ARTICLE NB-3000

Figures

Tables

IPEC00239096

ARTICLE NB-3000
DESIGN

NB-3100 GENERAL DESIGN

NB-3110 LOADING CRITERIA

NB-3111 Loading Conditions

The loadings that shall be taken into account in designing a component include, but are not limited to, those in (a) through (g) below:

(a) internal and external pressure;

(b) impact loads, including rapidly fluctuating pressures;

(c) weight of the component and normal contents under operating or test conditions, including additional pressure due to static and dynamic head of liquids;

(d) superimposed loads such as other components, operating equipment, insulation, corrosion resistant or erosion resistant linings, and piping;

(e) wind loads, snow loads, vibrations, and earthquake loads where specified;

(f) reactions of supporting lugs, rings, saddles, or other types of supports;

(g) temperature effects.

NB-3112 Design Loadings

The Design Loadings shall be established in accordance with NCA-2142.1 and the following subparagraphs.

NB-3112.1 Design Pressure

(a) The specified internal and external Design Pressures to be used in this Subsection shall be established in accordance with NCA-2142.1(a). They shall be used in the computations made to show compliance with the stress intensity limits of NB-3221, NB-3227.1, NB-3227.2, NB-3227.4, NB-3228.1, NB-3228.2, and NB-3231. The specified service pressure at the appropriate time shall be used in the computations made to show compliance with the stress intensity limits of NB-3222, NB-3228.3, and NB-3232. When the occurrence of different pressures during operation can be predicted for different zones of a component, the Design Pressure of the different zones may be based on their predicted pressures.

(b) All pressures referred to in this Article are to be taken as pounds per square inch, psi, above atmospheric pressure, unless otherwise stated.

NB-3112.2 Design Temperature

(a) The specified Design Temperature shall be established in accordance with NCA-2142.1(b). It shall be used in computations involving the Design Pressure and coincidental Design Mechanical Loads. The actual metal temperature at the point under consideration shall be used in all computations where the use of the specified service pressure is required.

(b) All temperatures referred to in this Subsection are the metal temperatures expressed in degrees Fahrenheit (°F) unless otherwise stated.

(c) Where a component is heated by tracing, induction coils, jacketing, or internal heat generation, the effect of such heating shall be incorporated in the establishment of the Design Temperature.

NB-3112.3 Design Mechanical Loads. The specified Design Mechanical Loads shall be established in accordance with NCA-2142.1(c). They shall be used in conjunction with the Design Pressure.

NB-3112.4 Design Stress Intensity Values. Design stress intensity values for materials are listed in Table I-1.0. The material shall not be used at metal and design temperatures above those for which stress intensity values are listed. The values in the Table may be interpolated for intermediate temperatures.

NB-3113 Service Conditions

Each service condition to which the components may be subjected shall be classified in accordance with NCA-2142 and Service Limits [NCA-2142.4(b)] designated in the Design Specifications in such detail as will provide a complete basis for design, construction, and inspection in accordance with this Article. The requirements of (a) and (b) below shall also apply.

(a) Level B Conditions. The estimated duration of service conditions for which Level B Limits are specified shall be included in the Design Specifications.

(b) Level C Conditions. The total number of postulated occurrences for all specified service conditions for which Level C Limits are specified shall not cause more than 25 stress cycles having an S_a value greater than that for 10^6 cycles from the applicable fatigue design curves of Figs. I-9.0.

NB-3120 SPECIAL CONSIDERATIONS

NB-3121 Corrosion

Material subject to thinning by corrosion, erosion, mechanical abrasion, or other environmental effects shall have provision made for these effects during the design or specified life of the component by a suitable increase in or addition to the thickness of the base metal over that determined by the design formulas. Material added or included for these purposes need not be of the same thickness for all areas of the component if different rates of attack are expected for the various areas. It should be noted that the tests on which the design fatigue curves (Figs. I-9.0) are based did not include tests in the presence of corrosive environments which might accelerate fatigue failure.

NB-3122 Cladding

The rules of the following subparagraphs apply to the analysis of clad components constructed of material permitted in Tables I-1.0.

NB-3122.1 Primary Stresses. No structural strength shall be attributed to the cladding in satisfying NB-3221.

NB-3122.2 Design Dimensions. The dimensions given in (a) and (b) below shall be used in the design of the component:

(a) for components subjected to internal pressure, the inside diameter shall be taken at the nominal inner face of the cladding;

(b) for components subjected to external pressure, the outside diameter shall be taken at the outer face of the base metal.

NB-3122.3 Secondary and Peak Stresses. In satisfying NB-3222.2 and NB-3222.4, the presence of the cladding shall be considered with respect to both the thermal analysis and the stress analysis. The stresses in both materials shall be limited to the values specified in NB-3222.2 and NB-3222.4. However, when the cladding is of the integrally bonded type and the nominal thickness of the cladding is 10% or less of the total thickness of the component, the presence of the cladding may be neglected.

NB-3122.4 Bearing Stresses. In satisfying NB-3227.1, the presence of cladding shall be included.

NB-3123 Welding

NB-3123.1 Dissimilar Welds. In satisfying the requirements of this Subarticle, caution should be exercised in design and construction involving dissimilar metals having different coefficients of thermal expansion in order to avoid difficulties in service.

NB-3123.2 Fillet Welded Attachments. Fillet welds conforming to Fig. NB-4427-1 may be used for attachments to components except as limited by NB-4433. Evaluation for cyclic loading shall be made in accordance with the appropriate Subarticle of NB-3000, and shall include consideration of temperature differences between the component and the attachment, and of expansion or contraction of the component produced by internal or external pressure.

NB-3124 Environmental Effects

Changes in material properties may occur due to environmental effects. In particular, fast neutron irradiation (> 1 MeV) above a certain level may result in significant increase in the brittle fracture transition temperature and deterioration in the resistance to fracture at temperatures above the transition range (upper shelf energy). Therefore, nozzles or other structural discontinuities in ferritic vessels should preferably not be placed in regions of high neutron flux.

NB-3125 Configuration

Accessibility to permit the examinations required by the Edition and Addenda of Section XI as specified in the Design Specification for the component shall be provided in the design of the component.

NB-3130 GENERAL DESIGN RULES

NB-3131 Scope

Design rules generally applicable to all components are provided in the following paragraphs. The design Subarticle for the specific component provides rules applicable to that particular component. In case of conflict between NB-3130 and the design rules for a particular component, the component design rules govern.

NB-3132 Dimensional Standards for Standard Products

Dimensions of standard products shall comply with the standards and specifications listed in Table NB-3132-1 when the standard or specification is referenced in the specific design Subarticle. However, compliance with these standards does not replace or eliminate the requirements for stress analysis when called for by the design Subarticle for a specific component.

NB-3133 Components Under External Pressure

NB-3133.1 General. Rules are given in this paragraph for determining the stresses under external pressure loading in spherical and cylindrical shells with or without stiffening rings, and tubular products consisting of pipes, tubes, and fittings. Charts for determining the stresses in shells, hemispherical heads, and tubular products are given in Appendix VII.

NB-3133.2 Nomenclature. The symbols used in this paragraph are defined as follows:

A = factor determined from Fig. VII-1100-1 in Appendix VII and used to enter the applicable material chart in Appendix VII. For the case of cylinders having D_o/T values less than 10, see NB-3133.3(b). Also, factor determined from the applicable chart in Appendix VII for the material used in a stiffening ring, corresponding to the factor B and the design metal temperature for the shell under consideration.

A_s = cross-sectional area of a stiffening ring, sq in.

B = factor determined from the applicable chart in Appendix VII for the material used in a shell or stiffening ring at the design metal temperature, psi

D_o = outside diameter of the cylindrical shell course or tube under consideration, in.

E = modulus of elasticity of material at Design Temperature, psi. For external pressure and axial compression design in accordance with this Section, the modulus of elasticity to be used shall be taken from the applicable materials chart in Appendix VII. (Interpolation may be made between lines for intermediate temperatures.) The modulus of elasticity values shown in Appendix VII for material groups may differ from those values listed in Tables I-6.0 for specific materials. Appendix VII values shall be applied only to external pressure and axial compression design.

I = available moment of inertia of the combined ring-shell section about its neutral axis, parallel to the axis of the shell, in.[4] The width of the shell which is taken as contributing to the combined moment of inertia shall not be greater than $1.10\sqrt{D_o T_n}$ and shall be taken as lying one-half on each side of the centroid of the ring. Portions of shell plates shall not be considered as contributing area to more than one stiffening ring.

I_s = required moment of inertia of the combined ring-shell section about its neutral axis parallel to the axis of the shell, in.[4]

L = total length, in., of a tube between tubesheets, or the design length of a vessel section, taken as the largest of the following:

(1) the distance between head tangent lines plus one-third of the depth of each head if there are no stiffening rings;

(2) the greatest center-to-center distance between any two adjacent stiffening rings; or

(3) the distance from the center of the first stiffening ring to the head tangent line plus one-third of the depth of the head, all measured parallel to the axis of the vessel, in.

L_s = one-half the distance, in., from the center line of the stiffening ring to the next line of support on one side, plus one-half of the center line distance to the next line of support on the other side of the stiffening ring, both measured parallel to the axis of the component. A line of support is:

(1) a stiffening ring that meets the requirements of this paragraph;

(2) a circumferential line on a head at one-third the depth of the head from the head tangent line; or

(3) circumferential connection to a jacket for a jacketed section of a cylindrical shell.

P = external design pressure, psi (gage or absolute, as required)

P_a = allowable external pressure, psi (gage or absolute, as required)

R = inside radius of spherical shell, in.

S = the lesser of 1.5 times the stress intensity at design metal temperature from Tables I-1.0 or 0.9 times the tabulated yield strength at design metal temperature from Tables I-2.0, psi

TABLE NB-3132-1
DIMENSIONAL STANDARDS

Standard	Designation
Pipe and Tubes	
Welded and Seamless Wrought Steel Pipe	ANSI B36.10-1979
Stainless Steel Pipe	ANSI B36.19-1976
Fittings, Flanges, and Gaskets	
Steel Pipe Flanges and Flanged Fittings	ANSI B16.5-1981
Factory Made Wrought Steel Buttwelding Fittings	ANSI B16.9-1978 [Note (1)]
Forged Steel Fittings, Socket-Welding and Threaded	ANSI B16.11-1973
Ring-Joint Gaskets and Grooves for Steel Pipe Flanges	ANSI B16.20-1973
Nonmetallic Gaskets for Pipe Flanges	ANSI B16.21-1978
Buttwelding Ends	ANSI B16.25-1979
Wrought Steel Buttwelding Short Radius Elbows and Returns	ANSI B16.28-1978 [Note (3)]
Refrigeration Flare Type Fittings	SAE J513 F-1981
Stainless Steel Buttwelding Fittings	MSS SP-43-1982
Steel Pipeline Flanges	MSS SP-44-1982
Factory Made Buttwelding Fittings for Class 1 Nuclear Piping Applications	MSS SP-87-1982 [Note (2)]
Large Diameter Carbon Steel Flanges	API 605-1980
Standard for Steel Pipe Flanges	AWWA C207-1978
Bolting	
Square and Hex Bolts and Screws, Including Askew Head Bolts, Hex Cap Screws and Lag Screws	ANSI B18.2.1-1981
Square and Hex Nuts	ANSI B18.2.2-1972(R1983)
Socket Cap, Shoulder, and Set Screws	ANSI B18.3-1982
Threads	
Unified Inch Screw Threads (UN and UNR Thread Form)	ANSI B1.1-1982
Pipe Threads (Except Dryseal)	ANSI B1.20.1-1983
Dryseal Pipe Threads	ANSI B1.20.3-1976(R1981)
Gaging for Dryseal Piping Threads	ANSI B1.20.5-1978
Valves	
Valves — Flanged and Buttwelding End	ANSI B16.34-1981

NOTES:
(1) Analysis per ANSI B16.9, paragraph 2.2, is acceptable only for caps and reducers.
(2) Analysis per MSS-SP-87 is acceptable only for caps and reducers.
(3) Analysis per ANSI B16.28, paragraph 2.2, is not acceptable.

T = minimum required thickness of cylindrical shell or tube, or spherical shell, in.

T_n = nominal thickness used, less corrosion allowance, of cylindrical shell or tube, in.

NB-3133.3 Cylindrical Shells and Tubular Products

(a) The minimum thickness of cylindrical shells or tubular products under external pressure having D_o/T values equal to or greater than 10 shall be determined by the procedure given in Steps 1 through 8 below.

Step 1: Assume a value for T. Determine the ratios L/D_o and D_o/T.

Step 2: Enter Fig. VII-1100-1 in Appendix VII at the value of L/D_o determined in Step 1. For values of L/D_o greater than 50, enter the chart at a value of L/D_o of 50. For values of L/D_o less than 0.05, enter the chart at a value of L/D_o of 0.05.

Step 3: Move horizontally to the line for the value of D_o/T determined in Step 1. Interpolation may be made for intermediate values of D_o/T. From this intersection move vertically downward and read the value of factor A.

Step 4: Using the value of A calculated in Step 3, enter the applicable material chart in Appendix VII for the material/temperature under consideration. Move vertically to an intersection with the material/temperature line for the design temperature. Interpolation may be made between lines for intermediate temperatures. In cases where the value at A falls to the right of the end of the material/temperature line, assume an intersection with the horizontal projection of the upper end of the material/temperature line. For values of A falling to the left of the material line, see Step 7.

Step 5: From the intersection obtained in Step 4 move horizontally to the right and read the value of B.

Step 6: Using this value of B, calculate the maximum allowable external pressure P_a using the following formula:

$$P_a = \frac{4B}{3D_o/T}$$

Step 7: For values of A falling to the left of the applicable material/temperature line, the value of P_a can be calculated using the following formula:

$$P_a = \frac{2AE}{3D_o/T}$$

Step 8: Compare P_a with P. If P_a is smaller than P, select a larger value for T and repeat the design procedure until a value for P_a is obtained that is equal to or greater than P.

(b) The minimum thickness of cylindrical shells or tubular products under external pressure having D_o/T values less than 10 shall be determined by the procedure given in Steps 1 through 4 below.

Step 1: Using the same procedure as given in (a) above, obtain the value of B. For values of D_o/T less than 4, the value of factor A can be calculated using the following formula:

$$A = \frac{1.1}{(D_o/T)^2}$$

For values of A greater than 0.10 use a value of 0.10.

Step 2: Using the value of B obtained in Step 1, calculate a value P_{a1} using the following formula:

$$P_{a1} = \left(\frac{2.167}{D_o/T} - 0.0833\right) B$$

Step 3: Calculate a value P_{a2} using the following formula:

$$P_{a2} = \frac{2S}{D_o/T}\left(1 - \frac{1}{D_o/T}\right)$$

Step 4: The smaller of the values of P_{a1} calculated in Step 2 or P_{a2} calculated in Step 3 shall be used for the maximum allowable external pressure P_a. Compare P_a with P. If P_a is smaller than P, select a larger value for T and repeat the design procedure until a value for P_a is obtained that is equal to or greater than P.

NB-3133.4 Spherical Shells. The minimum required thickness of a spherical shell under external pressure, either seamless or of built-up construction with butt joints, shall be determined by the procedure given in Steps 1 through 6 below.

Step 1: Assume a value for T and calculate the value of factor A using the following formula:

$$A = \frac{0.125}{R/T}$$

Step 2: Using the value of A calculated in Step 1, enter the applicable material chart in Appendix VII for the material under consideration. Move vertically to an intersection with the material/temperature line for the design temperature. Interpolation may be made between lines for intermediate temperatures. In cases where the value of A falls to the right of the end of the material/temperature line, assume an intersection with the horizontal projection of the upper end of the material/temperature line. For values at A falling to the left of the material/temperature line, see Step 5.

Step 3: From the intersection obtained in Step 2, move horizontally to the right and read the value of factor B.

Step 4: Using the value of B obtained in Step 3, calculate the value of the maximum allowable external pressure P_a using the following formula:

$$P_a = \frac{B}{R/T}$$

Step 5: For values of A falling to the left of the applicable material/temperature line for the Design Temperature, the value of P_a can be calculated using the following formula:

$$P_a = \frac{0.0625E}{(R/T)^2}$$

Step 6: Compare P_a obtained in Step 4 or 5 with P. If P_a is smaller than P, select a larger value for T, and repeat the design procedure until a value for P_a is obtained that is equal to or greater than P.

NB-3133.5 Stiffening Rings for Cylindrical Shells

(a) The required moment of inertia of the combined ring-shell section is given by the formula:

$$I_s = \frac{D_o^2 L_s (T + A_s/L_s)A}{10.9}$$

The available moment of inertia I for a stiffening ring shall be determined by the procedure given in Steps 1 through 6 below.

