SUBSECTION B
REQUIREMENTS PERTAINING TO METHODS OF FABRICATION OF PRESSURE VESSELS

PART UW
REQUIREMENTS FOR PRESSURE VESSELS FABRICATED BY WELDING

GENERAL

UW-1 SCOPE

The rules in Part UW are applicable to pressure vessels and vessel parts that are fabricated by welding and shall be used in conjunction with the general requirements in Subsection A, and with the specific requirements in Subsection C that pertain to the class of material used.

(19) UW-2 SERVICE RESTRICTIONS

(a) When vessels are to contain lethal[64] substances, either liquid or gaseous, all butt-welded joints shall be fully radiographed in accordance with UW-51, except for butt welds subject to the provisions of (2) and (3) below and UW-11(a)(4), and butt welds in stiffening rings designed under the rules of UG-29. ERW pipe or tube is not permitted to be used as a shell or nozzle in lethal service applications. When fabricated of carbon or low alloy steel, such vessels shall be postweld heat treated in accordance with Table UCS-56, unless otherwise exempted by General Note (b)(3) of Table UCS-56-1. When a vessel is to contain fluids of such a nature that a very small amount mixed or unmixed with air is dangerous to life when inhaled, it shall be the responsibility of the user and/or his designated agent to determine if it is lethal. If determined as lethal, the user and/or his designated agent [see U-2(a)] shall so advise the designer and/or Manufacturer. It shall be the responsibility of the Manufacturer to comply with the applicable Code provisions (see UCI-2 and UCD-2).

(1) The joints of various categories (see UW-3) shall be as follows:

(-a) Except for welded tubes and pipes internal to heat exchanger shells, all joints of Category A shall be Type No. (1) of Table UW-12.

(-b) All Category B and C joints shall be Type No. (1) or No. (2) of Table UW-12.

(-c) Category C joints for lap joint stub ends shall be as follows:

(-1) The finished stub end shall be attached to its adjacent shell with a Type No. (1) or Type No. (2) joint of Table UW-12. The finished stub end can be made from a forging or can be machined from plate material. [See UW-13(h).]

(-2) The lap joint stub end shall be fabricated as follows:

(+a) The weld is made in two steps as shown in Figure UW-13.5.

(+b) Before making weld No. 2, weld No. 1 is examined by full radiography in accordance with UW-51, regardless of size. The weld and fusion between the weld buildup and neck is examined by ultrasonics in accordance with Mandatory Appendix 12.

(+c) Weld No. 2 is examined by full radiography in accordance with UW-51.

(-3) The finished stub end may either conform to ASME B16.9 dimensional requirements or be made to a non-standard size, provided all requirements of this Division are met.

(-d) All joints of Category D shall be full penetration welds extending through the entire thickness of the vessel wall or nozzle wall.

(2) Radiographic examination of the welded seam in exchanger tubes and pipes, to a material specification permitted by this Division, which are butt welded without the addition of filler metal may be waived, provided the tube or pipe is totally enclosed within a shell of a vessel which meets the requirements of (a).

(3) If only one side of a heat exchanger contains a lethal substance, the other side need not be built to the rules for a vessel in lethal service if:

(-a) exchanger tubes are seamless; or

(-b) exchanger tubes conform to a tube specification permitted by this Division, are butt welded without addition of filler metal, and receive in lieu of full radiography all of the following nondestructive testing and examination:

(-1) hydrotest in accordance with the applicable specification;

(-2) pneumatic test under water in accordance with the applicable material specification, or if not specified, in accordance with SA-688;

(-3) ultrasonic or nondestructive electric examination of sufficient sensitivity to detect surface calibration notches in any direction in accordance with SA-557, S1 or S3.

No improvement in longitudinal joint efficiency is permitted because of the additional nondestructive tests.

(4) All elements of a combination vessel in contact with a lethal substance shall be constructed to the rules for lethal service.

(b) When vessels are to operate below certain temperatures designated by Part UCS (see UCS-68), or impact tests of the material or weld metal are required by Part UHA, the joints of various categories (see UW-3) shall be as follows:

(1) All joints of Category A shall be Type No. (1) of Table UW-12 except that for austenitic chromium–nickel stainless steel Types 304, 304L, 316, 316L, 321, and 347, which satisfy the requirements of UHA-51(f), Type No. (2) joints may be used.

(2) All joints of Category B shall be Type No. (1) or No. (2) of Table UW-12.

(3) All joints of Category C shall be full penetration welds extending through the entire section at the joint.

(4) All joints of Category D shall be full penetration welds extending through the entire thickness of the vessel wall or nozzle wall except that partial penetration welds may be used between materials listed in Table UHA-23 as follows:

(-a) for materials shown in UHA-51(d)(1)(-a) and UHA-51(d)(2)(-a) at minimum design metal temperatures (MDMTs) of −320°F (−196°C) and warmer;

(-b) for materials shown in UHA-51(d)(1)(-b) and UHA-51(d)(2)(-b) at MDMTs of −50°F (−45°C) and warmer.

(c) Unfired steam boilers with design pressures exceeding 50 psi (343 kPa)shall satisfy all of the following requirements:

(1) All joints of Category A (see UW-3) shall be in accordance with Type No. (1) of Table UW-12, and all joints in Category B shall be in accordance with Type No. (1) or No. (2) of Table UW-12.

(2) All butt-welded joints shall be fully radiographed except under the provisions of UW-11(a)(4) and except for ERW pipe weld seams. When using ERW pipe as the shell of an unfired steam boiler, its thickness shall not exceed ½ in. (13 mm), its diameter shall not exceed 24 in. (DN 600), and the ERW weld shall be completed using high frequency (HFI) welding.

(3) When fabricated of carbon or low-alloy steel, such vessels shall be postweld heat treated.

(4) See also U-1(g)(1), UG-16(b), and UG-125(b).

(d) Pressure vessels or parts subject to direct firing [see U-1(h)] may be constructed in accordance with all applicable rules of this Division and shall meet the following requirements:

(1) All welded joints in Category A (see UW-3) shall be in accordance with Type No. (1) of Table UW-12, and all welded joints in Category B, when the thickness exceeds $^5/_8$ in. (16 mm), shall be in accordance with Type No. (1) or No. (2) of Table UW-12. No welded joints of Type No. (3) of Table UW-12 are permitted for either Category A or B joints in any thickness.

(2) When the thickness at welded joints exceeds $^5/_8$ in. (16 mm) for carbon (P-No. 1) steels and for all thicknesses for low alloy steels (other than P-No. 1 steels), postweld heat treatment is required. For all other material and in any thickness, the requirements for postweld heat treatment shall be in conformance with the applicable Subsections of this Division. See also U-1(h), UG-16(b), and UCS-56.

(3) The user, his designated agent, or the Manufacturer of the vessel shall make available to the Inspector the calculations used to determine the design temperature of the vessel. The provisions of UG-20 shall apply except that pressure parts in vessel areas having joints other than Type Nos. (1) and (2) of Table UW-12, subject to direct radiation and/or the products of combustion, shall be designed for temperatures not less than the maximum surface metal temperatures expected under operating conditions.

UW-3 WELDED JOINT CATEGORY

The term “Category” as used herein defines the location of a joint in a vessel, but not the type of joint. The “Categories” established by this paragraph are for use elsewhere in this Division in specifying special requirements regarding joint type and degree of inspection for certain welded pressure joints. Since these special requirements, which are based on service, material, and thickness, do not apply to every welded joint, only those joints to which special requirements apply are included in the categories. The special requirements will apply to joints of a given category only when specifically so stated. The joints included in each category are designated as Category A, B, C, and D joints below. Figure UW-3 illustrates typical joint locations included in each category. Welded joints not defined by the category designations include but are not limited to Figure 5-1, sketches (a), (c), and (d) corner joints; Figure 9-5 jacket-closure-to-shell welds; and Figure 26-13 fillet welds. Unless limited elsewhere in this Division, the UW-9(a) permissible weld joint types may be used with welded joints that are not assigned a category.

(a) Category A. Longitudinal and spiral welded joints within the main shell, communicating chambers,[65] transitions in diameter, or nozzles; any welded joint within a sphere, within a formed or flat head, or within the side plates[66] of a flat-sided vessel; any butt-welded joint within a flat tubesheet; circumferential welded joints connecting hemispherical heads to main shells, to transitions in diameters, to nozzles, or to communicating chambers.[65]

(b) Category B. Circumferential welded joints within the main shell, communicating chambers,[65] 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.[65] Circumferential welded joints are butt joints if the half-apex angle, α, is equal to or less than 30 deg and angle joints when α is greater than 30 deg. (See Figure UW-3.)

(c) Category 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[65] any welded joint connecting one side plate[66] to another side plate of a flat-sided vessel.

(d) Category D. Welded joints connecting communicating chambers[65] or nozzles to main shells, to spheres, to transitions in diameter, to heads, or to flat-sided vessels, and those joints connecting nozzles to communicating chambers[65] (for nozzles at the small end of a transition in diameter, see Category B).

MATERIALS

UW-5 GENERAL

(a) Pressure Parts. Materials used in the construction of welded pressure vessels shall comply with the requirements for materials given in UG-4 through UG-15, and shall be proven of weldable quality. Satisfactory qualification of the welding procedure under Section IX is considered as proof.

(b) Nonpressure Parts. Materials used for nonpressure parts that are welded to the pressure vessel shall be proven of weldable quality as described below.

(1) For material identified in accordance with UG-10, UG-11, UG-15, or UG-93, satisfactory qualification of the welding procedure under Section IX is considered as proof of weldable quality.

(2) For materials not identifiable in accordance with UG-10, UG-11, UG-15, or UG-93, but identifiable as to nominal chemical analysis and mechanical properties, P-Number under Section IX, Table QW/QB-422, or to a material specification not permitted in this Division, satisfactory qualification of the welding procedure under Section IX is considered as proof of weldable quality. For materials identified by P-Numbers, the provisions of Section IX, Table QW/QB-422 may be followed for welding procedure qualification. The welding procedure need only be qualified once for a given nominal chemical analysis and mechanical properties or material specification not permitted in this Division.

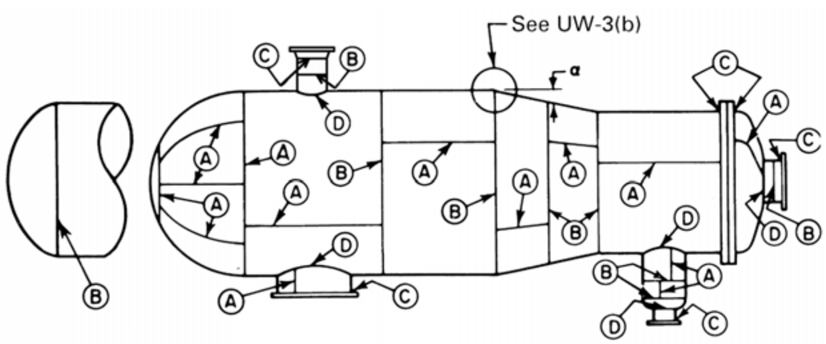
(3) Materials that cannot be identified are to be considered as unassigned material and qualified per the requirements of Section IX.

(c) Two materials of different specifications may be joined by welding provided the requirements of Section IX are met.

(d) Materials joined by the electroslag and electrogas welding processes shall be limited to ferritic steels and the following austenitic steels which are welded to

(19)

Figure UW-3
Illustration of Welded Joint Locations Typical of Categories A, B, C, and D



produce a ferrite containing weld metal: SA-240 Types 304, 304L, 316, and 316L; SA-182 F304, F304L, F316, and F316L; SA-351 CF3, CF3A, CF3M, CF8, CF8A, and CF8M.

(e) Welding of SA-841 by the electroslag or electrogas welding process is prohibited.

(f) Materials joined by the inertia and continuous drive friction welding processes shall be limited to materials assigned P-Numbers in Section IX and shall not include rimmed or semikilled steel.

UW-6 NONMANDATORY GUIDELINES FOR WELDING MATERIAL SELECTIONS

The Manufacturer is responsible for the selection of welding consumables and welding processes. These nonmandatory guidelines for welding material selections are intended to achieve suitable vessel performance for the intended service conditions, but may not be appropriate for every condition in the absence of specific technical reasons to do otherwise. The user or his designated agent should inform the Manufacturer when a specific filler metal selection is necessary to achieve satisfactory vessel performance for the intended service conditions.

(a) The tensile strength of the weld should equal or exceed that of the base metals to be joined. When base metals of different strengths are to be joined by welding, the tensile strength of the weld metal should equal or exceed that of the weaker of the two base metals.

(b) When considerations such as corrosion resistance, toughness, or fatigue resistance require selecting welding consumables or processes that produce weld joints of a lesser strength than either of the base metals, the strength of the resulting joint should be reviewed and the design adjusted as appropriate for the intended service conditions.

(c) When welding materials of like composition, the nominal composition of the weld metal should be analogous to the nominal composition of the base metal, except when creep or corrosion performance is an overriding consideration.

(d) When welding materials of different nominal composition, the nominal composition of the weld metal should be analogous to one of the base metals, or be of an acceptable alternative composition.

(e) When joining nonferrous base metals, filler metal selections should follow the recommendations of the manufacturer of the nonferrous metal or applicable industry associations.

DESIGN

UW-8 GENERAL

The rules in the following paragraphs apply specifically to the design of pressure vessels and vessel parts that are fabricated by welding and shall be used in conjunction with the general requirements for *Design* in Subsection A, and with the specific requirements for *Design* in Subsection C that pertain to the class of material used.

UW-9 DESIGN OF WELDED JOINTS (19)

(a) Permissible Types. The types of welded joints permitted for Category A, B, C, and D joints are listed in Table UW-12, together with the limiting plate thickness permitted for each type. Other types of welded joints are specifically allowed in this Subsection. Only butt-type joints may be used with the permitted welding processes in UW-27 that include the application of pressure.

(b) Welding Grooves. The dimensions and shape of the edges to be joined shall be such as to permit complete fusion and complete joint penetration. Qualification of the welding procedure, as required in UW-28, is acceptable as proof that the welding groove is satisfactory.

(c) Tapered Transitions

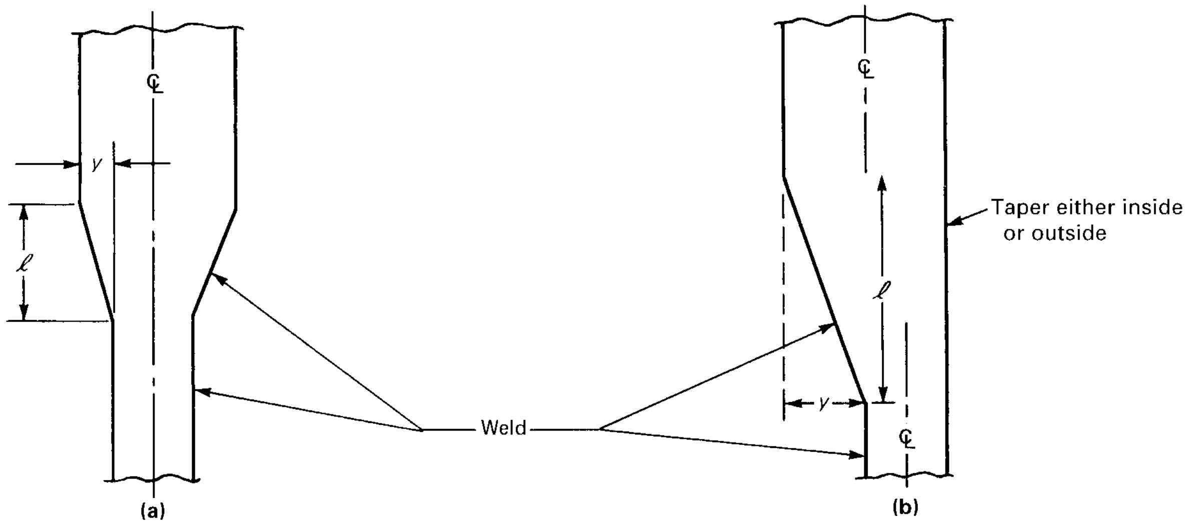
(1) A tapered transition having a length not less than *three* times the offset between the adjacent surfaces of abutting sections, as shown in Figure UW-9-1, shall be provided at joints between sections that differ in thickness by more than one-fourth of the thickness of the thinner section, or by more than $^1/_8$ in. (3 mm), whichever is less. The transition may be formed by any process that will provide a uniform taper. When the transition is formed by removing material from the thicker section, the minimum thickness of that section, after the material is removed, shall not be less than that required by UG-23(c). When the transition is formed by adding additional weld metal beyond what would otherwise be the edge of the weld, such additional weld metal buildup shall be subject to the requirements of UW-42. The butt weld may be partly or entirely in the tapered section or adjacent to it. This paragraph also applies when there is a reduction in thickness within a spherical shell or cylindrical shell course and to a taper at a Category A joint within a formed head. Provisions for tapers at circumferential, butt welded joints connecting formed heads to main shells are contained in UW-13.

(2) The centerline of a butt weld attaching a component (flange, pipe, etc.) to a thickened neck nozzle that has a taper transition angle, α, less than 71.5 deg shall be located a minimum of $1.5t_n$ from the taper (see Figure UW-9-2), where t_n is the nominal thickness of the nozzle wall at the butt weld.

(d) Except when the longitudinal joints are radiographed 4 in. (100 mm) each side of each circumferential welded intersection, vessels made up of two or more courses shall have the centers of the welded longitudinal joints of adjacent courses staggered or separated by a distance of at least five times the thickness of the thicker plate.

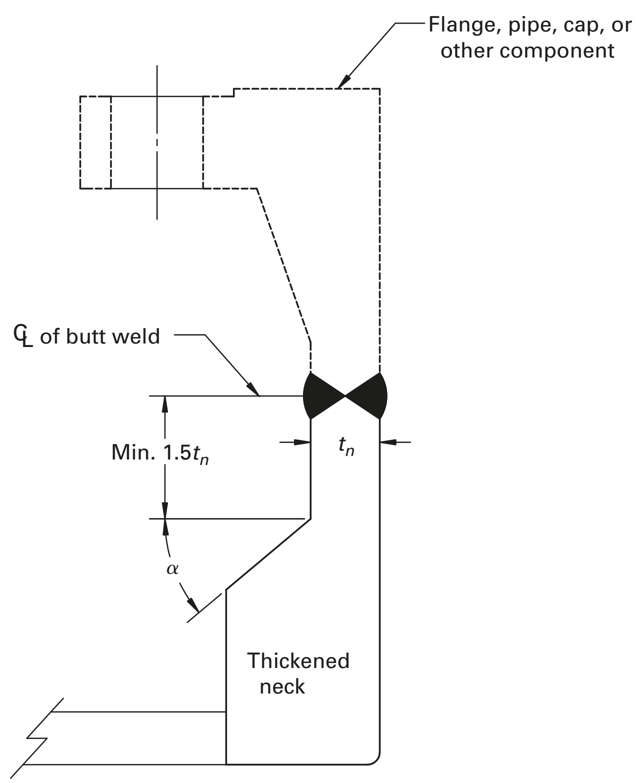
(e) Lap Joints. For lapped joints, the surface overlap shall be not less than four times the thickness of the inner plate except as otherwise provided for heads in UW-13.

Figure UW-9-1
Butt Welding of Plates of Unequal Thickness



GENERAL NOTES:
(a) $\ell \geq 3y$, where ℓ is the required length of taper and y is the offset between the adjacent surfaces of abutting sections.
(b) Length of required taper, ℓ, may include the width of the weld.
(c) In all cases, ℓ shall be not less than $3y$.

Figure UW-9-2
Butt Welding of Components to Thickened Neck Nozzles



(f) Welded Joints Subject to Bending Stresses. Except where specific details are permitted in other paragraphs, fillet welds shall be added where necessary to reduce stress concentration. Corner joints, with fillet welds only, shall not be used unless the plates forming the corner are properly supported independently of such welds. (See UW-18.)

(g) Minimum Weld Sizes. Sizing of fillet and partial penetration welds shall take into consideration the loading conditions in UG-22 but shall not be less than the minimum sizes specified elsewhere in this Division.

UW-10 POSTWELD HEAT TREATMENT

Pressure vessels and pressure vessel parts shall be postweld heat treated as prescribed in UW-40 when postweld heat treatment is required in the applicable part of Subsection C.

(19) UW-11 RADIOGRAPHIC AND ULTRASONIC EXAMINATION

(a) Full Radiography. The following welded joints shall be examined radiographically for their full length in the manner prescribed in UW-51:

(1) all butt welds in the shell and heads of vessels used to contain lethal substances [see UW-2(a)];

(2) all butt welds in the shell and heads of vessels in which the nominal thickness [see (g) below] at the welded joint exceeds $1^1/_2$ in. (38 mm), or exceeds the lesser thicknesses prescribed in UCS-57, UNF-57, UHA-33, UCL-35, or UCL-36 for the materials covered therein, or as otherwise prescribed in UHT-57, ULW-51, ULW-52(d), ULW-54, or ULT-57;

(3) all butt welds in the shell and heads of unfired steam boilers having design pressures

(-a) exceeding 50 psi (350 kPa) [see UW-2(c)];

(-b) not exceeding 50 psi (350 kPa) [see UW-2(c)] but with nominal thickness at the welded joint exceeding the thickness specified in (2) above;

(4) all butt welds in nozzles, communicating chambers, etc., with the nominal thickness at the welded joint that exceeds the thickness in (2) above or attached to the shell or heads of vessels under (1)or (3) above that are required to be fully radiographed; however, except as required by UHT-57(a), Category B and C butt welds in nozzles and communicating chambers that neither exceed NPS 10 (DN 250) nor $1^1/_8$ in. (29 mm) wall thickness do not require any radiographic examination;

(5) all Category A and D butt welds in the shell and heads of vessels where the design of the joint or part is based on a joint efficiency permitted by UW-12(a), in which case:

(-a) Category A and B welds connecting the shell or heads of vessels shall be of Type No. (1) or Type No. (2) of Table UW-12;

(-b) Category B or C butt welds [but not including those in nozzles and communicating chambers except as required in (4) above] which intersect the Category A butt welds in the shell or heads of vessels or connect seamless vessel shell or heads shall, as a minimum, meet the requirements for spot radiography in accordance with UW-52. Spot radiographs required by this paragraph shall not be used to satisfy the spot radiography rules as applied to any other weld increment.

(6) all butt welds joined by electrogas welding with any single pass greater than $1^1/_2$ in. (38 mm) and all butt welds joined by electroslag welding;

(7) all Category A welds in a tubesheet shall be of Type (1) of Table UW-12;

(8) exemptions from radiographic examination for certain welds in nozzles and communicating chambers as described in (2), (4), and (5) above take precedence over the radiographic requirements of Subsection C of this Division.

(b) Spot Radiography. Except when spot radiography is required for Category B or C butt welds by (a)(5)(-b) above, butt-welded joints made in accordance with Type No. (1) or (2) of Table UW-12 which are not required to be fully radiographed by (a) above, may be examined by spot radiography. Spot radiography shall be in accordance with UW-52. If spot radiography is specified for the entire vessel, radiographic examination is not required of Category B and C butt welds in nozzles and communicating chambers that exceed neither NPS 10 (DN 250) nor $1^1/_8$ in. (29 mm) wall thickness.

NOTE: This requirement specifies spot radiography for butt welds of Type No. (1) or No. (2) that are used in a vessel, but does not preclude the use of fillet and/or corner welds permitted by other paragraphs, such as for nozzle and manhole attachments, welded stays, flat heads, etc., which need not be spot radiographed.

(c) No Radiography. Except as required in (a) above, no radiographic examination of welded joints is required when the vessel or vessel part is designed for external pressure only, or when the joint design complies with UW-12(c).

(d) Electrogas welds in ferritic materials with any single pass greater than $1^1/_2$ in. (38 mm) and electroslag welds in ferritic materials shall be ultrasonically examined throughout their entire length in accordance with the requirements of Mandatory Appendix 12. This ultrasonic examination shall be done following the grain refining (austenitizing) heat treatment or postweld heat treatment.

(e) In addition to the requirements in (a) and (b) above, all welds made by the electron beam or laser beam process shall be ultrasonically examined for their entire length in accordance with the requirements of Mandatory Appendix 12. Ultrasonic examination may be waived if the following conditions are met:

(1) The nominal thickness at the welded joint does not exceed $^1/_4$ in. (6 mm).

(2) For ferromagnetic materials, the welds are either examined by the magnetic particle examination technique in accordance with Mandatory Appendix 6 or examined by the liquid penetrant examination technique in accordance with Mandatory Appendix 8.

(3) For nonferromagnetic materials, the welds are examined by the liquid penetrant examination technique in accordance with Mandatory Appendix 8.

(f) When radiography is required for a welded joint in accordance with (a) and (b) above, and the weld is made by the inertia and continuous drive friction welding processes, the welded joints shall also be ultrasonically examined for their entire length in accordance with Mandatory Appendix 12.

(g) For radiographic and ultrasonic examination of butt welds, the definition of nominal thickness at the welded joint under consideration shall be the nominal thickness of the thinner of the two parts joined. Nominal thickness is defined in 3-2.

(19) UW-12 JOINT EFFICIENCIES

Table UW-12 gives the joint efficiencies, E, to be used in the equations of this Division for welded joints. Except as required by UW-11(a)(5), a joint efficiency depends only on the type of joint and on the extent of examination of the joint and does not depend on the extent of examination of any other joint. The user or his designated agent [see U-2(a)] shall establish the type of joint and the extent of examination when the rules of this Division do not mandate specific requirements. Rules for determining the applicability of the efficiencies are found in the various paragraphs covering design equations [for example, see UG-24(a) and UG-27]. For further guidance, see Nonmandatory Appendix L.

(a) A value of E not greater than that given in column (a) of Table UW-12 shall be used in the design calculations for fully radiographed butt joints [see UW-11(a)], except that when the requirements of UW-11(a)(5) are not met, a value of E not greater than that given in column (b) of Table UW-12 shall be used.

(b) A value of E not greater than that given in column (b) of Table UW-12 shall be used in the design calculations for spot radiographed butt-welded joints [see UW-11(b)].

(c) A value of E not greater than that given in column (c) of Table UW-12 shall be used in the design calculations for welded joints that are neither fully radiographed nor spot radiographed [see UW-11(c)].

(d) Seamless vessel sections or heads shall be considered equivalent to welded parts of the same geometry in which all Category A welds are Type No. 1. For calculations involving circumferential stress in seamless vessel sections or for thickness of seamless heads, E = 1.0 when the spot radiography requirements of UW-11(a)(5)(-b) are met. E = 0.85 when the spot radiography requirements of UW-11(a)(5)(-b) are not met, or when the Category A or B welds connecting seamless vessel sections or heads are Type No. 3, 4, 5, 6, or 8 of Table UW-12.

(e) Welded pipe or tubing shall be treated in the same manner as seamless, but with allowable tensile stress taken from the welded product values of the stress tables, and the requirements of (d) applied.

(f) A value of E not greater than 0.80 may be used in the equations of this Division for joints completed by any of the permitted welding processes in UW-27(b) that include the application of pressure, except for electric resistance welding, provided the welding process used is permitted by the rules in the applicable parts of Subsection C for the material being welded. The quality of such welds used in vessels or parts of vessels shall be proved as follows: Test specimens shall be representative of the production welding on each vessel. They may be removed from the shell itself or from a prolongation of the shell including the longitudinal joint, or, in the case of vessels not containing a longitudinal joint, from a test plate of the same material and thickness as the vessel and welded in accordance with the same procedure. One reduced-section tension test and two side-bend tests shall be made in accordance with, and shall meet the requirements of Section IX, QW-150 and QW-160.

UW-13 ATTACHMENT DETAILS

(a) Definitions

t_h = nominal thickness of head
t_p = minimum distance from outside surface of flat head to edge of weld preparation measured as shown in Figure UW-13.2
t_s = nominal thickness of shell

(See UG-27, UG-28, UG-32, UG-34, and other paragraphs for additional definitions.)

(b) See below.

(1) Ellipsoidal, torispherical, and other types of formed heads shall be attached to the shell with a butt weld, or as illustrated in the applicable Figure UW-13.1, sketches (a), (b), (c), (d), and (i). The construction shown in sketch (e) may also be used for end heads when the thickness of the shell section of the vessel does not exceed $^5/_8$ in. (16 mm) [see also (c) below]. Limitations relative to the use of these attachments shall be as given in the sketches and related notes and in Table UW-12. Figure UW-13.1, sketches (f), (g), and (h) are examples of attachment methods which are not permissible.

(2) Formed heads, concave or convex to the pressure, shall have a skirt length not less than that shown in Figure UW-13.1, using the applicable sketch. Heads that are fitted inside or over a shell shall have a driving fit before welding.

(3) A tapered transition having a length not less than three times the offset between the adjacent surfaces of abutting sections as shown in Figure UW-13.1, sketches (j) and (k) shall be provided at joints between formed

(19)

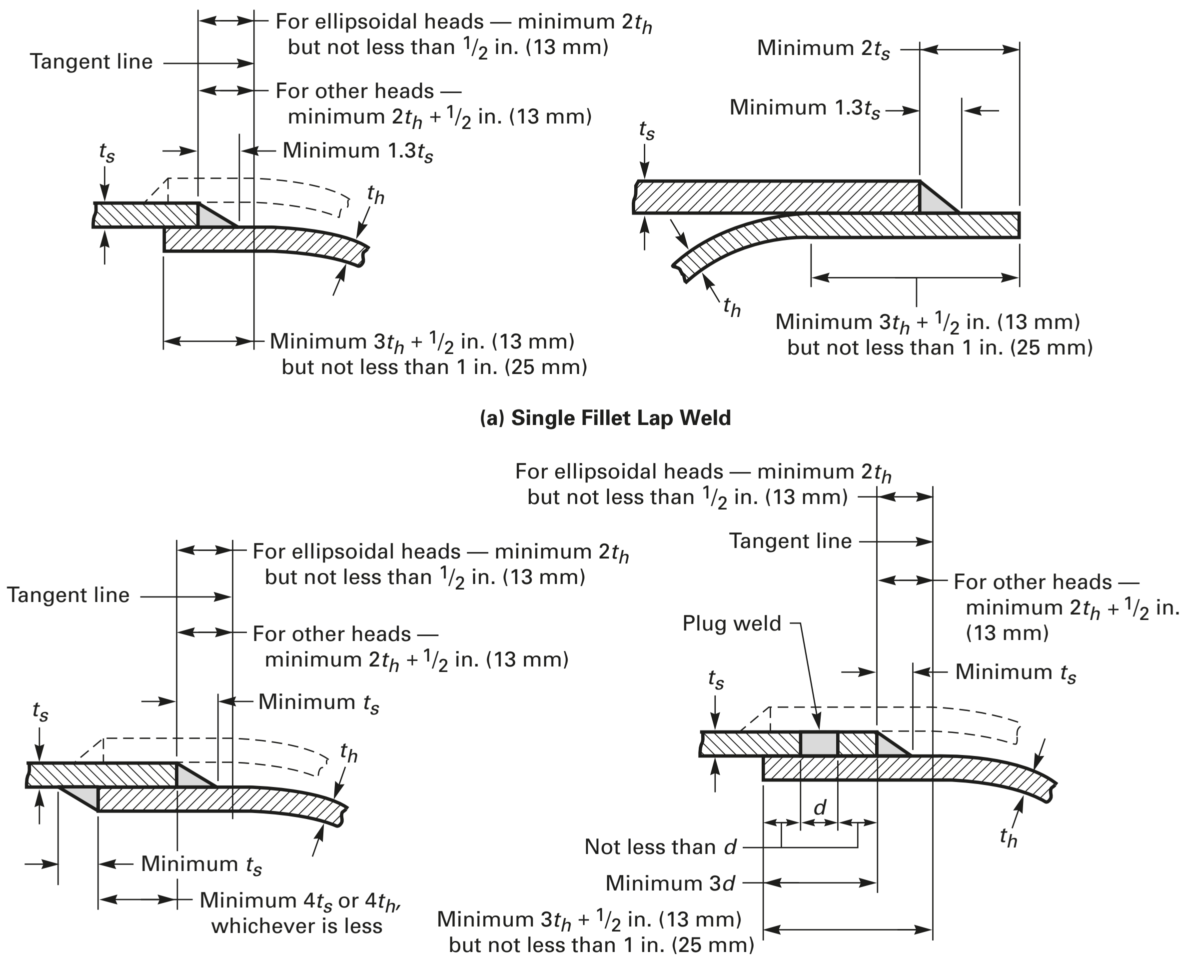
Table UW-12
Maximum Allowable Joint Efficiencies for Welded Joints

Type No.	Joint Description	Limitations	Joint Category	Extent of Radiographic or Ultrasonic Examination [Note (1), Note (2), Note (3)] (a) Full [Note (4)]	(b) Spot [Note (5)]	(c) None
(1)	Butt joints as attained by double-welding or by other means that will obtain the same quality of deposited weld metal on the inside and outside weld surfaces to agree with the requirements of UW-35. Welds using metal backing strips that remain in place are excluded.	None	A, B, C, and D	1.00	0.85	0.70
(2)	Single-welded butt joint with backing strip other than those included under (1)	*(a)* None except as in (b) below	A, B, C, and D	0.90	0.80	0.65
		(b) Circumferential butt joints with one plate offset; see UW-13(b)(4) and Figure UW-13.1, sketch (i)	A, B, and C	0.90	0.80	0.65
(3)	Single-welded butt joint without use of backing strip	Circumferential butt joints only, not over $^5/_8$ in. (16 mm) thick and not over 24 in. (600 mm) outside diameter	A, B, and C	NA	NA	0.60
(4)	Double full fillet lap joint	*(a)* Longitudinal joints not over $^3/_8$ in. (10 mm) thick	A	NA	NA	0.55
		(b) Circumferential joints not over $^5/_8$ in. (16 mm) thick	B and C [Note (6)]	NA	NA	0.55
(5)	Single full fillet lap joints with plug welds conforming to UW-17	*(a)* Circumferential joints [Note (7)] for attachment of heads not over 24 in. (600 mm) outside diameter to shells not over $^1/_2$ in. (13 mm) thick	B	NA	NA	0.50
		(b) Circumferential joints for the attachment to shells of jackets not over $^5/_8$ in. (16 mm) in nominal thickness where the distance from the center of the plug weld to the edge of the plate is not less than $1^1/_2$ times the diameter of the hole for the plug.	C	NA	NA	0.50
(6)	Single full fillet lap joints without plug welds	*(a)* For the attachment of heads convex to pressure to shells not over $^5/_8$ in. (16 mm) required thickness, only with use of fillet weld on inside of shell; or	A and B	NA	NA	0.45
		(b) for attachment of heads having pressure on either side, to shells not over 24 in. (600 mm) inside diameter and not over $^1/_4$ in. (6 mm) required thickness with fillet weld on outside of head flange only	A and B	NA	NA	0.45
(7)	Corner joints, full penetration, partial penetration, and/or fillet welded	As limited by Figure UW-13.2 and Figure UW-16.1	C and D [Note (8)]	NA	NA	NA
(8)	Angle joints	Design per U-2(g) for Category B and C joints	B, C, and D	NA	NA	NA

GENERAL NOTE: E = 1.00 for butt joints in compression.

NOTES:
(1) Some welding processes require ultrasonic examination in addition to radiographic examination, and other processes require ultrasonic examination in lieu of radiographic examination. See UW-11 for some additional requirements and limitations that may apply.
(2) Joint efficiency assignment rules of UW-12(d) and UW-12(e) shall be considered and may further reduce the joint efficiencies to be used in the required thickness calculations.
(3) The rules of UW-12(f) may be used in lieu of the rules of this Table at the Manufacturer's option.
(4) See UW-12(a) and UW-51.
(5) See UW-12(b) and UW-52.
(6) For Type No. 4 Category C joint, limitation not applicable for bolted flange connections.
(7) Joints attaching hemispherical heads to shells are excluded.
(8) There is no joint efficiency E in the design equations of this Division for Category C and D corner joints. When needed, a value of E not greater than 1.00 may be used.

Figure UW-13.1
Heads Attached to Shells

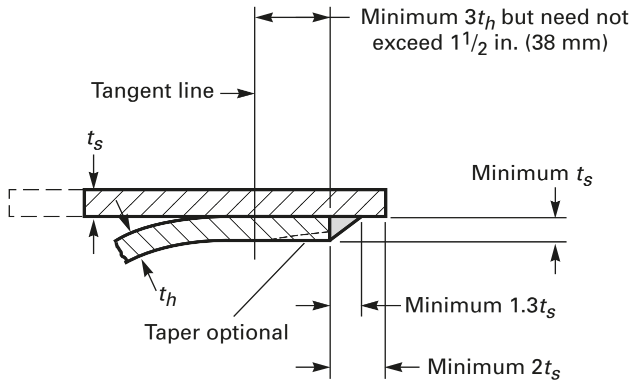


(a) Single Fillet Lap Weld

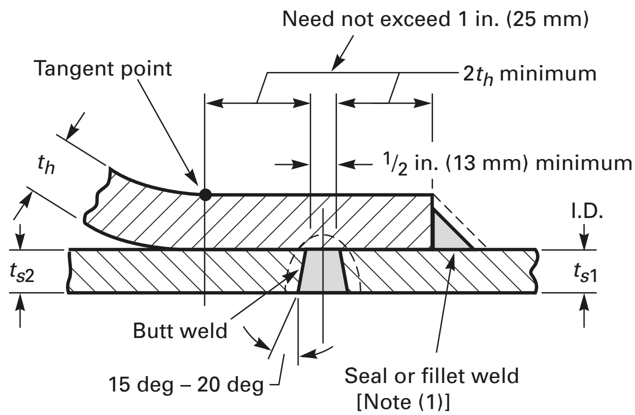
(b) Double Fillet Lap Weld

(c) Single Fillet Lap Weld With Plug Welds

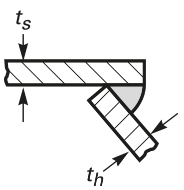
Figure UW-13.1
Heads Attached to Shells (Cont'd)



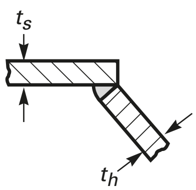
(d) Single Fillet Lap Weld



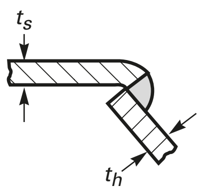
(e) Intermediate Head [See Notes (2) and (3)]



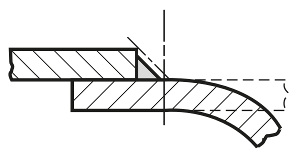
(f-1) Not Permissible



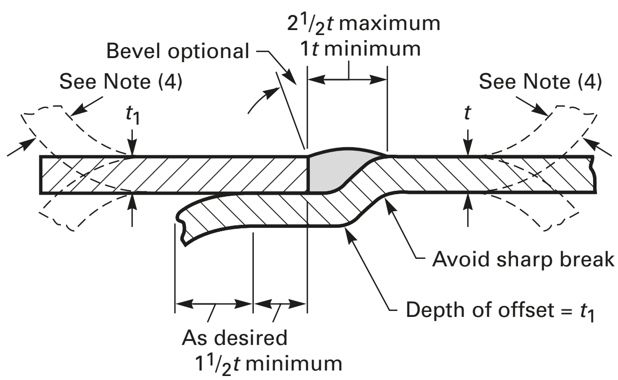
(f-2) Not Permissible



(g) Not Permissible



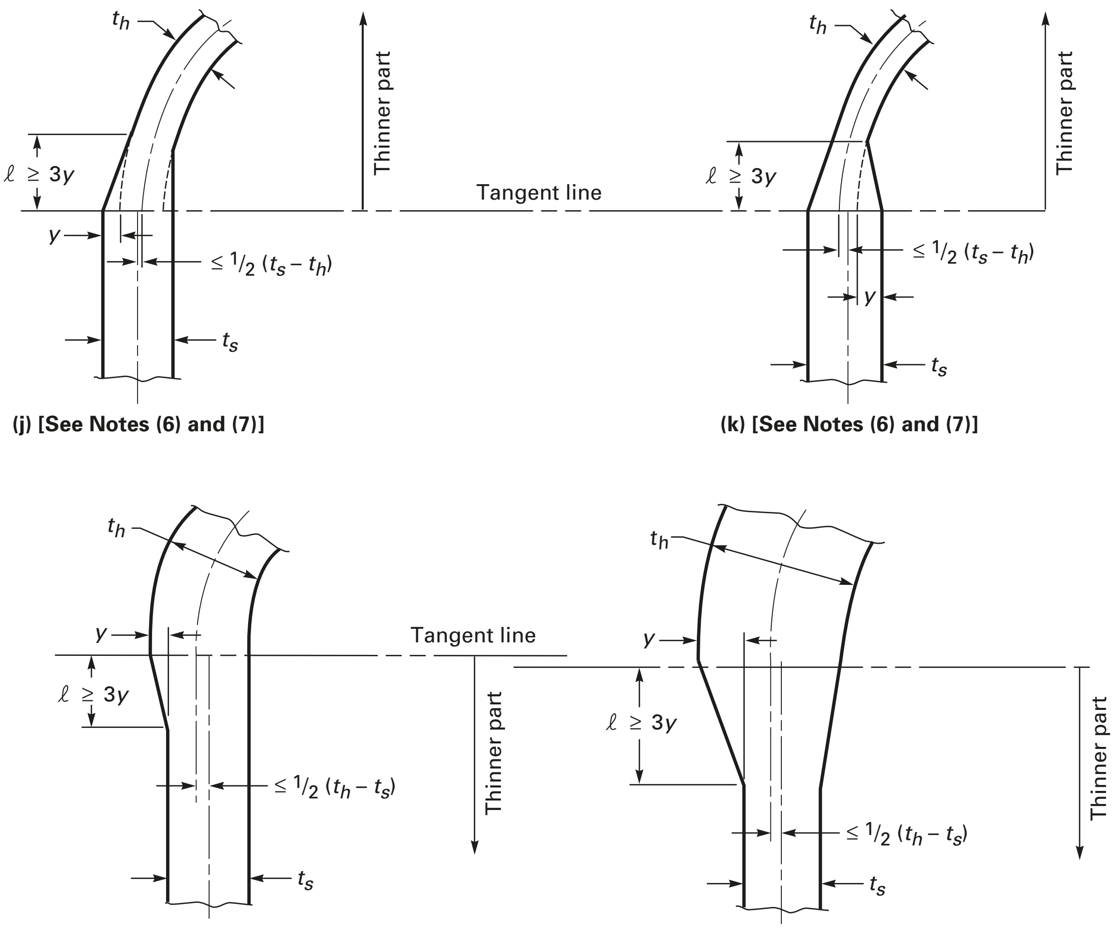
(h) Not Permissible



t or t_1 = $^5/_8$ maximum [see Note (5)]

(i) Butt Weld With One Plate Edge Offset

Figure UW-13.1
Heads Attached to Shells (Cont'd)



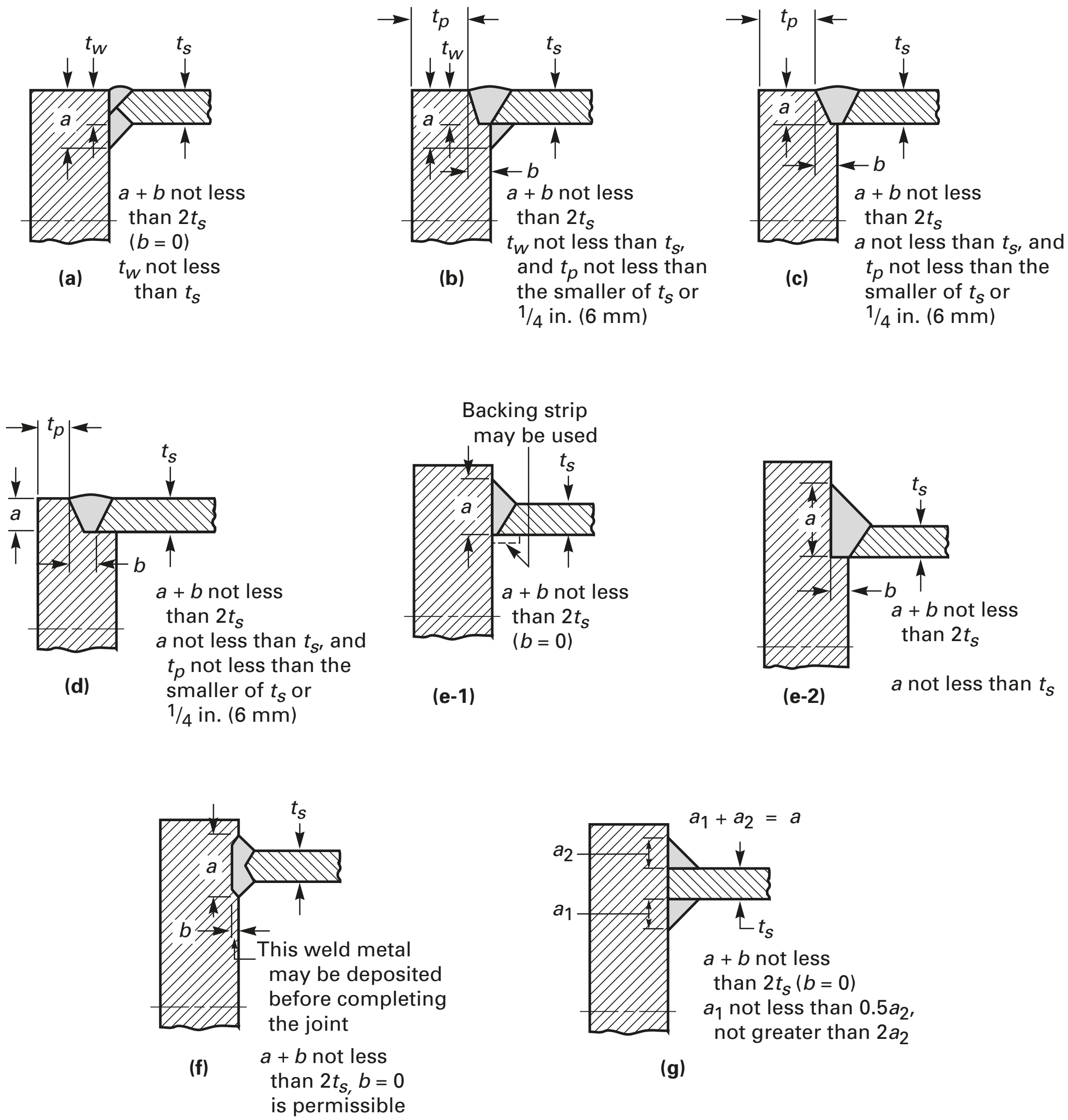
(j) [See Notes (6) and (7)] **(k) [See Notes (6) and (7)]**

(l) [See Notes (7) and (8)] **(m) [See Notes (7) and (8)]**

GENERAL NOTE: See Table UW-12 for limitations.

NOTES:
(1) See UW-13(c)(2).
(2) Butt weld and fillet weld, if used, shall be designed to take shear at $1\frac{1}{2}$ times the differential pressure than can exist.
(3) t_{s1} and t_{s2} may be different.
(4) See UW-13(b)(4) for limitation when weld bead is deposited from inside.
(5) For joints connecting hemispherical heads to shells, the following shall apply:
(a) t or t_1 = $^3/_8$ in. (10 mm) maximum.
(b) Maximum difference in thickness between t or t_1 = $^3/_{32}$ in. (2.5 mm).
(c) Use of this figure for joints connecting hemispherical heads to shells shall be noted in the "Remarks" part of the Data Report Form.
(6) In all cases, the projected length of taper, ℓ, shall be not less than $3y$.
(7) Length of required taper, ℓ, may include the width of the weld. The shell plate centerline may be on either side of the head plate centerline.
(8) In all cases, ℓ shall be not less than $3y$ when t_h exceeds t_s. Minimum length of skirt is $3t_h$ but need not exceed $1\frac{1}{2}$ in. (38 mm) except when necessary to provide required length of taper. When t_h is equal to or less than $1.25t_s$, length of skirt shall be sufficient for any required taper.

Figure UW-13.2
Attachment of Pressure Parts to Flat Plates to Form a Corner Joint



Typical Unstayed Flat Heads, Tubesheets Without a Bolting Flange, and Side Plates of Rectangular Vessels [See Note (1)]

Figure UW-13.2
Attachment of Pressure Parts to Flat Plates to Form a Corner Joint (Cont'd)

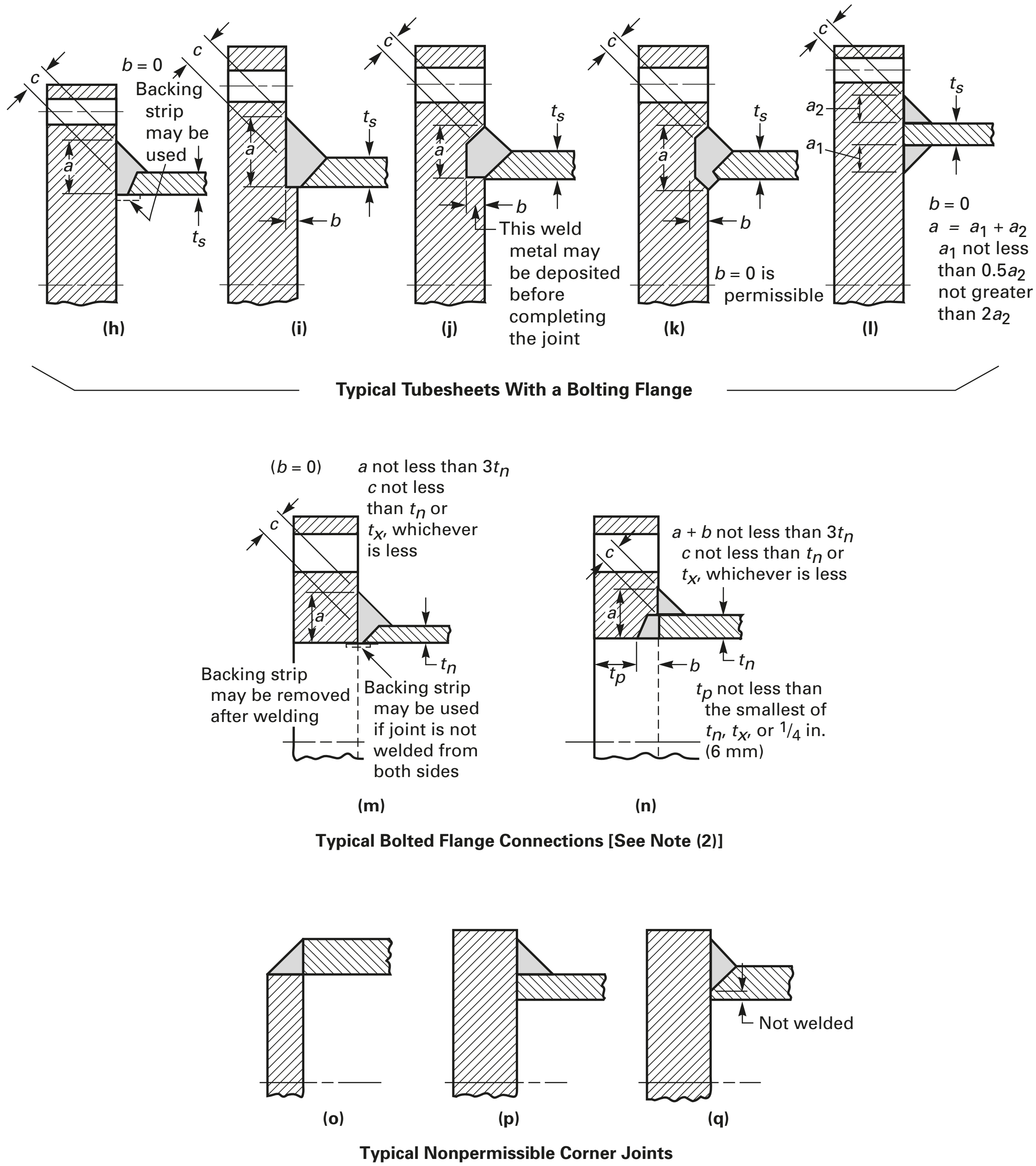
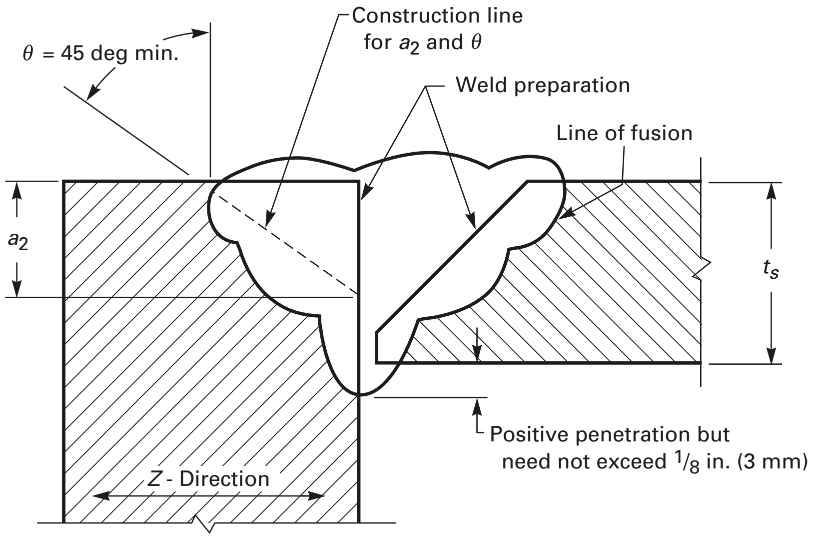
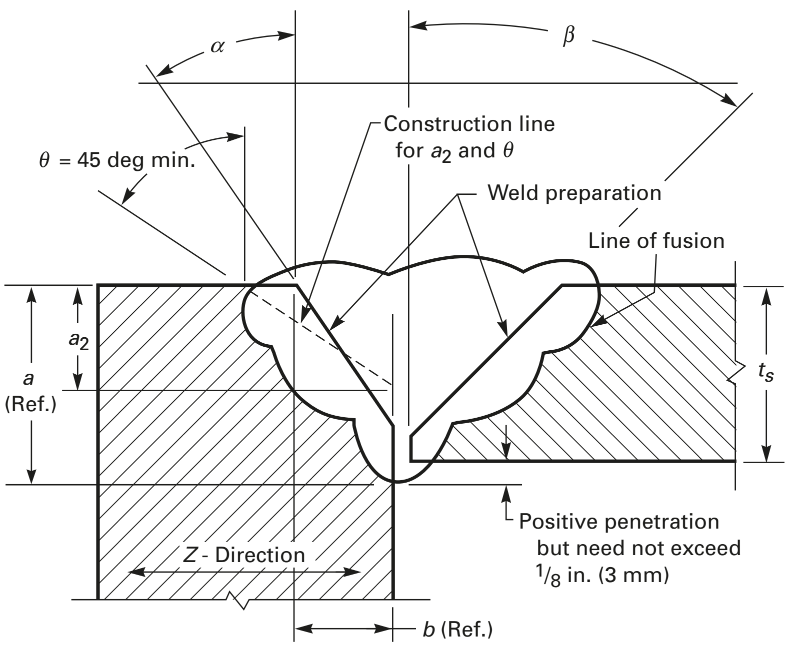


Figure UW-13.2
Attachment of Pressure Parts to Flat Plates to Form a Corner Joint (Cont'd)



K	a_2/t_s Not Less Than
0.6	0.29
0.7	0.23
0.8	0.17
0.9	0.09
1.0	0

(r) Details for One Member Beveled [See Note (3)]



K	Min. a_2/t_s for α Not Less Than 15 deg	Min. a_2/t_s for α Not Less Than 30 deg	Min. a_2/t_s for α Not Less Than 45 deg
0.6	0.85	0.55	0.29
0.7	0.81	0.47	0.23
0.8	0.74	0.38	0.17
0.9	0.58	0.23	0.09
1.0	0	0	0

See sketch (r) above for table with values of K and a_2/t_s

(s) Details for Both Members Beveled [See Note (3)]

GENERAL NOTES:

(a) $a + b$ not less than $2t_s$; c not less than $0.7t_s$ or $1.4t_r$, whichever is less.

(b) t_s and t_r are as defined in UG-34(b).

(c) Dimension b is produced by the weld preparation and shall be verified after fit up and before welding.

NOTES:

(1) For unstayed flat heads, see also UG-34.

(2) c, t_n, and t_x are as defined in 2-3.

(3) Interpolation of α and K is permitted.

heads and shells that differ in thickness by more than one-fourth the thickness of the thinner section or by more than $^1/_8$ in. (3 mm), whichever is less. When a taper is required on any formed head thicker than the shell and intended for butt-welded attachment [Figure UW-13.1, sketches (l) and (m)], the skirt shall be long enough so that the required length of taper does not extend beyond the tangent line. When the transition is formed by removing material from the thicker section, the minimum thickness of that section, after the material is removed, shall not be less than that required by UG-23(c). When the transition is formed by adding additional weld metal beyond what would otherwise be the edge of the weld, such additional weld metal buildup shall be subject to the requirements of UW-42. The centerline misalignment between shell and head shall be no greater than one-half the difference between the actual shell and head thickness, as illustrated in Figure UW-13.1, sketches (j), (k), (l), and (m).

(4) Shells and heads may be attached to shells or heads using a butt weld with one plate offset as shown in Figure UW-13.1, sketch (i). The weld bead may be deposited on the inside of the vessel only when the weld is accessible for inspection after the vessel is completed. The offset shall be smooth and symmetrical and shall not be machined or otherwise reduced in thickness. There shall be a uniform force fit with the mating section at the root of the weld. Should the offset contain a longitudinal joint, the following shall apply:

(-a) The longitudinal weld within the area of the offset shall be ground substantially flush with the parent metal prior to the offsetting operation.

(-b) The longitudinal weld from the edge of the plate through the offset shall be examined by the magnetic particle method after the offsetting operation. Cracks and cracklike defects are unacceptable and shall be repaired or removed.

(-c) As an acceptable alternative to magnetic particle examination or when magnetic particle methods are not feasible because of the nonferromagnetic character of the weld deposit, a liquid penetrant method shall be used. Cracks and cracklike defects are unacceptable and shall be repaired or removed.

(5) Non-butt-welded bolting flanges shall be attached to formed heads as illustrated in Figure 1-6.

(c) See below.

(1) Intermediate heads, without limit to thickness, of the type shown in Figure UW-13.1, sketch (e) may be used for all types of vessels provided that the outside diameter of the head skirt is a close fit inside the overlapping ends of the adjacent length of cylinder.

(2) The butt weld and fillet weld shall be designed to take shear based on $1^1/_2$ times the maximum differential pressure that can exist. The allowable stress value for the butt weld shall be 70% of the stress value for the vessel material and that of the fillet 55%. The area of the butt weld in shear is the width at the root of the weld times the length of weld. The area of the fillet weld is the minimum leg dimension times the length of weld. The fillet weld may be omitted if the construction precludes access to make the weld, and the vessel is in noncorrosive service.

(d) The requirements for the attachment of welded unstayed flat heads to shells are given in UG-34 and in (e) and (f) hereunder.

(e) When shells, heads, or other pressure parts are welded to a forged or rolled plate to form a corner joint, as in Figure UW-13.2, the joint shall meet the following requirements [see also UG-93(d)(3)]:

(1) On the cross section through the welded joint, the line of fusion between the weld metal and the forged or rolled plate being attached shall be projected on planes both parallel to and perpendicular to the surface of the plate being attached, in order to determine the dimensions a and b, respectively (see Figure UW-13.2).

(2) For flange rings of bolted flanged connections, as (**19**)
shown in Figure UW-13.2, sketches (m) and (n), the sum of a and b shall be not less than three times the nominal wall thickness of the abutting pressure part.

(3) For other components, the sum a and b shall be not less than two times the nominal wall thickness of the abutting pressure part unless the provisions of (f) are satisfied. Examples of such components are flat heads, tubesheets with or without a projection having holes for a bolted connection, and the side plates of a rectangular vessel.

(4) Other dimensions at the joint shall be in accordance with details as shown in Figure UW-13.2.

(5) Joint details that have a dimension through the joint less than the thickness of the shell, head or other pressure part, or that provide attachment eccentric thereto, are not permissible. See Figure UW-13.2, sketches (o), (p), and (q).

(f) When a multipass corner weld joint is constructed in accordance with Figure UW-13.2, sketch (r) or sketch (s), all rules in the Code pertaining to welded joints shall apply except that the requirement "$a + b$ not less than $2t_s$" of (e)(3) shall be replaced with the following requirements:

(1) A sample corner weld joint shall be prepared to qualify the weld procedure, and a sample corner weld joint shall be prepared to qualify each welder or welding operator. The Manufacturer shall prepare the sample corner weld joint with nominal thickness and configuration matching that to be employed with the following tolerances:

(-a) The sample thinner plate shall match the thickness of the production thinner plate within $\pm^1/_4$ in. (±6 mm).

(-b) The sample thicker plate shall be at least 1.5 times the thickness of the sample thinner plate.

The sample shall be sectioned, polished, and etched to clearly delineate the line of fusion. Acceptability shall be determined by measurements of the line of fusion for use in the calculations for compliance with Figure

UW-13.2, sketch (r) or sketch (s). The sample shall be free from slag, cracks, and lack of fusion. A sample corner weld shall be prepared for each P-Number, except that a sample prepared to qualify a joint made from material with a given value for K [see (4)] may be used to qualify a joint made from material having an equal or higher value for K but not vice versa.

(2) This sample corner weld joint is an addition to the Welding Procedure Specification Qualification and the Welder and Welding Operator Performance Qualification requirements of Section IX. The following essential variables apply for both the procedure and performance qualification, in addition to those of Section IX:

(-a) a change in the nominal size of the electrode or electrodes used and listed in the PQR;

(-b) a change in the qualified root gap exceeding ±$^1/_{16}$ in. (±1.5 mm);

(-c) addition or deletion of nonmetallic retainers or nonfusing metal retainers;

(-d) a change in the SFA specification filler metal classification or to a weld metal or filler metal composition not covered in the specifications;

(-e) the addition of welding positions other than those qualified;

(-f) for fill passes, a change in amperage exceeding ±25 amp, change in voltage exceeding ±3 V;

(-g) a change in contact tube to work distance exceeding $^1/_4$ in. (6 mm);

(-h) a change from single electrode to multiple electrodes, or vice versa;

(-i) a change in the electrode spacing;

(-j) a change from manual or semiautomatic to machine or automatic welding or vice versa.

(3) After production welding, the back side of the weld shall be subjected to a visual examination to ensure that complete fusion and penetration have been achieved in the root, except where visual examination is locally prevented by an internal member covering the weld.

(4) K, the ratio of through-thickness (Z direction) tensile strength to the specified minimum tensile strength, shall be taken as 0.6. Higher values for K, but not higher than 1.0, may be used if through-thickness tensile strength is determined in accordance with Specification SA-770. The test results, including the UTS in addition to the reduction in area, shall be reported on the Material Test Report, in addition to the information required by Specification SA-20 when the testing in accordance with Specification SA-770 is performed by the material manufacturer. If the testing is performed by the vessel Manufacturer, the test result shall be reported on the Manufacturer's Data Report. See UG-93(b) and UG-93(c).

(5) The maximum value of t_s [see Figure UW-13.2, sketch (r) or sketch (s)] shall be limited to 3 in. (75 mm).

(6) Both members may be beveled as shown in Figure UW-13.2, sketch (s). When the bevel angle, α, is large enough to satisfy the (e)(3) requirements, these alternative rules do not apply. When the bevel angle, α, results in weld fusion dimensions that do not satisfy the (e)(3) requirement that $a + b$ is not less than $2t_s$, the following shall be satisfied:

(-a) The angle α shall be equal to or greater than 15 deg.

(-b) The dimension a_2 shall be measured from the projected surface of the plate being attached as shown in Figure UW-13.2, sketch (s).

(-c) The angle β shall be equal to or greater than 15 deg.

(-d) When a_2/t_s is equal to or exceeds the value corresponding to the K shown in the table in Figure UW-13.2, sketch (s), the requirements in (1) and (2) need not be satisfied. When a_2/t_s is less than this value, all other requirements of (f) shall be satisfied.

(g) When used, the hub of a tubesheet or flat head shall have minimum dimensions in accordance with Figure UW-13.3 and shall meet the following requirements:

(1) When the hub is integrally forged with the tubesheet or flat head, or is machined from a forging, the hub shall have the minimum tensile strength and elongation specified for the material, measured in the direction parallel to the axis of the vessel. Proof of this shall be furnished by a tension test specimen (subsize if necessary) taken in this direction and as close to the hub as practical.[67]

(2) When the hub is machined from plate, the requirements of Mandatory Appendix 20 shall be met.

(h) When the hub of a lap joint stub end is machined from plate with the hub length in the through thickness direction of the plate, the requirements of Mandatory Appendix 20 shall be met.

(i) In the case of nozzle necks which attach to piping [see U-1(e)(1)(-a)] of a lesser wall thickness, a tapered transition from the weld end of the nozzle may be provided to match the piping thickness although that thickness is less than otherwise required by the rules of this Division. This tapered transition shall meet the limitations as shown in Figure UW-13.4.

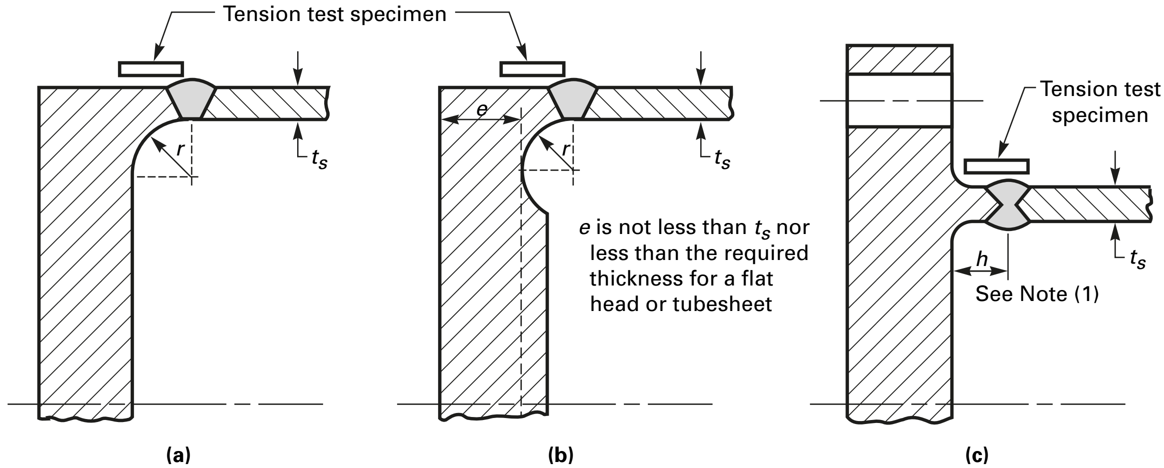
UW-14 OPENINGS IN OR ADJACENT TO WELDS

(a) Any type of opening that meets the requirements for reinforcement given in UG-37 or UG-39 may be located in a welded joint.

(b) Single openings meeting the requirements given in UG-36(c)(3) may be located in head-to-shell or Category B or C butt-welded joints, provided the weld meets the radiographic requirements in UW-51 for a length equal to three times the diameter of the opening with the center of the hole at midlength. Defects that are completely removed in cutting the hole shall not be considered in judging the acceptability of the weld.

(c) In addition to meeting the radiographic requirements of (b) above, when multiple openings meeting the requirements given in UG-36(c)(3) are in line in a

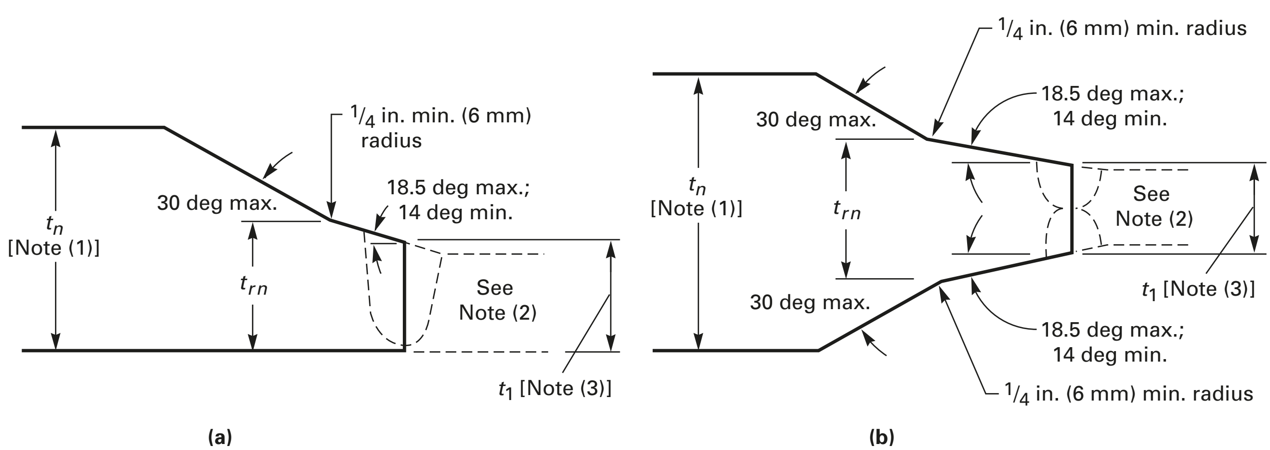
Figure UW-13.3
Typical Pressure Parts With Butt-Welded Hubs



GENERAL NOTES:
(a) Refer to Figure UG-34, sketch (b-2) for dimensional requirements.
(b) Not permissible if machined from rolled plate unless in accordance with Mandatory Appendix 20. See UW-13(g).
(c) Tension test specimen may be located inside or outside the hub.

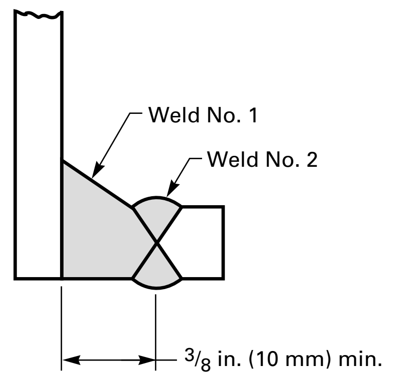
NOTE:
(1) h is the greater of $^3/_4$ in. (19 mm) or $1.5t_s$, but need not exceed 2 in. (50 mm).

Figure UW-13.4
Nozzle Necks Attached to Piping of Lesser Wall Thickness



NOTES:
(1) As defined in UG-40. t_n shall not be less than the thickness required by UG-45.
(2) Weld bevel is shown for illustration only.
(3) t_1 is not less than the greater of
(a) $0.8t_{rn}$, where t_{rn} = required thickness of seamless nozzle wall
(b) minimum wall thickness of connecting pipe

Figure UW-13.5
Fabricated Lap Joint Stub Ends for Lethal Service



head-to-shell or Category B or C butt-welded joint, the requirements of UG-53 shall be met or the openings shall be reinforced in accordance with UG-37 through UG-42.

(d) Except when the adjacent butt weld satisfies the requirement for radiography in (b) above, the edge of openings in solid plate meeting the requirements of UG-36(c)(3) shall not be placed closer than $^1/_2$ in. (13 mm) from the edge of a Category A, B, or C weld for material $1^1/_2$ in. (38 mm) thick or less.

(19) UW-15 WELDED CONNECTIONS

(a) Nozzles, other connections, and their reinforcements may be attached to pressure vessels by welding. Sufficient welding shall be provided on either side of the line through the center of the opening parallel to the longitudinal axis of the shell to develop the strength of the reinforcing parts as prescribed in UG-41 through shear or tension in the weld, whichever is applicable. The strength of groove welds shall be based on the area subjected to shear or to tension. The strength of fillet welds shall be based on the area subjected to shear (computed on the minimum leg dimension). The inside diameter of a fillet weld shall be used in figuring its length.

(b) Strength calculations for nozzle attachment welds for pressure loading are not required for the following:

(1) Figure UW-16.1, sketches (a), (b), (c), (d), (e), (f-1), (f-2), (f-3), (f-4), (g), (x-1), (y-1), and (z-1), and all the sketches in Figures UHT-18.1 and UHT-18.2

(2) openings that are exempt from the reinforcement requirements by UG-36(c)(3)

(3) openings designed in accordance with the rules for ligaments in UG-53

(c) The allowable stress values for groove and fillet welds in percentages of stress values for the vessel material, which are used with UG-41 calculations, are as follows:

(1) groove-weld tension, 74%

(2) groove-weld shear, 60%

(3) fillet-weld shear, 49%

NOTE: These values are obtained by combining the following factors: $87^1/_2$% for combined end and side loading, 80% for shear strength, and the applicable joint efficiency factors.

UW-16 MINIMUM REQUIREMENTS FOR ATTACHMENT WELDS AT OPENINGS

(a) General

(1) The terms: nozzles, connections, reinforcements, necks, tubes, fittings, pads, and other similar terms used in this paragraph define essentially the same type construction and form a Category D weld joint between the nozzle (or other term) and the shell, head, etc., as defined in UW-3(d).

(2) The location and minimum size of attachment welds for nozzles and other connections shall conform to the requirements of this paragraph in addition to the strength calculations required in UW-15.

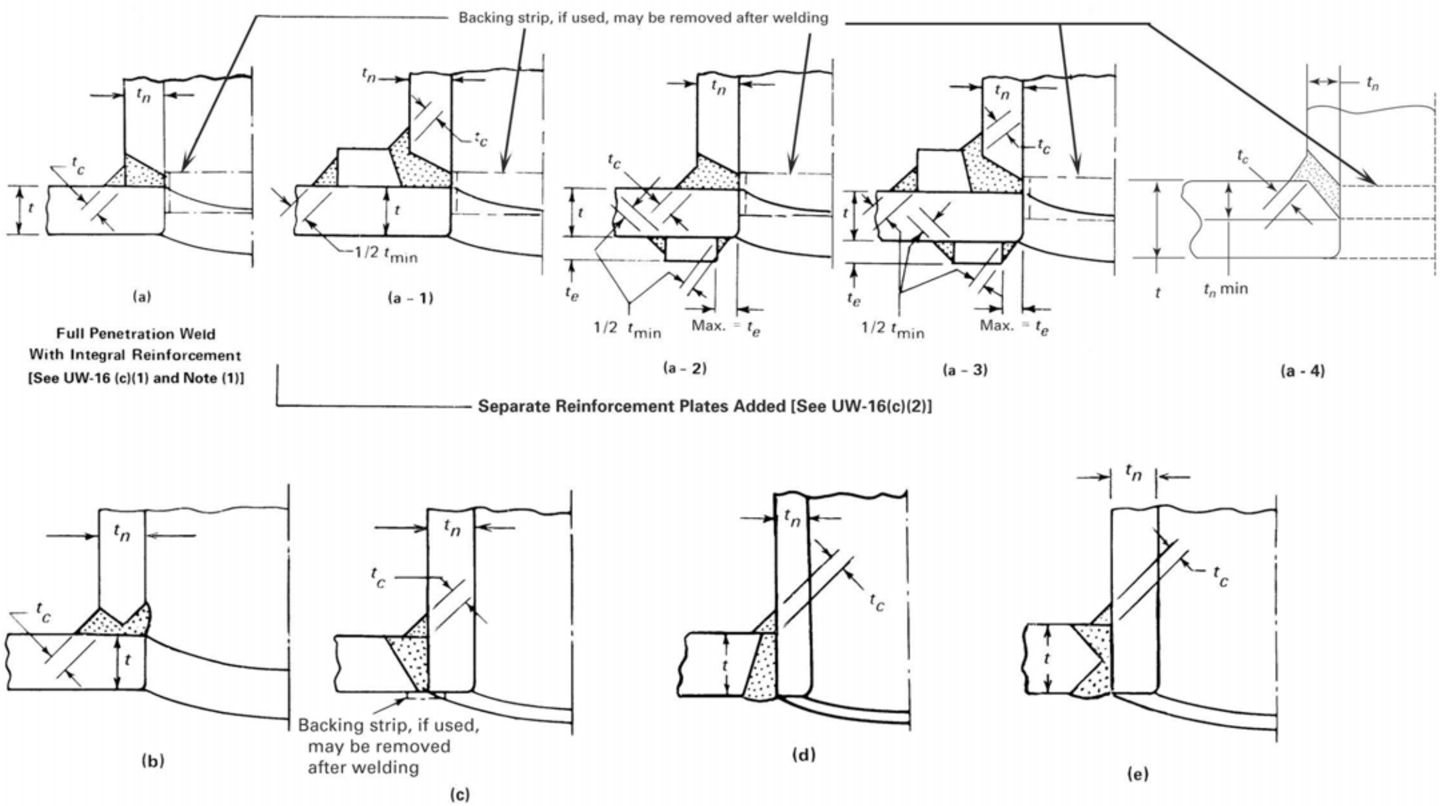
(b) Symbols. The symbols used in this paragraph and in Figures UW-16.1 and UW-16.2 are defined as follows:

D_o = outside diameter of neck or tube attached by welding on inside of vessel shell only
G = radial clearance between hole in vessel wall and outside diameter of nozzle neck or tube
r_1 = minimum inside corner radius, the lesser of $^1/_4 t$ or $^1/_8$ in. (3 mm)
Radius = $^1/_8$ in. (3 mm) minimum blend radius
t = nominal thickness of vessel shell or head,
t_1 or t_2 = not less than the smaller of $^1/_4$ in. (6 mm) or $0.7t_{\min}$
t_c = not less than the smaller of $^1/_4$ in. (6 mm) or $0.7t_{\min}$ (inside corner welds may be further limited by a lesser length of projection of the nozzle wall beyond the inside face of the vessel wall)
t_e = thickness of reinforcing plate, as defined in UG-40
$t_{\min}$ = the smaller of $^3/_4$ in. (19 mm) or the thickness of the thinner of the parts joined by a fillet, single-bevel, or single-J weld
t_n = nominal thickness of nozzle wall
t_w = dimension of attachment welds (fillet, single-bevel, or single-J), measured as shown in Figure UW-16.1

(c) Necks Attached by a Full Penetration Weld. Necks abutting a vessel wall shall be attached by a full penetration groove weld. See Figure UW-16.1, sketches (a) and (b) for examples. Necks inserted through the vessel wall may be attached by a full penetration groove weld. See Figure UW-16.1, sketches (c), (d), and (e). When complete joint penetration cannot be verified by visual inspection or other means permitted in this Division, backing strips or equivalent shall be used with full penetration welds deposited from one side.

