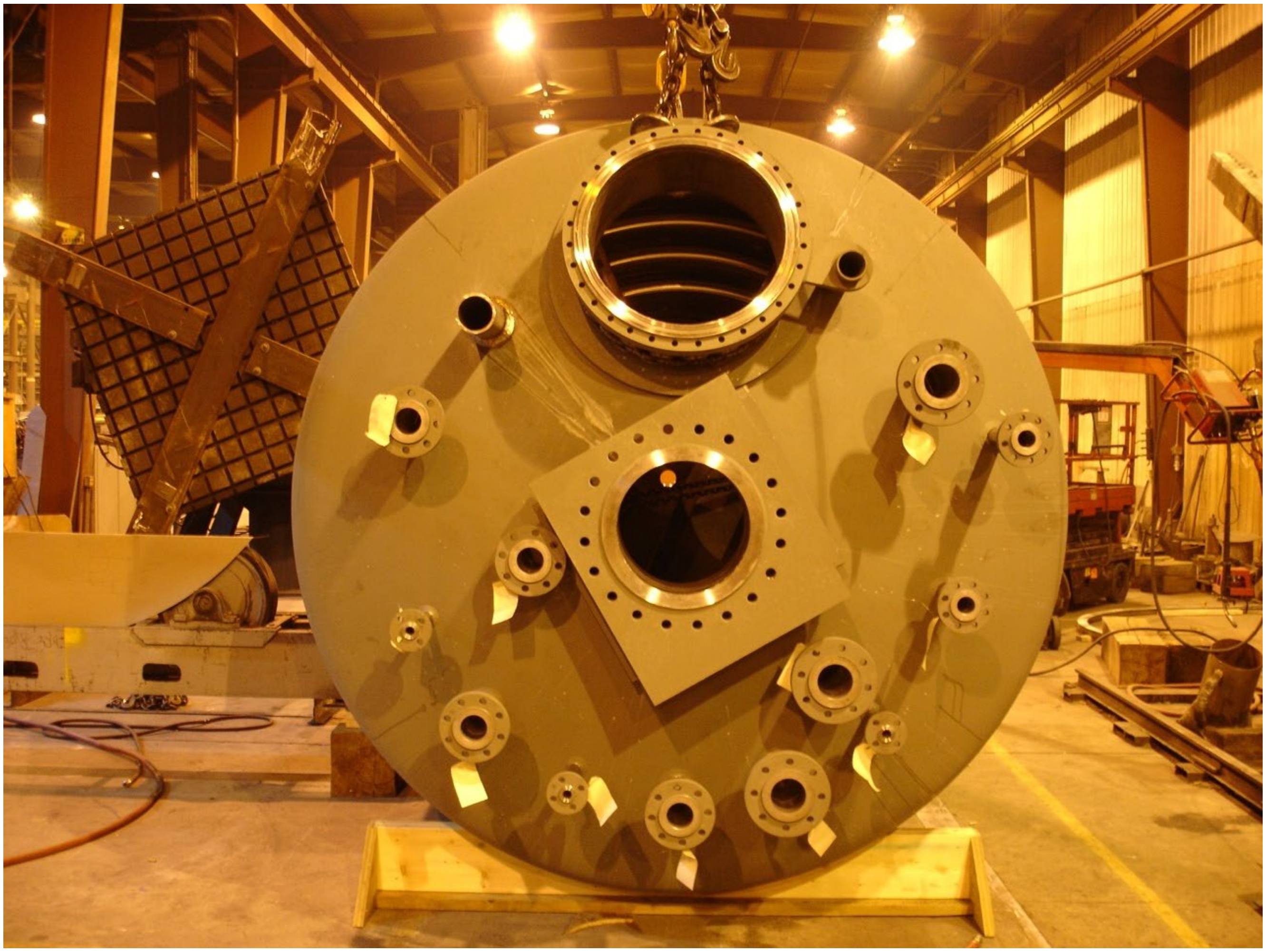
Fixed Equipment Newsletter

Volume 2025, January Issue



NOZZLE ATTACHMENTS

From the Editor's Desk:

We have a new year, and a new administration in the US. The US economy ended 2024 on a low note, and the global situation at the end of 2024 and into the first month of 2025 remains a little uncertain. All in all, a mix of optimism and some concern. 2025 also happens to be the year when a new edition of the ASME Boiler and Pressure Vessel Code will be issued. By all accounts, the upcoming changes in the ASME Section VIII, Division 1 are massive – rivalling the changes in the 2007 edition of ASME Section VIII, Division 2 code. 2025 edition of the code will be issued on July 1, 2025, and compliance with the code requirements will become mandatory for all pressure vessels contracted after January 1, 2026.

So, what are the upcoming changes in the ASME Section VIII, Division 1 code? Well… there are several. But two changes stand out. First – the long running push to harmonize ASME Section VIII, Division 1 design rules with those of ASME Section VIII, Division 2, Part 4 continues. It may even have accelerated. Will ASME Section VIII, Division 1 become obsolete in the future? Only time will tell. But it sure looks that way.

The second change is much more interesting. The ASME Section VIII, Division1 code currently has 42 mandatory appendices and 26 nonmandatory appendices. Some people have jokingly called it the Code of Appendices. There has been a long-standing demand from the pressure vessel community to rationalize the Appendix section and bring the number of mandatory and nonmandatory appendices to a manageable level. It appears that the voice of the pressure vessel community has been heard. This, however, has resulted in the introduction of a new Subsection and transfer of some of the Appendices to this new Subsection.

So, the changes in the 2025 edition of the ASME Section VIII, Division 1 code can be summarized as follows: a) continued harmonization of the ASME Section VIII, Division 1 design rules with ASME Section VIII, Division 2, Part 4 design rules, and b) Introduction of a new Subsection that contains many of the rules in the old Appendices. There are other changes as well of course… however, these two are the major ones. Stay tuned to a newsletter article in the near future with a detailed description of all the changes in the 2025 edition of ASME Section VIII, Division 1.

Best wishes to everyone for a very happy new year.

Ramesh K Tiwari

The graphic on the cover page is courtesy of Fourinox - located in Green Bay, Wisconsin.

Featured in this issue…

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NOZZLE ATTACHMENT WELDS

This article is a summary of paragraph UW-16 of ASME Section VIII, Division 1 (ASME VIII-1). This paragraph classifies the nozzle attachments into various types, namely:

1) *Necks attached by full penetration weld*
2) *Necks attached by fillet or partial penetration weld*
3) *Necks and tubes up to and including NPS6 attached from one side only*
4) *Standard fittings: ASME/ ANSI or Manufacturer's Standard*
5) *Bolting pads: Manufacturer's Standard*

ASME VIII-1 requirements as they apply to various types of nozzle attachments are described below.

NECKS ATTACHED BY A FULL PENETRATION WELD

1) Necks abutting a vessel wall (set-on nozzles) shall be attached by full penetration groove welds.

   See Figure 1 for examples.

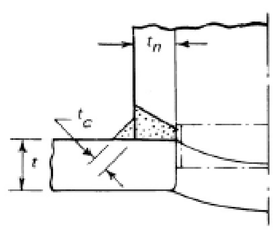

Figure: UW-16.1, Sketch (a)

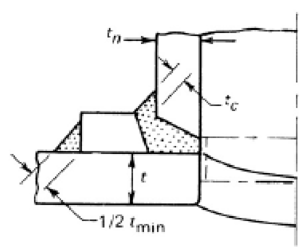

Figure: UW-16.1, Sketch (a-1)

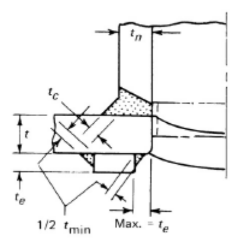

Figure: UW-16.1, Sketch (a-2)

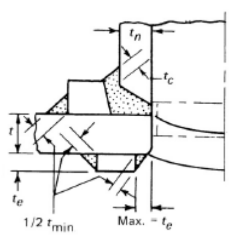

Figure: UW-16.1, Sketch (a-3)

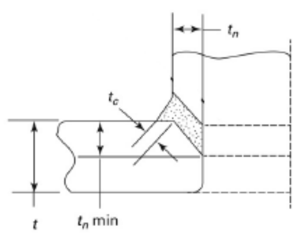

Figure: UW-16.1, Sketch (a-4)

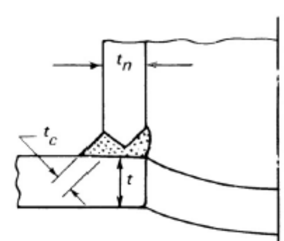

Figure: UW-16.1, Sketch (b)

Figure 1: Examples for Set-on Nozzles Attached by Full Penetration Weld

2) Necks inserted through the vessel wall (set-in nozzles) may be attached by full penetration groove welds.

   See Figure 2 for examples. *Note: The configurations shown in Figure 2 are all full penetration groove welds; however, it is not a requirement for set-in nozzles.*

3) If additional reinforcement is required, it shall be provided as an integral reinforcement, or by the addition of separate reinforcement plates.

a) Integral reinforcement is that reinforcement provided in the form of extra thickness in nozzle necks and/or shells, forging type inserts, or weld build-up which is an integral part of shell or nozzle wall, and where required, is attached by full penetration welds. Correction factor, F factor from Figure UG-37 may be used for reinforcement calculations.

See Figure 3 for examples.

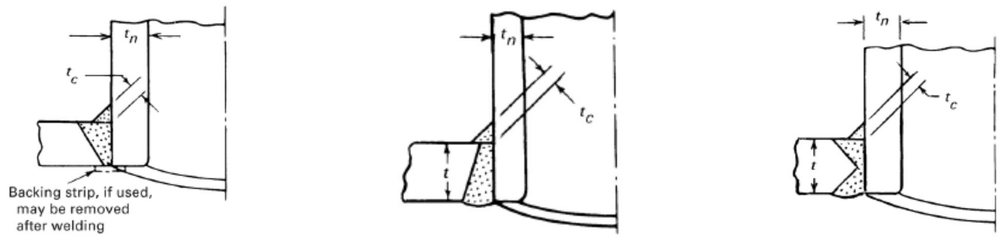

Figure: UW-16.1, Sketch (c) | Figure: UW-16.1, Sketch (d) | Figure: UW-16.1, Sketch (e)

Figure 2: Examples for Set-in Nozzles Attached by Full Penetration Welds

b) Separate reinforcement plates may be added to the outside surface of shell wall, the inside surface of shell wall, or to both surfaces of shell wall. The nozzle is no longer considered a nozzle with integral reinforcement, and the F factor shall be taken as 1.0.