Step 1: Assuming that the shell has been designed and D_o, L_s, and T_n are known, select a member to be used for the stiffening ring and determine its area A and the value of I defined in NB-3133.2. Then calculate B by the formula:

$$B = \frac{3}{4}\left(\frac{PD_o}{T_n + A_s/L_s}\right)$$

Step 2: Enter the right-hand side of the applicable material chart in Appendix VII for the material under consideration at the value of B determined in Step 1. If different materials are used for the shell and stiffening ring, then use the material chart resulting in the larger value for factor A in Steps 4 or 5 below.

Step 3: Move horizontally to the left to the material/temperature line for the design metal temperature. For values of B falling below the left end of the material/temperature line, see Step 5.

Step 4: Move vertically to the bottom of the chart and read the value of A.

Step 5: For values of B falling below the left end of the material/temperature line for the design temperature, the value of A can be calculated using the following formula:

$$A = 2B/E$$

Step 6: If the required I_s is greater than the computed moment of inertia I for the combined-ring shell section selected in Step 1, a new section with a larger moment of inertia must be selected and a new I_s determined. If the required I_s is smaller than the computed I for the section selected in Step 1, that section should be satisfactory.

(b) Stiffening rings may be attached to either the outside or the inside of the component by continuous welding.

NB-3133.6 Cylinders Under Axial Compression. The maximum allowable compressive stress to be used in the design of cylindrical shells and tubular products subjected to loadings that produce longitudinal compressive stresses in the shell or wall shall be the lesser of the values given in (a) or (b) below:

(a) the S_m value for the applicable material at design temperature given in Table I-1.1 or Table I-1.2;

(b) the value of the factor B determined from the applicable chart contained in Appendix VII, using the following definitions for the symbols on the charts:

T = minimum required thickness of the shell or tubular product, exclusive of the corrosion allowance, in.

R = inside radius of the cylindrical shell or tubular product, in.

The value of B shall be determined from the applicable chart contained in Appendix VII as given in Steps 1 through 5 below.

Step 1: Using the selected values of T and R, calculate the value of factor A using the following formula:

$$A = \frac{0.125}{R/T}$$

Step 2: Using the value of A calculated in Step 1, enter the applicable material chart in Appendix VII for the material under consideration. Move vertically to an intersection with the material/temperature line for the design temperature. Interpolation may be made between lines for intermediate temperatures. In cases where the value at A falls to the right of the end of the material/temperature line, assume an intersection with the horizontal projection of the upper end of the material/temperature line. For values of A falling to the left of the material/ temperature line, see Step 4.

Step 3: From the intersection obtained in Step 2, move horizontally to the right and read the value of factor B. This is the maximum allowable compressive stress for the values of T and R used in Step 1.

Step 4: For values of A falling to the left of the applicable material/temperature line, the value of B shall be calculated using the following formula:

$$B = \frac{AE}{2}$$

Step 5: Compare the value of B determined in Step 3 or 4 with the computed longitudinal compressive stress in the cylindrical shell or tube, using the selected values of T and R. If the value of B is smaller than the computed compressive stress, a greater value of T must be selected and the design procedure repeated until a value of B is obtained which is greater than the compressive stress computed for the loading on the cylindrical shell or tube.

NB-3134 Leak Tightness

Where a system leak tightness greater than that required or demonstrated by a hydrostatic test is required, the leak tightness requirements for each component shall be set forth in the Design Specifications.

NB-3135 Attachments

(a) Except as in (d) and (e) below, attachments and connecting welds within the jurisdictional boundary of the component as defined in NB-1130 shall meet the stress limits of the component or NB-3200.

(b) The design of the component shall include consideration of the interaction effects and loads transmitted through the attachment to and from the pressure retaining portion of the component. Thermal stresses, stress concentrations, and restraint of the pressure retaining portion of the component shall be considered.

(c) The first welded structural attachment within $2t$ of the pressure retaining portion of the component, where t is the nominal thickness of the pressure retaining material, shall be evaluated for cyclic loading. Evaluation shall be in accordance with the appropriate Subarticle of NB-3000 and shall be made at the juncture of the attachment to the component.

(d) Beyond $2t$ the appropriate design rules of NF-3000 may be used as a substitute for the design rules of NB-3000 for portions of attachments which are in the component support load path.

(e) Nonstructural attachments shall meet the requirements of NB-4435.

NB-3137 Reinforcement for Openings

The requirements applicable to vessels and piping are contained in NB-3330 and NB-3643, respectively.

NB-3200 DESIGN BY ANALYSIS

NB-3210 DESIGN CRITERIA

NB-3211 Requirements for Acceptability

The requirements for the acceptability of a design by analysis are given in (a) through (d) below.

(a) The design shall be such that stress intensities

will not exceed the limits described in this Subarticle and in NB-3100 and tabulated in Tables I-1.0.

(b) The design details shall conform to the rules given in NB-3100 and those given in the Subarticle applicable to the specific component.

(c) For configurations where compressive stresses occur, in addition to the requirements in (a) and (b) above, the critical buckling stress shall be taken into account. For the special case of external pressure, NB-3133 applies.

(d) Protection against nonductile fracture shall be provided. An acceptable procedure for nonductile failure prevention is given in Appendix G.

NB-3212 Basis for Determining Stresses

The theory of failure, used in the rules of this Subsection for combining stresses, is the maximum shear stress theory. The maximum shear stress at a point is equal to one-half the difference between the algebraically largest and the algebraically smallest of the three principal stresses at the point.

NB-3213 Terms Relating to Stress Analysis

Terms used in this Subsection relating to stress analysis are defined in the following subparagraphs.

NB-3213.1 Stress Intensity.[1] Stress intensity is the equivalent intensity of combined stress, or, in short, the stress intensity is defined as twice the maximum shear stress. In other words, the stress intensity is the difference between the algebraically largest principal stress and the algebraically smallest principal stress at a given point. Tensile stresses are considered positive and compressive stresses are considered negative.

NB-3213.2 Gross Structural Discontinuity. Gross structural discontinuity is a geometric or material discontinuity which affects the stress or strain distribution through the entire wall thickness of the pressure retaining member. Gross discontinuity type stresses are those portions of the actual stress distributions that produce net bending and membrane force resultants when integrated through the wall thickness. Examples of gross structural discontinuities are head-to-shell and flange-to-shell junctions, nozzles (NB-3331), and junctions between shells of different diameters or thicknesses.

[1]This definition of stress intensity is not related to the definition of stress intensity applied in the field of Fracture Mechanics.

NB-3213.3 Local Structural Discontinuity. Local structural discontinuity is a geometric or material discontinuity which affects the stress or strain distribution through a fractional part of the wall thickness. The stress distribution associated with a local discontinuity causes only very localized types of deformation or strain and has no significant effect on the shell type discontinuity deformations. Examples are small fillet radii, small attachments, and partial penetration welds.

NB-3213.4 Normal Stress. Normal stress is the component of stress normal to the plane of reference. This is also referred to as direct stress. Usually the distribution of normal stress is not uniform through the thickness of a part, so this stress is considered to be made up in turn of two components, one of which is uniformly distributed and equal to the average value of stress across the thickness under consideration, and the other of which varies from this average value with the location across the thickness.

NB-3213.5 Shear Stress. Shear stress is the component of stress tangent to the plane of reference.

NB-3213.6 Membrane Stress. Membrane stress is the component of normal stress which is uniformly distributed and equal to the average value of stress across the thickness of the section under consideration.

NB-3213.7 Bending Stress. Bending stress is the variable component of normal stress described in NB-3213.4. The variation may or may not be linear across the thickness.

NB-3213.8 Primary Stress. Primary stress is any normal stress or a shear stress developed by an imposed loading which is necessary to satisfy the laws of equilibrium of external and internal forces and moments. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses which considerably exceed the yield strength will result in failure or, at least, in gross distortion. A thermal stress is not classified as a primary stress. Primary membrane stress is divided into general and local categories. A general primary membrane stress is one which is so distributed in the structure that no redistribution of load occurs as a result of yielding. Examples of primary stresses are:

(a) general membrane stress in a circular cylindrical or a spherical shell due to internal pressure or to distributed live loads;

(b) bending stress in the central portion of a flat head due to pressure.

NB-3213.9 Secondary Stress. Secondary stress is a normal stress or a shear stress developed by the constraint of adjacent material or by self-constraint of the structure. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the conditions which cause the stress to occur and failure from one application of the stress is not to be expected. Examples of secondary stresses are:

(a) general thermal stress [NB-3213.13(a)];

(b) bending stress at a gross structural discontinuity.

NB-3213.10 Local Primary Membrane Stress. Cases arise in which a membrane stress produced by pressure or other mechanical loading and associated with a primary or a discontinuity effect produces excessive distortion in the transfer of load to other portions of the structure. Conservatism requires that such a stress be classified as a local primary membrane stress even though it has some characteristics of a secondary stress. A stressed region may be considered local if the distance over which the membrane stress intensity exceeds $1.1S_m$ does not extend in the meridional direction more than $1.0\sqrt{Rt}$, where R is the minimum midsurface radius of curvature and t is the minimum thickness in the region considered. Regions of local primary stress intensity involving axisymmetric membrane stress distributions which exceed $1.1S_m$ shall not be closer in the meridional direction than $2.5\sqrt{Rt}$, where R is defined as $(R_1 + R_2)/2$ and t is defined as $(t_1 + t_2)/2$ (where t_1 and t_2 are the minimum thicknesses at each of the regions considered, and R_1 and R_2 are the minimum midsurface radii of curvature at these regions where the membrane stress intensity exceeds $1.1S_m$). Discrete regions of local primary membrane stress intensity, such as those resulting from concentrated loads acting on brackets, where the membrane stress intensity exceeds $1.1S_m$, shall be spaced so that there is no overlapping of the areas in which the membrane stress intensity exceeds $1.1S_m$.

NB-3213.11 Peak Stress. Peak stress is that increment of stress which is additive to the primary plus secondary stresses by reason of local discontinuities or local thermal stress [NB-3213.13(b)] including the effects, if any, of stress concentrations. The basic characteristic of a peak stress is that it does not cause any noticeable distortion and is objectionable only as a possible source of a fatigue crack or a brittle fracture. A stress which is not highly localized falls into this category if it is of a type which cannot cause noticeable distortion. Examples of peak stresses are:

(a) the thermal stress in the austenitic steel cladding of a carbon steel component;

(b) certain thermal stresses which may cause fatigue but not distortion;

(c) the stress at a local structural discontinuity;

(d) surface stresses produced by thermal shock.

NB-3213.12 Load Controlled Stresses. Load controlled stresses are the stresses resulting from application of a loading, such as internal pressure, inertial loads, or the effects of gravity, whose magnitude is not reduced as a result of displacement.

NB-3213.13 Thermal Stress. Thermal stress is a self-balancing stress produced by a nonuniform distribution of temperature or by differing thermal coefficients of expansion. Thermal stress is developed in a solid body whenever a volume of material is prevented from assuming the size and shape that it normally should under a change in temperature. For the purpose of establishing allowable stresses, two types of thermal stress are recognized, depending on the volume or area in which distortion takes place, as described in (a) and (b) below.

(a) General thermal stress is associated with distortion of the structure in which it occurs. If a stress of this type, neglecting stress concentrations, exceeds twice the yield strength of the material, the elastic analysis may be invalid and successive thermal cycles may produce incremental distortion. Therefore this type is classified as secondary stress in Table NB-3217-1. Examples of general thermal stresses are:

(1) stress produced by an axial temperature distribution in a cylindrical shell;

(2) stress produced by the temperature difference between a nozzle and the shell to which it is attached;

(3) the equivalent linear stress[2] produced by the radial temperature distribution in a cylindrical shell.

(b) Local thermal stress is associated with almost complete suppression of the differential expansion and thus produces no significant distortion. Such stresses shall be considered only from the fatigue standpoint and are therefore classified as local stresses in Table NB-3217-1. In evaluating local thermal stresses the procedures of NB-3228.1(c) shall be used. Examples of local thermal stresses are:

(1) the stress in a small hot spot in a vessel wall;

(2) the difference between the actual stress and the equivalent linear stress resulting from a radial temperature distribution in a cylindrical shell;

(3) the thermal stress in a cladding material which

[2]*Equivalent linear stress* is defined as the linear stress distribution which has the same net bending moment as the actual stress distribution.

has a coefficient of expansion different from that of the base metal.

NB-3213.14 Total Stress. Total stress is the sum of the primary, secondary, and peak stress contributions. Recognition of each of the individual contributions is essential to establishment of appropriate stress limitations.

NB-3213.15 Operational Cycle. Operational cycle is defined as the initiation and establishment of new conditions followed by a return to the conditions which prevailed at the beginning of the cycle. The types of operating conditions which may occur are further defined in NB-3113.

NB-3213.16 Stress Cycle. Stress cycle is a condition in which the alternating stress difference [NB-3222.4(e)] goes from an initial value through an algebraic maximum value and an algebraic minimum value and then returns to the initial value. A single operational cycle may result in one or more stress cycles. Dynamic effects shall also be considered as stress cycles.

NB-3213.17 Fatigue Strength Reduction Factor. Fatigue strength reduction factor is a stress intensification factor which accounts for the effect of a local structural discontinuity (stress concentration) on the fatigue strength. Values for some specific cases, based on experiment, are given in NB-3338 and NB-3339. In the absence of experimental data, the theoretical stress concentration factor may be used.

NB-3213.18 Free End Displacement. Free end displacement consists of the relative motions that would occur between a fixed attachment and connected piping if the two members were separated and permitted to move.

NB-3213.19 Expansion Stresses. Expansion stresses are those stresses resulting from restraint of free end displacement of the piping system.

NB-3213.20 Deformation. Deformation of a component part is an alteration of its shape or size.

NB-3213.21 Inelasticity. Inelasticity is a general characteristic of material behavior in which the material does not return to its original shape and size after removal of all applied loads. Plasticity and creep are special cases of inelasticity.

NB-3213.22 Creep. Creep is the special case of inelasticity that relates to the stress-induced, time-dependent deformation under load. Small time-dependent deformations may occur after the removal of all applied loads.

NB-3213.23 Plasticity. Plasticity is the special case of inelasticity in which the material undergoes time-independent nonrecoverable deformation.

NB-3213.24 Plastic Analysis. Plastic analysis is that method which computes the structural behavior under given loads considering the plasticity characteristics of the materials, including strain hardening and the stress redistribution occurring in the structure.

NB-3213.25 Plastic Analysis — Collapse Load. A plastic analysis may be used to determine the collapse load for a given combination of loads on a given structure. The following criterion for determination of the collapse load shall be used. A load–deflection or load–strain curve is plotted with load as the ordinate and deflection or strain as the abscissa. The angle that the linear part of the load–deflection or load–strain curve makes with the ordinate is called θ. A second straight line, hereafter called the collapse limit line, is drawn through the origin so that it makes an angle $\phi = \tan^{-1}(2 \tan \theta)$ with the ordinate. The collapse load is the load at the intersection of the load–deflection or load–strain curve and the collapse limit line. If this method is used, particular care should be given to ensure that the strains or deflections that are used are indicative of the load carrying capacity of the structure.

NB-3213.26 Plastic Instability Load. The plastic instability load for members under predominantly tensile or compressive loading is defined as that load at which unbounded plastic deformation can occur without an increase in load. At the plastic tensile instability load, the true stress in the material increases faster than strain hardening can accommodate.

NB-3213.27 Limit Analysis. Limit analysis is a special case of plastic analysis in which the material is assumed to be ideally plastic (nonstrain-hardening). In limit analysis, the equilibrium and flow characteristics at the limit state are used to calculate the collapse load. The two bounding methods which are used in limit analysis are the lower bound approach, which is associated with a statically admissible stress field, and the upper bound approach, which is associated with a kinematically admissible velocity field. For beams and frames, the term *mechanism* is commonly used in lieu of *kinematically admissible velocity* field.

NB-3213.28 Limit Analysis — Collapse Load. The methods of limit analysis are used to compute the maximum load that a structure assumed to be made of ideally plastic material can carry. At this load, which is termed the collapse load, the deformations of the structure increase without bound.

NB-3213.29 Collapse Load — Lower Bound. If, for a given load, any system of stresses can be found which everywhere satisfies equilibrium, and nowhere exceeds the material yield strength, the load is at or below the collapse load. This is the lower bound theorem of limit analysis which permits calculations of a lower bound to the collapse load.

NB-3213.30 Plastic Hinge. A plastic hinge is an idealized concept used in Limit Analysis. In a beam or a frame, a plastic hinge is formed at the point where the moment, shear, and axial force lie on the yield interaction surface. In plates and shells, a plastic hinge is formed where the generalized stresses lie on the yield surface.

NB-3213.31 Strain Limiting Load. When a limit is placed upon a strain, the load associated with the strain limit is called the strain limiting load.