(19)

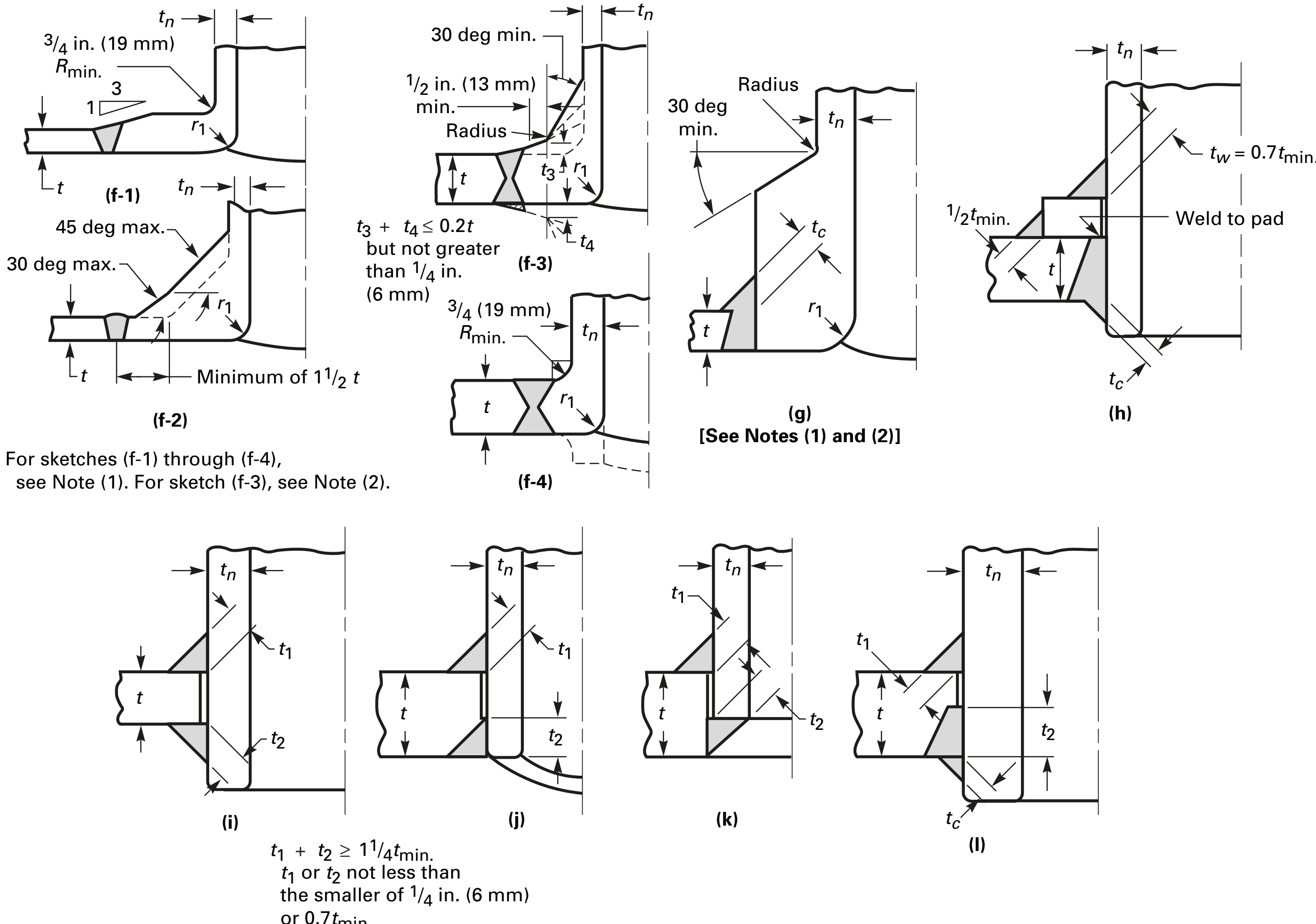
Figure UW-16.1
Some Acceptable Types of Welded Nozzles and Other Connections to Shells, Heads, etc.



Full Penetration Welds to Which Separate Reinforcement Plates May Be Added [See UW-16(c)(2) and Note (1)]

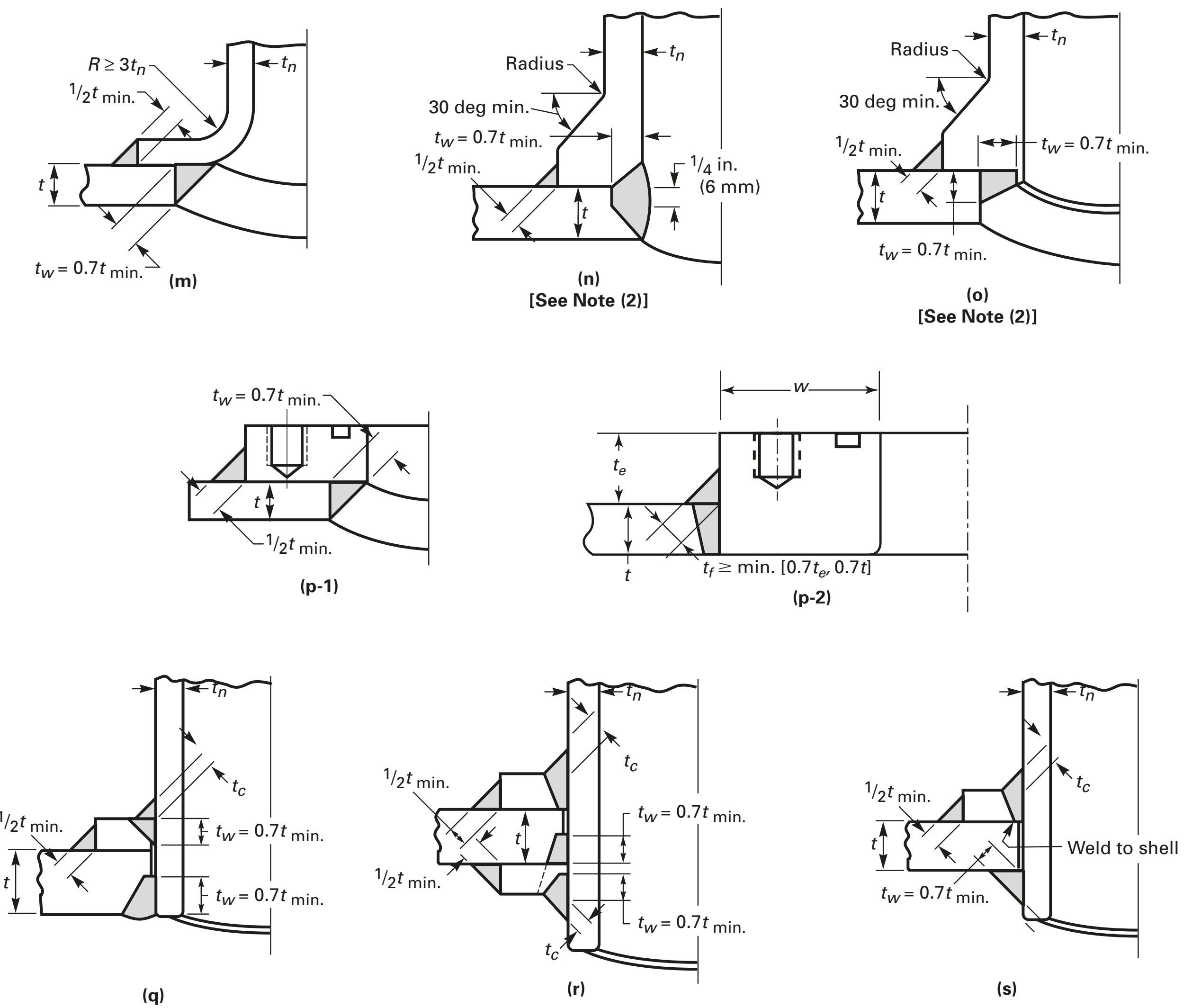
Notes follow on last page of this Figure.

Figure UW-16.1
Some Acceptable Types of Welded Nozzles and Other Connections to Shells, Heads, etc. (Cont'd)



Notes follow on last page of this Figure

Figure UW-16.1
Some Acceptable Types of Welded Nozzles and Other Connections to Shells, Heads, etc. (Cont'd)



Notes follow on last page of this Figure

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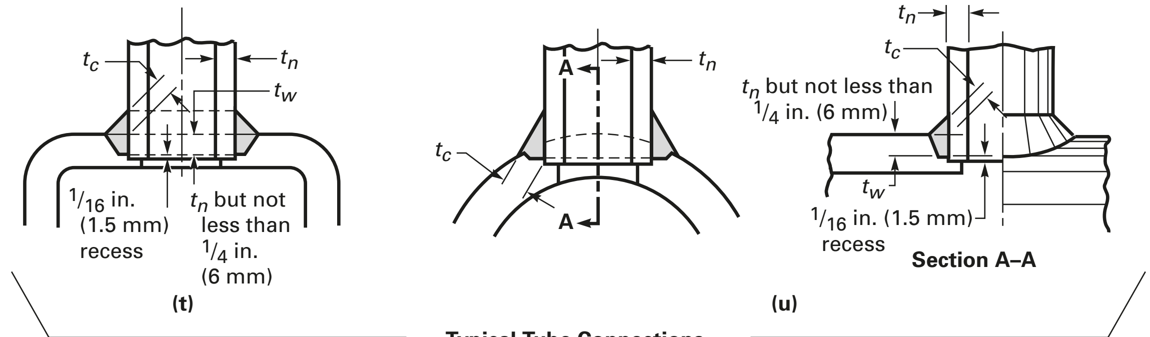
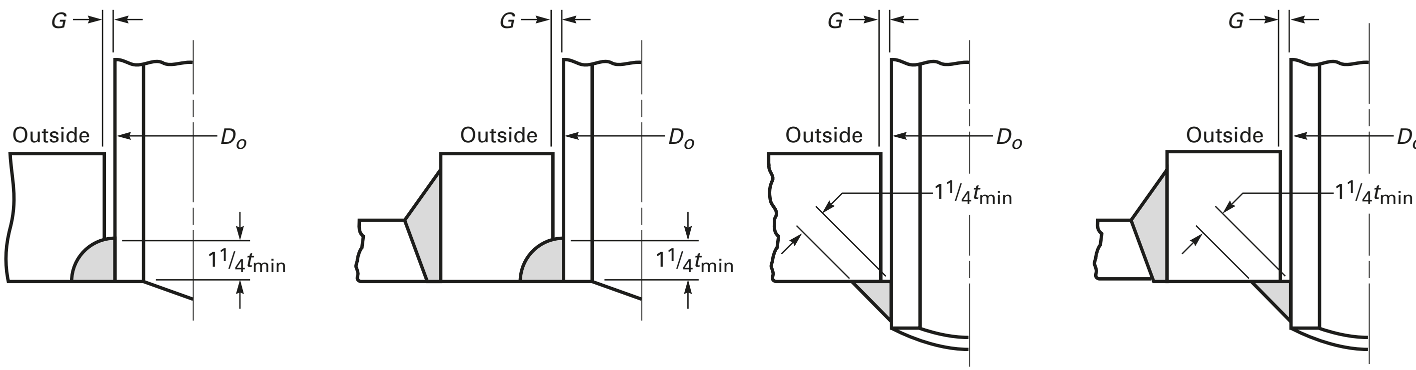
Figure UW-16.1
Some Acceptable Types of Welded Nozzles and Other Connections to Shells, Heads, etc. (Cont'd)



(When used for other than square, round, or oval headers, round off corners)



(v-1)
[See Note (3)]

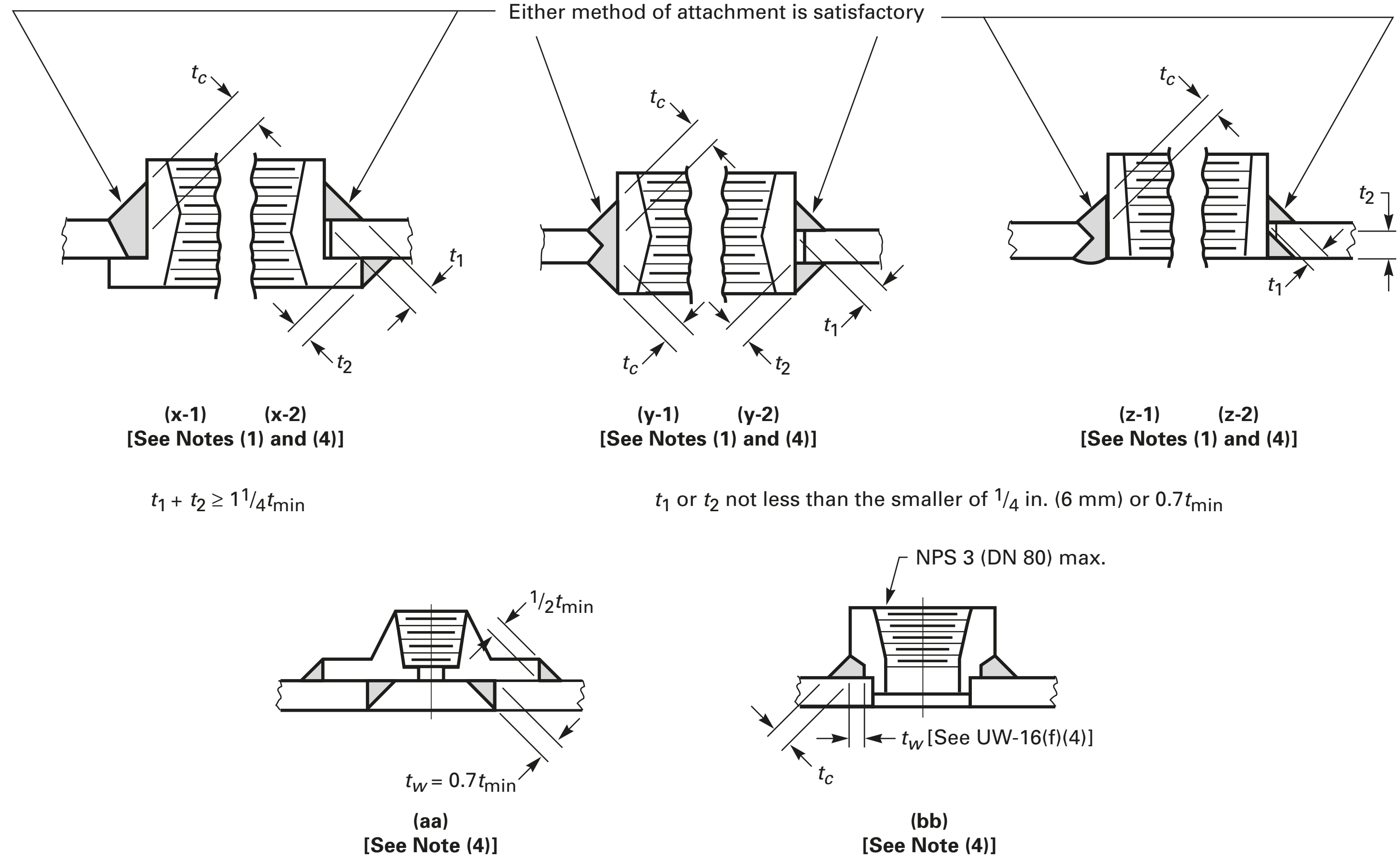
(v-2)
[See Note (3)]

(w-1)
[See Note (3)]

(w-2)
[See Note (3)]

Notes follow on last page of this Figure.

Figure UW-16.1
Some Acceptable Types of Welded Nozzles and Other Connections to Shells, Heads, etc. (Cont'd)



NOTES:
(1) Sketches (a), (b), (c), (d), (e), (f-1) through (f-4), (g), (x-1), (y-1), and (z-1) are examples of nozzles with integral reinforcement.
(2) Where the term *Radius* appears, provide a $^1/_8$ in. (3 mm) minimum blend radius.
(3) For sketches (v-1) through (w-2):
- *(a)* For applications where there are no external loads, G = $^1/_8$ in. (3 mm) max.
- *(b)* With external loads
  G = 0.005 for $D_o \leq 1$ in. (25 mm); G = 0.010 for 1 in. (25 mm) < $D_o \leq 4$ in. (100 mm); G = 0.015 for 4 in. (100 mm) < $D_o \leq 6^5/_8$ in. (170 mm)

(4) For NPS 3 (DN 80) and smaller, see exemptions in UW-16(f)(2).

If additional reinforcement is required, it shall be provided as integral reinforcement as described in (1) below, or by the addition of separate reinforcement elements (plates) attached by welding as described in (2) below.

(1) Integral reinforcement is that reinforcement provided in the form of extended or thickened necks, thickened shell plates, forging type inserts, or weld buildup which is an integral part of the shell or nozzle wall and, where required, is attached by full penetration welds. See Figure UW-16.1, sketches (a), (b), (c), (d), (e), (f-1), (f-2), (f-3), (f-4), (g), (x-1), (y-1), and (z-1) for examples of nozzles with integral reinforcement where the F factor in Figure UG-37 may be used.

(2) Separate reinforcement elements (plates) may be added to the outside surface of the shell wall, the inside surface of the shell wall, or to both surfaces of the shell wall. When this is done, the nozzle and reinforcement is no longer considered a nozzle with integral reinforcement and the F factor in UG-37(a) shall be F = 1.0. Figure UW-16.1, sketches (a-1), (a-2), and (a-3) depict various applications of reinforcement elements added to sketch (a). Any of these applications of reinforcement elements may be used with necks of the types shown in Figure UW-16.1, sketches (b), (c), (d), and (e) or any other integral reinforcement types listed in (1) above. The reinforcement 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 no nozzle neck is adjacent to the plate.

(-a) The weld at the outer edge of the reinforcement plate shall be a continuous fillet weld with a minimum throat dimension of $\frac{1}{2}t_{\min}$.

(-b) The weld at the inner edge of the reinforcement plate which does not abut a nozzle neck shall be a continuous fillet weld with a minimum throat dimension $\frac{1}{2}t_{\min}$ [see Figure UW-16.1, sketches (a-2) and (a-3)].

(-c) The weld at the inner edge of the reinforcement plate when the reinforcement plate is full penetration welded to the nozzle neck shall be a continuous fillet weld with a minimum throat dimension of t_c [see Figure UW-16.1, sketches (a-1) and (a-3)].

(-d) 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 a continuous fillet weld with a minimum throat dimension of $t_w = 0.7t_{\min}$ [see Figure UW-16.1, sketch (h)].

(d) Neck Attached by Fillet or Partial Penetration Welds

(1) Necks inserted into or through the vessel wall 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 dimension of t_1 or t_2 for each weld shall be not less than the smaller of $\frac{1}{4}$ in. (6 mm) or $0.7t_{\min}$, and their sum shall be not less than $1\frac{1}{4}t_{\min}$. See Figure UW-16.1, sketches (i), (j), (k), and (l).

If additional reinforcement is required, it may be provided in the form of extended or thickened necks, thickened shell plates, forgings, and/or separate reinforcement elements (plates) attached by welding. Weld requirements shall be the same as given in (c)(2) above, except as follows. The welds attaching the neck to the vessel wall or to the reinforcement plate shall consist of one of the following:

(-a) a single-bevel or single-J weld in the shell plate, and a single-bevel or single-J weld in each reinforcement plate. The dimension t_w of each weld shall be not less than $0.7t_{\min}$. See Figure UW-16.1, sketches (q) and (r).

(-b) a full penetration groove weld in each reinforcement plate, and a fillet, single-bevel, or single-J weld with a weld dimension t_w not less than $0.7t_{\min}$ in the shell plate. See Figure UW-16.1, sketch (s).

(2) Nozzle necks, flared necks, 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 $\frac{1}{2}t_{\min}$. The dimension t_w of the weld at the inside of the shell cutout shall not be less than $0.7t_{\min}$. See Figure UW-16.1, sketches (m), (n), (o), and (p-1). Studding-outlet-type flanges may also be attached by full-penetration welds as shown in Figure UW-16.1, sketch (p-2). (**19**)

(e) Necks and Tubes Up to and Including NPS 6 (DN 150) Attached From One Side Only. Necks and tubes not exceeding NPS 6 (DN 150) may be attached from one side only on either the outside or inside surface of the vessel.

(1) The depth of the welding groove or the throat of the fillet weld shall be at least equal to $1\frac{1}{4}t_{\min}$. The radial clearance between the vessel hole and the nozzle outside diameter at the unwelded side shall not exceed the tolerances given in Figure UW-16.1, sketches (v-1), (v-2), (w-1), and (w-2). When welded from the outside only, the neck or tube shall extend to be at least flush to the inside surface of the vessel wall. Such attachments shall satisfy the rules for reinforcement of openings, except that no material in the nozzle neck shall be counted as reinforcement.

(2) As an alternative to (1) above, when the neck or tube is attached from the outside only, 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 $\frac{1}{16}$ in. (1.5 mm) deep be provided at the bottom of the groove, in which to center the nozzle. The dimension t_w of the attachment weld shall be not less than t_n nor less than $\frac{1}{4}$ in. (6 mm). See Figure UW-16.1, sketches (t) and (u).

(f) Standard Fittings: ASME/ANSI or Manufacturer's Standard. The attachment of standard fittings shall meet the following requirements; see (g) for the attachment of bolting pads:

(1) Except as provided for in (2), (3), (4), (5), and (6) below, fittings shall be attached by a full penetration groove weld or by two fillet or partial penetration welds, one on each face of the vessel wall. The minimum weld dimensions shall be as shown in Figure UW-16.1, sketches (x), (y), (z), and (aa).

(2) Fittings not exceeding NPS 3 (DN 80) shown on Figure UW-16.1, sketches (x), (y), (z), (aa), and (bb) may be attached by welds that are exempt from size requirements with the following limitations:

(-a) UW-15(a) requirements shall be satisfied for UG-22 loadings.

(-b) For partial penetration welds or fillet welds, t_1 or t_2 shall not be less than the smaller of $^3/_{32}$ in. (2.5 mm) or $0.7t_{\min}$.

(3) See below.

(-a) Fittings not exceeding NPS 3 (DN 80), as shown in Figure UW-16.2, may be attached to vessels that are not subject to rapid fluctuations in pressure by a fillet weld deposited from the outside only without additional reinforcement other than is inherent in the fitting and its attachment to the vessel wall provided all of the following conditions are met

(-1) maximum vessel wall thickness of $^3/_8$ in. (10 mm);

(-2) the maximum size of the opening in the vessel is limited to the outside diameter of the attached pipe plus $^3/_4$ in. (19 mm), but not greater than one-half of the vessel inside diameter;

(-3) the attachment weld throat shall be the greater of the following:

(+a) the minimum nozzle neck thickness required by UG-45 for the same nominal size connection; or

(+b) that necessary to satisfy the requirements of UW-18 for the applicable loadings of UG-22.

Table UW-16.1
Minimum Thickness Requirements for Fittings

NPS	in.	mm
$^1/_8$	0.11	2.7
$^1/_4$	0.11	2.7
$^3/_8$	0.11	2.7
$^1/_2$	0.14	3.6
$^3/_4$	0.16	4.2
1	0.22	5.5
$1^1/_4$	0.30	7.5
$1^1/_2$	0.30	7.5
2	0.31	7.9
$2^1/_2$	0.37	9.5
3	0.38	9.5

(-4) the typical fitting dimension t_f as shown in Figure UW-16.2, sketch (p) shall be sufficient to accommodate a weld leg which will provide a weld throat dimension as required in (-3) above.

(-5) The openings shall meet the requirements provided in UG-36(c)(3)(-c) and UG-36(c)(3)(-d).

(-6) In lieu of the thickness requirements in UG-45, the minimum wall thickness for fittings shall not be less than that shown in Table UW-16.1 for the nearest equivalent nominal pipe size.

(-b) If the opening does not meet the requirements of (-a)(-5) or exceeds the requirements of (-a)(-2) above or (5)(-d) below in any direction, or is greater than one-half the vessel inside diameter, the part of the vessel affected shall be subjected to a proof test as required in UG-36(a)(2), or the opening shall be reinforced in accordance with UG-37 and the nozzle or other connection attached, using a suitable detail in Figure UW-16.1, if welded. In satisfying the rules for reinforcement of openings, no material in the nozzle neck shall be counted as reinforcement.

(4) Fittings not exceeding NPS 3 (DN 80) may be attached by a fillet groove weld from the outside only as shown in Figure UW-16.1, sketch (bb). The groove weld t_w shall not be less than the thickness of Schedule 160 pipe (ASME B36.10M) for the nearest equivalent pipe size. [For fittings smaller than NPS ½ (DN 15), use Schedule 160 taken from Table 8 of ASME B16.11.]

(5) Flange-type fittings not exceeding NPS 2 (DN 50), with some acceptable types such as those shown in Figure UW-16.2, may be attached without additional reinforcement other than that in the fitting and its attachment to the vessel wall. The construction satisfies the requirements of this Division without further calculation or proof test as permitted in UG-36(c)(3) provided all of the following conditions are met:

(-a) Maximum vessel wall thickness shall not exceed $^3/_8$ in. (10 mm).

(-b) Maximum design pressure shall not exceed 350 psi (2.5 MPa).

(-c) Minimum fillet leg t_f is $^3/_{32}$ in. (2.45 mm).

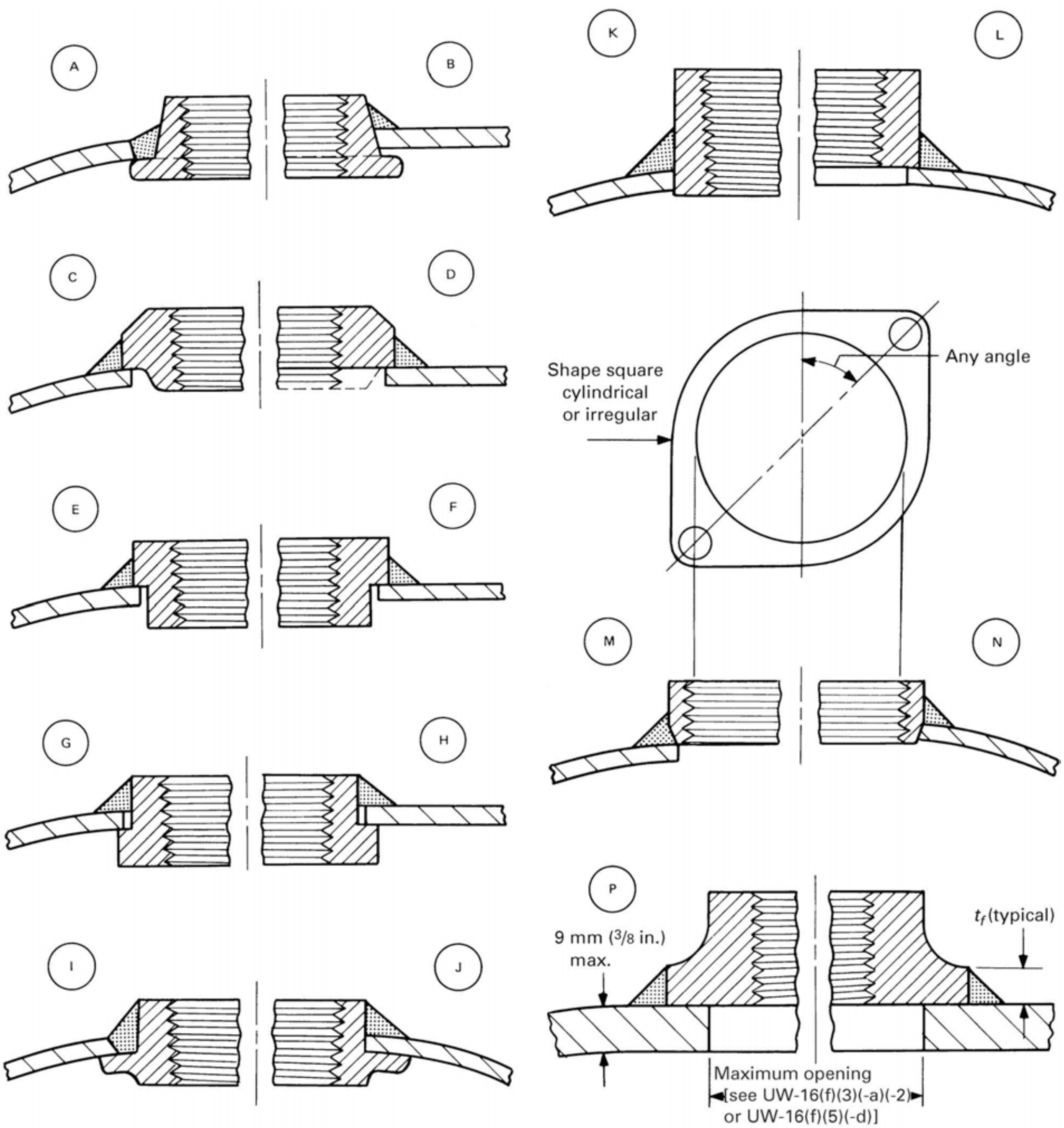
(-d) The finished opening, defined as the hole in the vessel wall, shall not exceed the outside diameter of the nominal pipe size plus $^3/_4$ in. (19 mm).

(6) Fittings conforming to Figure UW-16.2, sketch (k) not exceeding NPS 3 (DN 80) may be attached by a single fillet weld on the inside of the vessel only, provided the criteria of Figure UW-16.1, sketch (w) and (e)(1) are met.

(g) *Bolting Pads: Manufacturer's Standard*. The attachment of standard bolting pads shall meet the following requirements:

(1) Except as provided for in (2) and (3), bolting pads shall be attached by a full penetration groove weld or by two fillet or partial penetration welds, one on each face of

Figure UW-16.2
Some Acceptable Types of Small Standard Fittings



GENERAL NOTE: See UW-16(f) for limitations.

the vessel wall. The minimum weld dimensions shall be as shown in Figure UW-16.1, sketches (p), (x), (y), (z), and (aa).

(2) Bolting pads as shown in Figure UW-16.3, sketches (a) and (b) may be attached to vessels by a fillet weld deposited from the outside only with the following limitations:

(-a) The maximum vessel wall thickness is $^3/_8$ in. (10 mm), and the bolting pad outside the diameter is not greater than $4^3/_4$ in. (120 mm).

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

(-1) $4^3/_4$ in. (120 mm) for bolting pads that are installed through wall; see Figure UW-16.3, sketch (a)

(-2) $^1/_4$ in. (6 mm) less than the bolting pad diameter for those that are attached to the outside of the vessel; see Figure UW-16.3, sketch (b)

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

(-1) the minimum nozzle neck thickness required by UG-45 for the same nominal size connection

(-2) $1.0t_{\min}$

(-3) that necessary to satisfy the requirements of UW-18 for the applicable loadings of UG-22

(-d) The typical bolting pad dimension, t_f, as shown in Figure UW-16.3, sketch (a), 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.

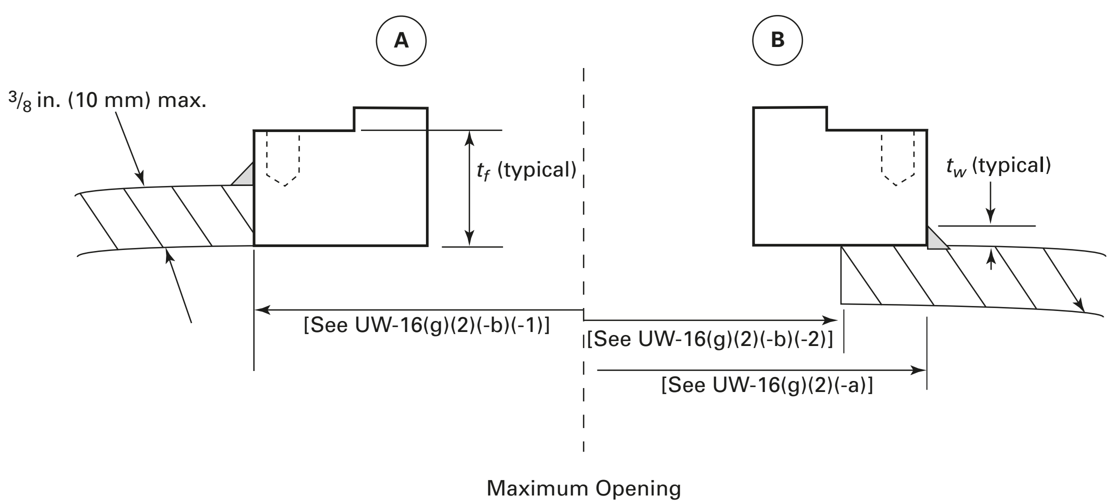
(3) If the opening exceeds the requirements of (2)(-b) above, or is greater than one-half the vessel inside diameter, the part of the vessel affected shall be subjected to a proof test as required in UG-36(a)(2), or the opening shall be reinforced in accordance with UG-37 and the nozzle or other connection attached, using a suitable detail in Figure UW-16.1, if welded.

(h) The minimum throat dimensions of fillet welds defined in UW-16.1 shall be maintained around the circumference of the attachment, except as provided below. (**19**)

(1) For a radial nozzle attachment on a cylindrical shell as shown in Figure UW-16.1, sketches (a) through (e), the fillet weld leg dimensions that meet the minimum throat dimensions shall be determined at the plane through the longitudinal axis of the cylindrical shell (other planes need not be considered), and these fillet weld leg dimensions shall be used around the circumference of the attachment.

(2) For a radial nozzle attachment on a cylindrical shell as shown in Figure UW-16.1, sketches (a) through (e), where the outside diameter of the nozzle is the same as the outside diameter of the cylindrical shell or when the outside diameter of the nozzle is too large to make a fillet weld, the fillet weld leg dimensions that meet the minimum throat dimensions shall be determined at the plane through the longitudinal axis of the cylindrical shell (other planes need not be considered), and these fillet weld leg dimensions shall be used around the circumference of the attachment to the maximum extent possible, and from that point, the fillet weld may be transitioned into the full-penetration weld.

Figure UW-16.3
Some Acceptable Types of Small Bolting Pads



GENERAL NOTE: See UW-16(g)(2) for limitations.

UW-17 PLUG WELDS

(a) Plug welds may be used in lap joints, in reinforcements around openings and in nonpressure structural attachments. They shall be properly spaced to carry their proportion of the load, but shall not be considered to take more than 30% of the total load to be transmitted.

(b) Plug weld holes shall have a diameter not less than $t + \frac{1}{4}$ in. (6 mm) and not more than $2t + \frac{1}{4}$ in. (6 mm), where t is the thickness in inches of the plate or attached part in which the hole is made.

(c) Plug weld holes shall be completely filled with weld metal when the thickness of the plate, or attached part, in which the weld is made is $\frac{5}{16}$ in. (8 mm) or less; for thicker plates or attached parts the holes shall be filled to a depth of at least half the plate thickness or $\frac{5}{16}$ of the hole diameter, whichever is larger, but in no case less than $\frac{5}{16}$ in. (8 mm).

(d) The allowable working load on a plug weld in either shear or tension shall be computed by the following formula:

(U.S. Customary Units)

$$P = 0.63S\left(d - \frac{1}{4}\right)^2$$

(SI Units)

$$P = 0.63S(d - 6)^2$$

where

d = the bottom diameter of the hole in which the weld is made
P = total allowable working load on the plug weld
S = maximum allowable stress value for the material in which the weld is made (see UG-23)

UW-18 FILLET WELDS

(a) Fillet welds may be employed as strength welds for pressure parts within the limitations given elsewhere in this Division. Particular care shall be taken in the layout of joints in which fillet welds are to be used in order to assure complete fusion at the root of the fillet.

(b) Corner or tee joints may be made with fillet welds provided the plates are properly supported independently of such welds, except that independent supports are not required for joints used for the purposes enumerated in UG-55.

(c) Figures UW-13.1 and UW-13.2 show several construction details that are not permissible.

(d) Unless the sizing basis is given elsewhere in this Division, the maximum allowable load on fillet welds shall equal the product of the weld area (based on minimum leg dimension), the maximum allowable stress value in tension of the material being welded, and a joint efficiency of 55%.

UW-19 WELDED STAYED CONSTRUCTION (19)

(a) Welded-in staybolts shall meet the following requirements:

(1) the arrangement shall substantially conform to one of those illustrated in Figure UW-19.1;

(2) the required thickness of the plate shall not exceed $1\frac{1}{2}$ in. (38 mm), except for Figure UW-19.1, sketches (e), (g), and (h). For plate thicknesses greater than $\frac{3}{4}$ in. (19 mm), the staybolt pitch shall not exceed the smaller of 20 in. (500 mm) or the limits established in UG-47(f).

(3) the provisions of UG-47 and UG-49 shall be followed; and

(4) the required area of the staybolt shall be determined in accordance with the requirements in UG-50.

(b) Welded stays, substantially as shown in Figure UW-19.2, may be used to stay jacketed pressure vessels provided:

(1) the pressure does not exceed 300 psi (2 MPa);

(2) the required thickness of the plate does not exceed $\frac{1}{2}$ in. (13 mm);

(3) the size of the fillet welds is not less than the plate thickness;

(4) the inside welds are properly inspected before the closing plates are attached;

(5) the allowable load on the fillet welds is computed in accordance with UW-18(d);

(6) the maximum diameter or width of the hole in the plate does not exceed $1\frac{1}{4}$ in. (32 mm);

(7) the welders are qualified under the rules of Section IX;

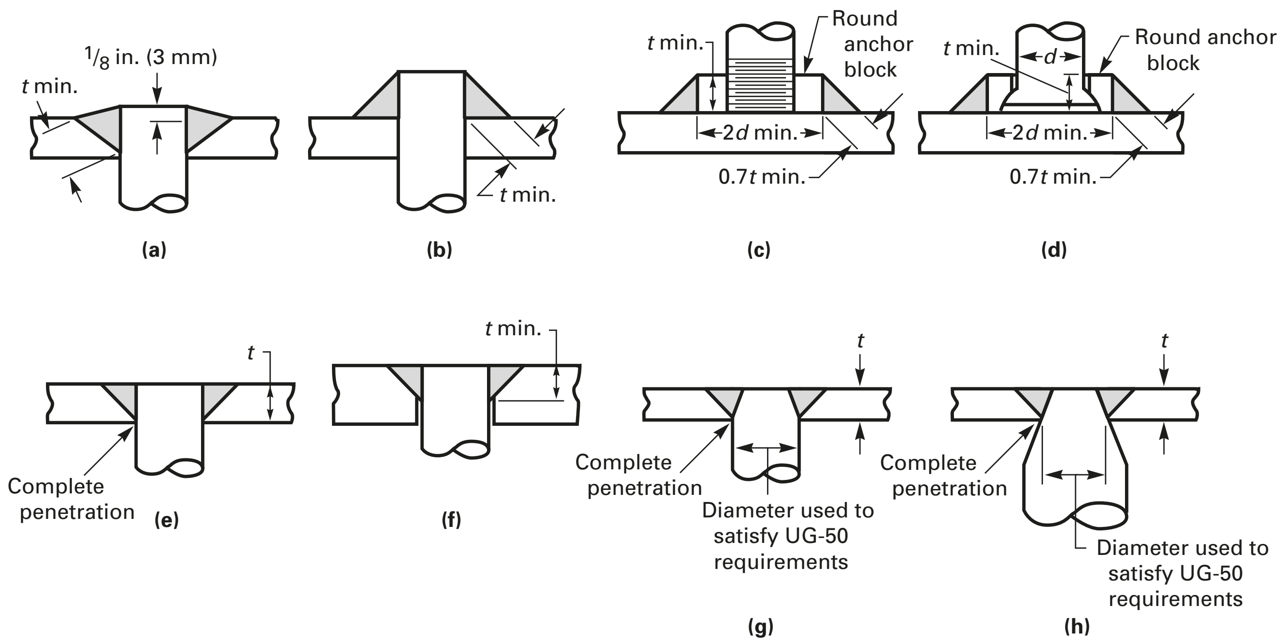
(8) the maximum spacing of stays is determined by the formula in UG-47(a), using C = 2.1 if either plate is not over $\frac{7}{16}$ in. (11 mm) thick, C = 2.2 if both plates are over $\frac{7}{16}$ in. (11 mm) thick.

(c) Welded stayed construction, as shown in Figure UW-19.2 or consisting of a dimpled or embossed plate welded to another like plate or to a plain plate, may be used, provided

(1) the welded attachment is made by fillet welds around holes or slots as shown in Figure UW-19.2 or if the thickness of the plate having the hole or slot is $\frac{1}{2}$ in. (12 mm) or less, and the hole is 1 in. (25 mm) or less in diameter, the holes may be completely filled with weld metal. The allowable load on the weld shall equal the product of the thickness of the plate having the hole or slot, the circumference or perimeter of the hole or slot, the allowable stress value in tension of the weaker of the materials being joined and a joint efficiency of 55%.

(2) the maximum allowable working pressure of the dimpled or embossed components is established in accordance with the requirements of UG-101. The joint efficiency, E, used in UG-101 to calculate the MAWP of the dimpled panel shall be taken as 0.80. This proof test may be carried out on a representative panel. If a representative panel is used, it shall be rectangular in shape and at least 5 pitches in each direction, but not less than

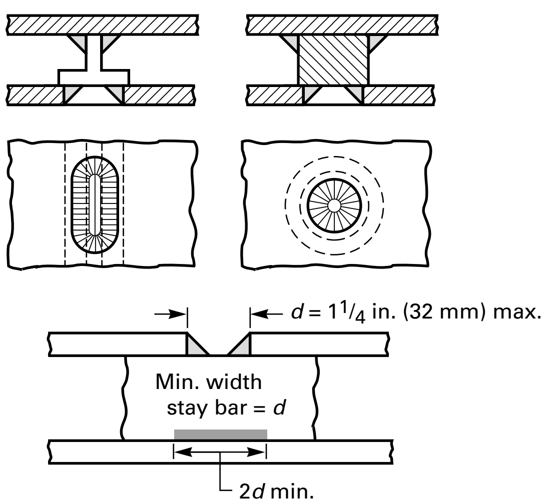
Figure UW-19.1
Typical Forms of Welded Staybolts



Legend:

t = nominal thickness of the thinner stayed plate

Figure UW-19.2
Use of Plug and Slot Welds for Staying Plates



24 in. (600 mm) in either direction. The representative panel shall utilize the same weld details as will be used in the final construction.

(3) the plain plate, if used, shall meet the requirements for braced and stayed surfaces.

(d) The welds need not be radiographed, nor need they be postweld heat treated unless the vessel or vessel part in which they occur is required to be postweld heat treated.

UW-20 TUBE-TO-TUBESHEET WELDS

UW-20.1 Scope. These rules provide a basis for establishing weld sizes and allowable joint loads for full strength and partial strength tube-to-tubesheet welds.

UW-20.2 Definitions.

(a) Full Strength Weld. A full strength tube-to-tubesheet weld is one in which the design strength is equal to or greater than the axial tube strength, F_t. When the weld in a tube-to-tubesheet joint meets the requirements of UW-20.4, it is a full strength weld and the joint does not require qualification by shear load testing. Such a weld also provides tube joint leak tightness.

(b) Partial Strength Weld. A partial strength weld is one in which the design strength is based on the mechanical and thermal axial tube loads (in either direction) that are determined from the actual design conditions. The maximum allowable axial load of this weld may be determined in accordance with UW-20.5, Nonmandatory Appendix A, or UW-18(d). When the weld in a tube-to-tubesheet joint meets the requirements of UW-20.5 or UW-18(d), it is a partial strength weld and the joint does not require qualification by shear load testing. Such a weld also provides tube joint leak tightness.

(c) Seal Weld. A tube-to-tubesheet seal weld is one used to supplement an expanded tube joint to ensure leak tightness. Its size has not been determined based on axial tube loading.

(19) **UW-20.3 Nomenclature.** The symbols described below are used for the design of tube-to-tubesheet welds.

a_c = length of the combined weld legs measured parallel to the longitudinal axis of the tube at its outside diameter
a_f = fillet weld leg
a_g = groove weld leg
a_r = minimum required length of the weld leg(s) under consideration
d_o = tube outside diameter
F_d = design strength, but not greater than F_t
f_d = ratio of the design strength to the tube strength
= 1.0 for full strength welds
= F_d/F_t for partial strength welds
F_f = fillet weld strength, but not greater than F_t
= $0.55\pi a_f\,(d_o + 0.67a_f)\,S_w$
f_f = ratio of the fillet weld strength to the design strength
= $1 - F_g/(f_dF_t)$
F_g = groove weld strength, but not greater than F_t
= $0.85\pi a_g\,(d_o + 0.67a_g)\,S_w$
F_t = axial tube strength
= $\pi t\,(d_o - t)\,S_a$
f_w = weld strength factor
= S_a/S_w
$L_{\max}$ = maximum allowable axial load in either direction on the tube-to-tubesheet joint
S = allowable stress value at the design temperature as given in the applicable part of Section II, Part D
S_a = allowable stress in tube (see S, above)
S_t = allowable stress of the material to which the tube is welded (see S, above). See UW-20.7(d)
S_w = allowable stress in weld (lesser of S_a or S_t, above)
t = nominal tube thickness

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

UW-20.4 Full Strength Welds. Full strength welds shown in Figure UW-20.1 shall conform to the following requirements:

(a) The size of a full strength weld shall be determined in accordance with UW-20.6.

(b) The maximum allowable axial load in either direction on a tube-to-tubesheet joint with a full strength weld shall be determined as follows:

(1) For loads due to pressure-induced axial forces, $L_{\max} = F_t$.

(2) For loads due to thermally induced or pressure plus thermally induced axial forces:

(-a) $L_{\max} = F_t$ for welded only tube-to-tubesheet joints, where the thickness through the weld throat is less than the nominal tube thickness t;

(-b) $L_{\max} = 2F_t$ for all other welded tube-to-tubesheet joints.

UW-20.5 Partial Strength Welds. Partial strength welds shown in Figure UW-20.1 shall conform to the following requirements:

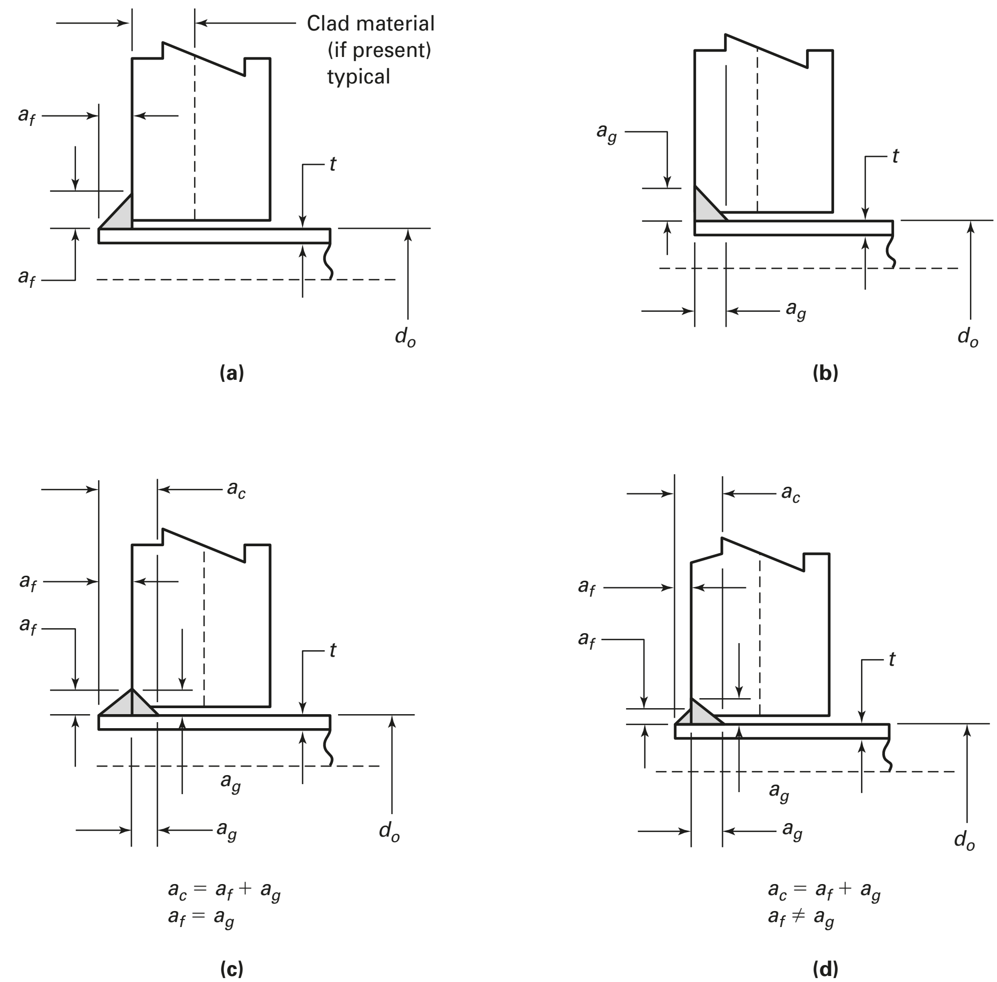
(a) The size of a partial strength weld shall be determined in accordance UW-20.6.

(b) The maximum allowable axial load in either direction on a tube-to-tubesheet joint with a partial strength weld shall be determined as follows:

(1) For loads due to pressure-induced axial forces, $L_{\max} = F_f + F_g$, but not greater than F_t.

(2) For loads due to thermally induced or pressure plus thermally induced axial forces:

Figure UW-20.1
Some Acceptable Types of Tube-to-Tubesheet Strength Welds



(-a) $L_{\max} = F_f + F_g$, but not greater than F_t, for welded only tube-to-tubesheet joints, where the thickness through the weld throat is less than the nominal tube thickness t;

(-b) $L_{\max} = 2(F_f + F_g)$, but not greater than $2F_t$, for all other welded tube-to-tubesheet joints.

UW-20.6 Weld Size Design Formulas. The size of tube-to-tubesheet strength welds shown in Figure UW-20.1 shall conform to the following requirements:

(a) For fillet welds shown in sketch (a),

$$a_r = \sqrt{(0.75d_o)^2 + 2.73t(d_o - t)f_w f_d} - 0.75d_o$$

(1) For full strength welds, a_f shall not be less than the greater of a_r or t.

(2) For partial strength welds, a_f shall not be less than a_r.

(b) For groove welds shown in sketch (b),

$$a_r = \sqrt{(0.75d_o)^2 + 1.76t(d_o - t)f_w f_d} - 0.75d_o$$

(1) For full strength welds, a_g shall not be less than the greater of a_r or t.

(2) For partial strength welds, a_g shall not be less than a_r.

(c) For combined groove and fillet welds shown in sketch (c), where a_f is equal to a_g,

$$a_r = 2\left[\sqrt{(0.75d_o)^2 + 1.07t(d_o - t)f_w f_d} - 0.75d_o\right]$$

(1) For full strength welds, a_c shall not be less than the greater of a_r or t.

(2) For partial strength welds, a_c shall not be less than a_r.

Calculate a_f and a_g: $a_f = a_c/2$ and $a_g = a_c/2$.

(d) For combined groove and fillet welds shown in sketch (d), where a_f is not equal to a_g, a_r shall be determined as follows: Choose a_g. Calculate a_r:

$$a_r = \sqrt{(0.75d_o)^2 + 2.73t(d_o - t)f_w f_d f_f} - 0.75d_o$$

(1) For full strength welds, a_c shall not be less than the greater of $(a_r + a_g)$ or t.

(2) For partial strength welds, a_c shall not be less than $(a_r + a_g)$.

Calculate a_f : $a_f = a_c - a_g$.

(19) **UW-20.7 Clad Tubesheets.**

(a) Tube-to-tubesheet welds in the cladding of either integral or weld metal overlay clad tubesheets may be considered strength welds (full or partial), provided the welds meet the design requirements of UW-20. In addition, when the strength welds are to be made in the clad material of integral clad tubesheets, the integral clad material to be used for tubesheets shall meet the requirements in (1) and (2) for any combination of clad and base materials. The shear strength test and ultrasonic examination specified in (1) and (2) are not required for weld metal overlay clad tubesheets.

(1) Integral clad material shall be shear strength tested in accordance with SA-263. One shear test shall be made on each integral clad plate or forging and the results shall be reported on the material test report.

(2) Integral clad material shall be ultrasonically examined for bond integrity in accordance with SA-578, including Supplementary Requirement S1, and shall meet the acceptance criteria given in SA-263 for Quality Level Class 1.

(b) When the design calculations for clad tubesheets are based on the total thickness including the cladding, the clad material shall meet any additional requirements specified in Part UCL.

(c) When tubesheets are constructed using linings, or integral cladding that does not meet the requirements of (a)(1) and (a)(2), the strength of the tube-to-tubesheet joint shall not be dependent upon the connection between the tubes and the lining or integral cladding, as applicable.

(d) When the tubes are strength welded (full or partial) to integral or weld metal overlay clad tubesheets, S_t shall be the allowable stress value of the integral cladding or the wrought material whose chemistry most closely approximates that of the weld metal overlay cladding. The thickness of the integral or weld metal clad overlay material shall be sufficient to prevent any of the strength weld from extending into the base material.

UW-21 ASME B16.5 SOCKET AND SLIP-ON FLANGE WELDS

(a) ASME B16.5 socket weld flanges shall be welded using an external fillet weld. See Figure UW-21, sketch (4).

(b) ASME B16.5 slip-on flanges shall be welded using an internal and an external weld. See Figure UW-21, sketches (1), (2), and (3).

(c) Nomenclature

t_n = nominal thickness of the shell or nozzle

X_{min} = the lesser of $1.4t_n$ or the thickness of the hub

(d) When ASME B16.5 slip-on flanges are shown to comply with all the requirements provided in Mandatory Appendix 2 of this Division, the weld sizes in Mandatory Appendix 2 may be used as an alternative to the requirements in (b).

FABRICATION

UW-26 GENERAL

(a) The rules in the following paragraphs apply specifically to the fabrication of pressure vessels and vessel parts that are fabricated by welding and shall be used in conjunction with the general requirements for *Fabrication* in Subsection A, and with the specific requirements for *Fabrication* in Subsection C that pertain to the class of material used.

(b) Each Manufacturer or parts Manufacturer shall be responsible for the quality of the welding done by his organization and shall conduct tests not only of the welding procedure to determine its suitability to ensure welds that will meet the required tests, but also of the welders and welding operators to determine their ability to apply the procedure properly.

(c) No production welding shall be undertaken until after the welding procedures which are to be used have been qualified. Only welders and welding operators who are qualified in accordance with Section IX shall be used in production.

(d) The Manufacturer (Certificate Holder) may engage individuals by contract or agreement for their services as welders[68] at the shop location shown on the Certificate of Authorization and at field sites (if allowed by the Certificate of Authorization) for the construction of pressure vessels or vessel parts, provided all of the following conditions are met:

(1) All Code construction shall be the responsibility of the Manufacturer.

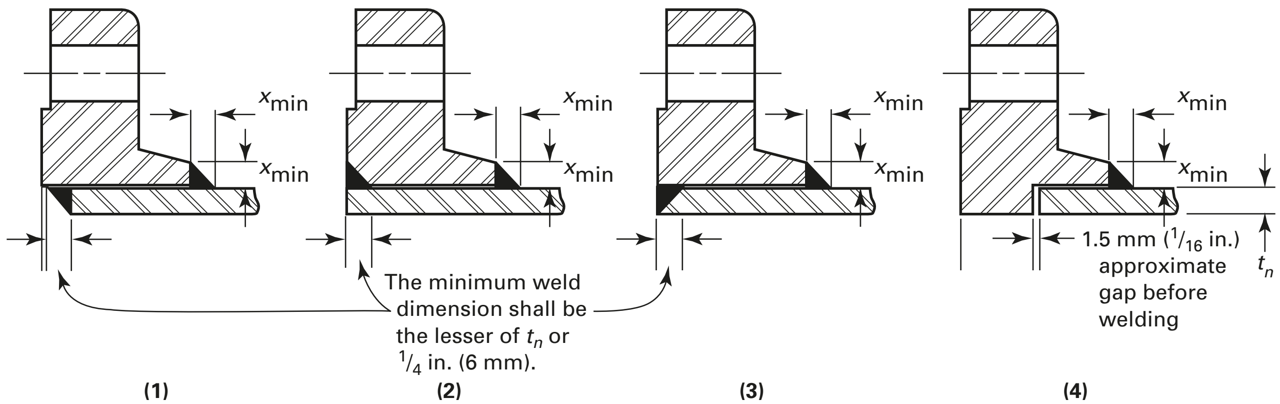
(2) All welding shall be performed in accordance with the Manufacturer's welding procedure specifications in accordance with the requirements of Section IX.

(3) All welders shall be qualified by the Manufacturer in accordance with the requirements of Section IX.

(4) The Manufacturer's Quality Control System shall include as a minimum:

(-a) a requirement for complete and exclusive administrative and technical supervision of all welders by the Manufacturer;

Figure UW-21
Welds of Socket Weld Flanges to Nozzle Necks



(-b) evidence of the Manufacturer's authority to assign and remove welders at his discretion without involvement of any other organization;

(-c) a requirement for assignment of welder identification symbols;

(-d) evidence that this program has been accepted by the Manufacturer's Authorized Inspection Agency which provides the inspection service.

(5) The Manufacturer shall be responsible for Code compliance of the vessel or part, including Certification Mark stamping and providing Data Report Forms properly executed and countersigned by the Inspector.

(19) UW-27 WELDING PROCESSES

The welding processes that may be used in the construction of vessels under this Part of this Division are limited to those listed in Section IX, Article II with the following additional restrictions:

(a) Other than pressure inherent to the welding processes, no mechanical pressure or blows shall be applied except as permitted for peening in UW-39.

(b) Arc stud welding and resistance stud welding may be used only for non-pressure-bearing attachments having a load- or non-load-carrying function, except for material listed in Table UHT-23, provided that, in the case of ferrous materials, the heat treatment requirements of UCS-56 are complied with and the requirements of UW-28(b) and UW-29(a) are met prior to start of production welding. Studs shall be limited to 1 in. (25 mm) diameter maximum for round studs and an equivalent cross-sectional area for studs with other shapes.

(c) The electroslag welding process may be used for butt welds only in ferritic steels and austenitic stainless steels of types listed in UW-5(d), provided the requirements of UW-11(a)(6) and UW-11(d) are satisfied. [See UW-5(e).]

(d) The electrogas welding process may be used for butt welds only in ferritic steels and austenitic stainless steels of types listed in UW-5(d), provided the requirements of UW-11(a)(6) are satisfied. When a single pass is greater than $1^1/_2$ in. (38 mm) in ferritic materials, the joint shall be given a grain refining (austenitizing) heat treatment. [See UW-5(e).]

(e) Welding processes accepted under Section IX Code Cases shall not be used unless those Code Cases are explicitly accepted by this Division.

UW-28 QUALIFICATION OF WELDING PROCEDURE

(a) Each procedure of welding that is to be followed in construction shall be recorded in detail by the manufacturer.

(b) The procedure used in welding pressure parts and in joining load-carrying nonpressure parts, such as all permanent or temporary clips and lugs, to pressure parts shall be qualified in accordance with Section IX.

(c) The procedure used in welding non-pressure-bearing attachments which have essentially no load-carrying function (such as extended heat transfer surfaces, insulation support pins, etc.), to pressure parts shall meet the following requirements.

(1) When the welding process is manual, machine, or semiautomatic, procedure qualification is required in accordance with Section IX.

(2) When the welding is any automatic welding process performed in accordance with a Welding Procedure Specification (in compliance with Section IX as far as applicable), procedure qualification testing is not required.

(d) Welding of all test coupons shall be conducted by the Manufacturer. Testing of all test coupons shall be the responsibility of the Manufacturer. Alternatively, AWS Standard Welding Procedure Specifications that have been accepted by Section IX may be used provided they meet all other requirements of this Division.

Qualification of a welding procedure by one Manufacturer shall not qualify that procedure for any other Manufacturer except as provided in Section IX, QG-106.

UW-29 TESTS OF WELDERS AND WELDING OPERATORS

(a) The welders and welding operators used in welding pressure parts and in joining load-carrying nonpressure parts (attachments) to pressure parts shall be qualified in accordance with Section IX.

(1) The qualification test for welding operators of machine welding equipment shall be performed on a separate test plate prior to the start of welding or on the first workpiece.

(2) When stud welding is used to attach load-carrying studs, a production stud weld test of each welder or welding operator shall be performed on a separate test plate or tube prior to the start of welding on each work shift. This weld test shall consist of five studs, welded and tested by the bend or torque stud weld testing procedure described in Section IX.

(b) The welders and welding operators used in welding non-pressure-bearing attachments, which have essentially no load-carrying function (such as extended heat transfer surfaces, insulation support pins, etc.), to pressure parts shall comply with the following:

(1) When the welding process is manual, machine, or semiautomatic, qualification in accordance with Section IX is required.

(2) When welding is done by any automatic welding process, performance qualification testing is not required.

(3) When stud welding is used, a production stud weld test, appropriate to the end use application requirements, shall be specified by the Manufacturer and carried out on a separate test plate or tube at the start of each shift.

(c) Each welder and welding operator shall be assigned an identifying number, letter, or symbol by the manufacturer which shall be used to identify the work of that welder or welding operator in accordance with UW-37(f).

(d) The Manufacturer shall maintain a record of the welders and welding operators showing the date and result of tests and the identification mark assigned to each. These records shall be maintained in accordance with Section IX.

(e) Welding of all test coupons shall be conducted by the Manufacturer. Testing of all test coupons shall be the responsibility of the Manufacturer. A performance qualification test conducted by one Manufacturer shall not qualify a welder or welding operator to do work for any other Manufacturer except as provided in Section IX, QG-106.

UW-30 LOWEST PERMISSIBLE TEMPERATURES FOR WELDING

It is recommended that no welding of any kind be done when the temperature of the base metal is lower than 0°F (−20°C). At temperatures between 32°F (0°C) and 0°F (−20°C), the surface of all areas within 3 in. (75 mm) of the point where a weld is to be started should be heated to a temperature at least warm to the hand [estimated to be above 60°F (15°C)] before welding is started. It is recommended also that no welding be done when surfaces are wet or covered with ice, when snow is falling on the surfaces to be welded, or during periods of high wind, unless the welders or welding operators and the work are properly protected.

UW-31 CUTTING, FITTING, AND ALIGNMENT

(a) When plates are shaped by oxygen or arc cutting, the edges to be welded shall be uniform and smooth and shall be freed of all loose scale and slag accumulations before welding (see UG-76 and UCS-5).

(b) Plates that are being welded shall be fitted, aligned, and retained in position during the welding operation.

(c) Bars, jacks, clamps, tack welds, or other appropriate means may be used to hold the edges of parts in alignment. Tack welds used to secure alignment shall either be removed completely when they have served their purpose, or their stopping and starting ends shall be properly prepared by grinding or other suitable means so that they may be satisfactorily incorporated into the final weld. Tack welds, whether removed or left in place, shall be made using a fillet weld or butt weld procedure qualified in accordance with Section IX. Tack welds to be left in place shall be made by welders qualified in accordance with Section IX, and shall be examined visually for defects, and if found to be defective shall be removed.

Provided that the work is done under the provisions of U-2(b), it is not necessary that a subcontractor making such tack welds for a vessel or parts manufacturer be a holder of a Code Certificate of Authorization. The requirements of UW-26(d) do not apply to such tack welds.

(d) The edges of butt joints shall be held during welding so that the tolerances of UW-33 are not exceeded in the completed joint. When fitted girth joints have deviations exceeding the permitted tolerances, the head or shell ring, whichever is out-of-true, shall be reformed until the errors are within the limits specified. Where fillet welds are used, the lapped plates shall fit closely and be kept in contact during welding.

(e) When joining two parts by the inertia and continuous drive friction welding processes, one of the two parts must be held in a fixed position and the other part rotated. The two faces to be joined must be essentially symmetrical with respect to the axis of rotation. Some of the basic types of applicable joints are solid round to solid round, tube to tube, solid round to tube, solid round to plate, and tube to plate.

UW-32 CLEANING OF SURFACES TO BE WELDED

(a) The surfaces to be welded shall be clean and free of scale, rust, oil, grease, slag, detrimental oxides, and other deleterious foreign material. The method and extent of cleaning should be determined based on the material to be welded and the contaminants to be removed. When weld metal is to be deposited over a previously welded surface, all slag shall be removed by a roughing tool, chisel, chipping hammer, or other suitable means so as to prevent inclusion of impurities in the weld metal.

(b) Cast surfaces to be welded shall be machined, chipped, or ground to remove foundry scale and to expose sound metal.

(c) The requirements in (a) and (b) above are not intended to apply to any process of welding by which proper fusion and penetration are otherwise obtained and by which the weld remains free from defects.

UW-33 ALIGNMENT TOLERANCE

(a) Alignment of sections at edges to be butt welded shall be such that the maximum offset is not greater than the applicable amount for the welded joint category (see UW-3) under consideration, as listed in Table UW-33. The section thickness t is the nominal thickness of the thinner section at the joint.

(b) Any offset within the allowable tolerance provided above shall be faired at a three to one taper over the width of the finished weld, or if necessary, by adding additional weld metal beyond what would otherwise be the edge of the weld. Such additional weld metal buildup shall be subject to the requirements of UW-42.

UW-34 SPIN-HOLES

Spin-holes are permitted within heads or segments thereof to facilitate forming. Spin-holes not exceeding the size limitations of UG-36(c)(3)(-a) may be closed with a full-penetration weld using either a welded plug or weld metal. The weld and plug shall be no thinner than the head material adjacent to the spin-hole.

The finished weld shall be examined[69] and shall meet the acceptance requirements of Mandatory Appendix 6 or Mandatory Appendix 8 of this Division. Radiographic examination, if required by UW-11(a), and additional inspections, if required by the material specification, shall be performed.

Table UW-33

Customary Units

Section Thickness, in.	Joint Category A	Joint Category B, C, and D
Up to $^1/_2$, incl.	$^1/_4t$	$^1/_4t$
Over $^1/_2$ to $^3/_4$, incl.	$^1/_8$ in.	$^1/_4t$
Over $^3/_4$ to $1^1/_2$, incl.	$^1/_8$ in.	$^3/_{16}$ in.
Over $1^1/_2$ to 2, incl.	$^1/_8$ in.	$^1/_8t$
Over 2	Lesser of $^1/_{16}t$ or $^3/_8$ in.	Lesser of $^1/_8t$ or $^3/_4$ in.

SI Units

Section Thickness, mm	Joint Category A	Joint Category B, C, and D
Up to 13, incl.	$^1/_4t$	$^1/_4t$
Over 13 to 19, incl.	3 mm	$^1/_4t$
Over 19 to 38, incl.	3 mm	5 mm
Over 38 to 51, incl.	3 mm	$^1/_8t$
Over 51	Lesser of $^1/_{16}t$ or 10 mm	Lesser of $^1/_8t$ or 19 mm

This weld is a butt weld, but it is not categorized. It shall not be considered in establishing the joint efficiency of any part of the head or of the head-to-shell weld.

UW-35 FINISHED LONGITUDINAL AND CIRCUMFERENTIAL JOINTS

(a) Butt-welded joints shall have complete penetration and full fusion. As-welded surfaces are permitted; however, the surface of welds shall be sufficiently free from coarse ripples, grooves, overlaps, and abrupt ridges and valleys to permit proper interpretation of radiographic and other required nondestructive examinations. If there is a question regarding the surface condition of the weld when interpreting a radiographic film, the film shall be compared to the actual weld surface for determination of acceptability.

(b) A reduction in thickness due to the welding process is acceptable provided all of the following conditions are met:

(1) The reduction in thickness shall not reduce the material of the adjoining surfaces below the design thickness at any point.

(2) The reduction in thickness shall not exceed $^1/_{32}$ in. (1 mm) or 10% of the nominal thickness of the adjoining surface, whichever is less.[70]

(c) When a single-welded butt joint is made by using a backing strip which is left in place [Type No. (2) of Table UW-12], the requirement for reinforcement applies only to the side opposite the backing strip.

(d) To assure that the weld grooves are completely filled so that the surface of the weld metal at any point does not fall below the surface of the adjoining base

materials,[71] weld metal may be added as reinforcement on each face of the weld. The thickness of the weld reinforcement on each face shall not exceed the following:

Customary Units		
	Maximum Reinforcement, in.	
Material Nominal Thickness, in.	Category B and C Butt Welds	Other Welds
Less than $^3/_{32}$	$^3/_{32}$	$^1/_{32}$
$^3/_{32}$ to $^3/_{16}$, incl.	$^1/_8$	$^1/_{16}$
Over $^3/_{16}$ to $^1/_2$, incl.	$^5/_{32}$	$^3/_{32}$
Over $^1/_2$ to 1, incl.	$^3/_{16}$	$^3/_{32}$
Over 1 to 2, incl.	$^1/_4$	$^1/_8$
Over 2 to 3, incl.	$^1/_4$	$^5/_{32}$
Over 3 to 4, incl.	$^1/_4$	$^7/_{32}$
Over 4 to 5, incl.	$^1/_4$	$^1/_4$
Over 5	$^5/_{16}$	$^5/_{16}$

SI Units		
	Maximum Reinforcement, mm	
Material Nominal Thickness, mm	Category B and C Butt Welds	Other Welds
Less than 2.4	2.5	0.8
2.4 to 4.8, incl.	3	1.5
Over 4.8 to 13, incl.	4	2.5
Over 13 to 25, incl.	5	2.5
Over 25 to 51, incl.	6	3
Over 51 to 76, incl.	6	4
Over 76 to 102, incl.	6	5.5
Over 102 to 127, incl.	6	6
Over 127	8	8

UW-36 FILLET WELDS

In making fillet welds, the weld metal shall be deposited in such a way that adequate penetration into the base metal at the root of the weld is secured. The reduction of the thickness of the base metal due to the welding process at the edges of the fillet weld shall meet the same requirements as for butt welds [see UW-35(b)].

UW-37 MISCELLANEOUS WELDING REQUIREMENTS

(a) The reverse side of double-welded joints shall be prepared by chipping, grinding, or melting out, so as to secure sound metal at the base of weld metal first deposited, before applying weld metal from the reverse side.

(b) The requirements in (a) above are not intended to apply to any process of welding by which proper fusion and penetration are otherwise obtained and by which the base of the weld remains free from defects.

(c) If the welding is stopped for any reason, extra care shall be taken in restarting to get the required penetration and fusion. For submerged arc welding, chipping out a groove in the crater is recommended.

(d) Where single-welded joints are used, particular care shall be taken in aligning and separating the components to be joined so that there will be complete penetration and fusion at the bottom of the joint for its full length.

(e) In welding plug welds, a fillet around the bottom of the hole shall be deposited first.

(f) *Welder and Welding Operator Identification*

(1) Each welder and welding operator shall stamp the identifying number, letter, or symbol assigned by the Manufacturer, on or adjacent to and at intervals of not more than 3 ft (1 m) along the welds which he makes in steel plates $^1/_4$ in. (6 mm) and over in thickness and in nonferrous plates $^1/_2$ in. (13 mm) and over in thickness; or a record shall be kept by the Manufacturer of welders and welding operators employed on each joint which shall be available to the Inspector. For identifying welds on vessels in which the wall thickness is less than $^1/_4$ in. (6 mm) for steel material and less than $^1/_2$ in. (13 mm) for nonferrous material, suitable stencil or other surface markings shall be used; or a record shall be kept by the Manufacturer of welders and welding operators employed on each joint which shall be available to the Inspector; or a stamp may be used provided the vessel part is not deformed and the following additional requirements are met:

(-a) for ferrous materials:

(-1) the materials shall be limited to P-No. 1 Gr. Nos. 1 and 2;

(-2) the minimum nominal plate thickness shall be $^3/_{16}$ in. (5 mm), or the minimum nominal pipe wall thickness shall be 0.154 in. (3.91 mm);

(-3) the minimum design metal temperature shall be no colder than −20°F (−29°C);

(-b) for nonferrous materials:

(-1) the materials shall be limited to aluminum as follows: SB-209 Alloys 3003, 5083, 5454, and 6061; SB-241 Alloys 3003, 5083, 5086, 5454, 6061, and 6063; and SB-247 Alloys 3003, 5083, and 6061;

(-2) the minimum nominal plate thickness shall be 0.249 in. (6.32 mm), or the minimum nominal pipe thickness shall be 0.133 in. (3.37 mm).

(2) When a multiple number of permanent nonpressure part load bearing attachment welds, nonload-bearing welds such as stud welds, or special welds such as tube-to-tubesheet welds are made on a vessel, the Manufacturer need not identify the welder or welding operator that welded each individual joint provided:

(-a) the Manufacturer's Quality Control System includes a procedure that will identify the welders or welding operators that made such welds on each vessel so that the Inspector can verify that the welders or welding operators were all properly qualified;

(-b) the welds in each category are all of the same type and configuration and are welded with the same welding procedure specification.

(3) Permanent identification of welders or welding operators making tack welds that become part of the final pressure weld is not required provided the Manufacturer's Quality Control System includes a procedure to permit the Inspector to verify that such tack welds were made by qualified welders or welding operators.

(g) The welded joint between two members joined by the inertia and continuous drive friction welding processes shall be a full penetration weld. Visual examination of the as-welded flash roll of each weld shall be made as an in-process check. The weld upset shall meet the specified amount within ±10%. The flash shall be removed to sound metal.

(h) Capacitor discharge welding may be used for welding temporary attachments and permanent nonstructural attachments without postweld heat treatment, provided the following requirements are met:

(1) A welding procedure specification shall be prepared in accordance with Section IX, insofar as possible describing the capacitor discharge equipment, the combination of materials to be joined, and the technique of application. Qualification of the welding procedure is not required.

(2) The energy output shall be limited to 125 W-sec.

UW-38 REPAIR OF WELD DEFECTS

Defects, such as cracks, pinholes, and incomplete fusion, detected visually or by the hydrostatic or pneumatic test or by the examinations prescribed in UW-11 shall be removed by mechanical means or by thermal gouging processes, after which the joint shall be rewelded [see UW-40(e)].

UW-39 PEENING

(a) Weld metal and heat-affected zones may be peened by manual, electric, or pneumatic means when it is deemed necessary or helpful to control distortion, to relieve residual stresses, or to improve the quality of the weld. Peening shall not be used on the initial (root) layer of weld metal nor on the final (face) layer unless the weld is subsequently postweld heat treated. In no case, however, is peening to be performed in lieu of any postweld heat treatment required by these rules.

(b) Controlled shot peening and other similar methods which are intended only to enhance surface properties of the vessel or vessel parts shall be performed after any nondestructive examinations and pressure tests required by these rules.

UW-40 PROCEDURES FOR POSTWELD HEAT TREATMENT

(a) The operation of postweld heat treatment shall be performed in accordance with the requirements given in the applicable Part in Subsection C using one of the following procedures. In the procedures that follow, the soak band is defined as the volume of metal required to meet or exceed the minimum PWHT temperatures listed in Tables UCS-56-1 through UCS-56-11. As a minimum, the soak band shall contain the weld, heat-affected zone, and a portion of base metal adjacent to the weld being heat treated. The minimum width of this volume is the widest width of weld plus 1t or 2 in. (50 mm), whichever is less, on each side or end of the weld. The term t is the nominal thickness as defined in (f) below. For additional detailed recommendations regarding implementation and performance of these procedures, refer to Welding Research Council (WRC) Bulletin 452, June 2000, "Recommended Practices for Local Heating of Welds in Pressure Vessels."

(1) heating the vessel as a whole in an enclosed furnace. This procedure is preferable and should be used whenever practicable.

(2) heating the vessel in more than one heat in a furnace, provided the overlap of the heated sections of the vessel is at least 5 ft (1.5 m). When this procedure is used, the portion outside of the furnace shall be shielded so that the temperature gradient is not harmful. The cross section where the vessel projects from the furnace shall not intersect a nozzle or other structural discontinuity.

(3) heating of shell sections and/or portions of vessels to postweld heat treat longitudinal joints or complicated welded details before joining to make the completed vessel. When the vessel is required to be postweld heat treated, and it is not practicable to postweld heat treat the completed vessel as a whole or in two or more heats as provided in (2) above, any circumferential joints not previously postweld heat treated may be thereafter locally postweld heat treated by heating such joints by any appropriate means that will assure the required uniformity. For such local heating, the soak band shall extend around the full circumference. The portion outside the soak band shall be protected so that the temperature gradient is not harmful. This procedure may also be used to postweld heat treat portions of new vessels after repairs.

(4) heating the vessel internally by any appropriate means and with adequate indicating and recording temperature devices to aid in the control and maintenance of a uniform distribution of temperature in the vessel wall. Previous to this operation, the vessel should be fully enclosed with insulating material, or the permanent insulation may be installed provided it is suitable for the required temperature. In this procedure the internal pressure should be kept as low as practicable, but shall not exceed 50% of the maximum allowable working pressure at the highest metal temperature expected during the postweld heat treatment period.

(5) heating a circumferential band containing nozzles or other welded attachments that require postweld heat treatment in such a manner that the entire band shall be brought up uniformly to the required temperature and held for the specified time. Except as modified in this paragraph below, the soak band shall extend around the entire vessel, and shall include the nozzle or welded attachment. The circumferential soak band width may be varied away from the nozzle or attachment weld requiring PWHT, provided the required soak band around the nozzle or attachment weld is heated to the required temperature and held for the required time. As an alternative

to varying the soak band width, the temperature within the circumferential band away from the nozzle or attachment may be varied and need not reach the required temperature, provided the required soak band around the nozzle or attachment weld is heated to the required temperature, held for the required time, and the temperature gradient is not harmful throughout the heating and cooling cycle. The portion of the vessel outside of the circumferential soak band shall be protected so that the temperature gradient is not harmful. This procedure may also be used to postweld heat treat portions of vessels after repairs.

(6) heating the circumferential joints of pipe or tubing by any appropriate means using a soak band that extends around the entire circumference. The portion outside the soak band shall be protected so that the temperature gradient is not harmful. The proximity to the shell increases thermal restraint, and the designer should provide adequate length to permit heat treatment without harmful gradients at the nozzle attachment or heat a full circumferential band around the shell, including the nozzle.