See Figure 4 for examples. These are various applications of reinforcing plates added to Figure UW-16.1, Sketch (a) in ASME VIII-1.

Any of these applications of reinforcement plates may be used with necks of types shown in Figure UW-16.1, Sketches (b), (c), (d), and (e) in ASME VIII-1. The reinforcing plates shall be attached by welds at the outer edge of the plate and at the nozzle neck periphery or inner edge of the plate if there is no neck adjacent to the plate.

i) The weld at the outer edge of the reinforcement plate shall be a continuous fillet weld with a minimum throat dimension of ½ t_{min}.

$$t_{min} = \text{smaller of } (3/4", \text{thickness of the thinner of the parts joined})$$

ii) The weld at the inner edge of the reinforcement plate which does not abut a nozzle neck shall be continuous fillet weld with a minimum throat dimension of ½ t_{min}.

See Figure 5 for examples.

iii) The weld at the inner edge of the reinforcement plate when the reinforcement plate is full penetration welded to the nozzle neck shall be continuous fillet weld with a minimum throat dimension of t_c.

$$t_c \geq \text{smaller of } (1/4", 0.7t_{min})$$

See Figure 6 for examples.

iv) The weld at the inner edge of the reinforcement plate when the reinforcement plate is not full penetration welded to the nozzle neck shall be continuous fillet weld with a minimum throat dimension of $t_w = 0.7t_{min}$.

See Figure 7 for example.

NECKS ATTACHED BY FILLET OR PARTIAL PENETRATION WELDS

1) Set-in nozzles may be attached by fillet or partial penetration welds, one on each face of the vessel wall. The welds may be any desired combination of fillet, single-bevel, and single-J welds. The dimensions t_1 and t_2 in each weld shall be as follows:

$$t_1, t_2 \quad \geq \quad \text{smaller of } (1/4", 0.7t_{min})$$

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

See Figure 8 for examples.

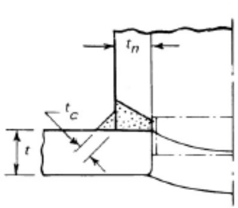

Figure: UW-16.1, Sketch (a)

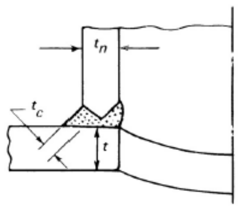

Figure: UW-16.1, Sketch (b)

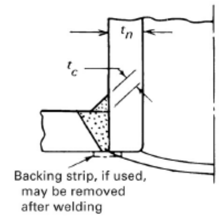

Figure: UW-16.1, Sketch (c)

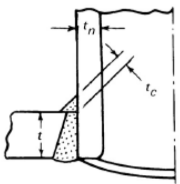

Figure: UW-16.1, Sketch (d)

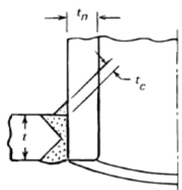

Figure: UW-16.1, Sketch (e)

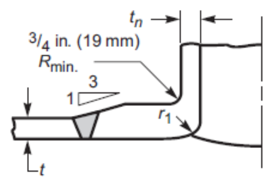

Figure: UW-16.1, Sketch (f-1)

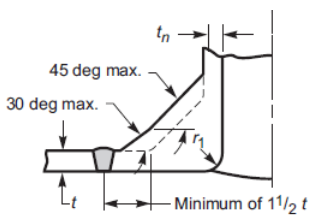

Figure: UW-16.1, Sketch (f-2)

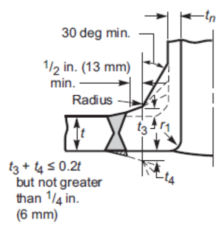

Figure: UW-16.1, Sketch (f-3)

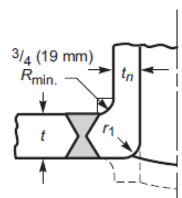

Figure: UW-16.1, Sketch (f-4)

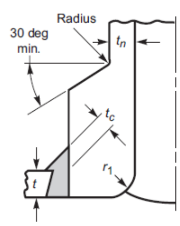

Figure: UW-16.1, Sketch (g)

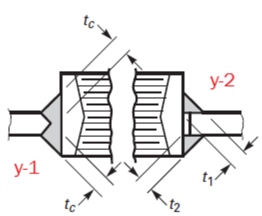

Figure: UW-16.1, Sketch (y-1)

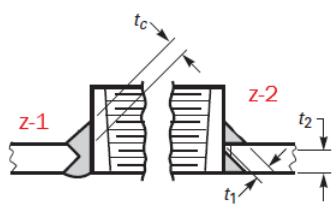

Figure: UW-16.1, Sketch (z-1)

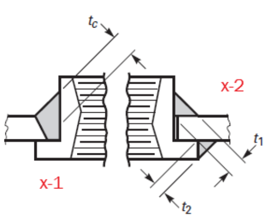

Figure: UW-16.1, Sketch (x-1)

Figure 3: Examples for Nozzles with Integral Reinforcement

2) If additional reinforcement is required, it may be provided in the form of extra thickness in nozzle necks and/or shells, forgings, and/or separate reinforcing plates attached by welding. Weld requirements shall be same as that for

separate reinforcing plate for nozzles attached by full penetration weld except that the welds attaching the nozzle to the vessel wall or to the reinforcing plate shall consist of either (-a) or (-b) below:

a) Single-bevel or single-J weld in the shell plate, and a single-bevel or a single-J weld in each reinforcing plate.

$$t_w \geq 0.7t_{min}$$

See Figure 9 for examples.

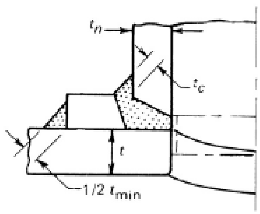

Figure: UW-16.1, Sketch (a-1)

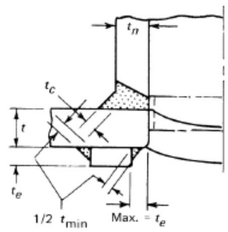

Figure: UW-16.1, Sketch (a-2)

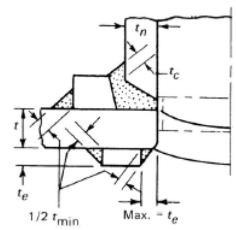

Figure: UW-16.1, Sketch (a-3)

Figure 4: Examples for Nozzles with Separate Reinforcing Plates

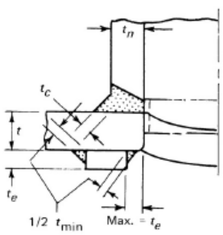

Figure: UW-16.1, Sketch (a-2)

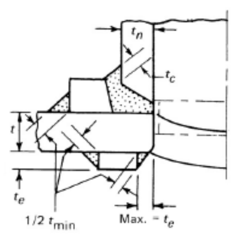

Figure: UW-16.1, Sketch (a-3)

Figure 5: Weld at Inner Edge of Reinforcing Plate that Does Not Abut Nozzle

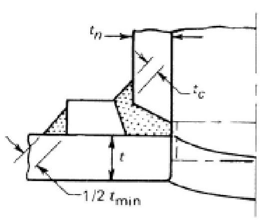

Figure: UW-16.1, Sketch (a-1)

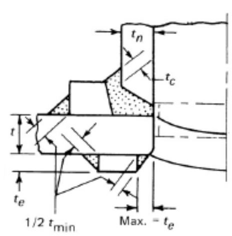

Figure: UW-16.1, Sketch (a-3)

Figure 6: Weld at Inner Edge of Reinforcing Plate When Reinforcing Plate is Full Penetration Welded to Nozzle

b) Full penetration groove weld in each reinforcing plate; and a fillet, single-bevel, or single-J weld with the following dimensions:

$$t_w \geq 0.7t_{min}$$

See Figure 10 for examples.

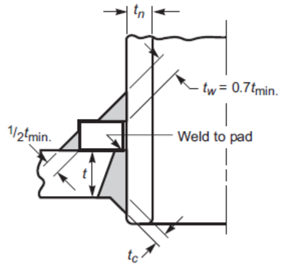

Figure: UW-16.1, Sketch (h)

Figure 7: Weld at Inner Edge of Reinforcing Plate When Reinforcing Plate is Not Full Penetration Welded to Nozzle

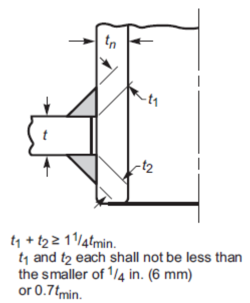

Figure: UW-16.1, Sketch (i)

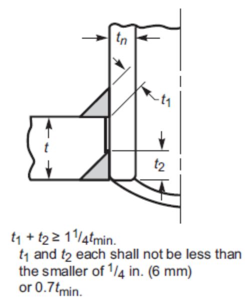

Figure: UW-16.1, Sketch (j)

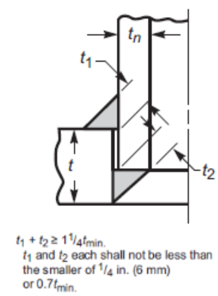

Figure: UW-16.1, Sketch (k)

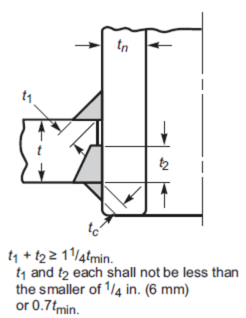

Figure: UW-16.1, Sketch (l)

Figure 8: Set-in Nozzles Attached by Fillet or Partial Penetration Welds, One on Each Face of Vessel Wall

3) Nozzles, flared nozzles, and studding outlet type flanges may be attached by fillet welds or partial penetration welds between the outside diameter or the attachment and the outside surface of the shell and at the inside of the opening in the shell. The throat dimension of the outer attachment weld shall not be less than $1/2\mathrm{t}_{\mathrm{min}}$. The dimension t_{w} of the weld at the inside of the shell cutout shall not be less than $0.7\mathrm{t}_{\mathrm{min}}$.