NB-3213.32 Test Collapse Load. Test collapse load is the collapse load determined by tests according to the criteria given in II-1430.

NB-3213.33 Ratcheting. Ratcheting is a progressive incremental inelastic deformation or strain which can occur in a component that is subjected to variations of mechanical stress, thermal stress, or both.

NB-3213.34 Shakedown. Shakedown of a structure occurs if, after a few cycles of load application, ratcheting ceases. The subsequent structural response is elastic, or elastic–plastic, and progressive incremental inelastic deformation is absent. Elastic shakedown is the case in which the subsequent response is elastic.

NB-3214 Stress Analysis

A detailed stress analysis of all major structural components shall be prepared in sufficient detail to show that each of the stress limitations of NB-3220 and NB-3230 is satisfied when the component is subjected to the loadings of NB-3110. As an aid to the evaluation of these stresses, formulas and methods for the solution of certain recurring problems have been placed in Appendix A.

NB-3215 Derivation of Stress Intensities

One requirement for the acceptability of a design (NB-3210) is that the calculated stress intensities shall not exceed specified allowable limits. These limits differ depending on the stress category (primary, secondary, etc.) from which the stress intensity is derived. This paragraph describes the procedure for the calculation of the stress intensities which are subject to the specified limits. The steps in the procedure are stipulated in (a) through (e) below.

(a) At the point on the component which is being investigated, choose an orthogonal set of coordinates, such as tangential, longitudinal, and radial, and designate them by the subscripts t, l, and r. The stress components in these directions are then designated σ_t, σ_l, and σ_r for direct stresses and τ_{lt}, τ_{lr}, and τ_{rt} for shear stresses.

(b) Calculate the stress components for each type of loading to which the part will be subjected, and assign each set of stress values to one or a group of the following categories:[3]

(1) general primary membrane stress P_m (NB-3213.8);

(2) local primary membrane stress P_L (NB-3213.10);

(3) primary bending stress P_b (NB-3213.7 and NB-3213.8);

(4) expansion stress P_e (NB-3213.20);

(5) secondary stress Q (NB-3213.9);

(6) peak stress F (NB-3213.11). NB-3217 provides guidance for this step.

(c) For each category, calculate the algebraic sum of the σ_t values which result from the different types of loadings and similarly for the other five stress components. Certain combinations of the categories must also be considered.

(d) Translate the stress components for the t, l, and r directions into principal stresses σ_1, σ_2, and σ_3. In many pressure component calculations, the t, l, and r directions may be so chosen that the shear stress components are zero and σ_1, σ_2, and σ_3 are identical to σ_t, σ_l, and σ_r.

(e) Calculate the stress differences S_{12}, S_{23}, and S_{31} from the relations:

$$S_{12} = \sigma_1 - \sigma_2$$
$$S_{23} = \sigma_2 - \sigma_3$$
$$S_{31} = \sigma_3 - \sigma_1$$

The stress intensity S is the largest absolute value of S_{12}, S_{23}, and S_{31}.

NOTE: Membrane stress intensity is derived from the stress components averaged across the thickness of the section. The averaging shall be performed at the component level in (b) or (c) above.

[3]See Tables NB-3217-1 and NB-3217-2 and Note (2) of Fig. NB-3221-1.

NB-3216 Derivation of Stress Differences

If the specified operation of the component does not meet the conditions of NB-3222.4(d), the ability of the component to withstand the specified cyclic service without fatigue failure shall be determined as provided in NB-3222.4(e). The determination shall be made on the basis of the stresses at a point of the component, and the allowable stress cycles shall be adequate for the specified service at every point. Only the stress differences due to cyclic service loadings as specified in the Design Specification need be considered.

NB-3216.1 Constant Principal Stress Direction. For any case in which the directions of the principal stresses at the point being considered do not change during the cycle, the steps stipulated in (a) through (c) below shall be taken to determine the alternating stress intensity.

(a) Principal Stresses. Consider the values of the three principal stresses at the point versus time for the complete stress cycle taking into account both the gross and local structural discontinuities and the thermal effects which vary during the cycle. These are designated as σ_1, σ_2, and σ_3 for later identification.

(b) Stress Differences. Determine the stress differences $S_{12}=\sigma_1-\sigma_2$, $S_{23}=\sigma_2-\sigma_3$, and $S_{31}=\sigma_3-\sigma_1$ versus time for the complete cycle. In what follows, the symbol S_{ij} is used to represent any one of these three stress differences.

(c) Alternating Stress Intensity. Determine the extremes of the range through which each stress difference S_{ij} fluctuates and find the absolute magnitude of this range for each S_{ij}. Call this magnitude S_{rij} and let $S_{\text{alt } ij}=0.5S_{rij}$. The alternating stress intensity S_{alt} is the largest $S_{\text{alt } ij}$ value.

NB-3216.2 Varying Principal Stress Direction. For any case in which the directions of the principal stresses at the point being considered do change during the stress cycle, it is necessary to use the more general procedure of (a) through (e) below.

(a) Consider the values of the six stress components σ_t, σ_l, σ_r, τ_{lt}, τ_{lr}, and τ_{rt} versus time for the complete stress cycle, taking into account both the gross and local structural discontinuities and the thermal effects which vary during the cycle.

(b) Choose a point in time when the conditions are one of the extremes for the cycle (either maximum or minimum, algebraically) and identify the stress components at this time by the subscript *i*. In most cases it will be possible to choose at least one time during the cycle when the conditions are known to be extreme. In some cases it may be necessary to try different points in time to find the one which results in the largest value of alternating stress intensity.

(c) Subtract each of the six stress components σ_{ti}, σ_{li}, etc., from the corresponding stress components σ_t, σ_l, etc., at each point in time during cycle and call the resulting components σ'_t, σ'_l, etc.

(d) At each point in time during the cycle, calculate the principal stresses σ'_1, σ'_2, and σ'_3 derived from the six stress components σ'_t, σ'_l, etc. Note that the directions of the principal stresses may change during the cycle but each principal stress retains its identity as it rotates.

(e) Determine the stress differences $S'_{12}=\sigma'_1-\sigma'_2$, $S'_{23}=\sigma'_2-\sigma'_3$, and $S'_{31}=\sigma'_3-\sigma'_1$ versus time for the complete cycle and find the largest absolute magnitude of any stress difference at any time. The alternating stress intensity S_{alt} is one-half of this magnitude.

NB-3217 Classification of Stresses

Tables NB-3217-1 and NB-3217-2 provide assistance in the determination of the category to which a stress should be assigned.

NB-3220 STRESS LIMITS FOR OTHER THAN BOLTS

NB-3221 Design Loadings

The stress intensity limits which must be satisfied for the Design Loadings (NB-3112) stated in the Design Specifications are the four limits of this paragraph and the Special Stress Limits of NB-3227. The provisions of NB-3228 may provide relief from certain of these stress limits if plastic analysis techniques are applied. The design stress intensity values S_m are given by NB-3229. The limits are summarized by Fig. NB-3221-1.

NB-3221.1 General Primary Membrane Stress Intensity. (Derived from P_m in Fig. NB-3221-1.) This stress intensity is derived from the average value across the thickness of a section of the general primary stresses (NB-3213.8) produced by design internal pressure and other specified Design Mechanical Loads, but excluding all secondary and peak stresses. Averaging is to be applied to the stress components prior to determination of the stress intensity values. The allowable value of this stress intensity is S_m at the Design Temperature.

NB-3221.2 Local Membrane Stress Intensity. (Derived from P_L in Fig. NB-3221-1.) This stress intensity is derived from the average value across the thickness of a section of the local primary stresses (NB-3213.10)

TABLE NB-3217-1
CLASSIFICATION OF STRESS INTENSITY IN VESSELS FOR SOME TYPICAL CASES[1]

Vessel Part	Location	Origin of Stress	Type of Stress	Classification
Cylindrical or spherical shell	Shell plate remote from discontinuities	Internal pressure	General membrane Gradient through plate thickness	P_m Q
		Axial thermal gradient	Membrane Bending	Q Q
	Junction with head or flange	Internal pressure	Membrane Bending	P_L Q [Note (2)]
Any shell or head	Any section across entire vessel	External load or moment, or internal pressure	General membrane averaged across full section	P_m
		External load or moment	Bending across full section	P_m
	Near nozzle or other opening	External load or moment, or internal pressure	Local membrane Bending Peak (fillet or corner)	P_L Q F
	Any location	Temperature difference between shell and head	Membrane Bending	Q Q
Dished head or conical head	Crown	Internal pressure	Membrane Bending	P_m P_b
	Knuckle or junction to shell	Internal pressure	Membrane Bending	P_L [Note (3)] Q
Flat head	Center region	Internal pressure	Membrane Bending	P_m P_b
	Junction to shell	Internal pressure	Membrane Bending	P_L Q [Note (2)]
Perforated head or shell	Typical ligament in a uniform pattern	Pressure	Membrane (averaged through cross section) Bending (averaged through width of ligament, but gradient through plate) Peak	P_m P_b F
	Isolated or atypical ligament	Pressure	Membrane Bending Peak	Q F F
Nozzle (NB-3227.5)	Within the limits of reinforcement defined by NB-3334	Pressure and external loads and moments, including those attributable to restrained free end displacements of attached piping	General membrane Bending (other than gross structural discontinuity stresses) averaged through nozzle thickness	P_m P_m

(Table NB-3217-1 continues on next page)

TABLE NB-3217-1 (CONT'D)
CLASSIFICATION OF STRESS INTENSITY IN VESSELS FOR SOME TYPICAL CASES[1]

Vessel Part	Location	Origin of Stress	Type of Stress	Classification
Nozzle (NB-3227.5)	Outside the limits of reinforcement defined by NB-3334	Pressure and external axial, shear, and torsional loads other than those attributable to restrained free end displacements of attached piping	General membrane stresses	P_m
		Pressure and external loads and moments other than those attributable to restrained free end displacements of attached piping	Membrane Bending	P_L P_b
		Pressure and all external loads and moments	Membrane Bending Peak	P_L Q F
	Nozzle wall	Gross structural discontinuities	Local membrane Bending Peak	P_L Q F
		Differential expansion	Membrane Bending Peak	Q Q F
Cladding	Any	Differential expansion	Membrane Bending	F F
Any	Any	Radial temperature distribution [Note (4)]	Equivalent linear stress [Note (5)]	Q
			Nonlinear portion of stress distribution	F
Any	Any	Any	Stress concentration (notch effect)	F

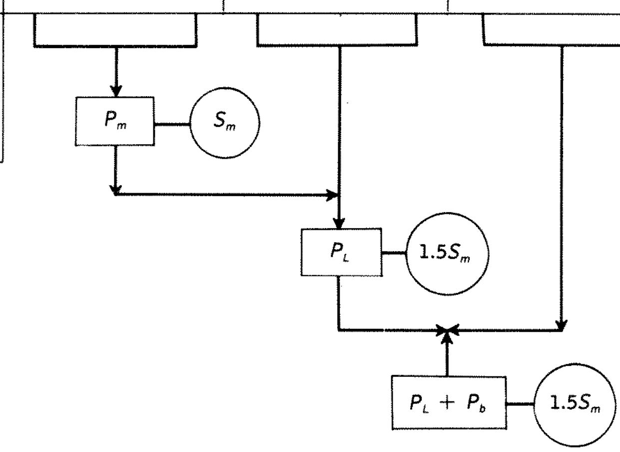
NOTES:
(1) Q & F classification of stresses refers to other than design condition (Fig. NB-3222-1).
(2) If the bending moment at the edge is required to maintain the bending stress in the middle to acceptable limits, the edge bending is classified as P_b. Otherwise, it is classified as Q.
(3) Consideration shall also be given to the possibility of wrinkling and excessive deformation in vessels with a large diameter–thickness ratio.
(4) Consider possibility of thermal stress ratchet.
(5) Equivalent linear stress if defined as the linear stress distribution which has the same net bending moment as the actual stress distribution.

TABLE NB-3217-2
CLASSIFICATION OF STRESS INTENSITY IN PIPING, TYPICAL CASES

Piping Component	Locations	Origin of Stress	Classification	Discontinuities Considered	
				Gross	Local
Pipe or tube, elbows, and reducers. Intersections and branch connections, except in crotch regions	Any, except crotch regions of intersections	Internal pressure	P_m P_L and Q F	No Yes Yes	No No Yes
		Sustained mechanical loads, including weight	P_b P_L and Q F	No Yes Yes	No No Yes
		Expansion Axial thermal gradient	P_e F Q F	Yes Yes Yes Yes	No Yes No Yes
Intersections, including tees and branch connections	In crotch region	Internal pressure, sustained mechanical loads, and expansion	P_L and Q [Note (1)] F	Yes Yes	No Yes
		Axial thermal gradient	Q F	Yes Yes	No Yes
Bolts and flanges	Any	Internal pressure, gasket compression, and bolt load	P_m Q F	No Yes Yes	No No Yes
		Thermal gradient	Q F	Yes Yes	No Yes
		Expansion	P_e F	Yes Yes	No Yes
Any	Any	Nonlinear radial thermal gradient	F	Yes	Yes
		Linear radial thermal gradient	F	Yes	No
		Anchor point motions, including those resulting from earthquake	Q	Yes	No

NOTE:
(1) Analysis is not required when reinforced in accordance with NB-3643.

Stress Category	Primary		
	General Membrane	Local Membrane	Bending
Description (for examples see Table NB-3217-1)	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Average stress across any solid section. Considers discontinuities but not concentrations. Produced only by mechanical loads.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads. [Note (1)]
Symbol [Note (2)]	P_m	P_L	P_b
Combination of stress components and allowable limits of stress intensities.			



Legend

—— Use Design Loads

NOTES:
(1) Bending component of primary stress for piping shall be the stress proportional to the distance from centroid of pipe cross section.
(2) The symbols P_m, P_L, and P_b do not represent single quantities, but rather sets of six quantities representing the six stress components σ_t, σ_l, σ_r, τ_{lt}, τ_{lr}, and τ_{rt}.

FIG. NB-3221-1 STRESS CATEGORIES AND LIMITS OF STRESS INTENSITY FOR DESIGN CONDITIONS

produced by Design Pressure and specified Design Mechanical Loads, but excluding all thermal and peak stresses. Averaging is to be applied to the stress components prior to the determination of the stress intensity values. The allowable value of this stress intensity is $1.5S_m$.

NB-3221.3 Primary Membrane (General or Local) Plus Primary Bending Stress Intensity. (Derived from $P_L \pm P_b$ in Fig. NB-3221-1.) This stress intensity is derived from the highest value across the thickness of a section of the general or local primary membrane stresses plus primary bending stresses produced by Design Pressure and other specified Design Mechanical Loads, but excluding all secondary and peak stresses. For solid rectangular sections, the allowable value of this stress intensity is $1.5S_m$. For other than solid rectangular sections, a value of α times the limit established in NB-3221.1 may be used, where the factor α is defined as the ratio of the load set producing a fully plastic section to the load set producing initial yielding in the extreme fibers of the section. In the evaluation of the initial yield and fully plastic section capacities, the ratios of each individual load in the respective load set to each other load in that load set shall be the same as the respective ratios of the individual loads in the specified design load set. The value of α shall not exceed the value calculated for bending only ($P_m = 0$). In no case shall the value of α exceed 1.5. The propensity for buckling of the part of the section that is in compression shall be investigated. The α factor is not permitted for Level D Service Limits when inelastic component analysis is used as permitted in Appendix F.

NB-3221.4 External Pressure. The provisions of NB-3133 apply.

NB-3222 Level A Service Limits

Level A Service Limits must be satisfied for the Service Conditions [NCA-2142.2(b)(1)] for which these limits are designated in the Design Specifications and are the four limits of this paragraph and the Special Stress Limits of NB-3227. The provisions of NB-3228 may provide relief from certain of these stress limits if plastic analysis techniques are applied. The design stress intensity values S_m are given by NB-3229. The limits are summarized by Fig. NB-3222-1.

NB-3222.1 Primary Membrane and Bending Stress Intensities. There are no specific limits established on the primary stresses in the Level A Limits. However, the stresses due to primary loads presented during normal service must be computed and combined with the effects of other loadings in satisfying the remaining limits.

NB-3222.2 Primary Plus Secondary Stress Intensity.[4] This stress intensity is derived from the highest value at any point across the thickness of a section of the general or local primary membrane stresses, plus primary bending stresses plus secondary stresses, produced by the specified service pressure and other specified mechanical loads and by general thermal effects associated with normal Service Conditions. The allowable value of the maximum range of this stress intensity is $3S_m$.

NB-3222.3 Expansion Stress Intensity. (Not applicable to vessels; P_e in Fig. NB-3222-1.) This stress intensity is the highest value of stress, neglecting local structural discontinuities, produced at any point across the thickness of a section by the loadings that result from restraint of free end displacement. The allowable value of the maximum range of this stress intensity is $3S_m$.