(7) heating a local area around nozzles or welded attachments in the larger radius sections of a double curvature head or a spherical shell or head in such a manner that the area is brought up uniformly to the required temperature and held for the specified time. The soak band shall include the nozzle or welded attachment. The soak band shall include a circle that extends beyond the edges of the attachment weld in all directions by a minimum of t or 2 in. (50 mm), whichever is less. The portion of the vessel outside of the soak band shall be protected so that the temperature gradient is not harmful.

(8) heating of other configurations. Local area heating of other configurations such as "spots" or "bulls eye" local heating not addressed in (1) through (7) above is permitted, provided that other measures (based upon sufficiently similar, documented experience or evaluation) are taken that consider the effect of thermal gradients, all significant structural discontinuities (such as nozzles, attachments, head to shell junctures), and any mechanical loads which may be present during PWHT. The portion of the vessel or component outside the soak band shall be protected so that the temperature gradient is not harmful.

(b) The temperatures and rates of heating and cooling to be used in postweld heat treatment of vessels constructed of materials for which postweld heat treatment may be required are given in UCS-56, UHT-56, UNF-56, and UHA-32.

(c) The minimum temperature for postweld heat treatment given in Tables UCS-56-1 through UCS-56-11, Table UHT-56, and Tables UHA-32-1 through UHA-32-7, and in UNF-56, shall be the minimum temperature of the plate material of the shell or head of any vessel. Where more than one pressure vessel or pressure vessel part are postweld heat treated in one furnace charge, thermocouples shall be placed on vessels at the bottom, center, and top of the charge, or in other zones of possible temperature variation so that the temperature indicated shall be true temperature for all vessels or parts in those zones.[72]

(d) It is recognized that some postweld heat treatments may have detrimental effects on the properties of some materials. When pressure parts of two different P-Numbers are joined by welding, engineering judgment shall be applied when selecting the postweld heat treatment temperature and holding time to produce material properties suitable for the intended service. Alternatives such as welding with buttering as described in Section IX, QW-283 may be considered.

(e) Postweld heat treatment, when required, shall be done before the hydrostatic test and after any welded repairs except as permitted by UCS-56(f). A preliminary hydrostatic test to reveal leaks prior to postweld heat treatment is permissible.

(f) The term nominal thickness as used in Tables UCS-56-1 through UCS-56-11, UCS-56.1, UHA-32-1 through UHA-32-7, and UHT-56, is the thickness of the welded joint as defined below. For pressure vessels or parts of pressure vessels being postweld heat treated in a furnace charge, it is the greatest weld thickness for all weld types as defined in (1) through (6) below in any vessel or vessel part that has not previously been postweld heat treated.

(1) When the welded joint connects parts of the same thickness, using a full penetration buttweld, the nominal thickness is the total depth of the weld exclusive of any permitted weld reinforcement.

(2) For groove welds, the nominal thickness is the depth of the groove. For single- or double-sided groove welds, the nominal thickness is the total depth of the groove.

(3) For fillet welds, the nominal thickness is the throat dimension. If a fillet weld is used in conjunction with a groove weld, the nominal thickness is the depth of the groove or the throat dimension, whichever is greater.

(4) For stud welds, the nominal thickness shall be the diameter of the stud.

(5) When a welded joint connects parts of unequal thicknesses, the nominal thickness shall be the following:

(-a) the thinner of two adjacent butt-welded parts including head to shell connections

(-b) the thickness of the shell or the fillet weld, whichever is greater, in connections to intermediate heads of the type shown in Figure UW-13.1, sketch (e);

(-c) the thickness of the shell in connections to tubesheets, flat heads, covers, flanges (except for welded parts depicted in Figure 2-4, sketch (7), where the thickness of the weld shall govern), or similar constructions;

(-d) in Figures UW-16.1 and UW-16.2, the thickness of the weld across the nozzle neck or shell or head or reinforcing pad or attachment fillet weld, whichever is the greater;

(-e) the thickness of the nozzle neck at the joint in nozzle neck to flange connections;

(-f) the thickness of the weld at the point of attachment when a nonpressure part is welded to a pressure part;

(-g) the thickness of the tube in tube-to-tubesheet connections.

(-h) the thickness of the weld metal overlay when weld metal overlay is the only welding applied

(6) For repairs, the nominal thickness is the depth of the repair weld.

(7) The thickness of the head, shell, nozzle neck, or other parts as used in (1) through (6) above shall be the wall thickness of the part at the welded joint under consideration. For plate material, the thickness as shown on the Material Test Report or material Certificate of Compliance before forming may be used, at the Manufacturer's option, in lieu of measuring the wall thickness at the welded joint.

UW-41 SECTIONING OF WELDED JOINTS

Welded joints may be examined by sectioning when agreed to by user and Manufacturer, but this examination shall not be considered a substitute for spot radiographic examination. This type of examination has no effect on the joint factors in Table UW-12. The method of closing the hole by welding is subject to acceptance by the Inspector. Some acceptable methods are given in Nonmandatory Appendix K.

UW-42 SURFACE WELD METAL BUILDUP

(a) Construction in which deposits of weld metal are applied to the surface of base metal for the purpose of restoring the thickness of the base metal for strength consideration; or modifying the configuration of weld joints in order to provide the tapered transition requirements of UW-9(c) and UW-33(b) shall be performed in accordance with the rules in (b) and (c).

(b) Procedure Qualification. A butt welding procedure qualification in accordance with provisions of Section IX shall be performed for the thickness of weld metal deposited, prior to production welding.

(c) Examination Requirements

(1) All weld metal buildup shall be examined over the full surface of the deposit by either magnetic particle examination to the requirements of Mandatory Appendix 6, or by liquid penetrant examination to the requirements of Mandatory Appendix 8.

(2) When such surface weld metal buildup is used in welded joints which require full or spot radiographic examination, the weld metal buildup shall be included in the examination.

INSPECTION AND TESTS

UW-46 GENERAL

The rules in the following paragraphs apply specifically to the inspection and testing of pressure vessels and vessel parts that are fabricated by welding and shall be used in conjunction with the general requirements for *Inspection and Tests* in Subsection A, and with the specific requirements for *Inspection and Tests* in Subsection C that pertain to the class of material used. [For tests on reinforcing plates, see UG-37(g).]

UW-47 CHECK OF WELDING PROCEDURE

The Inspector shall assure himself that the welding procedure employed in the construction of a vessel has been qualified under the provisions of Section IX. The Manufacturer shall submit evidence to the Inspector that the requirements have been met.

UW-48 CHECK OF WELDER AND WELDING OPERATOR QUALIFICATIONS

(a) The Manufacturer shall certify that the welding on a vessel has been done only by welders and welding operators who have been qualified under the requirements of Section IX and the Inspector shall assure himself that only qualified welders and welding operators have been used.

(b) The Manufacturer shall make available to the Inspector the record of the qualification tests of each welder and welding operator. The Inspector shall have the right at any time to call for and witness tests of the welding procedure or of the ability of any welder and welding operator.

UW-49 CHECK OF POSTWELD HEAT TREATMENT PRACTICE

The Inspector shall satisfy himself that all postweld heat treatment has been correctly performed and that the temperature readings conform to the requirements.

UW-50 NONDESTRUCTIVE EXAMINATION OF WELDS ON PNEUMATICALLY TESTED VESSELS (19)

(a) On welded pressure vessels to be pneumatically tested in accordance with UG-100, the full length of the following welds shall be examined[69] before the pneumatic test is performed, for the purpose of detecting cracks:

(1) all welds around openings

(2) all attachment welds having a throat thickness greater than $^1/_4$ in. (6 mm), including welds attaching nonpressure parts to pressure parts

(b) The weld joint examination requirements given in (a) may be waived when the maximum allowable working pressure of the vessel is no greater than 500 psi (3.5 MPa) and the following applicable requirement is met:

(1) For Part UCS materials, the governing thickness as defined in UCS-66(a) shall be limited to a maximum governing thickness of $^1/_2$ in. (13 mm) for materials assigned to Curve A, and 1 in. (25 mm) for materials assigned to Curve B, C, or D in Figure UCS-66.

(2) For austenitic chromium–nickel stainless steels 304, 304L, 316, 316L, 321, and 347 in Part UHA, the maximum nominal material thickness shall be $^3/_4$ in. (19 mm).

(3) For aluminum, aluminum alloy 3000 series, aluminum alloy 5000 series, and aluminum alloy 6061-T6 in Part UNF, the maximum nominal material thickness shall be 1 in. (25 mm).

(19) UW-51 RADIOGRAPHIC EXAMINATION OF WELDED JOINTS

(a) All welded joints to be radiographed shall be examined in accordance with Section V, Article 2, except as specified below.

(1) A complete set of radiographic images and examination records, as described in Section V, Article 2, for each vessel or vessel part shall be retained by the Manufacturer, as follows:

(-a) radiographic images until the Manufacturer's Data Report has been signed by the Inspector

(-b) examination records as required by this Division (10-13)

(2) Demonstration of acceptable density on radiographic films and the ability to see the prescribed image quality indicator (IQI) image and the specified hole or the designated wire of a wire IQI shall be considered satisfactory evidence of compliance with Section V, Article 2.

(3) The requirements of Section V, Article 2, T-274.2, are to be used only as a guide for film-based radiography.

(4) As an alternative to the radiographic examination requirements above, all welds in which the thinner of the members joined is $^1/_4$ in. (6 mm) thick and greater may be examined using the ultrasonic (UT) method specified by UW-53(b).

(b) Indications revealed by radiography within a weld that exceed the following criteria are unacceptable and therefore are defects. Defects shall be repaired as provided in UW-38, and the repaired area shall be reexamined. In lieu of reexamination by radiography, the repaired weld may be ultrasonically examined in accordance with Mandatory Appendix 12 at the Manufacturer's option. For material thicknesses in excess of 1 in. (25 mm), the concurrence of the user shall be obtained. This ultrasonic examination shall be noted under Remarks on the Manufacturer's Data Report Form:

(1) any indication characterized as a crack or zone of incomplete fusion or penetration;

(2) any other elongated indication on the radiograph which has length greater than:

(-a) $^1/_4$ in. (6 mm) for t up to $^3/_4$ in. (19 mm)

(-b) $^1/_3t$ for t from $^3/_4$ in. (19 mm) to $2^1/_4$ in. (57 mm)

(-c) $^3/_4$ in. (19 mm) for t over $2^1/_4$ in. (57 mm)

where

t = the thickness of the weld excluding any allowable reinforcement. For a butt weld joining two members having different thicknesses at the weld, t is the thinner of these two thicknesses. If a full penetration weld includes a fillet weld, the thickness of the throat of the fillet shall be included in t.

(3) any group of aligned indications that have an aggregate length greater than t in a length of $12t$, except when the distance between the successive imperfections exceeds $6L$ where L is the length of the longest imperfection in the group;

(4) rounded indications in excess of that specified by the acceptance standards given in Mandatory Appendix 4.

UW-52 SPOT EXAMINATION OF WELDED JOINTS

NOTE: Spot radiographing of a welded joint is recognized as an effective inspection tool. The spot radiography rules are also considered to be an aid to quality control. Spot radiographs made directly after a welder or an operator has completed a unit of weld proves that the work is or is not being done in accordance with a satisfactory procedure. If the work is unsatisfactory, corrective steps can then be taken to improve the welding in the subsequent units, which unquestionably will improve the weld quality.

Spot radiography in accordance with these rules will not ensure a fabrication product of predetermined quality level throughout. It must be realized that an accepted vessel under these spot radiography rules may still contain defects which might be disclosed on further examination. If all radiographically disclosed weld defects must be eliminated from a vessel, then 100% radiography must be employed.

(a) Butt-welded joints that are to be spot radiographed shall be examined locally as provided herein.

(b) Minimum Extent of Spot Radiographic Examination

(1) One spot shall be examined on each vessel for each 50 ft (15 m) increment of weld or fraction thereof for which a joint efficiency from column (b) of Table UW-12 is selected. However, for identical vessels or parts, each with less than 50 ft (15 m) of weld for which a joint efficiency from column (b) of Table UW-12 is selected, 50 ft (15 m) increments of weld may be represented by one spot examination.

(2) For each increment of weld to be examined, a sufficient number of spot radiographs shall be taken to examine the welding of each welder or welding operator. Under conditions where two or more welders or welding operators make weld layers in a joint, or on the two sides of a double-welded butt joint, one spot may represent the work of all welders or welding operators.

(3) Each spot examination shall be made as soon as practicable after completion of the increment of weld to be examined. The location of the spot shall be chosen by the Inspector after completion of the increment of welding to be examined, except that when the Inspector has been notified in advance and cannot be present or otherwise make the selection, the Manufacturer may exercise his own judgment in selecting the spots.

(4) Radiographs required at specific locations to satisfy the rules of other paragraphs, such as UW-9(d), UW-11(a)(5)(-b), and UW-14(b), shall not be used to satisfy the requirements for spot radiography.

(c) Standards for Spot Radiographic Examination. Spot examination by radiography shall be made in accordance with the technique prescribed in UW-51(a). The minimum length of spot radiograph shall be 6 in. (150 mm). Spot radiographs may be retained or be discarded by the Manufacturer after acceptance of the vessel by the Inspector. The acceptability of welds examined by spot radiography shall be judged by the following standards:

(1) Welds in which indications are characterized as cracks or zones of incomplete fusion or penetration shall be unacceptable.

(2) Welds having indications characterized as slag inclusions or cavities are unacceptable when the indication length exceeds $^2/_3t$, where t is defined as shown in UW-51(b)(2). For all thicknesses, indications less than $^1/_4$ in. (6 mm) are acceptable, and indications greater than $^3/_4$ in. (19 mm) are unacceptable. Multiple aligned indications meeting these acceptance criteria are acceptable when the sum of their longest dimensions indications does not exceed t within a length of $6t$ (or proportionally for radiographs shorter than $6t$), and when the longest length L for each indication is separated by a distance not less than $3L$ from adjacent indications.

(3) Rounded indications are not a factor in the acceptability of welds not required to be fully radiographed.

(d) Evaluation and Retests

(1) When a spot, radiographed as required in (b)(1) or (b)(2) above, is acceptable in accordance with (c)(1) and (c)(2) above, the entire weld increment represented by this radiograph is acceptable.

(2) When a spot, radiographed as required in (b)(1) or (b)(2) above, has been examined and the radiograph discloses welding which does not comply with the minimum quality requirements of (c)(1) or (c)(2) above, two additional spots shall be radiographically examined in the same weld increment at locations away from the original spot. The locations of these additional spots shall be determined by the Inspector or fabricator as provided for the original spot examination in (b)(3) above.

(-a) If the two additional spots examined show welding which meets the minimum quality requirements of (c)(1) and (c)(2) above, the entire weld increment represented by the three radiographs is acceptable provided the defects disclosed by the first of the three radiographs are removed and the area repaired by welding. The weld repaired area shall be radiographically examined in accordance with the foregoing requirements of UW-52.

(-b) If either of the two additional spots examined shows welding which does not comply with the minimum quality requirements of (c)(1) or (c)(2) above, the entire increment of weld represented shall be rejected. The entire rejected weld shall be removed and the joint shall be rewelded or, at the fabricator's option, the entire increment of weld represented shall be completely radiographed and only defects need be corrected.

(-c) Repair welding shall be performed using a qualified procedure and in a manner acceptable to the Inspector. The rewelded joint, or the weld repaired areas, shall be spot radiographically examined at one location in accordance with the foregoing requirements of UW-52.

UW-53 ULTRASONIC EXAMINATION OF WELDED JOINTS

(a) Ultrasonic examination of welded joints whose joint efficiency is not determined by ultrasonic examinations may be performed and evaluated in accordance with Mandatory Appendix 12.

(b) Ultrasonic examination of welds per UW-51(a)(4) shall be performed and evaluated in accordance with the requirements of Section VIII, Division 2, 7.5.5.

UW-54 QUALIFICATION OF NONDESTRUCTIVE EXAMINATION PERSONNEL (19)

Personnel performing nondestructive examinations in accordance with UW-51, UW-52, or UW-53 shall be qualified and certified in accordance with the requirements of Section V, Article 1, T-120(e), T-120(f), T-120(g), T-120(i), T-120(j), or T-120(k), as applicable.

MARKING AND REPORTS

UW-60 GENERAL

The provisions for marking and reports, UG-115 through UG-120, shall apply without supplement to welded pressure vessels.

PART UHX
RULES FOR SHELL-AND-TUBE HEAT EXCHANGERS

UHX-1 SCOPE

(a) The rules in Part UHX cover the minimum requirements for design, fabrication, and inspection of shell-and-tube heat exchangers.

(b) The rules in Part UHX cover the common types of shell-and-tube heat exchangers and their elements but are not intended to limit the configurations or details to those illustrated or otherwise described herein. Designs that differ from those covered in this Part shall be in accordance with U-2(g).

UHX-2 MATERIALS AND METHODS OF FABRICATION

Materials and methods of fabrication of heat exchangers shall be in accordance with Subsections A, B, and C.

UHX-3 TERMINOLOGY

UHX-3.1 U-Tube Heat Exchanger

Heat exchanger with one stationary tubesheet attached to the shell and channel. The heat exchanger contains a bundle of U-tubes attached to the tubesheet [see Figure UHX-3, sketch (a)].

UHX-3.2 Fixed Tubesheet Heat Exchanger

Heat exchanger with two stationary tubesheets, each attached to the shell and channel. The heat exchanger contains a bundle of straight tubes connecting both tubesheets [see Figure UHX-3, sketch (b)].

UHX-3.3 Floating Tubesheet Heat Exchanger

Heat exchanger with one stationary tubesheet attached to the shell and channel, and one floating tubesheet that can move axially. The heat exchanger contains a bundle of straight tubes connecting both tubesheets [see Figure UHX-3, sketch (c)].

(19) UHX-4 DESIGN

(a) The design of all components shall be in accordance with the applicable rules of Subsection A, Mandatory Appendices, and this Part.

(b) Flanges with pass partitions, including those covered by UG-44(a), shall be designed in accordance with Mandatory Appendix 2, and the effects of pass partition gasketing shall be considered in determining the minimum required bolt loads, W_{m1} and W_{m2}, of Mandatory Appendix 2. When the tubesheet is gasketed between the shell and channel flanges, the shell and channel flange bolt loads are identical and shall be treated as flange pairs in accordance with Mandatory Appendix 2.

(c) Requirements for distribution and vapor belts shall be as follows:

(1) Distribution and vapor belts where the shell is not continuous across the belt shall be designed in accordance with UHX-17.

(2) Distribution and vapor belts, where the shell is continuous across the belt, shall be designed in accordance with a Type 1 jacket in Mandatory Appendix 9. The longitudinal stress in the shell section with openings (for flow into the shell) shall be based on the net area of the shell (the shell area less that removed by the openings) and shall not exceed the applicable allowable stress criteria. For U-tube and floating head exchangers, the allowable axial stress is the maximum allowable stress for the shell material (see UG-23), and for fixed tubesheet exchangers, the allowable stress is as defined in UHX-13.5.10.

(d) Requirements for tubes shall be as follows:

(1) The allowable axial tube stresses in fixed and floating tubesheet heat exchangers given in this Part UHX-13 and UHX-14 supersede the requirements of UG-23.

(2) The thickness of U-tubes after forming shall not be less than the design thickness.

(e) Rules for U-tube heat exchangers are covered in UHX-12.

(f) Rules for fixed tubesheet heat exchangers are covered in UHX-13.

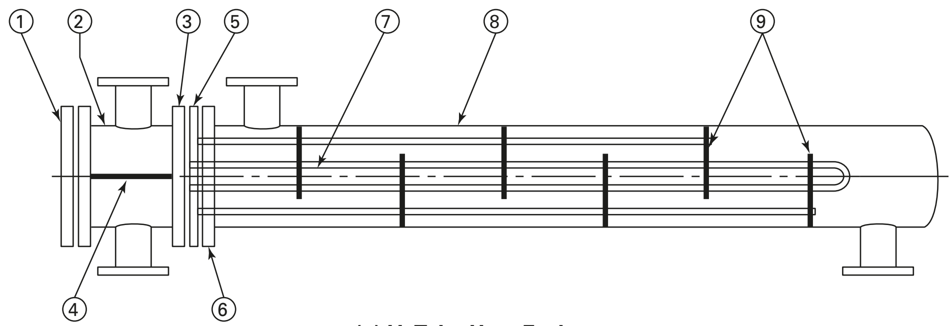
(g) Rules for floating tubesheet heat exchangers are covered in UHX-14.

(h) Except as limited in (1) and (2) below, nozzles in cylindrical shells or cylindrical channels adjacent to integral tubesheets (see Figure UHX-4-1) may be located at any distance from the tubesheet (refer to UG-37 and Figure UG-40 for nomenclature not defined in this paragraph). These requirements do not apply to nozzles in shells or channels having tubesheets that are calculated as simply supported (see UHX-12.6, UHX-13.9, and UHX-14.7).

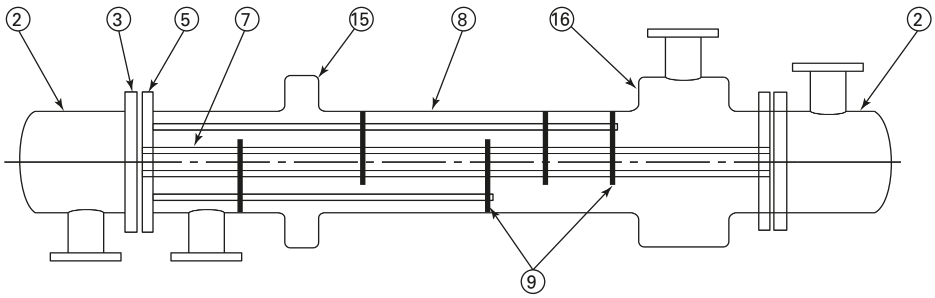
(1) For a circular nozzle with d greater than 30% of D, no part of d may be located within $1.8(Dt)^{1/2}$ of the adjacent tubesheet face (see Figure UHX-4-1).

(2) For a noncircular nozzle, d_{max} (major axis) is defined as the maximum diameter of d, and d_{min} is defined as the minimum diameter of d.

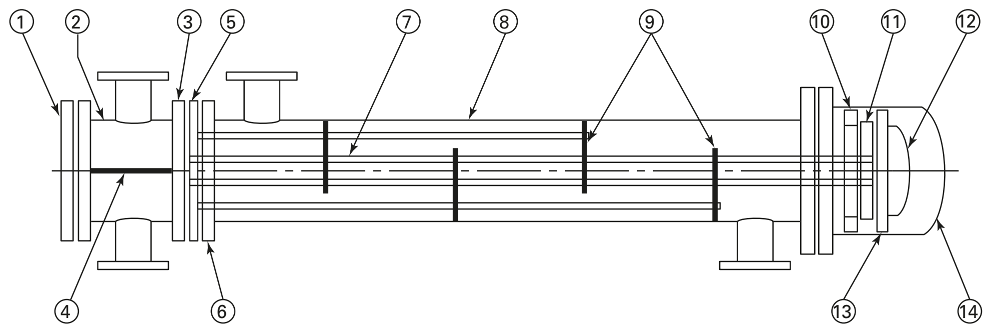
Figure UHX-3
Terminology of Heat Exchanger Components



(a) U–Tube Heat Exchanger



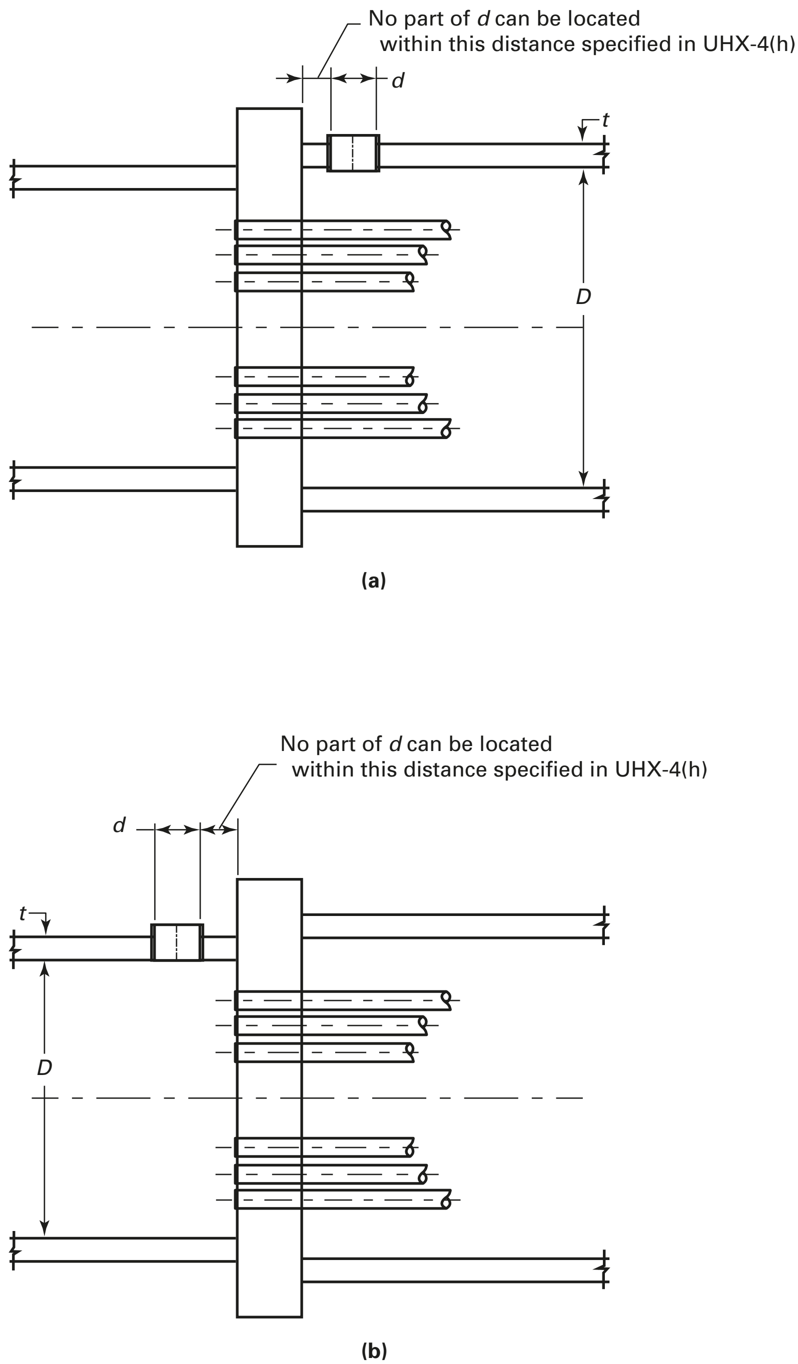
(b) Fixed Tubesheet Heat Exchanger



(c) Floating Tubesheet Heat Exchanger

1 Channel cover (bolted flat cover)
2 Channel
3 Channel flange
4 Pass partition
5 Stationary tubesheet
6 Shell flange
7 Tubes
8 Shell
9 Baffles or support plates
10 Floating head backing device
11 Floating tubesheet
12 Floating head
13 Floating head flange
14 Shell cover
15 Expansion joint
16 Distribution or vapor belt

Figure UHX-4-1
Nozzles Adjacent to Tubesheets (19)



(-a) For a noncircular nozzle having its major axis not parallel to the tubesheet face and d_{max}/D > 30%, d is limited to the distance specified in (1).

(-b) For a noncircular nozzle having its major axis parallel to the tubesheet face and d_{max}/D > 30%, no part of d may be within $1.8(Dt)^{1/2} + (d_{max} - d_{min})/2$ of the adjacent tubesheet face.

(3) Nozzles subject to the limitations in (1) or (2) above may have their required reinforcement (see UG-37) located within $1.8(Dt)^{1/2}$ of the adjacent tubesheet face.

NOTE: Tubesheet deflection, especially when the tubesheet thickness is less than the tube diameter, may contribute to tube-to-tubesheet joint leakage; likewise, deflection of a tubesheet or flat bolted cover may result in fluid leakage across a gasketed pass partition plate. Such leakages can be detrimental to the thermal performance of the heat exchanger, and deflection may need to be considered by the designer.

UHX-8 TUBESHEET EFFECTIVE BOLT LOAD, W^*

UHX-8.1 Scope

Table UHX-8.1 provides the tubesheet effective bolt load, W^*, transmitted to the perforated region of the tubesheet for each combination of Configuration and Loading Case. The bolt loads shall be calculated using the appropriate formula from Mandatory Appendix 2 considering the requirements in UHX-4(b).

UHX-8.2 Nomenclature

W_c = channel flange design bolt load for the gasket seating condition (see Mandatory Appendix 2)

W_{m1c} = channel flange design bolt load (see definition for W_{m1} in Mandatory Appendix 2, 2-3)

W_{m1max} = MAX[(W_{m1c}), (W_{m1s})]

W_{m1s} = shell flange design bolt load (see definition for W_{m1} in Mandatory Appendix 2, 2-3)

W_{max} = MAX[(W_c), (W_s)]

W_s = shell flange design bolt load for the gasket seating condition (see Mandatory Appendix 2)

W^* = tubesheet effective bolt load selected from Table UHX-8.1 for the respective Configuration and Loading Case

Table UHX-8.1
Tubesheet Effective Bolt Load, W^*

Configuration	Design Loading Cases				Operating Loading Cases
	1	2	3	4	1–4
a	0	0	0	0	0
b	W_{m1c}	0	W_{m1c}	0	W_c
c	W_{m1c}	0	W_{m1c}	0	W_c
d	W_{m1c}	W_{m1s}	W_{m1max}	0	W_{max}
e	0	W_{m1s}	W_{m1s}	0	W_s
f	0	W_{m1s}	W_{m1s}	0	W_s
A	0	0	0	0	0
B	W_{m1c}	0	W_{m1c}	0	W_c
C	W_{m1c}	0	W_{m1c}	0	W_c
D	0	0	0	0	0

UHX-9 TUBESHEET EXTENSION

UHX-9.1 Scope

(a) Tubesheet extensions, if present, may be extended as a flange (flanged) or not extended as a flange (unflanged).

(1) Configuration a tubesheets may have no extension or an unflanged extension.

(2) Configurations b, e, and B tubesheets have flanged extensions.

(3) Configurations c, f, and C tubesheets have unflanged extensions.

(4) Configuration d may have a flanged or unflanged tubesheet extension.

(5) Configurations A and D do not have tubesheet extensions.

(b) These rules cover the design of tubesheet extensions that have loads applied to them.

(c) The required thickness of the tubesheet extension may differ from that required for the interior of the tubesheet as calculated in UHX-12, UHX-13, or UHX-14.

UHX-9.2 Conditions of Applicability

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

(b) These rules do not apply to Configurations a, A, and D.

(c) These rules apply to flanged extensions that have bolt loads applied to them (Configurations b, e, and B). This includes Configuration d if the extension is flanged and there are bolt loads applied to the extension.

(d) These rules apply to unflanged extensions (Configurations c, d, f, and C) and flanged extensions that have no bolt loads applied to them (Configuration d), if the thickness of the extension is less than the tubesheet thickness, h. If the tubesheet extension is equal to or greater than the tubesheet thickness, h, no analysis is required.

UHX-9.3 Nomenclature

The following symbols are used for determining the required thickness of the tubesheet extension:

D_E = maximum of the shell and channel gasket inside diameters, but not less than the maximum of the shell and channel flange inside diameters

G = diameter of gasket load reaction

= G_c for tubesheet configuration b of a U-tube tubesheet heat exchanger
= G_s for tubesheet configuration e of a U-tube tubesheet heat exchanger
= G_c for tubesheet configuration b of a fixed tubesheet heat exchanger
= G_c for stationary tubesheet configuration b of a floating tubesheet exchanger
= G_s for stationary tubesheet configuration e of a floating tubesheet exchanger
= G_c for floating tubesheet configuration B of a floating tubesheet exchanger
= G_c or G_s for tubesheet configuration d when applicable (e.g., hydrotest)

h_G = gasket moment arm, equal to the radial distance from the centerline of the bolts to the line of the gasket reaction as shown in Table 2-5.2

h_r = minimum required thickness of the tubesheet extension

MAX *[(a), (b),(c),...]* = greatest of *a, b, c,...*

P_s = shell side design pressure. For shell side vacuum, use a negative value for P_s.

P_t = tube side design pressure. For tube side vacuum, use a negative value for P_t.

S_a = allowable stress for the material of the tubesheet extension at ambient temperature (see UG-23)

S_{fe} = allowable stress for the material of the tubesheet extension at tubesheet extension design temperature (see UG-23)

W = flange design bolt load from eq. 2-5(e)(5) considering UHX-4(b)

W_{m1} = flange design bolt load from eq. 2-5(c)(1)(1) considering UHX-4(b)

UHX-9.4 Design Considerations

(a) The designer shall take appropriate consideration of the stresses resulting from the pressure test required by UG-99 or UG-100 [see UG-99(b) and UG-99(d)]. Special consideration shall be required for tubesheets that are gasketed on both sides when the pressure test in each chamber is conducted independently and the bolt loading is only applied to the flanged extension during the pressure test.

(b) If the tubesheet is grooved for a peripheral gasket, the net thickness under the groove or between the groove and the outer edge of the tubesheet shall not be less than h_r. Figure UHX-9 depicts thickness h_r for some representative configurations.

UHX-9.5 Calculation Procedure

(a) For flanged extensions that have bolt loads applied to them [Configurations b, d (extended for bolting), e, and B], the procedure for calculating the minimum required thickness of the extension, h_r, is as follows:

$$h_r = \mathrm{MAX}\left[\sqrt{\frac{1.9Wh_G}{S_aG}}, \sqrt{\frac{1.9W_{m1}h_G}{S_{fe}G}}\right]$$

(b) For unflanged Configurations c and f, the minimum required thickness of the extension, h_r, shall be calculated in accordance with Mandatory Appendix 2, 2-8(c) for loose-type flanges with laps.

(c) For unflanged Configurations d and C and for flanged Configuration d having no bolt loads applied to the extension, the minimum required thickness of the extension, h_r, shall be the maximum of the values determined for each design loading case as follows:

$$h_r = \left(\frac{D_E}{3.2S_{fe}}\right)\left|P_s - P_t\right|$$

UHX-10 GENERAL CONDITIONS OF APPLICABILITY FOR TUBESHEETS (19)

(a) The tubesheet shall be flat and circular.

(b) The tubesheet shall be of uniform thickness, except that the thickness of a tubesheet extension as determined in UHX-9 may differ from the center thickness as determined in UHX-12, UHX-13, and UHX-14. The outside diameter, A, used for the tubesheet calculations shall not exceed the diameter at which the thickness of the tubesheet extension is less than the minimum of 0.75h or h – 0.375 in. (h – 10 mm).

(c) The tubesheet shall be uniformly perforated over a nominally circular area, in either equilateral triangular or square patterns. However, untubed lanes for pass partitions are permitted.

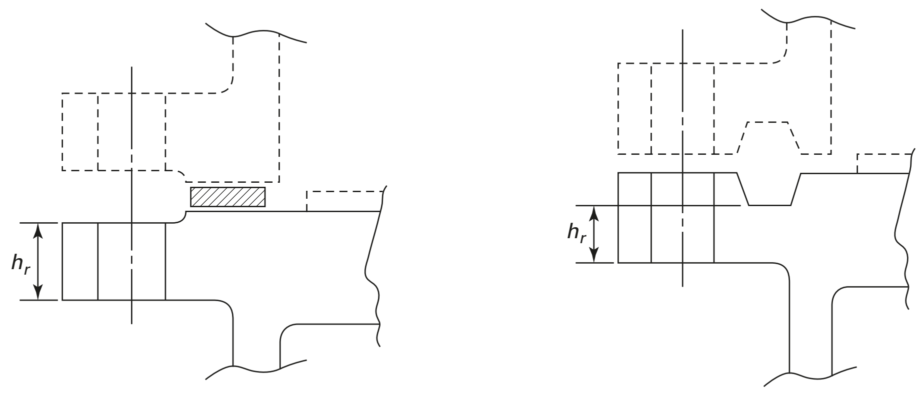
(d) The channel component integral with the tubesheet (UHX-12.5, UHX-13.5, and UHX-14.5; configurations a, e, f, and A) shall be either a cylinder or a hemispherical head (see Figure UHX-10). The hemispherical head rules shall be used when the head is attached directly to the tubesheet and there are no cylindrical sections between the head and the tubesheet. If a hemispherical head is attached to the hub of a tubesheet, the hub may be considered part of the hemispherical head and not require an intervening cylinder, provided the hub complies with one of the following conditions:

(1) It is shaped as a continuation of the head in accordance with Figure UHX-10, sketch (b).

(2) It meets the requirements of Figure UHX-10, sketch (c).

For both cases, the tangent line of the head is coincident with the adjacent face of the tubesheet.

Figure UHX-9
Some Representative Configurations Describing the Minimum Required Thickness of the Tubesheet Flanged Extension, h_r



(a) Raised Face (b) Grooved for Ring Gasket

(e) The tube side and shell side pressures are assumed to be uniform. These rules do not cover weight loadings or pressure drop.

(f) The design pressure or operating pressure defined in the nomenclature is the applicable pressure in the shell side or tube side chamber, including any static head, not the coincident pressure defined in UG-21. For the design-pressure-only conditions (design loading cases), the design pressure shall be used. For the operating-thermal-pressure conditions (operating loading cases), either the operating pressure or design pressure shall be used.

(g) The design rules in UHX-12, UHX-13, and UHX-14 are based on a fully assembled heat exchanger. If pressure is to be applied to a partially assembled heat exchanger having a Configuration d tubesheet that is extended for bolting, special consideration, in addition to the rules given in UHX-9, UHX-12, UHX-13, and UHX-14, shall be given to ensure that the tubesheet is not overstressed for the condition considered.

UHX-11 TUBESHEET CHARACTERISTICS

UHX-11.1 Scope

These rules cover the determination of the ligament efficiencies, effective depth of the tube side pass partition groove, and effective elastic constants to be used in the calculation of U-tube, fixed, and floating tubesheets.

UHX-11.2 Conditions of Applicability

The general conditions of applicability given UHX-10 apply.

(19) UHX-11.3 Nomenclature

The symbols described below are used for determining the effective elastic constants.

A_L = total area of untubed lanes
= $U_{L1}L_{L1} + U_{L2}L_{L2} + \ldots$ (limited to $4D_o p$)

c_t = tubesheet corrosion allowance on the tube side
= 0 in the uncorroded condition

D_o = equivalent diameter of outer tube limit circle [see Figure UHX-11.3-1, sketch (a)]

d = diameter of tube hole

d_t = nominal outside diameter of tubes

d^* = effective tube hole diameter

E = modulus of elasticity for tubesheet material at tubesheet design temperature

E_{tT} = modulus of elasticity for tube material at tubesheet design temperature

E^* = effective modulus of elasticity of tubesheet in perforated region

h = tubesheet thickness

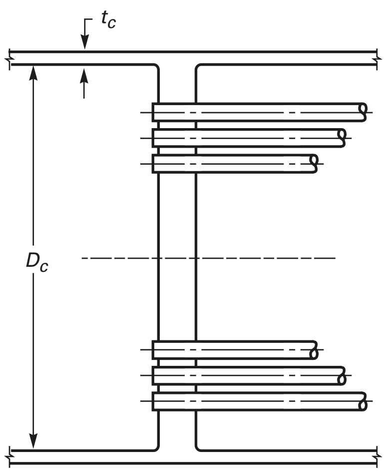
h_g = tube side pass partition groove depth [see Figure UHX-11.3-1, sketch (c)]

h'_g = effective tube side pass partition groove depth

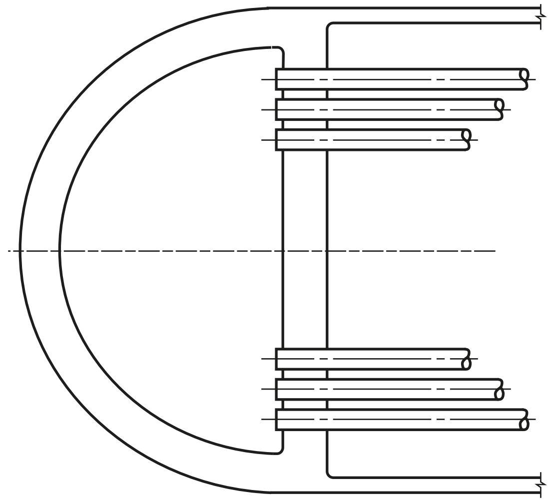
$L_{L1}, L_{L2} \ldots$ = length(s) of untubed lane(s) (see Figure UHX-11.3-2)

ℓ_{tx} = expanded length of tube in tubesheet ($0 \le \ell_{tx} \le h$) [see Figure UHX-11.3-1, sketch (b)]. An expanded tube-to-tubesheet joint is produced by applying pressure inside the tube such that contact is established between the tube and tubesheet. In selecting an appropriate value of expanded length, the designer shall consider the degree of initial expansion, differences in thermal expansion, or other factors that could result in loosening of the tubes within the tubesheet.

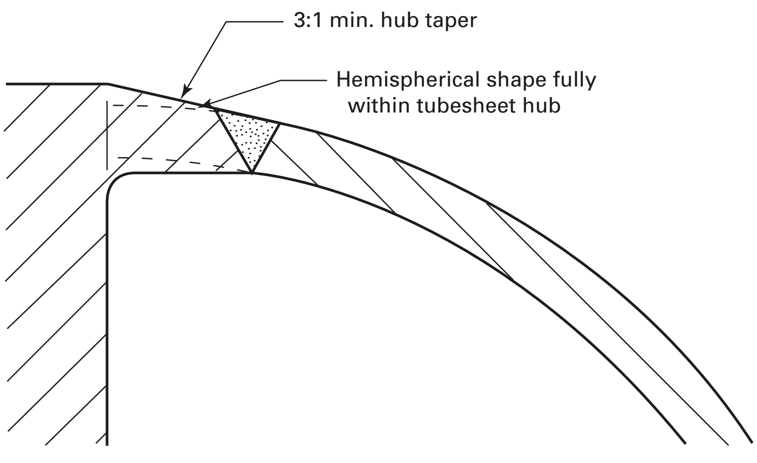
Figure UHX-10 Integral Channels (19)



(a) Cylindrical Channel [Note (1)]



(b) Hemispherical Channel [Notes (2) and (3)]



(c) Hemispherical Channel With Tubesheet Hub Thicker Than Channel

NOTES:

(1) Length of cylinder shall be $\geq 1.8\sqrt{D_c t_c}$.

(2) Head shall be 180 deg with no intervening cylinders.

(3) These rules also apply to channels integral with tubesheets having extensions.

MAX [(a), (b),(c),...] = greatest of a, b, c,...
MIN [(a), (b),(c),...] = smallest of a, b, c,...
p = tube pitch
p^* = effective tube pitch
r_o = radius to outermost tube hole center [see Figure UHX-11.3-1, sketch (a)]
S = allowable stress for tubesheet material at tubesheet design temperature (see UG-23)
S_{tT} = allowable stress for tube material at tubesheet design temperature (see UG-23)

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

T' = tubesheet metal temperature at the rim (see Figure UHX-11.3-3)
t_t = nominal tube wall thickness
U_{L1}, U_{L2}... = center-to-center distance(s) between adjacent tube rows of untubed lane(s), but not to exceed $4p$ (see Figure UHX-11.3-2)
μ = basic ligament efficiency for shear
μ^* = effective ligament efficiency for bending
ν^* = effective Poisson's ratio in perforated region of tubesheet
ρ = tube expansion depth ratio = ℓ_{tx}/h, $(0 \le \rho \le 1)$

UHX-11.4 Design Considerations

(a) Elastic moduli and allowable stresses shall be taken at the design temperatures. However, for cases involving thermal loading, it is permitted to use the operating temperatures instead of the design temperatures.

(b) When the values calculated in this section are to be used for fixed tubesheets, they shall be determined in both the corroded and uncorroded conditions.

(c) ρ may be either calculated or chosen as a constant.

UHX-11.5 Calculation Procedure

UHX-11.5.1 Determination of Effective Dimensions and Ligament Efficiencies. From the geometry (see Figure UHX-11.3-1 and Figure UHX-11.3-2) and material properties of the exchanger, calculate the required parameters in accordance with (a) or (b) below.

(a) For geometries where the tubes extend through the tubesheet [see Figure UHX-11.3-1, sketch (b)], calculate D_o, μ, d^*, p^*, μ^*, and h'_g.

$$D_o = 2r_o + d_t$$

$$\mu = \frac{p - d_t}{p}$$

$$d^* = \text{MAX}\left\{\left[d_t - 2t_t\left(\frac{E_{tT}}{E}\right)\left(\frac{S_{tT}}{S}\right)\rho\right], \left[d_t - 2t_t\right]\right\}$$

$$p^* = \frac{p}{\left(1 - \frac{4\text{MIN}\left[\left(A_L\right), \left(4D_o p\right)\right]}{\pi D_o^2}\right)^{\frac{1}{2}}}$$

$$\mu^* = \frac{p^* - d^*}{p^*}$$

$$h'_g = \text{MAX}\left[\left(h_g - c_t\right), (0)\right]$$

(b) For tubes welded to the backside of the tubesheet [see Figure UHX-11.3-1, sketch (d)], calculate D_o, μ, p^*, μ^*, and h'_g.

$$D_o = 2r_o + d$$

$$\mu = \frac{p - d}{p}$$

$$p^* = \frac{p}{\left(1 - \frac{4\text{MIN}\left[\left(A_L\right), \left(4D_o p\right)\right]}{\pi D_o^2}\right)^{\frac{1}{2}}}$$

$$\mu^* = \frac{p^* - d}{p^*}$$

$$h'_g = \text{MAX}\left[\left(h_g - c_t\right), (0)\right]$$

UHX-11.5.2 Determination of Effective Elastic Properties. Determine the values for E^*/E and ν^* relative to h/p using either Figure UHX-11.5.2-1 (equilateral triangular pattern) or Figure UHX-11.5.2-2 (square pattern).

UHX-12 RULES FOR THE DESIGN OF U-TUBE TUBESHEETS

UHX-12.1 Scope

These rules cover the design of tubesheets for U-tube heat exchangers. The tubesheet may have one of the six configurations shown in Figure UHX-12.1:

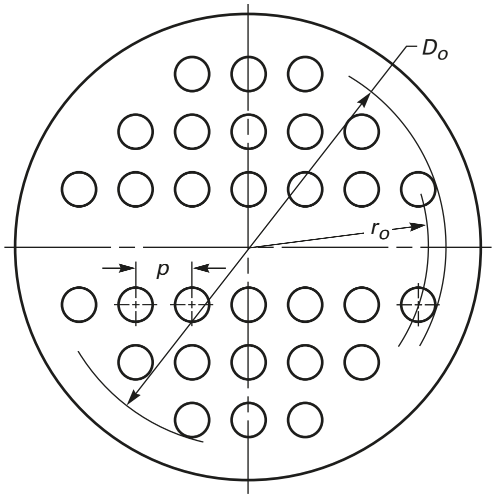
(a) Configuration a: tubesheet integral with shell and channel;

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

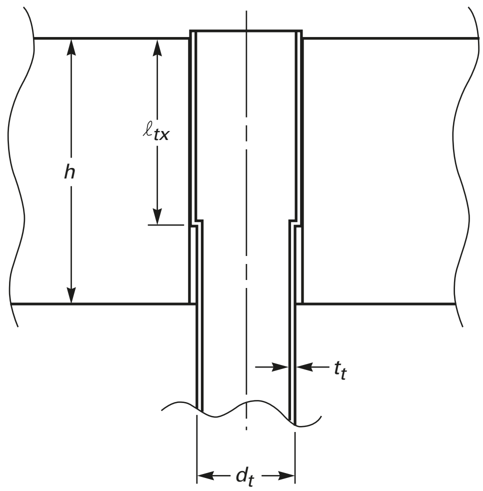
(c) Configuration c: tubesheet integral with shell and gasketed with channel, not extended as a flange;

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

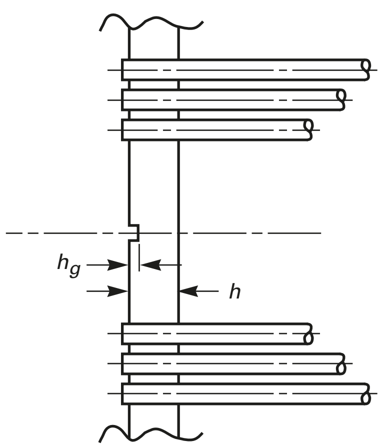
Figure UHX-11.3-1 Tubesheet Geometry (19)



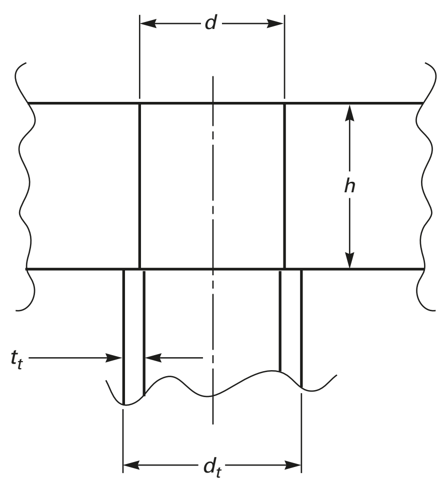
(a) Tubesheet Layout



(b) Expanded Tube Joint



(c) Tube Side Pass Partition Groove Depth

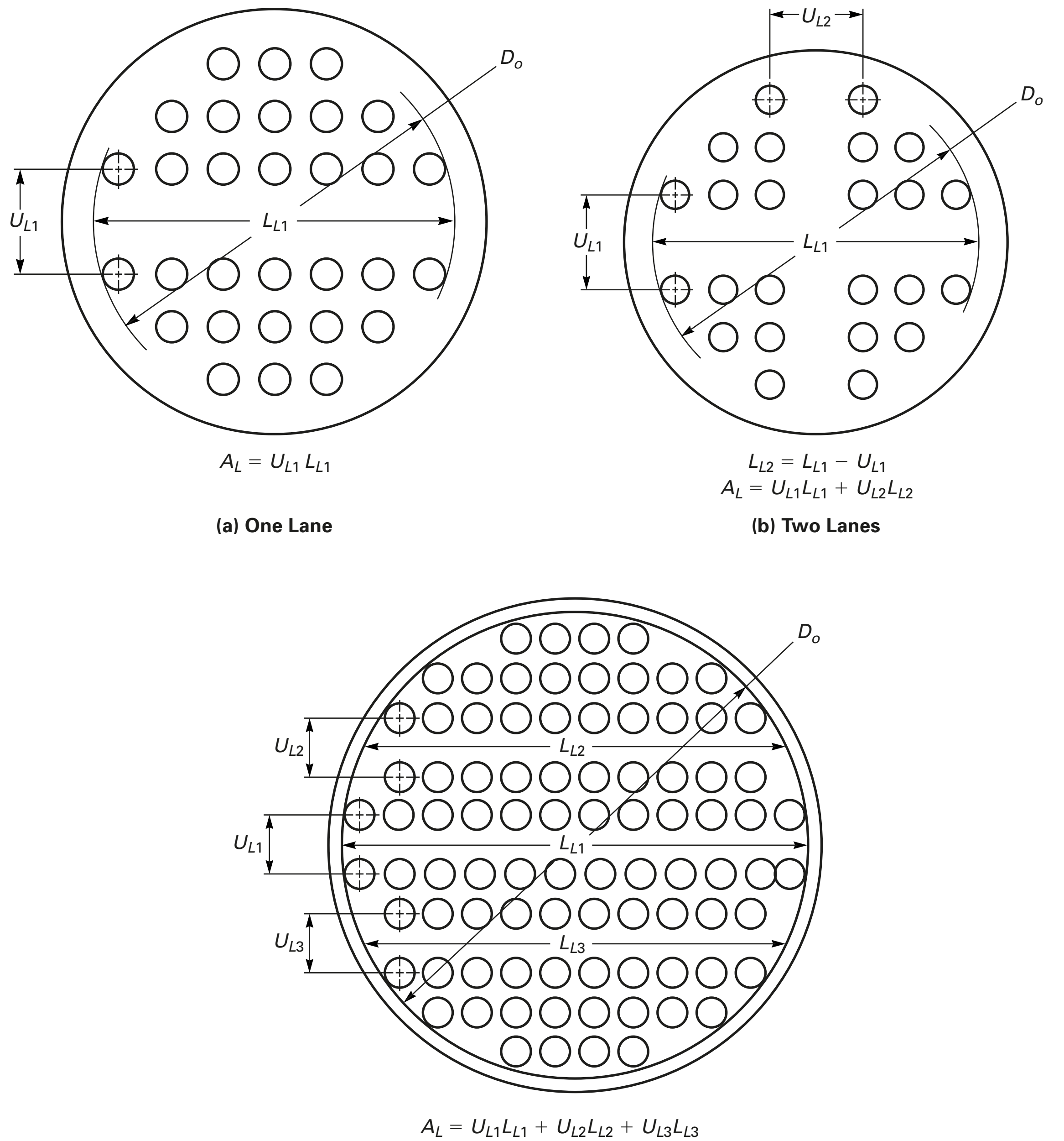


(d) Tubes Welded to Back Side of Tubesheet [See Note (1)]

NOTE:
(1) $d_t - 2t_t \le d < d_t$

(19)

Figure UHX-11.3-2
Typical Untubed Lane Configurations



$A_L = U_{L1}\,L_{L1}$

(a) One Lane

$L_{L2} = L_{L1} - U_{L1}$
$A_L = U_{L1}L_{L1} + U_{L2}L_{L2}$

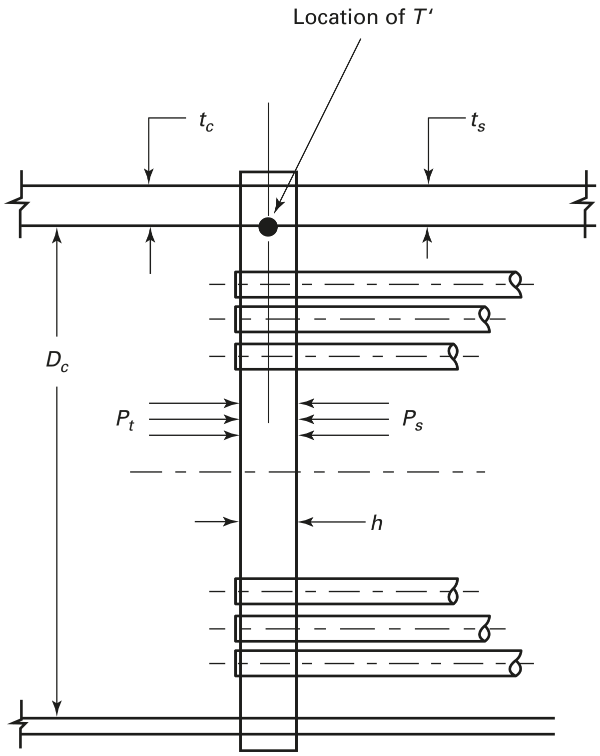
(b) Two Lanes

$A_L = U_{L1}L_{L1} + U_{L2}L_{L2} + U_{L3}L_{L3}$

(c) Three Lanes

(19)

Figure UHX-11.3-3
Location of Tubesheet Metal Temperature, T', at Rim



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

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

UHX-12.2 Conditions of Applicability

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

(19) UHX-12.3 Nomenclature

The symbols described below are used for the design of the tubesheet. Symbols D_o, E^*, h'_g, μ, μ^*, and ν^* are defined in UHX-11.

A = outside diameter of tubesheet, except as limited by UHX-10(b)

A_p = total area enclosed by C_p

C = bolt circle diameter (see Mandatory Appendix 2)

C_p = perimeter of the tube layout measured stepwise in increments of one tube pitch from the center-to-center of the outermost tubes (see Figure UHX-12.2)

D_c = inside channel diameter

D_s = inside shell diameter

E = modulus of elasticity for tubesheet material at design temperature

E_c = modulus of elasticity for channel material at design temperature

E_s = modulus of elasticity for shell material at design temperature

G_1 = midpoint of contact between flange and tubesheet

G_c = diameter of channel gasket load reaction (see Mandatory Appendix 2)

G_s = diameter of shell gasket load reaction (see Mandatory Appendix 2)

h = tubesheet thickness

MAX *[(a), (b),(c),...]* = greatest of *a, b, c,...*

P_s = shell side design pressure. For shell side vacuum, use a negative value for P_s

$P_{sd,\max}$ = maximum shell side design pressure

$P_{sd,\min}$ = minimum shell side design pressure (negative if vacuum is specified, otherwise zero)

P_t = tube side design pressure. For tube side vacuum, use a negative value for P_t

$P_{td,\max}$ = maximum tube side design pressure

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

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

S_c = allowable stress for channel material at design temperature

S_s = allowable stress for shell material at design temperature

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

S_y = yield strength for tubesheet material at tubesheet design temperature

$S_{y,c}$ = yield strength for channel material at design temperature

$S_{y,s}$ = yield strength for shell material at design temperature

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

t_c = channel thickness

t_s = shell thickness

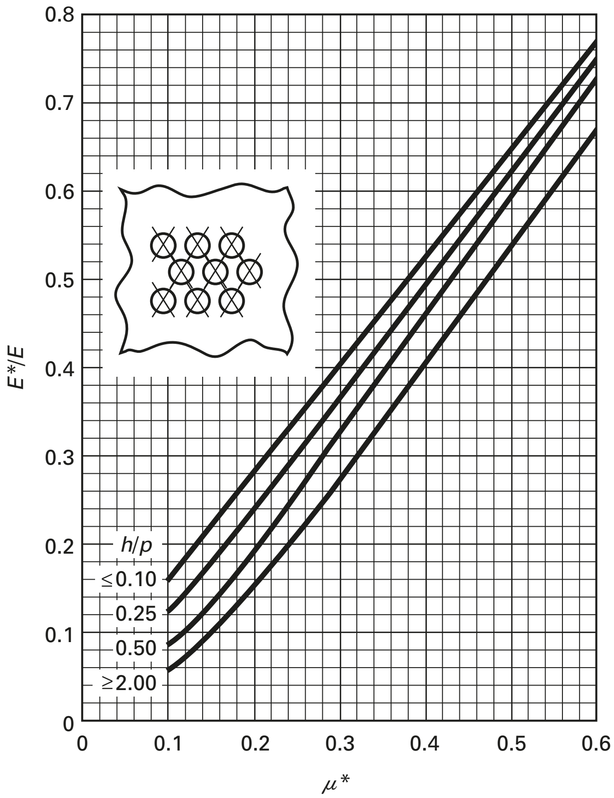
W^* = tubesheet effective bolt load determined in accordance with UHX-8

ν_c = Poisson's ratio of channel material

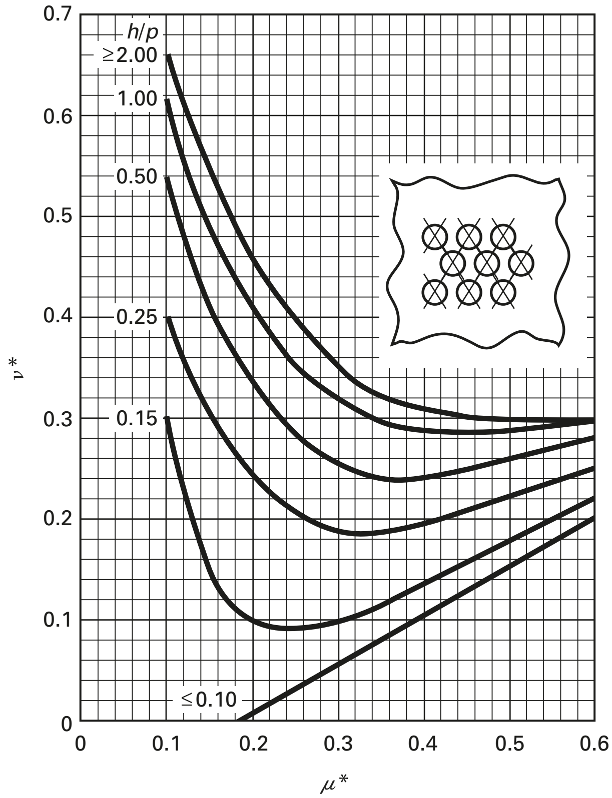
ν_s = Poisson's ratio of shell material

(**19**)

Figure UHX-11.5.2-1
Curves for the Determination of E^*/E and ν^* (Equilateral Triangular Pattern)



(a) E^*/E (Equilateral Triangular Pattern)



(b) ν^* (Equilateral Triangular Pattern)

(a) Equilateral Triangular Pattern: $E^*/E = \alpha_0 + \alpha_1\mu^* + \alpha_2\mu^{*2} + \alpha_3\mu^{*3} + \alpha_4\mu^{*4}$

h/p	α_0	α_1	α_2	α_3	α_4
0.10	0.0353	1.2502	−0.0491	0.3604	−0.6100
0.25	0.0135	0.9910	1.0080	−1.0498	0.0184
0.50	0.0054	0.5279	3.0461	−4.3657	1.9435
2.00	−0.0029	0.2126	3.9906	−6.1730	3.4307

(b) Equilateral Triangular Pattern: $\nu^* = \beta_0 + \beta_1\mu^* + \beta_2\mu^{*2} + \beta_3\mu^{*3} + \beta_4\mu^{*4}$

h/p	β_0	β_1	β_2	β_3	β_4
0.10	−0.0958	0.6209	−0.8683	2.1099	−1.6831
0.15	0.8897	−9.0855	36.1435	−59.5425	35.8223
0.25	0.7439	−4.4989	12.5779	−14.2092	5.7822
0.50	0.9100	−4.8901	12.4325	−12.7039	4.4298
1.00	0.9923	−4.8759	12.3572	−13.7214	5.7629
2.0	0.9966	−4.1978	9.0478	−7.9955	2.2398

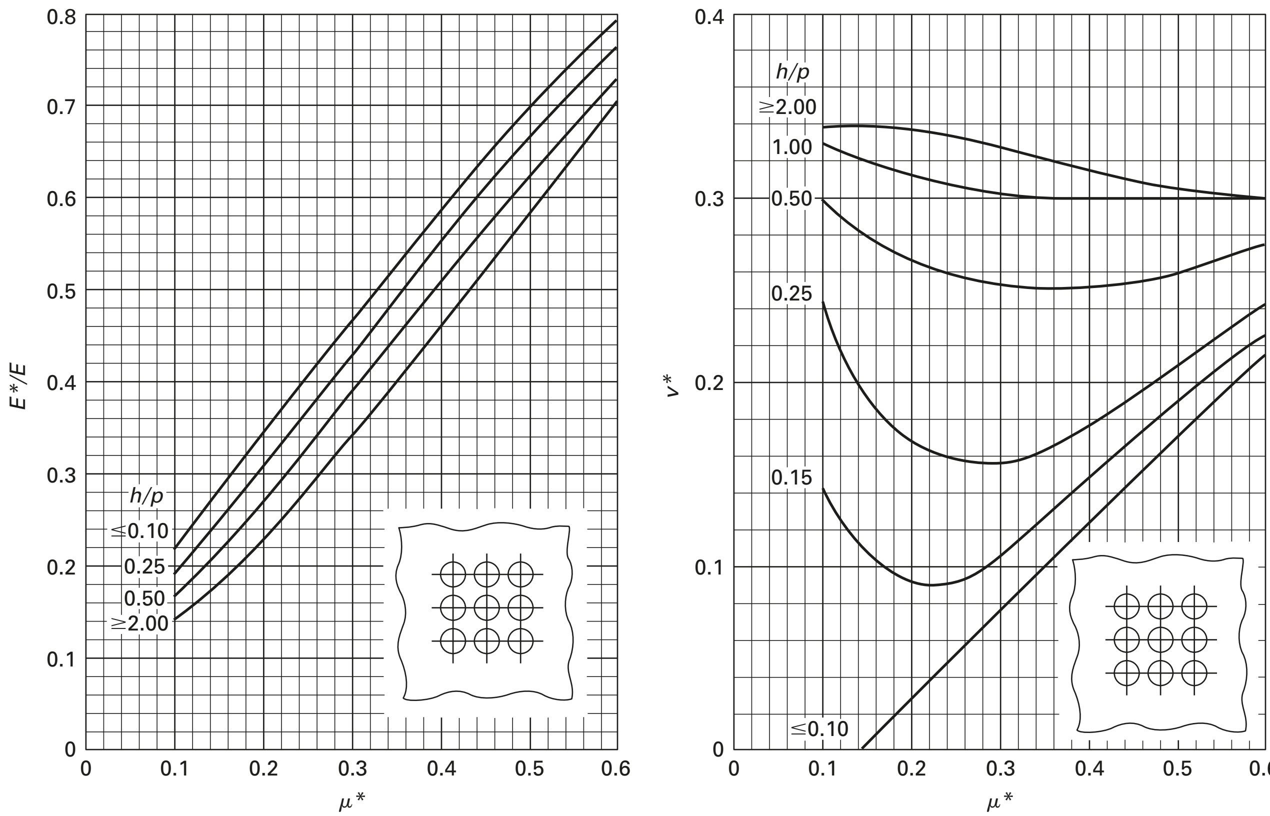
GENERAL NOTES:

(a) The polynomial equations given in the tabular part of this Figure can be used in lieu of the curves.

(b) For both parts (a) and (b) in the tabular part of this Figure, these coefficients are only valid for $0.1 \le \mu^* \le 0.6$.

(c) For both parts (a) and (b) in the tabular part of this Figure: for values of h/p lower than 0.1, use $h/p = 0.1$; for values of h/p higher than 2.0, use $h/p = 2.0$.

Figure UHX-11.5.2-2
Curves for the Determination of E^*/E and ν^* (Square Pattern)
(19)



(a) E^*/E (Square Pattern) **(b) ν^* (Square Pattern)**

(a) Square Pattern: $E^*/E = \alpha_0 + \alpha_1\mu^* + \alpha_2\mu^{*2} + \alpha_3\mu^{*3} + \alpha_4\mu^{*4}$

h/p	α_0	α_1	α_2	α_3	α_4
0.10	0.0676	1.5756	−1.2119	1.7715	−1.2628
0.25	0.0250	1.9251	−3.5230	6.9830	−5.0017
0.50	0.0394	1.3024	−1.1041	2.8714	−2.3994
2.00	0.0372	1.0314	−0.6402	2.6201	−2.1929

(b) Square Pattern: $\nu^* = \beta_0 + \beta_1\mu^* + \beta_2\mu^{*2} + \beta_3\mu^{*3} + \beta_4\mu^{*4}$

h/p	β_0	β_1	β_2	β_3	β_4
0.10	−0.0791	0.6008	−0.3468	0.4858	−0.3606
0.15	0.3345	−2.8420	10.9709	−15.8994	8.3516
0.25	0.4296	−2.6350	8.6864	−11.5227	5.8544
0.50	0.3636	−0.8057	2.0463	−2.2902	1.1862
1.00	0.3527	−0.2842	0.4354	−0.0901	−0.1590
2.00	0.3341	0.1260	−0.6920	0.6877	−0.0600

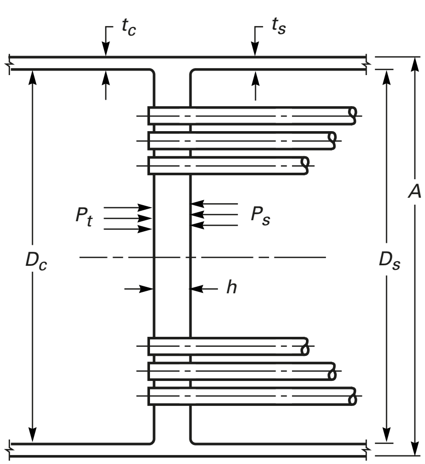
GENERAL NOTES:

(a) The polynomial equations given in the tabular part of this Figure can be used in lieu of the curves.

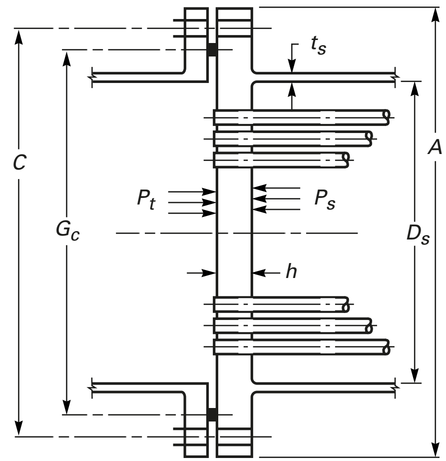
(b) For both parts (a) and (b) in the tabular part of this Figure, these coefficients are only valid for $0.1 \leq \mu^* \leq 0.6$.

(c) For both parts (a) and (b) in the tabular part of this Figure: for values of h/p lower than 0.1, use h/p = 0.1; for values of h/p higher than 2.0, use h/p = 2.0.

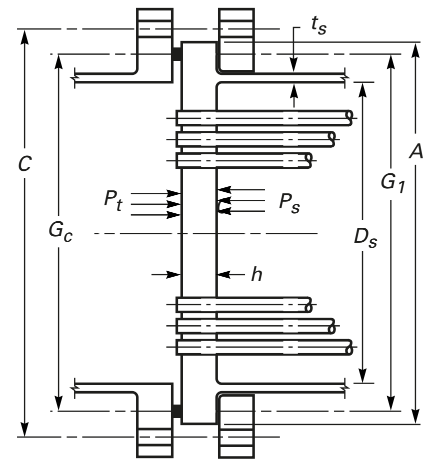
Figure UHX-12.1
U-Tube Tubesheet Configurations



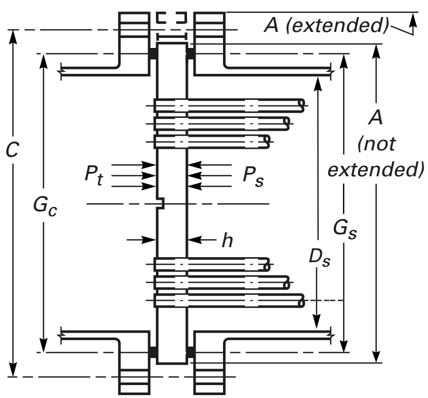
(a) Configuration a:
Tubesheet Integral With Shell and Channel



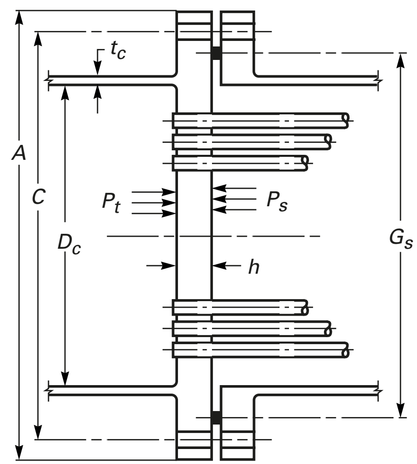
(b) Configuration b:
Tubesheet Integral With Shell and Gasketed With Channel, Extended as a Flange



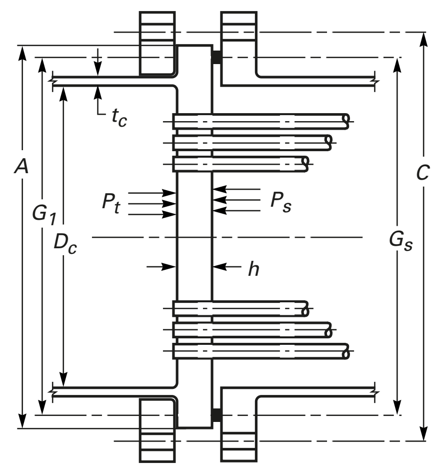
(c) Configuration c:
Tubesheet Integral With Shell and Gasketed With Channel, Not Extended as a Flange



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

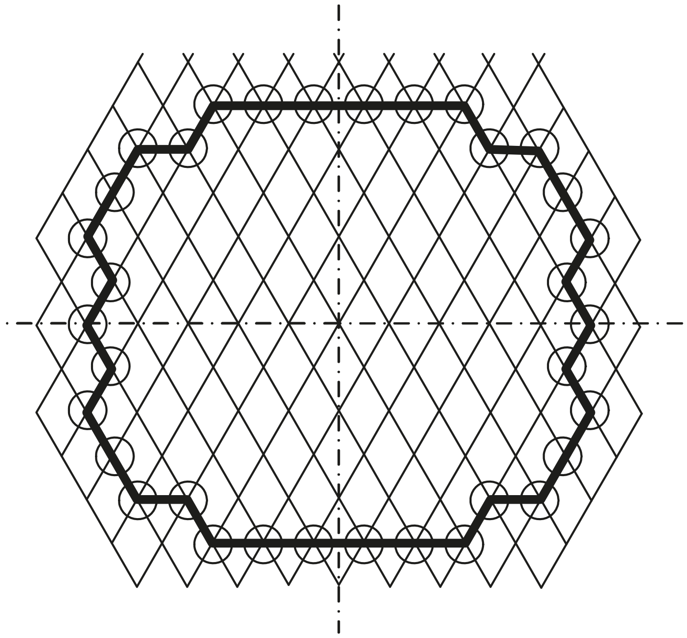


(e) Configuration e:
Tubesheet Gasketed With Shell and Integral With Channel, Extended as a Flange

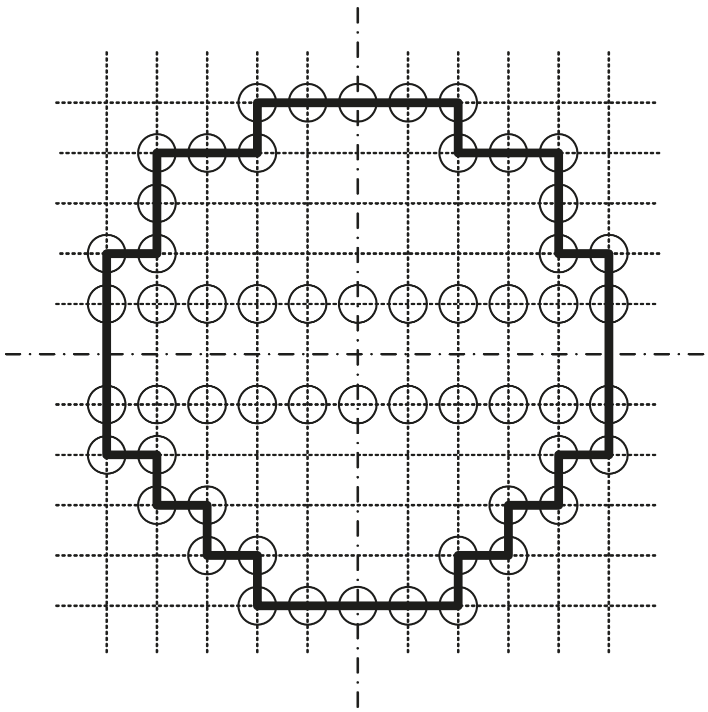


(f) Configuration f:
Tubesheet Gasketed With Shell and Integral With Channel, Not Extended as a Flange

Figure UHX-12.2
Tube Layout Perimeter



(a) Equilateral Triangular Pattern



(b) Square Pattern

GENERAL NOTE: C_p (perimeter) is the length of the heavy line.

UHX-12.4 Design Considerations

(a) The various loading conditions to be considered shall include, but not be limited to, normal operating, startup, shutdown, cleaning, and upset conditions, which may govern the design of the tubesheet.

For each of these conditions, the following loading cases shall be considered:

(1) Design Loading Cases. Table UHX-12.4-1 provides the load combinations required to evaluate the heat exchanger for the design condition. When $P_{sd,\min}$ and $P_{td,\min}$ are both zero, design loading case 4 does not need to be considered.

(2) When differential design pressure is specified by the user or his designated agent, the design shall be based only on loading case 3. If the tube side is the higher-pressure side, P_t shall be the tube side design pressure and P_s shall be P_t less the differential design pressure. If the shell side is the higher-pressure side, P_s shall be the shell side design pressure and P_t shall be P_s less the differential design pressure.

(3) The designer should take appropriate consideration of the stresses resulting from the pressure test required by UG-99 or UG-100 [see UG-99(d)].

(b) As the calculation procedure is iterative, a value h shall be assumed for the tubesheet thickness to calculate and check that the maximum stresses in tubesheet, shell, and channel are within the maximum permissible stress limits.

(c) The designer may consider the tubesheet as simply supported in accordance with UHX-12.6.

Table UHX-12.4-1

Design Loading Case	Shell Side Design Pressure, P_s	Tube Side Design Pressure, P_t
1	$P_{sd,\min}$	$P_{td,\max}$
2	$P_{sd,\max}$	$P_{td,\min}$
3	$P_{sd,\max}$	$P_{td,\max}$
4	$P_{sd,\min}$	$P_{td,\min}$

UHX-12.5 Calculation Procedure

The procedure for the design of a tubesheet for a U-tube heat exchanger is as follows:

UHX-12.5.1 Step 1. Determine D_o, μ, μ^*, and h'_g from UHX-11.5.1.

UHX-12.5.2 Step 2. Calculate diameter ratios ρ_s and ρ_c. Configurations a, b, and c:

$$\rho_s = \frac{D_s}{D_o}$$

Configurations d, e, and f:

$$\rho_s = \frac{G_s}{D_o}$$

Configurations a, e, and f:

$$\rho_c = \frac{D_c}{D_o}$$

Configurations b, c, and d:

$$\rho_c = \frac{G_c}{D_o}$$

For each loading case, calculate moment M_{TS} due to pressures P_s and P_t acting on the unperforated tubesheet rim.

$$M_{TS} = \frac{{D_o}^2}{16}\left[(\rho_s - 1)(\rho_s^2 + 1)P_s - (\rho_c - 1)(\rho_c^2 + 1)P_t\right]$$

UHX-12.5.3 Step 3. Calculate h/p. If ρ changes, recalculate d^* and μ^* from UHX-11.5.1.

Determine E^*/E and ν^* relative to h/p from UHX-11.5.2.

Configurations a, b, c, e, and f: Proceed to UHX-12.5.4.

Configuration d: Proceed to UHX-12.5.5.