See Figure 11 for examples.

Studding outlet type flanges may also be attached by full penetration welds as shown in Figure UW-16.1, Sketch (p-2) in ASME VIII-1.

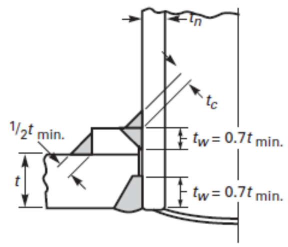

Figure: UW-16.1, Sketch (q)

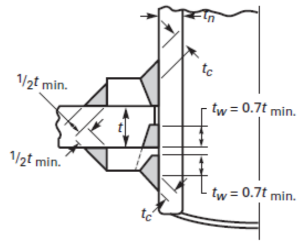

Figure: UW-16.1, Sketch (r)

Figure 9: Single-bevel or Single-J Weld in Shell Plate, and Single-bevel or Single-J Weld in Each Reinforcing Plate

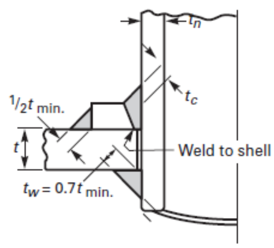

Figure: UW-16.1, Sketch (s)

Figure 10: Full Penetration Groove Weld in Each Reinforcing Plate, and Fillet, Single-bevel, or Single-J Weld in Each Reinforcing Plate

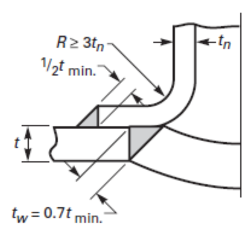

Figure: UW-16.1, Sketch (m)

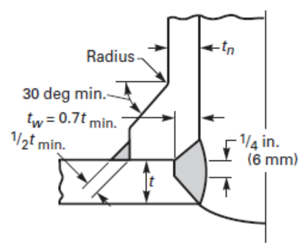

Figure: UW-16.1, Sketch (n)

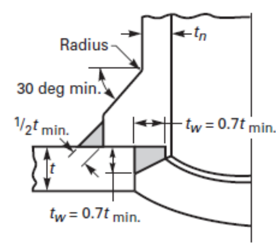

Figure: UW-16.1, Sketch (0)

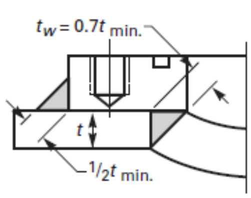

Figure: UW-16.1, Sketch (p-1)

Figure 11: Nozzle Necks, Flared Necks, and Studding Outlet Type Flanges Attached by Fillet Welds or Partial Penetration Welds

NECKS AND TUBES UP TO AND INCLUDING NPS6 ATTACHED FROM ONE SIDE ONLY

1) Necks and tubes not exceeding NPS6 may be attached from one side only on either the outside or inside surface of the vessel.

   a) Depth of the welding groove or the throat of the fillet weld shall be at least equal to $1\frac{1}{4}t_{min}$.

   See Figure 12 for examples.

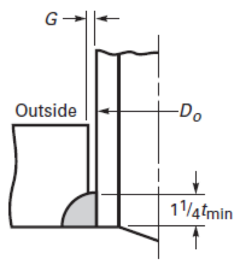

Figure: UW-16.1, Sketch (v-1)

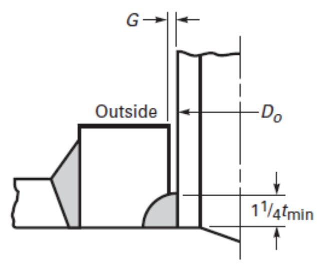

Figure: UW-16.1, Sketch (v-2)

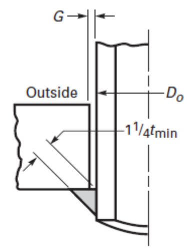

Figure: UW-16.1, Sketch (w-1)

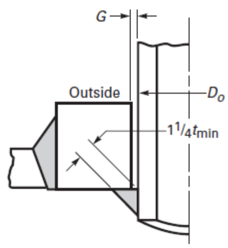

Figure: UW-16.1, Sketch (w-2)

Figure 12: Nozzles and Tubes Not Exceeding NPS6 Attached From One Side Only

The radial clearance between the vessel opening and the nozzle outside diameter at the unwelded side shall not exceed the tolerances given in the sketches in Figure 12.

For applications where there are no external loads, G = 1/8” max.

For applications where there are external loads, $G = 0.005$ in for $D_o \leq 1$ in

$G = 0.010$ in for $1 \text{ in} < D_o \leq 4$ in

$G = 0.015$ in for $4 \text{ in} < D_o \leq 6\frac{5}{8}$ in

When welded from outside, the nozzle or tube shall extend to be at least flush to the inside surface of the vessel wall.

These attachments shall satisfy rules for reinforced openings, except that no material in the nozzle shall be counted as reinforcement.

   b) As an alternative to a) above, when the nozzle or tube is attached from outside, a welding groove shall be cut into the surface to a depth of not less than t_n on the longitudinal axis of the opening. It is recommended that a recess of 1/16” deep be provided at the bottom of the groove, in which the nozzle can be centered. The dimension t_w of the attachment weld shall not be less than t_n nor less than 1/4".

   See Figure 13 for examples.

STANDARD FITTINGS: ASME/ ANSI OR MANUFACTURER'S STANDARD

Attachment of standard fittings shall meet the following requirements:

1) Except as provided in 2) through 6) below, fittings shall be attached by

   a) Two fillet or partial penetration welds, one on each face of the vessel wall; and the minimum weld dimensions shall be as shown in Figure 14; or

   b) Full penetration groove weld [See "Necks Attached by a Full Penetration Weld"].

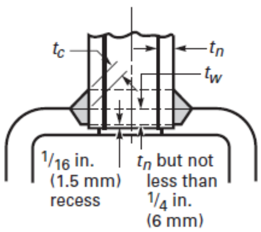

Figure: UW-16.1, Sketch (t)

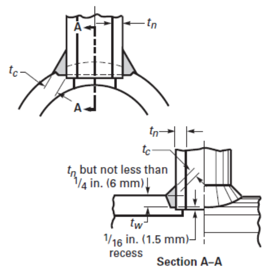

Figure: UW-16.1, Sketch (u)

Figure 13: Nozzles or Tubes Not Exceeding NPS6 Attached From Outside Only

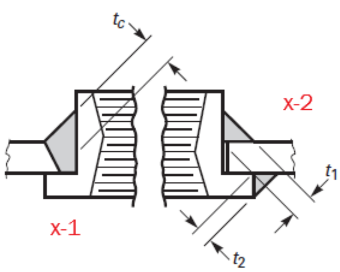

Figure: UW-16.1, Sketch (x)

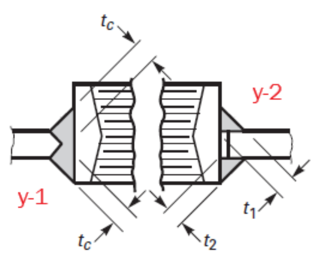

Figure: UW-16.1, Sketch (y)

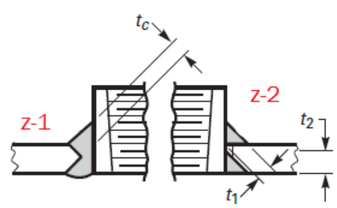

Figure: UW-16.1, Sketch (z)

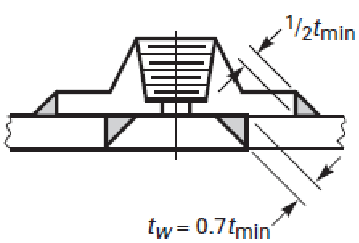

Figure: UW-16.1, Sketch (aa)

Figure 14: Two Fillet or Partial Penetration Welds, One on Each Face of Vessel Wall

2) Fittings not exceeding NPS3 shown in Figure 15 may be attached by welds that are exempt from size requirements with the following limitations:

   a) The strength of groove welds and the fillet welds shall be calculated for all applied loadings.

   b) For partial penetration welds or fillet welds,

$$t_1 \text{ or } t_2 \geq \text{smaller of } (3/32", 0.7t_{min})$$

3) See below.

a) If a pressure vessel is not subject to rapid fluctuations in pressure, fittings not exceeding NPS3 may be attached by a fillet weld deposited from outside only without any additional reinforcement provided all of following conditions are satisfied:

i) The maximum vessel thickness is 3/8”.

ii) The maximum size of opening is limited to the outside diameter of the attached pipe plus 3/4", but not greater than one-half the vessel inside diameter.

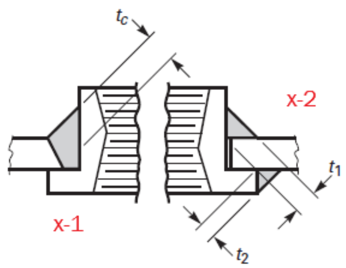

Figure: UW-16.1, Sketch (x)

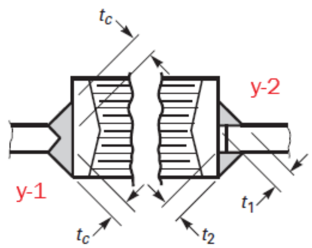

Figure: UW-16.1, Sketch (y)

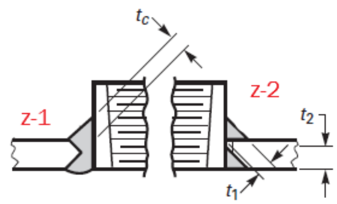

Figure: UW-16.1, Sketch (z)

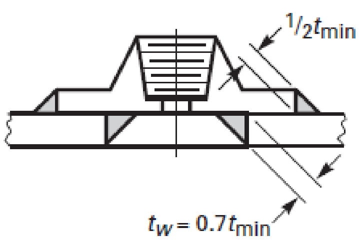

Figure: UW-16.1, Sketch (aa)

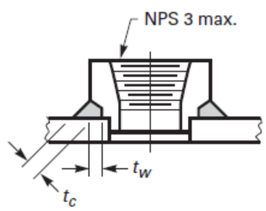

Figure: UW-16.1, Sketch (bb)

Figure 15: Fittings Not Exceeding NPS3 Attached by Welds That Are Exempt From Size Requirements

iii) The attachment weld throat shall be greater of the minimum nozzle neck thickness required by UG-45 for the same nominal size connection AND that necessary to limit the maximum allowable load to the product of weld area (based on minimum leg dimension), the maximum allowable stress value in tension of the material being used, and a joint efficiency of 0.55.

iv) The typical fitting dimension t_f as shown in Figure 16 shall be sufficient to accommodate a weld leg which will provide a weld throat dimension detailed in iii) above.