NB-3222.4 Analysis for Cyclic Operation

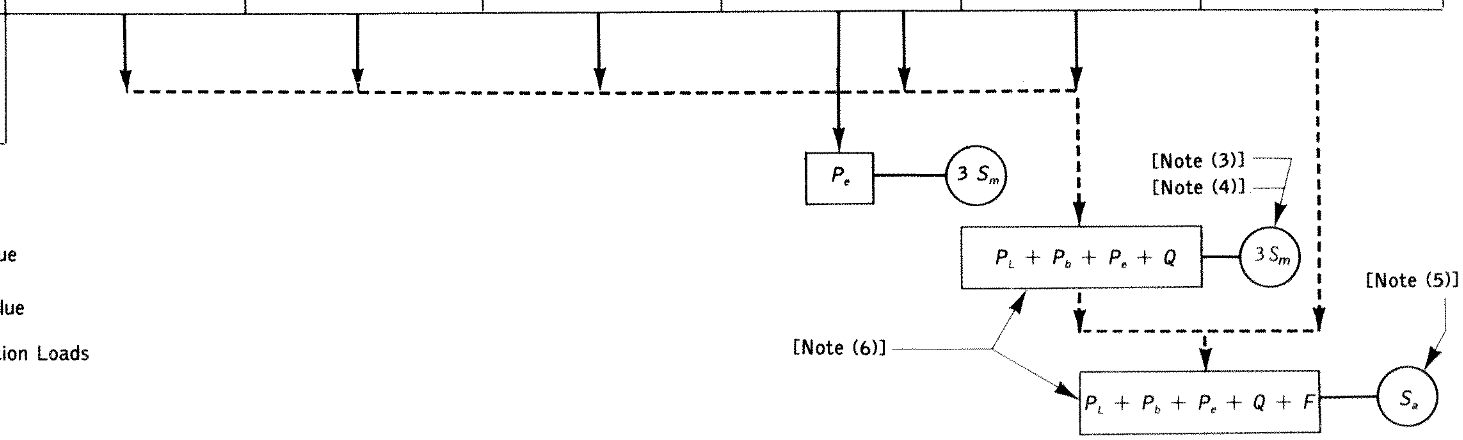
(a) Suitability for Cyclic Condition. The suitability of a component for specified service loadings involving cyclic application of loads and thermal conditions shall be determined by the methods described herein, except that the suitability of high strength bolts shall be determined by the methods of NB-3232.3(b) and the possibility of thermal stress ratchet shall be investigated in accordance with NB-3222.5. If the specified Service Loadings of the component meet all of the conditions of (d) below, no analysis for cyclic service is required, and it may be assumed that the limits on peak stress intensities as governed by fatigue have been satisfied by compliance with the applicable requirements for material, design, fabrication, examination, and testing of this Subsection. If the Service Loadings do not meet all the conditions of (d) below, a fatigue analysis shall be made in accordance with (e) below or a fatigue test shall be made in accordance with II-1200.

(b) Peak Stress Intensity. This stress intensity is derived from the highest value at any point across the thickness of a section of the combination of all primary, secondary, and peak stresses produced by specified service pressures and other mechanical loads, and by general and local thermal effects associated with nor-

[4]The concept of stress differences discussed in NB-3216 is essential to determination to the maximum range, since algebraic signs must be retained in the computation. Note that this limitation on range is applicable to the entire history of normal Service Conditions, not just to the stresses resulting from each individual transient.

Stress Category	Primary: General Membrane	Primary: Local Membrane	Primary: Bending	Secondary: Expansion	Secondary: Membrane plus Bending	Peak
Description (for examples see Table NB-3217-1)	Average primary stress across solid section. Excludes effects of discontinuities and concentrations. Produced by pressure and mechanical loads.	Average stress across any solid section. Considers effects of discontinuities but not concentrations. Produced by pressure and mechanical loads, including inertia earthquake effects.	Component of primary stress proportional to distance from centroid of solid section. Excludes effects of discontinuities and concentrations. Produced by pressure and mechanical loads, including inertia earthquake effects. [Note (1)]	Stresses which result from the constraint of free end displacement. Considers effects of discontinuities but not local stress concentration (not applicable to vessels).	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by pressure, mechanical loads, or differential thermal expansion. Excludes local stress concentrations.	(1) Increment added to primary or secondary stress by a concentration (notch). (2) Certain thermal stresses which may cause fatigue but not distortion.
Symbol [Note (2)]	P_m	P_L	P_b	P_e	Q	F

Combination of stress components and allowable limits of stress intensities



Legend

Allowable Value

Calculated Value

Service Condition Loads

NOTES:

(1) Bending component of primary stress for piping shall be the stress proportional to the distance from centroid of pipe cross section.

(2) The symbols P_m, P_L, P_b, P_e, Q, and F do not represent single quantities, but sets of six quantities representing the six stress components σ_t, σ_l, σ_r, τ_{lt}, τ_{lr}, and τ_{rt}.

(3) When the secondary stress is due to a temperature transient at the point at which the stresses are being analyzed or to restraint of free end deflection, the value of S_m shall be taken as the average of the tabulated S_m values for the highest and the lowest temperatures of the metal during the transient. When part or all of the secondary stress is due to mechanical load, the value of S_m shall not exceed the value for the highest temperature during the transient.

(4) Special rules for exceeding $3S_m$ are provided in NB-3228.5.

(5) S_a is obtained from the fatigue curves, Figs. I-9.0. The allowable stress intensity for the full range of fluctuation is $2S_a$.

(6) The stresses in category Q are those parts of the total stress that are produced by thermal gradients, structural discontinuities, etc., and they do not include primary stresses that may also exist at the same point. However, it should be noted that a detailed stress analysis frequently gives the combination of primary and secondary stresses directly and, when appropriate, the calculated value represents the total of $P_m + P_b + Q$, and not Q alone. Similarly, if the stress in category F is produced by a stress concentration, the quantity F is the additional stress produced by the notch over and above the nominal stress. For example, if a point has a nominal stress intensity P_m and has a notch with a stress concentration factor K, then $P_m \leq S_m$, $P_b = Q$, $Q = 0$, $F = P_m(K - 1)$, and the peak stress intensity equals $P_m + P_m(K - 1) = KP_m$. However, P_L is the total membrane stress that results from mechanical loads, including discontinuity effects, rather than a stress increment. Therefore, the P_L value always includes the P_m contribution.

FIG. NB-3222-1 STRESS CATEGORIES AND LIMITS OF STRESS INTENSITY FOR LEVEL A AND LEVEL B SERVICE LIMITS mal service conditions, and including the effects of gross and local structural discontinuities.

(c) Conditions and Procedures. The conditions and procedures of NB-3222.4 are based on a comparison of peak stresses with strain cycling fatigue data. The strain cycling fatigue data are represented by design fatigue strength curves of Figs. I-9.0. These curves show the allowable amplitude S_a of the alternating stress intensity component (one-half of the alternating stress intensity range) plotted against the number of cycles. This stress intensity amplitude is calculated on the assumption of elastic behavior and, hence, has the dimensions of stress, but it does not represent a real stress when the elastic range is exceeded. The fatigue curves are obtained from uniaxial strain cycling data in which the imposed strains have been multiplied by the elastic modulus and a design margin has been provided so as to make the calculated stress intensity amplitude and the allowable stress intensity amplitude directly comparable.[5] Where necessary, the curves have been adjusted to include the maximum effects of mean stress, which is the condition where the stress fluctuates about a mean value that is different from zero. As a consequence of this procedure, it is essential that the requirements of NB-3222.2 be satisfied at all times with transient stresses included, and that the calculated value of the alternating stress intensity be proportional to the actual strain amplitude. To evaluate the effect of alternating stresses of varying amplitudes, a linear damage relation is assumed in (e)(4) below.

(d) Components Not Requiring Analysis for Cyclic Service. An analysis for cyclic service is not required, and it may be assumed that the limits on peak stress intensities as governed by fatigue have been satisfied for a component by compliance with the applicable requirements for material, design, fabrication, examination, and testing of this Subsection, provided the specified Service Loading[6] of the component or portion thereof meets all the conditions stipulated in (1) through (6) below.

(1) Atmospheric to Service Pressure Cycle. The specified number of times (including startup and shutdown) that the pressure will be cycled from atmospheric pressure to service pressure and back to atmospheric pressure during normal service does not exceed the number of cycles on the applicable fatigue curve of Figs. I-9.0 corresponding to an S_a value of three times the S_m value for the material at service temperature.

(2) Normal Service Pressure Fluctuation. The specified full range of pressure fluctuations during normal service does not exceed the quantity $\frac{1}{3} \times$ Design Pressure $\times (S_a/S_m)$, where S_a is the value obtained from the applicable design fatigue curve for the total specified number of significant pressure fluctuations and S_m is the allowable stress intensity for the material at service temperature. If the total specified number of significant pressure fluctuations exceeds the maximum number of cycles defined on the applicable design fatigue curve, the S_a value corresponding to the maximum number of cycles defined on the curve may be used. Significant pressure fluctuations are those for which the total excursion exceeds the quantity:

$$\text{Design Pressure} \times \tfrac{1}{3} \times (S/S_m)$$

where S is defined as follows.

(a) If the total specified number of service cycles is 10^6 cycles or less, S is the value of S_a obtained from the applicable design fatigue curve for 10^6 cycles.

(b) If the total specified number of service cycles exceeds 10^6 cycles, S is the value of S_a obtained from the applicable design fatigue curve for the maximum number of cycles defined on the curve.

(3) Temperature Difference — Startup and Shutdown. The temperature difference, °F, between any two adjacent points[7] of the component during normal service does not exceed $S_a/2E\alpha$, where S_a is the value obtained from the applicable design fatigue curves for the specified number of startup–shutdown cycles, α is the value of the instantaneous coefficient of thermal expansion at the mean value of the temperatures at the two points as given by Table I-5.0, and E is taken from Table I-6.0 at the mean value of the temperature at the two points.

[5]As an exception to the use of strain controlled test data, Fig. I-9.2.2, Curves B and C are based on load controlled fatigue data.

[6]As is stated in NB-3223, for components operating within the temperature limits of this Subsection, Service Loadings for which Level B Limits are designated must be considered as though Level A Limits were designated in evaluating exemptions from fatigue analysis.

[7]Adjacent points are defined in (a), (b), and (c) below.

(a) For surface temperature differences on surfaces of revolution in the meridional direction, adjacent points are defined as points that are less than the distance $2\sqrt{Rt}$, where R is the radius measured normal to the surface from the axis of rotation to the midwall and t is the thickness of the part at the point under consideration. If the product Rt varies, the average value of the points shall be used.

(b) For surface temperature differences on surfaces of revolution in the circumferential direction and on flat parts, such as flanges and flat heads, adjacent points are defined as any two points on the same surface.

(c) For through-thickness temperature differences, adjacent points are defined as any two points on a line normal to any surface.

(4) Temperature Difference — Normal Service.[8] The temperature difference, °F, between any two adjacent points[7] does not change[9] during normal service by more than the quantity $S_a/2E\alpha$, where S_a is the value obtained from the applicable design fatigue curve of Figs. I-9.0 for the total specified number of significant temperature difference fluctuations. A temperature difference fluctuation shall be considered to be significant if its total algebraic range exceeds the quantity $S/2E\alpha$, where S is defined as follows.

(a) If the total specified number of service cycles is 10^6 cycles or less, S is the value of S_a obtained from the applicable design fatigue curve for 10^6 cycles.

(b) If the total specified number of service cycles exceeds 10^6 cycles, S is the value of S_a obtained from the applicable design fatigue curve for the maximum number of cycles defined on the curve.

(5) Temperature Difference — Dissimilar Materials. For components fabricated from materials of differing moduli of elasticity or coefficients of thermal expansion, the total algebraic range of temperature fluctuation, °F, experienced by the component during normal service does not exceed the magnitude $S_a/2(E_1\alpha_1 - E_2\alpha_2)$, where S_a is the value obtained from the applicable design fatigue curve for the total specified number of significant temperature fluctuations, E_1 and E_2 are the moduli of elasticity (Table I-6.0), and α_1 and α_2 are the values of the instantaneous coefficients of thermal expansion (Table I-5.0) at the mean temperature value involved for the two materials of construction. A temperature fluctuation shall be considered to be significant if its total excursion exceeds the quantity $S/2(E_1\alpha_1 - E_2\alpha_2)$, where S is defined as follows.

(a) If the total specified number of service cycles is 10^6 cycles or less, S is the value of S_a obtained from the applicable design fatigue curve for 10^6 cycles.

(b) If the total specified number of service cycles exceeds 10^6 cycles, S is the value of S_a obtained from the applicable design fatigue curve for the maximum number of cycles defined on the curve. If the two materials used have different applicable design fatigue curves, the lower value of S_a shall be used in applying the rules of this paragraph.

(6) Mechanical Loads. The specified full range of mechanical loads, excluding pressure but including pipe reactions, does not result in load stresses whose range exceeds the S_a value obtained from the applicable design fatigue curve of Figs. I-9.0 for the total specified number of significant load fluctuations. If the total specified number of significant load fluctuations exceeds the maximum number of cycles defined on the applicable design fatigue curve, the S_a value corresponding to the maximum number of cycles defined on the curve may be used. A load fluctuation shall be considered to be significant if the total excursion of load stress exceeds the quantity S, where S is defined as follows.

(a) If the total specified number of service cycles is 10^6 cycles or less, S is the value of S_a obtained from the applicable design fatigue curve for 10^6 cycles.

(b) If the total specified number of service cycles exceeds 10^6 cycles, S is the value of S_a obtained from the applicable design fatigue curve for the maximum number of cycles defined on the curve.

(e) Procedure for Analysis for Cyclic Loading. If the specified Service Loadings for the component do not meet the conditions of NB-3222.4(d), the ability of the component to withstand the specified cyclic service without fatigue failure shall be determined as provided in this subsubparagraph. The determination shall be made on the basis of the stresses at a point, and the allowable stress cycles shall be adequate for the specified Service Loadings at every point. Only the stress differences due to service cycles as specified in the Design Specifications need be considered. Compliance with these requirements means only that the component is suitable from the standpoint of possible fatigue failure; complete suitability for the specified Service Loadings is also dependent on meeting the general stress limits of NB-3222 and any applicable special stress limits of NB-3227.

(1) Stress Differences. For each condition of normal service, determine the stress differences and the alternating stress intensity S_a in accordance with NB-3216.

(2) Local Structural Discontinuities. These effects shall be evaluated for all conditions using stress concentration factors determined from theoretical, experimental, or photoelastic studies, or numerical stress analysis techniques. Experimentally determined fatigue strength reduction factors may be used when determined in accordance with the procedures of II-1600, except for high strength alloy steel bolting for which the requirements of NB-3232.3(c) shall apply when using the design fatigue curve of Fig. I-9.4. Except for the case of crack-like defects and specified piping geometries for which specific values are given in NB-3680, no fatigue strength reduction factor greater than five need be used.

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[8] *Normal service* is defined as service, other than startup and shutdown, resulting in specified Service Loadings for which Level A Limits, Level B Limits, or Testing Limits are designated.

[9] The algebraic range of the difference shall be used.

(3) Design Fatigue Curves. Figures I-9.0 contain the applicable fatigue design curves for the materials permitted by this Subsection. When more than one curve is presented for a given material, the applicability of each is identified. Where curves for various strength levels of a material are given, linear interpolation may be used for intermediate strength levels of these materials. The strength level is the specified minimum room temperature value. The design fatigue curves of Figs. I-9.0 are defined over a cyclic range of 10 to 10^6 cycles, except that for austenitic steels, nickel–chromium–iron alloys, and nickel–iron–chromium alloys, nickel–chromium–molybdenum–iron alloys, and nickel–copper alloys, the design fatigue curve is extended to 10^{11} cycles in Figs. I-9.2.2 and I-9.5. Criteria for the use of the latter curve are given in Fig. I-9.2.2 and are also presented graphically by the flow chart given in Fig. I-9.2.3.

(4) Effect of Elastic Modulus. Multiply S_{alt} (as determined in NB-3216.1 or NB-3216.2) by the ratio of the modulus of elasticity given on the design fatigue curve to the value of the modulus of elasticity used in the analysis. Enter the applicable design fatigue curve of Figs. I-9.0 at this value on the ordinate axis and find the corresponding number of cycles on the abscissa. If the service cycle being considered is the only one which produces significant fluctuating stresses, this is the allowable number of cycles.

(5) Cumulative Damage. If there are two or more types of stress cycle which produce significant stresses, their cumulative effect shall be evaluated as stipulated in Steps 1 through 6 below.