UHX-12.5.4 Step 4. Configurations a, b, and c: Calculate shell coefficients β_s, k_s, λ_s, δ_s, and ω_s.

$$\beta_s = \frac{\sqrt[4]{12(1-\nu_s^2)}}{\sqrt{(D_s + t_s)t_s}}$$

$$k_s = \beta_s \frac{E_s t_s^3}{6(1-\nu_s^2)}$$

$$\lambda_s = \frac{6D_s}{h^3} k_s \left(1 + h\beta_s + \frac{h^2\beta_s^2}{2}\right)$$

$$\delta_s = \frac{D_s^2}{4E_s t_s}\left(1 - \frac{\nu_s}{2}\right)$$

$$\omega_s = \rho_s k_s \beta_s \delta_s (1 + h\beta_s)$$

Configurations a, e, and f: Calculate channel coefficients β_c, k_c, λ_c, δ_c, and ω_c.

$$\beta_c = \frac{\sqrt[4]{12\left(1 - \nu_c^2\right)}}{\sqrt{\left(D_c + t_c\right)t_c}}$$

$$k_c = \beta_c \frac{E_c t_c^3}{6\left(1 - \nu_c^2\right)}$$

$$\lambda_c = \frac{6\,D_c}{h^3} k_c \left(1 + h\beta_c + \frac{h^2\beta_c^2}{2}\right)$$

For a cylinder:

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

For a hemispherical head:

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

$$\omega_c = \rho_c\,k_c\,\beta_c\,\delta_c\left(1 + h\,\beta_c\right)$$

UHX-12.5.5 Step 5. Calculate diameter ratio K.

$$K = \frac{A}{D_o}$$

Calculate coefficient F.
Configuration a:

$$F = \frac{1 - \nu^*}{E^*}\left(\lambda_s + \lambda_c + E\ \ln\ K\right)$$

Configurations b and c:

$$F = \frac{1 - \nu^*}{E^*}\left(\lambda_s + E\ \ln\ K\right)$$

Configuration d:

$$F = \frac{1 - \nu^*}{E^*}\left(E\ \ln\ K\right)$$

Configurations e and f:

$$F = \frac{1 - \nu^*}{E^*}\left(\lambda_c + E\ \ln\ K\right)$$

UHX-12.5.6 Step 6. For each loading case, calculate moment M^* acting on the unperforated tubesheet rim.
Configuration a:

$$M^* = M_{TS} + \omega_c P_t - \omega_s P_s$$

Configuration b:

$$M^* = M_{TS} - \omega_s P_s - \frac{\left(C - G_c\right)}{2\pi\ D_o} W^*$$

Configuration c:

$$M^* = M_{TS} - \omega_s P_s - \frac{\left(G_1 - G_c\right)}{2\pi\ D_o} W^*$$

Configuration d:

$$M^* = M_{TS} + \frac{\left(G_c - G_s\right)}{2\pi\ D_o} W^*$$

Configuration e:

$$M^* = M_{TS} + \omega_c P_t + \frac{\left(C - G_s\right)}{2\pi\ D_o} W^*$$

Configuration f:

$$M^* = M_{TS} + \omega_c P_t + \frac{\left(G_1 - G_s\right)}{2\pi\ D_o} W^*$$

UHX-12.5.7 Step 7. For each loading case, calculate the maximum bending moments acting on the tubesheet at the periphery M_p and at the center M_o.

$$M_p = \frac{M^* - \frac{D_o{}^2}{32} F\left(P_s - P_t\right)}{1 + F}$$

$$M_o = M_p + \frac{D_o{}^2}{64}\left(3 + \nu^*\right)\left(P_s - P_t\right)$$

For each loading case, determine the maximum bending moment M acting on the tubesheet.

$$M = \mathrm{MAX}\left[\left|M_p\right|, \left|M_o\right|\right]$$

UHX-12.5.8 Step 8. For each loading case, calculate the tubesheet bending stress σ.

$$\sigma = \frac{6M}{\mu^* \left(h - h'_g\right)^2}$$

If $\sigma \le 2S$, the assumed tubesheet thickness is acceptable for bending. Otherwise, increase the assumed tubesheet thickness h and return to the step in UHX-12.5.1.

(19) **UHX-12.5.9 Step 9.** For each loading case, calculate the average shear stress in the tubesheet at the outer edge of the perforated region, if required.

(a) If $|P_s - P_t| \le \dfrac{3.2S\mu h}{D_o}$, the shear stress is not required to be calculated. Proceed to the step in (c).

(b) Calculate the average shear stress, τ.

$$\tau = \left(\frac{1}{4\mu}\right)\left(\frac{1}{h}\left\{\frac{4A_p}{C_p}\right\}\right)|P_s - P_t|$$

If $\tau \le \mathrm{MIN}[0.8S, 0.533S_y]$, the assumed tubesheet thickness is acceptable for shear. Otherwise, increase the assumed tubesheet thickness, h, and return to UHX-12.5.1.

(c) Configurations a, b, c, e, and f: Proceed to UHX-12.5.10. Configuration d: The calculation procedure is complete.

UHX-12.5.10 Step 10. For each loading case, calculate the stresses in the shell and/or channel integral with the tubesheet.

Configurations a, b, and c: The shell shall have a uniform thickness of t_s for a minimum length of $1.8\sqrt{D_s t_s}$ adjacent to the tubesheet. Calculate the axial membrane stress $\sigma_{s,m}$, axial bending stress $\sigma_{s,b}$, and total axial stress σ_s, in the shell at its junction to the tubesheet.

$$\sigma_{s,m} = \frac{D_s^2}{4t_s\left(D_s + t_s\right)}P_s$$

$$\sigma_{s,b} = \frac{6}{t_s^2}k_s\left[\beta_s\delta_s P_s + 6\frac{1-\nu^*}{E^*}\frac{D_o}{h^3}\left(1 + \frac{h\beta_s}{2}\right)\times\left(M_p + \frac{D_o^2}{32}\left(P_s - P_t\right)\right)\right]$$

$$\sigma_s = \left|\sigma_{s,m}\right| + \left|\sigma_{s,b}\right|$$

Configurations a, e, and f: A cylindrical channel shall have a uniform thickness of t_c for a minimum length of $1.8\sqrt{D_c t_c}$ adjacent to the tubesheet. Calculate the axial membrane stress $\sigma_{c,m}$, axial bending stress $\sigma_{c,b}$, and total axial stress σ_c, in the channel at its junction to the tubesheet.

$$\sigma_{c,m} = \frac{D_c^2}{4t_c\left(D_c + t_c\right)}P_t$$

$$\sigma_{c,b} = \frac{6}{t_c^2}k_c\left[\beta_c\delta_c P_t - 6\frac{1-\nu^*}{E^*}\frac{D_o}{h^3}\left(1 + \frac{h\beta_c}{2}\right)\times\left(M_p + \frac{D_o^2}{32}\left(P_s - P_t\right)\right)\right]$$

$$\sigma_c = \left|\sigma_{c,m}\right| + \left|\sigma_{c,b}\right|$$

Configuration a: If $\sigma_s \le 1.5S_s$ and $\sigma_c \le 1.5S_c$, the shell and channel designs are acceptable and the calculation procedure is complete. Otherwise, proceed to UHX-12.5.11.

Configurations b and c: If $\sigma_s \le 1.5S_s$, the shell design is acceptable and the calculation procedure is complete. Otherwise, proceed to UHX-12.5.11.

Configurations e and f: If $\sigma_c \le 1.5S_c$, the channel design is acceptable and the calculation procedure is complete. Otherwise, proceed to UHX-12.5.11.

UHX-12.5.11 Step 11. The design shall be reconsidered. One or a combination of the following three options may be used.

Option 1. Increase the assumed tubesheet thickness h and return to UHX-12.5.1.

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

Configurations a, b, and c: If $\sigma_s > 1.5S_s$, increase the shell thickness t_s.

Configurations a, e, and f: If $\sigma_c > 1.5S_c$ increase the channel thickness t_c.

Return to UHX-12.5.1.

Option 3. Perform a simplified elastic–plastic calculation for each applicable loading case by using a reduced effective modulus for the integral shell and/or channel to reflect the anticipated load shift resulting from plastic action at the integral shell and/or channel-to-tubesheet junction. This may result in a higher tubesheet bending stress σ. This option shall not be used at temperatures where the time-dependent properties govern the allowable stress.

Configuration a: This option may only be used when $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$. In UHX-12.5.4, if $\sigma_s > 1.5\ S_s$, replace E_s with $E_s^* = E_s\sqrt{1.5S_s/\sigma_s}$ and recalculate k_s and λ_s. If $\sigma_c > 1.5\ S_c$, replace E_c with $E_c^* = E_c\sqrt{1.5S_c/\sigma_c}$ and recalculate k_c and λ_c.

Configurations b and c: This option may only be used when $\sigma_s \le S_{PS,s}$. In UHX-12.5.4, replace E_s with $E_s^* = E_s\sqrt{1.5S_s/\sigma_s}$ and recalculate k_s and λ_s.

Configurations e and f: This option may only be used when $\sigma_c \le S_{PS,c}$. In UHX-12.5.4, replace E_c with $E_c^* = E_c\sqrt{1.5S_c/\sigma_c}$ and recalculate k_c and λ_c.

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

(1) UHX-12.5.4 (Step 4):

Configurations a, b, and c: $\beta_s = 0$, $k_s = 0$, $\lambda_s = 0$, $\delta_s = 0$. Configurations a, e, and f: $\beta_c = 0$, $k_c = 0$, $\lambda_c = 0$, $\delta_c = 0$.

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

UHX-13 RULES FOR THE DESIGN OF FIXED TUBESHEETS

UHX-13.1 Scope

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

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

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

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

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

UHX-13.2 Conditions of Applicability

The two tubesheets shall have the same thickness, material and edge conditions.

UHX-13.3 Nomenclature

The symbols described below are used for the design of the tubesheets. Symbols D_o, E^*, h'_g, μ, μ* and ν^* are defined in UHX-11.

A = outside diameter of tubesheet, except as limited by UHX-10(b)
a_c = radial channel dimension
Configuration a: $a_c = D_c/2$
Configurations b, c, and d: $a_c = G_c/2$
a_o = equivalent radius of outer tube limit circle
A_P = total area enclosed by C_P
a_s = radial shell dimension
Configurations a, b, and c: $a_s = D_s/2$
Configuration d: $a_s = G_s/2$
C = bolt circle diameter (see Mandatory Appendix 2)
C_P = perimeter of the tube layout measured stepwise in increments of one tube pitch from the center-to-center of the outermost tubes (see Figure UHX-12.2)
D_c = inside channel diameter
D_J = inside diameter of the expansion joint at its convolution height
D_s = inside shell diameter
d_t = nominal outside diameter of tubes
E = modulus of elasticity for tubesheet material at T
E_c = modulus of elasticity for channel material at T_c
E_s = modulus of elasticity for shell material at T_s
$E_{s,w}$ = joint efficiency (longitudinal stress) for shell
E_t = modulus of elasticity for tube material at T_t
G_1 = midpoint of contact between flange and tubesheet
G_c = diameter of channel gasket load reaction (see Mandatory Appendix 2)
G_s = diameter of shell gasket load reaction (see Mandatory Appendix 2)
h = tubesheet thickness
J = ratio of expansion joint to shell axial rigidity (J = 1.0 if no expansion joint)

k = constant accounting for the method of support for the unsupported tube span under consideration
= 0.6 for unsupported spans between two tubesheets
= 0.8 for unsupported spans between a tubesheet and a tube support
= 1.0 for unsupported spans between two tube supports

K_J = axial rigidity of expansion joint, total force/elongation

L = tube length between inner tubesheet faces
= $L_t - 2h$

L_t = tube length between outer tubesheet faces

MAX *[(a), (b),(c),...]* = greatest of *a, b, c*,...

N_t = number of tubes

P_e = effective pressure acting on tubesheet

P_s = shell side design or operating pressure, as applicable. For shell side vacuum, use a negative value for P_s

$P_{sd,\max}$ = maximum shell side design pressure

$P_{sd,\min}$ = minimum shell side design pressure (negative if vacuum is specified, otherwise zero)

$P_{sox,\max}$ = max.(0, maximum shell side operating pressure for operating condition x)

$P_{sox,\min}$ = min.(0, minimum shell side operating pressure for operating condition x)

P_t = tube side design or operating pressure, as applicable. For tube side vacuum, use a negative value for P_t

$P_{td,\max}$ = maximum tube side design pressure

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

$P_{tox,\max}$ = max.(0, maximum tube side operating pressure for operating condition x)

$P_{tox,\min}$ = min.(0, minimum tube side operating pressure for operating condition x)

S = allowable stress for tubesheet material at T

S_c = allowable stress for channel material at T_c

S_{PS} = allowable primary plus secondary stress for tubesheet material at T per UG-23(e)

$S_{PS,c}$ = allowable primary plus secondary stress for channel material at T_c per UG-23(e)

$S_{PS,s}$ = allowable primary plus secondary stress for shell material at T_s per UG-23(e)

S_s = allowable stress for shell material at T_s

$S_{s,b}$ = maximum allowable longitudinal compressive stress in accordance with UG-23(b) for the shell

S_t = allowable stress for tube material at T_t

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

S_y = yield strength for tubesheet material at T

$S_{y,c}$ = yield strength for channel material at T_c

$S_{y,s}$ = yield strength for shell material at T_s

$S_{y,t}$ = yield strength for tube material at T_t

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

T = tubesheet design temperature for the design condition or operating metal temperature for operating condition x, as applicable [see UHX-13.4(b)]

T_c = channel design temperature for the design condition or operating metal temperature for operating condition x, as applicable [see UHX-13.4(b)]

t_c = channel thickness

T_s = shell design temperature for the design condition or operating metal temperature for operating condition x, as applicable [see UHX-13.4(b)]

t_s = shell thickness

$T_{s,m}$ = mean shell metal temperature along shell length

$T_{s,mx}$ = shell axial mean metal temperature for operating condition x, as applicable

T_t = tube design temperature for the design condition or operating metal temperature for operating condition x, as applicable [see UHX-13.4(b)]

t_t = nominal tube wall thickness

$T_{t,m}$ = mean tube metal temperature along tube length

$T_{t,mx}$ = tube axial mean metal temperature for operating condition x, as applicable

W_t = tube-to-tubesheet joint load

W^* = tubesheet effective bolt load determined in accordance with UHX-8

x = 1, 2, 3, ..., n, integer denoting applicable operating condition under consideration (e.g., normal operating, start-up, shutdown, cleaning, upset)

ℓ = unsupported tube span under consideration

$\alpha_{s,m}$ = mean coefficient of thermal expansion of shell material at $T_{s,m}$

$\alpha_{t,m}$ = mean coefficient of thermal expansion of tube material at $T_{t,m}$

γ = axial differential thermal expansion between tubes and shell

Δ_J = axial displacement over the length of the thin-walled bellows element (see UHX-16)

Δ_s = shell axial displacement over the length between the inner tubesheet faces, L [see UHX-17(c)]

ν = Poisson's ratio of tubesheet material

ν_c = Poisson's ratio of channel material

ν_s = Poisson's ratio of shell material

ν_t = Poisson's ratio of tube material

UHX-13.4 Design Considerations

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

The user or his designated agent shall specify all the design and operating conditions that govern the design of the main components of the heat exchanger (i.e., tubesheets, tubes, shell, channel, tube-to-tubesheet joint). These shall include, but not be limited to, normal operating, start-up, shutdown, cleaning, and upset conditions.

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

(1) Design Loading Cases. Table UHX-13.4-1 provides the load combinations required to evaluate the heat exchanger for the design condition. When $P_{sd,\min}$ and $P_{td,\min}$ are both zero, design loading case 4 does not need to be considered.

(2) Operating Loading Cases. Table UHX-13.4-2 provides the load combinations required to evaluate the heat exchanger for each operating condition x.

(3) When differential pressure design is specified by the user or his designated agent, the design shall be based only on design loading case 3 and operating loading cases 3 and 4 for each specified operating condition. If the tube side is the higher-pressure side, P_t shall be the tube side design pressure and P_s shall be P_t less the differential design pressure. If the shell side is the higher-pressure side, P_s shall be the shell side design pressure and P_t shall be the P_s less the differential pressure. For the operating loading cases, the differential pressure and the individual operating pressures shall not exceed the values used for design.

(4) The designer should take appropriate consideration of the stresses resulting from the pressure test required by UG-99 or UG-100 [see UG-99(d)].

(b) The elastic moduli, yield strengths, and allowable stresses shall be taken at the design temperatures for the design loading cases and may be taken at the operating metal temperature of the component under consideration for operating condition x.

(c) As the calculation procedure is iterative, a value h shall be assumed for the tubesheet thickness to calculate and check that the maximum stresses in tubesheet, tubes, shell, and channel are within the maximum permissible stress limits, and that the resulting tube-to-tubesheet joint load is acceptable.

Because any increase of tubesheet thickness may lead to overstresses in the tubes, shell, channel, or tube-to-tubesheet joint, a final check shall be performed, using in the equations the nominal thickness of tubesheet, tubes, shell, and channel, in both corroded and uncorroded conditions.

(d) The designer shall consider the effect of radial differential thermal expansion between the tubesheet and integral shell or channel (configurations a, b, and c) in accordance with UHX-13.8, if required by UHX-13.8.1.

(e) The designer may consider the tubesheet as simply supported in accordance with UHX-13.9.

UHX-13.5 Calculation Procedure

The procedure for the design of tubesheets for a fixed tubesheet heat exchanger is as follows.

UHX-13.5.1 Step 1. Determine D_o, μ, μ^*, and h'_g from UHX-11.5.1.

Operating loading cases: $h'_g = 0$

Calculate a_o, ρ_s, ρ_c, x_s, and x_t.

$$a_o = \frac{D_o}{2}$$

$$\rho_s = \frac{a_s}{a_o}$$

$$\rho_c = \frac{a_c}{a_o}$$

$$x_s = 1 - N_t\left(\frac{d_t}{2a_o}\right)^2$$

$$x_t = 1 - N_t\left(\frac{d_t - 2t_t}{2a_o}\right)^2$$

UHX-13.5.2 Step 2. Calculate the shell axial stiffness K_s, tube axial stiffness K_t, and stiffness factors $K_{s,t}$ and J.

$$K_s = \frac{\pi\, t_s(D_s + t_s)E_s}{L}$$

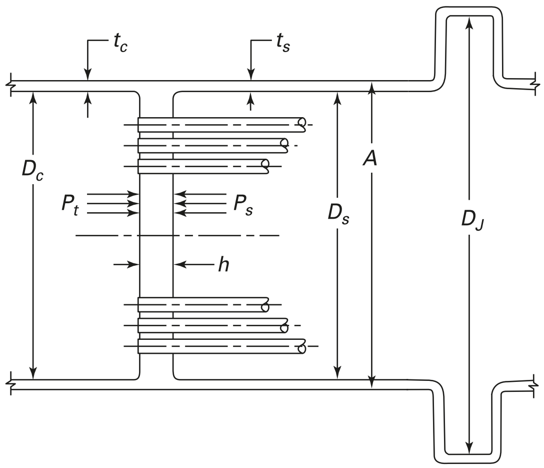
$$K_t = \frac{\pi\, t_t(d_t - t_t)E_t}{L}$$

$$K_{s,t} = \frac{K_s}{N_t K_t}$$

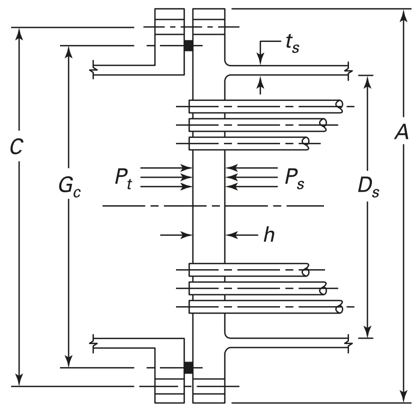
$$J = \frac{1}{1 + \dfrac{K_s}{K_J}}$$

Calculate shell coefficients β_s, k_s, λ_s, and δ_s.

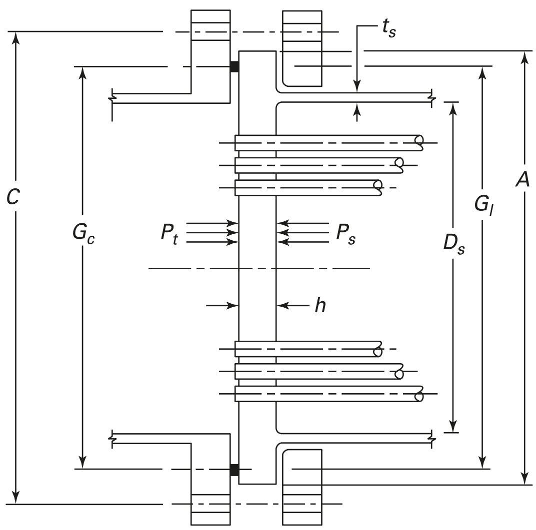
Figure UHX-13.1
Fixed Tubesheet Configurations



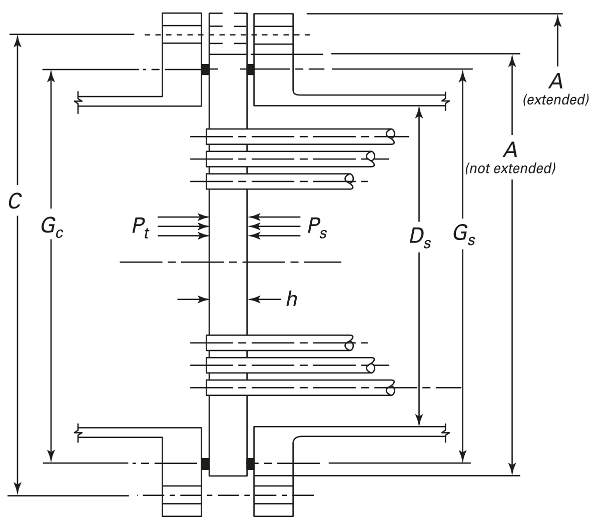
(a) Configuration a:
Tubesheet Integral With Shell and Channel



(b) Configuration b:
Tubesheet Integral With Shell and Gasketed With Channel, Extended as a Flange



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



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

GENERAL NOTE: The expansion joint detail in Configuration a applies to bellows, flanged-and-flued, and flanged-only expansion joints for Configurations a, b, c, and d.

Table UHX-13.1 Formulas for Determination of Z_d, Z_v, Z_m, Z_w, and F_m

(19)

(1) Calculate Kelvin functions of order 0 relative to x, where x varies from 0 to X_a such that $0 < x \le X_a$ [Note (1)]:

$$\mathrm{ber}(x) = \sum_{n=0}^{n=m-1} (-1)^n \frac{(x/2)^{4n}}{[(2n)!]^2} = 1 - \frac{(x/2)^4}{(2!)^2} + \frac{(x/2)^8}{(4!)^2} - \frac{(x/2)^{12}}{(6!)^2} + \ldots$$

$$\mathrm{bei}(x) = \sum_{n=1}^{n=m} (-1)^{n-1} \frac{(x/2)^{4n-2}}{[(2n-1)!]^2} = \frac{(x/2)^2}{(1!)^2} - \frac{(x/2)^6}{(3!)^2} + \frac{(x/2)^{10}}{(5!)^2} - \ldots$$

and their derivatives:

$$\mathrm{ber}'(x) = \sum_{n=1}^{n=m} (-1)^n \frac{(2n)(x/2)^{4n-1}}{[(2n)!]^2} = -\frac{2(x/2)^3}{(2!)^2} + \frac{4(x/2)^7}{(4!)^2} - \frac{6(x/2)^{11}}{(6!)^2} + \ldots$$

$$\mathrm{bei}'(x) = \sum_{n=1}^{n=m} (-1)^{n-1} \frac{(2n-1)(x/2)^{4n-3}}{[(2n-1)!]^2} = \frac{(x/2)^1}{(1!)^2} - \frac{3(x/2)^5}{(3!)^2} + \frac{5(x/2)^9}{(5!)^2} - \ldots$$

(2) Calculate functions $\psi_1(x)$ and $\psi_2(x)$ relative to x:

$$\psi_1(x) = \mathrm{bei}(x) + \frac{1-\nu^*}{x} \cdot \mathrm{ber}'(x)$$

$$\psi_2(x) = \mathrm{ber}(x) - \frac{1-\nu^*}{x} \cdot \mathrm{bei}'(x)$$

(3) Calculate Z_a, Z_d, Z_v, Z_w, and Z_m relative to X_a:

$$Z_a = \mathrm{bei}'(X_a) \cdot \psi_2(X_a) - \mathrm{ber}'(X_a) \cdot \psi_1(X_a)$$

$$Z_d = \frac{\mathrm{ber}(X_a) \cdot \psi_2(X_a) + \mathrm{bei}(X_a) \cdot \psi_1(X_a)}{X_a^3 \cdot Z_a}$$

$$Z_v = \frac{\mathrm{ber}'(X_a) \cdot \psi_2(X_a) + \mathrm{bei}'(X_a) \cdot \psi_1(X_a)}{X_a^2 \cdot Z_a}$$

$$Z_w = \frac{\mathrm{ber}'(X_a) \cdot \mathrm{ber}(X_a) + \mathrm{bei}'(X_a) \cdot \mathrm{bei}(X_a)}{X_a^2 \cdot Z_a}$$

$$Z_m = \frac{\mathrm{ber}'^2(X_a) + \mathrm{bei}'^2(X_a)}{X_a \cdot Z_a}$$

(4) Calculate functions $Q_m(x)$ and $Q_v(x)$ relative to x:

$$Q_m(x) = \frac{\mathrm{bei}'(X_a) \cdot \psi_2(x) - \mathrm{ber}'(X_a) \cdot \psi_1(x)}{Z_a}$$

$$Q_v(x) = \frac{\psi_1(X_a) \cdot \psi_2(x) - \psi_2(X_a) \cdot \psi_1(x)}{X_a \cdot Z_a}$$

(5) For each loading case, calculate $F_m(x)$ relative to x:

$$F_m(x) = \frac{Q_v(x) + Q_3 \cdot Q_m(x)}{2}$$

(6) F_m is the maximum of the absolute value of $F_m(x)$ when x varies from 0 to X_a such that $0 < x \le X_a$:

$$F_m = \mathrm{MAX}\,\left|F_m(x)\right|$$

NOTE:
(1) Use $m = 4 + X_a/2$ (rounded to the nearest integer) to obtain an adequate approximation of the Kelvin functions and their derivatives.

Table UHX-13.2
Formulas for Determination of $F_{t,\min}$ and $F_{t,\max}$

Step No.	Description
1	Follow steps (1), (2), and (3) in Table UHX-13.1.
2	Calculate functions $Z_d(x)$ and $Z_w(x)$ relative to x: $$Z_d(x) = \frac{\psi_2(X_a)\cdot \mathrm{ber}(x) + \psi_1(X_a)\cdot \mathrm{bei}(x)}{X_a^3 \cdot Z_a}$$ $$Z_w(x) = \frac{\mathrm{ber}'(X_a)\cdot \mathrm{ber}(x) + \mathrm{bei}'(X_a)\cdot \mathrm{bei}(x)}{X_a^2 \cdot Z_a}$$
3	For each loading case, calculate $F_t(x)$ relative to x in accordance with a or b below. *(a)* When $P_e \neq 0$ $$F_t(x) = \left[Z_d(x) + Q_3 \cdot Z_w(x)\right]\cdot \frac{X_a^4}{2}$$ *(b)* When $P_e = 0$ $$F_t(x) = Z_w(x)\cdot \frac{X_a^4}{2}$$
4	Calculate the minimum and maximum values, $F_{t,\min}$ and $F_{t,\max}$, of $F_t(x)$ when x varies from 0 to X_a such that $0 \le x \le X_a$. $F_{t,\min}$ and $F_{t,\max}$ may be positive or negative. $$F_{t,\min} = \mathrm{MIN}\left[F_t(x)\right]$$ $$F_{t,\max} = \mathrm{MAX}\left[F_t(x)\right]$$
	When $P_e \neq 0$, see Figures LL-1 and LL-2 in Nonmandatory Appendix LL for a graphical representation of $F_{t,\min}$ and $F_{t,\max}$.

Configurations a, b, and c:

$$\beta_s = \frac{\sqrt[4]{12\left(1-\nu_s^2\right)}}{\sqrt{\left(D_s + t_s\right)t_s}}$$

$$k_s = \beta_s \frac{E_s t_s^3}{6\left(1-\nu_s^2\right)}$$

$$\lambda_s = \frac{6D_s}{h^3} k_s \left(1 + h\beta_s + \frac{h^2\beta_s^2}{2}\right)$$

$$\delta_s = \frac{D_s^2}{4E_s t_s}\left(1 - \frac{\nu_s}{2}\right)$$

Configuration d: $\beta_s = 0$, $k_s = 0$, $\lambda_s = 0$, $\delta_s = 0$
Calculate channel coefficients β_c, k_c, λ_c, and δ_c.

Configuration a:

$$\beta_c = \frac{\sqrt[4]{12\left(1-\nu_c^2\right)}}{\sqrt{\left(D_c + t_c\right)t_c}}$$

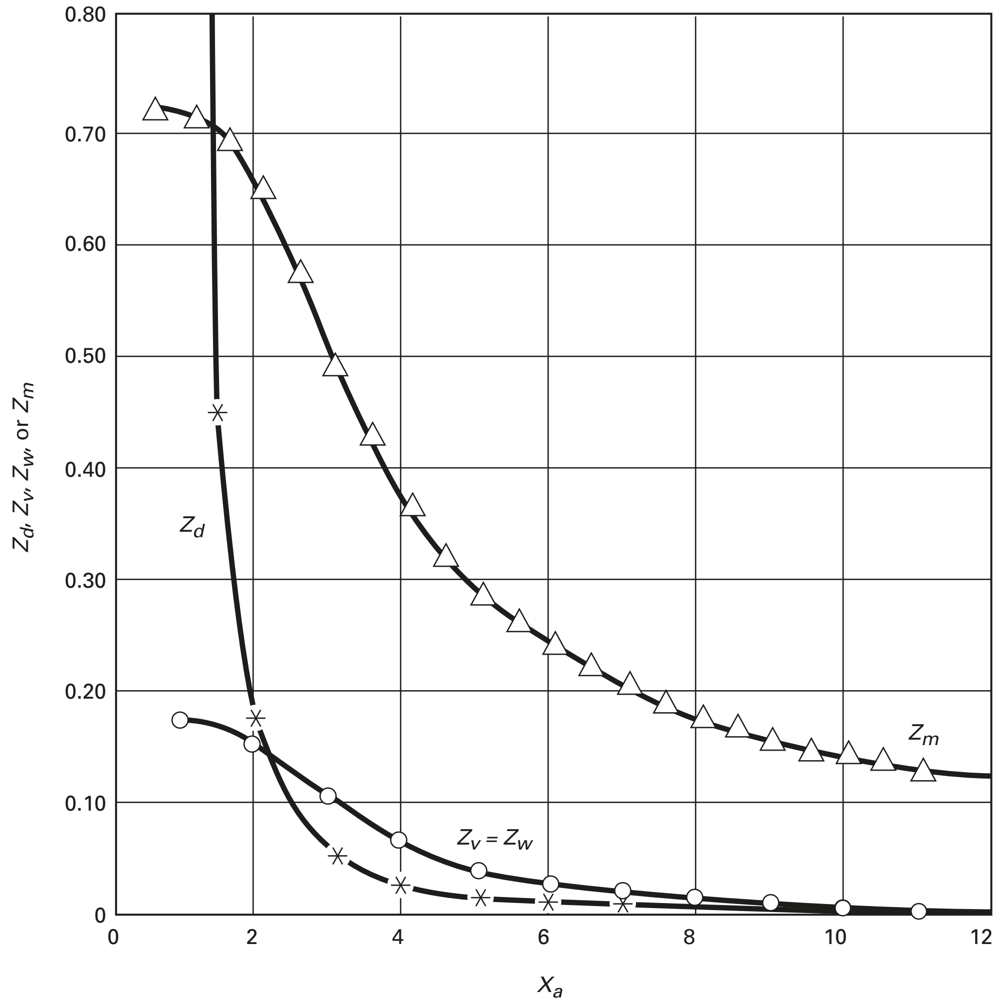
$$k_c = \beta_c \frac{E_c t_c^3}{6\left(1-\nu_c^2\right)}$$

$$\lambda_c = \frac{6D_c}{h^3} k_c \left(1 + h\beta_c + \frac{h^2\beta_c^2}{2}\right)$$

For a cylinder:

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

Figure UHX-13.2
Z_d, Z_v, Z_w, and Z_m Versus X_a



GENERAL NOTES:
(a) Curves giving Z_d, Z_v, Z_w, or Z_m are valid for $\nu^* = 0.4$. They are sufficiently accurate to be used for other values of ν^*.
(b) For $X_a > 12.0$, see Table UHX-13.1.

Table UHX-13.4-1

Design Loading Case	Shell Side Design Pressure, P_s	Tube Side Design Pressure, P_t
1	$P_{sd,\min}$	$P_{td,\max}$
2	$P_{sd,\max}$	$P_{td,\min}$
3	$P_{sd,\max}$	$P_{td,\max}$
4	$P_{sd,\min}$	$P_{td,\min}$

For a hemispherical head:

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

Configurations b, c, d: $\beta_c = 0$, $k_c = 0$, $\lambda_c = 0$, $\delta_c = 0$

UHX-13.5.3 Step 3. Calculate h/p. If ρ changes, recalculate d^* and μ^* from UHX-11.5.1.

Determine E^*/E and ν^* relative to h/p from UHX-11.5.2.

Calculate X_a.

Table UHX-13.4-2

Operating Loading Case	Operating Pressure		Axial Mean Metal Temperature	
	Shell Side, P_s	Tube Side, P_t	Tubes, $T_{t,m}$	Shell, $T_{s,m}$
1	$P_{sox,\min}$	$P_{tox,\max}$	$T_{t,mx}$	$T_{s,mx}$
2	$P_{sox,\max}$	$P_{tox,\min}$	$T_{t,mx}$	$T_{s,mx}$
3	$P_{sox,\max}$	$P_{tox,\max}$	$T_{t,mx}$	$T_{s,mx}$
4	$P_{sox,\min}$	$P_{tox,\min}$	$T_{t,mx}$	$T_{s,mx}$

$$X_a = \left[24\left(1 - \nu^{*2}\right)N_t \frac{E_t t_t\left(d_t - t_t\right)a_o^2}{E^* L h^3}\right]^{\frac{1}{4}}$$

Using the calculated value of X_a, enter either Table UHX-13.1 or Figure UHX-13.2 to determine Z_d, Z_v, Z_w, and Z_m.

UHX-13.5.4 Step 4. Calculate diameter ratio K and coefficient F.

$$K = \frac{A}{D_o}$$

$$F = \frac{1 - \nu^*}{E^*}\left(\lambda_s + \lambda_c + E \ln K\right)$$

Calculate Φ, Q_1, Q_{Z1}, Q_{Z2}, and U.

$$\Phi = \left(1 + \nu^*\right)F$$

$$Q_1 = \frac{\rho_s - 1 - \Phi Z_v}{1 + \Phi Z_m}$$

$$Q_{Z1} = \frac{\left(Z_d + Q_1 Z_w\right)X_a^4}{2}$$

$$Q_{Z2} = \frac{\left(Z_v + Q_1 Z_m\right)X_a^4}{2}$$

$$U = \frac{\left[Z_w + \left(\rho_s - 1\right)Z_m\right]X_a^4}{1 + \Phi Z_m}$$

UHX-13.5.5 Step 5.

(a) Calculate γ.

Design loading cases: $\gamma = 0$.

Operating loading cases:

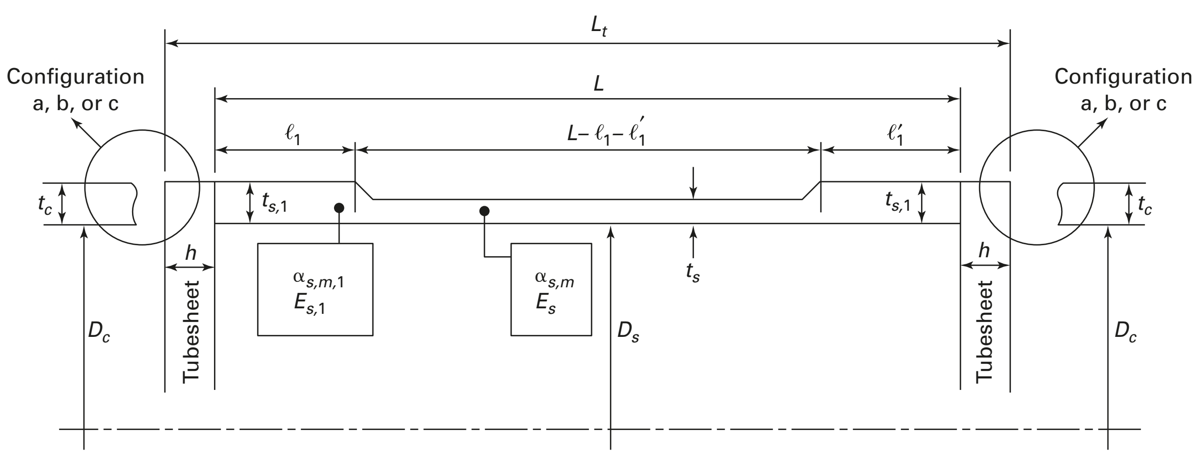
$$\gamma = \left[\alpha_{t,m}\left(T_{t,m} - T_a\right) - \alpha_{s,m}\left(T_{s,m} - T_a\right)\right]L$$

(b) Calculate ω_s, ω_s^*, and ω_c, ω_c^*.

$$\omega_s = \rho_s k_s \beta_s \delta_s\left(1 + h\beta_s\right)$$

(19)

Figure UHX-13.4
Different Shell Thicknesses and/or Material Adjacent to the Tubesheets



$$\omega_s^* = a_o^2 \frac{\left(\rho_s^2 - 1\right)\left(\rho_s - 1\right)}{4} - \omega_s$$

$$\omega_c = \rho_c k_c \beta_c \delta_c \left(1 + h\beta_c\right)$$

$$\omega_c^* = a_o^2 \left[\frac{\left(\rho_c^2 + 1\right)\left(\rho_c - 1\right)}{4} - \frac{\left(\rho_s - 1\right)}{2} \right] - \omega_c$$

(c) Calculate γ_b.
Configuration a:

$$\gamma_b = 0$$

Configuration b:

$$\gamma_b = \frac{G_c - C}{D_o}$$

Configuration c:

$$\gamma_b = \frac{G_c - G_1}{D_o}$$

Configuration d:

$$\gamma_b = \frac{G_c - G_s}{D_o}$$

UHX-13.5.6 Step 6. For each loading case, calculate P_s', P_t', P_γ, P_W, P_{rim}, and effective pressure P_e.

$$P_s' = \left(x_s + 2\left(1 - x_s\right)\nu_t + \frac{2}{K_{s,t}}\left(\frac{D_s}{D_o}\right)^2 \nu_s - \frac{\rho_s^2 - 1}{J K_{s,t}} - \frac{\left(1 - J\right)}{2 J K_{s,t}} \frac{\left[D^2 J - \left(D_s\right)^2\right]}{D_o^2} \right) P_s$$

$$P_t' = \left(x_t + 2\left(1 - x_t\right)\nu_t + \frac{1}{J K_{s,t}} \right) P_t$$

$$P_\gamma = \frac{N_t K_t}{\pi a_o^2} \gamma$$

$$P_W = -\frac{U}{a_o^2} \frac{\gamma_b}{2\pi} W^*$$

$$P_{\text{rim}} = -\frac{U}{a_o^2}\left(\omega_s^* P_s - \omega_c^* P_t\right)$$

$$P_e = \frac{J K_{s,t}}{1 + J K_{s,t}\left[Q_{Z1} + \left(\rho_s - 1\right) Q_{Z2}\right]} \times \left(P_s' - P_t' + P_\gamma + P_W + P_{\text{rim}}\right)$$

UHX-13.5.7 Step 7. For each loading case, calculate Q_2.

$$Q_2 = \frac{\left(\omega_s^* P_s - \omega_c^* P_t\right) + \frac{\gamma_b}{2\pi} W^*}{1 + \Phi Z_m}$$

For each loading case, calculate the maximum bending stress in the tubesheet in accordance with (a) or (b) below.

(a) When $P_e \neq 0$:

(1) Calculate Q_3.

$$Q_3 = Q_1 + \frac{2 Q_2}{P_e a_o^2}$$

(2) For each loading case, determine coefficient F_m from either Table UHX-13.1 or Figures UHX-13.5.7-1 and UHX-13.5.7-2 and calculate the maximum bending stress σ.

$$\sigma = \left(\frac{1.5 F_m}{\mu^*}\right)\left(\frac{2 a_o}{h - h_g'}\right)^2 P_e$$

(b) When $P_e = 0$, calculate the maximum bending stress σ.

$$\sigma = \frac{6 Q_2}{\mu^* \left(h - h_g'\right)^2}$$

For the design loading cases, if $|\sigma| \leq 1.5S$, and for the operating loading cases, if $|\sigma| \leq S_{PS}$, the assumed tubesheet thickness is acceptable for bending. Otherwise, increase the assumed tubesheet thickness h and return to the step in UHX-13.5.1.

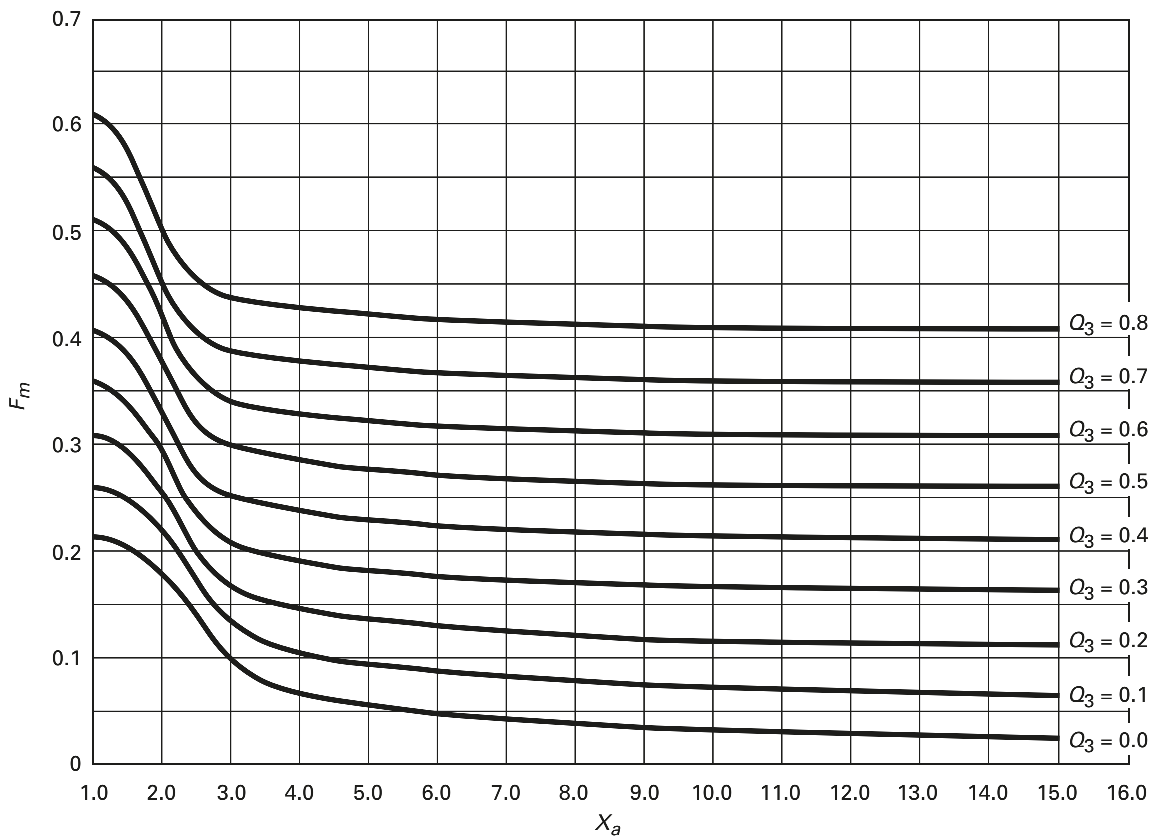
UHX-13.5.8 Step 8. For each loading case, calculate the average shear stress in the tubesheet at the outer edge of the perforated region, if required. (19)

(a) If $\left|P_e\right| \leq \dfrac{1.6 S \mu h}{a_o}$, the shear stress is not required to be calculated. Proceed to UHX-13.5.9.

(b) Calculate the average shear stress, τ.

(19)

Figure UHX-13.5.7-1
F_m Versus X_a ($0.0 \le Q_3 \le 0.8$)



GENERAL NOTES:
(a) Curves giving F_m are valid for ν^*= 0.4. They are sufficiently accurate to be used for other values of ν^*.
(b) For values of X_a and Q_3 beyond those given by the curves, see Table UHX-13.1.

$$\tau = \left(\frac{1}{4\mu}\right)\left[\frac{1}{h}\left(\frac{4A_p}{C_p}\right)\right]P_e$$

If $|\tau| \le$ MIN[$0.8S$, $0.533S_y$], the assumed tubesheet thickness is acceptable for shear. Otherwise, increase the assumed tubesheet thickness, h, and return to UHX-13.5.1.

UHX-13.5.9 Step 9. Perform this step for each loading case.

(a) Check the axial tube stress.

(1) For each loading case, determine coefficients $F_{t,\min}$ and $F_{t,\max}$ from Table UHX-13.2 and calculate the two extreme values of tube stress, $\sigma_{t,1}$ and $\sigma_{t,2}$. The values for $\sigma_{t,1}$ and $\sigma_{t,2}$ may be positive or negative.

(-a) When $P_e \neq 0$:

$$\sigma_{t,1} = \frac{1}{x_t - x_s}\left[\left(P_s x_s - P_t x_t\right) - P_e F_{t,\min}\right]$$

$$\sigma_{t,2} = \frac{1}{x_t - x_s}\left[\left(P_s x_s - P_t x_t\right) - P_e F_{t,\max}\right]$$

(-b) When $P_e = 0$:

$$\sigma_{t,1} = \frac{1}{x_t - x_s}\left[\left(P_s x_s - P_t x_t\right) - \frac{2Q_2}{a_o^2}F_{t,\min}\right]$$

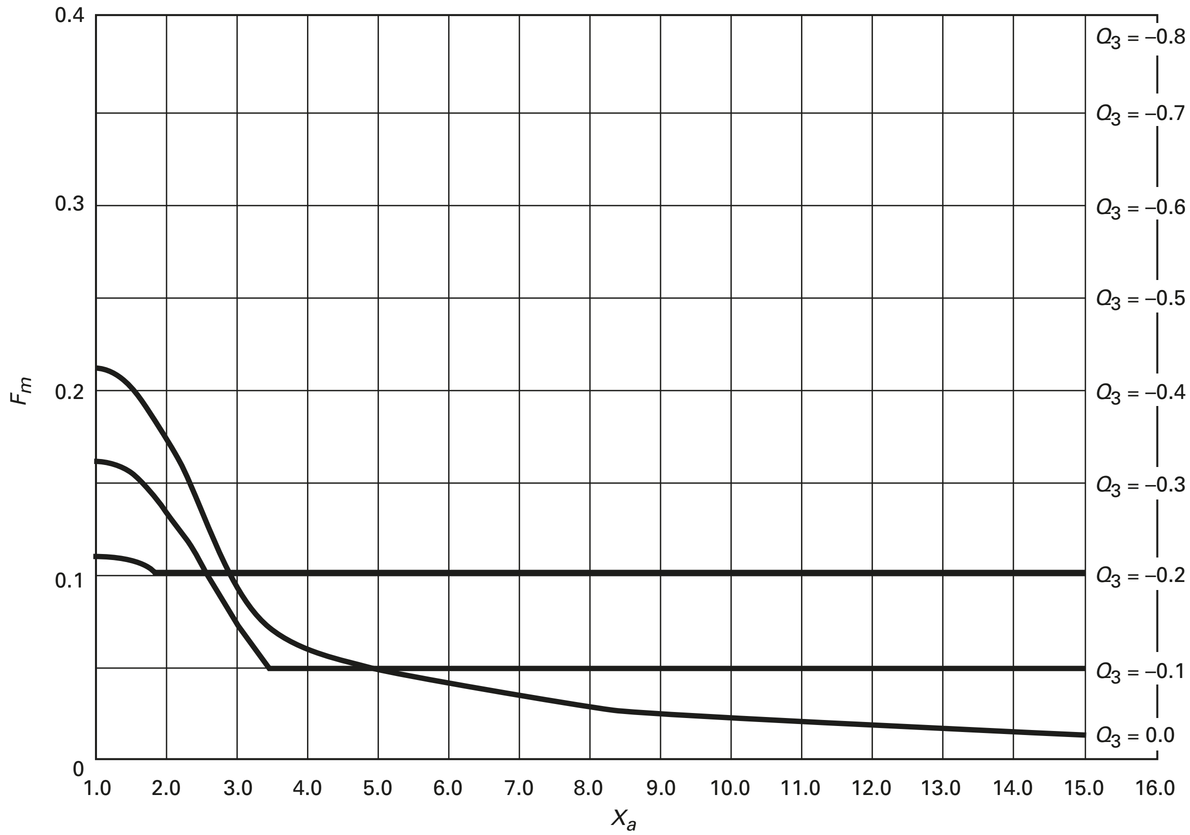
$$\sigma_{t,2} = \frac{1}{x_t - x_s}\left[\left(P_s x_s - P_t x_t\right) - \frac{2Q_2}{a_o^2}F_{t,\max}\right]$$

(2) Determine $\sigma_{t,\max}$ = MAX ($|\sigma_{t,1}|$, $|\sigma_{t,2}|$).

For the design loading cases, if $\sigma_{t,\max} > S_t$, and for the operating loading cases, if $\sigma_{t,\max} > 2S_t$, reconsider the tube design and return to the step in UHX-13.5.1.

(b) Check the tube-to-tubesheet joint design.

Figure UHX-13.5.7-2
F_m Versus X_a ($-0.8 \leq Q_3 \leq 0.0$) (19)



GENERAL NOTES:
(a) Curves giving F_m are valid for $\nu^* = 0.4$. They are sufficiently accurate to be used for other values of ν^*.
(b) For values of X_a and Q_3 beyond those given by the curves, see Table UHX-13.1.

(1) Calculate the largest tube-to-tubesheet joint load, W_t.

$$W_t = \sigma_{t,\max} \pi (d_t - t_t) t_t$$

(2) Determine the maximum allowable load for the tube-to-tubesheet joint design, L_{max}. For tube-to-tubesheet joints with full strength welds, L_{max} shall be determined in accordance with UW-20. For tube-to-tubesheet joints with partial strength welds, L_{max} shall be determined in accordance with UW-20, UW-18(d), or Nonmandatory Appendix A, as applicable. For all other tube joints, L_{max} shall be determined in accordance with Nonmandatory Appendix A.

If $W_t > L_{max}$, reconsider the tube-to-tubesheet joint design.

If $W_t \leq L_{max}$, tube-to-tubesheet joint design is acceptable.

If $\sigma_{t,1}$ or $\sigma_{t,2}$ is negative, proceed to (c) below.

If $\sigma_{t,1}$ and $\sigma_{t,2}$ are positive, the tube design is acceptable. Proceed to the step in UHX-13.5.10.

(c) Check the tubes for buckling.

(1) Calculate the largest equivalent unsupported buckling length of the tube ℓ_t considering the unsupported tube spans ℓ and their corresponding method of support k.

$$\ell_t = k\ell$$

(2) Calculate r_t, F_t, and C_t.

$$r_t = \frac{\sqrt{d_t^2 + (d_t - 2t_t)^2}}{4}$$

$$F_t = \frac{\ell_t}{r_t}$$

$$C_t = \sqrt{\frac{2\pi^2 E_t}{S_{y,t}}}$$

(3) Determine the factor of safety F_s in accordance with (-a) or (-b) below:

(-a) When $P_e \neq 0$,

$$F_s = \text{MAX}\left\{\left[3.25\text{-}0.25(Z_d + Q_3 Z_w)X_a^{\,4}\right], [1.25]\right\}$$

F_s need not be taken greater than 2.0.

(-b) When $P_e = 0$, $F_s = 1.25$.

(4) Determine the maximum permissible buckling stress limit S_{tb} for the tubes in accordance with (-a) or (-b) below:

(-a) When $C_t \leq F_t$

$$S_{tb} = \text{MIN}\left\{\left[\frac{1}{F_s}\frac{\pi^2 E_t}{F_t^2}\right], [S_t]\right\}$$

(-b) When $C_t > F_t$

$$S_{tb} = \text{MIN}\left\{\left[\frac{S_{y,t}}{F_s}\left(1 - \frac{F_t}{2C_t}\right)\right], [S_t]\right\}$$

(5) Determine $\sigma_{t,\min} = \text{MIN}(\sigma_{t,1}, \sigma_{t,2})$.

If $|\sigma_{t,\min}| > S_{tb}$, reconsider the tube design and return to the step in UHX-13.5.1.

If $|\sigma_{t,\min}| \leq S_{tb}$, the tube design is acceptable. Proceed to the step in UHX-13.5.10.

UHX-13.5.10 Step 10. Perform this step for each loading case.

(a) Calculate the axial membrane stress, $\sigma_{s,m}$, in each different shell section. For shell sections integral with the tubesheet having a different material and/or thickness than the shell, refer to UHX-13.6 for the nomenclature.

$$\sigma_{s,m} = \frac{a_o^2}{t_s(D_s + t_s)}\left[P_e + \left(\rho_s^2 - 1\right)(P_s - P_t)\right] + \frac{a_s^2}{t_s(D_s + t_s)}P_t$$

For the design loading cases, if $|\sigma_{s,m}| > S_s E_{s,w}$, and for the operating loading cases, if $|\sigma_{s,m}| > S_{PS,s}$, reconsider the shell design and return to the step in UHX-13.5.1.

If $\sigma_{s,m}$ is negative, proceed to (b) below.

If $\sigma_{s,m}$ is positive, the shell design is acceptable.

Configurations a, b, and c: Proceed to the step in UHX-13.5.11.

Configuration d: The calculation procedure is complete.

(b) Determine the maximum allowable longitudinal compressive stress, $S_{s,b}$.

If $|\sigma_{s,m}| > S_{s,b}$, reconsider the shell design and return to the step in UHX-13.5.1.

If $|\sigma_{s,m}| \leq S_{s,b}$, the shell design is acceptable.

Configurations a, b, and c: Proceed to the step in UHX-13.5.11.

Configuration d: The calculation procedure is complete.

UHX-13.5.11 Step 11. For each loading case, calculate the stresses in the shell and/or channel when integral with the tubesheet (Configurations a, b, and c).

(a) Shell Stresses (Configurations a, b, and c). The shell shall have a uniform thickness of t_s for a minimum length of $1.8\sqrt{D_s t_s}$ adjacent to the tubesheet. Calculate the axial membrane stress $\sigma_{s,m}$, axial bending stress $\sigma_{s,b}$, and total axial stress σ_s, in the shell at its junction to the tubesheet.

$$\sigma_{s,m} = \frac{a_o^2}{t_s(D_s + t_s)}\left[P_e + \left(\rho_s^2 - 1\right)(P_s - P_t)\right] + \frac{a_s^2}{t_s(D_s + t_s)}P_t$$

$$\sigma_{s,b} = \frac{6}{t_s^2}k_s\left\{\beta_s\delta_s P_s + \frac{6\left(1 - \nu^{*2}\right)}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1 + \frac{h\beta_s}{2}\right)\left[P_e\left(Z_v + Z_m Q_1\right) + \frac{2}{a_o^2}Z_m Q_2\right]\right\}$$

$$\sigma_s = \left|\sigma_{s,m}\right| + \left|\sigma_{s,b}\right|$$

(b) Channel Stresses (Configuration a). When the channel is cylindrical, it shall have a uniform thickness of t_c for a minimum length of $1.8\sqrt{D_c t_c}$ adjacent to the tubesheet. Calculate the axial membrane stress $\sigma_{c,m}$, axial bending stress $\sigma_{c,b}$, and total axial stress σ_c, in the channel at its junction to the tubesheet.

$$\sigma_{c,m} = \frac{a_c^2}{t_c(D_c + t_c)}P_t$$

$$\sigma_{c,b} = \frac{6}{t_c^2}k_c\left\{\beta_c\delta_c P_t - \frac{6\left(1 - \nu^{*2}\right)}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right) \times \left[P_e\left(Z_v + Z_m Q_1\right) + \frac{2}{a_o^2}Z_m Q_2\right]\right\}$$

$$\sigma_c = \left|\sigma_{c,m}\right| + \left|\sigma_{c,b}\right|$$

(c) Stress Limitations

Configuration a: For the design loading cases, if $\sigma_s \leq 1.5\ S_s$ and $\sigma_c \leq 1.5\ S_c$, and for the operating loading cases, if $\sigma_s \leq S_{PS,s}$ and $\sigma_c \leq S_{PS,c}$, the shell and channel designs are acceptable, and the calculation procedure is complete. Otherwise, proceed to the step in UHX-13.5.12.

Configurations b and c: For the design loading cases, if $\sigma_s \leq 1.5\ S_s$, and for the operating loading cases, if $\sigma_s \leq S_{PS,s}$, the shell design is acceptable, and the calculation procedure is complete. Otherwise, proceed to the step in UHX-13.5.12.

UHX-13.5.12 Step 12. The design shall be reconsidered by using one or a combination of the following three options:

UHX-13.5.12.1 Option 1. Increase the assumed tubesheet thickness h and return to the step in UHX-13.5.1.

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

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

Configuration a: If $\sigma_c > 1.5\ S_c$, increase the channel thickness t_c and return to the step in UHX-13.5.1.

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

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

UHX-13.6.1 Scope.

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

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

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

UHX-13.6.3 Additional Nomenclature.

$E_{s,1}$ = modulus of elasticity for shell material adjacent to tubesheets at T_s

ℓ_1, ℓ_1' = lengths of shell of thickness $t_{s,1}$ adjacent to tubesheets

$S_{PS,s1}$ = allowable primary plus secondary stress for shell material at T_s per UG-23(e)

$S_{s,1}$ = allowable stress for shell material adjacent to tubesheets at T_s

$S_{s,b,1}$ = maximum allowable longitudinal compressive stress in accordance with UG-23(b) for the shell adjacent to the tubesheets

$S_{y,s,1}$ = yield strength for shell material adjacent to tubesheets at T_s. The yield strength shall be taken from Section II, Part D, Subpart 1, Table Y-1. When a yield strength value is not listed in Table Y-1, one may be obtained by using the procedure in UG-28(c)(2), Step 3.

$t_{s,1}$ = shell thickness adjacent to tubesheets

$\alpha_{s,m,1}$ = mean coefficient of thermal expansion of shell material adjacent to tubesheets at $T_{s,m}$

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

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

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

$$K_s^* = \frac{\pi\left(D_s + t_s\right)}{\dfrac{L-\ell_1-\ell_1'}{E_s t_s} + \dfrac{\ell_1+\ell_1'}{E_{s,1} t_{s,1}}}$$

Calculate $K_{s,t}$ and J, replacing K_s with K_s^*. Calculate β_s, k_s, and δ_s, replacing t_s with $t_{s,1}$ and E_s with $E_{s,1}$.

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

$$\gamma^* = \left(T_{t,m} - T_a\right)\alpha_{t,m}L - \left(T_{s,m} - T_a\right) \times\left[\alpha_{s,m}\left(L-\ell_1-\ell_1'\right) + \alpha_{s,m,1}\left(\ell_1+\ell_1'\right)\right]$$

(d) In the step in UHX-13.5.6, calculate P_γ, replacing γ with γ^*.

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

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

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

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

UHX-13.7 Calculation Procedure for Effect of Plasticity at Tubesheet/Channel or Shell Joint (19)

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

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

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

UHX-13.7.2 Conditions of Applicability.

(a) This procedure shall not be used at temperatures where the time-dependent properties govern the allowable stress.

(b) This procedure applies only for the design loading cases.

(c) This procedure applies to Configuration a when $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$.

(d) This procedure applies to Configurations b and c when $\sigma_s \le S_{PS,s}$.

(e) This procedure may only be used once for each iteration of tubesheet, shell, and channel thicknesses and materials.

UHX-13.7.3 Additional Nomenclature.

fact_c = factor used in the elastic–plastic analysis to account for any yielding of the channel

fact_s = factor used in the elastic–plastic analysis to account for any yielding of the shell

UHX-13.7.4 Calculation Procedure. After the calculation procedure given in UHX-13.5.1 through UHX-13.5.11 has been performed for the elastic solution, an elastic–plastic calculation using the referenced steps from UHX-13.5 shall be performed in accordance with the following procedure for each applicable loading case. Except for those quantities modified below, the quantities to be used for the elastic–plastic calculation shall be the same as those calculated for the corresponding elastic loading case.

(a) Define the maximum permissible bending stress limit in the shell and channel.

Configurations a, b, and c:

$$S_s^* = \text{MIN}\left[\left(S_{y,s}\right), \left(\frac{S_{PS,s}}{2}\right)\right]$$

Configuration a:

$$S_c^* = \text{MIN}\left[\left(S_{y,c}\right), \left(\frac{S_{PS,c}}{2}\right)\right]$$

(b) Using bending stresses $\sigma_{s,b}$ and $\sigma_{c,b}$ computed in UHX-13.5.11 (Step 11) for the elastic solution, determine fact_s and fact_c as follows:

Configurations a, b, and c:

$$\text{fact}_s = \text{MIN}\left[\left(1.4 - 0.4\frac{\left|\sigma_{s,b}\right|}{S_s^*}\right), (1)\right]$$

Configuration a:

$$\text{fact}_c = \text{MIN}\left[\left(1.4 - 0.4\frac{\left|\sigma_{c,b}\right|}{S_c^*}\right), (1)\right]$$

Configuration a: If $\text{fact}_s = 1.0$ and $\text{fact}_c = 1.0$, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to (c) below.

Configurations b and c: If $\text{fact}_s = 1.0$, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to (c) below.

(c) Calculate reduced values of E_s and E_c as follows:

Configurations a, b, and c: $E_s^* = E_s \text{fact}_s$

Configuration a: $E_c^* = E_c \text{fact}_c$

(d) In UHX-13.5.2 (Step 2), recalculate k_s, λ_s, k_c, and λ_c replacing E_s by E_s^* and E_c by E_c^*.

(e) In UHX-13.5.4 (Step 4), recalculate F, Φ, Q_1, Q_{Z1}, Q_{Z2}, and U.

(f) In UHX-13.5.6 (Step 6), recalculate P_W, P_{rim}, and P_e.

(g) In UHX-13.5.7 (Step 7), recalculate Q_2, Q_3, and F_m as applicable, and the tubesheet bending stress σ.

If $|\sigma| \le 1.5S$, the design is acceptable and the calculation procedure is complete. Otherwise, the unit geometry shall be reconsidered.

UHX-13.8 Calculation Procedure for Effect of Radial Differential Thermal Expansion Adjacent to the Tubesheet

UHX-13.8.1 Scope.

(a) This procedure describes how to use the rules of UHX-13.5 when the effect of radial differential thermal expansion between the tubesheet and integral shell or channel is to be considered.

(b) This procedure shall be used when cyclic or dynamic reactions due to pressure or thermal variations are specified [see UG-22(e)].

(c) This procedure shall be used when specified by the user or his designated agent. The user or his designated agent shall provide the Manufacturer with the data necessary to determine the required tubesheet, channel, and shell metal temperatures.

(d) Optionally, the designer may use this procedure to consider the effect of radial differential thermal expansion even when it is not required by (b) or (c) above.

UHX-13.8.2 Conditions of Applicability. This calculation procedure applies only when the tubesheet is integral with the shell or channel (Configurations a, b, and c).

UHX-13.8.3 Additional Nomenclature. (19)

T' = tubesheet metal temperature at the rim (see Figure UHX-11.3-3)

T_c' = channel metal temperature at the tubesheet

T_{cx}' = channel metal temperature at the tubesheet for operating condition x

T_s' = shell metal temperature at the tubesheet

T_{sx}' = shell metal temperature at the tubesheet for operating condition x

T_x' = tubesheet metal temperature at the rim for operating condition x

α' = mean coefficient of thermal expansion of tubesheet material at T'

α_c' = mean coefficient of thermal expansion of channel material at T_c'

α_s' = mean coefficient of thermal expansion of shell material at T_s'

UHX-13.8.4 Calculation Procedure. The calculation procedure given in UHX-13.5 and UHX-13.6, if applicable, shall be performed only for the operating loading cases accounting for the modifications given in (a) through (g).

Table UHX-13.8.4-1 provides the load combinations required to evaluate the heat exchanger for each operating condition x.

(a) Determine the average temperature of the unperforated rim T_r.

Configuration a:

$$T_r = \frac{T' + T_s' + T_c'}{3}$$

Configurations b and c:

$$T_r = \frac{T' + T_s'}{2}$$

For conservative values of P_s^* and P_c^*, $T_r = T'$ may be used.

(b) Determine the average temperature of the shell T_s^* and channel T_c^* at their junction to the tubesheet as follows:

Configurations a, b, and c:

$$T_s^* = \frac{T_s' + T_r}{2}$$

Configuration a:

$$T_c^* = \frac{T_c' + T_r}{2}$$

For conservative values of P_s^* and P_c^*, $T_s^* = T_s'$ and $T_c^* = T_c'$ may be used.

(c) Calculate P_s^* and P_c^*.

Configurations a, b, and c:

$$P_s^* = \frac{E_s t_s}{a_s}\left[\alpha_s'\left(T_s^* - T_a\right) - \alpha'\left(T_r - T_a\right)\right]$$

Configuration a:

$$P_c^* = \frac{E_c t_c}{a_c}\left[\alpha_c'\left(T_c^* - T_a\right) - \alpha'\left(T_r - T_a\right)\right]$$

Configurations b and c:

$$P_c^* = 0$$

(d) Calculate P_ω.

$$P_\omega = \frac{U}{a_o^2}\left(\omega_s P_s^* - \omega_c P_c^*\right)$$

(e) In UHX-13.5.6 (Step 6), replace the formula for P_e with:

$$P_e = \frac{J K_{s,t}}{1 + J K_{s,t}\left[Q_{Z1} + \left(\rho_s - 1\right)Q_{Z2}\right]} \times \left(P_s' - P_t' + P_\gamma + P_\omega + P_W + P_{\text{rim}}\right)$$

Table UHX-13.8.4-1

Operating Loading Case	Operating Pressure		Axial Mean Metal Temperature		Metal Temperature		
	Shell Side, P_s	Tube Side, P_t	Tubes, $T_{t,m}$	Shell, $T_{s,m}$	Tubesheet at the Rim, T'	Channel at Tubesheet, T_c'	Shell at Tubesheet, T_s'
1	$P_{sox,\min}$	$P_{tox,\max}$	$T_{t,mx}$	$T_{s,mx}$	T_x'	T_{cx}'	T_{sx}'
2	$P_{sox,\max}$	$P_{tox,\min}$	$T_{t,mx}$	$T_{s,mx}$	T_x'	T_{cx}'	T_{sx}'
3	$P_{sox,\max}$	$P_{tox,\max}$	$T_{t,mx}$	$T_{s,mx}$	T_x'	T_{cx}'	T_{sx}'
4	$P_{sox,\min}$	$P_{tox,\min}$	$T_{t,mx}$	$T_{s,mx}$	T_x'	T_{cx}'	T_{sx}'

(f) In UHX-13.5.7 (Step 7), replace the formula for Q_2 with:

$$Q_2 = \frac{\left(\omega_s^* P_s - \omega_c^* P_t\right) - \left(\omega_s P_s^* - \omega_c P_c^*\right) + \frac{\gamma_b}{2\pi} W^*}{1 + \Phi Z_m}$$

(g) In UHX-13.5.11 (Step 11), replace the equations for $\sigma_{s,b}$ and $\sigma_{c,b}$ with:

$$\sigma_{s,b} = \frac{6}{t_s^2} k_s \left\{ \beta_s \left[\delta_s P_s + \frac{a_s^2}{E_s t_s} P_s^* \right] + \frac{6\left(1 - \nu^{*2}\right)}{E^*} \times \left(\frac{a_o^3}{h^3} \right) \left(1 + \frac{h \beta_s}{2} \right) \left[P_e \left(Z_v + Z_m Q_1 \right) + \frac{2}{a_o^2} Z_m Q_2 \right] \right\}$$

$$\sigma_{c,b} = \frac{6}{t_c^2} k_c \left\{ \beta_c \left[\delta_c P_t + \frac{a_c^2}{E_c t_c} P_c^* \right] - \frac{6\left(1 - \nu^{*2}\right)}{E^*} \times \left(\frac{a_o^3}{h^3} \right) \left(1 + \frac{h \beta_c}{2} \right) \left[P_e \left(Z_v + Z_m Q_1 \right) + \frac{2}{a_o^2} Z_m Q_2 \right] \right\}$$

UHX-13.9 Calculation Procedure for Simply Supported Fixed Tubesheets

UHX-13.9.1 Scope. This procedure describes how to use the rules of UHX-13.5 when the effect of the stiffness of the integral channel and/or shell is not considered.

UHX-13.9.2 Conditions of Applicability. This calculation procedure applies only when the tubesheet is integral with the shell or channel (configurations a, b, and c).

UHX-13.9.3 Calculation Procedure. The calculation procedure given in UHX-13.5 shall be performed accounting for the following modifications.

(a) Perform Steps 1 through 10.

(b) Perform Step 11 except as follows:

(1) The shell (configurations a, b, and c) is not required to meet a minimum length requirement. The shell is exempt from the minimum length requirement in UHX-13.6.4(a).

(2) The channel (configuration a) is not required to meet a minimum length requirement.

(3) Configuration a: If $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$, the shell and channel are acceptable. Otherwise, increase the thickness of the overstressed component(s) (shell and/or channel) and return to Step 1.

Configurations b and c: If $\sigma_s \le S_{PS,s}$, the shell is acceptable. Otherwise, increase the thickness of the shell and return to Step 1.

(c) Do not perform Step 12.

(d) Repeat Steps 1 through 7 for the design loading cases, with the following changes to Step 2, until the tubesheet stress criteria have been met:

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

Configuration a: $\beta_c = 0$, $k_c = 0$, $\lambda_c = 0$, $\delta_c = 0$.

UHX-13.10 Calculation Procedure for Kettle Shell Exchangers With Fixed Tubesheets

UHX-13.10.1 Scope. This procedure describes how to use the rules of UHX-13.5 when an eccentric cone and small cylinder exist between the large shell side cylinder and the tubesheet on both sides.

UHX-13.10.2 Conditions of Applicability.

(a) The two eccentric cones are identical in geometry and material.

(b) The small shell cylinders adjacent to the tubesheet are identical in geometry and material. They shall meet the length requirements of UHX-13.5.11(a) unless the simply supported rules of UHX-13.9 are applied. The rules of UHX-13.6 shall not be used. The rules of UHX-13.8 may be used only if the length requirements of UHX-13.5.11(a) are met by the small shell cylinders.

(c) This procedure applies only when $\theta_{ecc} \le 30$ deg. This procedure accounts for the stiffness and loadings in the shell of the eccentric cones used in the design of the tubesheet. This procedure does not evaluate the acceptability of the shell-to-cone transition. Other requirements in this Division pertaining to shell-to-cone transitions shall be satisfied [e.g., UW-3(b), 1-5, and 1-8].

(d) This procedure applies only when $0.5 \le \frac{L_{ecc}}{D_{ecc,S}} \le 1.5$.

(e) This procedure applies only when $D_{ecc,L} \le 2.17 D_{ecc,S}$.

(f) These rules assume that an expansion joint, if present, is located in the small shell cylinder.

(g) For cone-to-cylinder junctions without a transition knuckle, use the following for design cases (pressure-only cases) in 1-5. The cone-to-cylinder junctions do not need to be evaluated for the operating cases (cases including differential thermal expansion).

$$f_1 = f_1' + f_1''$$

$$f_2 = f_2' + f_2''$$

where

$$f_1' = \sigma_{ecc,L,m} t_{ecc} \cos\left(\theta_{ecc}\right) - \frac{P_s D_{ecc,L}}{4}$$

$$f_2' = \sigma_{ecc,S,m} t_{ecc} \cos\left(\theta_{ecc}\right) - \frac{P_s D_{ecc,S}}{4}$$

(h) For cone-to-cylinder junctions without a transition knuckle, use the following for design cases (pressure-only cases) in 1-8. The cone-to-cylinder junctions do not need to be evaluated for the operating cases (cases including differential thermal expansion).

$$f_1 = f_1' + f_1''$$

$$f_2 = f_2' + f_2''$$

where

$$f_1' = -\sigma_{ecc,L,m} t_{ecc} \cos\left(\theta_{ecc}\right) + \frac{P_s D_{ecc,L}}{4}$$

$$f_2' = -\sigma_{ecc,S,m} t_{ecc} \cos\left(\theta_{ecc}\right) + \frac{P_s D_{ecc,S}}{4}$$

UHX-13.10.3 Additional Nomenclature.

$D_{ecc,L}$ = eccentric cone inside diameter at the large end (see Figure UHX-13.10.3-1)

$D_{ecc,S}$ = eccentric cone inside diameter at the small end (see Figure UHX-13.10.3-1)

$D_{s,L}$ = large cylinder inside diameter (see Figure UHX-13.10.3-1)

E_{ecc} = modulus of elasticity for eccentric cone material at T_s

$E_{ecc,w}$ = joint efficiency (longitudinal stress) for eccentric cone

$E_{s,L}$ = modulus of elasticity for large cylinder material at T_s

$E_{s,L,w}$ = joint efficiency (longitudinal stress) for large cylinder

f_1 = axial load per unit circumference at conical reducer large end due to wind, dead load, heat exchanger constraint, etc., excluding pressure, for use in 1-5 or 1-8 cone-to-cylinder junction analyses. Note that per 1-5(d)(1), tension is positive, and that per 1-8(b)(1), compression is positive.

$f_1{}'$ = axial load per unit circumference at conical reducer large end due to heat exchanger constraint, excluding pressure. See definition of f_1 for signs.

$f_1{}''$ = axial load per unit circumference at conical reducer large end due to wind, dead load, etc., excluding pressure. See definition of f_1 for signs.

f_2 = axial load per unit circumference at conical reducer small end due to wind, dead load, heat exchanger constraint, etc., excluding pressure, for use in 1-5 or 1-8 cone-to-cylinder junction analyses. Note that per 1-5(e)(1), tension is positive, and that per 1-8(c)(1), compression is positive.

$f_2{}'$ = axial load per unit circumference at conical reducer small end due to heat exchanger constraint, excluding pressure. See definition of f_2 for signs.

$f_2{}''$ = axial load per unit circumference at conical reducer small end due to wind, dead load, etc., excluding pressure. See definition of f_2 for signs.