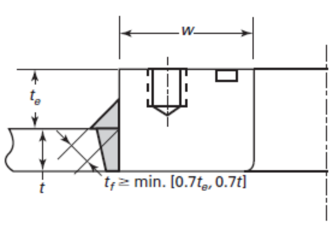

Figure: UW-16.1, Sketch (p-2)

Figure 16: Fitting Dimension t_f

v) The openings shall meet the requirements of small openings defined in ASME VIII-1.

vi) In lieu of thickness requirements in UG-45, the minimum wall thickness for fittings shall not be less than that shown in Table 1 plus any thickness needed for corrosion allowance.

Table 1: Minimum Thickness Requirements for Fittings

NPS	in	NPS	in
1/8	0.11	1 ¼	0.30
1/4	0.11	1 ½	0.30
3/8	0.11	2	0.31
1/2	0.14	2 ½	0.37
3/4	0.16	3	0.38
1	0.22		

In the table above, for fittings having a specified outside diameter not equal to the outside diameter of an equivalent standard NPS size, the NPS size chosen from the table shall be one having an equivalent outside diameter larger than the fittings outside diameter.

b) If the size of the opening does not meet the requirements for a small opening; OR exceeds the outside diameter of attached pipe plus 3/4"; OR is greater than one-half of the vessel inside diameter, the affected part shall be subject to a proof test, or the opening shall be reinforced in accordance with the code requirements given in paragraph UG-37. In satisfying the rules for reinforcement, no material in the nozzle neck shall be counted as reinforcement.

4) Fittings not exceeding NPS3 may be attached by a fillet groove weld from the outside only as shown in Figure 17. The groove weld, t_w, shall not be less than the thickness of Schedule 160 pipe for the nearest equivalent pipe size.

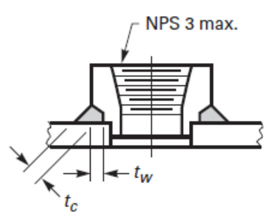

Figure: UW-16.1, Sketch (bb)

Figure 17: Fittings Not Exceeding NPS3 Attached by Fillet Groove Weld From Outside Only

5) Flange type fittings not exceeding NPS2, with some acceptable type shown in Figure UW-16.2, may be attached without any additional reinforcement provided all of the following conditions are met,

a) Maximum vessel wall thickness is 3/8”.

b) Maximum design pressure is 350 psi.

c) Minimum fillet leg, t_f, is 3/32”.

d) Finished opening shall not exceed the outside diameter of the nominal pipe size plus 3/4".

6) Fittings conforming to Figure 18 not exceeding NPS3 may be attached by a single fillet weld on the inside of the vessel only provided criteria of Figure 19 are satisfied.

BOLTING PADS: MANUFACTURER'S STANDARD

The attachment of standard bolting pads shall meet the following requirements:

1) Bolting pads shall be attached by full penetration groove weld OR by two fillet OR partial penetration welds, one on each face of the vessel wall.

   See Figure 20 for examples.

   Exceptions are provided in 2) and 3) below.

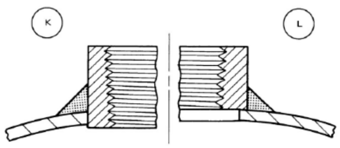

Figure: UW-16.2, Sketch (k)

Figure 18: Fittings Not Exceeding NPS3 Attached by Fillet Groove Weld

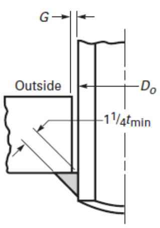

Figure: UW-16.1, Sketch (w-1)

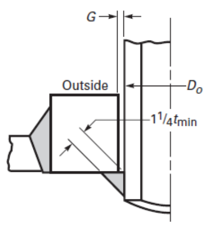

Figure: UW-16.1, Sketch (w-2)

Figure 19: Fittings Not Exceeding NPS3 Attached by Fillet Groove Weld

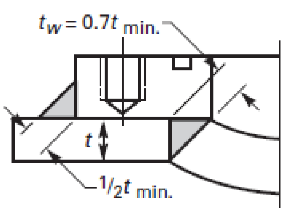

Figure: UW-16.1, Sketch (p-1)

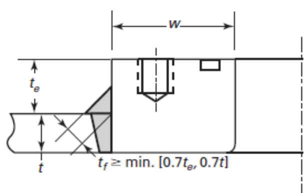

Figure: UW-16.1, Sketch (p-2)

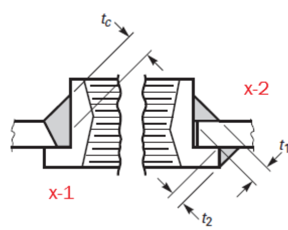

Figure: UW-16.1, Sketch (x)

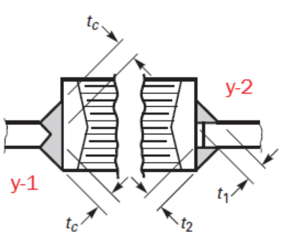

Figure: UW-16.1, Sketch (y)

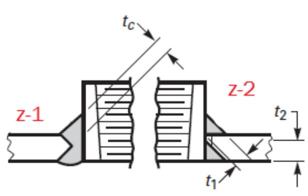

Figure: UW-16.1, Sketch (z)

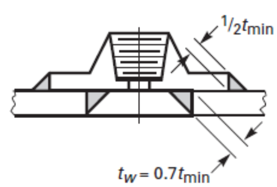

Figure: UW-16.1, Sketch (aa)

Figure 20: Bolting Pads Attached by Full Penetration Groove Weld OR by Two Fillet OR Partial Penetration Welds, One on Each Face of Vessel Wall

2) Bolting pads as shown in Figure 21 may be attached to vessels by a fillet weld deposited from outside only with the following limitations:

a) The maximum vessel wall thickness is 3/8” and the bolting pad outside diameter is not greater than 4-¾”.

b) The maximum size of the opening in the vessel is limited to the following:

   i) 4-¾” for bolting pads that are inserted through the vessel wall. See UW-16.3, Sketch (a) in Figure 21 below.

   ii) 1/4" less than the bolting pad diameter for those that are set-on the vessel. See UW-16.3, Sketch (b) in Figure 21 below.

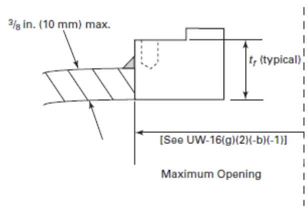

Figure: UW-16.3, Sketch (a)

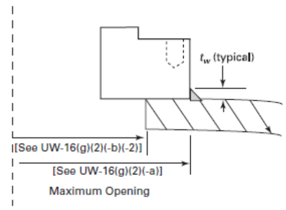

Figure: UW-16.3, Sketch (b)

Figure 21: Bolting Pads Attached by Fillet Weld Deposited From Outside Only

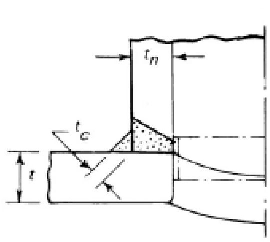

Figure: UW-16.1, Sketch (a)

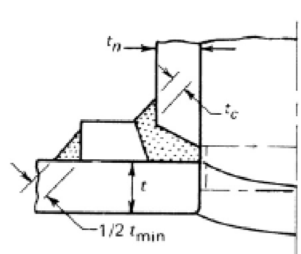

Figure: UW-16.1, Sketch (a-1)

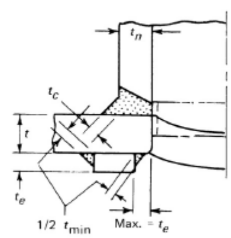

Figure: UW-16.1, Sketch (a-2)

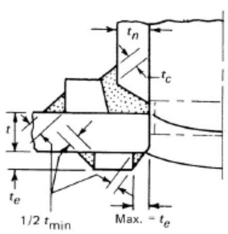

Figure: UW-16.1, Sketch (a-3)

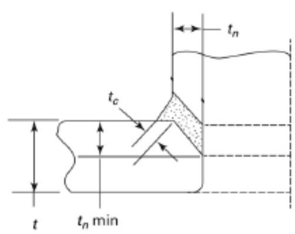

Figure: UW-16.1, Sketch (a-4)

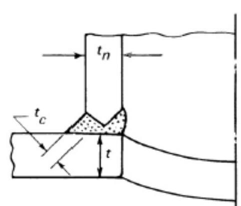

Figure: UW-16.1, Sketch (b)

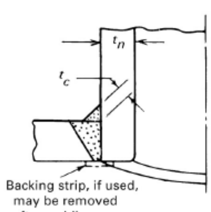

Figure: UW-16.1, Sketch (c)

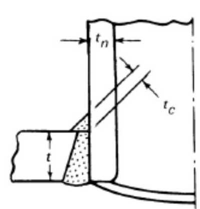

Figure: UW-16.1, Sketch (d)

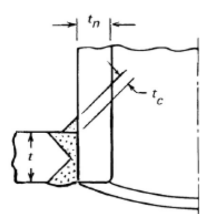

Figure: UW-16.1, Sketch (e)

Figure 22: Fillet Weld Leg Dimensions for Sketches Above

c) The attachment weld throat shall be the greatest of the following:

   i) Minimum nozzle neck thickness required by UG-45 for the same nominal size connection.

ii) $1.0t_{min}$

iii) That necessary to limit the maximum allowable load to the product of weld area (based on minimum leg dimension), the maximum allowable stress value in tension of the material being used, and a joint efficiency of 0.55.

d) The typical bolting pad dimension, t_f, as shown in UW-16.3, Sketch (a) in Figure 21 above shall be sufficient to accommodate a weld leg that will provide a weld throat dimension.

e) In satisfying the rules for reinforcement of openings, no material in the bolting pad shall be counted as reinforcement.