Step 1: Designate the specified number of times each type of stress cycle of types 1, 2, 3, . . ., n, will be repeated during the life of the component as n_1, n_2, n_3, . . ., n_n, respectively.

NOTE: In determining n_1, n_2, n_3, . . ., n_n, consideration shall be given to the superposition of cycles of various origins which produce a total stress difference range greater than the stress difference ranges of the individual cycles. For example, if one type of stress cycle produces 1000 cycles of a stress difference variation from zero to +60,000 psi and another type of stress cycle produces 10,000 cycles of a stress difference variation from zero to −50,000 psi, the two types of cycle to be considered are defined by the following parameters:

(a) for type 1 cycle, n_1 = 1000 and $S_{alt\ 1}$ = (60,000 + 50,000)/2 = 55,000 psi;

(b) for type 2 cycle, n_2 = 9000 and $S_{alt\ 2}$ = (50,000 + 0)/2 = 25,000 psi.

Step 2: For each type of stress cycle, determine the alternating stress intensity S_{alt} by the procedures of NB-3216.1 or NB-3216.2 above. Call these quantities $S_{alt\ 1}$, $S_{alt\ 2}$, $S_{alt\ 3}$, . . ., $S_{alt\ n}$.

Step 3: For each value $S_{alt\ 1}$, $S_{alt\ 2}$, $S_{alt\ 3}$, . . ., $S_{alt\ n}$, 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, . . ., N_n.

Step 4: For each type of stress cycle, calculate the usage factors U_1, U_2, U_3, . . ., U_n, from $U_1 = n_1/N_1$, $U_2 = n_2/N_2$, $U_3 = n_3/N_3$, . . . , $U_n = n_n/N_n$.

Step 5: Calculate the cumulative usage factor U from $U = U_1 + U_2 + U_3 + \ldots + U_n$.

Step 6: The cumulative usage factor U shall not exceed 1.0.

NB-3222.5 Thermal Stress Ratchet. It should be noted that under certain combinations of steady state and cyclic loadings there is a possibility of large distortions developing as the result of ratchet action; that is, the deformation increases by a nearly equal amount for each cycle. Examples of this phenomenon are treated in this subparagraph and in NB-3227.3.

(a) The limiting value of the maximum cyclic thermal stress permitted in a portion of an axisymmetric shell loaded by steady state internal pressure in order to prevent cyclic growth in diameter is as follows. Let

y' = maximum allowable range of thermal stress computed on an elastic basis divided by the yield strength[10] S_y

x = maximum general membrane stress due to pressure divided by the yield strength S_y

Case 1: Linear variation of temperature through the wall: for $0 < x < 0.5$, $y' = 1/x$ and, for $0.5 < x < 1.0$, $y' = 4(1 - x)$.

Case 2: Parabolic constantly increasing or constantly decreasing variation of temperature through the wall: for $0.615 < x < 1.0$, $y' = 5.2(1 - x)$ and, approximately for $x < 0.615$, y' = 4.65, 3.55, and 2.70 for x = 0.3, 0.4, and 0.5, respectively.

(b) Use of yield strength S_y in the above relations instead of the proportional limit allows a small amount of growth during each cycle until strain hardening raises the proportional limit to S_y. If the yield strength of the material is higher than the endurance limit[11] for

[10] It is permissible to use $1.5S_m$ whenever it is greater than S_y.

[11] The endurance limit shall be taken as two times the S_a value at 10^7 cycles in the applicable fatigue curve of Figs. I-9.0, except that for the curves of Figs. I-9.2.1 and I-9.2.2 the endurance limit shall be taken as two times the S_a value at 10^{11} cycles obtained from Curve A.

the material, the latter value shall be used if there is to be a large number of cycles because strain softening may occur.

NB-3222.6 Deformation Limits. Any deformation limits prescribed by the Design Specifications shall be satisfied.

NB-3223 Level B Service Limits

For components operating within the temperature limits of this Subsection the requirements of (a), (b), and (c) below apply.

(a) The values of Level A Service Limits shall apply for Level B Service Limits. In addition, if a pressure for which Level B Limits are designated exceeds the design pressure, the stress limits of Fig. NB-3221-1 shall apply using allowable stress intensity values of 110% of those given on Fig. NB-3221-1 and the loadings for which Level B Limits are designated.

(b) In evaluating possible exemption from fatigue analysis by the methods of NB-3222.4(d), Service Loadings for which Level B Limits are designated shall be considered as though Level A Limits were designated.

(c) Any deformation limits prescribed by the Design Specifications shall be satisfied.

NB-3224 Level C Service Limits

If the Design Specifications specify any Service Loadings for which Level C Service Limits are designated [NCA-2142.2(b)(3)], the rules used in evaluating these loadings shall be those used for other loadings, except as modified by the following subparagraphs and as summarized in Fig. NB-3224-1.

NB-3224.1 Primary Stress Limits. The primary stress limits of NB-3221 shall be satisfied using an S_m value equal to the greater of 120% of the tabulated S_m value or 100% of the tabulated yield strength, with both values taken at the appropriate temperature. In addition, for ferritic material, the P_m elastic analysis limits for pressure loadings alone shall be equal to the greater of $1.1S_m$ or $0.9S_y$.

NB-3224.2 External Pressure. The permissible external pressure shall be taken as 120% of that given by the rules of NB-3133.

NB-3224.3 Special Stress Limits. The permissible values for special stress limit shall be taken as 120% of the values given in NB-3227.4 and NB-3228.

NB-3224.4 Secondary and Peak Stresses. The requirements of NB-3222.2, NB-3222.4, NB-3222.5, and NB-3227.3 need not be satisfied.

NB-3224.5 Fatigue Requirements. Service Loadings for which Level C Service Limits are designated need not be considered when applying the procedures of NB-3222.4(a) to determine whether or not a fatigue analysis is required.

NB-3224.6 Deformation Limits. Any deformation limits prescribed by the Design Specifications shall be considered.

NB-3225 Level D Service Limits

If the Design Specifications specify any Service Loadings for which Level D Limits are designated [NCA-2142.2(b)(4)], the rules contained in Appendix F may be used in evaluating these loadings, independently of all other Design and Service Loadings.

NB-3226 Testing Limits

The evaluation of pressure test loadings (NCA-2142.3) shall be in accordance with (a) through (e) below, except that these rules do not apply to the items in NB-3500.

(a) The general primary membrane stress intensity P_m shall not exceed 90% of the tabulated yield strength S_y at test temperature.

(b) The primary membrane plus bending stress intensity $P_m + P_b$ shall not exceed the applicable limits given in (1) or (2) below.

(1) For $P_m \leq 0.67S_y$

$$P_m + P_b \leq 1.35S_y$$

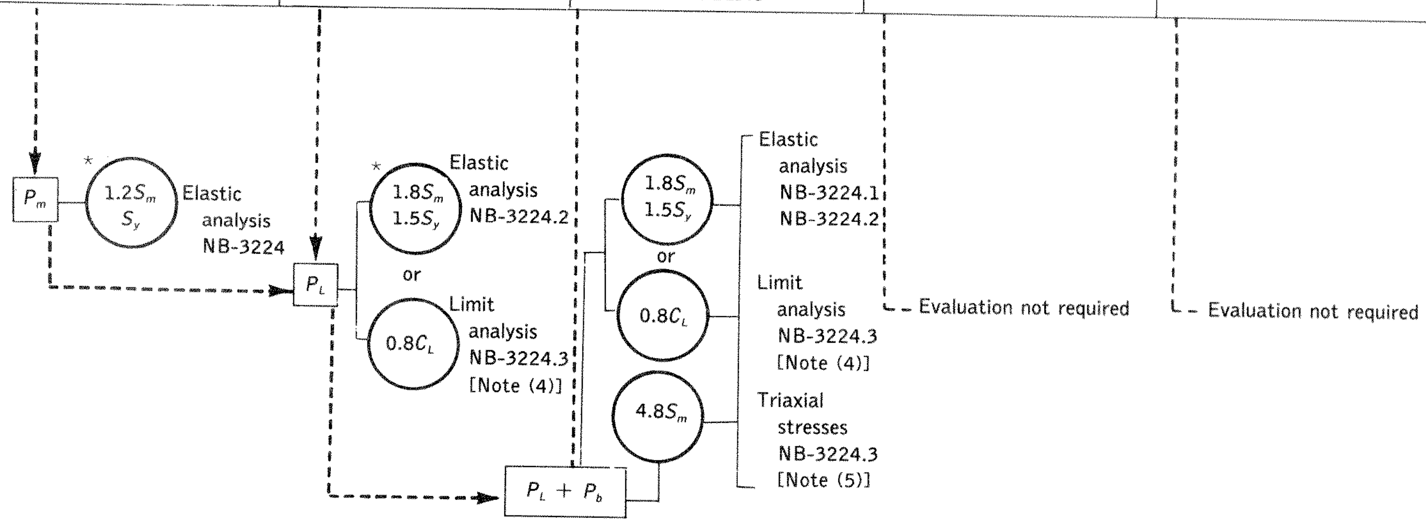
(2) For $0.67S_y < P_m \leq 0.90S_y$

$$P_m + P_b \leq (2.15S_y - 1.2P_m)$$

where S_y is the tabulated yield strength at test temperature.

(c) The external pressure shall not exceed 135% of the value determined by the rules of NB-3133. Alternatively, an external hydrostatic test pressure may be applied up to a maximum of 80% of the lower of the collapse or elastic instability pressures determined by analysis or experimental procedures (NB-3228 and Appendix II) including consideration of allowable toler-

Stress Category	Primary [Notes (1) and (2)]			Secondary [Note (1)] Membrane plus Bending	Peak [Note (1)]
	General Membrane	Local Membrane	Bending		
Description (for examples see Table NB-3217-1)	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Average stress across any solid section. Considers discontinuities but not concentrations. Produced only by mechanical loads.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by mechanical load or by differential thermal expansion. Excludes local stress concentration.	(1) Increment added to primary or secondary stress by a concentration (notch). (2) Certain thermal stresses which may cause fatigue but not distortion of vessel shape.
Symbol [Note (1)]	P_m NB-3213.6 and NB-3213.8 [Note (3)]	P_L NB-3213.10	P_b NB-3213.7 and NB-3213.8	Q NB-3213.9	F NB-3213.11
Combination of stress components and allowable limits of stress intensities					



NOTES:

(1) The symbols P_m, P_L, P_b, Q, and F do not represent single quantities, but rather sets of six quantities representing the six stress components σ_t, σ_l, σ_r, τ_{lt}, τ_{lr}, and τ_{rt}.

(2) For configurations where compressive stresses occur, the stress limits shall be revised to take into account critical buckling stresses [NB-3211(c)].

(3) The limits shown are for stresses resulting from pressure in combination with other mechanical loads. For ferritic materials, the P_m elastic analysis limits for pressure loadings alone shall be equal to the greater of $1.1S_m$ or $0.9S_y$.

(4) C_L the collapse load calculated on the basis of the lower bound theorem of limit analysis and yield strength values specified in Table I-2.1 or I-2.2 (NB-3213.22).

(5) The triaxial stresses represent the algebraic sum of the three primary principal stresses ($\sigma_1 + \sigma_2 + \sigma_3$) for the combination of stress components.

* Use the greater of the values specified.

FIG. NB-3224-1 STRESS CATEGORIES AND LIMITS OF STRESS INTENSITY FOR LEVEL C SERVICE LIMITS

ances. If a collapse analysis is performed, it shall be a lower bound limit analysis assuming ideally elastic–plastic (nonstrain-hardening) material having a yield strength equal to its tabulated yield strength at test temperature.

(d) For the 1.25 Design Pressure hydrostatic test of NB-6221, or the 1.20 to 1.25 Design Pressure pneumatic tests of NB-6321, the stresses shall be calculated and compared to the limits of (a), (b), and (c) above. This calculation and the fatigue evaluation of (e) below need not be revised unless the actual hydrostatic test pressure of NB-6221 exceeds the 1.25 Design Pressure by more than 6%, or the actual pneumatic test pressure of NB-6321 exceeds 1.25 Design Pressure.

(e) Tests, with the exception of either the first 10 hydrostatic tests in accordance with NB-6220, the first 10 pneumatic tests in accordance with NB-6320, or any combination of 10 such tests, shall be considered in the fatigue evaluation of the component. In this fatigue evaluation, the limits on the primary plus secondary stress intensity range (NB-3222.2) may be taken as the larger of $3S_m$ or $2S_y$ when at least one extreme of the stress intensity range is determined by the Test Loadings.

NB-3227 Special Stress Limits

The following deviations from the basic stress limits are provided to cover special Service Loadings or configurations. Some of these deviations are more restrictive, and some are less restrictive, than the basic stress limits. In cases of conflict between these requirements and the basic stress limits, the rules of NB-3227 take precedence for the particular situations to which they apply.

NB-3227.1 Bearing Loads

(a) The average bearing stress for resistance to crushing under the maximum load, experienced as a result of Design Loadings, Test Loadings, or any Service Loadings, except those for which Level D Limits are designated, shall be limited to S_y at temperature, except that when the distance to a free edge is larger than the distance over which the bearing load is applied, a stress of $1.5S_y$ at temperature is permitted. For clad surfaces, the yield strength of the base metal may be used if, when calculating the bearing stress, the bearing area is taken as the lesser of the actual contact area or the area of the base metal supporting the contact surface.

(b) When bearing loads are applied near free edges, such as at a protruding ledge, the possibility of a shear failure shall be considered. In the case of load stress only (NB-3213.12) the average shear stress shall be limited to $0.6S_m$. In the case of load stress plus secondary stress (NB-3213.10) the average shear stress shall not exceed (1) or (2) below:

(1) for materials to which Note (2) of Table I-1.2 applies, the lower of $0.5S_y$ at 100°F and $0.675S_y$ at temperature;

(2) for all other materials, $0.5S_y$ at temperature.

For clad surfaces, if the configuration or thickness is such that a shear failure could occur entirely within the clad material, the allowable shear stress for the cladding shall be determined from the properties of the equivalent wrought material. If the configuration is such that a shear failure could occur across a path that is partially base metal and partially clad material, the allowable shear stresses for each material shall be used when evaluating the combined resistance to this type of failure.

(c) When considering bearing stresses in pins and similar members, the S_y at temperature value is applicable, except that a value of $1.5S_y$ may be used if no credit is given to bearing area within one pin diameter from a plate edge.

NB-3227.2 Pure Shear

(a) The average primary shear stress across a section loaded in pure shear, experienced as a result of Design Loadings, Test Loadings, or any Service Loadings, except those for which Level D Limits are designated (for example, keys, shear rings, screw threads), shall be limited to $0.6S_m$.

(b) The maximum primary shear that is experienced as a result of Design Loadings, Test Loadings, or any Service Loadings (except those for which Level D Limits are designated), exclusive of stress concentration, at the periphery of a solid circular section in torsion shall be limited to $0.8S_m$. Primary plus secondary and peak shear stresses shall be converted to stress intensities (equal to two times the pure shear stress) and as such shall not exceed the basic stress limits of NB-3222.2 and NB-3222.4.

NB-3227.3 Progressive Distortion of Nonintegral Connections. Screwed on caps, screwed in plugs, shear ring closures, and breech lock closures are examples of nonintegral connections which are subject to failure by bell mouthing or other types of progressive deformation. If any combination of applied loads produces yielding, such joints are subject to ratcheting because the mating members may become loose at the end of each complete operating cycle and start the next cycle in a new relationship with each other, with or without manual manipulation. Additional distortion may occur in each cycle so that interlocking parts, such as threads, can eventually lose engagement. Therefore, primary plus secondary stress intensities (NB-3222.2), which result in slippage between the parts of a nonintegral connection in which disengagement could occur as a result of progressive distortion, shall be limited to the value S_y (Table I-2.1 or I-2.2).

NB-3227.4 Triaxial Stresses. The algebraic sum of the three primary principal stresses ($\sigma_1 + \sigma_2 + \sigma_3$) shall not exceed four times the tabulated value of S_m, except for Service Level D.