L_{ecc} = eccentric cone shortest length from small end to large end (see Figure UHX-13.10.3-1)

L_s = axial length of small cylinder (see Figure UHX-13.10.3-1)

$L_{s,L}$ = axial length of large cylinder (see Figure UHX-13.10.3-1)

S_{ecc} = allowable stress for eccentric cone material at T_s

$S_{ecc,b}$ = maximum allowable longitudinal compressive stress for eccentric cone material at T_s; see U-2(g)

$S_{PS,ecc}$ = allowable primary plus secondary stress for eccentric cone material at T_s per UG-23(e)

$S_{PS,s,L}$ = allowable primary plus secondary stress for large cylinder material at T_s per UG-23(e)

$S_{s,L}$ = allowable stress for large cylinder material at T_s

$S_{s,L,b}$ = maximum allowable longitudinal compressive stress in accordance with UG-23(b) for large cylinder material at T_s

t_{ecc} = eccentric cone wall thickness (see Figure UHX-13.10.3-1)

$t_{s,L}$ = large cylinder wall thickness (see Figure UHX-13.10.3-1)

$\alpha_{ecc,m}$ = mean coefficient of thermal expansion of eccentric cone material at $T_{s,m}$

$\alpha_{s,m,L}$ = mean coefficient of thermal expansion of large cylinder material at $T_{s,m}$

θ_{ecc} = eccentric cone half-apex angle, deg (see Figure UHX-13.10.3-1)

ν_{ecc} = Poisson's ratio of eccentric cone material

$\nu_{s,L}$ = Poisson's ratio of large cylinder material

UHX-13.10.4 Calculation Procedure. The calculation procedure outlined in UHX-13.5 shall be performed accounting for the following modifications:

(a) Perform Step 2 (UHX-13.5.2) with the following changes:

$$K_{ecc} = 0.8\frac{\pi t_{ecc}\left(D_{ecc,S} + t_{ecc}\right)E_{ecc}}{L_{ecc}}$$

$$K_{s,L} = \frac{\pi t_{s,L}\left(D_{s,L} + t_{s,L}\right)E_{s,L}}{L_{s,L}}$$

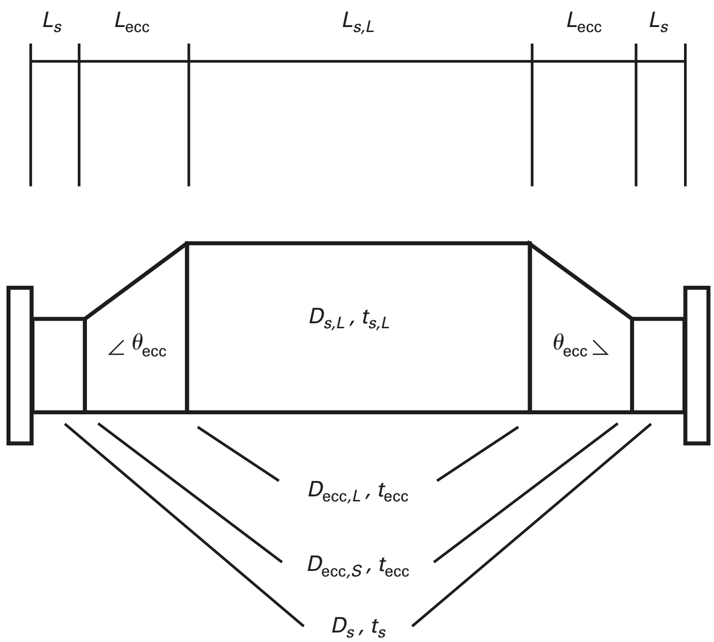
$$K_s = \frac{\pi t_s\left(D_s + t_s\right)E_s}{L_s}$$

$$K_s{}^* = \frac{K_s K_{s,L} K_{ecc}}{2K_{ecc}K_{s,L} + 2K_{s,L}K_s + K_s K_{ecc}}$$

$$K_{s,t} = \frac{K_s{}^*}{N_t K_t}$$

$$J = \frac{1}{1 + \frac{K_s{}^*}{K_J}}$$

Figure UHX-13.10.3-1
Kettle Shell



(b) Perform Step 5 (UHX-13.5.5) with the following change:

$$\gamma = \alpha_{t,m}(T_{t,m} - T_a)L - \alpha_{s,m,L}(T_{s,m} - T_a)L_{s,L} - 2\alpha_{ecc,m}(T_{s,m} - T_a)L_{ecc} - 2\alpha_{s,m}(T_{s,m} - T_a)L_s$$

(c) Perform Step 6 (UHX-13.5.6) with the following changes; use $\nu_s{}^*$ instead of ν_s:

$$A_s = D_s(D_s + t_s)$$

$$A_{s,L} = D_{s,L}(D_{s,L} + t_{s,L})$$

$$\Delta_{ecc} = D_{ecc,L} - D_{ecc,S}$$

$$\nu_s{}^* = \frac{2K_s{}^*}{A_s}\left\{\frac{A_s\nu_s}{K_s} + \left[\nu_{ecc} - \frac{L_{ecc}{}^2 - 2\nu_{ecc}L_{ecc}{}^2 - 3\Delta_{ecc}{}^2}{8L_{ecc}{}^3} \times \left(\Delta_{ecc}{}^2 + L_{ecc}{}^2\right)^{0.5}\right]\frac{(D_{ecc,L} + D_{ecc,S})(D_{ecc,S} + t_{ecc})}{5K_{ecc}} + \frac{A_{s,L}\nu_{s,L}}{2K_{s,L}}\right\}$$

(d) Perform Step 10 (UHX-13.5.10) with the following changes:

(1) Calculate the axial membrane stress for the small cylinder.

$$\sigma_{s,m} = \frac{a_o{}^2}{t_s(D_s + t_s)}\left[P_e + \left(\frac{D_s{}^2}{4a_o{}^2} - 1\right)(P_s - P_t)\right] + \frac{D_s{}^2}{4t_s(D_s + t_s)}P_t$$

(2) Calculate the axial membrane stress for the eccentric cone at the small end.

$$\sigma_{ecc,S,m} = \frac{a_o{}^2}{t_{ecc}(D_{ecc,S} + t_{ecc})\cos(\theta_{ecc})}\left[P_e + \left(\frac{D_{ecc,S}{}^2}{4a_o{}^2} - 1\right) \times (P_s - P_t)\right] + \frac{D_{ecc,S}{}^2}{4t_{ecc}(D_{ecc,S} + t_{ecc})\cos(\theta_{ecc})}P_t$$

(3) Calculate the axial membrane stress for the eccentric cone at the large end.

$$\sigma_{ecc,L,m} = \frac{a_o{}^2}{t_{ecc}(D_{ecc,L} + t_{ecc})\cos(\theta_{ecc})}\left[P_e + \left(\frac{D_{ecc,L}{}^2}{4a_o{}^2} - 1\right) \times (P_s - P_t)\right] + \frac{D_{ecc,L}{}^2}{4t_{ecc}(D_{ecc,L} + t_{ecc})\cos(\theta_{ecc})}P_t$$

(4) Calculate the axial membrane stress for the large cylinder.

$$\sigma_{s,L,m} = \frac{a_o{}^2}{t_{s,L}(D_{s,L} + t_{s,L})}\left[P_e + \left(\frac{D_{s,L}{}^2}{4a_o{}^2} - 1\right)(P_s - P_t)\right] + \frac{D_{s,L}{}^2}{4t_{s,L}(D_{s,L} + t_{s,L})}P_t$$

(5) Acceptance Criteria

(-a) Design loading case acceptance criteria: $|\sigma_{s,m}| \le S_s E_{s,w}$ and $|\sigma_{ecc,S,m}| \le S_{ecc} E_{ecc,w}$ and $|\sigma_{ecc,L,m}| \le S_{ecc} E_{ecc,w}$ and $|\sigma_{s,L,m}| \le S_{s,L} E_{s,L,w}$

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

(-c) If axial membrane stress is negative (design and operating): $|\sigma_{s,m}| \le S_{s,b}$ and $|\sigma_{ecc,S,m}| \le S_{ecc,b}$ and $|\sigma_{ecc,L,m}| \le S_{ecc,b}$ and $|\sigma_{s,L,m}| \le S_{s,L,b}$

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

UHX-14 RULES FOR THE DESIGN OF FLOATING TUBESHEETS

UHX-14.1 Scope

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

(1) Sketch (a), immersed floating head;

(2) Sketch (b), externally sealed floating head;

(3) Sketch (c), internally sealed floating tubesheet.

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

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

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

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

(4) Configuration d: tubesheet gasketed with shell and channel;

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

(6) Configuration f: tubesheet gasketed with shell and integral with channel, not extended as a flange.

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

(1) Configuration A: tubesheet integral;

(2) Configuration B: tubesheet gasketed, extended as a flange;

(3) Configuration C: tubesheet gasketed, not extended as a flange;

(4) Configuration D: tubesheet internally sealed.

UHX-14.2 Conditions of Applicability

The two tubesheets shall have the same thickness and material.

UHX-14.3 Nomenclature

The symbols described below are used for the design of the stationary and floating tubesheets. Symbols D_o, E^*, h'_g, μ, μ^*, and ν^* are defined in UHX-11.

A = outside diameter of tubesheet, except as limited by UHX-10(b)

a_c = radial channel dimension
Configurations a, e, f, and A: $a_c = D_c/2$
Configurations b, c, d, B, and C: $a_c = G_c/2$
Configuration D: $a_c = A/2$

a_o = equivalent radius of outer tube limit circle

A_P = total area enclosed by C_P

a_s = radial shell dimension
Configurations a, b, and c: $a_s = D_s/2$
Configurations d, e, and f: $a_s = G_s/2$
Configurations A, B, C, and D: $a_s = a_c$

C = bolt circle diameter (see Mandatory Appendix 2)

Cp = perimeter of the tube layout measured stepwise in increments of one tube pitch from the center-to-center of the outermost tubes (see Figure UHX-12.2)

D_c = inside channel diameter

D_s = inside shell diameter

d_t = nominal outside diameter of tubes

E = modulus of elasticity for tubesheet material at T

E_c = modulus of elasticity for channel material at T_c

E_s = modulus of elasticity for shell material at T_s

E_t = modulus of elasticity for tube material at T_t

G_1 = midpoint of contact between flange and tubesheet

G_c = diameter of channel gasket load reaction (see Mandatory Appendix 2)

G_s = diameter of shell gasket load reaction (see Mandatory Appendix 2)

h = tubesheet thickness

k = constant accounting for the method of support for the unsupported tube span under consideration
= 0.6 for unsupported spans between two tubesheets
= 0.8 for unsupported spans between a tubesheet and a tube support
= 1.0 for unsupported spans between two tube supports

L = tube length between inner tubesheet faces
= $L_t - 2h$

l = unsupported tube span under consideration

L_t = tube length between outer tubesheet faces

MAX *[(a),(b),(c),...]* = greatest of *a, b, c,* ...

N_t = number of tubes

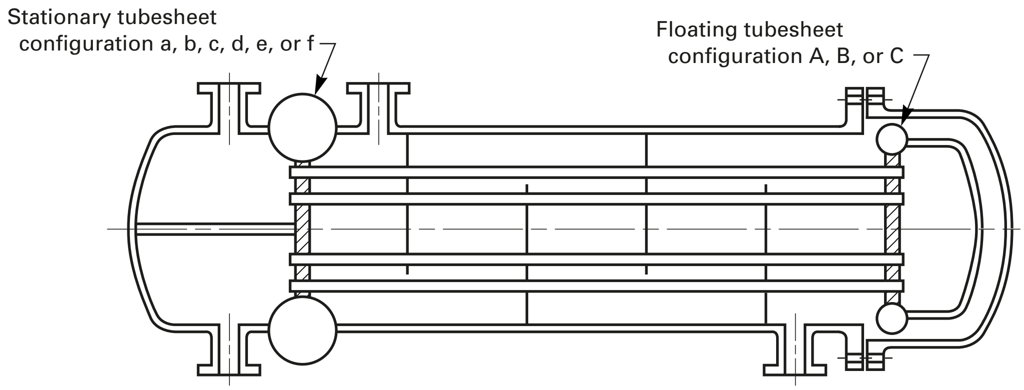
P_e = effective pressure acting on tubesheet

P_s = shell side design or operating pressure, as applicable. For shell side vacuum, use a negative value for P_s.

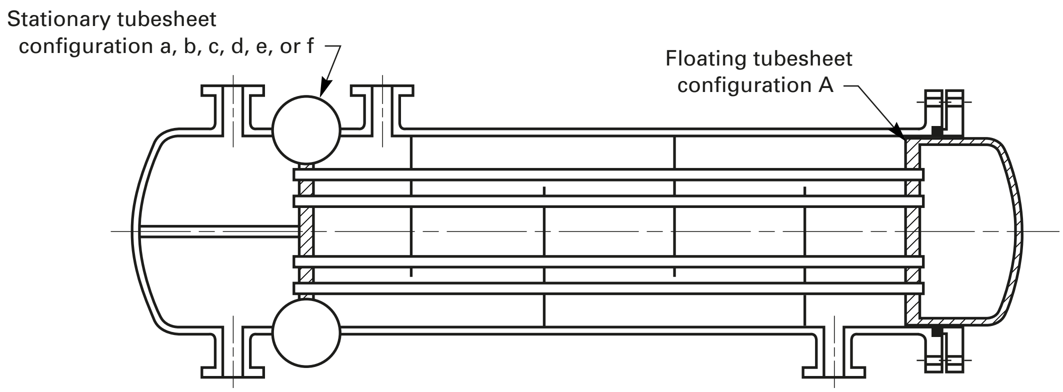
$P_{sd,max}$ = maximum shell side design pressure

$P_{sd,min}$ = minimum shell side design pressure (negative if vacuum is specified, otherwise zero)

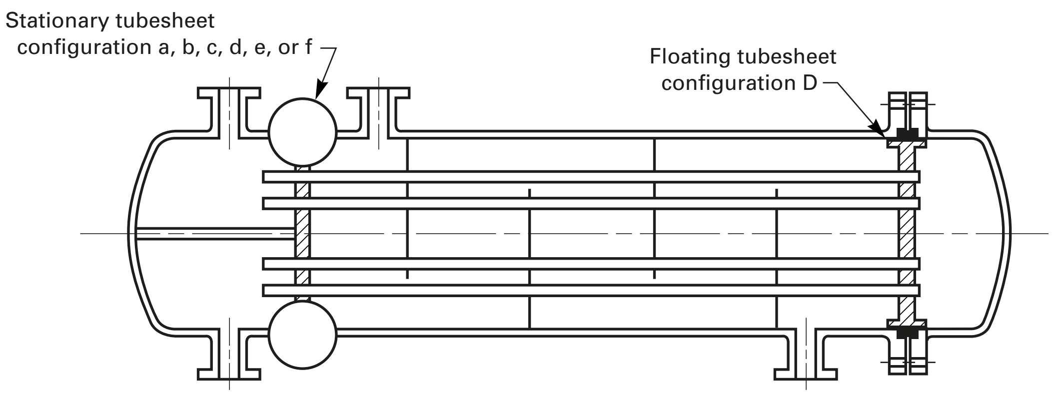
Figure UHX-14.1
Floating Tubesheet Heat Exchangers



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

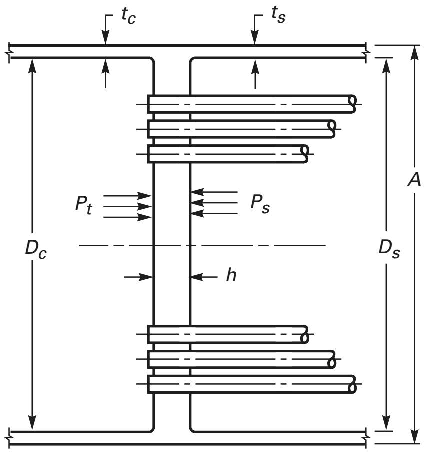


(b) Typical Floating Tubesheet Exchanger With an Externally Sealed Floating Head

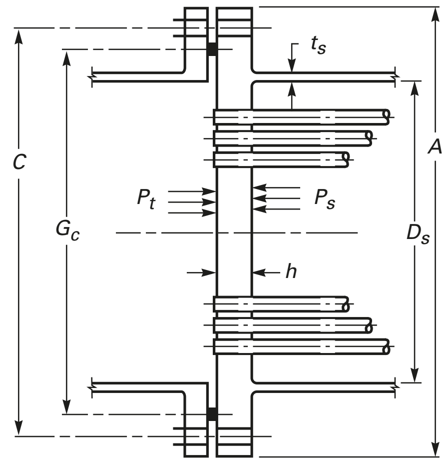


(c) Typical Floating Tubesheet Exchanger With an Internally Sealed Floating Tubesheet

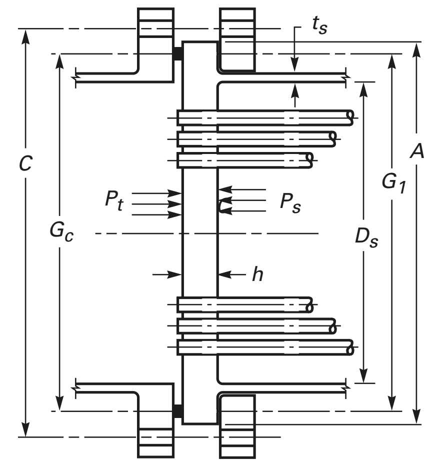
Figure UHX-14.2
Stationary Tubesheet Configurations



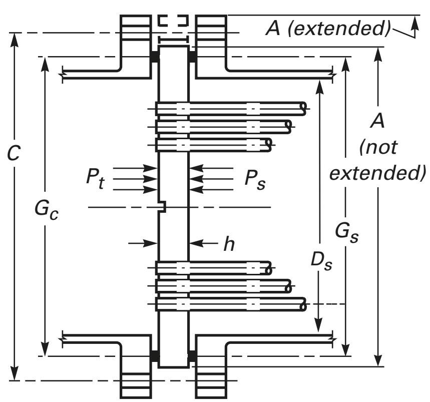
(a) Configuration a:
Tubesheet Integral With Shell and Channel



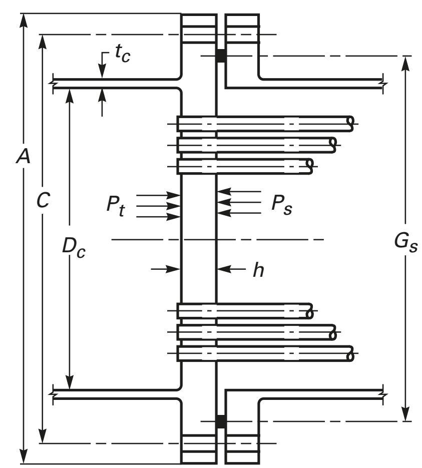
(b) Configuration b:
Tubesheet Integral With Shell and Gasketed With Channel, Extended as a Flange



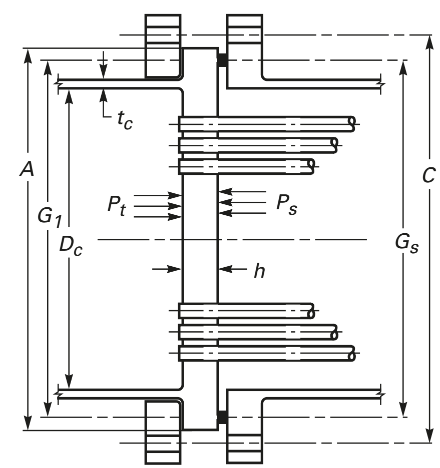
(c) Configuration c:
Tubesheet Integral With Shell and Gasketed With Channel, Not Extended as a Flange



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



(e) Configuration e:
Tubesheet Gasketed With Shell and Integral With Channel, Extended as a Flange



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

$P_{sox,\max}$ = max.(0, maximum shell side operating pressure for operating condition x)

$P_{sox,\min}$ = min.(0, minimum shell side operating pressure for operating condition x)

P_t = tube side design or operating pressure, as applicable. For tube side vacuum, use a negative value for P_t.

$P_{td,\max}$ = maximum tube side design pressure

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

$P_{tox,\max}$ = max.(0, maximum tube side operating pressure for operating condition x)

$P_{tox,\min}$ = min.(0, minimum tube side operating pressure for operating condition x)

S = allowable stress for tubesheet material at T

S_c = allowable stress for channel material at T_c

S_{PS} = allowable primary plus secondary stress for tubesheet material at T per UG-23(e)

$S_{PS,c}$ = allowable primary plus secondary stress for channel material at T_c per UG-23(e)

$S_{PS,s}$ = allowable primary plus secondary stress for shell material at T_s per UG-23(e)

S_s = allowable stress for shell material at T_s

S_t = allowable stress for tube material at T_t

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

S_y = yield strength for tubesheet material at T

$S_{y,c}$ = yield strength for channel material at T_c

$S_{y,s}$ = yield strength for shell material at T_s

$S_{y,t}$ = yield strength for tube material at T_t

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

T = tubesheet design temperature for the design condition or operating metal temperature for operating condition x, as applicable [see UHX-14.4(c)]

T_a = ambient temperature, 70°F (20°C)

T_c = channel design temperature for the design condition or operating metal temperature for operating condition x, as applicable [see UHX-14.4(c)]

t_c = channel thickness

T_s = shell design temperature for the design condition or operating metal temperature for operating condition x, as applicable [see UHX-14.4(c)]

t_s = shell thickness

T_t = tube design temperature for the design condition or operating metal temperature for operating condition x, as applicable [see UHX-14.4(c)]

t_t = nominal tube wall thickness

W_t = tube-to-tubesheet joint load

W^* = tubesheet effective bolt load determined in accordance with UHX-8

x = 1, 2, 3, ... n, integer denoting applicable operating condition under consideration (e.g., normal operating, start-up, shutdown, cleaning, upset)

ν = Poisson's ratio of tubesheet material

ν_c = Poisson's ratio of channel material

ν_s = Poisson's ratio of shell material

ν_t = Poisson's ratio of tube material

UHX-14.4 Design Considerations (19)

(a) The calculation shall be performed for the stationary end and for the floating end of the exchanger. Since the edge configurations of the stationary and floating tubesheets are different, the data may be different for each set of calculations. However, the conditions of applicability given in UHX-14.2 must be maintained. For the stationary end, diameters A, C, D_s, D_c, G_s, G_c, G_1, and thickness t_c shall be taken from Figure UHX-14.2. For the floating end, diameters A, C, D_c, G_c, G_1, and thickness t_c shall be taken from Figure UHX-14.3, and the radial shell dimension a_s shall be taken equal to a_c.

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

The user or his designated agent shall specify all the design and operating conditions that govern the design of the main components of the heat exchanger (i.e., tubesheets, tubes, shell, channel, tube-to-tubesheet joint). These shall include, but not be limited to, normal operating, start-up, shutdown, cleaning, and upset conditions.

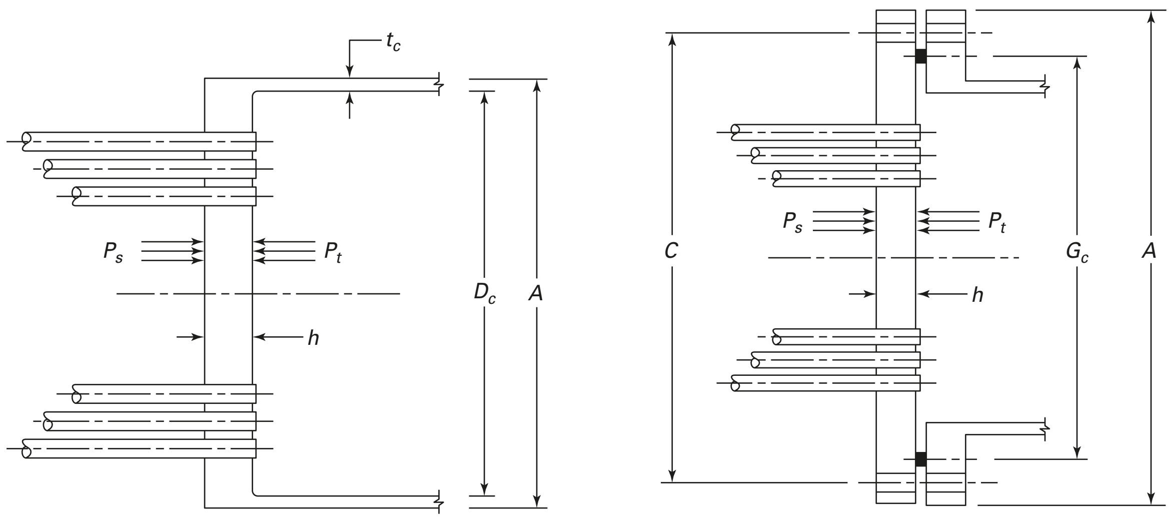
For each of these conditions, the following loading cases shall be considered to determine the effective pressure P_e to be used in the design equations:

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

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

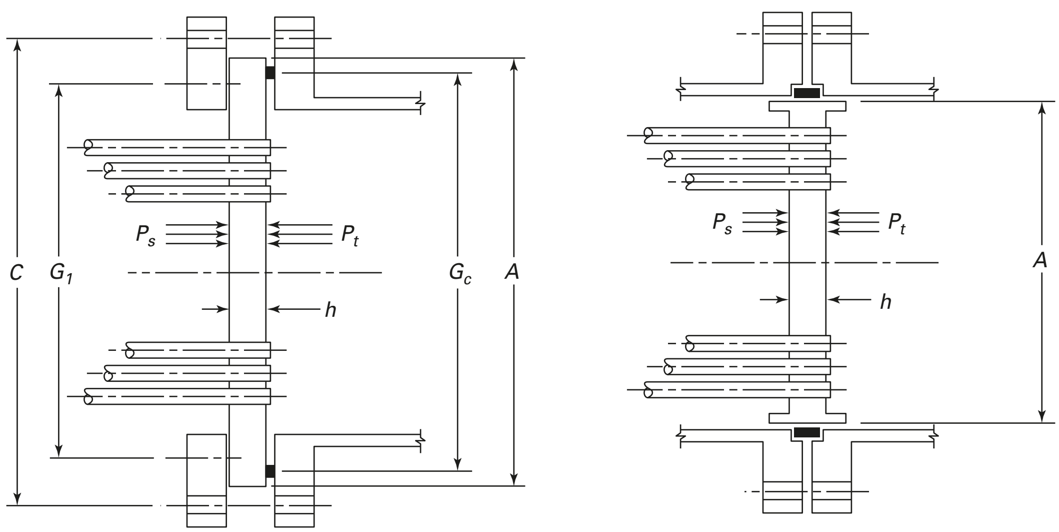
(3) When differential pressure design is specified by the user or his designated agent, the design shall be based only on design loading case 3 and operating loading cases 3 and 4 for each specified operating condition. If the tube side is the higher-pressure side, P_t shall be the tube side design pressure,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

Figure UHX-14.3
Floating Tubesheet Configurations



(a) Configuration A:
Tubesheet Integral

(b) Configuration B:
Tubesheet Gasketed, Extended as a Flange



(c) Configuration C:
Tubesheet Gasketed, Not Extended as a Flange

(d) Configuration D:
Tubesheet Internally Sealed

Table UHX-14.4-1

Design Loading Case	Shell Side Design Pressure, P_s	Tube Side Design Pressure, P_t
1	$P_{sd,\min}$	$P_{td,\max}$
2	$P_{sd,\max}$	$P_{td,\min}$
3	$P_{sd,\max}$	$P_{td,\max}$
4	$P_{sd,\min}$	$P_{td,\min}$

loading cases, the differential pressure and the individual operating pressures shall not exceed the values used for design.

(4) The designer should take appropriate consideration of the stresses resulting from the pressure test required by UG-99 or UG-100 [see UG-99(d)].

(c) The elastic moduli, yield strengths, and allowable stresses shall be taken at the design temperatures for the design loading cases and may be taken at the operating metal temperature of the component under consideration for operating condition x.

(d) As the calculation procedure is iterative, a value h shall be assumed for the tubesheet thickness to calculate and check that the maximum stresses in tubesheet, tubes, shell, and channel are within the maximum permissible stress limits and that the resulting tube-to-tubesheet joint load is acceptable.

(e) The designer shall consider the effect of radial differential thermal expansion adjacent to the tubesheet in accordance with UHX-14.6, if required by UHX-14.6.1.

(f) The designer may consider the tubesheet as simply supported in accordance with UHX-14.7.

UHX-14.5 Calculation Procedure

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

UHX-14.5.1 Step 1. Determine D_o, μ, μ^*, and h'_g from UHX-11.5.1.

Operating loading cases: $h'_g = 0$

Calculate a_o, ρ_s, ρ_c, x_s, and x_t.

$$a_o = \frac{D_o}{2}$$

$$\rho_s = \frac{a_s}{a_o}$$

$$\rho_c = \frac{a_c}{a_o}$$

$$x_s = 1 - N_t\left(\frac{d_t}{2a_o}\right)^2$$

$$x_t = 1 - N_t\left(\frac{d_t - 2t_t}{2a_o}\right)^2$$

UHX-14.5.2 Step 2. Calculate shell coefficients β_s, k_s, λ_s, and δ_s.

Configurations a, b, and c:

$$\beta_s = \frac{\sqrt[4]{12\left(1 - \nu_s^2\right)}}{\sqrt{\left(D_s + t_s\right)t_s}}$$

$$k_s = \beta_s \frac{E_s t_s^3}{6\left(1 - \nu_s^2\right)}$$

$$\lambda_s = \frac{6D_s}{h^3}k_s\left(1 + h\beta_s + \frac{h^2\beta_s^2}{2}\right)$$

$$\delta_s = \frac{D_s^2}{4E_s t_s}\left(1 - \frac{\nu_s}{2}\right)$$

Configurations d, e, f, A, B, C, and D: $\beta_s = 0$, $k_s = 0$, $\lambda_s = 0$, $\delta_s = 0$

Calculate channel coefficients β_c, k_c, λ_c, and δ_c.

Configurations a, e, f, and A:

$$\beta_c = \frac{\sqrt[4]{12\left(1 - \nu_c^2\right)}}{\sqrt{\left(D_c + t_c\right)t_c}}$$

$$k_c = \beta_c \frac{E_c t_c^3}{6\left(1 - \nu_c^2\right)}$$

$$\lambda_c = \frac{6D_c}{h^3}k_c\left(1 + h\beta_c + \frac{h^2\beta_c^2}{2}\right)$$

For a cylinder:

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

For a hemispherical head:

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

Configurations b, c, d, B, C, and D: $\beta_c = 0$, $k_c = 0$, $\lambda_c = 0$, $\delta_c = 0$

UHX-14.5.3 Step 3. Calculate h/p. If ρ changes, recalculate d^* and μ^* from UHX-11.5.1.

Determine E^*/E and ν^* relative to h/p from UHX-11.5.2.

Calculate X_a.

$$X_a = \left[24\left(1-\nu^{*2}\right)N_t \frac{E_t\, t_t\left(d_t - t_t\right)a_o^2}{E^*\, L\, h^3}\right]^{\frac{1}{4}}$$

Using the calculated value of X_a, enter either Table UHX-13.1 or Figure UHX-13.2 to determine Z_d, Z_v, Z_w, and Z_m.

UHX-14.5.4 Step 4. Calculate diameter ratio K and coefficient F.

$$K = \frac{A}{D_o}$$

$$F = \frac{1-\nu^*}{E^*}\left(\lambda_s + \lambda_c + E \ln K\right)$$

Calculate Φ and Q_1.

$$\Phi = \left(1+\nu^*\right)F$$

$$Q_1 = \frac{\rho_s - 1 - \Phi Z_v}{1 + \Phi Z_m}$$

UHX-14.5.5 Step 5.

(a) Calculate ω_s, ω_s^* and ω_c, ω_c^*.

$$\omega_s = \rho_s k_s \beta_s \delta_s\left(1 + h\beta_s\right)$$

$$\omega_s^* = a_o^2 \frac{\left(\rho_s^2 - 1\right)\left(\rho_s - 1\right)}{4} - \omega_s$$

$$\omega_c = \rho_c k_c \beta_c \delta_c\left(1 + h\beta_c\right)$$

$$\omega_c^* = a_o^2\left[\frac{\left(\rho_c^2 + 1\right)\left(\rho_c - 1\right)}{4} - \frac{\left(\rho_s - 1\right)}{2}\right] - \omega_c$$

(b) Calculate γ_b.

Configurations a, A, and D:

$$\gamma_b = 0$$

Configurations b and B:

$$\gamma_b = \frac{G_c - C}{D_o}$$

Configurations c and C:

$$\gamma_b = \frac{G_c - G_1}{D_o}$$

Configuration d:

$$\gamma_b = \frac{G_c - G_s}{D_o}$$

Configuration e:

$$\gamma_b = \frac{C - G_s}{D_o}$$

Configuration f:

$$\gamma_b = \frac{G_1 - G_s}{D_o}$$

UHX-14.5.6 Step 6. For each loading case, calculate the effective pressure P_e.

For an exchanger with an immersed floating head [Figure UHX-14.1, sketch (a)]: $P_e = P_s - P_t$

For an exchanger with an externally sealed floating head [Figure UHX-14.1, sketch (b)]: $P_e = P_s\,(1 - \rho_s^2) - P_t$

For an exchanger with an internally sealed floating tubesheet [Figure UHX-14.1, sketch (c)]: $P_e = (P_s - P_t)(1 - \rho_s^2)$

UHX-14.5.7 Step 7. For each loading case, calculate Q_2.

$$Q_2 = \frac{\left(\omega_s^* P_s - \omega_c^* P_t\right) + \frac{\gamma_b}{2\pi} W^*}{1 + \Phi Z_m}$$

For each loading case, calculate the maximum bending stress in the tubesheet in accordance with (a) or (b) below.

(a) When $P_e \neq 0$:

(1) Calculate Q_3.

$$Q_3 = Q_1 + \frac{2Q_2}{P_e a_o^2}$$

(2) For each loading case, determine coefficient F_m from either Table UHX-13.1 or Figures UHX-13.5.7-1 and UHX-13.5.7-2 and calculate the maximum bending stress σ.

$$\sigma = \left(\frac{1.5F_m}{\mu^*}\right)\left(\frac{2a_o}{h-h'_g}\right)^2 P_e$$

(b) When $P_e = 0$, calculate the maximum bending stress σ.

$$\sigma = \frac{6Q_2}{\mu^*\left(h-h'_g\right)^2}$$

For the design loading cases, if $|\sigma| \leq 1.5S$, and for the operating loading cases, if $|\sigma| \leq S_{PS}$, the assumed tubesheet thickness is acceptable for bending. Otherwise, increase the assumed tubesheet thickness h and return to UHX-14.5.1 (Step 1).

Configurations a, b, c, d, e, and f: Proceed to UHX-14.5.8 (Step 8).

Configuration A: Proceed to UHX-14.5.10 (Step 10).

Configurations B, C, and D: The calculation procedure is complete.

(19) **UHX-14.5.8 Step 8.** For each loading case, calculate the average shear stress in the tubesheet at the outer edge of the perforated region, if required.

(a) If $\left|P_e\right| \leq \dfrac{1.6S\mu h}{a_o}$, the shear stress is not required to be calculated. Proceed to UHX-14.5.9.

(b) Calculate the average shear stress, τ.

$$\tau = \left(\frac{1}{4\mu}\right)\left(\frac{1}{h}\left\{\frac{4A_p}{C_p}\right\}\right)P_e$$

If $|\tau| \leq \text{MIN}[0.8S, 0.533S_y]$, the assumed tubesheet thickness is acceptable for shear. Otherwise, increase the assumed tubesheet thickness, h, and return to UHX-14.5.1.

UHX-14.5.9 Step 9. Perform this step for each loading case.

(a) Check the axial tube stress.

(1) For each loading case, determine coefficients $F_{t,\min}$ and $F_{t,\max}$ from Table UHX-13.2 and calculate the two extreme values of tube stress, $\sigma_{t,1}$ and $\sigma_{t,2}$. The values for $\sigma_{t,1}$ and $\sigma_{t,2}$ may be positive or negative.

(-a) When $P_e \neq 0$:

$$\sigma_{t,1} = \frac{1}{x_t - x_s}\left[\left(P_s x_s - P_t x_t\right) - P_e F_{t,\min}\right]$$

$$\sigma_{t,2} = \frac{1}{x_t - x_s}\left[\left(P_s x_s - P_t x_t\right) - P_e F_{t,\max}\right]$$

(-b) When $P_e = 0$:

$$\sigma_{t,1} = \frac{1}{x_t - x_s}\left[\left(P_s x_s - P_t x_t\right) - \frac{2Q_2}{a_o^2}F_{t,\min}\right]$$

$$\sigma_{t,2} = \frac{1}{x_t - x_s}\left[\left(P_s x_s - P_t x_t\right) - \frac{2Q_2}{a_o^2}F_{t,\max}\right]$$

(2) Determine $\sigma_{t,\max} = \text{MAX}(|\sigma_{t,1}|, |\sigma_{t,2}|)$.

For the design loading cases, if $\sigma_{t,\max} > S_t$, and for the operating loading cases, if $\sigma_{t,\max} > 2S_t$, reconsider the tube design and return to UHX-14.5.1 (Step 1).

(b) Check the tube-to-tubesheet joint design.

(1) Calculate the largest tube-to-tubesheet joint load, W_t.

$$W_t = \sigma_{t,\max}\,\pi(d_t - t_t)t_t$$

(2) Determine the maximum allowable load for the tube-to-tubesheet joint design, $L_{\max}$. For tube-to-tubesheet joints with full strength welds, $L_{\max}$ shall be determined in accordance with UW-20. For tube-totubesheet joints with partial strength welds, $L_{\max}$ shall be in accordance with UW-18(d), UW-20, or Nonmandatory Appendix A, as applicable. For all other tube joints, $L_{\max}$ shall be determined in accordance with Nonmandatory Appendix A.

If $W_t > L_{\max}$, reconsider the tube-to-tubesheet joint design.

If $W_t \leq L_{\max}$, tube-to-tubesheet joint design is acceptable.

If $\sigma_{t,1}$ or $\sigma_{t,2}$ is negative, proceed to (c) below.

If $\sigma_{t,1}$ and $\sigma_{t,2}$ are positive, the tube design is acceptable. Proceed to UHX-14.5.10 (Step 10).

(c) Check the tubes for buckling.

(1) Calculate the largest equivalent unsupported buckling length of the tube ℓ_t considering the unsupported tube spans ℓ and their corresponding method of support k.

$$\ell_t = k\,\ell$$

(2) Calculate r_t, F_t, and C_t.

$$r_t = \frac{\sqrt{d_t^2 + (d_t - 2t_t)^2}}{4}$$

$$F_t = \frac{\ell_t}{r_t}$$

$$C_t = \sqrt{\frac{2\pi^2 E_t}{S_{y,t}}}$$

(3) Determine the factor of safety F_s in accordance with (-a) or (-b) below:

(-a) When $P_e \neq 0$,

$$F_s = \mathrm{MAX}\left\{\left[3.25 - 0.25(Z_d + Q_3 Z_w) X_a{}^4\right], (1.25)\right\}$$

F_s need not be taken greater than 2.0.

(-b) When $P_e = 0$, $F_s = 1.25$

(4) Determine the maximum permissible buckling stress limit S_{tb} for the tubes in accordance with (-a) or (-b) below:

(-a) When $C_t \leq F_t$,

$$S_{tb} = \mathrm{MIN}\left\{\left[\frac{1}{F_s}\frac{\pi^2 E_t}{F_t^2}\right], [S_t]\right\}$$

(-b) When $C_t > F_t$,

$$S_{tb} = \mathrm{MIN}\left\{\left[\frac{S_{y,t}}{F_s}\left(1 - \frac{F_t}{2\,C_t}\right)\right], [S_t]\right\}$$

(5) Determine $\sigma_{t,\min} = \mathrm{MIN}\ (\sigma_{t,1}, \sigma_{t,2})$.

If $|\sigma_{t,\min}| > S_{tb}$, reconsider the tube design and return to UHX-14.5.1 (Step 1).

If $|\sigma_{t,\min}| \leq S_{tb}$, the tube design is acceptable. Proceed to UHX-14.5.10 (Step 10).

UHX-14.5.10 Step 10. For each loading case, calculate the stresses in the shell and/or channel integral with the tubesheet.

Configurations a, b, and c: The shell shall have a uniform thickness of t_s for a minimum length of $1.8\sqrt{D_s t_s}$ adjacent to the tubesheet. Calculate the axial membrane stress $\sigma_{s,m}$, axial bending stress $\sigma_{s,b}$, and total axial stress σ_s in the shell at its junction to the tubesheet.

$$\sigma_{s,m} = \frac{a_o^2}{t_s(D_s + t_s)}\left[P_e + \left(\rho_s^2 - 1\right)(P_s - P_t)\right] + \frac{a_s^2}{t_s(D_s + t_s)}P_t$$

$$\sigma_{s,b} = \frac{6}{t_s^2}k_s\left\{\beta_s\delta_s P_s + \frac{6(1 - \nu^{*2})}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1 + \frac{h\beta_s}{2}\right) \times \left[P_e\left(Z_v + Z_m Q_1\right) + \frac{2}{a_o^2}Z_m Q_2\right]\right\}$$

$$\sigma_s = \left|\sigma_{s,m}\right| + \left|\sigma_{s,b}\right|$$

Configurations a, e, f, and A: A cylindrical channel shall have a uniform thickness of t_c for a minimum length of $1.8\sqrt{D_c t_c}$ adjacent to the tubesheet. Calculate the axial membrane stress $\sigma_{c,m}$, axial bending stress $\sigma_{c,b}$, and total axial stress σ_c, in the channel at its junction to the tubesheet.

$$\sigma_{c,m} = \frac{a_c^2}{t_c(D_c + t_c)}P_t$$

$$\sigma_{c,b} = \frac{6}{t_c^2}k_c\left\{\beta_c\delta_c P_t - \frac{6(1 - \nu^{*2})}{E^*}\left(\frac{a_o^3}{h^3}\right)\left(1 + \frac{h\beta_c}{2}\right) \times \left[P_e\left(Z_v + Z_m Q_1\right) + \frac{2}{a_o^2}Z_m Q_2\right]\right\}$$

$$\sigma_c = \left|\sigma_{c,m}\right| + \left|\sigma_{c,b}\right|$$

Configuration a: For the design loading cases, if $\sigma_s \leq 1.5S_s$ and $\sigma_c \leq 1.5S_c$, and for the operating loading cases, if $\sigma_s \leq S_{PS,s}$ and $\sigma_c \leq S_{PS,c}$, the shell and channel designs are acceptable, and the calculation procedure is complete. Otherwise, proceed to UHX-14.5.11 (Step 11).

Configurations b and c: For the design loading cases, if $\sigma_s \leq 1.5S_s$, and for the operating loading cases, if $\sigma_s \leq S_{PS,s}$, the shell design is acceptable, and the calculation procedure is complete. Otherwise, proceed to UHX-14.5.11 (Step 11).

Configurations e, f, and A: For the design loading cases, if $\sigma_c \leq 1.5S_c$, and for the operating loading cases, if $\sigma_c \leq S_{PS,c}$, the channel design is acceptable and the calculation procedure is complete. Otherwise, proceed to UHX-14.5.11 (Step 11).

UHX-14.5.11 Step 11. The design shall be reconsidered by using one or a combination of the following three options.

UHX-14.5.11.1 Option 1. Increase the assumed tubesheet thickness h and return to UHX-14.5.1 (Step 1).

UHX-14.5.11.2 Option 2. Increase the integral shell and/or channel thickness as follows:

Configurations a, b, and c: If $\sigma_s > 1.5S_s$, increase the shell thickness t_s and return to UHX-14.5.1 (Step 1).

Configurations a, e, f, and A: If $\sigma_c > 1.5S_c$, increase the channel thickness t_c and return to UHX-14.5.1 (Step 1).

UHX-14.5.11.3 Option 3. Perform the elastic–plastic calculation procedure as defined in UHX-14.8 only when the conditions of applicability stated in UHX-14.8.2 are satisfied.

UHX-14.6 Calculation Procedure for Effect of Radial Thermal Expansion Adjacent to the Tubesheet

UHX-14.6.1 Scope.

(a) This procedures describes how to use the rules of UHX-14.5 when the effect of radial differential thermal expansion between the tubesheet and integral shell or channel is to be considered.

(b) This procedure shall be used when cyclic or dynamic reactions due to pressure or thermal variations are specified [see UG-22(e)].

(c) This procedure shall be used when specified by the user or his designated agent. The user or his designated agent shall provide the Manufacturer with the data necessary to determine the required tubesheet, channel, and shell metal temperatures.

(d) Optionally, the designer may use this procedure to consider the effect of radial differential thermal expansion even when it is not required by (b) or (c) above.

UHX-14.6.2 Conditions of Applicability. This calculation procedure applies only when the tubesheeet is integral with the shell or channel (Configurations a, b, c, e, f, and A).

(19) **UHX-14.6.3 Additional Nomenclature.**

T' = tubesheet metal temperature at the rim (see Figure UHX-11.3-3)
T'_c = channel metal temperature at the tubesheet
T'_{cx} = channel metal temperature at the tubesheet for operating condition x
T'_s = shell metal temperature at the tubesheet
T'_{sx} = shell metal temperature at the tubesheet for operating condition x
T'_x = tubesheet metal temperature at the rim for operating condition x
α' = mean coefficient of thermal expansion of tubesheet material at T'
α'_c = mean coefficient of thermal expansion of channel material at T'_c
α'_s = mean coefficient of thermal expansion of shell material at T'_s

UHX-14.6.4 Calculation Procedure. The calculation procedure given in UHX-14.5 shall be performed for the operating loading cases accounting for the modifications in (a) through (e).

Table UHX-14.6.4-1 provides the load combinations required to evaluate the heat exchanger for each operating condition x.

(a) Determine the average temperature of the unperforated rim T_r.

Configuration a:

$$T_r = \frac{T' + T'_s + T'_c}{3}$$

Configurations b and c:

$$T_r = \frac{T' + T'_s}{2}$$

Configurations e, f, and A:

$$T_r = \frac{T' + T'_c}{2}$$

For conservative values of P_s^* and P_c^*, $T_r = T'$ may be used.

(b) Determine the average temperature of the shell T_s^* and channel T_c^* at their junction to the tubeshet as follows:

Configurations a, b, and c:

$$T_s^* = \frac{T'_s + T_r}{2}$$

Configurations a, e, f, and A:

$$T_c^* = \frac{T'_c + T_r}{2}$$

For conservative values of P_s^* and P_c^*, $T_s^* = T'_s$ and $T_c^* = T'_c$ may be used.

(c) Calculate P_s^* and P_c^*.

Configurations a, b, and c:

$$P_s^* = \frac{E_s t_s}{a_s}\left[\alpha'_s\left(T_s^* - T_a\right) - \alpha'\left(T_r - T_a\right)\right]$$

Configurations e, f, and A:

$$P_s^* = 0$$

Configurations a, e, f, and A:

$$P_c^* = \frac{E_c t_c}{a_c}\left[\alpha'_c\left(T_c^* - T_a\right) - \alpha'\left(T_r - T_a\right)\right]$$

Configurations b and c:

$$P_c^* = 0$$

Table UHX-14.6.4-1

Operating Loading Case	Operating Pressure		Metal Temperature		
	Shell Side, P_s	Tube Side, P_t	Tubesheet at the Rim, T'	Channel at Tubesheet, T'_c	Shell at Tubesheet, T'_s
1	$P_{sox,\min}$	$P_{tox,\max}$	T'_x	T'_{cx}	T'_{sx}
2	$P_{sox,\max}$	$P_{tox,\min}$	T'_x	T'_{cx}	T'_{sx}
3	$P_{sox,\max}$	$P_{tox,\max}$	T'_x	T'_{cx}	T'_{sx}
4	$P_{sox,\min}$	$P_{tox,\min}$	T'_x	T'_{cx}	T'_{sx}

(d) In UHX-14.5.7 (Step 7), replace the formula for Q_2 with:

$$Q_2 = \frac{\left(\omega_s^* P_s - \omega_c^* P_t\right) - \left(\omega_s P_s^* - \omega_c P_c^*\right) + \frac{\gamma_b}{2\pi} W^*}{1 + \Phi Z_m}$$

(e) In UHX-14.5.10 (Step 10), replace the equations for $\sigma_{s,b}$ and $\sigma_{c,b}$ with:

$$\sigma_{s,b} = \frac{6}{t_s^2} k_s \left\{ \beta_s \left[\delta_s P_s + \frac{a_s^2}{E_s t_s} P_s^* \right] + \frac{6\left(1 - \nu^{*2}\right)}{E^*} \left(\frac{a_o^3}{h^3} \right) \right.$$
$$\left. \times \left(1 + \frac{h \beta_s}{2} \right) \left[P_e \left(Z_v + Z_m Q_1 \right) + \frac{2}{a_o^2} Z_m Q_2 \right] \right\}$$

$$\sigma_{c,b} = \frac{6}{t_c^2} k_c \left\{ \beta_c \left[\delta_c P_t + \frac{a_c^2}{E_c t_c} P_c^* \right] - \frac{6\left(1 - \nu^{*2}\right)}{E^*} \left(\frac{a_o^3}{h^3} \right) \right.$$
$$\left. \times \left(1 + \frac{h \beta_c}{2} \right) \left[P_e \left(Z_v + Z_m Q_1 \right) + \frac{2}{a_o^2} Z_m Q_2 \right] \right\}$$

UHX-14.7 Calculation Procedure for Simply Supported Floating Tubesheets

UHX-14.7.1 Scope. This procedure describes how to use the rules of UHX-14.5 when the effect of the stiffness of the integral channel and/or shell is not considered.

UHX-14.7.2 Conditions of Applicability. This calculation procedure applies only when the tubesheet is integral with the shell or channel (configurations a, b, c, e, f, and A).

UHX-14.7.3 Calculation Procedure. The calculation procedure outlined in UHX-14.5 shall be performed accounting for the following modifications.

(a) Perform the steps in UHX-14.5.1 through UHX-14.5.9.

(b) Perform the step in UHX-14.5.10 except as follows:

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

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

(3) Configuration a: If $\sigma_s \le S_{PS,s}$ and $\sigma_c \le S_{PS,c}$, then the shell and channel are acceptable. Otherwise, increase the thickness of the overstressed component(s) (shell and/or channel) and return to UHX-14.5.1 (Step 1).

Configurations b and c: If $\sigma_s \le S_{PS,s}$, then the shell is acceptable. Otherwise, increase the thickness of the shell and return to UHX-14.5.1 (Step 1).

Configurations e, f, and A: If $\sigma_c \le S_{PS,c}$, then the channel is acceptable. Otherwise increase the thickness of the channel and return to UHX-14.5.1 (Step 1).

(c) Do not perform UHX-14.5.11 (Step 11).

(d) Repeat the steps in UHX-14.5.1 through UHX-14.5.7 for the design loading cases, with the following changes to UHX-14.5.2 (Step 2), until the tubesheet stress criteria have been met:

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

Configurations a, e, f, and A: $\beta_c = 0$, $k_c = 0$, $\lambda_c = 0$, $\delta_c = 0$.

UHX-14.8 Calculation Procedure for Effect of Plasticity at Tubesheet/Channel or Shell Joint (19)

UHX-14.8.1 Scope. This procedure describes how to use the rules of UHX-14.5 when the effect of plasticity at the shell–tubesheet and/or channel–tubesheet joint is to be considered.

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

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

UHX-14.8.2 Conditions of Applicability.

(a) This procedure shall not be used at temperatures where the time-dependent properties govern the allowable stress.

(b) This procedure applies only for loading cases 1, 2, and 3.

(c) This procedure applies to Configuration a when $\sigma_s \le S_{PS,c}$ and $\sigma_s \le S_{PS,c}$.

(d) This procedure applies to Configurations b and c when $\sigma_s \le S_{PS,s}$.

(e) This procedure applies to Configurations e, f, and A when $\sigma_c \le S_{PS,c}$.

(f) This procedure may only be used once for each iteration of tubesheet, shell, and channel thicknesses and materials.

UHX-14.8.3 Additional Nomenclature.

fact_c = factor used in the elastic–plastic analysis to account for any yielding of the channel

fact_s = factor used in the elastic–plastic analysis to account for any yielding of the shell

UHX-14.8.4 Calculation Procedure. After the calculation procedure given in the steps in UHX-14.5.1 through UHX-14.5.10 has been performed for the elastic solution, an elastic–plastic calculation using the referenced steps from UHX-14.5 shall be performed in accordance with the following procedure for each applicable loading case. Except for those quantities modified below, the quantities to be used for the elastic–plastic calculation shall be the same as those calculated for the corresponding elastic loading case.

(a) Define the maximum permissible bending stress limit in the shell and channel.

Configurations a, b, and c:

$$S_S^* = \text{MIN}\left[\left(S_{y,s}\right),\left(\frac{S_{PS,s}}{2}\right)\right]$$

Configurations a, e, f, and A:

$$S_C^* = \text{MIN}\left[\left(S_{y,c}\right),\left(\frac{S_{PS,c}}{2}\right)\right]$$

(b) Using bending stresses $\sigma_{s,b}$ and $\sigma_{c,b}$ computed in the step in UHX-14.5.10 for the elastic solution, determine fact_s and fact_c as follows:

Configurations a, b, and c:

$$\text{fact}_S = \text{MIN}\left[\left(1.4 - 0.4\frac{\left|\sigma_{s,b}\right|}{S_S^*}\right),(1.0)\right]$$

Configurations a, e, f, and A:

$$\text{fact}_C = \text{MIN}\left[\left(1.4 - 0.4\frac{\left|\sigma_{c,b}\right|}{S_C^*}\right),(1.0)\right]$$

Configuration a: If fact_s = 1.0 and fact_c = 1.0, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to (c) below.

Configurations b and c: If fact_s = 1.0, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to (c) below.

Configurations e, f, and A: If fact_c = 1.0, the design is acceptable, and the calculation procedure is complete. Otherwise, proceed to (c) below.

(c) Calculate reduced values of E_s and E_c as follows:

Configurations a, b, and c: $E_S^* = E_s\ \text{fact}_s$

Configurations a, e, f, and A: $E_C^* = E_c\ \text{fact}_c$

(d) In Step 2, recalculate k_s, λ_s, k_c, and λ_c replacing E_s by E_S^* and E_c by E_C^*.

(e) In Step 4, recalculate F, Φ, and Q_1.

(f) In Step 7, recalculate Q_2, Q_3, and F_m, as applicable, and the tubesheet bending stress, σ.

If $|\sigma| \le 1.5S$, the design is acceptable and the calculation procedure is complete. Otherwise, the unit geometry shall be reconsidered.

UHX-16 BELLOWS EXPANSION JOINTS

Bellows expansion joints shall be designed in accordance with Mandatory Appendix 26, as applicable. The expansion joint shall be designed for the axial displacement range over all load cases from one of the following equations for the axial displacement over the length of the thin-walled bellows element. Note that these may be used for flanged-and-flued or flanged-only expansion joints when the expansion joint analysis method uses the displacement over the expansion element only [see UHX-17(c)].

(a) For heat exchangers with constant shell thickness and material, use the following equation:

$$\Delta_J = \frac{\sigma_{s,m}\left[t_s\left(D_s + t_s\right)\pi\right]}{JK_s} + \frac{\pi}{8}\frac{D_J^2 - D_S^2}{K_J}P_s$$

(b) For heat exchangers that have a different shell thickness and/or material adjacent to the tubesheet per UHX-13.6, use the following equation:

$$\Delta_J = \frac{\sigma_{s,m}\left[t_{s,1}\left(D_s + t_{s,1}\right)\pi\right]}{JK_S^*} + \frac{\pi}{8}\frac{D_J^2 - D_S^2}{K_J}P_s$$

Table UHX-17
Flexible Shell Element Expansion Joint Load Cases and Stress Limits

Loading Case	Shell Side Pressure, P_s	Tube Side Pressure, P_t	Differential Thermal Expansion	Maximum Stress: Membrane — Corners and Torus	Maximum Stress: Membrane Plus Bending — Corners and Torus	Maximum Stress: Membrane Plus Bending — Annular Plates	Maximum Stress: Membrane Plus Bending — Straight Flanges
Design							
1	$P_{sd,\min}$	$P_{td,\max}$	No	1.5S	S_{PS}	1.5S	1.5S
2	$P_{sd,\max}$	$P_{td,\min}$	No	1.5S	S_{PS}	1.5S	1.5S
3	$P_{sd,\max}$	$P_{td,\max}$	No	1.5S	S_{PS}	1.5S	1.5S
4	$P_{sd,\min}$	$P_{td,\min}$	No	1.5S	S_{PS}	1.5S	1.5S
Operating							
1	$P_{sox,\min}$	$P_{tox,\max}$	Yes	S_{PS}	S_{PS}	S_{PS}	S_{PS}
2	$P_{sox,\max}$	$P_{tox,\min}$	Yes	S_{PS}	S_{PS}	S_{PS}	S_{PS}
3	$P_{sox,\max}$	$P_{tox,\max}$	Yes	S_{PS}	S_{PS}	S_{PS}	S_{PS}
4	$P_{sox,\min}$	$P_{tox,\min}$	Yes	S_{PS}	S_{PS}	S_{PS}	S_{PS}

UHX-17 FLEXIBLE SHELL ELEMENT EXPANSION JOINTS

(a) Flexible shell element expansion joints shall be designed in accordance with Mandatory Appendix 5, as applicable.

(b) The higher stress limits shown in Table UHX-17 may be applied in lieu of the limits of 5-3(a). These limits allow the expansion joint to yield, which decreases its stiffness. All calculations shall be performed in both the corroded and noncorroded condition. To apply these limits, it shall be shown that

(1) the design of the other components of the heat exchanger (i.e., tubesheet, tubes, shell, channel, etc.) is acceptable considering the decreased stiffness of the expansion joint. This may be accomplished by performing an additional evaluation of all the components of the exchanger for design loading cases 1 through 4 (when $P_{sd,\min}$ and $P_{td,\min}$ are both zero, design loading case 4 does not need to be considered) with zero expansion joint stiffness. In UHX-13, this may be accomplished by replacing the Step 6 formula for P_e with

$$P_e = \left[1 - \frac{1}{2}\left(\rho_S{}^2 + \frac{D_J{}^2}{D_o{}^2}\right)\right] P_S - P_t$$

(2) the rotational stiffness at the expansion joint corners and torus is not necessary to meet the stress limits for annular plates and straight flanges for the design loading cases shown in Table UHX-17. This may be accomplished by modeling the corners and torus as simply supported to determine the stress in the annular plates and straight flanges.

(c) Displacements arising from pressure and differential thermal expansion shall be calculated for use in the expansion joint analysis. The length over which the displacement is taken is dependent upon the expansion joint analysis method. If the expansion joint analysis method utilizes displacements over the length of the expansion joint only, use the appropriate equation from UHX-16. If the expansion joint analysis method utilizes displacements over the length between the inner tubesheet faces, L, use the appropriate equation from below.

(1) For heat exchangers with a constant shell thickness and material, use one of the following:

(-a) If the expansion joint analysis includes thermal expansion effects

$$\Delta_S^T = \frac{\sigma_{s,m}\left[t_s\left(D_s + t_s\right)\pi\right]}{JK_s} + L\alpha_{s,m}\left(T_{s,m} - T_a\right) - \frac{\pi D_S^2 P_s}{2K_s}\nu_s + \frac{\pi}{8}\frac{D_J^2 - D_S^2}{K_J}P_s$$

(-b) If the expansion joint analysis does not include thermal expansion effects

$$\Delta_S^M = \frac{\sigma_{s,m}\left[t_s\left(D_s + t_s\right)\pi\right]}{JK_s} - \frac{\pi D_S^2 P_s}{2K_s}\nu_s + \frac{\pi}{8}\frac{D_J^2 - D_S^2}{K_J}P_s$$

(2) For heat exchangers that have a different shell thickness and/or material adjacent to the tubesheet per UHX-13.6, use one of the following:

(-a) If the expansion joint analysis includes thermal expansion effects

$$\Delta_S^T = \frac{\sigma_{s,m}\left[t_{s,1}\left(D_s + t_{s,1}\right)\pi\right]}{JK_S^*} + \left[\left(L - \ell_1 - \ell_1'\right)\alpha_{s,m} + \left(\ell_1 + \ell_1'\right)\alpha_{s,m,1}\right]\left(T_{s,m} - T_a\right) - \frac{\pi D_S^2 P_s}{2K_S^*}\nu_s + \frac{\pi}{8}\frac{D_J^2 - D_S^2}{K_J}P_s$$

(-b) If the expansion joint analysis does not include thermal expansion effects

$$\Delta_s^M = \frac{\sigma_{s,m}\left[t_{s,1}\left(D_s + t_{s,1}\right)\pi\right]}{JK_s^*} - \frac{\pi D_s^2 P_s}{2K_s^*} v_s + \frac{\pi}{8} \frac{D_J^2 - D_s^2}{K_J} P_s$$

UHX-18 PRESSURE TEST REQUIREMENTS

(a) The shell side and the tube side of the heat exchanger shall be subjected to a pressure test in accordance with UG-99 or UG-100.

(b) Shipping bars on bellows expansion joints may be required to maintain assembly length during shipment and vessel fabrication. Shipping bars shall not be engaged or otherwise provide any restraint of the expansion joint during vessel pressure testing and operation [see 26-4.1(c) and 26-4.1(d)].

UHX-19 HEAT EXCHANGER MARKING AND REPORTS

UHX-19.1 Required Marking

The marking of heat exchangers shall be in accordance with UG-116 using the specific requirements of UG-116(j) for combination units (multi-chamber vessels). When the markings are grouped in one location in accordance with requirements of UG-116(j)(1) and abbreviations for each chamber are used, they shall be as follows:

(a) For markings in accordance with UG-116(a)(3) and UG-116(a)(4), the chambers shall be abbreviated as:

(1) SHELL for shell side

(2) TUBES for tube side

This abbreviation shall precede the appropriate design data. For example, use:

(3) SHELL FV&300 psi (FV&2000 kPa) at 500°F (260°C) for the shell side maximum allowable working pressure

(4) TUBES 150 psi (1 000 kPa) at 350°F (175°C) for the tube side maximum allowable working pressure

(b) When the markings in accordance with UG-116(b)(1), UG-116(c), UG-116(e) and UG-116(f) are different for each chamber, the chambers shall be abbreviated as:

(1) S for shell side

(2) T for tube side

This abbreviation shall follow the appropriate letter designation and shall be separated by a hyphen. For example, use:

(3) L-T for lethal service tube side

(4) RT 1-S for full radiography on the shell side

UHX-19.2 Supplemental Marking

A supplemental tag or marking shall be supplied on the heat exchanger to caution the user if there are any restrictions on the design, testing, or operation of the heat exchanger. The marking shall meet the requirements of UG-118 or UG-119, except that height of the characters for the caution required by UHX-19.2.2 shall be at least $^1/_8$ in. (3 mm) high. Supplemental marking shall be required for, but not limited to, the following:

UHX-19.2.1 Common Elements. Shell-and-tube heat exchangers are combination units as defined in UG-19(a) and the tubes and tubesheets are common elements. The following marking is required when the common elements are designed for conditions less severe than the design conditions for which its adjacent chambers are stamped.

(a) Differential Pressure Design. When common elements such as tubes and tubesheets are designed for a differential design pressure, the heat exchanger shall be marked "Differential Design" in addition to meeting all the requirements of UG-19(a)(2) [see UG-116(j)]. If the tubes and tubesheets are designed for a differential pressure of 150 psi, an example of the marking would be

DIFFERENTIAL DESIGN: TUBES
& TUBESHEETS 150 psi

(b) Mean Metal Temperature Design. When common elements such as tubes and tubesheets are designed for a maximum mean metal design temperature that is less than the maximum of the shell side and tube side design temperatures, the heat exchanger shall be marked "Max Mean Metal Temp" in addition to meeting all the requirements of UG-19(a)(3) [see UG-116(j)]. If the tubes are designed for a maximum mean metal temperature of 400°F, an example of the marking would be

MAX MEAN METAL TEMP: TUBES 400°F

UHX-19.2.2 Fixed Tubesheet Heat Exchangers. Fixed tubesheet heat exchangers shall be marked with the following caution:

CAUTION: The heat exchanger design has been evaluated for the range of conditions listed on Form U-5 of the MDR. It shall be reevaluated for conditions outside this range before being operated at them.

UHX-19.3 Manufacturer's Data Reports

UHX-19.3.1 Common Elements. When common elements such as tubes and tubesheets are designed for a differential pressure, or a mean metal temperature, or both, that is less severe than the design conditions for which its adjacent chambers are stamped, the data for each common element that differs from the data for the corresponding chamber shall be indicated as required by UG-19(a) and UG-120(b) in the "Remarks" section of the Manufacturer's Data Report.

UHX-19.3.2 Fixed Tubesheet Heat Exchangers. For each design and operating condition, the following information shall be indicated on Form U-5 of the Manufacturer's Data Report Supplementary Sheet for Shell-and-Tube Heat Exchangers. The operating conditions may be combined on this form where they are bounded by the operating pressure range, maximum metal temperatures, and axial differential thermal expansion range.

(a) Name of Condition. The first condition shown shall be the design condition. If there is more than one design condition or a differential pressure design condition, multiple lines may be used. Each different operating condition or range of operating conditions shall be listed.

(b) Design/Operating Pressure Ranges. Range of shell side and tube side pressures for each condition shall be listed.

(c) Design/Operating Metal Temperatures. For each condition, the temperature at which the allowable stress was taken for the shell, channel, tube, and tubesheet shall be listed. Any metal temperature between the MDMT and the listed temperature is permitted, provided the resulting axial differential thermal expansion is within the listed range.

(d) Axial Differential Thermal Expansion Range. The minimum and maximum axial differential thermal expansion for each operating condition shall be listed. If the minimum value is positive, zero shall be used for the minimum value. If the maximum value is negative, zero shall be used for the maximum value. Within the listed range of operating temperature and pressure, any combination of shell and tube axial mean metal temperatures is permitted, provided the resulting axial differential thermal expansion is within the listed range.

UHX-20 EXAMPLES

See UG-16(f).

MANDATORY APPENDIX 2
RULES FOR BOLTED FLANGE CONNECTIONS WITH RING TYPE GASKETS

(19) 2-1 SCOPE

(a) The rules in Mandatory Appendix 2 apply specifically to the design of bolted flange connections with gaskets that are entirely within the circle enclosed by the bolt holes and with no contact outside this circle, and are to be used in conjunction with the applicable requirements in Subsections A, B, and C of this Division. The hub thickness of weld neck flanges designed to this Appendix shall also comply with the minimum thickness requirements in Subsection A of this Division. These rules are not to be used for the determination of the thickness of tubesheets integral with a bolting flange as illustrated in Figure UW-13.2, sketches (h) through (l) or Figure UW-13.3, sketch (c). Nonmandatory Appendix S provides discussion on Design Considerations for Bolted Flanged Connections.

These rules provide only for hydrostatic end loads and gasket seating. The flange design methods outlined in 2-4 through 2-8 are applicable to circular flanges under internal pressure. Modifications of these methods are outlined in 2-9 and 2-10 for the design of split and noncircular flanges. See 2-11 for flanges with ring type gaskets subject to external pressure, 2-12 for flanges with nut-stops, and 2-13 for reverse flanges. Rules for calculating rigidity factors for flanges are provided in 2-14. Recommendations for qualification of assembly procedures and assemblers are in 2-15. Proper allowance shall be made if connections are subject to external loads other than external pressure.

(b) The design of a flange involves the selection of the gasket (material, type, and dimensions), flange facing, bolting, hub proportions, flange width, and flange thickness. See Note in 2-5(c)(1). Flange dimensions shall be such that the stresses in the flange, calculated in accordance with 2-7, do not exceed the allowable flange stresses specified in 2-8. Except as provided for in 2-14(a), flanges designed to the rules of this Appendix shall also meet the rigidity requirements of 2-14. All calculations shall be made on dimensions in the corroded condition.

(c) It is recommended that bolted flange connections conforming to the standards listed in UG-44(a) be used for connections to external piping. These standards may be used for other bolted flange connections and dished covers within the limits of size in the standards and the pressure–temperature ratings permitted in UG-44(a). The ratings in these standards are based on the hub dimensions given or on the minimum specified thickness of flanged fittings of integral construction. Flanges fabricated from rings may be used in place of the hub flanges in these standards provided that their strength, calculated by the rules in this Appendix, is not less than that calculated for the corresponding size of hub flange.

(d) Except as otherwise provided in (c) above, bolted flange connections for unfired pressure vessels shall satisfy the requirements in this Appendix.

(e) The rules of this Appendix should not be construed to prohibit the use of other types of flanged connections, provided they are designed in accordance with good engineering practice and method of design is acceptable to the Inspector. Some examples of flanged connections which might fall in this category are as follows:

(1) flanged covers as shown in Figure 1-6;

(2) bolted flanges using full-face gaskets;

(3) flanges using means other than bolting to restrain the flange assembly against pressure and other applied loads.

2-2 MATERIALS (19)

(a) Materials used in the construction of bolted flange connections shall comply with the requirements given in UG-4 through UG-14.

(b) Flanges made from ferritic steel and designed in accordance with this Appendix shall be full-annealed, normalized, normalized and tempered, or quenched and tempered when the thickness of the flange, t (see Figure 2-4), exceeds 3 in. (75 mm).

(c) Material on which welding is to be performed shall be proved of good weldable quality. Satisfactory qualification of the welding procedure under Section IX is considered as proof. Welding shall not be performed on steel that has a carbon content greater than 0.35%. All welding on flange connections shall comply with the requirements for postweld heat treatment given in this Division.

(d) Flanges with hubs that are machined from plate, bar stock, or billet shall not be machined from plate or bar material [except as permitted in UG-14(b)] unless the material has been formed into a ring and the following additional conditions are met:

(1) In a ring formed from plate, the original plate surfaces are parallel to the axis of the finished flange. (This is not intended to imply that the original plate surface should be present in the finished flange.)

(2) The joints in the ring are welded butt joints that conform to the requirements of this Division. Thickness to be used to determine postweld heat treatment and radiography requirements shall be the lesser of

$$t \text{ or } \frac{(A-B)}{2}$$

where these symbols are as defined in 2-3.

(3) The back of the flange and the outer surface of the hub are examined by either the magnetic particle method as per Mandatory Appendix 6 or the liquid penetrant method as per Mandatory Appendix 8.

(e) Bolts, studs, nuts, and washers shall comply with the requirements in this Division. It is recommended that bolts and studs have a nominal diameter of not less than $^1/_2$ in. (13 mm). If bolts or studs smaller than $^1/_2$ in. (13 mm) are used, ferrous bolting material shall be of alloy steel. Precautions shall be taken to avoid overstressing small-diameter bolts.

(19) 2-3 NOTATION

The symbols described below are used in the equations for the design of flanges (see also Figure 2-4):

A = outside diameter of flange or, where slotted holes extend to the outside of the flange, the diameter to the bottom of the slots

a = nominal bolt diameter

A_b = cross-sectional area of the bolts using the root diameter of the thread or least diameter of unthreaded position, if less

A_m = total required cross-sectional area of bolts, taken as the greater of A_{m1} and A_{m2}

A_{m1} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the operating conditions
= W_{m1} / S_b

A_{m2} = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket seating
= W_{m2} / S_a

B = inside diameter of flange. When B is less than $20g_1$, it will be optional for the designer to substitute B_1 for B in the formula for longitudinal stress S_H.

b = effective gasket or joint-contact-surface seating width [see Note in 2-5(c)(1)]

B_1 = $B + g_1$ for loose type flanges and for integral type flanges that have calculated values h / h_o and g_1 / g_o which would indicate an f value of less than 1.0, although the minimum value of f permitted is 1.0.
= $B + g_o$ for integral type flanges when f is equal to or greater than one

b_o = basic gasket seating width (from Table 2-5.2)

B_s = bolt spacing. The bolt spacing may be taken as the bolt circle circumference divided by the number of bolts or as the chord length between adjacent bolt locations.

B_{sc} = bolt spacing factor

$B_{s\max}$ = maximum bolt spacing

C = bolt-circle diameter

c = basic dimension used for the minimum sizing of welds equal to t_n or t_x, whichever is less

C_b = conversion factor
= 0.5 for U.S. Customary calculations; 2.5 for SI calculations

d = factor
= $\frac{U}{V} h_o g_o^2$ for integral type flanges
= $\frac{U}{V_L} h_o g_o^2$ for loose type flanges

e = factor
= $\frac{F}{h_o}$ for integral type flanges
= $\frac{F_L}{h_o}$ for loose type flanges

F = factor for integral type flanges (from Figure 2-7.2)

f = hub stress correction factor for integral flanges from Figure 2-7.6 (When greater than one, this is the ratio of the stress in the small end of hub to the stress in the large end.) (For values below limit of figure, use f = 1.)

F_L = factor for loose type flanges (from Figure 2-7.4)

G = diameter at location of gasket load reaction. Except as noted in sketch (1) of Figure 2-4, G is defined as follows (see Table 2-5.2):

(a) when $b_o \leq {}^1/_4$ in. (6 mm), G = mean diameter of gasket contact face

(b) when $b_o > {}^1/_4$ in. (6 mm), G = outside diameter of gasket contact face less $2b$

g_1 = thickness of hub at back of flange

g_o = thickness of hub at small end

(a) for optional type flanges calculated as integral and for integral type flanges per Figure 2-4, sketch (7), $g_o = t_n$

(b) for other integral type flanges, g_o = the smaller of t_n or the thickness of the hub at the small end

H = total hydrostatic end force
= $0.785G^2P$

h = hub length

H_D = hydrostatic end force on area inside of flange
= $0.785B^2P$

h_D = radial distance from the bolt circle, to the circle on which H_D acts, as prescribed in Table 2-6

H_G = gasket load for the operating condition
= $W_{m1} - H$

h_G = radial distance from gasket load reaction to the bolt circle
= $(C - G)/2$

h_o = factor
= $\sqrt{Bg_o}$

H_p = total joint-contact surface compression load
= $2b \times 3.14\ GmP$

H_T = difference between total hydrostatic end force and the hydrostatic end force on area inside of flange
= $H - H_D$

h_T = radial distance from the bolt circle to the circle on which H_T acts as prescribed in Table 2-6

K = ratio of outside diameter of flange to inside diameter of flange
= A/B

L = factor
= $\dfrac{te + 1}{T} + \dfrac{t^3}{d}$

m = gasket factor, obtain from Table 2-5.1 [see Note in 2-5(c)(1)]

M_D = component of moment due to H_D,
= H_Dh_D

M_G = component of moment due to H_G,
= H_Gh_G

M_o = total moment acting upon the flange, for the operating conditions or gasket seating as may apply (see 12-4)

M_T = component of moment due to H_T
= H_Th_T

N = width used to determine the basic gasket seating with b_o, based upon the possible contact width of the gasket (see Table 2-5.2)

P = internal design pressure (see UG-21). For flanges subject to external design pressure, see 2-11.

R = radial distance from bolt circle to point of intersection of hub and back of flange. For integral and hub flanges,
= $\dfrac{C - B}{2} - g_1$

S_a = allowable bolt stress at atmospheric temperature (see UG-23)

S_b = allowable bolt stress at design temperature (see UG-23)

S_f = allowable design stress for material of flange at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply (see UG-23)

S_H = calculated longitudinal stress in hub

S_n = allowable design stress for material of nozzle neck, vessel or pipe wall, at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply (see UG-23)

S_R = calculated radial stress in flange

S_T = calculated tangential stress in flange

T = factor involving K (from Figure 2-7.1)

t = flange thickness

t_n = nominal thickness of shell or nozzle wall to which flange or lap is attached

t_x = two times the thickness g_0, when the design is calculated as an integral flange or two times the thickness of shell nozzle wall required for internal pressure, when the design is calculated as a loose flange, but not less than $^1/_4$ in. (6 mm)

U = factor involving K (from Figure 2-7.1)

V = factor for integral type flanges (from Figure 2-7.3)

V_L = factor for loose type flanges (from Figure 2-7.5)

W = flange design bolt load, for the operating conditions or gasket seating, as may apply [see 2-5(e)]

w = width used to determine the basic gasket seating width b_0, based upon the contact width between the flange facing and the gasket (see Table 2-5.2)

W_{m1} = minimum required bolt load for the operating conditions [see 2-5(c)]. For flange pairs used to contain a tubesheet for a floating head or a U-tube type of heat exchangers, or for any other similar design, W_{m1} shall be the larger of the values as individually calculated for each flange, and that value shall be used for both flanges.

W_{m2} = minimum required bolt load for gasket seating [see 2-5(c)]. For flange pairs used to contain a tubesheet for a floating head or U-tube type of heat exchanger, or for any other similar design where the flanges or gaskets are not the same, W_{m2} shall be the larger of the values calculated for each flange and that value shall be used for both flanges.

Y = factor involving K (from Figure 2-7.1)

y = gasket or joint-contact-surface unit seating load, [see Note 1, 2-5(c)]

Z = factor involving K (from Figure 2-7.1)

2-4 CIRCULAR FLANGE TYPES (19)

For purposes of computation, there are three types:

(a) Loose Type Flanges. This type covers those designs in which the flange has no direct connection to the nozzle neck, vessel, or pipe wall, and designs where the method of attachment is not considered to give the mechanical strength equivalent of integral attachment. See Figure 2-4, sketches (1), (1a), (2), (2a), (3), (3a), (4), (4a), (4b), and (4c) for typical loose type flanges and the location of the loads and moments. Welds and other details of

construction shall satisfy the dimensional requirements given in Figure 2-4, sketches (1), (1a), (2), (2a), (3), (3a), (4), (4a), (4b), and (4c).

(b) Integral Type Flanges. This type covers designs where the flange is cast or forged integrally with the nozzle neck, vessel or pipe wall, butt welded thereto, or attached by other forms of welding of such a nature that the flange and nozzle neck, vessel or pipe wall is considered to be the equivalent of an integral structure. In welded construction, the nozzle neck, vessel, or pipe wall is considered to act as a hub. See Figure 2-4, sketches (5), (6), (6a), (6b), and (7) for typical integral type flanges and the location of the loads and moments. Welds and other details of construction shall satisfy the dimensional requirements given in Figure 2-4, sketches (5), (6), (6a), (6b), and (7).

(c) Optional Type Flanges. This type covers designs where the attachment of the flange to the nozzle neck, vessel, or pipe wall is such that the assembly is considered to act as a unit, which shall be calculated as an integral flange, except that for simplicity the designer may calculate the construction as a loose type flange, provided none of the following values is exceeded:

$$g_o = {}^5\!/_8\ \text{in.}\,(16\ \text{mm})$$
$$B/g_o = 300$$
$$P = 300\ \text{psi}\,(2\ \text{MPa})$$
$$\text{operating temperature} = 700°\text{F}\,(370°\text{C})$$

See Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11) for typical optional type flanges. Welds and other details of construction shall satisfy the dimensional requirements given in Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11).

2-5 BOLT LOADS

(a) General Requirements

(1) In the design of a bolted flange connection, calculations shall be made for each of the two design conditions of operating and gasket seating, and the more severe shall control.

(2) In the design of flange pairs used to contain a tubesheet of a heat exchanger or any similar design where the flanges and/or gaskets may not be the same, loads must be determined for the most severe condition of operating and/or gasket seating loads applied to each side at the same time. This most severe condition may be gasket seating on one flange with operating on the other, gasket seating on each flange at the same time, or operating on each flange at the same time. Although no specific rules are given for the design of the flange pairs, after the loads for the most severe conditions are determined, calculations shall be made for each flange following the rules of Mandatory Appendix 2.

(3) Recommended minimum gasket contact widths for sheet and composite gaskets are provided in Table 2-4.

(b) Design Conditions

(1) Operating Conditions. The conditions required to resist the hydrostatic end force of the design pressure tending to part the joint, and to maintain on the gasket or joint-contact surface sufficient compression to assure a tight joint, all at the design temperature. The minimum load is a function of the design pressure, the gasket material, and the effective gasket or contact area to be kept tight under pressure, per eq. (c)(1)(1) below, and determines one of the two requirements for the amount of the bolting A_{m1}. This load is also used for the design of the flange, per eq. (d)(3) below.

(2) Gasket Seating. The conditions existing when the gasket or joint-contact surface is seated by applying an initial load with the bolts when assembling the joint, at atmospheric temperature and pressure. The minimum initial load considered to be adequate for proper seating is a function of the gasket material, and the effective gasket or contact area to be seated, per eq. (c)(2)(2) below, and determines the other of the two requirements for the amount of bolting A_{m2}. For the design of the flange, this load is modified per eq. (e)(4) below to take account of the operating conditions, when these govern the amount of bolting required A_m, as well as the amount of bolting actually provided A_b.

(c) Required Bolt Loads. The flange bolt loads used in calculating the required cross-sectional area of bolts shall be determined as follows.

(1) The required bolt load for the operating conditions W_{m1} shall be sufficient to resist the hydrostatic end force H exerted by the maximum allowable working pressure on the area bounded by the diameter of gasket reaction, and, in addition, to maintain on the gasket or joint-contact surface a compression load H_p, which experience has shown to be sufficient to ensure a tight joint. (This compression load is expressed as a multiple m of the internal pressure. Its value is a function of the gasket material and construction.)