3) If the opening exceeds either 4-¾" diameter or is greater than one-half the vessel inside diameter, the affected part shall be subject to proof test, or the opening shall be reinforced in accordance with UG-37 and the nozzle shall be attached using a suitable detail in Figure UW-16.1, if welded.

MINIMUM THROAT DIMENSIONS OF FILLET WELDS

The minimum throat dimensions of fillet welds defined in Figure UW-16.1 shall be maintained around the circumference of the attachment, except as provided below.

1) For nozzles attached to shells as shown in Figure 22, the fillet weld leg dimensions that meets the minimum throat dimensions shall be calculated for the equivalent thickness nozzle attached perpendicular to the surface of a flat head with thickness equal to the shell thickness.
2) The fillet weld leg dimensions along the shell and nozzle need not exceed the calculated fillet weld leg dimension.
3) For radial nozzles when the outside diameter of the nozzle approaches tangency with the shell, the fillet weld shall be smoothly transitioned into the full-penetration weld.
4) For non-radial and tangential nozzles where the angle between the nozzle and shell is too large or too small to make a fillet weld or when the nozzle approaches tangency with the shell, the fillet weld shall be smoothly transitioned into the full-penetration weld.

References:

ASME Boiler and Pressure Vessel Code, Section VIII, Division 1 (2023 Edition)

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DESIGN RULES FOR OPENINGS IN SHELLS AND HEADS (DIV. 2)

The rules for the design of nozzles in shells and heads subjected to internal pressure, external pressure, and external forces and moments from supplemental loads are provided in paragraph 4.5 of ASME Section VIII, Division 2. The rules described here are based on the 2023 edition of the code.

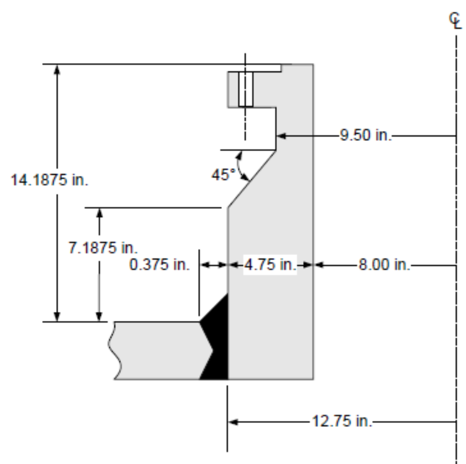
Dimensions and Shape of Nozzles

Nozzles shall be circular, elliptical, or any other shape that results from intersection of circular of circular or elliptical cylinder with the vessel. These rules can only be used if the ratio of inside diameter of the shell and the shell thickness is less than or equal to 400. The exception is that when nozzles are located on a spherical shell or a formed head there is no restriction on this ratio.

D_i	=	Inside diameter of shell
t	=	Nominal thickness of vessel wall

For nozzles located on vessel components other than spherical shell and formed head, D_i/t shall be less than or equal to 400. When this condition is not satisfied, the nozzles shall be designed in accordance with the requirements of Part 5.

Throughout this article, we will perform calculations for the following nozzle for a Class 2 vessel:
(*Taken from ASME PTB-3*)



Example Problem

The details for this nozzle are as follows:

<u>Design Conditions</u>

Design pressure, P	=	356 psig
Design temperature, T	=	300ºF
Corrosion allowance	=	0.125 in
Weld joint efficiency	=	1.0

<u>Shell Data</u>

Shell material	=	SA-516 70N	
Shell inside diameter	=	150 in	
Shell thickness	=	1.8125 in	
Shell allowable stress, S	=	22,400 psi	[*from Table 5A of ASME II-D*]

Nozzle Data

Nozzle material	=	SA-105	
Nozzle outside diameter, d_o	=	19.0 in	
Nozzle hub outside diameter, d_{ho}	=	25.5 in	
Nozzle hub height	=	7.1875 in	
Nozzle thickness	=	4.75 in	
External nozzle projection	=	14.1875 in	
Internal nozzle projection	=	0 in	
Nozzle allowable stress, S_n	=	21,200 psi	[*from Table 5A of ASME II-D*]

The nozzle is inserted through the shell, i.e. set-in type nozzle (see the Figure on previous page).

Method of Attachment

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

Welded Connections

Nozzles are attached by welding.

Studded Connections

Nozzles may be made by means of studded pad type connections. The vessel shall have a flat surface machined on the shell, or on a built-up pad. Drilled holes to be tapped shall not penetrate within one-fourth of the wall thickness from the inside surface of the vessel after deducting corrosion allowance. Where tapped holes are provided for studs, the threads shall be full and clean and shall engage the stud for a length, L_{st} defined by the following equations:

$L_{st} = \min(L_{st1}.\ 1.5d_{st})$ *Thread engagement length*

where,

$L_{st1} = \max\left[d_{st}, 0.75d_{st}\left(\frac{S_{st}}{S_{tp}}\right)\right]$

d_{st} = Nominal diameter of stud

S_{st} = Allowable stress of stud material at design temperature

S_{tp} = Allowable stress of the tapped material at design temperature

Threaded Connections

Pipes, tubes and other threaded connections that conform to ASME B1.20.1 may be screwed into a threaded hole in a vessel wall, provided the connection size is less than NPS2 and the pipe engages a minimum number of threads specified in table 1 below:

Table 1: Minimum Number of Pipe Threads for Connections

Size of Pipe	Threads engaged	Minimum Plate Thickness Required
NPS 0.5, 0.75	6	0.43 in
NPS 1, 1.25, 1.5	7	0.61 in
NPS 2	8	0.70 in

Expanded Connections

A pipe, tube or forging may be attached to the wall of a vessel by inserting through an unreinforced opening and expanding into the shell, provide the diameter is not greater than NPS2. A pipe, tube or forging not exceeding 6 in. in outside diameter may be attached to the wall of a vessel by inserting through a reinforced opening and expanding into the shell.

Nozzle Neck Minimum Thickness Requirements

The minimum nozzle neck thickness for nozzles excluding access openings and openings for inspection shall be determined for internal and external pressure using the applicable design rules given in the code. The resulting nozzle neck thickness shall not be less than the smaller of the shell thickness or the thickness given in Table 2. Corrosion allowance shall be added to the minimum nozzle neck thickness.

Table 2: Nozzle Minimum Thickness Requirements

Nominal Size	Minimum Thickness (in)
NPS 1/8	0.060
NPS 1/4	0.077
NPS 3/8	0.080
NPS 1/2	0.095
NPS 3/4	0.099
NPS 1	0.116
NPS 1-1/4	0.123
NPS 1-1/2	0.127
NPS 2	0.135
NPS 2-1/2	0.178
NPS 3	0.189
NPS 3-1/2	0.198
NPS 4	0.207
NPS 5	0.226
NPS 6	0.245
NPS 8	0.282
NPS 10	0.319
≥ NPS 12	0.328

Design rules are provided in the paragraph 4.5 for:

- Radial nozzle in a cylindrical shell
- Hillside nozzle in a cylindrical shell
- Nozzle in a cylindrical shell oriented at an angle from the longitudinal axis
- Radial nozzle in a conical shell
- Radial nozzle in a spherical shell or formed head
- Hillside or perpendicular nozzle in a spherical shell or formed head, and
- Circular nozzles in a flat head.

This article will address only the radial nozzles in a cylindrical shell. The parameters used in the design procedure are shown in Figures 1, 2 and 3.

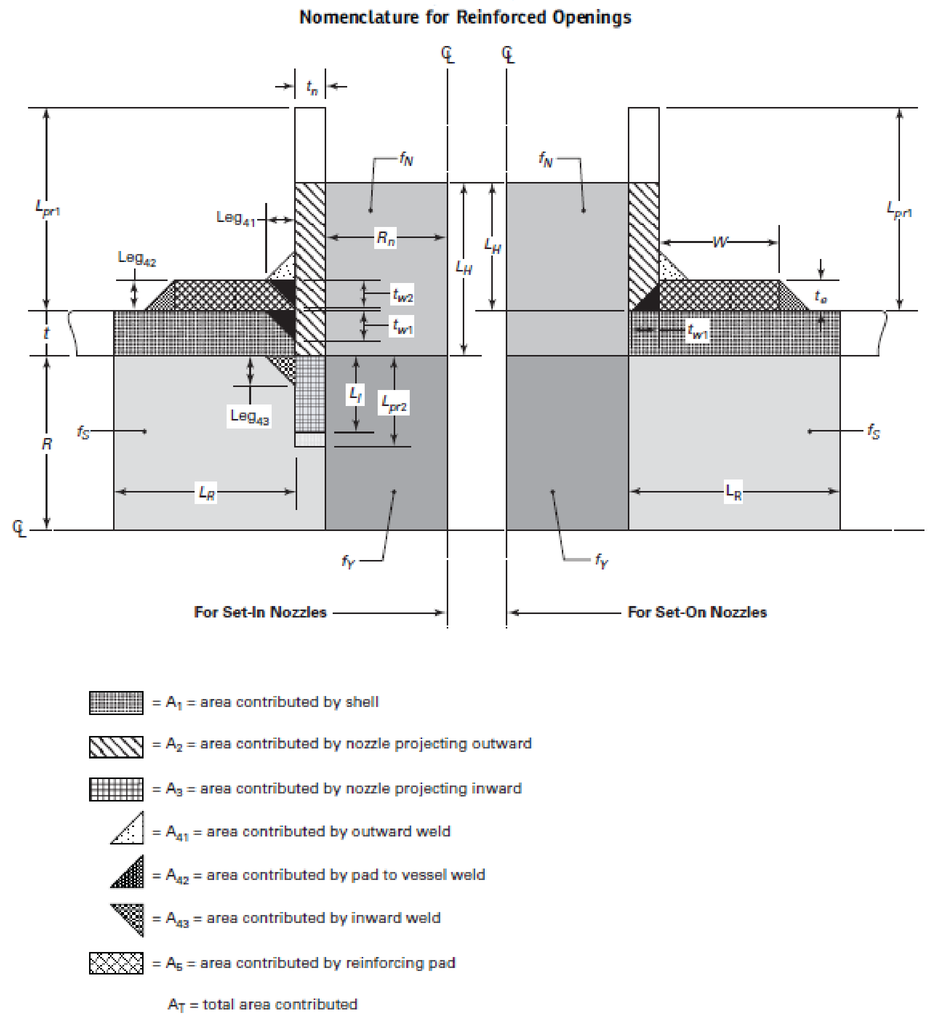


Figure 1: Nomenclature for Reinforced Openings

We will first establish the corroded dimensions for the example problem.

