 - If the allowable external pressure, P_a, is less than the design external pressure, increase the shell thickness or reduce the unsupported length of the shell (i.e. by addition of stiffening rings) and repeat from STEP 2.

Stiffening Ring Size

A combination of large and small stiffening rings may be used along the length of a shell. If a single size stiffener is used, then it shall be sized as a small stiffener. Alternatively, a combination of large and small stiffeners can be used to reduce the size of intermittent small stiffening rings. The large stiffening rings may be sized to function as end stiffeners, with small stiffeners spaced as required between end rings based on the shell thickness selected and loading combinations considered in the design.

Openings in Shells and Heads

The rules in paragraph 4.5 are applicable for design of nozzles in shells and heads. Configurations and/or loading conditions that do not satisfy the rules of this paragraph must be designed in accordance with Part 5. These design rules shall only be used if the ratio of the inside diameter of the shell and the shell thickness is less than or equal to 400. In addition, the ratio of the diameter along the major axis to the diameter along the minor axis of the finished nozzle opening shall be less than or equal to 1.5.

Nozzles may be attached to the shell or head of a vessel by the following methods:

a) Welded connections
b) Studded connections
c) Threaded connections
d) Expanded connections

Radial Nozzle in a Cylindrical Shell

STEP 1 - Calculate the limit of reinforcement along the vessel wall:

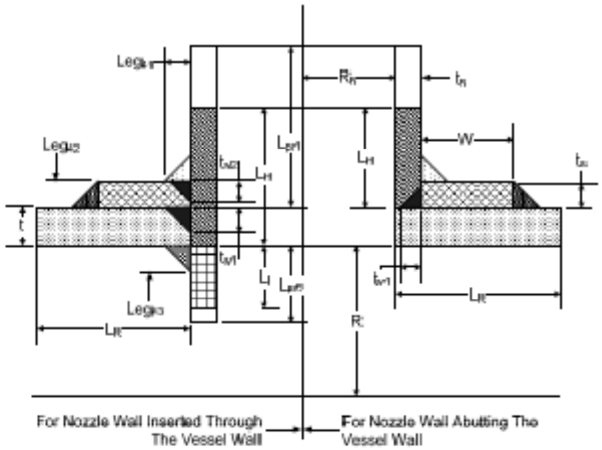
1) $L_R = min\left[\sqrt{R_{eff}t}, 2R_n\right]$ for integrally reinforced nozzles

2) $L_{R1} = \sqrt{R_{eff}t} + W$ for nozzles with reinforcing pads

$L_{R2} = \sqrt{\left(R_{eff} + t\right)(t + t_e)}$

$L_{R3} = 2R_n$

$L_R = min[L_{R1}, L_{R2}, L_{R3}]$



Determine the contributing areas as applicable where L_R, L_H and L_I are determined from the procedures in paragraphs 4.5.5 and 4.5.11.

= A_1 = Area contributed by shell

= A_2 = Area contributed by nozzle projecting outward

= A_3 = Area contributed by nozzle projecting inward

= A_{41} = Area contributed by outward weld

= A_{42} = Area contributed by pad to vessel weld

= A_{43} = Area contributed by inward weld

= A_5 = Area contributed by reinforcing pad

A_T = Total area contributed

Notes: 1. Do not include any area that falls outside of the limits defined by L_H, L_R, and L_I. For example, if $W \geq L_R$, then $W = L_R$ and $A_{42} = 0.0$.
2. In accordance paragraph 4.1.4.1, all dimensions are in the corroded condition.

Figure 6:

Nomenclature for Reinforced Openings

STEP 2 - Calculate the limit of reinforcement along the nozzle wall projecting outside the vessel surface:

$$L_{H1} = t + t_e + \sqrt{R_n t_n}$$

$$L_{H2} = L_{pr1} + t$$

[for nozzles inserted through the vessel wall]

$$L_{H2} = L_{pr1}$$

[for nozzles abutting the vessel wall]

$$L_{H3} = 8(t + t_e)$$

$$L_H = min[L_{H1}, L_{H2}, L_{H3}]$$

STEP 3 - Calculate the limit of reinforcement along the nozzle wall projecting inside the vessel surface

$$L_{I1} = \sqrt{R_n t_n}$$

$$L_{I2} = L_{pr2}$$

$$L_{I3} = 8(t + t_e)$$

$$L_I = min[L_{I1}, L_{I2}, L_{I3}]$$

STEP 4 - Determine the total available area near the nozzle opening

$A_T = A_1 + f_{rn}(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{rp}A_5$ [Please refer to paragraph 4.5.5.1 for formulas For various terms]