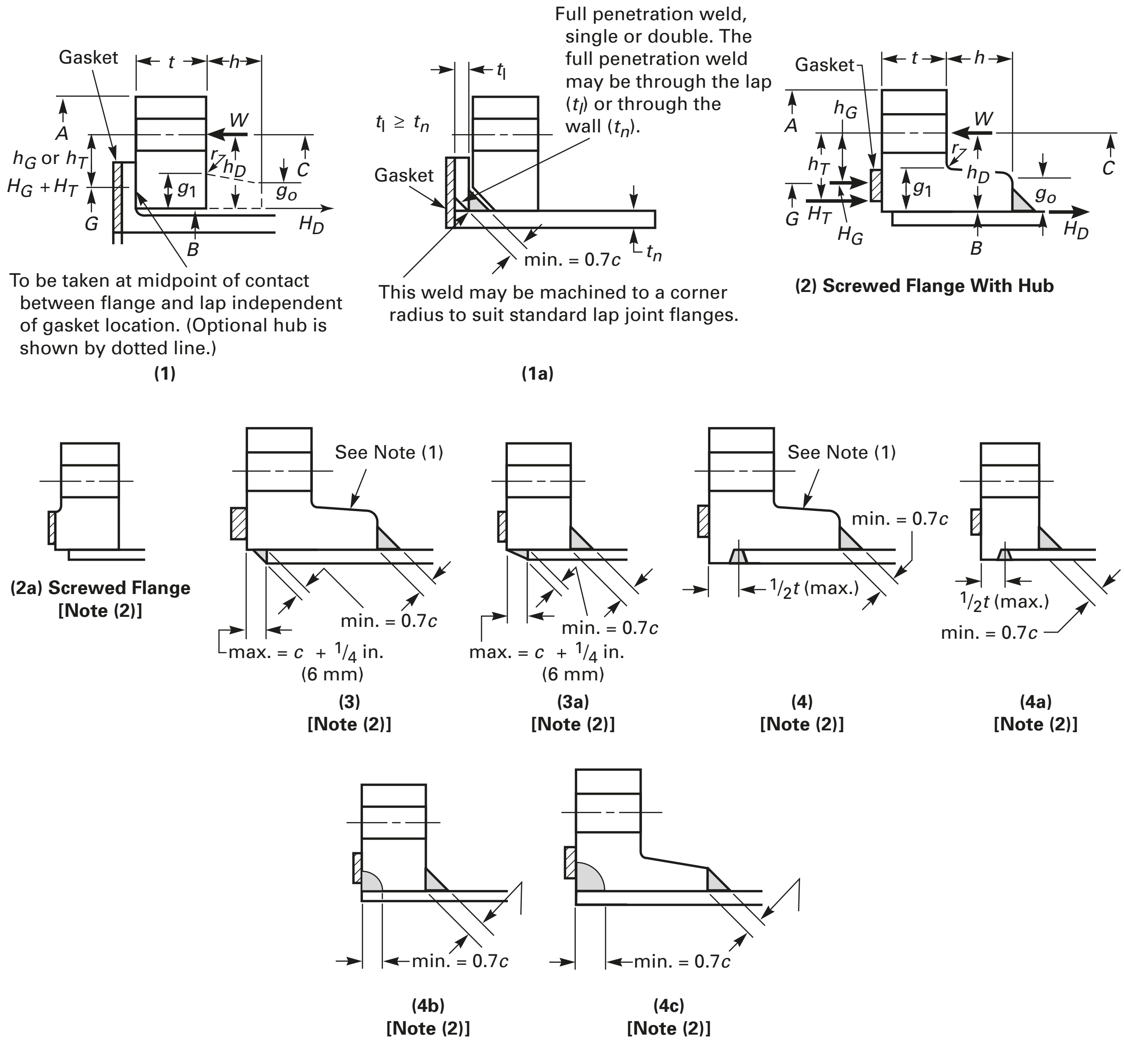
NOTE: Tables 2-5.1 and 2-5.2 give a list of many commonly used gasket materials and contact facings, with suggested values of m, b, and y that have proved satisfactory in actual service. These values are suggested only and are not mandatory.

The required bolt load for the operating conditions W_{m1} is determined in accordance with eq. (1).

$$W_{m1} = H + H_p = 0.785G^2P + (2b \times 3.14GmP) \qquad (1)$$

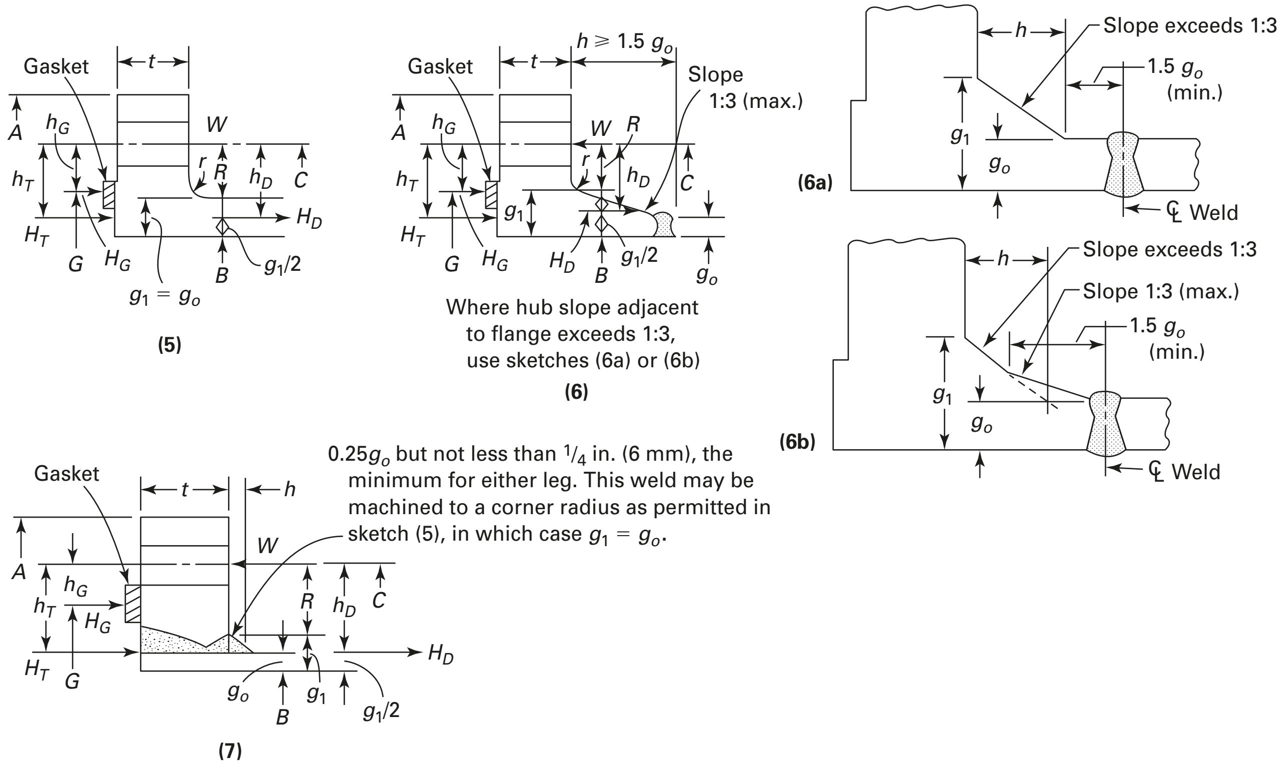
(2) Before a tight joint can be obtained, it is necessary to seat the gasket or joint-contact surface properly by applying a minimum initial load (under atmospheric temperature conditions without the presence of internal pressure), which is a function of the gasket material and

Figure 2-4
Types of Flanges



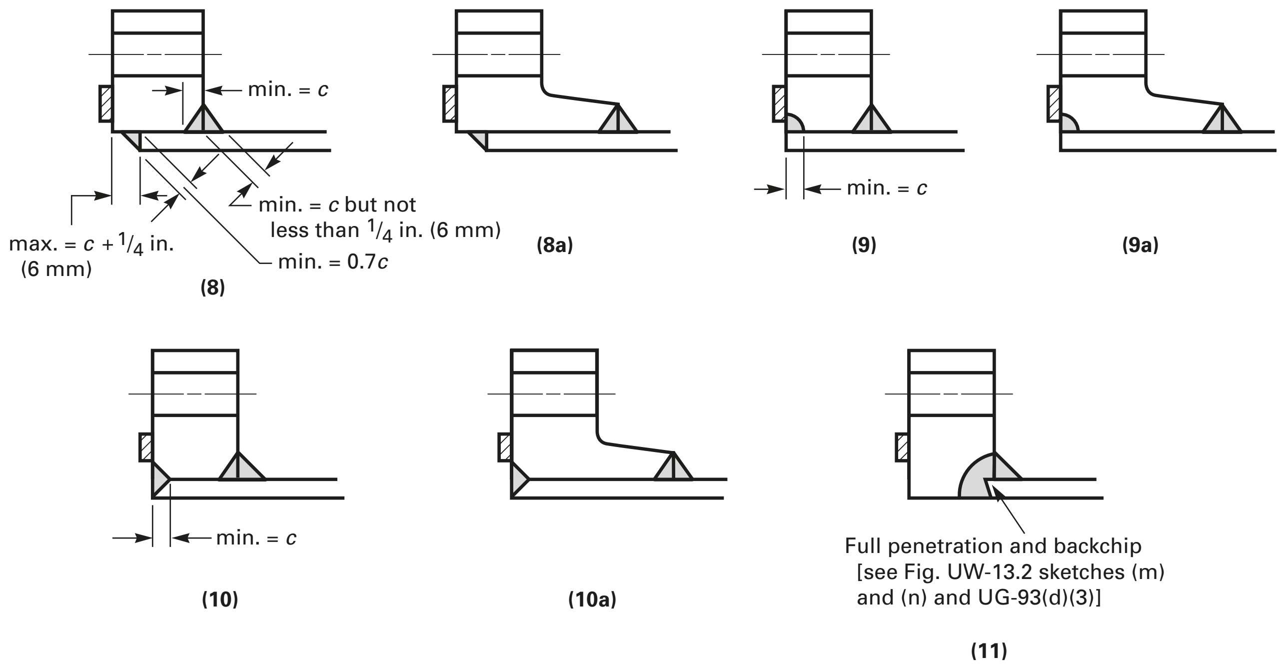
Loose-Type Flanges [Notes (3) and (4)]

Figure 2-4
Types of Flanges (Cont'd)

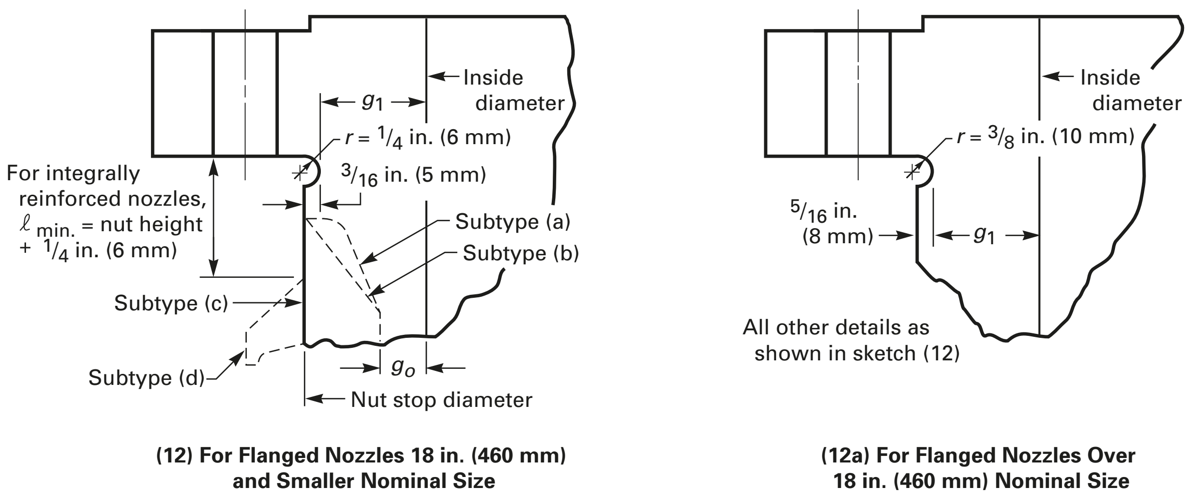


Integral-Type Flanges [Notes (3) and (4)]

Figure 2-4
Types of Flanges (Cont'd)



Optional-Type Flanges [Notes (5), (6), and (7)]



Flanges With Nut Stops [Note (8)]

NOTES:
(1) For hub tapers 6 deg or less, use $g_o = g_1$.
(2) Loading and dimensions for sketches (2a), (3), (3a), (4), (4a), (4b), and (4c) not shown are the same as for sketch (2).
(3) Fillet radius r to be at least 0.25 g_1 but not less than $^3/_{16}$ in. (5 mm).

Figure 2-4
Types of Flanges (Cont'd)

NOTES (CONT'D):
(4) Facing thicknesses or groove depths greater than $^1/_{16}$ in. (1.5 mm) shall be in excess of the required minimum flange thickness, t; those equal to or less than $^1/_{16}$ in. (1.5 mm) may be included in the overall flange thickness.
(5) Optional-type flanges may be calculated as either loose or integral type. See 2-4.
(6) Loadings and dimensions not shown in sketches (8), (8a), (9), (9a), (10), and (10a) are the same as shown in sketch (2) when the flange is calculated as a loose-type flange, and as shown in sketch (7) when the flange is calculated as an integral-type flange.
(7) The groove and fillet welds between the flange back face and the shell given in sketch (8) also apply to sketches (8a), (9), (9a), (10), and (10a).
(8) For subtypes (a) and (b), g_o is the thickness of the hub at the small end. For subtypes (c) and (d), $g_o = g_1$.

the effective gasket area to be seated. The minimum initial bolt load required for this purpose W_{m2} shall be determined in accordance with eq. (2).

$$W_{m2} = 3.14bGy \tag{2}$$

The need for providing sufficient bolt load to seat the gasket or joint-contact surfaces in accordance with eq. (2) will prevail on many low-pressure designs and with facings and materials that require a high seating load, and where the bolt load computed by eq. (1)(1) for the operating conditions is insufficient to seat the joint. Accordingly, it is necessary to furnish bolting and to pretighten the bolts to provide a bolt load sufficient to satisfy both of these requirements, each one being individually investigated. When eq. (2) governs, flange proportions will be a function of the bolting instead of internal pressure.

(3) Bolt loads for flanges using gaskets of the self-energizing type differ from those shown above.

(-a) The required bolt load for the operating conditions W_{m1} shall be sufficient to resist the hydrostatic end force H exerted by the maximum allowable working pressure on the area bounded by the outside diameter of the gasket. H_p is to be considered as 0 for all self-energizing gaskets except certain seal configurations which generate axial loads which must be considered.

(-b) $W_{m2} = 0$.

Self-energizing gaskets may be considered to require an inconsequential amount of bolting force to produce a seal. Bolting, however, must be pretightened to provide a bolt load sufficient to withstand the hydrostatic end force H.

(d) Total Required and Actual Bolt Areas, A_m and A_b. The total cross-sectional area of bolts A_m required for both the operating conditions and gasket seating is the greater of the values for A_{m1} and A_{m2}, where $A_{m1} = W_{m1}/S_b$ and $A_{m2} = W_{m2}/S_a$. A selection of bolts to be used shall be made such that the actual total cross-sectional area of bolts A_b will not be less than A_m. For vessels in lethal service or when specified by the user or his designated agent, the maximum bolt spacing shall not exceed the value calculated in accordance with eq. (3).

$$B_{s\,\max} = 2a + \frac{6t}{m + 0.5} \tag{3}$$

(e) Flange Design Bolt Load W. The bolt loads used in the design of the flange shall be the values obtained from eqs. (4) and (5). For operating conditions,

$$W = W_{m1} \tag{4}$$

For gasket seating,

$$W = \frac{(A_m + A_b)S_a}{2} \tag{5}$$

S_a used in eq. (5) shall be not less than that tabulated in the stress tables (see UG-23). In addition to the minimum requirements for safety, eq. (5) provides a margin against abuse of the flange from overbolting. Since the margin against such abuse is needed primarily for the initial, bolting-up operation which is done at atmospheric temperature and before application of internal pressure, the flange design is required to satisfy this loading only under such conditions.

Table 2-4
Recommended Minimum Gasket Contact Widths for Sheet and Composite Gaskets

Flange ID	Gasket Contact Width
24 in. (600 mm) < ID ≤ 36 in. (900 mm)	1 in. (25 mm)
36 in. (900 mm) < ID < 60 in. (1500 mm)	$1^1/_4$ in. (32 mm)
ID ≥ 60 in. (1500 mm)	$1^1/_2$ (38 mm)

Table 2-5.1
Gasket Materials and Contact Facings
Gasket Factors *m* for Operating Conditions and Minimum Design Seating Stress *y*

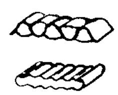
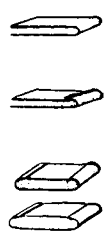
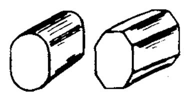
Gasket Material	Gasket Factor *m*	Min. Design Seating Stress *y*, psi (MPa)	Sketches	Facing Sketch and Column in Table 2-5.2
Self-energizing types (O-rings, metallic, elastomer, other gasket types considered as self-sealing)	0	0 (0)	...	...
Elastomers without fabric or high percent of mineral fiber:				
Below 75A Shore Durometer	0.50	0 (0)		(1a), (1b), (1c), (1d), (4), (5); Column II
75A or higher Shore Durometer	1.00	200 (1.4)		
Mineral fiber with suitable binder for operating conditions:				
$^1\!/_8$ in. (3.2 mm) thick	2.00	1,600 (11)		(1a), (1b), (1c), (1d), (4), (5); Column II
$^1\!/_{16}$ in. (1.6 mm) thick	2.75	3,700 (26)		
$^1\!/_{32}$ in. (0.8 mm) thick	3.50	6,500 (45)		
Elastomers with cotton fabric insertion	1.25	400 (2.8)		(1a), (1b), (1c), (1d), (4), (5); Column II
Elastomers with mineral fiber fabric insertion (with or without wire reinforcement):				
3-ply	2.25	2,200 (15)		(1a), (1b), (1c), (1d), (4), (5); Column II
2-ply	2.50	2,900 (20)		
1-ply	2.75	3,700 (26)		
Vegetable fiber	1.75	1,100 (7.6)		(1a), (1b), (1c), (1d), (4), (5); Column II
Spiral-wound metal, mineral fiber filled:				
Carbon	2.50	10,000 (69)		(1a), (1b); Column II
Stainless, Monel, and nickel-base alloys	3.00	10,000 (69)		
Corrugated metal, mineral fiber inserted, or corrugated metal, jacketed mineral fiber filled:				
Soft aluminum	2.50	2,900 (20)		(1a), (1b); Column II
Soft copper or brass	2.75	3,700 (26)		
Iron or soft steel	3.00	4,500 (31)		
Monel or 4–6% chrome	3.25	5,500 (38)		
Stainless steels and nickel-base alloys	3.50	6,500 (45)		
Corrugated metal:				
Soft aluminum	2.75	3,700 (26)		(1a), (1b), (1c), (1d); Column II
Soft copper or brass	3.00	4,500 (31)		
Iron or soft steel	3.25	5,500 (38)		
Monel or 4–6% chrome	3.50	6,500 (45)		
Stainless steels and nickel-base alloys	3.75	7,600 (52)		
Flat metal, jacketed mineral fiber filled:				
Soft aluminum	3.25	5,500 (38)		(1a), (1b), (1c) [Note (1)], (1d) [Note (1)], (2) [Note (1)]; Column II
Soft copper or brass	3.50	6,500 (45)		
Iron or soft steel	3.75	7,600 (52)		
Monel	3.50	8,000 (55)		
4–6% chrome	3.75	9,000 (62)		
Stainless steels and nickel-base alloys	3.75	9,000 (62)		

Table 2-5.1
Gasket Materials and Contact Facings
Gasket Factors m for Operating Conditions and Minimum Design Seating Stress y (Cont'd)

Gasket Material	Gasket Factor m	Min. Design Seating Stress y, psi (MPa)	Sketches	Facing Sketch and Column in Table 2-5.2
Grooved metal:				
Soft aluminum	3.25	5,500 (38)		(1a), (1b), (1c),
Soft copper or brass	3.50	6,500 (45)		(1d), (2), (3);
Iron or soft metal	3.75	7,600 (52)		Column II
Monel or 4–6% chrome	3.75	9,000 (62)		
Stainless steels and nickel-base alloys	4.25	10,100 (70)		
Solid flat metal:				
Soft aluminum	4.00	8,800 (61)		(1a), (1b), (1c),
Soft copper or brass	4.75	13,000 (90)		(1d), (2), (3), (4),
Iron or soft steel	5.50	18,000 (124)		(5); Column I
Monel or 4–6% chrome	6.00	21,800 (150)		
Stainless steels and nickel-base alloys	6.50	26,000 (180)		
Ring joint:				
Iron or soft steel	5.50	18,000 (124)		(6); Column I
Monel or 4–6% chrome	6.00	21,800 (150)		
Stainless steels and nickel-base alloys	6.50	26,000 (180)		

GENERAL NOTE: This Table gives a list of many commonly used gasket materials and contact facings with suggested design values of m and y that have generally proved satisfactory in actual service when using effective gasket seating width b given in Table 2-5.2. The design values and other details given in this Table are suggested only and are not mandatory.

NOTE:
(1) The surface of a gasket having a lap should not be against the nubbin.

NOTE: Where additional safety against abuse is desired, or where it is necessary that the flange be suitable to withstand the full available bolt load $A_b S_a$, the flange may be designed on the basis of this latter quantity.

(19) 2-6 FLANGE MOMENTS

In the calculation of flange stress, the moment of a load acting on the flange is the product of the load and its moment arm. The moment arm is determined by the relative position of the bolt circle with respect to that of the load producing the moment (see Figure 2-4). No consideration shall be given to any possible reduction in moment arm due to cupping of the flanges or due to inward shifting of the line of action of the bolts as a result thereof. It is recommended that the value of h_G $[(C-G)/2]$ be kept to a minimum to reduce flange rotation at the sealing surface.

For the operating conditions, the total flange moment M_o is the sum of the three individual moments M_D, M_T, and M_G, as defined in 2-3 and based on the flange design load of eq. 2-5(e)(4) with moment arms as given in Table 2-6.

For gasket seating, the total flange moment M_o is based on the flange design bolt load of eq. 2-5(e)(5), which is opposed only by the gasket load, in which case

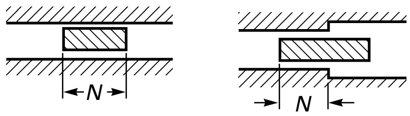
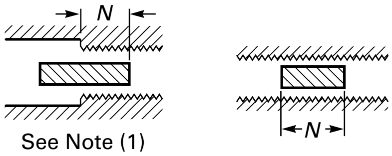
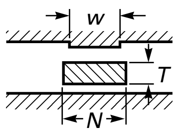
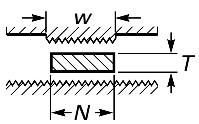
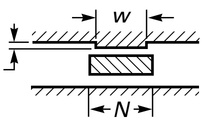
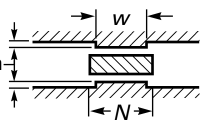
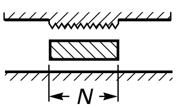
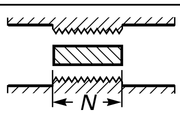
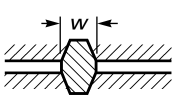
$$M_o = W\frac{(C-G)}{2} \tag{6}$$

For vessels in lethal service or when specified by the user or his designated agent, the bolt spacing correction shall be applied in calculating the flange stress in 2–7, 2-13(c), and 2-13(d). The flange moment M_o without correction for bolt spacing is used for the calculation of the rigidity index in 2-14.

When the bolt spacing exceeds $2a + t$, multiply M_O by the bolt spacing correction factor B_{SC} for calculating flange stress, where

$$B_{SC} = \sqrt{\frac{B_s}{2a + t}} \tag{7}$$

Table 2-5.2
Effective Gasket Width

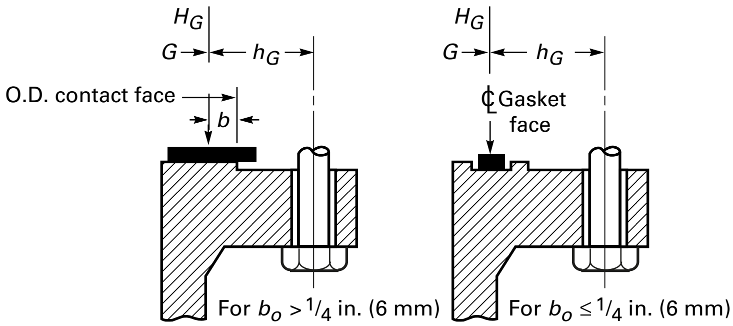
Facing Sketch (Exaggerated)	Basic Gasket Seating Width, b_o	
	Column I	**Column II**
(1a) 	$\frac{N}{2}$	$\frac{N}{2}$
(1b)  See Note (1)		
(1c)  $w \leq N$	$\frac{w+T}{2}; \left(\frac{w+N}{4}\max\right)$	$\frac{w+T}{2}; \left(\frac{w+N}{4}\max\right)$
(1d) See Note (1)  $w \leq N$		
(2) $^1/_{64}$ in. (0.4 mm) nubbin  $w \leq N/2$	$\frac{w+N}{4}$	$\frac{w+3N}{8}$
(3) $^1/_{64}$ in. (0.4 mm) nubbin  $w \leq N/2$	$\frac{N}{4}$	$\frac{3N}{8}$
(4) See Note (1) 	$\frac{3N}{8}$	$\frac{7N}{16}$
(5) See Note (1) 	$\frac{N}{4}$	$\frac{3N}{8}$
(6) 	$\frac{w}{8}$	...

Effective Gasket Seating Width, b

$b = b_o$, when $b_o \leq {}^1/_4$ in. (6 mm); $b = C_b\sqrt{b_o}$, when $b_o > {}^1/_4$ in. (6 mm)

Table 2-5.2
Effective Gasket Width (Cont'd)

Location of Gasket Load Reaction



GENERAL NOTE: The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.

NOTE:
(1) Where serrations do not exceed $^1/_{64}$ in. (0.4 mm) depth and $^1/_{32}$ in. (0.8 mm) width spacing, sketches (1b) and (1d) shall be used.

Table 2-6
Moment Arms for Flange Loads Under Operating Conditions

	h_D	h_T	h_G
Integral-type flanges [see Figure 2-4, sketches (5), (6), (6a), (6b), and (7)] and optional type flanges calculated as integral type [see Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)]	$R + 0.5g_1$	$\frac{R + g_1 + h_G}{2}$	$\frac{C - G}{2}$
Loose type, except lap-joint flanges [see Figure 2-4, sketches (2), (2a), (3), (3a), (4), and (4a)]; and optional type flanges calculated as loose type [see Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)]	$\frac{C - B}{2}$	$\frac{h_D + h_G}{2}$	$\frac{C - G}{2}$
Lap-type flanges [see Figure 2-4, sketches (1) and (1a)]	$\frac{C - B}{2}$	$\frac{C - G}{2}$	$\frac{C - G}{2}$

2-7 CALCULATION OF FLANGE STRESSES

The stresses in the flange shall be determined for both the operating conditions and gasket seating condition, whichever controls, in accordance with the following equations:

(a) for integral type flanges [Figure 2-4, sketches (5), (6), (6a), (6b), and (7)], for optional type flanges calculated as integral type [Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)], and for loose type flanges with a hub which is considered [Figure 2-4, sketches (1), (1a), (2), (2a), (3), (3a), (4), (4a), (4b), and (4c)]:

Longitudinal hub stress

$$S_H = \frac{f M_o}{L g_1^2 B} \tag{8}$$

Radial flange stress

$$S_R = \frac{(1.33te + 1)M_o}{Lt^2 B} \tag{9}$$

Tangential flange stress

$$S_T = \frac{Y M_o}{t^2 B} - Z S_R \tag{10}$$

(b) for loose type flanges without hubs and loose type flanges with hubs which the designer chooses to calculate without considering the hub [Figure 2-4, sketches (1), (1a), (2), (2a), (3), (3a), (4), (4a), (4b), and (4c)] and optional type flanges calculated as loose type [Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)]:

$$\begin{gathered} S_T = \frac{Y M_o}{t^2 B} \\ S_R = 0 \\ S_H = 0 \end{gathered} \tag{11}$$

(19)

2-8 ALLOWABLE FLANGE DESIGN STRESSES

(a) The flange stresses calculated by the equations in 2–7 shall not exceed the following values:

(1) longitudinal hub stress S_H not greater than S_f for cast iron[89] and, except as otherwise limited by (-a) and (-b) below, not greater than $1.5S_f$ for materials other than cast iron:

(-a) longitudinal hub stress S_H not greater than the smaller of $1.5S_f$ or $1.5S_n$ for optional type flanges designed as integral [Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)], also integral type [Figure 2-4, sketch (7)] where the neck material constitutes the hub of the flange;

(-b) longitudinal hub stress S_H not greater than the smaller of $1.5S_f$ or $2.5S_n$ for integral type flanges with hub welded to the neck, pipe or vessel wall [Figure 2-4, sketches (6), (6a), and (6b)].

(2) radial flange stress S_R not greater than S_f;

(3) tangential flange stress S_T not greater than S_f;

(4) also $(S_H + S_R)/2$ not greater than S_f and $(S_H + S_T)/2$ not greater than S_f.

(b) For hub flanges attached as shown in Figure 2-4, sketches (2), (2a), (3), (3a), (4), (4a), (4b), and (4c), the nozzle neck, vessel or pipe wall shall not be considered to have any value as a hub.

(c) In the case of loose type flanges with laps, as shown in Figure 2-4, sketches (1) and (1a), where the gasket is so located that the lap is subjected to shear, the shearing stress shall not exceed 0.8 S_n for the material of the lap, as defined in 2-3. In the case of welded flanges, shown in Figure 2-4, sketches (3), (3a), (4), (4a), (4b), (4c), (7), (8), (8a), (9), (9a), (10), and (10a) where the nozzle neck, vessel, or pipe wall extends near to the flange face and may form the gasket contact face, the shearing stress carried by the welds shall not exceed 0.8 S_n. The shearing stress shall be calculated on the basis of W_{m1} or W_{m2} as defined in 2-3, whichever is greater. Similar cases where flange parts are subjected to shearing stress shall be governed by the same requirements.

2-9 SPLIT LOOSE FLANGES[90] (19)

Loose flanges split across a diameter and designed under the rules given in this Appendix may be used under the following provisions.

(a) When the flange consists of a single split flange or flange ring, it shall be designed as if it were a solid flange (without splits), using 200% of the total moment M_o as defined in 12-4.

(b) When the flange consists of two split rings each ring shall be designed as if it were a solid flange (without splits), using 75% of the total moment M_o as defined in 12-4. The pair of rings shall be assembled so that the splits in one ring shall be 90 deg from the splits in the other ring.

(c) The splits should preferably be midway between bolt holes.

(d) It is not a requirement that the flange rigidity rules of 2-14 be applied to split loose flanges.

Figure 2-7.1
Values of *T*, *U*, *Y*, and *Z* (Terms Involving *K*)

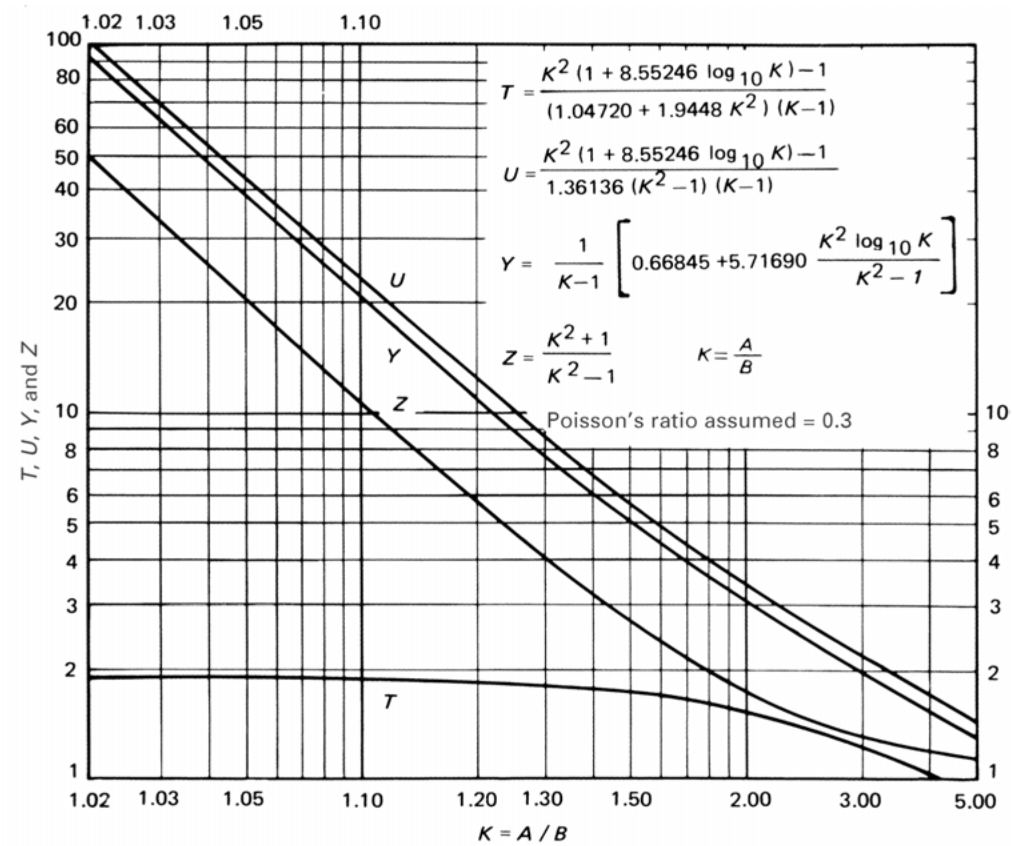
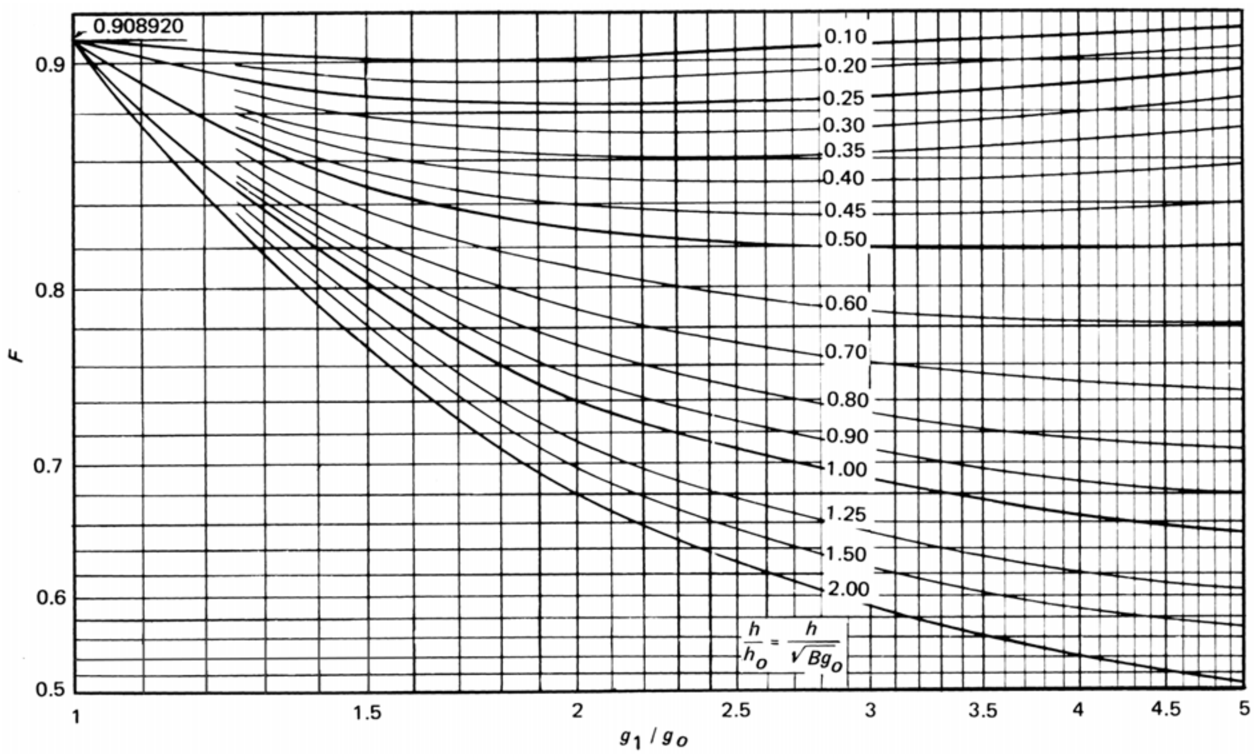
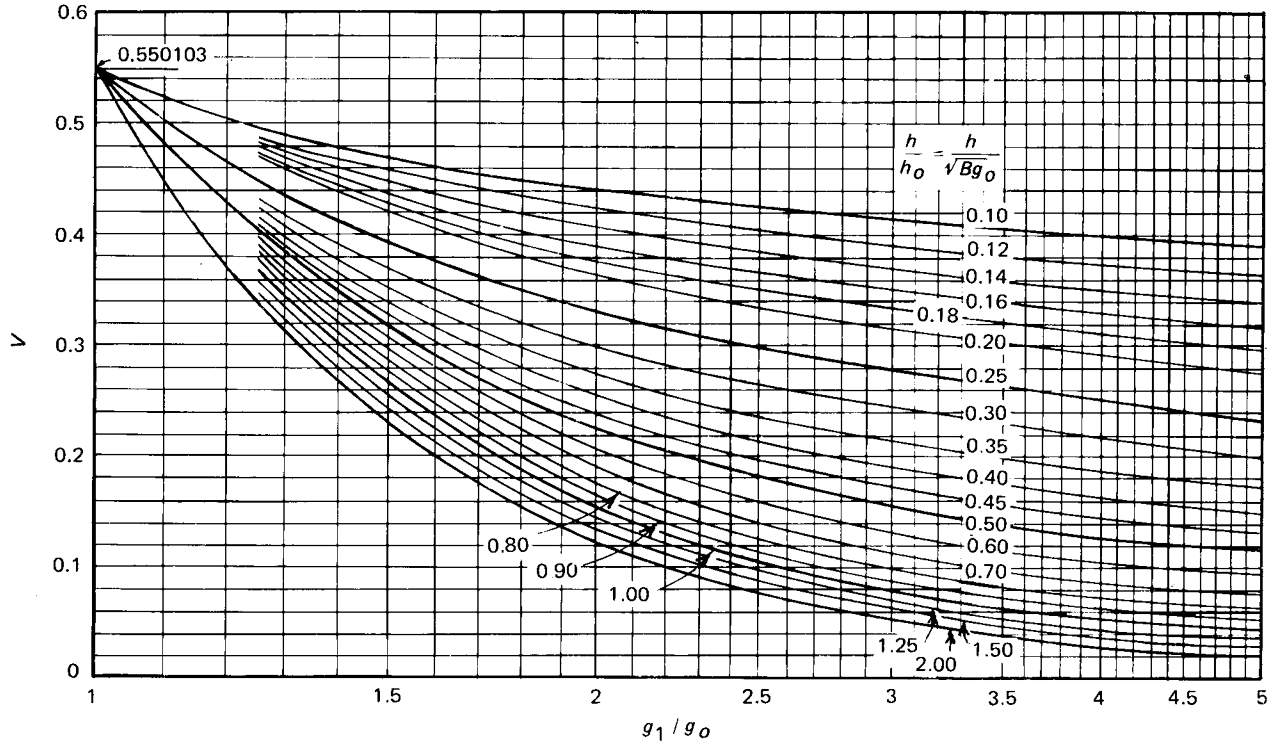


Figure 2-7.2
Values of F (Integral Flange Factors)



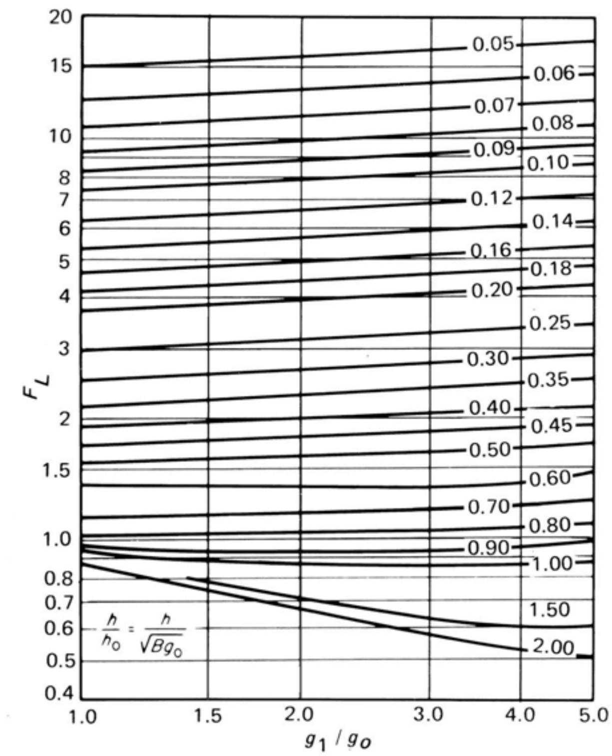
GENERAL NOTE: See Table 2-7.1 for equations.

Figure 2-7.3
Values of *V* (Integral Flange Factors)



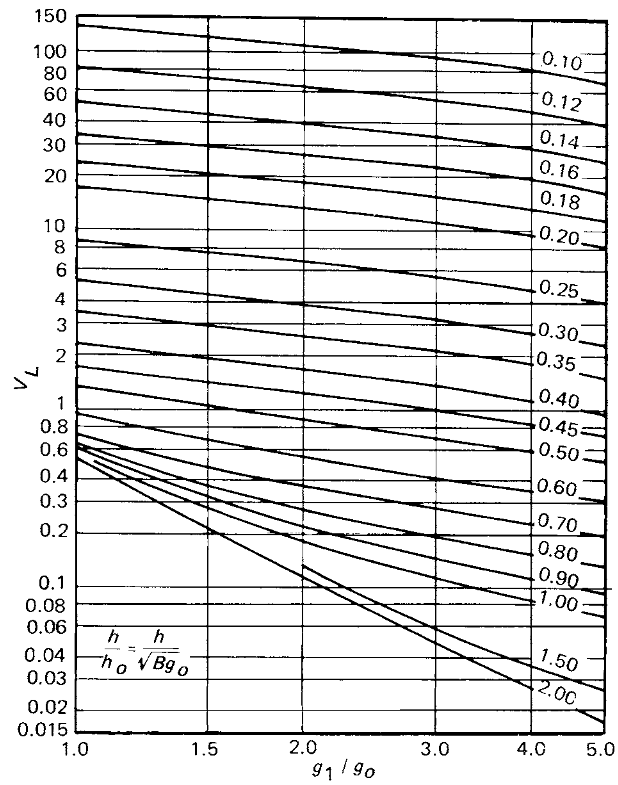
GENERAL NOTE: See Table 2-7.1 for equations.

Figure 2-7.4
Values of F_L (Loose Hub Flange Factors)



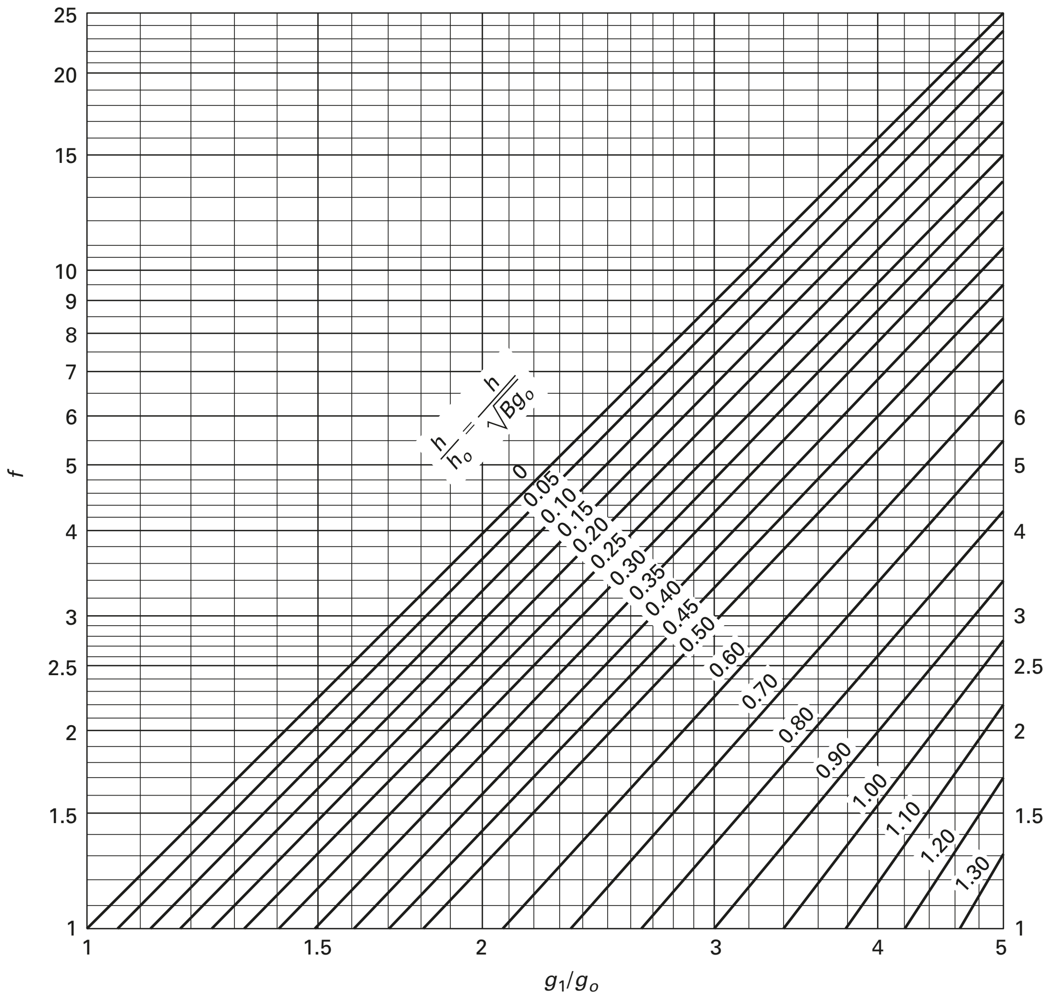
GENERAL NOTE: See Table 2-7.1 for equations.

Figure 2-7.5
Values of V_L (Loose Hub Flange Factors)



GENERAL NOTE: See Table 2-7.1 for equations.

Figure 2-7.6
Values of f (Hub Stress Correction Factor)



f = 1 (minimum)
= 1 for hubs of uniform thickness (g_1/g_0 = 1)
= 1 for loose hubbed flanges

GENERAL NOTE: See Table 2-7.1 for equations.

Table 2-7.1
Flange Factors in Formula Form

Integral Flange

Factor F per Figure 2-7.2 is then solved by

$$F = -\frac{E_6}{\left(\frac{C}{2.73}\right)^{\frac{1}{4}} \frac{(1+A)^3}{C}}$$

Factor V per Figure 2-7.3 is then solved by

$$V = \frac{E_4}{\left(\frac{2.73}{C}\right)^{\frac{1}{4}} (1+A)^3}$$

Factor f per Figure 2-7.6 is then solved by

$$f = C_{36} \ / \left[1 + A\right]$$

The values used in the above equations are solved using eqs. (1) through (45) below based on the values g_1, g_o, h, and h_o as defined by 2-3. When $g_1 = g_o$, $F = 0.908920$, $V = 0.550103$, and $f = 1$; thus eqs. (1) through (45) need not be solved.

Loose Hub Flange

Factor F_L per Figure 2-7.4 is solved by

$$F_L = -\frac{C_{18}\left(\frac{1}{2} + \frac{A}{6}\right) + C_{21}\left(\frac{1}{4} + \frac{11A}{84}\right) + C_{24}\left(\frac{1}{70} + \frac{A}{105}\right) - \left(\frac{1}{40} + \frac{A}{72}\right)}{\left(\frac{C}{2.73}\right)^{\frac{1}{4}} \frac{(1+A)^3}{C}}$$

Factor V_L per Figure 2-7.5 is solved by

$$V_L = \frac{\frac{1}{4} - \frac{C_{24}}{5} - \frac{3C_{21}}{2} - C_{18}}{\left(\frac{2.73}{C}\right)^{\frac{1}{4}} (1+A)^3}$$

Factor f per Figure 2-7.6 is set equal to 1.

$$f = 1$$

The values used in the above equations are solved using eqs. (1) through (5), (7), (9), (10), (12), (14), (16), (18), (20), (23), and (26) below based on the values of g_1, g_o, h, and h_o as defined by 2-3.

Equations

(1) $A = (g_1/g_o) - 1$
(2) $C = 43.68(h/h_o)^4$
(3) $C_1 = 1/3 + A/12$
(4) $C_2 = 5/42 + 17A/336$
(5) $C_3 = 1/210 + A/360$
(6) $C_4 = 11/360 + 59A/5040 + (1 + 3A)/C$
(7) $C_5 = 1/90 + 5A/1008 - (1 + A)^3/C$
(8) $C_6 = 1/120 + 17A/5040 + 1/C$
(9) $C_7 = 215/2772 + 51A/1232 + (60/7 + 225A/14 + 75A^2/7 + 5A^3/2)/C$
(10) $C_8 = 31/6930 + 128A/45{,}045 + (6/7 + 15A/7 + 12A^2/7 + 5A^3/11)/C$
(11) $C_9 = 533/30{,}240 + 653A/73{,}920 + (1/2 + 33A/14 + 39A^2/28 + 25A^3/84)/C$
(12) $C_{10} = 29/3780 + 3A/704 - (1/2 + 33A/14 + 81A^2/28 + 13A^3/12)/C$
(13) $C_{11} = 31/6048 + 1763A/665{,}280 + (1/2 + 6A/7 + 15A^2/28 + 5A^3/42)/C$
(14) $C_{12} = 1/2925 + 71A/300{,}300 + (8/35 + 18A/35 + 156A^2/385 + 6A^3/55)/C$
(15) $C_{13} = 761/831{,}600 + 937A/1{,}663{,}200 + (1/35 + 6A/35 + 11A^2/70 + 3A^3/70)/C$
(16) $C_{14} = 197/415{,}800 + 103A/332{,}640 - (1/35 + 6A/35 + 17A^2/70 + A^3/10)/C$
(17) $C_{15} = 233/831{,}600 + 97A/554{,}400 + (1/35 + 3A/35 + A^2/14 + 2A^3/105)/C$
(18) $C_{16} = C_1 C_7 C_{12} + C_2 C_8 C_3 + C_3 C_8 C_2 - (C_3^2 C_7 + C_8^2 C_1 + C_2^2 C_{12})$
(19) $C_{17} = [C_4 C_7 C_{12} + C_2 C_8 C_{13} + C_3 C_8 C_9 - (C_{13} C_7 C_3 + C_8^2 C_4 + C_{12} C_2 C_9)]/C_{16}$
(20) $C_{18} = [C_5 C_7 C_{12} + C_2 C_8 C_{14} + C_3 C_8 C_{10} - (C_{14} C_7 C_3 + C_8^2 C_5 + C_{12} C_2 C_{10})]/C_{16}$
(21) $C_{19} = [C_6 C_7 C_{12} + C_2 C_8 C_{15} + C_3 C_8 C_{11} - (C_{15} C_7 C_3 + C_8^2 C_6 + C_{12} C_2 C_{11})]/C_{16}$
(22) $C_{20} = [C_1 C_9 C_{12} + C_4 C_8 C_3 + C_3 C_{13} C_2 - (C_3^2 C_9 + C_{13} C_8 C_1 + C_{12} C_4 C_2)]/C_{16}$
(23) $C_{21} = [C_1 C_{10} C_{12} + C_5 C_8 C_3 + C_3 C_{14} C_2 - (C_3^2 C_{10} + C_{14} C_8 C_1 + C_{12} C_5 C_2)]/C_{16}$
(24) $C_{22} = [C_1 C_{11} C_{12} + C_6 C_8 C_3 + C_3 C_{15} C_2 - (C_3^2 C_{11} + C_{15} C_8 C_1 + C_{12} C_6 C_2)]/C_{16}$
(25) $C_{23} = [C_1 C_7 C_{13} + C_2 C_9 C_3 + C_4 C_8 C_2 - (C_3 C_7 C_4 + C_8 C_9 C_1 + C_2^2 C_{13})]/C_{16}$
(26) $C_{24} = [C_1 C_7 C_{14} + C_2 C_{10} C_3 + C_5 C_8 C_2 - (C_3 C_7 C_5 + C_8 C_{10} C_1 + C_2^2 C_{14})]/C_{16}$
(27) $C_{25} = [C_1 C_7 C_{15} + C_2 C_{11} C_3 + C_6 C_8 C_2 - (C_3 C_7 C_6 + C_8 C_{11} C_1 + C_2^2 C_{15})]/C_{16}$
(28) $C_{26} = -(C/4)^{1/4}$
(29) $C_{27} = C_{20} - C_{17} - 5/12 + C_{17} C_{26}$
(30) $C_{28} = C_{22} - C_{19} - 1/12 + C_{19} C_{26}$
(31) $C_{29} = -(C/4)^{1/2}$
(32) $C_{30} = -(C/4)^{3/4}$
(33) $C_{31} = 3A/2 - C_{17} C_{30}$
(34) $C_{32} = 1/2 - C_{19} C_{30}$
(35) $C_{33} = 0.5 C_{26} C_{32} + C_{28} C_{31} C_{29} - (0.5 C_{30} C_{28} + C_{32} C_{27} C_{29})$
(36) $C_{34} = 1/12 + C_{18} - C_{21} - C_{18} C_{26}$
(37) $C_{35} = -C_{18}(C/4)^{3/4}$
(38) $C_{36} = (C_{28} C_{35} C_{29} - C_{32} C_{34} C_{29})/C_{33}$
(39) $C_{37} = [0.5 C_{26} C_{35} + C_{34} C_{31} C_{29} - (0.5 C_{30} C_{34} + C_{35} C_{27} C_{29})]/C_{33}$
(40) $E_1 = C_{17} C_{36} + C_{18} + C_{19} C_{37}$

Table 2-7.1
Flange Factors in Formula Form (Cont'd)

Equations (Cont'd)

(41) $E_2 = C_{20}C_{36} + C_{21} + C_{22}C_{37}$
(42) $E_3 = C_{23}C_{36} + C_{24} + C_{25}C_{37}$
(43) $E_4 = 1/4 + C_{37}/12 + C_{36}/4 - E_3/5 - 3E_2/2 - E_1$
(44) $E_5 = E_1(1/2 + A/6) + E_2(1/4 + 11A/84) + E_3(1/70 + A/105)$
(45) $E_6 = E_5 - C_{36}\ (7/120 + A/36 + 3A/C) - 1/40 - A/72 - C_{37}(1/60 + A/120 + 1/C)$

2-10 NONCIRCULAR SHAPED FLANGES WITH CIRCULAR BORE

The outside diameter A for a noncircular flange with a circular bore shall be taken as the diameter of the largest circle, concentric with the bore, inscribed entirely within the outside edges of the flange. Bolt loads and moments, as well as stresses, are then calculated as for circular flanges, using a bolt circle drawn through the centers of the outermost bolt holes.

(19) ## 2-11 FLANGES SUBJECT TO EXTERNAL PRESSURES

(a) The design of flanges for external pressure only [see UG-99(f)][91] shall be based on the equations given in 2–7 for internal pressure except that for operating conditions:

$$M_o = H_D\left(h_D - h_G\right) + H_T\left(h_T - h_G\right) \tag{10}$$

For gasket seating,

$$M_o = W h_G \tag{11}$$

where

$$W = \frac{A_{m2} + A_b}{2} S_a \tag{11a}$$

$$H_D = 0.785 B^2 P_e \tag{11b}$$

$$H_T = H - H_D \tag{11c}$$

$$H = 0.785 G^2 P_e \tag{11d}$$

P_e = external design pressure

See 2-3 for definitions of other symbols. S_a used in eq. (11a) shall be not less than that tabulated in the stress tables (see UG-23).

(b) When flanges are subject at different times during operation to external or internal pressure, the design shall satisfy the external pressure design requirements given in (a) above and the internal pressure design requirements given elsewhere in this Appendix.

NOTE: The combined force of external pressure and bolt loading may plastically deform certain gaskets to result in loss of gasket contact pressure when the connection is depressurized. To maintain a tight joint when the unit is repressurized, consideration should be given to gasket and facing details so that excessive deformation of the gasket will not occur. Joints subject to pressure reversals, such as in heat exchanger floating heads, are in this type of service.

2-12 FLANGES WITH NUT-STOPS (19)

(a) When flanges are designed per this Appendix, or are fabricated to the dimensions of ASME B16.5 or other acceptable standards [see UG-44(a)(2)], except that the dimension R is decreased to provide a nut-stop, the fillet radius relief shall be as shown in Figure 2-4, sketches (12) and (12a) except that:

(1) for flanges designed to this Appendix, the minimum dimension g_1 must be the lesser of $2t$ (t from UG-27) or $4r$, but in no case less than $^1/_2$ in. (13 mm), where

r = the radius of the undercut

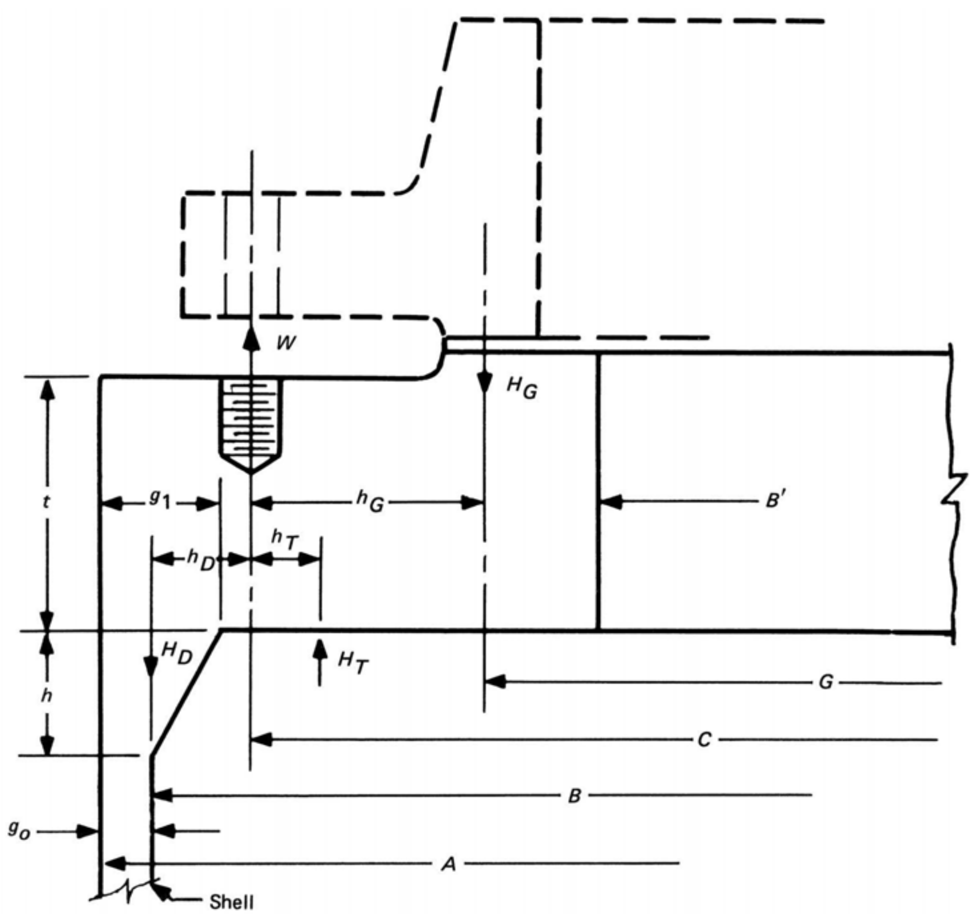
(2) for ASME B16.5 or other standard flanges, the dimension of the hub g_o shall be increased as necessary to provide a nut-stop.

2-13 REVERSE FLANGES

(a) Flanges with the configuration as indicated in Figure 2-13.1 shall be designed as integral reverse flanges and those in Figure 2-13.2 shall be designed as loose ring type reverse flanges. These flanges shall be designed in conformance with the rules in 2-3 through 2-8, but with the modifications as described in the following. Mandatory use of these rules is limited to $K \leq 2$. When $K > 2$, results become increasingly conservative and U-2(g) may be used.

(1) Integral Type Reverse Flange. The shell-to-flange attachment of integral type reverse flanges may be attached as shown in Figure 2-4, sketches (5) through (11), as well as Figure UW-13.2, sketches (a) and (b). The requirements of 2-4(c) apply to Figure 2-4, sketches (8) through (11) as well as Figure UW-13.2, sketches (a) and (b).

Figure 2-13.1
Reverse Flange



(2) *Loose Ring Type Reverse Flange*. The shell-to-flange attachment of loose ring type reverse flanges may be attached as shown in Figure 2-4, sketches (3a), (4a), (8), (9), (10), and (11) as well as Figure UW-13.2, sketches (c) and (d). When Figure UW-13.2, sketches (c) and (d) are used, the maximum wall thickness of the shell shall not exceed $^3/_8$ in. (10 mm), and the maximum design metal temperature shall not exceed 650°F (340°C).

The symbols and definitions in this paragraph pertain specifically to reverse flanges. Except as noted in (b) below, the symbols used in the equations of this paragraph are defined in 2-3.

The equations for S_H, S_R, and S_{T1} correspond, respectively, to eqs. 2-7(a)(8), 2-7(a)(9), and 2-7(a)(10), in direction, but are located at the flange *outside* diameter. The sole stress at the flange inside diameter is a tangential stress and is given by the formula for S_{T2}.

(b) *Notation*

B = inside diameter of shell

B' = inside diameter of reverse flange

d_r = $U_r h_{or} g_o^2 / V$

e_r = F / h_{or}

F = factor (use h_{or} for h_o in Figure 2-7.2)

f = factor (use h_{or} for h_o in Figure 2-7.6)

H = total hydrostatic end force on attached component
= $0.785 G^2 P$

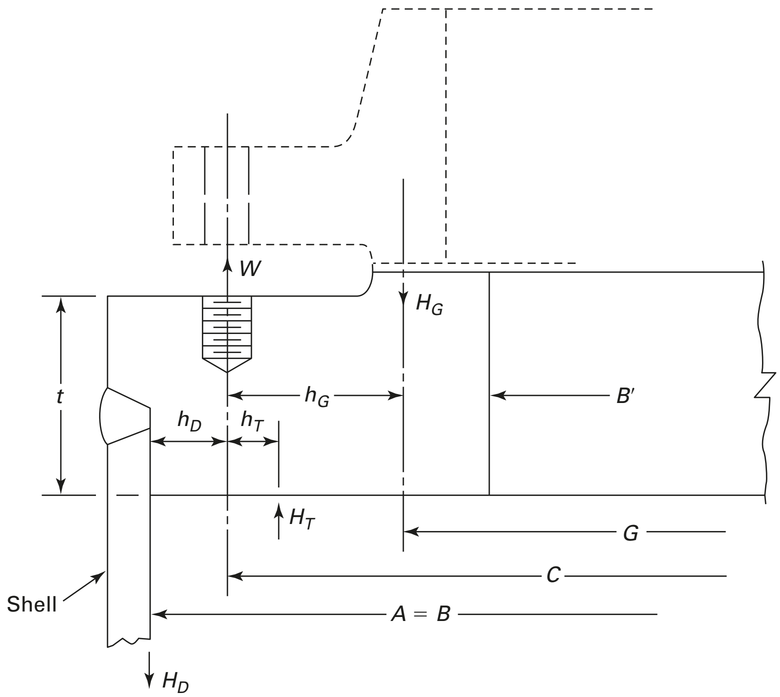
H_D = hydrostatic end force on area inside of flange
= $0.785 B^2 P$

H_T = difference between hydrostatic end force on attached component and hydrostatic end force on area inside of flange
= $H - H_D$

h_D = radial distance from the bolt circle to the circle on which H_D acts
= $(C + g_1 - 2g_o - B) / 2$ for integral type reverse flanges
= $(C - B) / 2$ for loose ring type reverse flanges

h_{or} = factor
= $\sqrt{A g_o}$

Figure 2-13.2
Loose Ring Type Reverse Flange



h_T = radial distance from the bolt circle, to the circle on which H_T acts

$= \frac{1}{2}\left(C - \frac{B+G}{2}\right)$

K = ratio of outside diameter of flange to inside diameter of flange

$= A/B'$

L_r = factor

$= \frac{te_r + 1}{T_r} + \frac{t^3}{d_r}$

M_o = total moment acting on the flange, for the operating conditions or gasket seating as may apply

= algebraic sum of M_D, M_T, and M_G. Values of load H_T and moment arm h_D are negative; value of moment arm h_T may be positive as in Figure 2-13.1, or negative. If M_o is negative, use its absolute value in calculating stresses to obtain positive stresses for comparison with allowable stresses.

$T_r = \left(\frac{Z+0.3}{Z-0.3}\right)\alpha_r T$

$U_r = \alpha_r U$

V = factor (use h_{or} for h_o in Figure 2-7.3)

$Y_r = \alpha_r Y$

$\alpha_r = \left[1 + \frac{0.668(K+1)}{Y}\right] / K^2$

(c) For Integral Type Reverse Flanges

(1) Stresses at the Outside Diameter

$$S_H = f M_o / L_r g_1^2 B'$$

$$S_R = (1.33te_r + 1) M_o / L_r t^2 B'$$

$$S_{T1} = \left(Y_r M_o / t^2 B'\right) - Z S_R (0.67te_r + 1) / (1.33te_r + 1)$$

(2) Stress at Inside Diameter B′

$$S_{T2} = \left(M_o / t^2 B'\right)\left[Y - \frac{2K^2\left(1 + \frac{2}{3}te_r\right)}{(K^2 - 1)L_r}\right]$$

(d) For Loose Ring Type Reverse Flanges

$$S_T = Y M_o / t^2 B'$$

$$S_R = 0$$

$$S_H = 0$$

2-14 FLANGE RIGIDITY

(a) Flanges that have been designed based on allowable stress limits alone may not be sufficiently rigid to control leakage. This paragraph provides a method of checking flange rigidity. The rigidity factors provided in Table 2-14 have been proven through extensive user experience for a wide variety of joint design and service conditions. The use of the rigidity index does not guarantee a leakage rate within established limits. The use of the factors must be considered as only part of the system of joint design and assembly requirements to ensure leak tightness. Successful service experience may be used as an alternative to the flange rigidity rules for fluid services that are nonlethal and nonflammable and designed within the temperature range of −20°F (−29°C) to 366°F (186°C) without exceeding design pressures of 150 psi (1 035 kPa).

(b) The notation is as follows:

E = modulus of elasticity for the flange material at design temperature (operating condition) or at atmospheric temperature (gasket seating condition), psi
J = rigidity index ≤ 1
K_I = rigidity factor for integral or optional flange types = 0.3
K_L = rigidity factor for loose-type flanges = 0.2

Experience has indicated that K_I and K_L provided above are sufficient for most services; other values may be used with the User's agreement.

Other notation is defined in 2-3 for flanges and 2-13 for reverse flanges.

(c) The rigidity criterion for an integral type flange and for a loose type flange without a hub is applicable to the reverse flanges in Figures 2-13.1 and 2-13.2, respectively. The values of h_{or} shall be substituted for h_o, and the value L_r shall be substituted for the value L in the rigidity equation for integral type flanges. Also substitute h_{or} for h_o in determining the factor V in the equation for integral type flanges.

(d) If the value of J, when calculated by the appropriate formula above, is greater than 1.0, the thickness of the flange, t, shall be increased and J recalculated until $J \le 1$ for both gasket seating and operating conditions.

2-15 QUALIFICATION OF ASSEMBLY PROCEDURES AND ASSEMBLERS

It is recommended that flange joints designed to this Appendix be assembled by qualified procedures and by qualified assemblers. ASME PCC-1 may be used as a guide.

Table 2-14
Flange Rigidity Factors

Flange Type	Rigidity Criterion
Integral-type flanges and optional type flanges designed as integral-type flanges	$J = \dfrac{52.14 V M_o}{L E g_o^2 K_I h_o} \le 1.0$
Loose-type flanges with hubs	$J = \dfrac{52.14 V_L M_o}{L E g_o^2 K_L h_o} \le 1.0$
Loose-type flanges without hubs and optional flanges designed as loose-type flanges	$J = \dfrac{109.4 M_o}{E t^3 K_L (\ln K)} \le 1.0$

MANDATORY APPENDIX 5
FLEXIBLE SHELL ELEMENT EXPANSION JOINTS

5-1 GENERAL

(a) Flexible shell element expansion joints used as an integral part of heat exchangers or other pressure vessels shall be designed to provide flexibility for thermal expansions and also function as pressure-containing elements. The rules in this Appendix are intended to apply to typical single-layer flexible shell element expansion joints shown in Figure 5-1 and are limited to applications involving only axial deflections. The suitability of the expansion joint for the specified design, pressure, and temperature shall be determined by methods described in this Appendix.

(b) In all vessels with expansion joints, the hydrostatic end force caused by pressure and/or the joint spring force shall be contained by adequate restraining elements (i.e., tube bundle, tubesheets or shell, external bolting, anchors, etc.). The average primary membrane stress [see UG-23(c)] in these restraining elements shall not exceed the maximum allowable stress at the design temperature for the material given in the tables given in Section II, Part D, Subpart 1.

(c) If expansion-joint flexible elements are to be extended, compressed, rotated, or laterally offset to accommodate connecting parts that are not properly aligned, such movements shall be considered in the design.

(d) The rules of this Appendix do not address cyclic loading conditions; therefore, consideration of cyclic loading for flexible shell element expansion joints is not required unless it is specified for the vessel. The user or his designated agent is cautioned that the design of some pressure vessels containing expansion joints (especially expansion joints with corners) may be governed by cyclic loading. It is recommended that cyclic conditions be included with the specification (see Nonmandatory Appendix KK).

(e) Elastic moduli, yield strengths, and allowable stresses shall be taken at the design temperatures. However, for cases involving thermal loading, it is permitted to use the operating metal temperature instead of the design temperature.

(f) The rules in this Appendix cover the common types of flexible shell element expansion joints but are not intended to limit configurations or details to those illustrated or otherwise described herein. Designs that differ from those covered in this Appendix (e.g., multilayer, asymmetric geometries or loadings having a thick liner or other attachments) shall be in accordance with U-2(g).

5-2 MATERIALS

Materials for pressure-retaining components shall conform to the requirements of UG-4. For carbon and low alloy steels, minimum thickness exclusive of corrosion allowance shall be 0.125 in. (3 mm) for all pressure-containing parts. The minimum thickness for high alloy steel shall conform to requirements of UG-16.

5-3 DESIGN

The design of expansion joints shall conform to the requirements of Part UG and those of (a) through (f) below.

(a) Except as permitted by UHX-17(b), the design of expansion joint flexible elements shall satisfy the following stress limits [see (b) below]. These stress limits shall be met in both the corroded and noncorroded conditions.

(1) Mechanical Loads Only. Mechanical loads include pressure and pressure-induced axial deflection. The maximum stress in the joint is limited to 1.5S [where S is the maximum allowable stress value (see UG-23) for the joint material].

(2) Thermally Induced Displacements Only. The maximum stress in the joint is limited to S_{PS} [see UG-23(e)].

(3) Mechanical Loads Plus Thermally Induced Displacements. The maximum stress in the joint is limited to S_{PS}.

(b) The calculation of the individual stress components and their combination shall be performed by a method of stress analysis that can be shown to be appropriate for expansion joints.

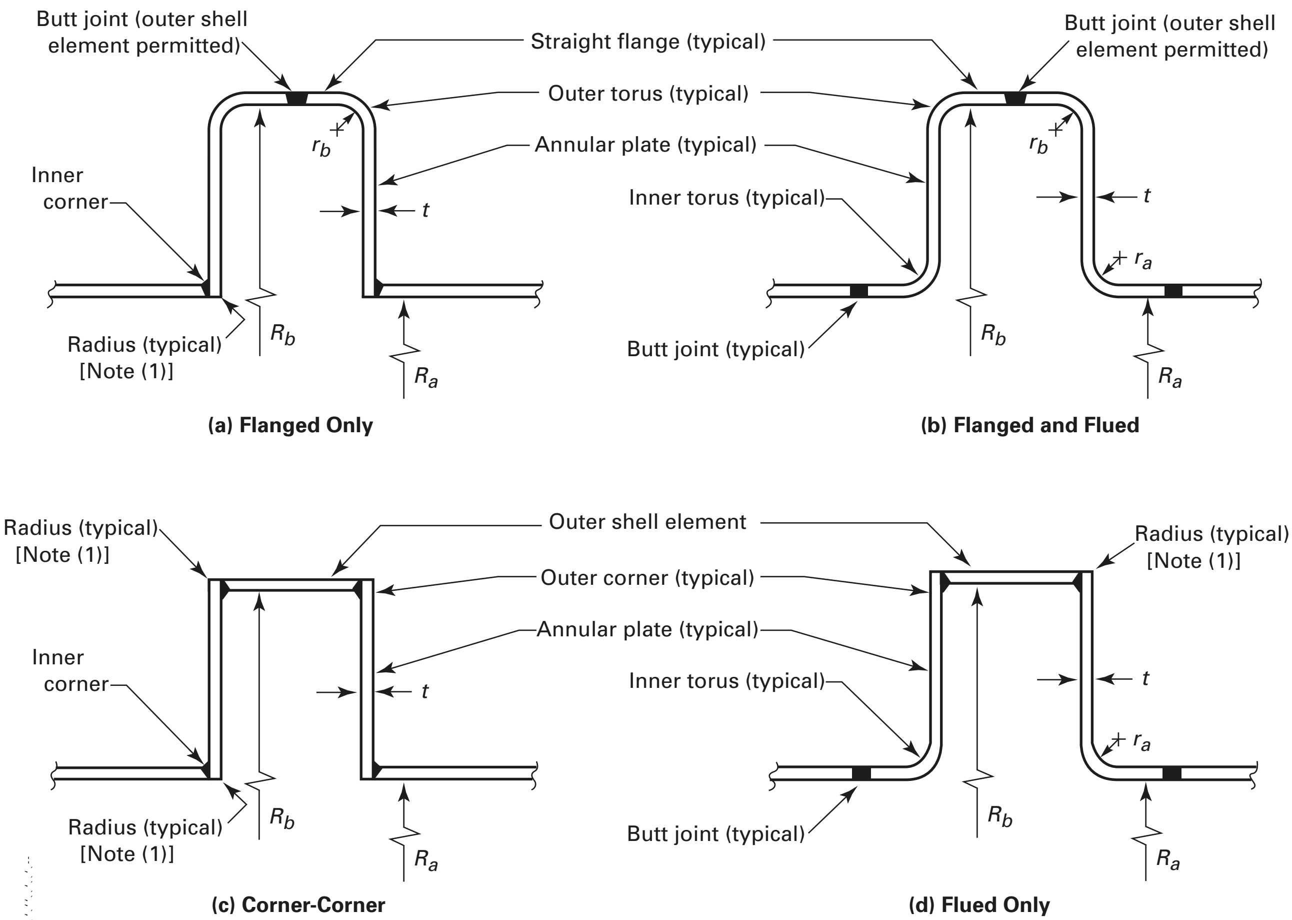
(c) The knuckle radius r_a or r_b of any formed element shall not be less than three times the element thickness t as shown in Figure 5-1.

(d) The spring rate of the expansion joint assembly may be determined either by calculation or by testing.

(e) Thinning of any flexible element as a result of forming operations shall be considered in the design and specifications of material thickness.

(f) Extended straight flanges between the inner torus and the shell and between both outer tori are permissible. An outer shell element between the outer tori is permissible. Extended straight flanges between the inner torus and the shell, between the outer tori and the outer shell element, and between both outer tori that do not have

Figure 5-1
Typical Flexible Shell Element Expansion Joints



Legend:

R_a, R_b = inside radius of expansion joint straight flange, shell, or outer shell element
t = thickness of expansion joint flexible element

GENERAL NOTE: r_a, $r_b \geq 3t$.

NOTE:
(1) Where the term "Radius" appears, provide a $^1\!/_8$-in. (3-mm) minimum blend radius.

an intermediate outer shell element with lengths in excess of $0.5\sqrt{Rt_f}$ shall satisfy all the requirements of UG-27 where

R = inside radius of expansion joint straight flange at the point of consideration
= R_a or R_b
t_f = uncorroded thickness of expansion joint straight flange

5-4 FABRICATION

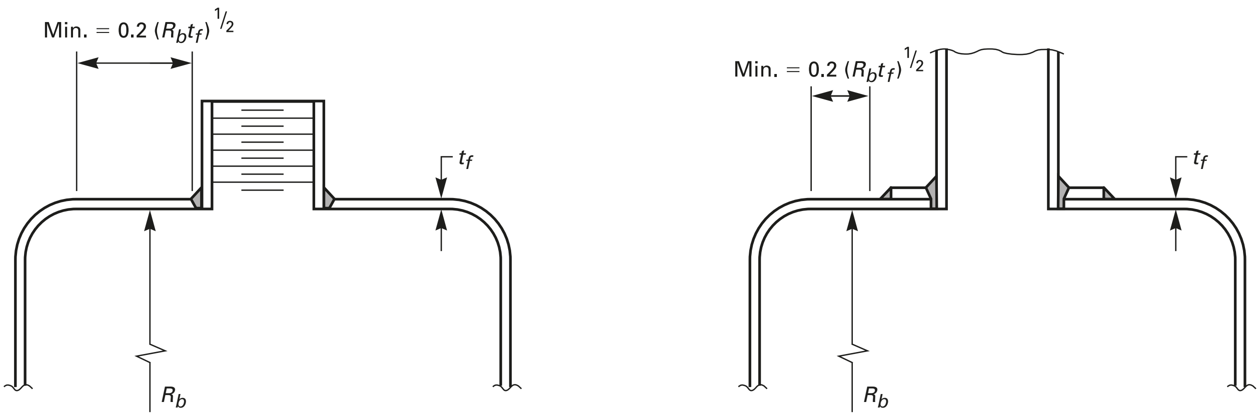
(a) The flexible element is the flanged-only head, the flanged-and-flued head, the annular plate, or the flued-only head, as appropriate to the expansion joint configuration per Figure 5-1. The flexible element may be fabricated from a single plate (without welds) or from multiple plates or shapes welded together. When multiple plates or shapes are used to fabricate the flexible element, the following requirements apply:

(1) Welds shall be butt-type full penetration welds, Type (1) of Table UW-12.