For Shell:

$$D_i = 150.0 + 2(\text{Corrosion Allowance}) = 150.0 + 2(0.125) = 150.25 \text{ in}$$

$$t = 1.8125 - \text{Corrosion Allowance} = 1.8125 - 0.125 = 1.6875 \text{ in}$$

For Nozzle:

$$t_n = 4.75 - \text{Corrosion Allowance} = 4.75 - 0.125 = 4.625 \text{ in}$$

$$R_n = \frac{D-2(t_n)}{2} = \frac{25.5-2(4.625)}{2} = 8.125 \text{ in}$$

Step 1: *Determine the effective radius of the shell as follows.*

$$R_{eff} = 0.5D_i = 0.5 \text{ x } 150.25 = 75.125 \text{ in}$$

Step 2: *Calculate the limit of reinforcement along the vessel wall.*

For set-in, integrally reinforced nozzles:

$$L_R = \min\left[\sqrt{R_{eff}t}, 2R_n\right] = \min\left[\sqrt{75.125 \text{ x } 1.6875}, 2 \text{ x } 8.125\right] = 11.2594 \text{ in}$$

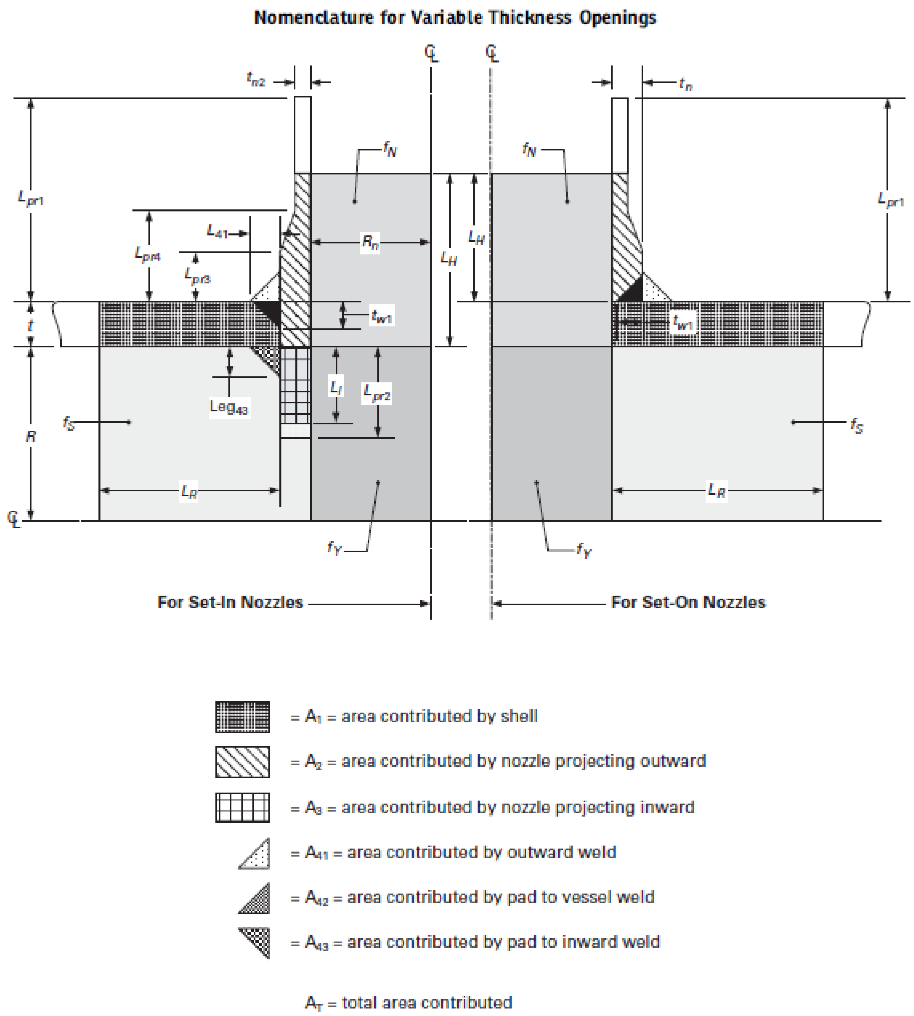


Figure 2: Nomenclature for Variable Thickness Openings

Step 3: *Calculate the limit of reinforcement along the nozzle wall projecting outside the vessel surface.*

$$L_{H1} = \min[1.5t, t_e] + \sqrt{R_n t_n} = \min[1.5 \text{ x } 1.6875, 0] + \sqrt{8.125 \text{ x } 4.625} = 6.1301 \text{ in}$$

(Since there is no reinforcing pad, t_e is zero)

$L_{H2} = L_{pr1} = 14.1875$ in (L_{pr1} is the nozzle external projection)

$$L_{H3} = 8(t + t_e) = 8(1.6875 + 0) = 13.5 \text{ in}$$

$$L_H = \min[L_{H1}, L_{H2}, L_{H3}] + t = \min[6.1301, 14.1875, 13.5] + 1.6875 = 7.8176 \text{ in}$$

Step 4: *Calculate the limit of reinforcement along the nozzle wall projecting inside the vessel surface, if applicable.*

$$L_{I1} = \sqrt{R_n t_n} = \sqrt{8.125 \text{ x } 4.625} = 6.1301 \text{ in}$$

$L_{I2} = L_{pr2} = 0.0$ in (L_{pr2} is the nozzle internal projection)

L_{I3} = $8(t + t_e)$ = $8(1.6875 + 0)$ = 13.5 in

L_H = $\min[L_{H1}, L_{H2}, L_{H3}] + t$ = $\min[6.1301, 14.1875, 13.5] + 1.6875$ = 7.8176 in

L_I = $\min[L_{I1}, L_{I2}, L_{I3}]$ = $\min[6.1301, 0 , 13.5]$ = 6.1301 in

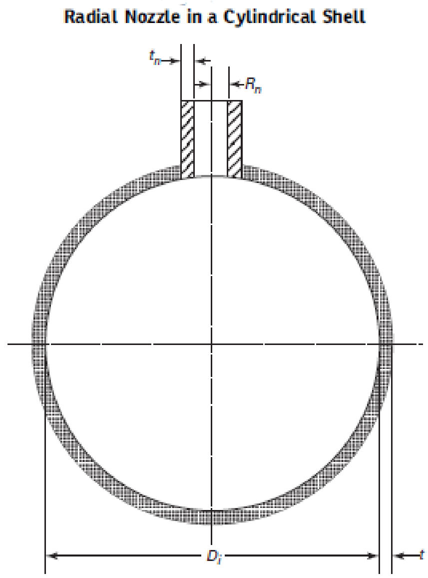


Figure 3: Radial Nozzle in a Cylindrical Shell

Step 5: *Determine the total available area near the nozzle opening. Do not include any area that falls outside the limits defined by L_H, L_R and L_I*

$$A_T = A_1 + f_{rn}(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{rp}A_5$$

$$f_{rn} = \min\left[\frac{S_n}{S}, 1\right] = \min\left[\frac{21{,}200}{22{,}400}, 1\right] = 0.9464$$

$$f_{rp} = \min\left[\frac{S_p}{S}, 1\right] = \min\left[\frac{0}{22{,}400}, 1\right] = 0.0$$ (There is no reinforcing pad)

$$t_{eff} = t + \left(\frac{A_5 f_{rp}}{L_R}\right) = 1.6875$$ (f_{rp} is zero; therefore, the second term is zero)

$$\lambda = \min\left[\left\{\frac{2R_n + t_n}{\sqrt{(D_i + t_{eff})t_{eff}}}\right\}, 12.0\right] = \min\left[\left\{\frac{2\text{x}8.125 + 4.625}{\sqrt{(150.25 + 1.6875)1.6875}}\right\}, 12.0\right] = 1.3037$$

$$A_1 = (tL_R)\max\left[\left(\frac{\lambda}{5}\right)^{0.85}, 1.0\right] = (1.6875 \text{ x } 11.2594)\max\left[\left(\frac{1.3037}{5}\right)^{0.85}, 1.0\right]$$

$$= 19.0002 \text{ in}^2$$

L_{x3} = $L_{pr3} + t$ = 7.1875 + 1.6875 = 8.875 in

A_2 = $t_n L_H$ = 4.625 x 7.8176 = 36.1564 in^2 ($L_H \le L_{x3}$)

A_3 = $t_n L_I$ = 0 (Since there is projection of nozzle inside the vessel, L_I = 0)

L_{41} = 0.375 A_{41} = $0.5L_{41}^2$ = $0.5 \text{ x } 0.375^2$ = 0.0703 in^2

L_{42} = 0 A_{42} = $0.5L_{42}^2$ = $0.5 \text{ x } 0^2$ = 0.0 in^2

L_{43} = 0 A_{43} = $0.5L_{43}^2$ = 0.5×0^2 = 0.0 in²

A_{5a} = Wt_e = 0 (There is no reinforcing pad)