NB-3227.5 Nozzle Piping Transition. Within the limits of reinforcement given by NB-3334, whether or not nozzle reinforcement is provided, the P_m classification is applicable to stress intensities resulting from pressure-induced general membrane stresses as well as stresses other than discontinuity stresses due to external loads and moments including those attributable to restrained free end displacements of the attached pipe. Also, within the limits of reinforcement, a P_L classification shall be applied to local primary membrane stress intensities derived from discontinuity effects plus primary bending stress intensities due to combined pressure and external loads and moments, including those attributable to restrained free end displacements of the attached pipe; and a $P_L + P_b + Q$ classification shall apply to primary plus secondary stress intensities resulting from a combination of pressure, temperature, and external loads and moments, including those due to restrained free end displacements of the attached pipe. Beyond the limits of reinforcement, a P_m classification is applicable to stress intensities resulting from pressure-induced general membrane stresses as well as the average stress across the nozzle thickness due to externally applied nozzle axial, shear, and torsional loads other than those attributable to restrained free end displacement of the attached pipe. Also, outside the limits of reinforcement a $P_L + P_b$ classification is applicable to the stress intensities that result from adding those stresses classified as P_m to those due to externally applied bending moments, except those attributable to restrained free end displacement of the pipe. Further, beyond the limits of reinforcement, a $P_L + P_b + Q$ classification is applicable to stress intensities resulting from all pressure, temperature, and external loads and moments, including those attributable to restrained free end displacements of the attached pipe. Beyond the limits of reinforcement, the $3S_m$ limit on the range of primary plus secondary stress intensity may be exceeded as provided in NB-3228.3, except that in the evaluation of NB-3228.3(a) stresses from restrained free end displacements of the attached pipe may also be excluded. The range of membrane plus bending stress intensity attributable solely to the restrained free end displacements of the attached pipe shall be $\leq 3S_m$. The nozzle, outside the reinforcement limit, shall not be thinner than the larger of the pipe thickness or the quantity $t_p(S_{mp}/S_{mn})$, where t_p is the nominal thickness of the mating pipe, S_{mp} is the allowable stress intensity value for the pipe material, and S_{mn} is the allowable stress intensity value for the nozzle material.

NB-3227.6 Applications of Elastic Analysis for Stresses Beyond the Yield Strength. Certain of the allowable stresses permitted in the design criteria are such that the maximum stress calculated on an elastic basis may exceed the yield strength of the material. The limit on primary plus secondary stress intensity of $3S_m$ (NB-3222.2) has been placed at a level which ensures shakedown to elastic action after a few repetitions of the stress cycle except in regions containing significant local structural discontinuities or local thermal stresses. These last two factors are considered only in the performance of a fatigue evaluation. Therefore:

(a) In evaluating stresses for comparison with the stress limits on other than fatigue allowables, stresses shall be calculated on an elastic basis.

(b) In evaluating stresses for comparison with fatigue allowables, all stresses except those which result from local thermal stresses (II-1124) shall be evaluated on an elastic basis. In evaluating local thermal stresses, the elastic equations shall be used, except that the numerical value substituted for Poisson's ratio shall be determined from the expression:

$$\nu = 0.5 - 0.2\,(S_y/S_a), \text{ but not less than } 0.3$$

where

S_y = yield strength of the material at the mean value of the temperature of the cycle

S_a = value obtained from the applicable design fatigue curve (Figs. I-9.0) for the specified number of cycles of the condition being considered

NB-3227.7 Requirements for Specially Designed Welded Seals

(a) Welded seals, such as omega and canopy seals (NB-4360), shall be designed to meet the pressure induced general primary membrane stress intensity limits specified in this Subsection. Note that the general primary membrane stress intensity varies around the toroidal cross section.

(b) All other membrane and bending stress intensities developed in the welded seals may be considered

as secondary stress intensities. The range of these stress intensities combined with the general primary membrane stress intensity may exceed the primary plus secondary stress intensity limit of $3S_m$, if they are analyzed in accordance with NB-3228.5 as modified in (1) and (2) below.

(1) In lieu of NB-3228.5(a), the range of the combined primary plus secondary membrane stress intensities shall be $\leq 3S_m$.

(2) NB-3228.5(d) need not apply.

NB-3228 Applications of Plastic Analysis

The following subparagraphs provide guidance in the application of plastic analysis and some relaxation of the basic stress limits which are allowed if plastic analysis is used.

NB-3228.1 Limit Analysis. The limits on General Membrane Stress Intensity (NB-3221.1), Local Membrane Stress Intensity (NB-3221.2), and Primary Membrane Plus Primary Bending Stress Intensity (NB-3221.3) need not be satisfied at a specific location if it can be shown by limit analysis that the specified loadings do not exceed two-thirds of the lower bound collapse load. The yield strength to be used in these calculations is $1.5S_m$. The use of $1.5S_m$ for the yield strength of those materials of Table I-1.2 to which Note (2) of the Table is applicable may result in small permanent strains during the first few cycles of loading. If these strains are not acceptable, the yield strength to be used shall be reduced according to the strain limiting factors of Table I-2.4. When two-thirds of the lower bound collapse load is used, the effects of plastic strain concentrations in localized areas of the structure such as the points where hinges form must be considered. The effects of these concentrations of strain on the fatigue behavior, ratcheting behavior, or buckling behavior of the structure must be considered in the design. The design shall satisfy the minimum wall thickness requirements.

NB-3228.2 Experimental Analysis. The limits of General Primary Membrane Stress Intensity (NB-3221.1), Local Membrane Stress Intensity (NB-3221.2), and Primary Membrane Plus Primary Bending Stress Intensity (NB-3221.3) need not be satisfied at a specific location if it can be shown that the specified loadings do not exceed two-thirds of the test collapse load determined by application of II-1430, in which case the effects of plastic strain concentrations in localized areas of the structure, such as the points where hinges form, must be considered. The effects of these concentrations of strain on the fatigue behavior, ratcheting behavior, or buckling behavior of the structure must be considered in the design. The design shall satisfy the minimum wall thickness requirements.

NB-3228.3 Plastic Analysis. Plastic analysis is a method of structural analysis by which the structural behavior under given loads is computed by considering the actual material stress–strain relationship and stress redistribution, and it may include either strain hardening or change in geometry, or both.

The limits of General Membrane Stress Intensity (NB-3221.1), Local Membrane Stress Intensity (NB-3221.2), and Primary Membrane Plus Primary Bending Stress Intensity (NB-3221.3) need not be satisfied at a specific location if it can be shown that the specified loadings do not exceed two-thirds of the plastic analysis collapse load determined by application of II-1430 to a load–deflection or load–strain relationship obtained by plastic analysis. When this rule is used, the effects of plastic strain concentrations in localized areas of the structure, such as the points where hinges form, must be considered. The effects of the concentrations of strain on the fatigue behavior, ratcheting behavior, or buckling behavior of the structure must be considered in the design. The design shall satisfy the minimum wall thickness requirements.

NB-3228.4 Shakedown Analysis. The limits on Thermal Stress Ratchet in Shell (NB-3222.5) and Progressive Distortion of Non-Integral Connections (NB-3227.3) need not be satisfied at a specific location, if, at the location, the procedures of (a) through (c) below are used.

(a) In evaluating stresses for comparison with the remaining stress limits, the stresses shall be calculated on an elastic basis.

(b) In lieu of satisfying the specific requirements of NB-3221.2, NB-3222.2, NB-3222.5, and NB-3227.3 at a specific location, the structural action shall be calculated on a plastic basis, and the design shall be considered to be acceptable if shakedown occurs (as opposed to continuing deformation), and if the deformations which occur prior to shakedown do not exceed specified limits.

(c) In evaluating stresses for comparison with fatigue allowables, the numerically maximum principal total strain range which occurs after shakedown shall be multiplied by one-half the modulus of elasticity of the material (Tables I-6.0) at the mean value of the temperature of the cycle.

NB-3228.5 Simplified Elastic–Plastic Analysis. The $3S_m$ limit on the range of primary plus secondary stress intensity (NB-3222.2) may be exceeded provided that the requirements of (a) through (f) below are met.

TABLE NB-3228.5(b)-1
VALUES OF m, n, AND T_{max} FOR VARIOUS CLASSES OF PERMITTED MATERIALS

Materials	m	n	T_{max}, °F
Carbon steel	3.0	0.2	700
Low alloy steel	2.0	0.2	700
Martensitic stainless steel	2.0	0.2	700
Austenitic stainless steel	1.7	0.3	800
Nickel–chromium–iron	1.7	0.3	800
Nickel–copper	1.7	0.3	800

(a) The range of primary plus secondary membrane plus bending stress intensity, excluding thermal bending stresses, shall be $\leq 3S_m$.

(b) The value of S_a used for entering the design fatigue curve is multiplied by the factor K_e, where:

$$K_e = 1.0, \text{ for } S_n \leq 3S_m$$

$$= 1.0 + [(1 - n)/n(m - 1)](S_n/3S_m - 1), \text{ for } 3S_m < S_n < 3mS_m$$

$$= 1/n, \text{ for } S_n \geq 3mS_m$$

S_n = range of primary plus secondary stress intensity, psi

The values of the material parameters m and n for the various classes of permitted materials are as given in Table NB-3228.5(b)-1.

(c) The rest of the fatigue evaluation stays the same as required in NB-3222.4, except that the procedure of NB-3227.6 need not be used.

(d) The component meets the thermal ratcheting requirement of NB-3222.5.

(e) The temperature does not exceed those listed in Table NB-3228.5(b)-1 for the various classes of materials.

(f) The material shall have a specified minimum yield strength to specified minimum tensile strength ratio of less than 0.80.

NB-3229 Design Stress Values

The design stress intensity values S_m are given in Tables I-1.1 and I-1.2 for component materials. Values for intermediate temperatures may be found by interpolation. These form the basis for the various stress limits. Values of yield strength are given in Tables I-2.1 and I-2.2. Values of the coefficient of thermal expansion are in Table I-5.0 and values of the modulus of elasticity are in Table I-6.0. The basis for establishing stress values is given in Appendix III. The design fatigue curves used in conjunction with NB-3222.4 are those of Figs. I-9.0.

NB-3230 STRESS LIMITS FOR BOLTS

NB-3231 Design Conditions

(a) The number and cross-sectional area of bolts required to resist the Design Pressure shall be determined in accordance with the procedures of Appendix E, using the larger of the bolt loads, given by the equations of Appendix E, as a Design Mechanical Load. The allowable bolt design stresses shall be the values given in Table I-1.3 for bolting material.

(b) When sealing is effected by a seal weld instead of a gasket, the gasket factor m and the minimum design seating stress y may be taken as zero.

(c) When gaskets are used for preservice testing only, the design is satisfactory if the above requirements are satisfied for $m=y=0$, and the requirements of NB-3232 are satisfied when the appropriate m and y factors are used for the test gasket.

NB-3232 Level A Service Limits

Actual service stresses in bolts, such as those produced by the combination of preload, pressure, and differential thermal expansion, may be higher than the values given in Table I-1.3.

NB-3232.1 Average Stress. The maximum value of service stress, averaged across the bolt cross section and neglecting stress concentrations, shall not exceed two times the stress values of Table I-1.3.

NB-3232.2 Maximum Stress. The maximum value of service stress, except as restricted by NB-3232.3(b), at the periphery of the bolt cross section resulting from direct tension plus bending and neglecting stress concentrations shall not exceed three times the stress values of Table I-1.3. Stress intensity, rather than maximum stress, shall be limited to this value when the bolts are tightened by methods other than heaters, stretchers, or other means which minimize residual torsion.

NB-3232.3 Fatigue Analysis of Bolts. Unless the components on which they are installed meet all the conditions of NB-3222.4(d) and thus require no fatigue analysis, the suitability of bolts for cyclic service shall be determined in accordance with the procedures of (a) through (e) below.

(a) Bolting Having Less Than 100.0 ksi Tensile Strength. Bolts made of material which has specified minimum tensile strength of less than 100.0 ksi shall be evaluated for cyclic service by the methods of NB-3222.4(e), using the applicable design fatigue curve of Figs. I-9.0 and an appropriate fatigue strength reduction factor [NB-3232.3(c)].

(b) High Strength Alloy Steel Bolting. High strength alloy steel bolts and studs may be evaluated for cyclic service by the methods of NB-3222.4(e) using the design fatigue curve of Fig. I-9.4 provided:

(1) the maximum value of the service stress (NB-3232.2) at the periphery of the bolt cross section, resulting from direct tension plus bending and neglecting stress concentration, shall not exceed $2.7S_m$ if the higher of the two fatigue design curves given in Fig. I-9.4 is used. The $2S_m$ limit for direct tension is unchanged.

(2) threads shall be of a Vee-type having a minimum thread root radius no smaller than 0.003 in.;

(3) fillet radii at the end of the shank shall be such that the ratio of fillet radius to shank diameter is not less than 0.060.

(c) Fatigue Strength Reduction Factor (NB-3213.17). Unless it can be shown by analysis or tests that a lower value is appropriate, the fatigue strength reduction factor used in the fatigue evaluation of threaded members shall not be less than 4.0. However, when applying the rules of NB-3232.3(b) for high strength alloy steel bolts, the value used shall not be less than 4.0.

(d) Effect of Elastic Modulus. Multiply S_{alt} (as determined in NB-3216.1 or NB-3216.2) by the ratio of the modulus of elasticity given on the design fatigue curve to the value of the modulus of elasticity used in the analysis. Enter the applicable design fatigue curve at this value on the ordinate axis and find the corresponding number of cycles on the abscissa. If the service cycle being considered is the only one which produces significant fluctuating stresses, this is the allowable number of cycles.

(e) Cumulative Damage. The bolts shall be acceptable for the specified cyclic application of loads and thermal stresses provided the cumulative usage factor U, as determined in NB-3222.4(e)(5), does not exceed 1.0.

NB-3233 Level B Service Limits

Level A Service Limits (NB-3232) apply.

NB-3234 Level C Service Limits

The stress limits of NB-3232.1 and NB-3232.2 apply.

NB-3235 Level D Service Limits

If the Design Specifications specify any Service Loadings for which Level D Limits are designated [NCA-2142.4(b)(4)], the rules contained in Appendix F may be used in evaluating these loadings independently of all other Design and Service Loadings.

NB-3236 Design Stress Intensity Values

The design stress intensity values S_m are given in Table I-1.3 for bolting. Values for intermediate temperature may be found by interpolation. The basis for establishing stress intensity values is given in Appendix III.

NB-3300 VESSEL DESIGN

NB-3310 GENERAL REQUIREMENTS

NB-3311 Acceptability

The requirements for acceptability of a vessel design are as follows.

(a) The design shall be such that the requirements of NB-3100 and NB-3200 shall be satisfied.

(b) The rules of this Subarticle shall be met. In cases of conflict between NB-3200 and NB-3300 the requirements of NB-3300 shall govern.

NB-3320 DESIGN CONSIDERATIONS

NB-3321 Design and Service Loadings

The provisions of NB-3110 apply.

NB-3322 Special Considerations

The provisions of NB-3120 apply.

NB-3323 General Design Rules

The provisions of NB-3130 apply except when they conflict with rules of this Subarticle. In case of conflict, this Subarticle governs in the design of vessels.

NB-3324 Tentative Pressure Thickness

The following formulas are given as an aid to the designer for determining a tentative thickness for use in the design. They are not to be construed as formulas for acceptable thicknesses. However, except in local regions (NB-3221.2), the wall thickness of a vessel shall never be less than that obtained from the formulas in NB-3324.1 and NB-3324.2, in which:

t = thickness of shell or head, in.
P = Design Pressure, psi
R = inside radius of shell or head, in.
R_o = outside radius of shell or head, in.
S_m = design stress intensity values (Tables I-1.0), psi

NB-3324.1 Cylindrical Shells

$$t = \frac{PR}{S_m - 0.5P} \quad \text{or} \quad t = \frac{PR_o}{S_m + 0.5P}$$

NB-3324.2 Spherical Shells

$$t = \frac{PR}{2S_m - P} \quad \text{or} \quad t = \frac{PR_o}{2S_m}$$

NB-3330 OPENINGS AND REINFORCEMENT

NB-3331 General Requirements for Openings

(a) For vessels or parts thereof which meet the requirements of NB-3222.4(d), analysis showing satisfaction of the requirements of NB-3221.1, NB-3221.2, NB-3221.3, and NB-3222.2 in the immediate vicinity of the openings is not required for pressure loading if the rules of NB-3330 are met.

(b) For vessels or parts thereof that do not meet the requirements of NB-3222.4(d) so that a fatigue analysis is required, the rules contained in NB-3330 ensure satisfaction of the requirements of NB-3221.1, NB-3221.2, and NB-3221.3 in the vicinity of openings, and no specific analysis showing satisfaction of those stress limits is required for pressure loading. The requirements of NB-3222.2 may also be considered to be satisfied if, in the vicinity of the nozzle, the stress intensity resulting from external nozzle loads and thermal effects, including gross but not local structural discontinuities, is shown by analysis to be less than $1.5S_m$. In this case, when evaluating the requirements of NB-3222.4(e), the peak stress intensity resulting from pressure loadings may be obtained by application of the stress index method of NB-3338 or NB-3339.

(c) The provisions of (a) and (b) above are not intended to restrict the design to any specified section thicknesses or other design details, provided the basic stress limits are satisfied. If it is shown by analysis that all the stress requirements have been met, the rules of NB-3330 are waived.