(2) Welds shall be ground flush and smooth on both sides. For flexible elements to be formed, this shall be done prior to forming.

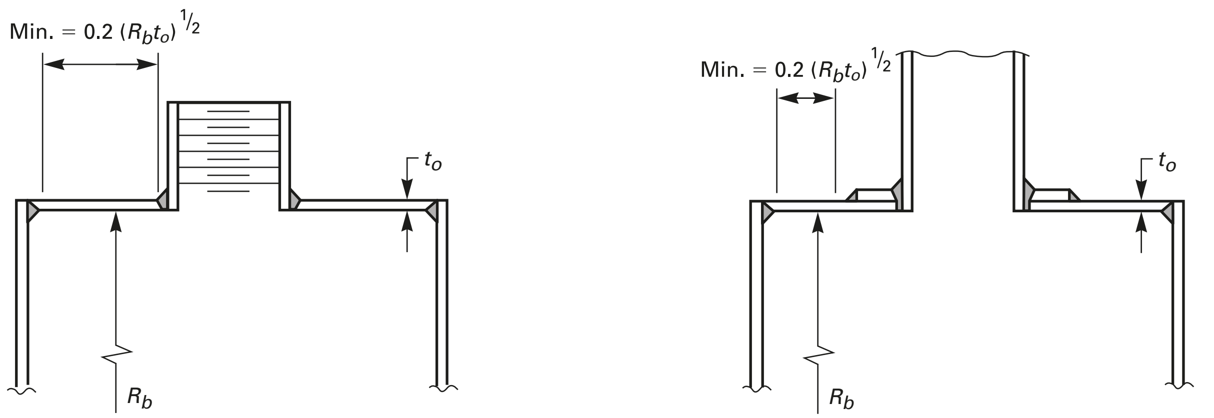
(b) The circumferential weld attaching the flexible element to the shell, mating flexible element, or outer shell element, as appropriate to the expansion joint configuration per Figure 5-1, shall be as follows:

Figure 5-2
Typical Nozzle Attachment Details Showing Minimum Length of Straight Flange or Outer Shell Element



(a) Nonreinforced Nozzle on Straight Flange

(b) Reinforced Nozzle on Straight Flange



(c) Nonreinforced Nozzle on Outer Shell Element

(d) Reinforced Nozzle on Outer Shell Element

Legend:

R_b = inside radius of expansion joint straight flange
t_f = uncorroded thickness of expansion joint straight flange
t_o = uncorroded thickness of expansion joint outer shell element

(1) Butt joints shall be full penetration welds, Type (1) of Table UW-12.

(2) Corner joints shall be full penetration welds with a covering fillet and no backing strip. The covering fillet shall be located on the inside of the corner and shall have a throat at least equal to 0.7 times the minimum thickness of the elements being joined, or $^1/_4$ in. (6 mm) (note that a fatigue evaluation may require a larger weld). It is permitted for the corner weld to be full penetration through either element being joined.

(c) Nozzles, backing strips, clips, or other attachments shall not be located in highly stressed areas of the expansion joint, i.e., inner torus, annular plate, and outer torus. As an exception, a thin cylindrical liner, having approximately the shell inside diameter, may be attached to an inner torus or an annular plate inner corner. A liner is considered thin when its thickness is no more than $t/3$; however, it need not be thinner than $^1/_{16}$ in. (1.6 mm). This liner shall be attached to only one side. The weld attaching the liner shall have a maximum dimension (groove depth or either fillet leg) no larger than the liner thickness. Nozzles or other attachments located in the outer straight flange or outer shell element shall satisfy the axial spacing requirements of Figure 5-2.

(d) The welds within the shell courses adjacent to flexible elements shall be full penetration butt welds, Type (1) of Table UW-12 , for a distance of $2.5\sqrt{Rt}$, where R is R_a or R_b, and t is the thickness of the shell or outer shell element, as applicable.

(e) Alignment tolerances of the completed expansion joint attached to the shell shall meet the tolerances specified by UW-33.

5-5 INSPECTION AND TESTS

(a) Expansion joint flexible elements shall be visually examined and found free of unacceptable surface conditions, such as notches, crevices, and weld spatter, which may serve as points of local stress concentration. Suspect surface areas shall be further examined by the magnetic particle or liquid penetrant method in accordance with Mandatory Appendix 6 or Mandatory Appendix 8.

(b) Welds within the flexible element shall be 100% examined in accordance with UW-51. These welds shall be examined 100% on both sides by the magnetic particle or liquid penetrant method in accordance with Mandatory Appendix 6 or Mandatory Appendix 8. For flexible elements to be formed, this surface inspection shall be after forming.

(c) The circumferential welds attaching the flexible element to the shell, mating flexible element, or outer shell element, as appropriate to the expansion joint configuration per Figure 5-1, shall be examined 100% on both sides, where accessible, by the magnetic particle or liquid penetrant method in accordance with Mandatory Appendix 6 or Mandatory Appendix 8. The accessibility of welds shall be subject to the acceptance of the Inspector.

(d) The completed expansion joint shall be pressure tested in accordance with UG-99 or UG-100. The pressure testing may be performed as a part of the final vessel pressure test, provided the joint is accessible for inspection during pressure testing.

(e) Expansion joint restraining elements shall also be pressure tested in accordance with UG-99 or UG-100 as a part of the initial expansion joint pressure test or as a part of the final vessel pressure test after installation of the joint.

(f) In addition to inspecting the expansion joint for leaks and structural integrity during the pressure test, expansion joints shall be inspected before, during, and after the pressure test for visible permanent distortion.

5-6 MARKING AND REPORTS

The expansion joint Manufacturer, whether the vessel Manufacturer or a parts Manufacturer, shall have a valid ASME Code U Certificate of Authorization and shall complete the appropriate Data Report in accordance with UG-120.

(a) The Manufacturer responsible for the expansion joint design shall include the following additional data and statements on the appropriate Data Report:

(1) uncorroded and corroded spring rate

(2) axial movement (+ and −) and associated loading condition, if applicable

(3) that the expansion joint has been constructed to the rules of this Appendix

(b) A parts Manufacturer shall identify the vessel for which the expansion joint is intended on the Partial Data Report.

(c) Markings shall not be stamped on the flexible elements of the expansion joint.

MANDATORY APPENDIX 23
EXTERNAL PRESSURE DESIGN OF COPPER, COPPER ALLOY, AND TITANIUM ALLOY CONDENSER AND HEAT EXCHANGER TUBES WITH INTEGRAL FINS

23-1 SCOPE

The rules in this Appendix cover the proof test procedure and criteria for determining the maximum allowable external working pressure of copper, copper alloy, and titanium alloy condenser and heat exchanger tubes with helical fins that are integrally extended from the tube wall as an alternative to the requirements of UG-8(b)(4). This Appendix may only be used when the specified corrosion allowance for the tubes is zero. In addition, when using SB-543, this Appendix may only be used when the finning operations are performed after the tubes have been welded, tested, and inspected according to SB-543.

23-2 MATERIALS

(a) Copper and copper alloy tubes shall meet SB-359, SB-543, or SB-956.

(b) Titanium alloy tubes shall meet SB-338.

23-3 TEST PROCEDURE

(a) Test three full size specimens to failure (visible collapse) by external hydrostatic pressure.

(b) The maximum allowable working pressure P shall be determined by

$$P = F\left(\frac{B}{3}\right)\left(\frac{Y_s}{Y_a}\right)$$

where

B = minimum collapse pressure, psi (kPa)
F = factor to adjust for change in strength due to design temperature
= S/S_2
S = maximum allowable stress value for the tube material at design temperature, as given in the tables referenced in UG-23 but not to exceed S_2, psi
S_2 = maximum allowable stress value for the tube material at test temperature, as given in the tables referenced in UG-23, psi
Y_a = actual average yield strength determined from the unfinned length of the three specimens tested at room temperature, psi (kPa)
Y_s = specified minimum yield strength at room temperature, psi (kPa)

23-4 CRITERIA

(a) The design of copper and copper alloy finned tubes to this Appendix shall meet the following requirements:

(1) Design temperature shall be limited to the maximum temperature listed in Section II, Part D, Subpart 1, Table 1B corresponding to the time independent allowable stress, or the maximum temperature shown on the external pressure chart for the corresponding material, whichever is less.

(2) Tubes shall have external and/or internal integrally extended helical fins and the sum of external plus internal fins shall be at least 10 fins/in. (10 fins/25 mm).

(3) Dimensions and permissible variations shall be as specified in Item 15 of SB-359 or SB-956.

(b) The design of titanium alloy finned tubes to this Appendix shall meet the following requirements:

(1) Design temperature shall not exceed 600°F (315°C).

(2) Tubes shall have external integrally extended helical fins only and shall have at least 10 fins/in. (10 fins/25 mm).

(3) Dimensions and permissible variations shall be as specified in item 15 of SB-359 (Specification for Copper and Copper-Alloy Seamless Condenser and Heat Exchanger Tubes With Integral Fins).

(c) Additional requirements for copper, copper alloy, and titanium alloy tubes designed to this Appendix are as follows:

(1) Test specimens shall be identical in fin geometry and pitch to production tubes.

(2) Test specimens of 50 outside diameters or more in length shall qualify all totally finned lengths.

(3) Unfinned length at the ends or at an intermediate section shall qualify that length and all lesser unfinned lengths.

(4) Nominal wall thickness under the fin and at the unfinned area shall qualify all thicker wall sections but with no increase in P.

(5) Outside diameter of the finned section shall not exceed the outside diameter of the unfinned section.

(6) Tests shall be done in accordance with 23-3, witnessed by and subjected to the acceptance of the Inspector.

23-5 DATA REPORTS

When all the requirements of this Division and the supplemental requirements of this Appendix have been met, the following notation shall be entered on the Manufacturer's Data Report under Remarks: "Constructed in Conformance with Mandatory Appendix 23, External Pressure Design of Copper, Copper Alloy, and Titanium Alloy Condenser and Heat Exchanger Tubes With Integral Fins."

MANDATORY APPENDIX 26
BELLOWS EXPANSION JOINTS

26-1 SCOPE

(a) The rules in this Appendix cover the minimum requirements for the design of bellows expansion joints used as an integral part of heat exchangers or other pressure vessels. These rules apply to single or multiple layer bellows expansion joints, unreinforced, reinforced or toroidal, as shown in Figure 26-1-1, subject to internal or external pressure and cyclic displacement. The bellows shall consist of single or multiple identically formed convolutions. They may be as formed (not heat treated), or annealed (heat treated). The suitability of an expansion joint for the specified design pressure, temperature, and axial displacement shall be determined by the methods described herein.

(b) The rules in this Appendix cover the common types of bellows expansion joints but are not intended to limit the configurations or details to those illustrated or otherwise described herein. Designs that differ from those covered in this Appendix (e.g., asymmetric geometries or loadings) shall be in accordance with U-2(g).

(19) 26-2 CONDITIONS OF APPLICABILITY

The design rules of this Appendix are applicable only when the following conditions of applicability are satisfied:

(a) The bellows shall be such that $Nq \le 3D_b$.

(b) The bellows nominal thickness shall be such that $nt \le 0.2$ in. (5.0 mm).

(c) The number of plies shall be such that $n \le 5$.

(d) The displacement shall be essentially axial. However angular and/or lateral deflection inherent in the fit-up of the expansion joint to the pressure vessel is permissible, provided the amount is specified and is included in the expansion joint design [see 26-4.1(d)].

(e) These rules are valid for design temperatures (see UG-20) up to the temperatures shown in Table 26-2-1. Above these temperatures, the effects of time-dependent behavior (creep and creep–fatigue interaction) shall be considered in accordance with U-2(g).

(f) The fatigue equations given in 26-6.6.3.2, 26-7.6.3.2, and 26-8.6.3.2 are valid for austenitic chromium-nickel stainless steels, UNS N066XX and UNS N04400. For other materials, the fatigue evaluation shall meet the requirements of 26-4.2.3.

(g) The length of the cylindrical shell on each side of the bellows shall not be less than $1.8\sqrt{D_s t_s}$. The length shall be taken from the beginning of the end convolution [point A in Figure 26-1-2, sketches (a) and (b)], except that for internally attached toroidal bellows, the length shall be taken from the extremity of the shell [point B in Figure 26-1-2, sketch (b)].

26-3 NOMENCLATURE (19)

Symbols used in this Appendix are as follows (see Figure 26-1-1):

A = cross-sectional metal area of one convolution

$$= \left[2\pi r_m + 2\sqrt{\left\{\frac{q}{2} - 2r_m\right\}^2 + \{w - 2r_m\}^2}\,\right] n t_p$$

A_f = cross-sectional metal area of one reinforcing fastener

A_r = cross-sectional metal area of one bellows reinforcing member for U-shaped bellows, and cross-sectional metal area of one reinforcing collar for toroidal bellows based on length L_r

A_{rt} = cross-sectional metal area of one reinforcing collar for toroidal bellows based on overall length

A_{tc} = cross-sectional metal area of one tangent collar

A_{ts} = cross-sectional metal area of shell based on length L_s

B_1, B_2, B_3 = coefficients used for toroidal bellows, given by Table 26-8

C_1, C_2 = coefficients given by equations, used to determine coefficients C_p, C_f, C_d

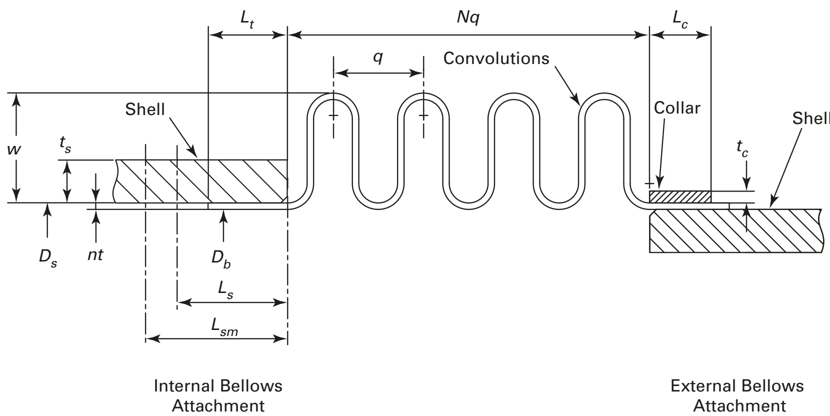
$$C_1 = \frac{2r_m}{w}$$

$$C_2 = \frac{1.82 r_m}{\sqrt{D_m t_p}}$$

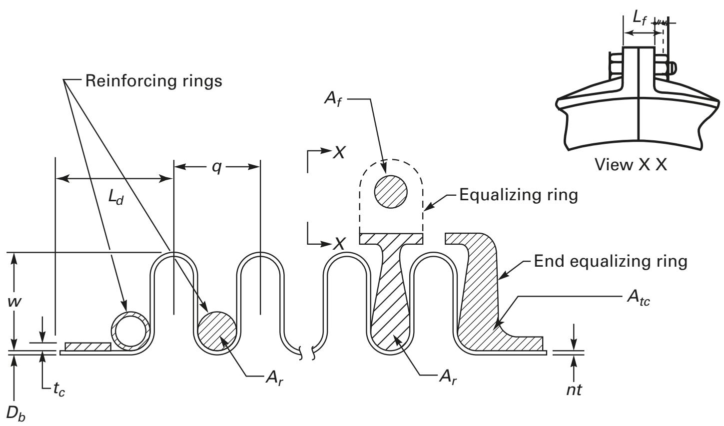
C_3 = coefficient used to determine coefficients B_1, B_2, and B_3

C_p, C_f, C_d = coefficients for U-shaped convolutions, given by Figure 26-4, 26-5, and 26-6

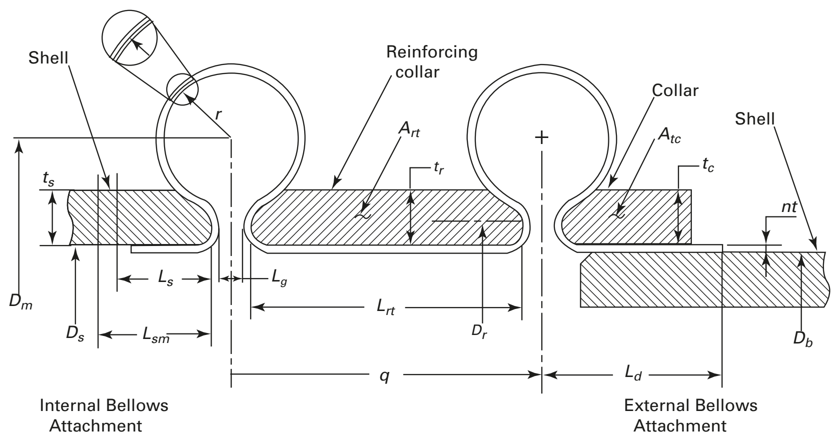
Figure 26-1-1
Typical Bellows Expansion Joints



(a) Unreinforced Bellows

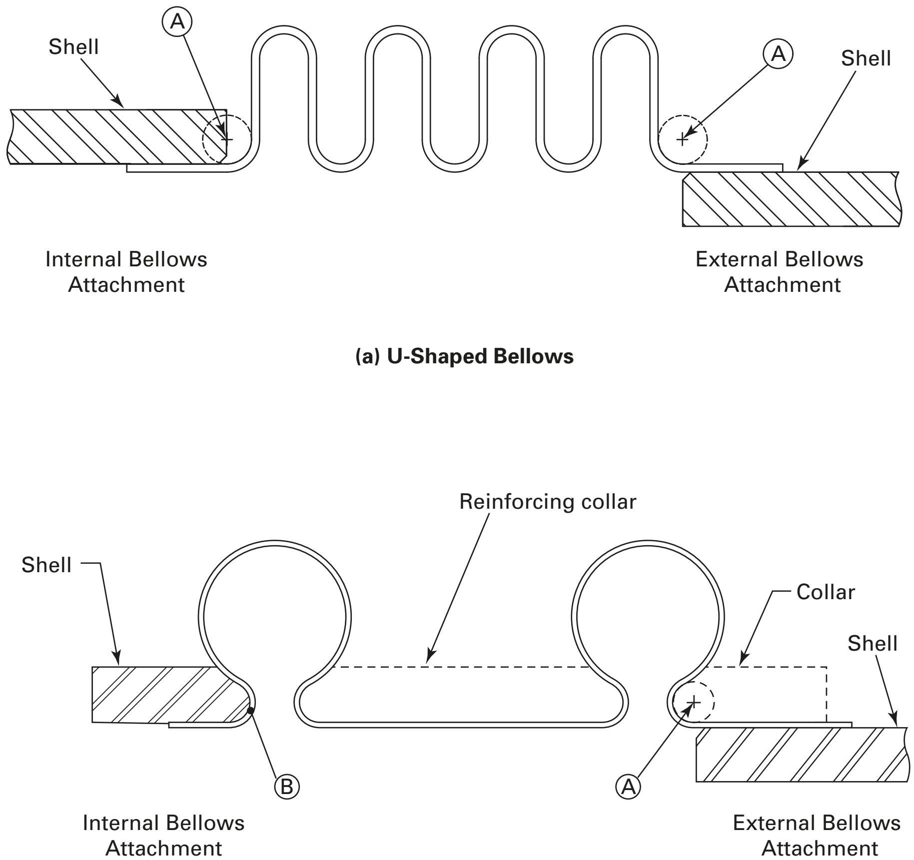


(b) Reinforced Bellows



(c) Toroidal Bellows

Figure 26-1-2
Starting Points for the Measurement of the Length of Shell on Each Side of the Bellows



(b) Toroidal Bellows

Table 26-2-1
Maximum Design Temperatures for Application of the Rules of Mandatory Appendix 26

Table in Which Material Is Listed	Maximum Temperature, °F	Maximum Temperature, °C
UNF-23.1	300	150
UNF-23.3	800	425
UNF-23.4	600	315
UNF-23.5	600	315
UHA-23	800	425

C_r = convolution height factor for reinforced bellows

$$= 0.3 - \left(\frac{100}{K_c P^{1.5} + 320}\right)^2$$

where

K_c = 0.6 if P is expressed in psi
= 1,048 if P is expressed in MPa

C_{wc} = longitudinal weld joint efficiency of tangent collar (see UW-12)

C_{wr} = longitudinal weld joint efficiency of reinforcing member (see UW-12)

C_{ws} = longitudinal weld joint efficiency of shell (see UW-12)

D_b = inside diameter of bellows convolution and end tangents

D_c = mean diameter of tangent collar

$= D_b + 2nt + t_c$

D_m = mean diameter of bellows convolution
= $D_b + w + nt$ for U-shaped bellows

D_r = mean diameter of reinforcing collar for toroidal bellows

D_s = inside diameter of cylindrical shell or weld end on which the bellows is attached

E_b = modulus of elasticity of bellows material at design temperature

E_c = modulus of elasticity of collar material at design temperature

E_f = modulus of elasticity of reinforcing fastener material at design temperature

E_o = modulus of elasticity of bellows material at room temperature

E_r = modulus of elasticity of reinforcing ring member material at design temperature

E_s = modulus of elasticity of shell or weld end material at design temperature

G_b = modulus of rigidity of bellows material at design temperature

$$= \frac{E_b}{2(1 + \nu_b)}$$

H = resultant total internal pressure force acting on the bellows and reinforcement
= PD_mq

k = factor considering the stiffening effect of the attachment weld and the end convolution on the pressure capacity of the end tangent

$$= \mathrm{MIN}\left[\left(\frac{L_t}{1.5\sqrt{D_b t}}\right), (1.0)\right]$$

K_0, K_1, K_2, K_3 = coefficients determined by best curve fit of bellows fatigue test data

K'_0, K'_1, K'_2, K'_3 = coefficients determined by best curve fit of bellows fatigue test data

K_b = bellows axial stiffness

K_f = forming method factor
= 1.0 for expanding mandrel or roll forming
= 0.6 for hydraulic, elastomeric, or pneumatic tube forming

L_c = bellows collar length

L_d = length from attachment weld to the center of the first convolution for externally attached bellows

L_{dt} = developed length of one convolution
= A/nt_p for U-shaped bellows

L_f = effective length of one reinforcing fastener. Distance between the mating face of the bolt head and mid-thickness of the nut or distance between mid-thickness of the two nuts, as applicable

L_g = maximum distance across the inside opening of a toroidal convolution considering all movements

L_r = effective reinforcing collar length
= $\sqrt{D_r t_r}/3$

L_{rt} = overall length of reinforcing collar

L_s = effective shell length
= $\sqrt{[(D_s + t_s)t_s]}/3$

L_{sm} = minimum required shell length having thickness t_s

L_t = end tangent length

M_2 = torsional load

N = number of convolutions

n = number of plies

N_{alw} = allowable number of fatigue cycles

N_{spe} = specified number of fatigue cycles

P = design pressure (see UG-21)

NOTE: If the MAWP of the bellows is significantly greater than the required design pressure of the vessel, use of the larger MAWP may adversely affect the allowable number cycles that the bellows can experience.

q = convolution pitch (see Figure 26-1-1)

R = ratio of the internal pressure force resisted by the bellows to the internal pressure force resisted by the reinforcement. Use R_1 or R_2 as designated in the equations.
= R_1 for integral reinforcing ring members

$$R_1 = \frac{A\,E_b}{A_r E_r}$$

= R_2 for reinforcing ring members joined by fasteners

$$R_2 = \frac{A\,E_b}{D_m}\left(\frac{L_f}{A_f E_f} + \frac{D_m}{A_r E_r}\right)$$

r = mean radius of toroidal bellows convolution

r_i = average internal torus radius of U-shaped bellows convolution (see 26-6.2)

r_m = mean torus radius of U-shaped bellows convolution
= $r_i + (nt/2)$

S = allowable stress of bellows material at design temperature

S_1 = circumferential membrane stress in bellows tangent, due to pressure P

S_1' = circumferential membrane stress in collar, due to pressure P

S_1''' = circumferential membrane stress in shell, due to pressure, P, for internally attached bellows

S_2 = circumferential membrane stress in bellows, due to pressure P

S_2' = circumferential membrane stress in reinforcing member, due to pressure P

S_2'' = membrane stress in fastener, due to pressure member P

S_3 = meridional membrane stress in bellows, due to pressure P

S_4 = meridional bending stress in bellows, due to pressure P

S_5 = meridional membrane stress in bellows, due to total equivalent axial displacement range Δq

S_6 = meridional bending stress in bellows, due to total equivalent axial displacement range Δq

S_c = allowable stress of collar material at design temperature

S_f = allowable stress of reinforcing fastener material at design temperature

S_q = total stress range due to cyclic displacement

S_r = allowable stress of reinforcing ring member material at design temperature

S_s = allowable stress of shell material at design temperature

S_t = total stress range due to cyclic displacement corrected by internal pressure

t = nominal thickness of one ply

t_c = collar thickness

t_{eq} = equivalent wall thickness

t_p = thickness of one ply, corrected for thinning during forming

$$= t\sqrt{\frac{D_b}{D_m}}$$

t_r = reinforcing collar thickness

t_s = nominal thickness of shell or weld end

w = convolution height

Y_{sm} = yield strength multiplier depending upon material

$= 1 + 9.94(K_f\varepsilon_f) - 7.59(K_f\varepsilon_f)^2 - 2.4(K_f\varepsilon_f)^3 + 2.21(K_f\varepsilon_f)^4$ for austenitic stainless steel

$= 1 + 6.8(K_f\varepsilon_f) - 9.11(K_f\varepsilon_f)^2 + 9.73(K_f\varepsilon_f)^3 - 6.43(K_f\varepsilon_f)^4$ for nickel alloys

= 1.0 for other materials

If Y_{sm} is less than 1.0, then Y_{sm} = 1.0

If Y_{sm} is greater than 2.0, then Y_{sm} = 2.0

Δq = total equivalent axial displacement range per convolution

ε_f = bellows forming strain

$$= \sqrt{\left[\ln\left(1+\frac{2w}{D_b}\right)\right]^2 + \left[\ln\left(1+\frac{nt_p}{2r_m}\right)\right]^2}$$

for bellows formed from cylinders with an inside diameter of D_b if forming is performed 100% to the outside of the initial cylinder

$$= \sqrt{\left[\ln\left(1+\frac{w}{D_b}\right)\right]^2 + \left[\ln\left(1+\frac{nt_p}{2r_m}\right)\right]^2}$$

for bellows formed from cylinders with an inside diameter of D_m if forming is performed 50% to the inside and 50% to the outside of the initial cylinder

θ_z = twist angle between the two extreme points of the end convolutions

ν_b = Poisson's ratio of bellows material

τ_z = shear stress due to torsional load or twist angle

Main subscripts:

b = for bellows
c = for collars
p = for ply
r = for reinforced
s = for shell
t = for end tangent

NOTE: No subscript is used for the bellows convolutions.

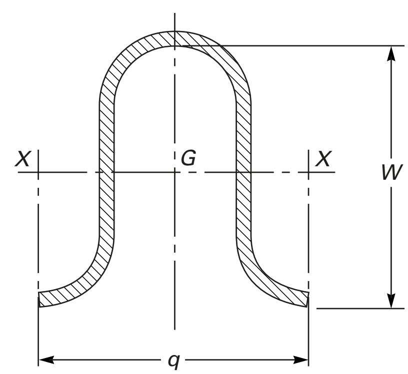
26-4 DESIGN CONSIDERATIONS

26-4.1 GENERAL

(a) Expansion joints shall be designed to provide flexibility for thermal expansion and also to function as a pressure-containing element.

(b) The vessel manufacturer shall specify the design conditions and requirements for the detailed design and manufacture of the expansion joint. Use of Specification Sheet Form 26-1 or Form 26-1M is recommended.

Figure 26-2
Dimensions to Determine I_{xx}



(c) In all vessels with integral expansion joints, the hydrostatic end force caused by pressure and/or the joint spring force shall be resisted by adequate restraint elements (e.g., exchanger tubes or shell, external restraints, anchors, etc.). The stress [see UG-23(c)] in these restraining elements shall not exceed the maximum allowable stress at the design temperature for the material given in the tables referenced by UG-23.

(d) See below.

(1) The expansion joints shall be provided with bars or other suitable members for maintaining the proper overall length dimension during shipment and vessel fabrication. During a heat exchanger pressure test, these bars or members shall not carry load or limit expansion joint movement.

(2) Expansion bellows shall not be extended, compressed, rotated, or laterally offset to accommodate connecting parts that are not properly aligned, unless the design considers such movements. See 26-9.

(3) Care should be taken to ensure that any torsional loads applied to expansion joints are kept to a minimum to prevent high shear stresses that may be detrimental to their use. If torsional loads are present or expected, they shall be considered in the design. See 26-4.3.

(e) The minimum thickness limitations of UG-16(b) do not apply to bellows designed to this Appendix.

(f) Bellows longitudinal weld seams shall have a joint efficiency of 1.0.

(g) Bellows circumferential attachment welds, shells or shell weld ends, and collars shall be in accordance with Figure 26-13, as applicable.

(h) The elastic moduli, yield strength, and allowable stresses shall be taken at the design temperatures. However, when performing the fatigue evaluation in accordance with 26-6.6 (unreinforced bellows), 26-7.6 (reinforced bellows), and 26-8.6 (toroidal bellows), it is permitted to use the operating metal temperature instead of the design temperature.

(19) 26-4.2 FATIGUE

26-4.2.1 Cumulative Damage. If there are two or more types of stress cycles that produce significant stresses, their cumulative effect shall be evaluated as given below.

(a) Procedure

(1) Designate the specified number of times each type of stress cycle of Types 1, 2, 3, etc., of stress range S_{t1}, S_{t2}, S_{t3}, etc., will be repeated during the life of the expansion joint as n_1, n_2, n_3, etc., respectively.

(2) For each value S_{t1}, S_{t_2}, S_{t_3}, etc., use the applicable design fatigue curve to determine the maximum number of repetitions which would be allowable if this type of cycle were the only one acting. Call these values N_1, N_2, N_3, etc.

(3) For each type of stress cycle, calculate the usage factors U_1, U_2, U_3, etc., from

$$U_1 = n_1 / N_1$$
$$U_2 = n_2 / N_2$$
$$U_3 = n_3 / N_3, \text{ etc.}$$

(4) Calculate the cumulative usage factor U from:

$$U = U_1 + U_2 + U_3 + \ldots$$

(5) The cumulative usage factor U shall not exceed 1.0.

(b) Cycle Counting. Stresses to be used for cycle counting shall be based on the total equivalent axial displacement of each convolution, Δq_e or Δq_c, at the top and bottom of each cycle, as determined in 26-9.5, not the range, Δq, determined in 26-9.6. Only the displacements shall be taken into account; pressure shall be neglected. The total equivalent axial displacement range, Δq, to be used for the calculation of the total stress range due to cyclic displacement, S_t, in the fatigue evaluation in 26-6.6, 26-7.6, or 26-8.6 shall be deduced from the stress range, S_q, obtained.

(1) Concurrent Conditions. When determining n_1, n_2, n_3, etc., and S_{q1}, S_{q2}, S_{q3}, etc., consideration shall be given to the superposition of cycles of various origins that produces a total stress range greater than the stress ranges of the individual cycles. For example, if one type of stress cycle produces 1,000 cycles of a stress variation from −1,000 psi to +60,000 psi and another type of stress cycle produces 10,000 cycles of a stress variation from −1,000 psi to −50,000 psi, the two types of cycles to be considered are defined by the following parameters:

(-a) Type 1 Cycle

$$n_1 = 1{,}000$$
$$S_{q1} = |60{,}000| - (-1{,}000)| + |-50{,}000 - (-1{,}000)| = 110{,}000 \text{ psi}$$

(-b) Type 2 Cycle

$$n_2 = 10{,}000 - 1{,}000 = 9{,}000$$
$$S_{q2} = |0| + |-50{,}000 - (-1{,}000)| = 49{,}000 \text{ psi}$$

(2) Independent Conditions. When no superposition of cycles can occur, cycle counting shall be simply based on the stress ranges of the individual cycles. For example, if one type of stress cycle produces 1,000 cycles of a stress variation from −1,000 psi to +60,000 psi and another type of stress cycle produces 10,000 cycles of a stress variation from −1,000 psi to −50,000 psi, the two types of cycles to be considered are defined by the following parameters:

(-a) Type 1 Cycle

$$n_1 = 1{,}000$$
$$S_{q1} = |60{,}000| - (-1{,}000)| = 61{,}000 \text{ psi}$$

(-b) Type 2 Cycle

$$n_2 = 10{,}000$$
$$S_{q2} = |-50{,}000 - (-1{,}000)| = 49{,}000 \text{ psi}$$

(3) Alternatively, when the cyclic displacement history is known, cycle counting may be performed by the Rainflow Method described in Section VIII, Division 2, Annex 5-B, or an equivalent method.

(4) If only the overall number of cycles of each range is known, or in case of doubt, cycle counting shall be performed considering concurrent conditions.

26-4.2.2 Fatigue Correlation Testing. Fatigue curves in 26-6.6.3.2, 26-7.6.3.2, or 26-8.6.3.2 may be used to design a bellows only if they have been correlated with actual bellows test results obtained by proof or strain gage testing (see UG-101) by the bellows Manufacturer to demonstrate predictability of cyclic life on a consistent series of bellows of the same basic design (convolution shape, reinforcement, number of plies, etc.) and forming process. Annealed and as-formed bellows are considered as separate designs.

(a) The substantiation of the fatigue curves shall be based on data obtained from five separate tests on bellows of the same basic design. When substantiating bellows designs with more than two convolutions in series, the test data shall have been obtained from bellows with a minimum of three convolutions. The effect of pressure shall be considered in the fatigue tests.

For each test data pair (S_t, N), two results shall be computed and compared to the applicable fatigue curve: one result with the number of cycles divided by a design factor of 2.6 (S_t, N/2.6) and the other result with the equivalent stress divided by a design factor of 1.25 (S_t/1.25, N). For a result to be accepted, it must be above the applicable fatigue curve.

If all the results meet the acceptance criterion, the substantiation shall be considered valid. If any result does not meet the acceptance criterion, a retest of five additional bellows shall be made. If all the results of the retest, including design factors, meet the acceptance criterion, the substantiation shall be considered valid. Otherwise a specific fatigue curve shall be established as described in 26-4.2.3 and used for the fatigue design of the bellows. The original test and retest results shall be taken into account to establish the specific fatigue curve.

(b) When S_t and the other appropriate factors are used in the cycle life equations in 26-6.6.3.2, 26-7.6.3.2, or 26-8.6.3.2, the specified number of fatigue cycles, N_{spe}, shall be less than the calculated cycles to failure based on the data obtained by testing. The allowable number of fatigue cycles, N_{alw}, may not be increased above that obtained from the equations in these paragraphs regardless of the test results.

(c) The test results shall be available for review by the Inspector.

(d) The substantiation of the fatigue curve used by the bellows Manufacturer for a bellows design that has shown a history of safe use can be waived provided the manufacturing process remains unchanged.

26-4.2.3 Fatigue Curves for Other Materials. For materials other than those specified in the applicable rules, 26-6.6.3.2, 26-7.6.3.2, or 26-8.6.3.2, specific fatigue curves shall be built. The Manufacturer shall determine the fatigue curve for the material intended for the bellows. This fatigue curve shall not be used for temperatures above the temperature shown in Table 26-2-1 for the tested material. Annealed and as-formed bellows shall be considered as being built with different materials. Different forming methods may have either individual curves established for each method or a single curve established by incorporating test results obtained from at least two bellows formed by each different anticipated forming method.

The procedure applied to determine the fatigue curve shall be as described below. The test results with the subsequent calculations used to determine the fatigue curve shall be available for review by the Inspector.

(a) A minimum of 25 fatigue tests shall be carried out. Each bellows in the test group shall have a minimum of three convolutions and varying geometries, including inside diameter, convolution profile, and thickness. A minimum of three different heats of the intended material shall be used.

(b) Each bellows in the test group shall be submitted to three to five different amplitudes of axial movements with a constant internal pressure applied. To ensure that the equivalent fatigue stress, S_t, is due primarily to cyclic displacement and not to pressure, the pressure-induced component stress shall not be higher than 30% of the equivalent fatigue stress.

(c) The test results shall be obtained by proof or strain gage testing (see UG-101) at room temperature.

(d) The fatigue curves shall be determined as follows:

(1) The best fit curve for the relation between the number of measured cycles to failure and the equivalent fatigue stress, S_t, calculated according to 26-6.6.3.2, 26-7.6.3.2, or 26-8.6.3.2, as applicable, shall be determined and expressed as

$$N = \left(\frac{K_0}{S_t - K'_0} \right)^2$$

(2) The curve shall then be adjusted such that all the test results are on or above the curve. The curve is now expressed as

$$N = \left(\frac{K_1}{S_t - K'_1} \right)^2$$

(3) The final fatigue curve shall be the lower bound curve of the curve obtained by applying a factor of 2.6 on numbers of cycles, expressed as

$$N = \left(\frac{K_2}{S_t - K'_2}\right)^2$$

and of the curve obtained by applying a factor of 1.25 on stresses, expressed as

$$N = \left(\frac{K_3}{S_t - K'_3}\right)^2$$

where

$K_2 = K_1/\sqrt{2.6}$
$K'_2 = K'_1$
$K_3 = K_1/1.25$
$K'_3 = K'_1/1.25$

26-4.3 TORSION

The shear stress due to torsion shall satisfy either of the following criteria:

(a) The shear stress due to torsional load, M_z,

$$\tau_z = \frac{2\left|M_z\right|}{\pi n t D_b^2}$$

shall comply with $\tau_z \leq 0.25S$.

(b) The shear stress due to twist angle, θ_z , expressed in radians,

$$\tau_z = \frac{\left|\theta_z\right| G_b D_b}{2NL_{dt}}$$

shall comply with $\tau_z \leq 0.25S$.

26-5 MATERIALS

Pressure-retaining component materials including the restraining elements covered by 26-4.1(c) shall comply with the requirements of UG-4.

26-6 DESIGN OF U-SHAPED UNREINFORCED BELLOWS

26-6.1 SCOPE

These rules cover the design of bellows having unreinforced U-shaped convolutions. The bellows can be attached to the shell either externally or internally.

Each half convolution consists of a sidewall and two quarter tori of nearly the same radius (at the crest and root of the convolution), in the neutral position, so that the convolution profile presents a smooth geometrical shape as shown in Figure 26-1-1.

26-6.2 CONDITIONS OF APPLICABILITY

These conditions of applicability apply in addition to those listed in 26-2.

(a) A variation of 10% between the crest convolution radius r_{ic} and the root convolution radius r_{ir} is permitted (see Figure 26-3 for the definitions of r_{ic} and r_{ir}).

(b) The torus radius shall be such that $r_i \geq 3t$, where

$$r_i = \frac{r_{ic} + r_{ir}}{2}$$

A smaller torus radius may be used, provided the rules of 26-4.2.2 or 26-4.2.3 are followed and the increased bending stress due to curvature is accounted for in the fatigue correlation testing.

(c) The offset angle of the sidewalls, α, in the neutral position shall be such that $-15 \leq \alpha \leq +15$ deg (see Figure 26-3).

(d) The convolution height shall be such that

$$w \leq \frac{D_b}{3}$$

(e) The type of attachment to the shell (external or internal) shall be the same on both sides.

(f) For internally attached bellows, the length of the shell on each side of the bellows having thickness t_s shall be at least equal to $L_{sm} = 1.8\sqrt{D_s t_s}$.

26-6.3 INTERNAL PRESSURE CAPACITY

26-6.3.1 End Tangent. For externally attached bellows, the circumferential membrane stress due to pressure

$$S_1 = \frac{1}{2}\frac{\left(D_b + nt\right)^2 L_t E_b k}{nt\left(D_b + nt\right) L_t E_b + t_c D_c L_c E_c k} P$$

shall comply with $S_1 \leq S$.

26-6.3.2 Collar or Shell.

(a) For externally attached bellows, the circumferential membrane stress in the collar due to pressure

Figure 26-3
Possible Convolution Profile in the Neutral Position

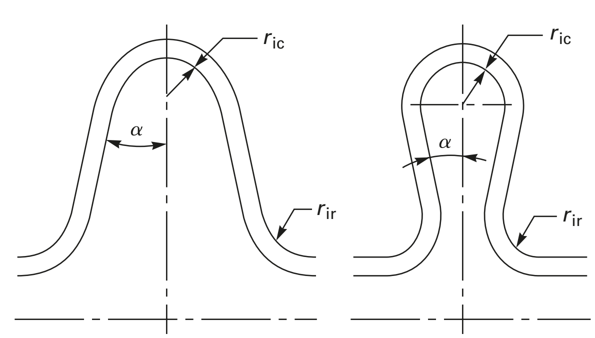
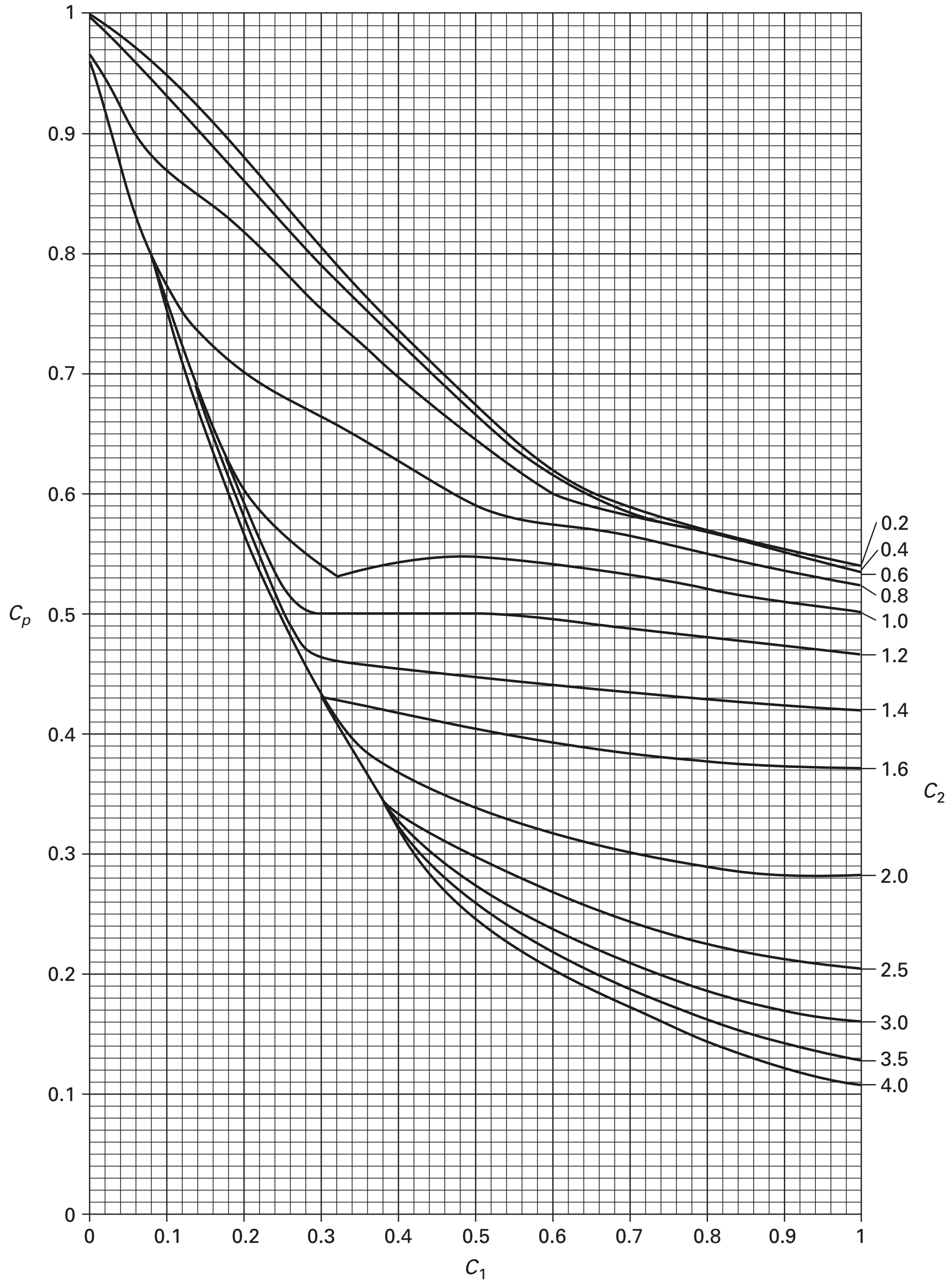
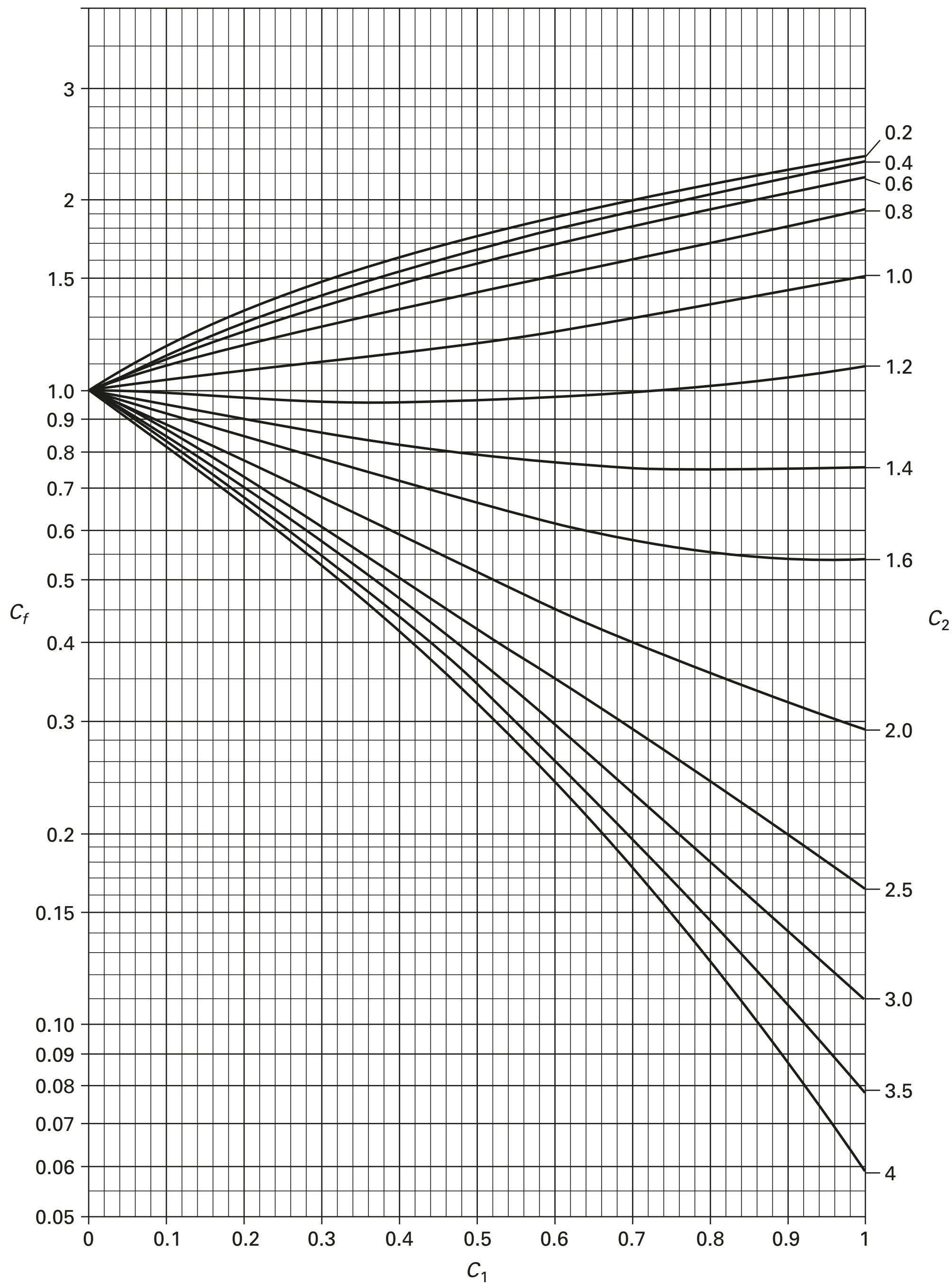


Figure 26-4
Coefficient C_p



GENERAL NOTE: Paragraph 26-15 gives polynomial approximations for these curves when $0.2 \le C_2 \le 4.0$.

Figure 26-5
Coefficient C_f



GENERAL NOTE: Paragraph 26-15 gives polynomial approximations for these curves when $0.2 \leq C_2 \leq 4.0$.

$$S_1' = \frac{1}{2}\frac{D_c^2\, L_t E_c k}{nt\left(D_b + nt\right)L_t E_b + t_c D_c L_c E_c k}P$$

shall comply with $S'_1 \leq C_{wc}\, S_c$.

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

$$S_1''' = \frac{1}{2}\frac{\left(D_s + t_s\right)^2\left(L_s + 0.5q\right)E_s}{nt\left(D_b + nt\right)L_t E_b + t_s\left(D_s + t_s\right)L_s E_s}P$$

shall comply with $S_1''' \leq C_{ws} S_s$.

26-6.3.3 Bellows Convolutions.

(a) The circumferential membrane stress due to pressure

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

$$S_{2,E} = \frac{1}{2}\frac{\left[qD_m + L_t\left(D_b + nt\right)\right]E_b}{\left(A + ntL_t\right)E_b + t_c L_c E_c}P$$

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

(2) for intermediate convolutions

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

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

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

$$S_3 = \frac{w}{2nt_p}P$$

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

$$S_4 = \frac{1}{2n}\left(\frac{w}{t_p}\right)^2 C_p P$$

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

$$S_3 + S_4 \leq K_m S$$

where

K_m = $1.5Y_{sm}$ for as-formed bellows
= 1.5 for annealed bellows

26-6.4 INSTABILITY DUE TO INTERNAL PRESSURE

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

$$P_{sc} = 0.34\frac{\pi K_b}{Nq}$$

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

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

$$P_{si} = \left(\pi - 2\right)\frac{AS_y^*}{D_m q\sqrt{\alpha}}$$

where

$$\alpha = 1 + 2\delta^2 + \sqrt{1 - 2\delta^2 + 4\delta^4}$$

$$\delta = \frac{1}{3}\frac{S_4}{S_{2,I}}$$

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

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

S_y^* = $2.3S_y$ for as-formed bellows
= $0.75S_y$ for annealed bellows

where S_y is the yield strength of bellows material at design temperature, given by Section II, Part D, Subpart 1, Table Y-1. For values not listed in Table Y-1, see UG-28(c)(2), Step 3.

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

The internal pressure shall not exceed P_{si}: $P \leq P_{si}$.

26-6.5 EXTERNAL PRESSURE STRENGTH

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

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

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

(a) an equivalent outside diameter D_{eq} given by

$$D_{eq} = D_b + w + 2t_{eq}$$

(b) an equivalent thickness t_{eq} given by

$$t_{eq} = \sqrt[3]{12\left(1 - \nu_b^2\right)\frac{I_{xx}}{q}}$$

where I_{xx} is the moment of inertia of one convolution cross section relative to the axis passing by the center of gravity and parallel to the axis of the bellows (see Figure 26-2).

NOTE: If L_t = 0, then I_{xx} is given by

$$I_{xx} = nt_p\left[\frac{(2w - q)^3}{48} + 0.4q(w - 0.2q)^2\right]$$

26-6.6 FATIGUE EVALUATION

26-6.6.1 Calculation of Stresses Due to the Total Equivalent Axial Displacement Range Δq of Each Convolution.

(a) Meridional membrane stress:

$$S_5 = \frac{1}{2}\frac{E_b t_p^2}{w^3 C_f}\Delta q$$

(b) Meridional bending stress:

$$S_6 = \frac{5}{3}\frac{E_b t_p}{w^2 C_d}\Delta q$$

26-6.6.2 Calculation of Total Stress Range Due to Cyclic Displacement.

$$S_t = 0.7\left[S_3 + S_4\right] + \left[S_5 + S_6\right]$$

26-6.6.3 Calculation of Allowable Number of Cycles.

(19) **26-6.6.3.1 General.**

(a) The specified number of cycles N_{spe} shall be stated as consideration of the anticipated number of cycles expected to occur during the operating life of the bellows. The allowable number of cycles N_{alw}, as derived in this subclause, shall be at least equal to N_{spe}: $N_{alw} \geq N_{spe}$.

The allowable number of cycles given by the following formulas includes a reasonable safety factor (2.6 on cycles and 1.25 on stresses) and represents the maximum number of cycles for the operating condition considered. Therefore, an additional safety factor should not be applied. An overly conservative estimate of cycles can necessitate a greater number of convolutions and result in a bellows more prone to instability.

(b) If the bellows is subjected to different cycles of pressure or displacement, such as those produced by startup or shutdown, their cumulative damage shall be considered as in 26-4.2.1.

26-6.6.3.2 Fatigue Equation. The following equations are valid for (19)

(a) austenitic chromium-nickel stainless steels, UNS N066XX and UNS N04400 for metal temperatures not exceeding 800°F (425°C). For other materials, the allowable number of cycles, N_{alw}, shall be calculated using the following equations, replacing the constants with those of curves determined according to 26-4.2.3.

(b) U-shaped unreinforced bellows, as-formed or annealed.

(c) basic designs and manufacturing processes that have successfully undergone fatigue correlation testing per 26-4.2.2.

The allowable number of cycles, N_{alw}, is given by the following:

If $K_g\frac{E_o}{E_b}S_t \geq 65,000$ psi (448 MPa) :

$$N_{alw} = \left(\frac{K_o}{K_g\frac{E_o}{E_b}S_t - S_o}\right)^2$$

If S_t is expressed in psi, $K_o = 5.2 \times 10^6$ and S_o = 38,300.
If S_t is expressed in MPa, K_o = 35 850 and S_o = 264.

If $K_g\frac{E_o}{E_b}S_t < 65,000$ psi (448 MPa)

$$N_{alw} = \left(\frac{K_o}{K_g\frac{E_o}{E_b}S_t - S_o}\right)^2$$

If S_t is expressed in psi, $K_o = 6.7 \times 10^6$ and S_o = 30,600.
If S_t is expressed in MPa, K_o = 46 200 and S_o = 211.

If $K_g\frac{E_o}{E_b}S_t \leq 37,300$ psi (257.2 MPa), then $N_{alw} = 10^6$ cycles.

In the above formulas,

K_g = fatigue strength reduction factor that accounts for geometrical stress concentration factors due to thickness variations, weld geometries, surface notches, and other surface or environmental conditions. The range K_g is $1.0 \leq K_g \leq 4.0$ with its minimum value for smooth geometrical shapes and its maximum for 90 deg welded corners and fillet welds. Fatigue strength reduction factors may be determined from theoretical, experimental, or photoelastic studies. A factor has already been included in the above equations for N to account for normal effects of size, environment, and surface finish. For expansion bellows without circumferential welds and meeting all the design and examination requirements of this Appendix, a K_g of 1.0 may be used.

26-6.7 AXIAL STIFFNESS

The theoretical axial stiffness of a bellows comprising N convolutions may be evaluated by the following formula:

$$K_b = \frac{\pi}{2\left(1 - \nu_b{}^2\right)} \frac{n}{N} E_b D_m \left(\frac{t_p}{w}\right)^3 \frac{1}{C_f}$$

This formula is valid only in the elastic range.

NOTE: Outside of the elastic range, lower values can be used, based upon manufacturer's experience or representative test results.

26-7 DESIGN OF U-SHAPED REINFORCED BELLOWS

26-7.1 SCOPE

These rules cover the design of bellows having U-shaped convolutions with rings to reinforce the bellows against internal pressure. The bellows shall be attached to the shell externally.

Each half convolution consists of a sidewall and two quarter tori of the same radius (at the crest and root of the convolution), in the neutral position, so that the convolution profile presents a smooth geometrical shape as shown in Figure 26-1-1.

26-7.2 CONDITIONS OF APPLICABILITY

The following conditions of applicability apply in addition to those listed in 26-2.

(a) A variation of 10% between the crest convolution radius r_{ic} and the root convolution radius r_{ir} is permitted (see Figure 26-3 for definitions of r_{ic} and r_{ir}).

(b) The torus radius shall be such that $r_i \geq 3t$, where

$$r_i = \frac{r_{ic} + r_{ir}}{2}$$

A smaller torus radius may be used, provided that the rules of 26-4.2.2 or 26-4.2.3 are followed and the increased bending stress due to curvature is accounted for in the fatigue correlation testing.

(c) The offset angle of the sidewalls, α, in the neutral position shall be such that $-15 \leq \alpha \leq +15$ deg (see Figure 26-3).

(d) The convolution height shall be such that:

$$w \leq \frac{D_b}{3}$$

26-7.3 INTERNAL PRESSURE CAPACITY

26-7.3.1 End Tangent. The circumferential membrane stress due to pressure

$$S_1 = \frac{1}{2} \frac{\left(D_b + nt\right)^2 L_d E_b}{(nt\, L_t + A/2)(D_b + nt)E_b + A_{tc} D_c E_c} P$$

shall comply with $S_1 \leq S$.

26-7.3.2 Collar. The circumferential membrane stress due to pressure

$$S_1' = \frac{1}{2} \frac{(D_c)^2 L_d E_c}{(nt\, L_t + A/2)(D_b + nt)E_b + A_{tc} D_c E_c} P$$

shall comply with $S_1' \leq C_{wc}\, S_c$.

26-7.3.3 Bellows Convolutions.

(a) The circumferential membrane stress due to pressure

$$S_2 = \frac{H}{2A}\left(\frac{R}{R+1}\right)$$

where

R = R_1 for integral reinforcing ring members
= R_2 for reinforcing fasteners

shall comply with $S_2 \leq S$.

NOTE: In the case of reinforcing members that are made in sections and joined by fasteners in tension, this equation assumes that the structure used to retain the fastener does not bend so as to permit the reinforcing member to expand diametrically. In addition, the end reinforcing members must be restrained against the longitudinal annular pressure load of the bellows.

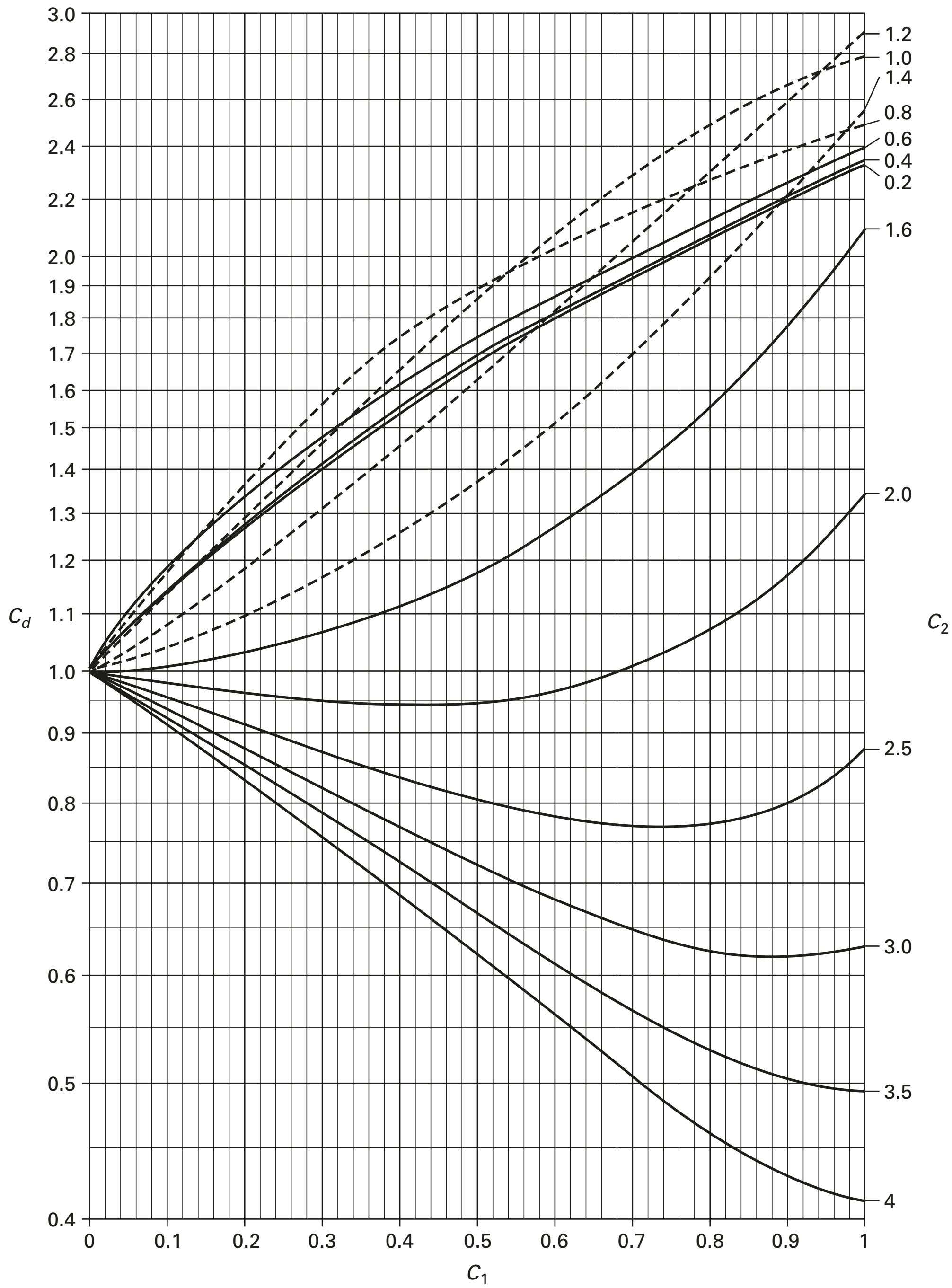
(b) The meridional membrane stress due to pressure is given by

$$S_3 = 0.85 \frac{w - 4C_r r_m}{2nt_p} P$$

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

$$S_4 = \frac{0.85}{2n}\left(\frac{w - 4C_r r_m}{t_p}\right)^2 C_p P$$

Figure 26-6
Coefficient C_d



GENERAL NOTE: Paragraph 26-15 gives polynomial approximations for these curves when $0.2 \le C_2 \le 4.0$.

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

$$S_3 + S_4 \le K_m\, S$$

where

K_m = $1.5Y_{sm}$ for as-formed bellows
= 1.5 for annealed bellows

26-7.3.4 Reinforcing Ring Member. The circumferential membrane stress due to pressure

$$S'_2 = \frac{H}{2A_r}\left(\frac{1}{R_1 + 1}\right)$$

shall comply with $S'_2 \le C_{wr}\, S_r$.

NOTE: In the case of equalizing rings, this equation provides only the simple membrane stress and does not include the bending stress caused by the eccentric fastener location. Elastic analysis and/or actual tests can determine these stresses.

26-7.3.5 Reinforcing Fastener. The membrane stress due to pressure

$$S''_2 = \frac{H}{2A_f}\left(\frac{1}{R_2 + 1}\right)$$

shall comply with $S''_2 \le S_f$.

26-7.4 INSTABILITY DUE TO INTERNAL PRESSURE

26-7.4.1 Column Instability. The allowable internal design pressure to avoid column instability is given by

$$P_{sc} = 0.3\frac{\pi K_b}{Nq}$$

The internal pressure shall not exceed P_{sc}: $P \le P_{sc}$.

26-7.4.2 In-Plane Instability. Reinforced bellows are not prone to in-plane instability.

26-7.5 EXTERNAL PRESSURE STRENGTH

26-7.5.1 External Pressure Capacity. The rules of 26-6.3 relative to unreinforced bellows shall be applied taking P as the absolute value of the external pressure.

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

26-7.5.2 Instability Due to External Pressure. The circumferential instability of a reinforced bellows shall be calculated in the same manner as for unreinforced bellows. See 26-6.5.2.

26-7.6 FATIGUE EVALUATION

26-7.6.1 Calculation of Stresses Due to the Total Equivalent Axial Displacement Range of Δq of Each Convolution.

(a) Meridional membrane stress:

$$S_5 = \frac{1}{2}\frac{E_b\, t_p^2}{\left(w - 4C_r r_m\right)^3 C_f}\Delta q$$

(b) Meridional bending stress:

$$S_6 = \frac{5}{3}\frac{E_b t_p}{\left(w - 4C_r r_m\right)^2 C_d}\Delta q$$

26-7.6.2 Calculation of Total Stress Range.

$$S_t = 0.7\left[S_3 + S_4\right] + \left[S_5 + S_6\right]$$

26-7.6.3 Calculation of Allowable Number of Cycles.

26-7.6.3.1 General. (19)

(a) The specified number of cycles N_{spe} shall be stated as consideration of the anticipated number of cycles expected to occur during the operating life of the bellows. The allowable number of cycles, N_{alw}, as derived in this subclause, shall be at least equal to N_{spe}: $N_{alw} \ge N_{spe}$.

The allowable number of cycles given by the following formulas includes a reasonable safety factor (2.6 on cycles and 1.25 on stresses) and represents the maximum number of cycles for the operating condition considered. Therefore, an additional safety factor should not be applied. An overly conservative estimate of cycles can necessitate a greater number of convolutions and result in a bellows more prone to instability.

(b) If the bellows is submitted to different cycles of pressure or displacement, such as those produced by startup or shutdown, their cumulative damage shall be considered as in 26-4.2.1.

26-7.6.3.2 Fatigue Equation. The following equations are valid for (19)

(a) austenitic chromium-nickel stainless steels, UNS N066XX and UNS N04400, for metal temperatures not exceeding 800°F (425°C). For other materials, the allowable number of cycles, N_{alw}, shall be calculated using the following equations, replacing the constants with those of curves determined according to 26-4.2.3.

(b) U-shaped reinforced bellows, as-formed or annealed.

(c) basic designs and manufacturing processes that have successfully undergone fatigue correlation testing per 26-4.2.2.

The allowable number of cycles, N_{alw}, is given by the following:

$$\text{If} \quad K_g \frac{E_o}{E_b} S_t \geq 82,200 \text{ psi} \left(567 \text{ MPa}\right):$$

$$N_{alw} = \left(\frac{K_o}{K_g \frac{E_o}{E_b} S_t - S_o} \right)^2$$

If S_t is expressed in psi, K_o = 6.6 × 10^6 and S_o = 48,500. If S_t is expressed in MPa, K_o = 45 505 and S_o = 334.

$$\text{If} \quad K_g \frac{E_o}{E_b} S_t < 82,200 \text{ psi} \left(567 \text{ MPa}\right):$$

$$N_{alw} = \left(\frac{K_o}{K_g \frac{E_o}{E_b} S_t - S_o} \right)^2$$

If S_t is expressed in psi, K_o = 8.5 × 10^6 and S_o = 38,800. If S_t is expressed in MPa, K_o = 58 605.4 and S_o = 267.5.

If $K_g \frac{E_o}{E_b} S_t$ ≤47,300 psi (326.1 MPa), then N_{alw} = 10^6 cycles.

In the above equations,

K_g = fatigue strength reduction factor that accounts for geometrical stress concentration factors due to thickness variations, weld geometries, surface notches, and other surface or environmental conditions. The range K_g is 1.0 ≤ K_g ≤ 4.0 with its minimum value for smooth geometrical shapes and its maximum for 90 deg welded corners and fillet welds. Fatigue strength reduction factors may be determined from theoretical, experimental, or photoelastic studies. A factor has already been included in the above equations for N to account for normal effects of size, environment, and surface finish. For expansion bellows without circumferential welds and meeting all the design and examination requirements of this Appendix, a K_g of 1.0 may be used.

26-7.7 AXIAL STIFFNESS

The theoretical axial stiffness of a bellows comprising N convolutions may be evaluated by the following formula:

$$K_b = \frac{\pi}{2\left(1 - \nu_b^2\right)} \frac{n}{N} E_b D_m \left(\frac{t_p}{w - 4C_r r_m} \right)^3 \frac{1}{C_f}$$

This formula is valid only in the elastic range.

NOTE: Outside of the elastic range lower values can be used, based upon manufacturer's experience or representative test results.

26-8 DESIGN OF TOROIDAL BELLOWS

26-8.1 SCOPE

These rules cover the design of bellows having toroidal convolutions. The bellows can be attached to the shell either externally or internally. Each convolution consists of a torus of radius r as shown in Figure 26-1-1.

26-8.2 CONDITIONS OF APPLICABILITY

The following conditions of applicability apply in addition to those listed in 26-2:

(a) The type of attachment to the shell (external or internal) shall be the same on both sides.

(b) Distance L_g shall be less than 0.75r in the maximum extended position.

(c) For internally attached bellows, the length of the shell on each side of the bellows having thickness t_s shall be at least equal to L_{sm} = $1.8\sqrt{D_s t_s}$.

26-8.3 INTERNAL PRESSURE CAPACITY

26-8.3.1 End Tangent. For externally attached bellows, the circumferential membrane stress due to pressure

$$S_1 = \frac{1}{2} \frac{\left(D_b + nt\right)^2 L_d E_b}{D_c E_c A_{tc}} P$$

shall comply with $S_1 \leq S$.

26-8.3.2 Tangent Collar or Shell.

(a) For externally attached bellows, the circumferential membrane stress in the collar due to pressure

$$S_1' = \frac{1}{2} \frac{D_c L_d}{A_{tc}} P$$

shall comply with $S_1' \leq C_{wc} S_c$.

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

$$S_1''' = \frac{(D_s + t_s)(L_s + 0.5L_g + nt)}{2A_{ts}} P$$

shall comply with $S_1''' \leq C_{ws} S_s$.

26-8.3.3 Bellows Convolutions.

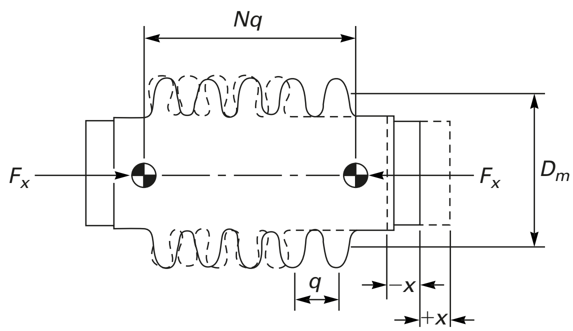
(a) The circumferential membrane stress due to pressure

$$S_2 = \frac{r}{2nt_p} P$$

shall comply with $S_2 \leq S$.

(b) The meridional membrane stress due to pressure

Figure 26-7
Bellows Subjected to an Axial Displacement *x*



$$S_3 = \frac{r}{nt_p}\left(\frac{D_m - r}{D_m - 2r}\right)P$$

shall comply with $S_3 \le S$.

26-8.3.4 Reinforcing Collars. The circumferential membrane stress due to pressure

$$S'_2 = \frac{D_r(L_{rt} + L_g + 2nt)}{2A_{rt}}P$$

if $L_{rt} \le 2\sqrt{D_r t_r}/3$

$$S'_2 = \frac{D_r(L_r + 0.5L_g + nt)}{2A_r}P$$

if $L_{rt} > 2\sqrt{D_r t_r}/3$

shall comply with $S'_2 \le C_{wr}S_r$.

26-8.4 INSTABILITY DUE TO INTERNAL PRESSURE

26-8.4.1 Column Instability. The allowable internal design pressure to avoid column instability is given by

$$P_{sc} = \frac{0.15\pi\, K_b}{Nr}$$

The internal pressure shall not exceed P_{sc}: $P \le P_{sc}$.

26-8.4.2 In-Plane Instability. Toroidal bellows are not subject to in-plane instability.

26-8.5 EXTERNAL PRESSURE STRENGTH

26-8.5.1 External Pressure Capacity. Toroidal bellows designed per the rules of this Division are suitable for external design pressures up to 15 psi (103 kPa) or full vacuum. For external design pressures greater than 15 psi (103 kPa), see U-2(g).

26-8.5.2 Instability Due to External Pressure. Instability due to external pressure is not covered by the present rules.

26-8.6 FATIGUE EVALUATION

26-8.6.1 Calculation of Stress Due to the Total Equivalent Axial Displacement Range Δq of Each Convolution.

(a) Meridional membrane stress:

$$S_5 = \frac{E_b t_p^2 B_1}{34.3r^3}\Delta q$$

(b) Meridional bending stress:

$$S_6 = \frac{E_b t_p B_2}{5.72r^2}\Delta q$$

26-8.6.2 Calculation of Total Stress Range.

$$S_t = 3\,S_3 + S_5 + S_6$$

26-8.6.3 Calculation of Allowable Number of Cycles.

26-8.6.3.1 General. (19)

(a) The specified number of cycles N_{spe} shall be stated as consideration of the anticipated number of cycles expected to occur during the operating life of the bellows. The allowable number of cycles, N_{alw}, as derived in this subclause, shall be at least equal to N_{spe}: $N_{alw} \ge N_{spe}$.

The allowable number of cycles given by the following formulas includes a reasonable safety factor (2.6 on cycles and 1.25 on stresses) and represents the maximum number of cycles for the operating condition considered. Therefore, an additional safety factor should not be applied. An overly conservative estimate of cycles can necessitate a greater number of convolutions and result in a bellows more prone to instability.

(b) If the bellows is submitted to different cycles of pressure or displacement, such as those produced by startup or shutdown, their cumulative damage shall be considered as in 26-4.2.1.

(19) **26-8.6.3.2 Fatigue Equation.** The following equations are valid for:

(a) austenitic chromium-nickel stainless steels, UNS N066XX and UNS N04400, for metal temperatures not exceeding 800°F (425°C). For other materials, the allowable number of cycles, N_{alw}, shall be calculated using the following equations, replacing the constants with those of curves determined according to 26-4.2.3.

(b) toroidal reinforced bellows, as-formed or annealed.

(c) basic designs and manufacturing processes that have successfully undergone fatigue correlation testing per 26-4.2.2.

The allowable number of cycles N_{alw} is given by the following:

$$\text{If} \quad K_g \frac{E_o}{E_b} S_t \geq 65{,}000 \text{ psi} (448 \text{ MPa}) :$$

$$N_{alw} = \left(\frac{K_o}{K_g \frac{E_o}{E_b} S_t - S_o} \right)^2$$

If S_t is expressed in psi, $K_o = 5.2 \times 10^6$ and $S_o = 38{,}300$. If S_t is expressed in MPa, $K_o = 35\ 850$ and $S_o = 264$.

$$\text{If} \quad K_g \frac{E_o}{E_b} S_t < 65{,}000 \text{ psi} (448 \text{ MPa}) :$$

$$N_{alw} = \left(\frac{K_o}{K_g \frac{E_o}{E_b} S_t - S_o} \right)^2$$

If S_t is expressed in psi, $K_o = 6.7 \times 10^6$ and $S_o = 30{,}600$. If S_t is expressed in MPa, $K_o = 46\ 200$ and $S_o = 211$.

If $K_g \frac{E_o}{E_b} S_t \leq 37{,}300$ psi (257.2 MPa), then $N_{alw} = 10^6$ cycles.

In the above formulas,

K_g = fatigue strength reduction factor that accounts for geometrical stress concentration factors due to thickness variations, weld geometries, surface notches, and other surface or environmental conditions. The range K_g is $1.0 \leq K_g \leq 4.0$ with its minimum value for smooth geometrical shapes and its maximum for 90 deg welded corners and fillet welds. Fatigue strength reduction factors may be determined from theoretical, experimental, or photoelastic studies. A factor has already been included in the above equations for N to account for normal effects of size, environment, and surface finish. For expansion bellows without circumferential welds and meeting all the design and examination requirements of this Appendix, a K_g of 1.0 may be used.

26-8.7 AXIAL STIFFNESS

The theoretical axial stiffness of a bellows comprising N convolutions may be evaluated by the following formula:

$$K_b = \frac{1}{12\left(1 - \nu_b^2\right)} \frac{n}{N} E_b D_m \left(\frac{t_p}{r} \right)^3 B_3$$

This formula is valid only in the elastic range.

NOTE: Outside of the elastic range lower values can be used, based upon manufacturer's experience or representative test results.

26-9 BELLOWS SUBJECTED TO AXIAL, LATERAL, OR ANGULAR DISPLACEMENTS

26-9.1 GENERAL

The purpose of this subclause is to determine the equivalent axial displacement of an expansion bellows subjected at its ends to:

(a) an axial displacement from the neutral position: x in extension ($x > 0$), or in compression ($x < 0$)

(b) a lateral deflection from the neutral position: y ($y > 0$)

(c) an angular rotation from the neutral position: θ ($\theta > 0$)

26-9.2 AXIAL DISPLACEMENT

When the ends of the bellows are subjected to an axial displacement x (see Figure 26-7), the equivalent axial displacement per convolution is given by

$$\Delta q_x = \frac{1}{N} x$$

where

x = positive for extension ($x > 0$)
= negative for compression ($x < 0$)

Values of x in extension and compression may be different.