A_{5b} = $L_R t_e$ = 0 (There is no reinforcing pad)

A_5 = $A_{5a} + A_{5b}$ = 0

A_T = $19.0002 + 0.9464\,(36.1564 + 0) + 0.0703 + 0 + 0 + 0$

= 53.2889 in²

<u>Step 6</u>: *Determine the applicable forces.*

$f_N = PR_{xn}L_H$

$R_{xn} = \frac{t_n}{\ln\left[1+\frac{t_n}{R_n}\right]} = \frac{4.625}{\ln\left[1+\frac{4.625}{8.125}\right]} = 10.2644$ in

$f_N = PR_{xn}L_H = 356 \times 10.2644 \times 7.8176 = 28{,}566.4985$ lb

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

$R_{xs} = \frac{t_{eff}}{\ln\left[1+\frac{t_{eff}}{R_{eff}}\right]} = \frac{1.6875}{\ln\left[1+\frac{1.6875}{75.125}\right]} = 75.9656$ in

$f_S = PR_{xs}(L_R + t_n) = 356 \times 75.9656 \times (11.2594 + 4.625) = 429{,}573.7997$ lb

$f_Y = PR_{xs}R_{nc}$ (For radial nozzles, $R_{nc} = R_n$)

$f_Y = PR_{xs}R_{nc} = 356 \times 75.9656 \times 8.125 = 219{,}730.4980$ lb

<u>Step 7</u>: *Determine the average local primary membrane stress and the general primary membrane stress at the nozzle intersection.*

$\sigma_{avg} = \frac{(f_N+f_S+f_Y)}{A_T} = \frac{(28{,}566.4985 + 429{,}573.7997 + 219{,}730.4980)}{53.2889} = 12{,}720.6753$ psi

$\sigma_{circ} = \frac{PR_{xs}}{t_{eff}} = \frac{356 \times 75.9656}{1.6875} = 16{,}025.9281$ psi

<u>Step 8</u>: *Determine the maximum local primary membrane stress at the nozzle intersection.*

$P_L = \max\left[\left(2\sigma_{avg} - \sigma_{circ}\right), \sigma_{circ}\right] = \max[(2 \times 12{,}720.6753 - 16{,}025.9281), 16{,}025.9281]$

= 16,025.9281 psi

<u>Step 9</u>: *The calculated maximum local primary membrane stress should satisfy the equation below.*

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

S_{allow} = 1.5 x 22,400 x 1.0 = 33,600 psi

P_L = 16,025.9281 psi

$P_L \leq S_{allow}$ (This equation is satisfied)

<u>Step 10</u>: *Determine the maximum allowable working pressure at the nozzle intersection.*

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

$A_p = \frac{f_N+f_S+f_Y}{P} = \frac{28{,}566.4985 + 429{,}573.7997 + 219{,}730.4980}{356}$

$$= 1904.1315 \text{ in}^2$$

$$P_{max1} = \frac{S_{allow}}{\frac{2A_p}{A_T} - \frac{R_{xs}}{t_{eff}}} = \frac{33{,}600}{\frac{2x\,1904.1315}{53.2889} - \frac{75.9656}{1.6875}} = 1270.4262 \text{ psi}$$

$$P_{max2} = S\left(\frac{t}{R_{xs}}\right) = 22{,}400\left(\frac{1.6875}{75.9656}\right) = 497.5936 \text{ psi}$$

$$P_{max} = \min[1270.4262, 497.5936] = 497.5936 \text{ psi}$$

The nozzle is acceptable because P_{max} = 497.5936 psi is greater than the specified design pressure of 356 psi.

Nozzle opening reinforcement calculations also involve analyzing nozzle spacing, strength of nozzle attachment welds, and local stresses in shell and nozzles from external loads. These topics are not addressed here in this article. Please look to future articles for treatment of these subjects.

References:

ASME Boiler and Pressure Vessel Code, Section VIII, Division 2

ASME PTB-3: ASME Section VIII – Division 2 Example Problem Manual

STRESSES IN PRESSURE VESSELS

Stresses due to internal pressure are a major part in pressure vessels. The tensile stress in thin (R/t > 10) shells due to pressure is considered to be uniform across thickness and is called membrane stress.

Due to internal pressure, the element of vessels is subjected to three principal stresses:

1. *Circumferential/hop*: This is approximately equal to PR/t, maximum on the inside surface, and minimum on the outside surface.
2. *Longitudinal*: This is approximately equal to $PR/2t$, maximum on the inside surface, and minimum on the outside surface.
3. *Radial*: This is equal to design pressure on the inside surface, and zero on the outside surface.

The difference of stress from inside to outside decreases with the increase in the R/t ratio and is insignificant above R/t ratio of 10. The radial stress above this ratio is insignificant compared to other two stresses and can be neglected. Codes give a slightly modified formula for stresses (average) to cover most of the practical shells up to *R/t* = 2. Almost all applications, the *R/t* ratio of shells will be above this value.

The shell thickness computed by the Code formulas for internal or external pressure alone is often not sufficient to withstand the combined effects of all other loadings. Detailed calculations consider the effects of each loading separately and then must be combined to give the total state of stress in that part.

Types of stress, stress categories, and allowable stresses are based on the type of loading that produced them and on the hazard they represent to the structure. Unrelenting loads produce primary stresses. Relenting loads (self-limiting) produce secondary stresses. Primary stresses must be kept lower than secondary stresses. Primary plus secondary stresses are allowed to be higher and so on. Before considering the combination of stress categories, we must first define the various types of stress and each category.

The following list of stresses describes types of stresses without regard to their effect on the vessel or component.

Tensile	Bearing	Load controlled
Compressive	Axial	Strain controlled
Normal	Discontinuity	Circumferential
Shear	Principal	Longitudinal
Membrane	Thermal	Radial
Bending	Tangential	

STRESS CATEGORIES

The foregoing list provides the categories and subcategories. Stress categories are defined by the type of loading which produces them and the hazard they represent to the vessel.

1) *Primary Stress*
    a. General membrane stress, P_m
    b. Local membrane stress, P_L
    c. Bending stress, P_b

2) *Secondary Stress*
    a. Secondary membrane stress, Q_m
    b. Secondary bending stress, Q_b
3) *Peak Stress*, F

The primary stress is related to mechanical loading directly and satisfies force and moment equilibrium. Primary stress that exceeds the yield stress will result in failure. By contrast, secondary stresses are those arising from geometric discontinuities or stress concentrations. For an increasing external load, at any point, both primary and secondary stresses increase in proportion to this load, until the yield point is reached. But secondary stresses are termed self-limiting; that is, once the yield point has been passed locally around the stress concentration, the direct relationship between load and stress is broken, due to the reduced post-yield stiffness of the material. This contrasts with primaries (sometimes called "load-controlled" stresses) that will continue to increase in overall magnitude, in direct proportion to the applied loads, irrespective of the shape of the stress-strain curve, until failure.

In a region away from any discontinuities, only primary stress will arise. Secondary stress cannot arise alone however – at a discontinuity, the secondary stress will be superimposed on the underlying primary stress.

Peak stress is the highest stress condition in a structure and is usually due to stress concentration caused by an abrupt change in geometry. This stress is important in considering fatigue failure because of cyclic load application.

PRIMARY STRESSES

Primary stresses are normal or shear stresses which are required to satisfy equilibrium. They are produced by mechanical loads (load controlled), such as pressure load, and when exceeding the yield strength can result in failure or gross distortion. The basic characteristic of a primary stress is that it is not self-limiting. Thermal stresses from thermal gradients and imposed displacements are never classified as primary stresses.

Primary stresses cause ductile rupture or a complete loss of load-carrying capability due to plastic collapse of the structure upon a single application of load. The purpose of the code limits on primary stress is to prevent gross plastic deformation and to provide a nominal factor of safety on the ductile burst pressure.

Primary general stresses are divided into membrane and bending stresses. The need for this division is that the calculated value of a primary bending stress may be allowed to go to a higher value than that of a primary general membrane stress.

1) *Primary general*: Primary general stresses are generally due to internal or external pressure or produced by sustained external forces and moments.

    a. *Primary general membrane stress*, P_m: Primary stress is the average primary stress across a solid section; it is produced by pressure or mechanical loads; and it is remote from discontinuities such as head-shell intersections, cone-cylinder intersections, nozzles, and supports. General primary membrane stress is so distributed in the structure that no redistribution of the load occurs as result of yielding. Examples are:

        - Shells away from discontinuities due to internal pressure.
        - Compressive and tensile axial stresses due to wind.
        - Axial compression due to weight.
        - Nozzles within the limits of reinforcement due to internal pressure.

    b. *Primary general bending stress*, P_b: Primary bending is the component of primary stress proportional to the distance from the centroid of the solid section and is produced by mechanical loads. The membrane stress is the component having a constant value through the section and represents an average value. These stresses are due to pressure or mechanical loads and are through thickness. These stresses are due to sustained loads and are capable of causing collapse of vessels. There are relatively few areas where primary bending occurs:

        - Center of flat head or crown of a dished head.
        - Shallow conical head
        - In the ligaments of closely spaced openings

2) *Primary local membrane stress, PL:* A primary local membrane stress is produced by sustained loads - either by design pressure alone or by other mechanical loads and are limited to a distance of $\sqrt{\mathrm{Rt}}$ in meridian direction. These stresses have self-limiting characteristics like secondary stresses. Since they are localized, once the yield strength of the material is reached, the load is redistributed to stiffer portions of the vessel. However, since any deformation associated with yielding would be unacceptable, an allowable stress lower than secondary stress is assigned. The ability of primary local membrane stresses to redistribute after the material yields allows for a higher allowable stress but only in the local area.

Quite often the concepts of general primary membrane stress and local primary membrane stress are used interchangeably; the local primary membrane stress representing a general primary membrane stress along a local structural discontinuity. The local primary membrane stress is the average across a solid section that includes discontinuities.