(d) Openings shall be circular, elliptical, or of any other shape which results from the intersection of a circular or elliptical cylinder with a vessel of the shapes permitted by this Subsection. Additional restrictions given in NB-3338.2(d) are applicable if the Stress Index Method is used. If fatigue analysis is not required, the restrictions on hole spacing are applicable unless there will be essentially no pipe reactions.

(e) Openings are not limited as to size except to the extent provided in NB-3338.2(d).

(f) All references to dimensions apply to the finished dimensions excluding material added as corrosion allowance. Rules regarding metal available for reinforcement are given in NB-3335.

(g) Any type of opening permitted in these rules may be located in a welded joint.

NB-3332 Reinforcement Requirements for Openings in Shells and Formed Heads

NB-3332.1 Openings Not Requiring Reinforcement. The rules for openings not requiring reinforcement are given in (a) through (c) below, where R is the mean radius and t is the nominal thickness of the vessel shell or head at the location of the opening; and *locally stressed area* means any area in the shell where the primary local membrane stress exceeds $1.1S_m$, but excluding those areas where such primary local membrane stress is due to an unreinforced opening.

(a) A single opening has a diameter not exceeding $0.2\sqrt{Rt}$, or if there are two or more openings within any circle of diameter $2.5\sqrt{Rt}$, but the sum of the diameters of such unreinforced openings shall not exceed $0.25\sqrt{Rt}$.

(b) No two unreinforced openings shall have their centers closer to each other, measured on the inside of the vessel wall, than 1.5 times the sum of their diameters.

(c) No unreinforced opening shall have its center closer than $2.5\sqrt{Rt}$ to the edge of a locally stressed area in the shell.

NB-3332.2 Required Area of Reinforcement. The total cross-sectional area of reinforcement A, required in any given plane for a vessel under internal pressure, shall not be less than:

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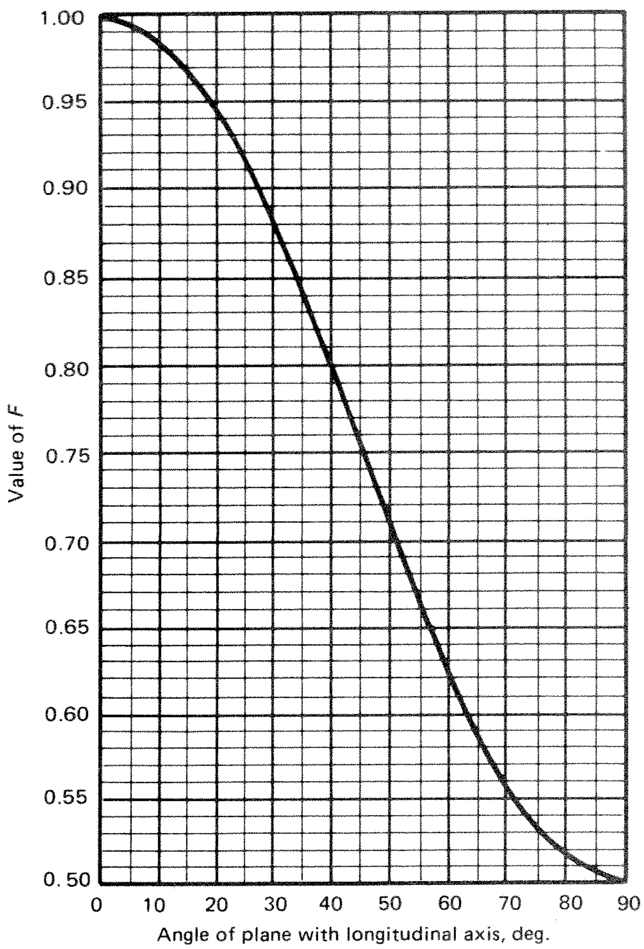


FIG. NB-3332.2-1 CHART FOR DETERMINING VALUE OF F

$$A = dt_rF$$

where

d = finished diameter of a circular opening or finished dimension (chord length) of an opening on the plane being considered for elliptical and obround openings in corroded condition, in.

F = a correction factor which compensates for the variation in pressure stresses on different planes with respect to the axis of a vessel. (A value of 1.00 shall be used for all configurations, except that Fig. NB-3332.2-1 may be used for integrally reinforced openings in cylindrical shells and cones.)

t_r = the thickness which meets the requirements of NB-3221.1 in the absence of the opening, in.

Not less than half the required material shall be on each side of the center line.

NB-3333 Reinforcement Required for Openings in Flat Heads

Flat heads that have an opening with a diameter that does not exceed one-half the head diameter shall have a total cross-sectional area of reinforcement not less than that given by the formula:

$$A = 0.5dt_r$$

where d is as defined in NB-3332 and t_r is the thickness, in., which meets the requirements of NB-3221.1 and NB-3221.3 in the absence of the opening.

NB-3334 Limits of Reinforcement

The boundaries of the cross-sectional area in any plane normal to the vessel wall and passing through the center of the opening and within which metal shall be located in order to have value as reinforcement are designated as the limits of reinforcement for that plane and are given in the following subparagraphs.

NB-3334.1 Limit of Reinforcement Along the Vessel Wall. The limits of reinforcement, measured along the midsurface of the nominal wall thickness, shall meet the following.

(a) One hundred percent of the required reinforcement shall be within a distance on each side of the axis of the opening equal to the greater of the following:

(1) the diameter of the finished opening in the corroded condition;

(2) the radius of the finished opening in the corroded condition plus the sum of the thicknesses of the vessel wall and the nozzle wall.

(b) Two-thirds of the required reinforcement shall be within a distance on each side of the axis of the opening equal to the greater of the following:

(1) $r + 0.5\sqrt{Rt}$, where R is the mean radius of shell or head, t is the nominal vessel wall thickness, and r is the radius of the finished opening in the corroded condition;

(2) the radius of the finished opening in the corroded condition plus two-thirds the sum of the thicknesses of the vessel wall and the nozzle wall.

NB-3334.2 Limit of Reinforcement Normal to the Vessel Wall. The limits of reinforcement, measured normal to the vessel wall, shall conform to the contour of the surface at a distance from each surface equal to the following limits as shown in Fig. NB-3338.2(a)-2.

(a) For Fig. NB-3338.2(a)-2 sketches (a), (b), (d), and (e):

$$\text{Limit} = 0.5\sqrt{r_m t_n} + 0.5 r_2$$

where

r_m = mean radius, in.
$= r_i + 0.5t_n$
r_i = inside radius, in.
t_n = nominal nozzle thickness, as indicated, in.
r_2 = transition radius, in., between nozzle and wall

For the case of a nozzle with a tapered inside diameter, the limit shall be obtained by using r_i and t_n values at the nominal outside diameters of the vessel wall [Fig. NB-3338.2(a)-2 sketch (e)].

(b) For Fig. NB-3338.2(a)-2 sketches (c) and (f):

$$\text{Limit} = 0.5\sqrt{r_m t_n}$$

where

$r_m = r_i + 0.5t_n$, in.
r_i = inside radius, in.
$t_n = t_p + 0.667X$
t_p = nominal thickness of the attached pipe, in.
X = slope offset distance, in.
θ = angle between vertical and slope, 45 deg. or less

For the case of a nozzle with a tapered inside diameter, the limit shall be obtained by using r_i and t_n values at the center of gravity of nozzle reinforcement area. These values must be determined by a trial and error procedure [Fig. NB-3338.2(a)-2 sketch (f)].

NB-3335 Metal Available for Reinforcement

Metal may be counted as contributing to the area of reinforcing called for in NB-3332, provided it lies within the limits of reinforcement specified in NB-3334, and shall be limited to material which meets the following requirements:

(a) metal forming a part of the vessel wall which is in excess of that required on the basis of membrane stress intensity (NB-3221.1) and is exclusive of corrosion allowance;

(b) similar excess metal in the nozzle wall, provided the nozzle is integral with the vessel wall or is joined to it by a full penetration weld;

(c) weld metal which is fully continuous with the vessel wall;

(d) the mean coefficient of thermal expansion of metal to be included as reinforcement under (b) and (c) above shall be within 15% of the value of the vessel wall material;

(e) metal not fully continuous with the shell, such as that in nozzles attached by partial penetration welds, shall not be counted as reinforcement;

(f) metal available for reinforcement shall not be considered as applying to more than one opening.

NB-3336 Strength of Reinforcing Material

Material used for reinforcement shall preferably be the same as that of the vessel wall. If the material of the nozzle wall or reinforcement has a lower design stress intensity value S_m than that for the vessel material, the amount of area provided by the nozzle wall or reinforcement in satisfying the requirements of NB-3332 shall be taken as the actual area provided multiplied by the ratio of the nozzle or reinforcement design stress intensity value to the vessel material design stress intensity value. No reduction in the reinforcing required may be taken for the increased strength of reinforcing material and weld metal having higher design stress intensity values than that of the material of the vessel wall. The strength of the material at the point under consideration shall be used in fatigue analyses.

NB-3337 Attachment of Nozzles and Other Connections

NB-3337.1 General Requirements. Nozzles and other Category D connections (NB-3351) shall be attached to the shell or head of the vessel by one of the methods provided in NB-3352.

NB-3337.2 Full Penetration Welded Nozzles. Full penetration welds, as shown in Figs. NB-4244(a)-1, NB-4244(b)-1, NB-4244(c)-1, and NB-4244(e)-1 may be used (except as otherwise provided in NB-3337.3) for the purpose of achieving continuity of metal and facilitating the required radiographic examination. When all or part of the required reinforcement is attributable to the nozzle, the nozzle shall be attached by full penetration welds through either the vessel or the nozzle thickness, or both.

NB-3337.3 Partial Penetration Welded Nozzles

(a) Partial penetration welds, as shown in Figs. NB-4244(d)-1 and NB-4244(d)-2, are allowed only for nozzles on which there are substantially no piping reactions, such as control rod housings, pressurizer heater wells, and openings for instrumentation. Earthquake loadings need not be considered in determining wheth-

er piping reactions are substantial. For such nozzles, all reinforcement shall be integral with the portion of the vessel penetrated. Partial penetration welds shall be of sufficient size to develop the full strength of the nozzles. Nozzles attached by partial penetration welds shall have an interference fit or a maximum diametral clearance between the nozzle and the vessel penetration of:

(1) 0.010 in. for $d \leq$ 1 in.
(2) 0.020 in. for 1 in. $< d \leq$ 4 in.
(3) 0.030 in. for $d >$ 4 in.

where d is the outside diameter of the nozzle, except that the above limits on maximum clearance need not be met for the full length of the opening, provided there is a region at the weld preparation and a region near the end of the opening opposite the weld that does meet the above limits on maximum clearance and the latter region is extensive enough (not necessarily continuous) to provide a positive stop for nozzle deflection.

(b) In satisfying the limit of NB-3222.2, the stress intensities resulting from pressure induced strains (dilation of hole) may be treated as secondary in the penetrating part of partial penetration welded construction, provided the requirements of NB-3352.4(d) and Fig. NB-4244(d)-1 are fulfilled.

NB-3338 Fatigue Evaluation of Stresses in Openings

NB-3338.1 General. For the purpose of determining peak stresses around the opening, three acceptable methods are listed below.

(a) Analytical Method. This method uses suitable analytical techniques such as finite element computer analyses, which provide detailed stress distributions around openings. In addition to peak stresses due to pressure, the effects of other loadings shall be included. The total peak stress at any given point shall be determined by combining stresses due to pressure, thermal, and external loadings in accordance with the rules of NB-3200.

(b) Experimental Stress Analysis. This is based on data from experiments (Appendix II).

(c) Stress Index Method. This uses various formulas together with available data obtained from an extensive series of tests covering a range of variation of applicable dimensional ratios and configurations (NB-3338.2). This method covers only single, isolated openings. Stress indices may also be determined by theoretical or experimental stress analysis.

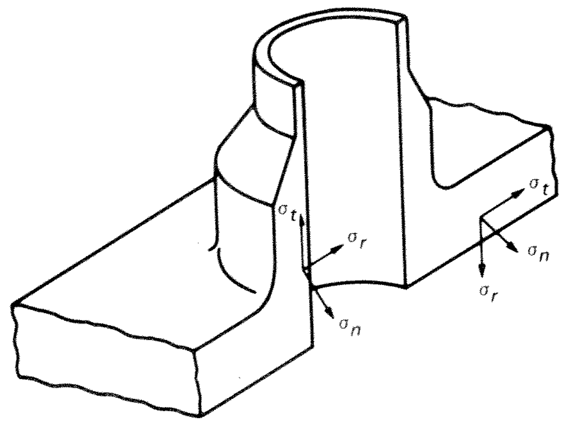


FIG. NB-3338.2(a)-1 DIRECTION OF STRESS COMPONENTS

NB-3338.2 Stress Index Method

(a) The term *stress index*, as used herein, is defined as the numerical ratio of the stress components σ_t, σ_n, and σ_r [Fig. NB-3338.2(a)-1] under consideration to the computed membrane hoop stress in the unpenetrated vessel material; however, the material which increases the thickness of a vessel wall locally at the nozzle shall not be included in the calculations of these stress components. When the thickness of the vessel wall is increased over that required to the extent provided hereinafter, the values of r_1 and r_2 in Fig. NB-3338.2(a)-2 shall be referred to the thickened section.

(b) The nomenclature used in NB-3338 is defined as follows.

σ_t = stress component in the plane of the section under consideration and parallel to the boundary of the section, psi
σ_n = stress component normal to the plane of the section (ordinarily the circumferential stress around the hole in the shell), psi
σ_r = stress component normal to the boundary of the section, psi
R = inside radius, in corroded condition, of cylindrical vessel, spherical vessel, or spherical head, in.
S = stress intensity (combined stress) at the point under consideration, psi
t = nominal wall thickness, less corrosion allowance, of vessel or head, in.

(c) When the conditions of (d) below are satisfied, the stress indices of Table NB-3338.2(c)-1 may be used for nozzles designed in accordance with the applicable rules of NB-3330. These stress indices deal only with the maximum stresses, at certain general locations, due

TABLE NB-3338.2(c)-1
STRESS INDICES FOR NOZZLES

Nozzles in Spherical Shells and Formed Heads		
Stress	Inside Corner	Outside Corner
σ_n	2.0	2.0
σ_t	−0.2	2.0
σ_r	$-2t/R$	0
S	2.2	2.0

Nozzles in Cylindrical Shells				
	Longitudinal Plane		Transverse Plane	
Stress	Inside	Outside	Inside	Outside
σ_n	3.1	1.2	1.0	2.1
σ_t	−0.2	1.0	−0.2	2.6
σ_r	$-t/R$	0	$-t/R$	0
S	3.3	1.2	1.2	2.6

to internal pressure. In the evaluation of stresses in or adjacent to vessel openings and connections, it is often necessary to consider the effect of stresses due to external loadings or thermal stresses. In such cases, the total stress at a given point may be determined by superposition. In the case of combined stresses due to internal pressure and nozzle loading, the maximum stresses for a given location shall be considered as acting at the same point and added algebraically unless positive evidence is available to the contrary.

(d) The indices of Table NB-3338.2(c)-1 apply when the conditions stipulated in (1) through (7) below exist.

(1) The opening is for a circular nozzle whose axis is normal to the vessel wall. If the axis of the nozzle makes an angle ϕ with the normal to the vessel wall and if $d/D \leq 0.15$, an estimate of the σ_n index on the inside may be obtained from one of the following formulas.

For hillside connections in spheres or cylinders:

$$K_2 = K_1\,(1 + 2\sin^2\phi)$$

For lateral connections in cylinders:

$$K_2 = K_1\,[1 + (\tan\phi)^{4/3}]$$

where

K_1 = the σ_n inside stress index of Table NB-3338.2(c)-1 for a radial connection

K_2 = the estimated σ_n inside stress index for the nonradial connection

(2) The arc distance measured between the center lines of adjacent nozzles along the inside surface of the shell is not less than three times the sum of their inside radii for openings in a head or along the longitudinal axis of a shell and is not less than two times the sum of their radii for openings along the circumference of a cylindrical shell. When two nozzles in a cylindrical shell are neither in a longitudinal line nor in a circumferential arc, their center line distance along the inside surface of the shell shall be such that $[(L_c/2)^2 + (L_l/3)^2]^{1/2}$ is not less than the sum of their inside radii, where L_c is the component of the center line distance in the circumferential direction and L_l is the component of the center line distance in the longitudinal direction.

(3) The dimensional ratios are not greater than the following:

Ratio	Cylinder	Sphere
D/t	100	100
d/D	0.50	0.50
$d/\sqrt{Dt}$	0.80	0.80

where D is the inside shell diameter, t is the shell thickness, and d is the inside nozzle diameter. In the case of cylindrical shells, the total nozzle reinforcement area on the transverse axis of the connections, including any outside of the reinforcement limits, shall not exceed 200% of that required for the longitudinal axis [compared to 50% permitted by Fig. NB-3332.2(a)-1] unless a tapered transition section is incorporated into the reinforcement and the shell, meeting the requirements of NB-3361.