STEP 5 - Determine the effective radius of the shell as follows:

$$R_{eff} = 0.5\ D_1$$

STEP 6 - Determine the effective shell thickness for nozzles in cylindrical shells as follows:

$$t_{eff} = t\left(\frac{tL_R + A_5 f_{rp}}{tL_R}\right)$$

STEP 7 - Determine the applicable forces

$$f_N = PR_{xn}(L_H - t)$$

$$f_S = PR_{xs}(L_R + t_n)$$

$$f_Y = PR_{xs}R_{nc}$$

$$R_{xn} = \frac{t_n}{ln\left[\frac{R_n+t_n}{R_n}\right]}$$

$$R_{xs} = \frac{t_{eff}}{ln\left[\frac{R_{eff}+t_{eff}}{R_{neff}}\right]}$$

STEP 8 - Determine the average local primary membrane stress and the general primary membrane stress in the vessel:

$$\sigma_{avg} = \frac{(f_N+f_S+f_Y)}{A_T}$$

$$\sigma_{circ} = \frac{PR_{xs}}{t_{eff}}$$

STEP 9 - Determine the maximum local primary membrane stress at the nozzle intersection:

$$P_L = max\left[\left(2\sigma_{avg} - \sigma_{circ}\right), \sigma_{circ}\right]$$

STEP 10 - The calculated maximum local primary membrane stress should satisfy the equation below:

$P_L \leq S_{allow}$ where $S_{allow} = 1.5$ SE for internal pressure

$S_{allow} = F_{ha}$ for external pressure

STEP 11 - Determine the maximum allowable working pressure of the nozzle:

$$P_{max1} = \frac{S_{allow}}{\frac{2A_p}{A_T} - \frac{R_{xs}}{t_{eff}}}$$

$$P_{max2} = S\left(\frac{t}{R_{xs}}\right)$$

$P_{max} = min[P_{max1}, P_{max2}]$ where $A_p = R_{xn}(L_H - t) + R_{xs}(L_R + t_n + R_{nc})$

Spacing Requirements for Nozzle

If the limits of reinforcement for nozzles do not overlap, no additional analysis is required. If the limits of reinforcement overlap, the maximum local primary membrane stress and the nozzle maximum allowable working pressure shall be determined for each nozzle with the value of L_R determined as follows:

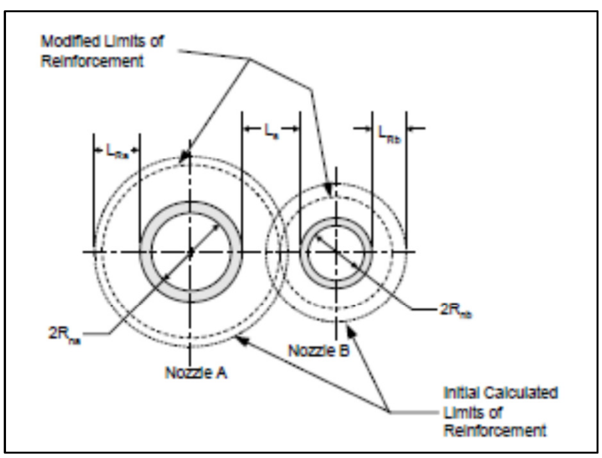


Figure 7: Example of Two Adjacent Nozzle Openings

$L_R = L_S\left(\frac{R_{na}}{R_{na}+R_{nb}}\right)$ for nozzle A

$L_R = L_S\left(\frac{R_{nb}}{R_{na}+R_{nb}}\right)$ for nozzle B

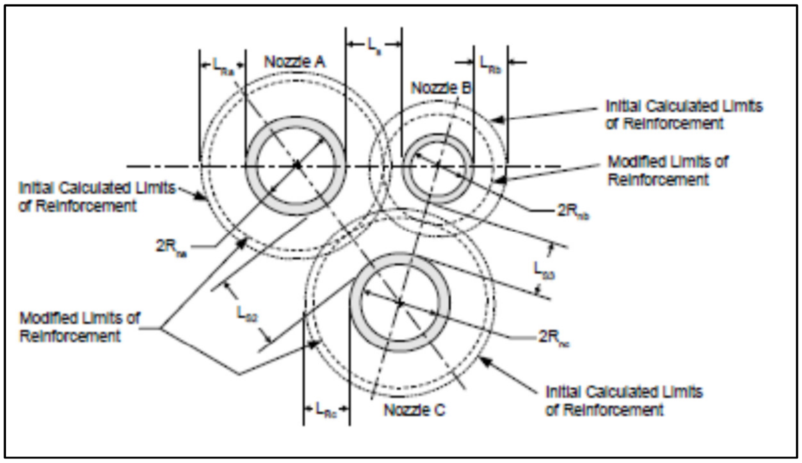


Figure 8:

Example of Three Adjacent Nozzle Openings

$$L_R = min\left[L_{S1}\left(\frac{R_{na}}{R_{na}+R_{nb}}\right), L_{S2}\left(\frac{R_{na}}{R_{na}+R_{nc}}\right)\right]$$

for nozzle A

$$L_R = min\left[L_{S1}\left(\frac{R_{nb}}{R_{na}+R_{nb}}\right), L_{S3}\left(\frac{R_{nb}}{R_{nb}+R_{nc}}\right)\right]$$

for nozzle B

$$L_R = min\left[L_{S2}\left(\frac{R_{nc}}{R_{na}+R_{nc}}\right), L_{S3}\left(\frac{R_{nc}}{R_{nb}+R_{nc}}\right)\right]$$

for nozzle C

Inspection Openings

All pressure vessels for use with compressed air and those subject to internal corrosion or having parts subject to erosion or mechanical abrasion must be provided with suitable inspection opening for examination and cleaning. Vessels that require access or inspection openings shall be equipped as follows:

a) All vessels less than 450 mm (18 in) and over 300 mm (12 in) inside diameter shall have at least two handholes or two plugged, threaded inspection openings of not less than DN 40 (NPS 1-1/2).

b) All vessels 450 mm (18 in) to 900 mm (36 in), inclusive, inside diameter shall have a manhole or at least two handholes or two plugged, threaded inspection openings of not less than DN 50 (NPS 2).

c) All vessels over 900 mm (36 in) inside diameter shall have a manhole or at least two handholes 100 mm x 150 mm (4 in x 6 in) or two equal openings of equivalent area.

d) Openings with removable heads or cover plates intended for other purposes may be used in place of the required inspection openings provided they are equal at least to the size of the required inspection openings.

e) A single opening with removable head or cover plate may be used in place of all the smaller inspection openings provided it is of such size and location as to afford at least an equal view of the interior.

An elliptical or obround manhole shall not be less than 300 mm x 400 mm (12 in x 16 in). A circular manhole shall not be less than 400 mm (16 in) inside diameter. A handhole opening should not be less than 50 mm x 75 mm (2 in x 3 in), but should be as large as is consistent with the size of the vessel and the location of the opening.

Flat Heads

Paragraph 4.6 gives the requirements for minimum thickness of unstayed flat heads, cover plates and blind flanges. The design methods in this paragraph provide adequate strength for design pressure. These design methods are very similar to those given in the ASME VIII-1.

This paragraph also has rules for flat heads which have a single, circular, centrally located opening that exceeds one half of the head diameter. For such flat heads, the head thickness does not have to meet the thickness requirements in paragraph 4.6.2; however, the thickness and other geometry parameters must satisfy the allowable stress limits in Table 4.6.3 (reproduced here as Table 5). The procedure to design such flat heads is given below:

STEP 1 - Determine the design pressure and temperature of the flat head opening.

STEP 2 - Determine the geometry of flat head opening (see Figure 9).