The bending stresses associated with local loading are almost always classified as secondary stresses. Therefore, the membrane stresses from a WRC-107 type analysis must be broken out separately and combined with general primary stresses due to internal pressure, for example.

Examples of primary local membrane stress are:

- Where internal pressure is the origin of stress and it is at a discontinuity,
  - On the shell near a nozzle or other opening
  - Head-shell juncture
  - Cone-cylinder juncture
  - Shell-flange juncture
  - Head-skirt juncture
  - Shell-stiffening ring juncture
- Where non-pressure applied loads are the origin of stress and they are at a discontinuity,
  - Support lugs
  - Nozzle external loads
  - Beam supports
  - Major attachments

SECONDARY STRESSES

Secondary stresses arise from geometric discontinuities or stress concentrations. They are normal or shear stresses which are required to satisfy the imposed strain or displacement (continuity requirement) as opposed to being in equilibrium with the external load.

The basic characteristic of a secondary stress is that it is self-limiting. This means that local yielding and minor distortions can satisfy the conditions which caused the stress to occur. Application of secondary stress cannot cause structural failure of the vessel due to the restraint offered by the body to which the part is attached. Secondary stress can develop at structural discontinuities but is also used to describe through-thickness gradients away from structural discontinuities. Secondary stresses are also produced by sustained loads other than internal or external pressure.

Structural discontinuities that develop secondary stresses should be placed apart by at least $2.5\sqrt{\mathrm{R_m t}}$. This restriction is to eliminate the additive effects of edge moments and forces.

Secondary stresses are divided into two additional groups – membrane and bending.

1. *Secondary membrane stress*, Q_m:

   Examples are,

   - Axial thermal gradients in shells, cones, or formed heads.
   - Thermal gradients between the shell and head.
   - Thermal stress due to differential thermal expansion within a nozzle wall.

- Pressure stress at an isolated ligament.

2. *Secondary bending stress, Q_b*:

   Examples are,

   - Axial thermal gradients in shells, cones, or formed heads.
   - Thermal gradients between the shell and head.
   - Head-shell juncture.
   - Nozzles outside the limits of reinforcement due to pressure and external loading.
   - Thermal stress due to differential thermal expansion within a nozzle wall.

PEAK STRESS, F

Peak stresses are additive to primary and secondary stresses present at the point of stress concentration (such as a notch or weld discontinuity). They can also be produced by certain thermal stress. Peak stress does not cause significant distortion but may cause fatigue failure.

Peak stresses apply to both sustained loads and self-limiting loads. They are only significant in fatigue conditions or brittle materials. Peak stresses are sources of fatigue cracks and apply to normal and shear stresses. Examples are:

- Stresses at corner of a discontinuity (e.g., fillet weld or corner).
- Thermal stresses due to differential thermal expansion within a nozzle wall.
- Thermal stresses in a cladding or wall overlay.
- stress due to the notch effect (stress concentration).

STRESS CLASSIFICATIONS

The stress classifications for various parts of a pressure vessel are indicated in Table 1. It can be observed that the membrane stress is considered primary for mechanical loads. For a number of geometries and loading situations, the bending stress is considered secondary. The bending stress is considered primary when the net section experiences the applied bending moment.

Table 1: Classification of Stresses

Vessel Part	Location	Origin of Stress	Type of Stress	Classification
Cylindrical or spherical shell	Remote from discontinuities	Internal pressure	General membrane	P_m
			Through thickness gradient	Q
	Junction with head or flange	Axial thermal gradient	Membrane	Q
			Bending	Q
Any shell or head	Any section across entire vessel	External load or moment, or internal pressure	General membrane averaged across full section	P_m
		External load or moment	Bending across full section	P_m
	Near nozzle or other opening	External load or moment, or internal pressure	Local membrane	P_L
			Bending	Q
			Peak (fillet or corner)	F
	Any location	Temperature difference between shell and head	Membrane	Q
			Bending	Q

Dished head or conical head	Crown	Internal pressure	Membrane	P_m
			Bending	P_b
	Knuckle or junction to shell	Internal pressure	Membrane	P_L
			Bending	Q
Flat head	Center region	Internal pressure	Membrane	P_m
			Bending	P_b
	Junction to shell	Internal pressure	Membrane	P_L
			Bending	Q

STRESS LIMITS

Potential failure modes and the various stress limits categories are related. Limits on primary stresses are set to prevent deformation and ductile burst. The primary plus secondary limits are set to prevent plastic deformation leading to incremental collapse and to validate using an elastic analysis to perform fatigue analysis. Finally, peak stress limits are set to prevent fatigue failure due to cyclic loadings.

Table 2 below is derived from ASME VIII-2. It should be used as a guide only because ASME VIII-1 recognizes only two categories of stresses – primary membrane stress and primary bending stress. In addition, ASME VIII-2 utilizes load combinations, by which short term loads (such as seismic) are reduced when combined with other loads. It also sets the allowable limits of combined stresses for fatigue loading where secondary and peak stresses are major considerations.

Table 2: Allowable Stresses for Stress Classifications and Categories

Stress Classification or Category	Allowable Stress
General primary membrane, P_m	SE
Primary membrane stress PLUS primary bending stress across the thickness, $P_m + P_b$	1.5SE
Local primary membrane, P_L	1.5SE
Local primary membrane PLUS primary bending, $P_L + P_b$	1.5SE
Secondary membrane PLUS secondary bending, $Q_m + Q_b$	$3SE < 2F_y$
P + Q	$3SE < 2F_y$
$P_m + P_b + Q_m + Q_b$	$3SE < 2F_y$
$P_L + P_b + Q_m + Q_b$	$3SE < 2F_y$
Peak, F	S_a
P + Q + F	S_a
$P_m + P_b + Q_m + Q_b + F$	S_a
$P_L + P_b + Q_m + Q_b + F$	S_a

Notes:

F_y = minimum specified yield strength at design temperature

E = joint efficiency

S = allowable stress at design temperature

S_a = alternating stress for any given number of cycles from design fatigue curves

The term 3SE shall be used in lieu of $2F_y$ when the ratio of minimum specified yield strength to ultimate strength exceeds 0.7 or S is governed by time-dependent properties

THERMAL STRESSES

Thermal stresses are developed whenever the expansion or contraction that would occur normally as a result of heating or cooling of components is prevented. These are “secondary stresses” because they are self-limiting. That is, yielding or deformation of the part relaxes the stress. Thermal stress will not cause failure by rupture in ductile material except by fatigue over repeated application. They can, however, cause failure due to excessive deformations.

Prevention of expansion or contraction, i.e., mechanical restraints are either internal or external.

External restraint occurs when an object or a component is supported or contained in a manner that restricts thermal movement. An example of external restraint occurs when piping expands into a vessel nozzle creating a radial load on the vessel shell.

Internal restraint occurs when the temperature through an object is not uniform. Stresses from a “thermal gradient” are due to internal restraint. Stress is caused by a thermal gradient whenever the temperature distribution or variation within a member creates a differential expansion such that the natural growth of one fiber is influenced by the different growth requirements of adjacent fibers. The result is distortion or warpage.

A transient thermal gradient occurs during heat-up and cool-down cycles where the thermal gradient is changing with time. Thermal gradients may be logarithmic or linear across a vessel wall. Given a steady heat input inside or outside a tube, the heat distribution will be logarithmic if there is a temperature difference between the inside and outside the tube. This effect is significant for thick-walled vessels. A linear temperature distribution may be assumed if the wall is thin. Stress calculations are much simpler for linear distribution.

Difference Between Mechanical Stress and Thermal Stress

The fundamental difference between mechanical stresses and thermal stresses lies in the nature of the loading. Thermal stresses are a result of restraint or temperature distribution. The stress pattern must only satisfy the requirements for equilibrium of the internal forces. The result being that yielding will relax the thermal stress. If the part is loaded mechanically beyond its yield strength, the part will continue to yield until it breaks. The external load remains constant, thus the internal stresses cannot relax.

Almost all equipment (except piping) are rigid between supports. Therefore, one support is designed as fixed and other sliding. If both supports are fixed, expansion joint (bellow) is provided between them. In such case, pressure thrust is induced in both fixed supports and generally not viable to design support. For low pressures, ties are provided between two parts separated by expansion joint to prevent supports from getting pressure thrust.

Piping systems are generally not rigid and providing one fixed and rest sliding will land in vibration problems. Several supports or restraints are required to prevent vibration. Pressure thrust is not high, and design of support or restraint is viable. A system can be designed to provide sufficient flexibility. Bellows are provided for low pressures and only when the required flexibility cannot be provided.

In general, thermal stresses are considered only in secondary and peak categories. Thermal stresses that cause a distortion of the structure are categorized as secondary stresses; thermal stresses caused by suppression of thermal expansion, but that may not cause distortion, are categorized as peak stresses.

DISCONTINUITY STRESSES

Vessel sections of different thickness, material, diameter, and change in directions would all have different displacements if allowed to expand freely. However, since they are connected in a continuous structure, they must deflect and rotate together. The stresses in the respective parts at or near the juncture are called discontinuity stresses. Discontinuity stresses are local in extent but can be of very high magnitude. They are self-limiting but some stresses require to be classified as local primary membrane stresses to avoid distortion. That is, once the structure has yielded, the forces causing excessive stresses are reduced. In a typical application they will not lead to failure. Discontinuity stresses become an important factor in fatigue design where cyclic loading is a consideration. Design of juncture of two parts is a major consideration in reducing discontinuity stresses.

It is necessary to superimpose the general membrane stresses with discontinuity stresses. From superimposition of these two states of stress. The total stresses are obtained. Generally, when combined, a higher allowable stress is permitted. The designer should be aware that for designs of high pressure (>1500 psi), brittle material, or cyclic loading, discontinuity stresses may be a major consideration.

REFERENCE:

Pressure Vessel Design Manual, *by* Dennis Moss and Michael Basic

Structural Analysis and Design of Process Equipment. *By* Maan Jawad and James Farr

Pressure Vessels Design and Practice, *by* Somnath Chattopadhyay

ASME Boiler and Pressure Vessel Code, Section VIII, Division 1 and Division 2

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