The corresponding axial force F_x applied to the ends of the bellows is given by

$$F_x = K_b x$$

26-9.3 LATERAL DEFLECTION

When the ends of the bellows are subjected to a lateral deflection y (see Figure 26-8), the equivalent axial displacement per convolution is given by

Table 26-8
Tabular Values for Coefficients B_1, B_2, B_3

C_3	B_1	B_2	B_3
0	1.0	1.0	1.0
1	1.1	1.0	1.1
2	1.4	1.0	1.3
3	2.0	1.0	1.5
4	2.8	1.0	1.9
5	3.6	1.0	2.3
6	4.6	1.1	2.8
7	5.7	1.2	3.3
8	6.8	1.4	3.8
9	8.0	1.5	4.4
10	9.2	1.6	4.9
11	10.6	1.7	5.4
12	12.0	1.8	5.9
13	13.2	2.0	6.4
14	14.7	2.1	6.9
15	16.0	2.2	7.4
16	17.4	2.3	7.9
17	18.9	2.4	8.5
18	20.3	2.6	9.0
19	21.9	2.7	9.5
20	23.3	2.8	10.0

GENERAL NOTE: Equations for B_1, B_2, and B_3 are shown below.

$$B_1 = \frac{1.00404 + 0.028725C_3 + 0.18961C_3^2 - 0.00058626C_3^3}{1 + 0.14069C_3 - 0.0052319C_3^2 + 0.00029867C_3^3 - 6.2088(10)^{-6}C_3^4}$$

$$\begin{aligned} B_2 &= 1.0 \text{ for } C_3 \le 5 \\ &= \frac{0.049198 - 0.77774C_3 - 0.13013C_3^2 + 0.080371C_3^3}{1 - 2.81257C_3 + 0.63815C_3^2 + 0.0006405C_3^3} \text{ for } C_3 > 5 \end{aligned}$$

$$B_3 = \frac{0.99916 - 0.091665C_3 + 0.040635C_3^2 - 0.0038483C_3^3 + 0.00013392C_3^4}{1 - 0.1527C_3 + 0.013446C_3^2 - 0.00062724C_3^3 + 1.4374(10)^{-5}C_3^4}$$

where

$$C_3 = \frac{6.61r^2}{D_m t_p}$$

$$\Delta q_y = \frac{3D_m}{N(Nq + x)}y$$

where y shall be taken positive.

The corresponding lateral force F_y applied to the ends of the bellows is given by

$$F_y = \frac{3K_b D_m^2}{2(Nq + x)^2}y$$

The corresponding moment M_y applied to the ends of the bellows is given by

$$M_y = \frac{3K_b D_m^2}{4(Nq + x)}y$$

26-9.4 ANGULAR ROTATION

When the ends of the bellows are subjected to an angular rotation θ (see Figure 26-9), the equivalent axial displacement per convolution is given by

$$\Delta q_\theta = \frac{D_m}{2N}\theta$$

where θ, expressed in radians, shall be taken positive.

The corresponding moment M_θ applied to the ends of the bellows is given by

$$M_\theta = \frac{K_b D_m^2}{8}\theta$$

26-9.5 TOTAL EQUIVALENT AXIAL DISPLACEMENT PER CONVOLUTION (19)

Axial displacement leads to uniform deformation of the convolutions. Lateral deflection and angular rotation lead to nonuniform deformation of the convolutions with one side extended and the other side compressed as shown in Figures 26-8 and 26-9. The total equivalent axial displacements per convolution, on the extended side and the compressed side, are given by

$$\Delta q_e = \Delta q_x + \Delta q_y + \Delta q_\theta \quad \text{(extended side)}$$

$$\Delta q_c = \Delta q_x - \Delta q_y - \Delta q_\theta \quad \text{(compressed side)}$$

NOTE: In case of axial displacement only, $\Delta q_e = \Delta q_c = \Delta q_x$

26-9.6 TOTAL EQUIVALENT AXIAL DISPLACEMENT RANGE PER CONVOLUTION (19)

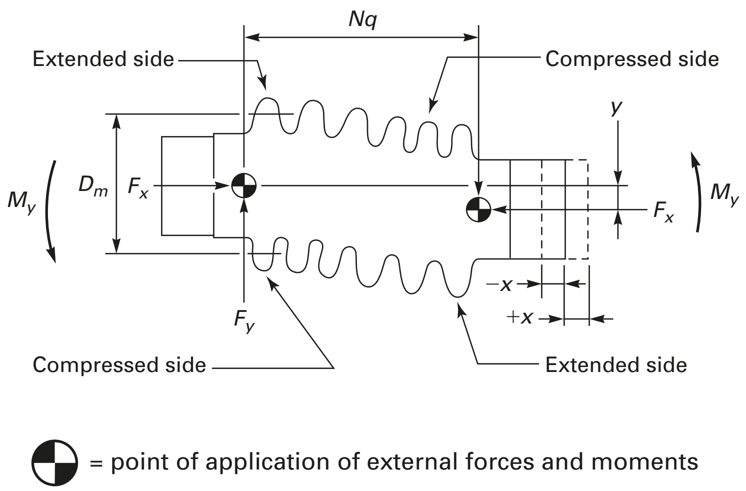
26-9.6.1 Bellows Installed Without Cold Spring. This subclause applies when the bellows is submitted to displacements (see Figure 26-10):

(a) from the neutral position ($x_0 = 0$, $y_0 = 0$, $\theta_0 = 0$)

(b) to the operating position (x_1, y_1, θ_1)

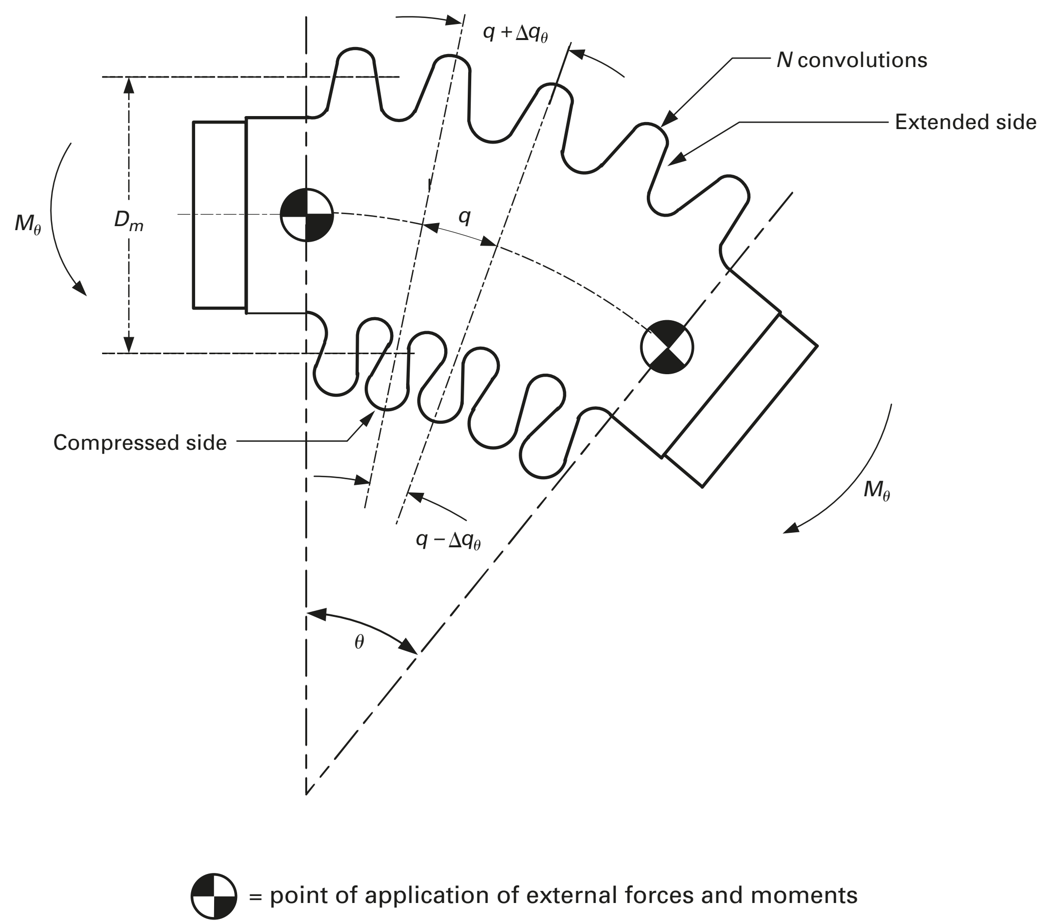
(19)

Figure 26-8
Bellows Subjected to a Lateral Deflection y



(19)

Figure 26-9
Bellows Subjected to an Angular Rotation θ



The total equivalent axial displacements per convolution, on the extended side and the compressed side, are given by

$$\Delta q_{e,1} = \Delta q_{x,1} + \Delta q_{y,1} + \Delta q_{\theta,1} \quad \text{(extended side)}$$

$$\Delta q_{c,1} = \Delta q_{x,1} - \Delta q_{y,1} - \Delta q_{\theta,1} \quad \text{(compressed side)}$$

If $x > 0$: first formula controls.
If $x < 0$: second formula controls.
The total equivalent axial displacement range is given by

$$\Delta q = \max\left[\left| \Delta q_{e,1} \right|, \left| \Delta q_{c,1} \right| \right]$$

NOTE: In case of axial displacement only, $\Delta q = |\Delta q_{x,1}|$.

26-9.6.2 Bellows Installed With Cold Spring. This subclause applies when the bellows is submitted to displacements (see Figure 26-11):

(a) from an initial position (x_0, y_0, θ_0), which is not the neutral position

$$\Delta q_{e,0} = \Delta q_{x,0} + \Delta q_{y,0} + \Delta q_{\theta,0} \quad \text{(extended side)}$$

$$\Delta q_{c,0} = \Delta q_{x,0} - \Delta q_{y,0} - \Delta q_{\theta,0} \quad \text{(compressed side)}$$

(b) to the operating position (x_1, y_1, θ_1)

$$\Delta q_{e,1} = \Delta q_{x,1} + \Delta q_{y,1} + \Delta q_{\theta,1} \quad \text{(extended side)}$$

$$\Delta q_{c,1} = \Delta q_{x,1} - \Delta q_{y,1} - \Delta q_{\theta,1} \quad \text{(compressed side)}$$

The total equivalent axial displacement range is given by

$$\Delta q = \max\left[\left| \Delta q_{e,1} - \Delta q_{c,0} \right|, \left| \Delta q_{c,1} - \Delta q_{e,0} \right| \right]$$

Alternatively, if the neutral position for lateral deflection and angular rotation is not passed between the initial position and the operating position, the total equivalent axial displacement range may be written as

$$\Delta q = \max\left[\left| \Delta q_{e,1} - \Delta q_{e,0} \right|, \left| \Delta q_{c,1} - \Delta q_{c,0} \right| \right]$$

NOTE: In case of axial displacement only, $\Delta q = |\Delta q_{x,1} - \Delta q_{x,0}|$.

26-9.6.3 Bellows Operating Between Two Operating Positions. This subclause applies when the bellows is submitted to displacements (see Figure 26-12):

(a) from the operating position 1 (x_1, y_1, θ_1)

$$\Delta q_{e,1} = \Delta q_{x,1} + \Delta q_{y,1} + \Delta q_{\theta,1} \quad \text{(extended side)}$$

$$\Delta q_{c,1} = \Delta q_{x,1} - \Delta q_{y,1} - \Delta q_{\theta,1} \quad \text{(compressed side)}$$

(b) to the operating position 2 (x_2, y_2, θ_2)

$$\Delta q_{e,2} = \Delta q_{x,2} + \Delta q_{y,2} + \Delta q_{\theta,2} \quad \text{(extended side)}$$

$$\Delta q_{c,2} = \Delta q_{x,2} - \Delta q_{y,2} - \Delta q_{\theta,2} \quad \text{(compressed side)}$$

The total equivalent axial displacement range is given by

$$\Delta q = \max\left[\left| \Delta q_{e,2} - \Delta q_{c,1} \right|, \left| \Delta q_{c,2} - \Delta q_{e,1} \right| \right]$$

Alternatively, if the neutral position for lateral deflection and angular rotation is not passed between operating positions 1 and 2, the total equivalent axial displacement range may be written as

$$\Delta q = \max\left[\left| \Delta q_{e,2} - \Delta q_{e,1} \right|, \left| \Delta q_{c,2} - \Delta q_{c,1} \right| \right]$$

NOTE: In case of axial displacement only, $\Delta q = |\Delta q_{x,2} - \Delta q_{x,1}|$.

An initial cold spring [initial position (0)] has no effect on the results.

26-10 FABRICATION

(a) Longitudinal weld seams shall be butt-type full penetration welds; Type (1) of Table UW-12.

(b) Circumferential welds attaching the bellows to the shell or weld end elements shall be full penetration groove welds or full fillet welds as shown in Figure 26-13.

(c) Other than the attachment welds, no circumferential welds are permitted in the fabrication of bellows convolutions.

(d) U-shaped unreinforced and reinforced bellows shall be manufactured to the tolerances listed in Table 26-10-1.

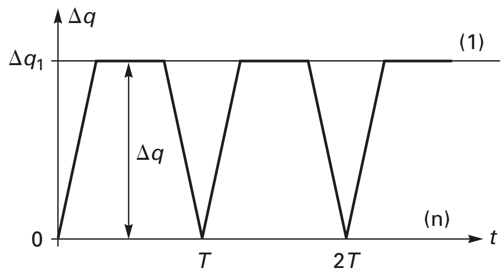
(e) Toroidal bellows shall be manufactured to the tolerances shown in Figure 26-14.

26-11 EXAMINATION

(a) Expansion joint flexible elements shall be visually examined and found free of unacceptable surface conditions, such as notches, crevices, material buildup or upsetting, and weld spatter, which may serve as points of local stress concentration. Suspect surface areas shall be further examined by the liquid penetrant method.

(19)

Figure 26-10
Cyclic Displacements



Legend:
(1) = operating position Δq_1
(n) = neutral position

(b) Bellows butt-type welds shall be examined 100% on both sides by the liquid penetrant method before forming. This examination shall be repeated after forming to the maximum extent possible considering the physical and visual access to the weld surfaces after forming.

(c) The circumferential attachment welds between the bellows and the weld ends shall be examined 100% by the liquid penetrant method.

(d) Liquid penetrant examinations shall be in accordance with Mandatory Appendix 8, except that linear indications shall be considered relevant if the dimension exceeds $0.25t_m$, but not less than 0.010 in. (0.25 mm), where t_m is the minimum bellows wall thickness before forming.

26-12 PRESSURE TEST REQUIREMENTS

26-12.1 DESIGN REQUIREMENTS

The designer shall consider the possibility of instability of the bellows due to internal pressure if the test pressure exceeds the value determined using the following applicable equation. In such a case, the designer shall redesign the bellows to satisfy the test condition.

(a) for unreinforced bellows

$$P_{t,s} = 1.5\,\mathrm{MIN}\left[\left(P_{sc}\right),\left(P_{si}\right)\right]$$

(b) for reinforced and toroidal bellows

$$P_{t,s} = 1.5\left(P_{sc}\right)$$

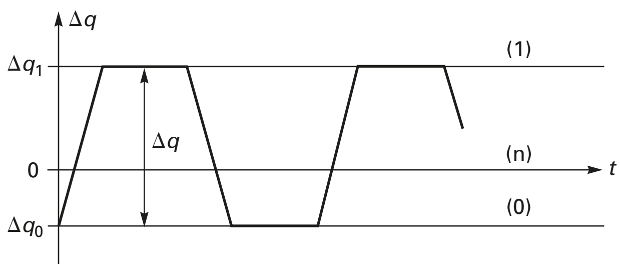
26-12.2 TEST REQUIREMENTS

(a) The completed expansion joint shall be pressure tested in accordance with UG-99 or UG-100. The pressure testing may be performed as a part of the final vessel pressure test, provided the joint is accessible for inspection during pressure testing.

Table 26-10-1
U-Shaped Unreinforced and Reinforced Bellows Manufacturing Tolerances

Bellows Dimension, in. (mm)	Manufacturing Tolerance, in. (mm)
Convolution pitch, *q*	
≤0.5 (≤12.7)	±0.063 (±1.6)
>0.5 to 1.0 (>12.7 to 25.4)	±0.125 (±3.2)
>1.0 to 1.5 (>25.4 to 38.1)	±0.188 (±4.7)
>1.5 to 2.0 (>38.1 to 50.8)	±0.250 (±6.4)
>2.0 (>50.8)	±0.313 (±7.9)
Convolution height, *w*	
≤0.5 (≤12.7)	±0.031 (±0.8)
>0.5 to 1.0 (>12.7 to 25.4)	±0.063 (±1.6)
>1.0 to 1.5 (>25.4 to 38.1)	±0.094 (±2.4)
>1.5 to 2.0 (>38.1 to 50.8)	±0.125 (±3.2)
>2.0 to 2.5 (>50.8 to 63.5)	±0.156 (±4.0)
>2.5 to 3.0 (>63.5 to 76.2)	±0.188 (±4.7)
>3.0 to 3.5 (>76.2 to 88.9)	±0.219 (±5.6)
>3.5 to 4.0 (>88.9 to 101.6)	±0.250 (±6.4)
>4.0 (>101.6)	±0.281 (±7.1)
Convolution inside diameter, D_b	
≤8.625 (≤219)	±0.063 (±1.6)
>8.625 to 24.0 (>219 to 610)	±0.125 (±3.2)
>24.0 to 48.0 (>610 to 1 219)	±0.188 (±4.7)
>48.0 to 60.0 (>1 219 to 1 524)	±0.250 (±6.4)
>60.0 (>1 524)	±0.313 (±7.9)

Figure 26-11
Cyclic Displacements



Legend:
(0) = initial position Δq_0
(1) = operating position Δq_1
(n) = neutral position

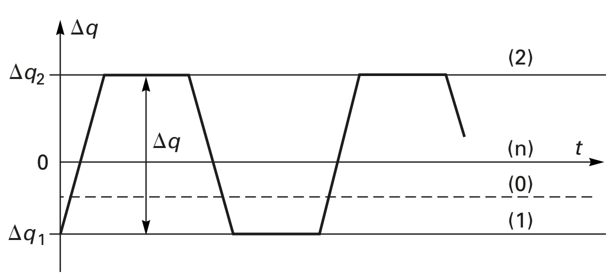
(b) Expansion joint restraining elements [see 26-4.1(c)] shall also be pressure tested in accordance with UG-99 or UG-100 as a part of the initial expansion joint pressure test or as a part of the final vessel pressure test after installation of the joint.

(c) In addition to inspecting the expansion joint for leaks and structural integrity during the pressure test, expansion joints shall be inspected before, during, and after the pressure test for visible permanent distortion.

26-13 MARKING AND REPORTS

The expansion joint Manufacturer, whether the vessel Manufacturer or a parts Manufacturer, shall have a valid ASME Code U Certificate of Authorization and shall complete the appropriate Data Report in accordance with UG-120.

Figure 26-12
Cyclic Displacements



Legend:
(0) = initial position 0
(n) = neutral position
(1) = operating position 1
(2) = operating position 2

(a) The Manufacturer responsible for the expansion joint design shall include the following additional data and statements on the appropriate Data Report:

(1) spring rate

(2) axial movement (+ and −), associated design life in cycles, and associated loading condition, if applicable

(3) that the expansion joint has been constructed to the rules of this Appendix

(b) A parts Manufacturer shall identify the vessel for which the expansion joint is intended on the Partial Data Report.

(c) Markings shall not be stamped on the flexible elements of the expansion joint.

26-14 EXAMPLES

See UG-16(f).

26-15 POLYNOMIAL APPROXIMATION FOR COEFFICIENTS C_p, C_f, C_d

26-15.1 COEFFICIENT C_p

$$C_p = \alpha_0 + \alpha_1 C_1 + \alpha_2 C_1^2 + \alpha_3 C_1^3 + \alpha_4 C_1^4 + \alpha_5 C_1^5$$

Coefficients α_i are given by Table 26-15.1a if $C_1 \le 0.3$ or Table 26-15.1b if $C_1 > 0.3$.

26-15.2 COEFFICIENT C_f

$$C_f = \beta_0 + \beta_1 C_1 + \beta_2 C_1^2 + \beta_3 C_1^3 + \beta_4 C_1^4 + \beta_5 C_1^5$$

Coefficients β_i are given by Table 26-15.2.

26-15.3 COEFFICIENT C_d

$$C_d = \gamma_0 + \gamma_1 C_1 + \gamma_2 C_1^2 + \gamma_3 C_1^3 + \gamma_4 C_1^4 + \gamma_5 C_1^5$$

Coefficients γ_i are given by Table 26-15.3.

Figure 26-13
Some Typical Expansion Bellows to Weld End Details

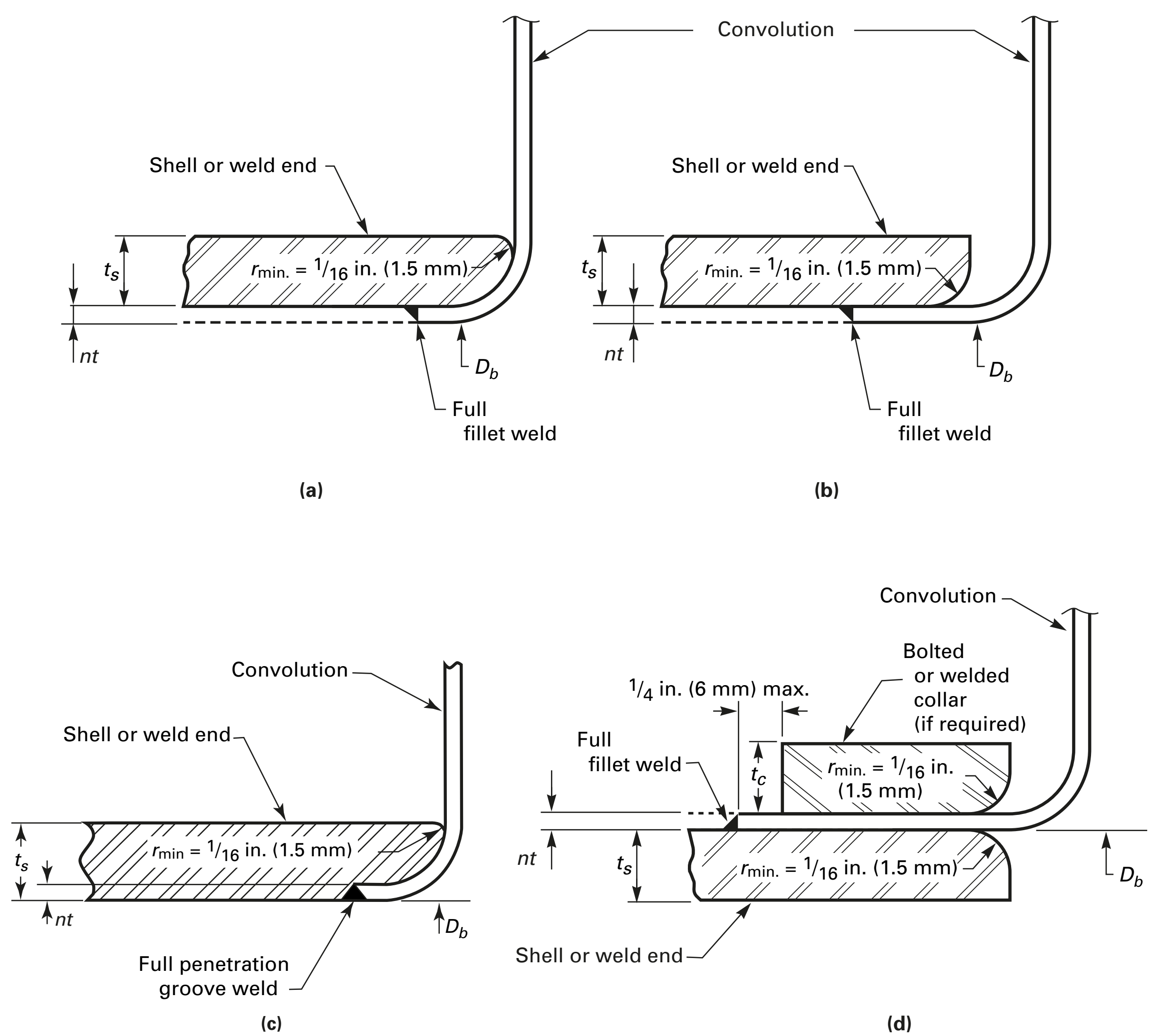
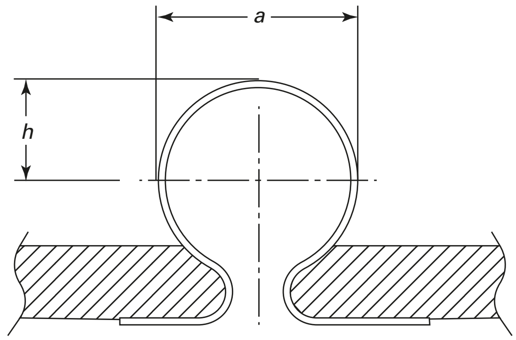


Figure 26-14
Toroidal Bellows Manufacturing Tolerances



$$0.8 \le \frac{a}{2h} \le 1.2$$

Table 26-15.1a
Polynomial Coefficients α_i for the Determination of C_p When $C_1 \le 0.3$

C_2	α_0	α_1	α_2	α_3	α_4	α_5
0.2	1.001	−0.448	−1.244	1.932	−0.398	−0.291
0.4	0.999	−0.735	0.106	−0.585	1.787	−1.022
0.6	0.961	−1.146	3.023	−7.488	8.824	−3.634
0.8	0.955	−2.708	7.279	14.212	−104.242	133.333
1	0.95	−2.524	10.402	−93.848	423.636	−613.333
1.2	0.95	−2.296	1.63	16.03	−113.939	240
1.4	0.95	−2.477	7.823	−49.394	141.212	−106.667
1.6	0.95	−2.027	−5.264	48.303	−139.394	160
2	0.95	−2.073	−3.622	29.136	−49.394	13.333
2.5	0.95	−2.073	−3.622	29.136	−49.394	13.333
3	0.95	−2.073	−3.622	29.136	−49.394	13.333
3.5	0.95	−2.073	−3.622	29.136	−49.394	13.333
4	0.95	−2.073	−3.622	29.136	−49.394	13.333

Table 26-15.1b
Polynomial Coefficients α_i for the Determination of C_p When $C_1 > 0.3$

C_2	α_0	α_1	α_2	α_3	α_4	α_5
0.2	1.001	−0.448	−1.244	1.932	−0.398	−0.291
0.4	0.999	−0.735	0.106	−0.585	1.787	−1.022
0.6	0.961	−1.146	3.023	−7.488	8.824	−3.634
0.8	0.622	1.685	−9.347	18.447	−15.991	5.119
1	0.201	2.317	−5.956	7.594	−4.945	1.299
1.2	0.598	−0.99	3.741	−6.453	5.107	−1.527
1.4	0.473	−0.029	−0.015	−0.03	0.016	0.016
1.6	0.477	−0.146	−0.018	0.037	0.097	−0.067
2	0.935	−3.613	9.456	−13.228	9.355	−2.613
2.5	1.575	−8.646	24.368	−35.239	25.313	−7.157
3	1.464	−7.098	17.875	−23.778	15.953	−4.245
3.5	1.495	−6.904	16.024	−19.6	12.069	−2.944
4	2.037	−11.037	28.276	−37.655	25.213	−6.716

Table 26-15.2
Polynomial Coefficients β_i for the Determination of C_f

C_2	β_0	β_1	β_2	β_3	β_4	β_5
0.2	1.006	2.375	−3.977	8.297	−8.394	3.194
0.4	1.007	1.82	−1.818	2.981	−2.43	0.87
0.6	1.003	1.993	−5.055	12.896	−14.429	5.897
0.8	1.003	1.338	−1.717	1.908	0.02	−0.55
1	0.997	0.621	−0.907	2.429	−2.901	1.361
1.2	1	0.112	−1.41	3.483	−3.044	1.013
1.4	1	−0.285	−1.309	3.662	−3.467	1.191
1.6	1.001	−0.494	−1.879	4.959	−4.569	1.543
2	1.002	−1.061	−0.715	3.103	−3.016	0.99
2.5	1	−1.31	−0.829	4.116	− 4.36	1.55
3	0.999	−1.521	−0.039	2.121	−2.215	0.77
3.5	0.998	−1.896	1.839	−2.047	1.852	−0.664
4	1	−2.007	1.62	−0.538	−0.261	0.249

Table 26-15.3
Polynomial Coefficients γ_i for the Determination of C_d

C_2	γ_0	γ_1	γ_2	γ_3	γ_4	γ_5
0.2	1	1.151	1.685	−4.414	4.564	−1.645
0.4	0.999	1.31	0.909	−2.407	2.273	−0.706
0.6	1.003	2.189	−3.192	5.928	−5.576	2.07
0.8	1.005	1.263	5.184	−13.929	13.828	−4.83
1	1.001	0.953	3.924	−8.773	10.44	−4.749
1.2	1.002	0.602	2.11	−3.625	5.166	−2.312
1.4	0.998	0.309	1.135	−1.04	1.296	−0.087
1.6	0.999	0.12	0.351	−0.178	0.942	−0.115
2	1	−0.133	−0.46	1.596	−1.521	0.877
2.5	1	−0.323	−1.118	3.73	−4.453	2.055
3	1	−0.545	−0.42	1.457	−1.561	0.71
3.5	1	−0.704	−0.179	0.946	−1.038	0.474
4	1.001	−0.955	0.577	−0.462	0.181	0.08

FORM 26-1 SPECIFICATION SHEET FOR ASME SECTION VIII, DIVISION 1 MANDATORY APPENDIX 26 BELLOWS EXPANSION JOINTS (19)

Date ______ / ______ / ______ Applicable ASME Code Edition ____________

1. Item Number ____________ Vessel Manufacturer ________________________
2. Drawing/Tag/Serial/Job Number ____________ Vessel Owner ________________________
3. Quantity ________ Installation Location ________________________
4. Size ______ O.D. ______ I.D. in. Expansion Joint Overall Length ________ in.

---

5. Internal Pressure: Design ______ psig
6. External Pressure: Design ______ psig
7. Vessel Manufacturer Hydrotest Pressure: Internal ______ psig External ______ psig

---

8. Temperature: Design ______ °F Operating ____ °F Upset _____ °F
9. Vessel Rating: MAWP ______ psig MDMT ______ °F Installed Position: Horiz. ___ Vert. ___

---

10. Design Movements [Note (1)]:
    Axial Compression (−) ______ in. Axial Extension (+) ______ in. Lateral ______ in. Angular ______ deg
11. Specified Number of Cycles ______________
12. Design Torsion: Moment ______________ in.-lb or Twist Angle ________________________ deg

---

13. Shell Material ______________________ Bellows Material ______________________
14. Shell Thickness ______ in. Shell Corrosion Allowance: Internal ______ in. External ______ in.
15. Shell Radiography: None / Spot / Full
16. End Preparation: Square Cut ___ Outside Bevel ___ Inside Bevel ___ Double Bevel ___ (Describe in Line 24 if special)
17. Heat Exchanger Tube Length Between Inner Tubesheet Faces _____ in.

(07/19)

FORM 26-1 SPECIFICATION SHEET FOR ASME SECTION VIII, DIVISION 1 MANDATORY APPENDIX 26 BELLOWS EXPANSION JOINTS (Cont'd)

18. Maximum Bellows Spring Rate: N Y - ____________________ lb/in.
19. Internal Liner: N Y - Material ____________________
20. Drain Holes in Liner: N Y - Quantity/Size ____________________
21. Liner Flush With Shell I.D.: N Y - Telescoping Liners? N ____ Y ____
22. External Cover: N Y - Material ____________________
23. Preproduction Approvals Required: N Y - Drawings / Bellows Calculations / Weld Procedures
24. Additional Requirements (e.g., bellows preset, ultrasonic inspection):

NOTE:
(1) For multiple movements, Design movements (line 10) can be replaced by operating movements and described in line 24. For each one of them axial compression or axial extension, lateral deflection and angular rotation at each extremity of cycle, together with the specified number of cycles, should be indicated. When known, the order of occurrence of the movements should also be indicated.

(07/19)

FORM 26-1M SPECIFICATION SHEET FOR ASME SECTION VIII, DIVISION 1 MANDATORY APPENDIX 26 BELLOWS EXPANSION JOINTS (19)

Date _____ / _____ / _____ Applicable ASME Code Edition __________

1. Item Number ________ Vessel Manufacturer ____________________
2. Drawing/Tag/Serial/Job Number ________ Vessel Owner ____________________
3. Quantity ______ Installation Location ____________________
4. Size _____ O.D. _____ I.D. mm Expansion Joint Overall Length _______ mm

5. Internal Pressure: Design _____ MPa
6. External Pressure: Design _____ MPa
7. Vessel Manufacturer Hydrotest Pressure: Internal _____ MPa External _____ MPa

8. Temperature: Design _____ °C Operating ____ °C Upset ____ °C
9. Vessel Rating: MAWP _____ MPa MDMT _____ °C Installed Position: Horiz. __ Vert. __

10. Design Movements [Note (1)]:
    Axial Compression (−)_____mm Axial Extension (+)_____ mm Lateral_____ mm Angular_____deg
11. Specified Number of Cycles __________
12. Design Torsion: Moment __________ N•mm or Twist Angle__________________ deg

13. Shell Material ____________________ Bellows Material ____________________
14. Shell Thickness _____ mm Shell Corrosion Allowance: Internal_____ mm External_____ mm
15. Shell Radiography: None / Spot / Full
16. End Preparation: Square Cut __ Outside Bevel __ Inside Bevel __ Double Bevel __ (Describe in Line 24 if special)
17. Heat Exchanger Tube Length Between Inner Tubesheet Faces _____ mm

(07/19)

FORM 26-1M SPECIFICATION SHEET FOR ASME SECTION VIII, DIVISION 1 MANDATORY APPENDIX 26 BELLOWS EXPANSION JOINTS (Cont'd)

18. Maximum Bellows Spring Rate: N Y - ____________________ N/ mm
19. Internal Liner: N Y - Material ____________________
20. Drain Holes in Liner: N Y - Quantity/Size ____________________
21. Liner Flush With Shell I.D.: N Y - Telescoping Liner? N____ Y____
22. External Cover: N Y - Material ____________________
23. Preproduction Approvals Required: N Y - Drawings / Bellows Calculations / Weld Procedures
24. Additional Requirements (e.g., bellows preset, ultrasonic inspection):

NOTE:
(1) For multiple movements, Design movements (line 10) can be replaced by operating movements and described in line 24. For each one of them axial compression or axial extension, lateral deflection and angular rotation at each extremity of cycle, together with the specified number of cycles, should be indicated. When known, the order of occurrence of the movements should also be indicated.

(07/19)

NONMANDATORY APPENDIX A BASIS FOR ESTABLISHING ALLOWABLE LOADS FOR TUBE-TO-TUBESHEET JOINTS

(19) A-1 GENERAL

(a) This Appendix provides a basis for establishing allowable tube-to-tubesheet joint loads, except for the following:

(1) Tube-to-tubesheet joints having full strength welds as defined in accordance with UW-20.2(a) shall be designed in accordance with UW-20.4 and do not require shear load testing.

(2) Tube-to-tubesheet joints having partial strength welds as defined in accordance with UW-20.2(b) and designed in accordance with UW-18(d) or UW-20.5 do not require shear load testing.

(b) The rules of this Appendix are not intended to apply to U-tube construction.

(c) Tubes used in the construction of heat exchangers or similar apparatus may be considered to act as stays which support or contribute to the strength of the tubesheets in which they are engaged. Tube-to-tubesheet joints shall be capable of transferring the applied tube loads. The design of tube-to-tubesheet joints depends on the type of joint, degree of examination, and shear load tests, if performed. Some acceptable geometries and combinations of brazed, welded, and mechanical joints are described in Table A-2. Some acceptable types of welded joints are illustrated in Figure A-2.

(1) Geometries, including variations in tube pitch, fastening methods, and combinations of fastening methods, not described or shown, may be used provided qualification tests have been conducted and applied in compliance with the procedures set forth in A-3 and A-4.

(2) Materials for welded or brazed tube-to-tubesheet joints that do not meet the requirements of UW-5 or UB-5, but in all other respects meet the requirements of Section VIII, Division 1, may be used if qualification tests of the tube-to-tubesheet joint have been conducted and applied in compliance with the procedures set forth in A-3 and A-4.

(d) Some combinations of tube and tubesheet materials, when welded, result in welded joints having lower ductility than required in the material specifications. Appropriate tube-to-tubesheet joint geometry, welding method, and/or heat treatment shall be used with these materials to minimize this effect.

(e) In the selection of joint type, consideration shall be given to the mean metal temperature of the joint at operating temperatures (see 3-2) and differential thermal expansion of the tube and tubesheet which may affect the joint integrity. The following provisions apply for establishing maximum operating temperature for tube-to-tubesheet joints:

(1) Tube-to-tubesheet joints where the maximum allowable axial load is controlled by the weld shall be limited to the maximum temperature for which there are allowable stresses for the tube or tubesheet material in Section II, Part D, Subpart 1, Table 1A or Table 1B.

Tube-to-tubesheet joints in this category are any of the following:

(-a) those complying with (a)(1) or (a)(2) with or without expansion

(-b) those welded and expanded joints, such as joint types f, g, and h, where the maximum allowable axial load is determined in accordance with A-2 and is controlled by the weld

(-c) those welded-only joints, such as joint types a, b, b-1, and e, where the maximum allowable load is determined in accordance with A-2

(2) Tube-to-tubesheet joints made by brazing, such as joint types c and d, shall be limited to temperatures in conformance with the requirements of Part UB.

(3) Tube-to-tubesheet joints where the maximum allowable axial load is determined in accordance with A-2 considering friction only, such as joint types i, j, and k, or is controlled by friction in welded and expanded joints, such as joint types f, g, and h, shall be limited to temperatures as determined by the following:

(-a) The operating temperature of the tube-to-tubesheet joint shall be within the tube and tubesheet time-independent properties of Section II, Part D, Subpart 1, Table 1A or Table 1B.

(-b) The maximum operating temperature is based on the interface pressure that exists between the tube and tubesheet. The maximum operating temperature is limited such that the interface pressure due to expanding the tube at joint fabrication plus the interface pressure due to differential thermal expansion, (P_o + P_T), does not exceed 58% of the smaller of the tube or tubesheet yield strength listed in Section II, Part D, Subpart 1, Table Y-1 at the operating temperature. If the tube or tubesheet yield strength is not listed in Table Y-1, the operating temperature limit shall be determined as described in (-d) below. The interface pressure due to

expanding the tube at fabrication or the interface pressure due to differential thermal expansion may be determined analytically or experimentally.

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

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

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

(19) A-2 MAXIMUM AXIAL LOADINGS

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

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

$$L_{\max} = A_t S_a f_r \tag{1}$$

For joint types f, g, h,

$$L_{\max} = \mathrm{MIN}\left(A_t S_a f_{re},\ A_t S_a\right) \tag{2}$$

For joint types i, j, k,

$$L_{\max} = \mathrm{MIN}\left(A_t S_a f_e f_r f_y f_T,\ A_t S_a\right) \tag{3}$$

where

A_t = tube cross-sectional area
= $\pi(d_o - t)t$

d_i = nominal tube inside diameter

d_o = nominal tube outside diameter

E = modulus of elasticity for tubesheet material at T

E_t = modulus of elasticity for tube material at T

f_e = factor for the length of the expanded portion of the tube. An expanded joint is a joint between tube and tubesheet produced by applying expanding force inside the portion of the tube to be engaged in the tubesheet. Expanding force shall be set to values necessary to effect sufficient residual interface pressure between the tube and hole for joint strength.
= MIN[(l/d_o), (1.0)] for expanded tube joints without enhancements
= 1.0 for expanded tube joints with enhancements

f_r = tube joint efficiency, which is set equal to the value of f_r (test) or f_r (no test)

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

f_r (no test) = tube joint efficiency taken from Table A-2 for tube joints not qualified by test

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

f_T = factor to account for the increase or decrease of tube joint strength due to radial differential thermal expansion at the tube-to-tubesheet joint
= $(P_o + P_T)/P_o$. Acceptable values of f_T may range from 0 to greater than 1. When the f_T value is negative, it shall be set to 0.

f_y = factor for differences in the mechanical properties of tubesheet and tube materials
= MIN[$(S_y/S_{y,t})$, (1.0)] for expanded joints. When f_y is less than 0.60, qualification tests in accordance with A-3 and A-4 are required.

k = 1.0 for loads due to pressure-induced axial forces
= 1.0 for loads due to thermally induced or pressure plus thermally induced axial forces on welded-only joints where the thickness through the weld throat is less than the nominal tube wall thickness t
= 2.0 for loads due to thermally induced or pressure plus thermally induced axial forces on all other tube-to-tubesheet joints

ℓ = length of the expanded portion of the tube

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

P_e = tube expanding pressure. The following equation may be used:

$$P_e = S_{y,t}\frac{t + r_o\left(\dfrac{S_y}{S_{y,t}}\right)}{t + r_o}\left(1.945 - 1.384\frac{d_i}{d_o}\right)$$

P_o = interface pressure between the tube and tubesheet that remains after expanding the tube at fabrication. This pressure may be established analytically or experimentally, but shall consider the effect of change in material strength at operating temperature. The following equation may be used:

$$P_o = P_e\left[1 - \left(\frac{d_i}{d_o}\right)^2\right] - \frac{2}{\sqrt{3}}S_{y,t}\left[\ln\frac{d_o}{d_i}\right]$$

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

$$P_T = \frac{\frac{R_m}{d_o}E_t\left[\alpha_t d_o\left(T - T_a\right) - \alpha d_o\left(T - T_a\right)\right]}{\left(\frac{d_o^2}{t} - R_m\right) + R_m\left(2.9\frac{E_t}{E} - 0.3\right)}$$

R_m = mean tube radius
= $(r_o - t/2)$

r_o = tube outside radius

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

S_a = allowable stress for tube material
= kS

S_y = yield strength for tubesheet material at T from Section II, Part D, Subpart 1, Table Y-1. When a yield strength value is not listed in Table Y-1, one may be obtained by using the procedure in UG-28(c)(2), Step 3.

$S_{y,t}$ = yield strength for tube material at T from Section II, Part D, Subpart 1, Table Y-1. When a yield strength value is not listed in Table Y-1, one may be obtained by using the procedure in UG-28(c)(2), Step 3.

T = tubesheet design temperature

t = nominal tube wall thickness

T_a = ambient temperature

α = mean coefficient of thermal expansion of tubesheet material at T

α_t = mean coefficient of thermal expansion of tube material at T

A-3 SHEAR LOAD TEST

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

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

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

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

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

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

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

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

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

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

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

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

(19)

Table A-2
Efficiencies f_r

Type Joint	Description [Note (1)]	Notes	f_r (Test) [Note (2)]	f_r (No Test)
a	Welded only, $a \geq 1.4t$	(3)	1.00	0.80
b	Welded only, $t \leq a < 1.4t$	(3)	0.70	0.55
b-1	Welded only, $a < t$	(4)	0.70	...
c	Brazed, examined	(5)	1.00	0.80
d	Brazed, not fully examined	(6)	0.50	0.40
e	Welded, $a \geq 1.4t$, and expanded	(3)	1.00	0.80
f	Welded, $a < 1.4t$, and expanded, enhanced with two or more grooves	(3) (7) (8) (9) (10)	0.95	0.75
g	Welded, $a < 1.4t$, and expanded, enhanced with single groove	(3) (7) (8) (9) (10)	0.85	0.65
h	Welded, $a < 1.4t$, and expanded, not enhanced	(3) (7) (8)	0.70	0.50
i	Expanded, enhanced with two or more grooves	(7) (8) (9) (10)	0.90	0.70
j	Expanded, enhanced with single groove	(7) (8) (9) (10)	0.80	0.65
k	Expanded, not enhanced	(7) (8)	0.60	0.50

GENERAL NOTE: The joint efficiencies listed in this Table apply only to allowable loads and do not indicate the degree of joint leak tightness.

NOTES:
(1) For joint types involving more than one fastening method, the sequence used in the joint description does not necessarily indicate the order in which the operations are performed.
(2) The use of the f_r (test) factor requires qualification in accordance with A-3 and A-4.
(3) The value of f_r (no test) applies only to material combinations as provided for under Section IX. For material combinations not provided for under Section IX, f_r shall be determined by test in accordance with A-3 and A-4.
(4) For f_r (no test), refer to UW-20.2(b).
(5) A value of 1.00 for f_r (test) or 0.80 for f_r (no test) can be applied only to joints in which visual examination assures that the brazing filler metal has penetrated the entire joint [see UB-14(a)] and the depth of penetration is not less than three times the nominal thickness of the tube wall.
(6) A value of 0.50 for f_r (test) or 0.40 for f_r (no test) shall be used for joints in which visual examination will not provide proof that the brazing filler metal has penetrated the entire joint [see UB-14(b)].
(7) When $d_o/(d_o - 2t)$ is less than 1.05 or greater than 1.410, f_r shall be determined by test in accordance with A-3 and A-4.
(8) When the nominal pitch (center-to-center distance of adjacent tube holes) is less than $d_o + 2t$, f_r shall be determined by test in accordance with A-3 and A-4.
(9) The Manufacturer may use other means to enhance the strength of expanded joints, provided, however, that the joints are tested in accordance with A-3 and A-4.
(10) For explosive and hydraulic expansion, grooves shall be a minimum of $1.1[(d_o - t)t]^{0.5}$ wide. For explosively or hydraulically expanded joints with single grooves meeting this requirement, f_r for joint type f may be used in lieu of that for joint type g, and f_r for joint type i may be used in lieu of that for joint type j, as applicable.

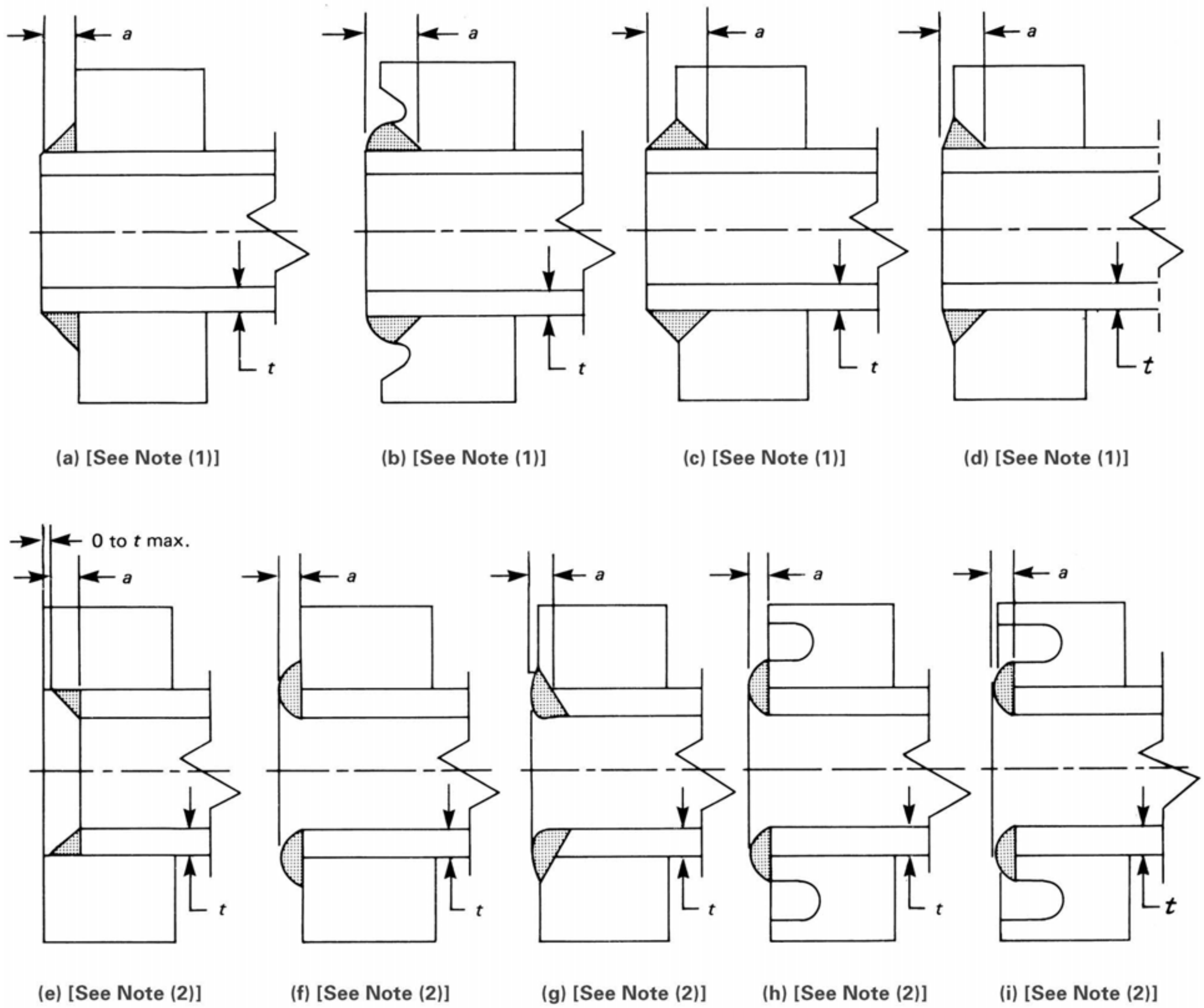
value of f_r (test) so determined is less than the value for f_r (test) given in Table A-2, retesting shall be performed in accordance with (k) below, or a new three specimen test shall be performed using a new joint configuration or fabrication procedure. All previous test data shall be rejected. To use a value of f_r (test) greater than the value given in Table A-2, a nine-specimen test shall be performed in accordance with (k) below.

(k) For joint types not listed in Table A-2, to increase the value of f_r (test) for joint types listed in Table A-2, or to retest joint types listed in Table A-2, the tests to determine f_r (test) shall conform to the following:

(1) A minimum of nine specimens from a single tube shall be tested. Additional tests of specimens from the same tube are permitted, provided all test data are used in the determination of f_r (test). If a change in the joint design or its manufacturing procedure is necessary to meet the desired characteristics, complete testing of the modified joint shall be performed.

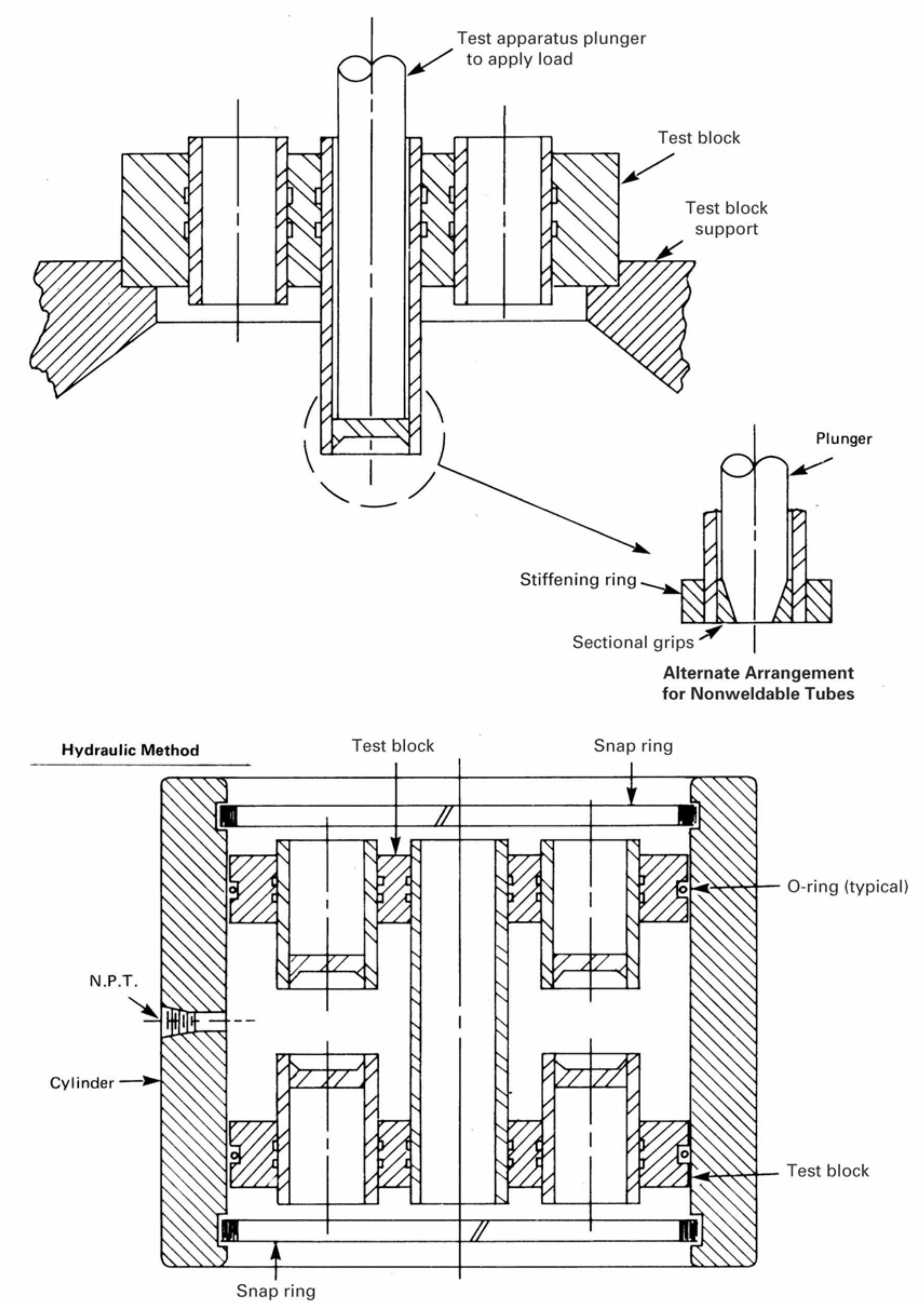
(2) In determining the value of f_r (test), the mean value of L (test) shall be determined and the standard deviation, sigma, about the mean shall be calculated. The value of f_r (test) shall be calculated using the value of L (test) corresponding to −2sigma, using the applicable equation in A-4. In no case shall f_r (test) exceed 1.0.

Figure A-2
Some Acceptable Types of Tube-to-Tubesheet Welds



NOTES:
(1) Sketches (a) through (d) show some acceptable weld geometries where a is not less than $1.4t$.
(2) Sketches (e) through (i) show some acceptable weld geometries where a is less than $1.4t$.

Figure A-3
Typical Test Fixtures for Expanded or Welded Tube-to-Tubesheet Joints



A-4 ACCEPTANCE STANDARDS FOR f_r DETERMINED BY TEST

(a) The value of f_r (test) shall be calculated as follows: For joint types a, b, b-1, c, d, e,

$$f_r(\text{test}) = \frac{L(\text{test})}{A_t S_T} \tag{4}$$

For joint types f, g, h, i, j, k,

$$f_r(\text{test}) = \frac{L(\text{test})}{A_t S_T f_e f_y} \tag{5}$$

where

L (test) = axial load at which failure of the test specimens occurs [refer to A-3(j) or A-3(k), as applicable]

S_T = tensile strength for tube material from material test report

A_t, f_e, f_y, and f_r (test) are as defined in A-2.

(b) The value of f_r (test) shall be used for f_r in the equation for L_{max}.

A-5 ACCEPTANCE STANDARDS FOR PROPOSED OPERATING TEMPERATURES DETERMINED BY TEST

The proposed operating conditions shall be acceptable if both of the following conditions are satisfied:

$$L_1(\text{test}) \geq A_t f_e f_y S_T \left(S_u / S_{ua}\right) \tag{6}$$

$$L_2(\text{test}) \geq A_t f_e f_y S_T \tag{7}$$

where

L_1 (test) = lowest axial load at which failure occurs at operating temperature

L_2 (test) = lowest axial load at which failure of heat soaked specimen tested at room temperature occurs

S_u = tensile strength for tube material at operating temperature taken from Section II, Part D, Subpart 1, Table U

S_{ua} = tensile strength for tube material at room temperature taken from Section II, Part D, Subpart 1, Table U

A_t, f_e, and f_y are as defined in A-2. S_T is as defined in A-4.

NONMANDATORY APPENDIX HH
TUBE EXPANDING PROCEDURES AND QUALIFICATION

HH-1 GENERAL

This Appendix establishes requirements for procedure specifications for expanded tube-to-tubesheet joints

(a) designed using the test joint efficiencies, f_r (test), listed in Table A-2 of Nonmandatory Appendix A;

(b) designed using the no-test joint efficiencies, f_r (no test), listed in Table A-2 of Nonmandatory Appendix A; and

(c) used in tubesheets designed in accordance with the rules of Part UHX when the effective tube hole diameter depends upon the expanded depth of the tube ($\rho > 0$).

Leak tightness of expanded joints is not a consideration in Part UHX and Nonmandatory Appendix A, and is therefore not considered in Nonmandatory Appendix HH.

HH-2 SCOPE

The rules in this Appendix apply to preparation and qualification of tube expanding procedures for the types of expanding processes permitted in this Appendix.

HH-3 TERMS AND DEFINITIONS

Some of the more common terms relating to tube expanding are as follows:

explosive expanding: uniform pressure expanding in which the force of an explosion is applied to the length of tube to be expanded.

groove: an annular machined depression in a tube hole.

hybrid expanding: hydroexpanding or explosive expanding to a percent wall reduction that ensures maintenance of tube-hole contact, followed by roller expanding to the final percent wall reduction.

hydroexpanding: uniform pressure expanding in which hydraulic pressure is applied to the length of tube to be expanded.

near contact kinetic expanding: see *explosive expanding*.

parallel tube roller: tube rolling tool in which the taper angle of the mandrel and the taper angle of the hardened pins are approximately equal and opposite, thereby causing the pins to bear uniformly on the tube surface.

percent wall reduction: reduction in tube wall thickness due to expanding, expressed as a percent of the measured thickness of the tube.

progressive rolling: step rolling in which the first step begins at or near the front face of the tubesheet and successive steps progress toward the rear face.

prosser: see *segmental expander*.

prossering: expanding tubes with a segmental expander.

regressive rolling: step rolling in which the first step begins at or near the rear face of the tubesheet and successive steps progress toward the front face.

roller expanding: expanding by inserting a tube rolling tool into a tube aligned with a tube hole.

segmental expander: thick-walled, flanged cylinder with a tapered interior wall, cut axially into segments and held together by bands. A mandrel with a reverse taper in contact with the taper of the interior of the cylinder is thrust forward, forcing the segments outward to contact and expand the tube. The flange bears against the tube end or tubesheet face to maintain the position of the expander relative to the tube.

self-feeding rolling tool: tube rolling tool with the slots in the cage at an angle with the tool centerline such that rotating the mandrel in a clockwise direction causes the tool to feed into the tube and reversing the direction causes it to back out.

serrations: parallel, narrow grooves machined in a tube hole or on the exterior of a tube end.

step rolling: tube rolling in which successive, overlapping applications of the tube roller are applied in order to roll tubes into tubesheets thicker than approximately 2 in. (50 mm).

torque control: an electronic, hydraulic control or cam-operated reversing mechanism that causes a rolling tool driver to reverse direction when a preset level of torque is reached.

transition zone: region of an expanded joint in which the expanded part of the tube transitions to the unexpanded part.

tube end enhancement: treatment to that part of the tube O.D. to be expanded into a tubesheet hole to increase the strength of the expanded tube-to-tubesheet joint.

tube expanding: process of expanding a tube to a fully plastic state into contact with the surrounding metal of a tube hole that creates residual interface pressure between the tube and tube hole when the expanding tool is withdrawn.

tube hole enhancement: treatment to the tube hole to increase the strength of an expanded tube-to-tubesheet joint. Enhancements may be by means of grooves or serrations.

tube rolling tool: tool consisting of a slotted cylindrical cage that holds hardened pins into which a hardened tapered mandrel is thrust and rotated, to expand the tube.

two-stage expanding: explosive, hydraulic, or roller expanding in which in the first stage all the tubes are expanded into firm contact with the holes, followed by a second stage of expanding to the final specified percent wall reduction.

uniform pressure expanding: tube expanding by applying force equally on the surfaces of the length of tube to be expanded.

HH-4 TUBE EXPANDING PROCEDURE SPECIFICATION (TEPS)

A TEPS is a written document that provides the tube expander operator with instructions for making production tube-to-tubesheet joint expansions in accordance with Code requirements (see Form QEXP-1). The Manufacturer is responsible for ensuring that production tube expanding is performed in accordance with a qualified TEPS that meets the requirements of HH-7.

NOTE: The instructions for completing Form QEXP-1 are provided in Table QEXP-1. The instructions are identified by parenthesized numbers corresponding to circled numbers in the form.

The TEPS shall address, as a minimum, the specific variables, both essential and nonessential, as provided in HH-7.1 for each process to be used in production expanding.

HH-5 TUBE EXPANDING PROCEDURE QUALIFICATION

The purpose for qualifying a TEPS is to demonstrate that the expanded joint proposed for construction will be suitable for its intended application. The tube expanding procedure qualification establishes the suitability of the expanded joint, not the skill of the tube expander operator.

HH-5.1 NO TEST QUALIFICATION

Tube expanding procedures not required to be qualified by HH-5.2 may be used for expanded tube joints meeting HH-1(b) or HH-1(c) without a qualification test, provided the Manufacturer maintains records indicating that the tube joints expanded using the tube expanding procedures were successfully tested in accordance with UG-99 or UG-100.

HH-5.2 TEST QUALIFICATION

Tube expanding procedures to be used for expanded tube joints meeting HH-1(a) shall be qualified by the Manufacturer in accordance with the requirements of A-1 and A-3, and the qualification shall be documented in accordance with HH-5.3.

HH-5.3 TUBE EXPANDING PROCEDURE QUALIFICATION RECORD (TEPQR) FOR TEST JOINT EFFICIENCIES

The TEPQR documents what occurred during expanding the test specimen and the results of the testing in accordance with the requirements of A-1 and A-3 of Nonmandatory Appendix A. In addition, the TEPQR shall document the essential variables and other specific information identified in HH-7 for each process used.

HH-6 TUBE EXPANDING PERFORMANCE QUALIFICATION (TEPQ)

The purpose of performing a TEPQ is to demonstrate that the operator of the equipment is qualified to make an expanded joint of the type specified in the TEPS.

HH-6.1 NO TEST QUALIFICATION

A tube expander operator not required to be qualified by HH-6.2 is qualified to expand tube joints meeting HH-1(b) or HH-1(c), provided the Manufacturer maintains records indicating that tube joints expanded by the operator were successfully tested in accordance with UG-99 or UG-100.

HH-6.2 TEST QUALIFICATION

A tube expander operator is qualified to expand tube joints using tube expanding procedures that have been qualified in accordance with HH-5.2, provided the operator, under the direction of the Manufacturer, has prepared at least one specimen that meets the requirements of A-1 and A-3 for the applicable procedure.

HH-7 TUBE EXPANDING VARIABLES

Variables are subdivided into essential variables that apply to all expanding processes, and essential and nonessential variables that apply to each expanding process. Essential variables are those in which a change, as described in specific variables, is considered to affect the mechanical properties of the expanded joint, and shall require requalification of the TEPS. Nonessential variables are those that may be changed at the Manufacturer's discretion and are included in the TEPS for instruction purposes.

HH-7.1 ESSENTIAL VARIABLES FOR ALL EXPANDING PROCESSES

The following essential variables shall be specified for all expanding processes. The Manufacturer may define additional essential variables.

(a) method of measuring and controlling tube hole diameter

(b) limit of percentage of tube holes that deviate from the specified diameter tolerance and maximum tolerance of hole-diameter deviation

(c) limiting ratio of tube diameter to tube wall thickness

(d) minimum ratio of tubesheet thickness to tube diameter

(e) minimum ratio of drilling pitch to tube diameter

(f) details of tube and/or tube hole treatments for joint strength enhancement, including surface finish of tube holes, tube-hole and tube end serrations, and tube hole annular grooves

(g) tube-to-hole diametral clearance prior to expanding (fit)

(h) range of modulus of elasticity of tube material

(i) range of modulus of elasticity of tubesheet material

(j) range of specified minimum tube yield stresses listed in Section II

(k) maximum permissible increase of tube yield stress above the minimum yield stress specified in Section II

(l) specified minimum tubesheet yield stress listed in Section II

(m) minimum ratio of tubesheet to tube yield stress;[109] a ratio below 0.6 requires shear load testing

(n) minimum and maximum percent wall reduction[110]

(o) for welded tube joints where tubes are to be expanded after welding, the method of fixing tube position before welding, the setback from the front face of the tubesheet to onset of expanding, the treatment of weld and tube-end shrinkage before inserting the expanding mandrel, and any post-expansion heat treatment

(p) for tubes to be expanded before welding, the procedure to be used to remove all traces of lubricants and moisture from the surfaces to be welded

(q) distance from front face of tubesheet to commencement of expanding

(r) distance from rear face of tubesheet to end of expanding

(s) unrolled length between front and rear expansion

(t) lubrication and cooling of the expanding mandrel

(u) measured actual amount of expansion

(v) range of tube wall thickness

HH-7.2 ESSENTIAL VARIABLES FOR ROLLER EXPANDING

The following are essential variables for roller expanding:

(a) tool driver type (electrical, air, hydraulic), power or torque rating

(b) number and length of overlapping steps

(c) direction of rolling (progressive or regressive)

(d) speed of rotation

(e) tool type (parallel or nonparallel)

(f) cage and pin length

(g) number of pins in the cage

(h) cage slot angle or tool manufacturer's tool number

(i) frequency of verifying percent wall reduction

(j) for tubes to be expanded after welding, amount of setback before expanding mandrel insertion due to weld and tube-end shrinkage

HH-7.3 ESSENTIAL VARIABLES FOR HYDRAULIC EXPANDING

The following are essential variables for hydraulic expanding:

(a) hydraulic mandrel details or mandrel manufacturer's mandrel number(s)

(b) hydraulic expanding pressure

(c) precision of pressure control

(d) number of applications of hydraulic pressure

(e) permissible + and − deviation from specified hydraulic expanding pressure

HH-7.4 ESSENTIAL VARIABLES FOR EXPLOSIVE EXPANDING

The following are essential variables for explosive expanding:

(a) number of applications of explosive force

(b) number of tubes to be simultaneously expanded

(c) tube supports in surrounding holes

(d) post-expanding tube-end cleaning

(e) size of the explosive load

(f) buffer material

(g) outside diameter of the buffer material

(h) inside diameter of the buffer material

(i) theoretical expanded O.D. of the tube based on original cross-sectional area and expanded I.D. of the tube as compared to the tubesheet hole diameter

HH-7.5 ESSENTIAL VARIABLES FOR HYBRID EXPANDING

The essential variables for hybrid expanding are the variables listed in HH-7.4 for the initial explosive expanding or HH-7.3 for the initial hydraulic expanding and the following:

(a) the range of percent wall reduction to be achieved by the initial expanding

(b) the range of total percent wall reduction to be achieved by the initial expanding and final rolling

HH-7.6 NONESSENTIAL VARIABLES

The Manufacturer shall specify nonessential variables for each process.

FORM QEXP-1 TUBE EXPANDING PROCEDURE SPECIFICATION (TEPS)

Company Name:				By:	
Tube Expanding Procedure Specification No.		Date		Supporting TEPQR No.(s)	
Revision No.		Date			
Expanding Process(es)			Driver Type(s)		
(Rolling, Hydroexpanding, Explosive Expanding, Hybrid Expanding)			(Electric, Air, Hydraulic, Hydroexpanded, Explosive)		

JOINTS

Measurement and Control of Tube Hole		Tube Pitch	
Tube Hole Diameter and Tolerance		Maximum Tube to Hole Clearance Before Expanding	
Ratio Tube Diameter/Tube Wall Thickness		Minimum Ratio Drilling Pitch/Tube Diameter	
Maximum % Wall Reduction		Minimum % Wall Reduction	
Maximum Permissible Deviation from Specified Hole Diameter		Maximum Permissible % of Holes that Deviate	
Details of Tube End Hole Enhancement and/or Tube End Enhancement		Minimum Ratio Tubesheet Thickness/Tube Diameter	
Method of Fixing Tubes in Position		Length of Expansion	
Setback from Front Tubesheet Face Before Start of Expanding		Setback from Rear Tubesheet Face after Expanding	
Method of Removing Weld Droop		Method of Tube End and Hole Cleaning	
Other Joint Details:			

EXPANDING EQUIPMENT

Manufacturer(s), Model No.(s), Range of Tube Diameters and Thicknesses, Maximum Torque Output or Pressure.

Expanding Tool Model and Description			
Expanded Length per Application of Expanding Mandrel		No. of Applications/ Expanded Length	
Torque or Pressure Calibration System and Frequency		Explosive Charge and No.(s) of Applications	

PROPERTIES

Range of Tube Elastic Modulus		Range of Plate Elastic Modulus		
Range of Tube Yield Stress (Mill Test Report Values)	Min.		Max.	
Range of Tubesheet Yield Stress (Mill Test Report Values)	Min.		Max.	
Minimum Tubesheet Yield Stress/Tube Yield Stress Note: Values below 0.6 require shear load testing				

TUBES

Diameter Range		Thickness Range		Maximum Ratio Tube Diameter/Thickness	
Material Specifications					

TUBESHEETS

Thickness Range		Minimum Ratio of Tubesheet Thickness to Tube Diameter	
Material Specifications			

REMARKS

FORM QEXP-1 TUBE EXPANDING PROCEDURE SPECIFICATION (TEPS)

1	Company Name:	1			By:	2
2	Tube Expanding Procedure Specification No.	3	Date	4	Supporting TEPQR No.(s)	5
3	Revision No.	6	Date	7		
4	Expanding Process(es)	8	Driver Type(s)	9		

JOINTS

5	Measurement and Control of Tube Hole	10	Tube Pitch	11
6	Tube Hole Diameter and Tolerance	12	Maximum Tube to Hole Clearance Before Expanding	13
7	Ratio Tube Diameter/Tube Wall Thickness	14	Minimum Ratio Drilling Pitch/Tube Diameter	15
8	Maximum % Wall Reduction	16	Minimum % Wall Reduction	17
9	Maximum Permissible Deviation from Specified Hole Diameter	18	Maximum Permissible % of Holes that Deviate	19
10	Details of Tube End Hole Enhancement and/or Tube End Enhancement	20	Minimum Ratio Tubesheet Thickness/Tube Diameter	21
11	Method of Fixing Tubes in Position	22	Length of Expansion	23
12	Setback from Front Tubesheet Face Before Start of Expanding	24	Setback from Rear Tubesheet Face After Expanding	25
13	Method of Removing Weld Droop	26	Method of Tube End and Hole Cleaning	27
14	Other Joint Details:	28		

EXPANDING EQUIPMENT

15	Manufacturer(s), Model No.(s), Range of Tube Diameters and Thicknesses, Maximum Torque Output or Pressure. 29			
16	Expanding Tool Model and Description	30		
17	Expanded Length per Application of Expanding Mandrel	31	No. of Applications/ Expanded Length	32
18	Torque or Pressure Calibration System and Frequency	33	Explosive Charge and No.(s) of Applications	34

PROPERTIES

19	Range of Tube Elastic Modulus		35	Range of Plate Elastic Modulus	36	
20	Range of Tube Yield Stress (mill test report values)	Min.	37	Max.	38	
21	Range of Tubesheet Yield Stress (mill test report values)	Min.	39	Max.	40	
22	Minimum Tubesheet Yield Stress/Tube Yield Stress NOTE: Values below 0.6 require shear load testing.		41			

TUBES

23	Diameter Range	42	Thickness Range	43	Maximum Ratio Tube Diameter/Thickness	44
24	Material Specifications	45				

TUBESHEETS

25	Thickness Range	46	Minimum Ratio of Tubesheet Thickness to Tube Diameter	47
26	Material Specifications	48		
27	**REMARKS:**	49, 50, 51		

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Table QEXP-1
Instructions for Filling Out TEPS Form

Ref. to Circled Nos. in Form QEXP-1	Explanation of Information to Be Provided
(1)	Show Manufacturer's name and address.
(2)	Show TEPS author's names.
(3)	Show Manufacturer's TEPS number.
(4)	Show applicable date of TEPS.
(5)	Insert number of supporting Tube Expanding Procedure Qualification Record (TEPQR).
(6)	Show revision number if any.
(7)	Insert date of revision if any.
(8)	Describe expanding process as torque-controlled expanding, hydraulic expanding, or explosive expanding. If hybrid expanding is to be performed, describe sequence, e.g., "hybrid expanding (hydraulic expanding to 3% wall reduction followed by torque-controlled roller expanding to 6% to 8% total wall reduction)."
(9)	Describe as hydraulic, explosive, air-driven torque controlled, electric torque controlled, or hydraulic torque controlled drive. If hybrid expanded, describe as hydraulic or explosive expanded + torque controlled air, torque controlled electric, or torque controlled hydraulic torque controlled drive.
(10)	Describe measuring equipment, e.g., "go–no/go gage," "internal 3 point micrometer," or similar measuring device. All equipment used for measurements shall be calibrated.
(11)	Minimum centerline distance between tube holes.
(12)	Show hole size and plus/minus tolerance.
(13)	Show diametrical clearance, e.g., 0.014 in. (for minimum of 96%) and 0.022 in. (for maximum of 4%).
(14)	Minimum and maximum ratio of tube O.D. to tube wall (O.D./t) for this TEPS.
(15)	Fill in nominal ratio of drilling pitch to tube diameter.
(16)	Fill in maximum percent wall reduction to which the TEPS applies.
(17)	Fill in minimum percent wall reduction to which the TEPS applies.
(18)	Enter maximum permissible deviation of hole from specified drilling size and tolerance, e.g., 0.01 in.
(19)	Enter maximum percent of holes that may deviate by the amount shown in (18).
(20)	Describe enhancements for joint strength, e.g., "(2)$^1/_8$ in. wide × $^1/_{64}$ in. grooves set 1 in. from inlet face with $^1/_2$ in. land between."
(21)	Fill in the maximum and minimum ratios of tubesheet thickness to tube diameter.
(22)	Describe how the tube will be fixed in position before expanding, e.g., "nose roll" or "hydraulically preset."
(23)	Fill in the length of tube end to be expanded into the hole, e.g., "tubesheet thickness - $^3/_{16}$ in." If hybrid expansion is to be performed, show length of expansion for each step.
(24)	Fill in the distance from the front face of the tubesheet to the point where expanding will begin.
(25)	Fill in the distance from the rear face of the tubesheet to the point where expanding will end.
(26)	If tube is welded to front face of tubesheet, describe how any weld metal that impedes access of the expanding tool(s) will be removed.
(27)	Describe how tube ends will be cleaned before expanding, e.g., "solvent wash and clean with felt plugs."
(28)	Describe any other pertinent details, e.g., "tubes to be welded to front face of tubesheet before expanding."
(29)	Show expanding tool manufacturer, e.g., name hydraulic expanding system or model no., "range of tube diameters $^1/_2$ in. to 2 in., range of thicknesses 0.028 in. to 0.109 in., maximum hydraulic pressure 60,000 psi."

Table QEXP-1
Instructions for Filling Out TEPS Form (Cont'd)

Ref. to Circled Nos. in Form QEXP-1	Explanation of Information to Be Provided
(30)	Fill in roller expanding tool or hydraulic mandrel number. If explosive expanding, fill in drawing number that describes the charges. If hybrid expanding, show this information for Steps 1 and 2.
(31)	Describe expanded length per application, e.g., "2 in. (roller length)."
(32)	Show number of applications of expanding tool, e.g., "two applications required for roll depth." If hydraulic or explosive expanding, show length of expansion per application of hydraulic expanding pressure or explosive charge, e.g., "tubesheet thickness - $^5/_8$ in."
(33)	Describe the system used to calibrate and control the rolling torque and frequency of verification. Alternatively, describe the use of production control holes and expansions.
(34)	Describe the explosive charge and whether it will be single- or two-stage explosive expansion.
(35)	List the minimum and maximum elastic modulus of the tubes for this TEPS.
(36)	List the minimum and maximum elastic modulus of the tubesheet(s) for this TEPS.
(37)	List minimum permissible tube yield stress.
(38)	List maximum permissible tube yield stress.
(39)	List minimum permissible tubesheet yield stress.
(40)	List maximum permissible tubesheet yield stress.
(41)	Show the minimum ratio of tubesheet to tube yield stresses.
(42)	List the range of tube diameters to which this TEPS applies.
(43)	List the range of tube thicknesses to which this TEPS applies.
(44)	Show the maximum ratio of tube diameter to thickness to which this TEPS applies.
(45)	Show the tube specification number, e.g., "SA-688 TP304N."
(46)	Show the range of tubesheet thicknesses to which this TEPS applies, e.g., 1 in. to 5 in.
(47)	Show the minimum ratio of tubesheet thickness to tube diameter to which this TEPS applies.
(48)	Show the tubesheet material specification numbers, e.g., "SA-350 LF2."
(49)	Describe pertinent job-specific information.
(50)	Describe such things as bundle setup and sequence of expansion operation. Refer to drawing numbers and manufacturer's standards as appropriate.
(51)	Refer to any attachment or supplement to the TEPS form.

FORM QEXP-2 SUGGESTED FORMAT FOR TUBE-TO-TUBESHEET EXPANDING PROCEDURE QUALIFICATION RECORD FOR TEST QUALIFICATION (TEPQR)

Company name ______

Procedure Qualification Record number ______ Date ______

TEPS no. ______

Expanding process(es) ______
(Rolling, hydroexpanding, explosive expanding, hybrid expanding)

Driver types ______
(Electric, air-driven, hydraulic, other)

Expanded tube length ______
(If there is a gap in the expanded zone, record the total expanded length)

Tube pitch ______

Joints (HH-7)

Sketch of Test Array

Tubesheet Material(s)

Material spec. ______ Type or grade ______

Diameter and thickness of test specimen ______ Hole diameter and pitch arrangement ______

No. and location of joints to be tested ______

No. and description of annular grooves ______

Hole surface finish ______

Yield stress (from mill test report) ______

Other ______

Testing Apparatus ______
(Manufacturer, type, calibration date)

Rate of loading to avoid impact ______
[Maximum $^1/_2$ in. (13 mm) per minute]

Tube Material(s) ______

Material spec. ______ Type or grade ______

Diameter and thickness (min./avg.) ______

Yield stress (from mill test report) ______

Other ______

(10/06)

FORM QEXP-2 (Back)

Shear Load Test (See Appendix A, Fig. A-3)

Tube No.	Position in Array	Diameter	Thickness	Cross-Sectional Area	Test Temp.	L_1 (test)	Manner of Failure

Tube No.	Position in Array	Diameter	Thickness	Cross-Sectional Area	Ambient Temp.	L_2 (test)	Manner of Failure

Mean value of L_1 (test) __________ Mean value of L_2 (test) __________

Standard deviation __________ Standard deviation __________ Satisfactory (see Appendix A, A-5) ________

f_r (test) (see Appendix A, A-4) ____________________

Operator's name ____________________________ Clock no. __________

Manufacturer ____________________________

Date ________________ By ____________________________

Remarks:

10/06