Head/Shell Junction Stresses	Opening/Head Junction Stresses
$S_{HS} \le 1.5 S_{ho}$	$S_{HO} \le 1.5 S_{ho}$
$S_{RS} \le S_{ho}$	$S_{RO} \le S_{ho}$
$S_{TS} \le S_{ho}$	$S_{TO} \le S_{ho}$
$\frac{(S_{HS}+S_{RS})}{2} \le S_{ho}$	$\frac{(S_{HO}+S_{RO})}{2} \le S_{ho}$
$\frac{(S_{HS}+S_{TS})}{2} \le S_{ho}$	$\frac{(S_{HO}+S_{TO})}{2} \le S_{ho}$

Table 5:

Stress Acceptance Criteria for an Integral Flat Head with Opening

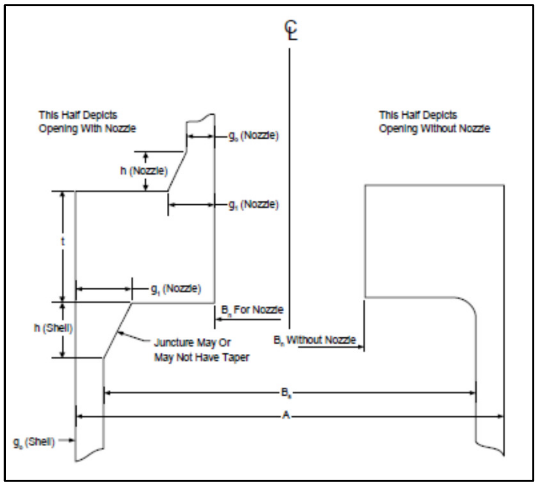


Figure 9: Integral Flat Head with a Large Central Opening

STEP 3 - Calculate the bending moment, M_o, using the following equation:

$$M_o = 0.785 B_n^2 P \left(R + \frac{g_{1n}}{2}\right) + 0.785(B_s^2 - B_n^2) P \left(R + \frac{g_{1n}}{2}\right)$$

where, $R = \frac{B_s - B_n}{2} - g_{1n}$

STEP 4 - Calculate F, V, and f based on B_n, g_{1n}, g_{0n} and h_n using the equations in Tables 4.16.4 and 4.16.5 of the Code, and designate the resulting values as F_n, V_n and f_n.

STEP 5 - Calculate F, V, and f based on B_s, g_{1s}, g_{0s} and h_s using the equations in Tables 4.16.4 and 4.16.5 of the Code, and designate the resulting values as F_s, V_s and f_s.

STEP 6 - Calculate Y, T, U, Z, L, e, and d based on $K = A/B_n$ using the equations in Table 4.16.4 of the Code.

STEP 7 - Calculate the quantity $(E\theta)^*$ for an opening with an integrally attached nozzle using the following equation:

$$(E\theta)^* = \frac{0.91\left(\frac{g_{1n}}{g_{0n}}\right)^2 (B_N + g_{0n}) V_n}{f_n \sqrt{B_n g_{0n}}} S_H$$

where S_H is evaluated using equation in Table4.6.2.

STEP 8 - Calculate the quantity M_H using the following equation:

$$M_H = \frac{(E\theta)^*}{\frac{1.74 V_S \sqrt{B_S g_{0S}}}{g_{0S}^3 (B_S + g_{0S})} + \frac{(E\theta)^*}{M_O}\left(1 + \frac{F_S t}{\sqrt{B_S g_{0S}}}\right)}$$

STEP 9 - Calculate the quantity X_1 using the following equation:

$$X_1 = \frac{M_O - M_H\left(1 + \frac{F_S t}{\sqrt{B_S g_{0S}}}\right)}{M_O}$$

STEP 10 - Calculate the stresses at the shell-to-flat head junction and opening-to-flat-head junction using Table 4.6.2.

STEP 11 - Check the flange stress acceptance criteria in Table 4.6.3. If the stress criteria are satisfied, then the design is complete. If not, then re-proportion the flat head and/or opening dimensions and go to STEP 3.

Sources:

1. ASME Boiler and Pressure Vessel Code, Section VIII, Division 2 - Alternate Rules, Edition 2010

***** E N D O F T H E A R T I C L E *****

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CoDesign Engineering is involved in providing training and consultancy services as described below:

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