(4) In the case of spherical shells and formed heads, at least 40% of the total nozzle reinforcement area shall be located beyond the outside surface of the minimum required vessel wall thickness.

(5) The inside corner radius r_1 [Fig. NB-3338.2(a)-2] is between 10% and 100% of the shell thickness t.

(6) The outer corner radius r_2 [Fig. NB-3338.2(a)-2] is large enough to provide a smooth transition between the nozzles and the shell. In addition, for opening diameters greater than $1\frac{1}{2}$ times the shell thickness in cylindrical shells and 2:1 ellipsoidal heads and greater than three shell thicknesses in spherical shells, the value of r_2 shall be not less than one-half the thickness of the shell or nozzle wall, whichever is greater.

(7) The radius r_3 [Fig. NB-3338.2(a)-2] is not less than the greater of the following:

(a) $0.002\theta d_o$, where d_o is the outside diameter of the nozzle and is as shown in Fig. NB-3338.2(a)-2, and the angle θ is expressed in degrees;

(b) $2(\sin\theta)^3$ times offset for the configuration shown in Fig. NB-3338.2(a)-2 sketches (a) and (b).

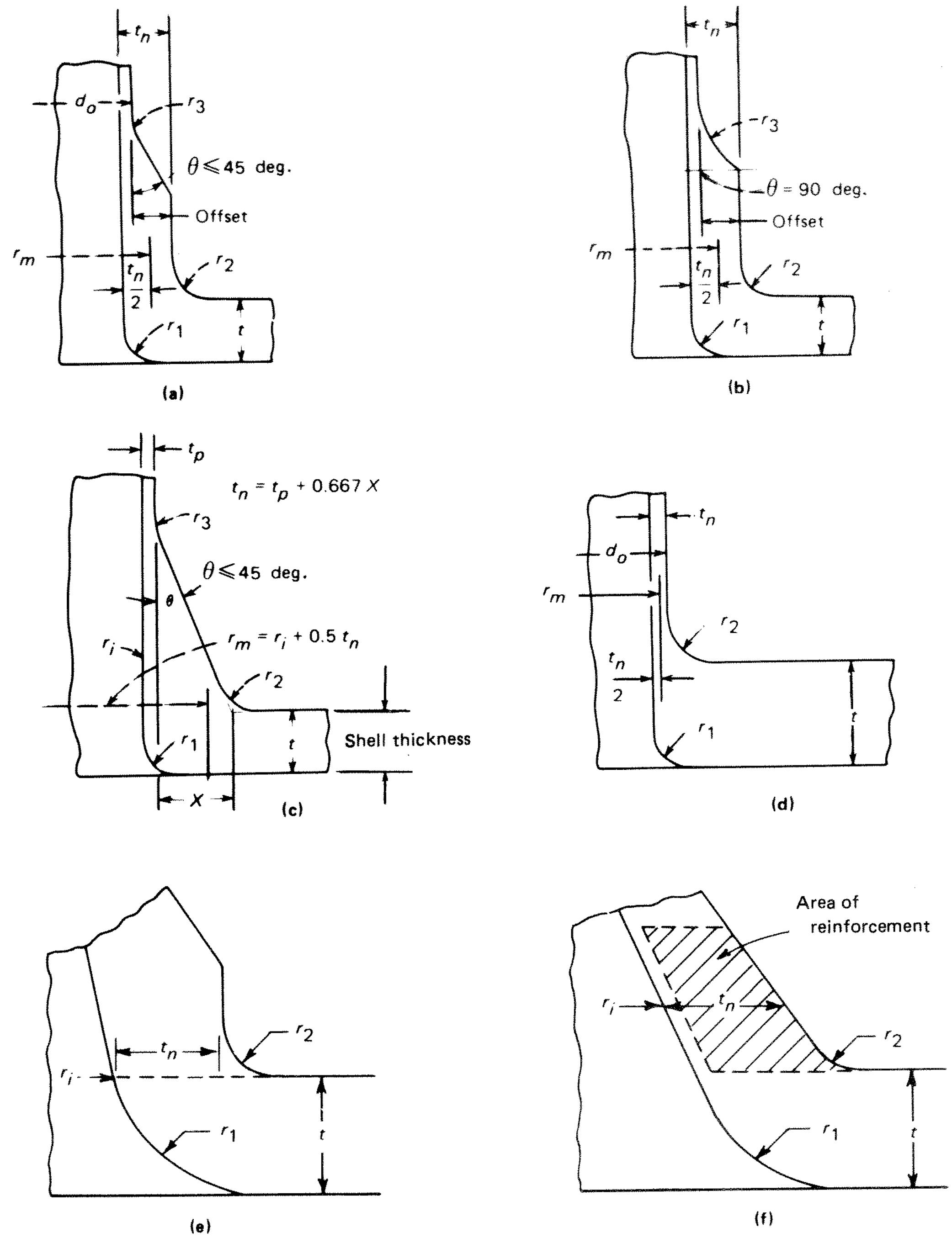
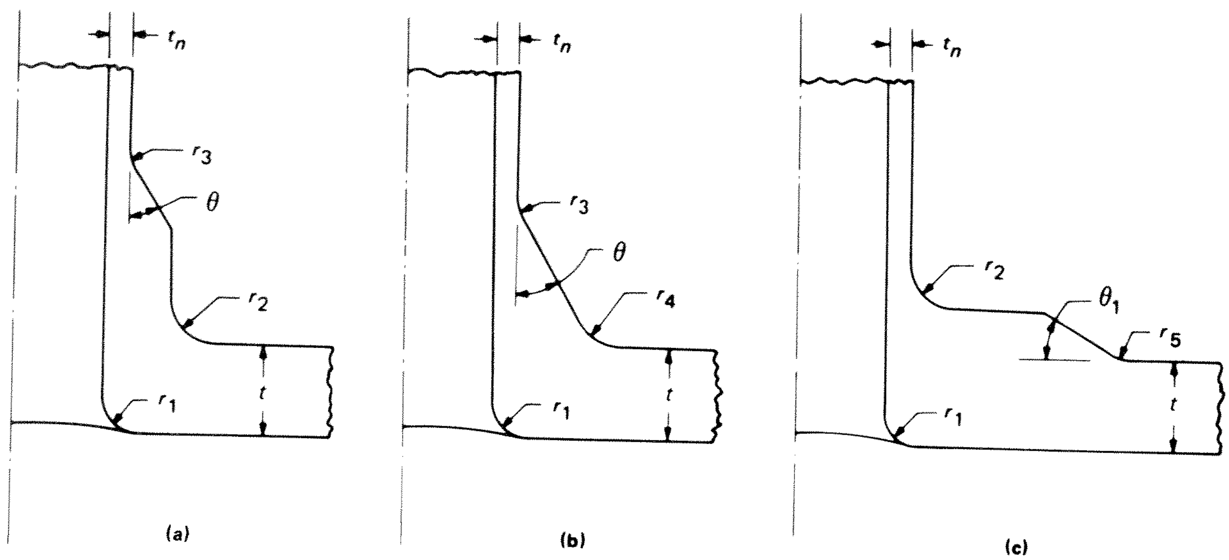


FIG. NB-3338.2(a)-2 NOZZLE DIMENSIONS



r_1 = 0.1t to 0.5t
$r_2 \geqslant$ larger of $\sqrt{dt_{rn}}$ or $t/2$
$r_3 \geqslant$ larger of $\sqrt{(\theta/90)\,(dt_{rn})}$ or $(\theta/90)t_n$
$r_4 \geqslant$ larger of $(1 - \sqrt{\theta/90})\sqrt{dt_{rn}}$ or (1-θ190) (t/2)
r_5 = $(\theta_1/90)t$
θ and θ_1 are in degrees

FIG. NB-3339.1(b)-1 EXAMPLES OF ACCEPTABLE TRANSITION DETAILS

NB-3339 Alternative Rules for Nozzle Design

Subject to the limitations stipulated in NB-3339.1, the requirements of this paragraph constitute an acceptable alternative to the rules of NB-3332 through NB-3336 and NB-3338.

NB-3339.1 Limitations. These alternative rules are applicable only to nozzles in vessels within the limitations stipulated in (a) through (f) below.

(a) The nozzle is circular in cross section and its axis is normal to the vessel or head surface.

(b) The nozzle and reinforcing (if required) are welded integrally into the vessel with full penetration welds. Details such as those shown in Figs. NB-4244(a)-1, NB-4244(b)-1, and NB-4244(c)-1 are acceptable. However, fillet welds shall be finished to a radius in accordance with Fig. NB-3339.1(b)-1.

(c) In the case of spherical shells and formed heads, at least 40% of the total nozzle reinforcement area shall be located beyond the outside surface of the minimum required vessel wall thickness.

(d) The spacing between the edge of the opening and the nearest edge of any other opening is not less than the smaller of $1.25(d_1 + d_2)$ or $2.5\sqrt{Rt}$, but in any case not less than $d_1 + d_2$, where d_1 and d_2 are the inside diameters of the openings.

(e) The material used in the nozzle, reinforcing, and vessel adjacent to the nozzle shall have a ratio of *UTS/YS* of not less than 1.5, where

UTS= specified minimum ultimate tensile strength, ksi

YS= specified minimum yield strength, ksi

(f) The following dimensional limitations are met:

	Nozzles in Cylindrical Vessels	Nozzles in Spherical Vessels or Hemispherical Heads
D/t	10 to 200	10 to 100
d/D	0.33 max.	0.5 max.
$d/\sqrt{Dt}$	0.8 max.	0.8 max.

NB-3339.2 Nomenclature. The nomenclature used in NB-3339 is defined as follows:

D= inside diameter, in corroded condition, of cylindrical vessel, spherical vessel, or spherical head, in.

d= inside diameter, in corroded condition, of the nozzle, in.

TABLE NB-3339.3-1
REQUIRED MINIMUM REINFORCING AREA A_r

	A_r, sq in.	
$d/\sqrt{Rt_r}$	Nozzles in Cylinders	Nozzles in Spherical Vessels or Heads
< 0.20	None [Note (1)]	None [Note (1)]
> 0.20 and < 0.40	$[4.05(d/\sqrt{Rt_r})^{1/2} - 1.81]dt_r$	$[5.40(d/\sqrt{Rt_r})^{1/2} - 2.41]dt_r$
> 0.40	$0.75dt_r$	$dt_r \cos \phi$ $\phi = \sin^{-1}(dt/D)$

NOTE:
(1) The transition radius r_2, shown in Fig. NB-3339.1(b)-1, or the equivalent thereof is required.

R = inside radius, in corroded condition, of cylindrical vessel, spherical vessel, or spherical head, in.
t_r = wall thickness of vessel or head, computed by the equations given in NB-3324.1 for cylindrical vessels and in NB-3324.2 for spherical vessels or spherical heads, in.
t_{rn} = wall thickness of nozzle, computed by the equation given in NB-3324.1, in.
t = nominal wall thickness, less corrosion allowance, of vessel or head, in.
t_n = nominal wall thickness, less corrosion allowance, of nozzle, in.
A_r = required minimum reinforcing area, sq in.
A_a = available reinforcing area, sq in.

For the definitions of r_1, r_2, r_3, r_4, θ, and θ_1 see Fig. NB-3339.1(a)-1; for L_c and L_n see Fig. NB-3339.4-1; for S, σ_t, σ_n, and σ_r see NB-3338.2 and Fig. NB-3338.2(a)-1.

NB-3339.3 Required Reinforcement Area. The required minimum reinforcing area is related to the value of $d/\sqrt{Rt_r}$ as tabulated in Table NB-3339.3-1. The required minimum reinforcing area shall be provided in all planes containing the nozzle axis.

NB-3339.4 Limits of Reinforcing Zone. Reinforcing metal included in meeting the minimum required reinforcing area specified in Table NB-3339.3-1 must be located within the reinforcing zone boundary shown in Fig. NB-3339.4-1.

NB-3339.5 Strength of Reinforcing Material Requirements. Material in the nozzle wall used for reinforcing shall preferably be the same as that of the vessel wall. If material with a lower design stress intensity value S_m is used, the area provided by such material shall be increased in proportion to the inverse ratio of the stress values of the nozzle and the vessel wall material. No reduction in the reinforcing area requirement shall be taken for the increased strength of nozzle material or weld metal which has a higher design stress intensity value than that of the material of the vessel wall. The strength of the material at the point under consideration shall be used in fatigue analyses. The mean coefficient of thermal expansion of metal to be included as reinforcement shall be within 15% of the value for the metal of the vessel wall.

NB-3339.6 Transition Details. Examples of acceptable transition tapers and radii are shown in Fig. NB-3339.1(b)-1. Other configurations which meet the reinforcing area requirements of NB-3339.3 and with equivalent or less severe transitions are also acceptable; e.g., larger radius–thickness ratios.

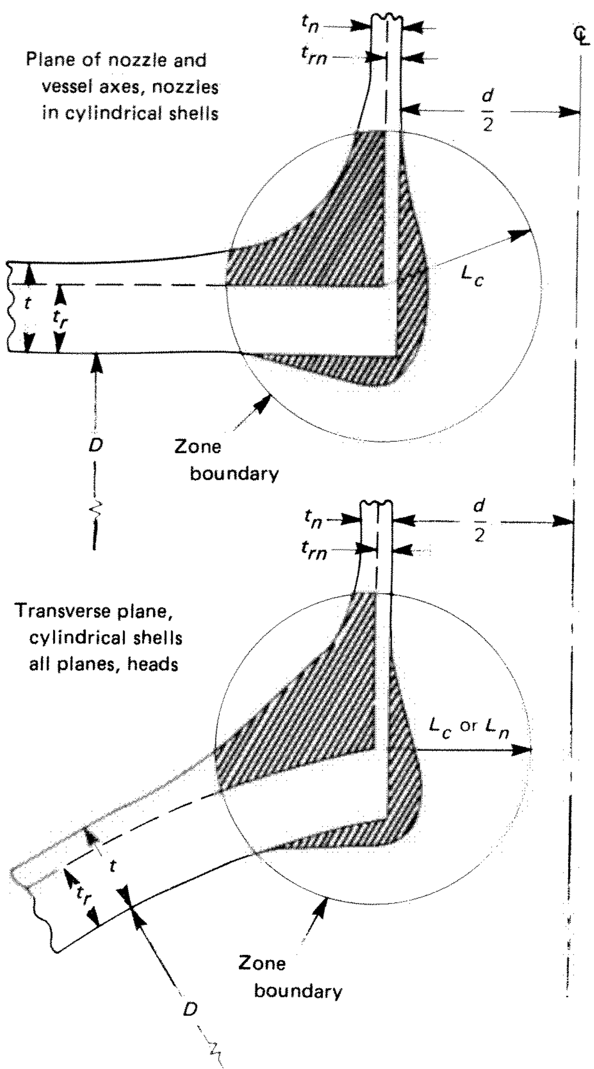
NB-3339.7 Stress Indices

(a) The term *stress index*, as used herein, is defined as the numerical ratio of the stress components σ_t, σ_n, and σ_r, under consideration, to the computed stress σ.

(b) The nomenclature for the stress components is shown in Fig. NB-3338.2(a)-1 and is defined as follows:

$\sigma = P(D + t)/4t$ for nozzles in spherical vessels or heads, psi
$= P(D + t)/2t$ for nozzles in cylindrical vessels, psi
σ_t = stress component in the plane of the section under consideration and parallel to the boundary of the section, psi
σ_n = stress component normal to the plane of the section (ordinarily the circumferential stress around the hole in the shell), psi
σ_r = stress component normal to the boundary of the section, psi
S = stress intensity (combined stress) at the point under consideration, psi
P = service pressure, psi

(c) When the conditions of NB-3339.1 through NB-3339.6 are satisfied, the stress indices given in Table NB-3339.7(c)-1 may be used. These stress indices deal only with the maximum stresses, at certain general locations, due to internal pressure. In the evaluation of stresses in or adjacent to vessel openings and connections, it is often necessary to consider the effect of stresses due to external loadings or thermal stresses. In such cases, the total stress at a given point may be determined by superposition. In the case of combined stresses due to internal pressure and nozzle loading,



GENERAL NOTES:

(a) Reinforcing Zone Limit

(1) For nozzles in cylindrical shells:

$$L_c = 0.75\,(t/D)^{2/3}D$$

(2) For nozzles in heads:

$$L_n = (t/D)^{2/3}(d/D + 0.5)D$$

(3) The center of L_c or L_n is at the juncture of the outside surfaces of the shell and nozzles of thickness t_r and t_{rn}.

(4) In constructions where the zone boundary passes through a uniform thickness wall segment, the zone boundary may be considered as L_c or L_n through the thickness.

(b) Reinforcing Area

(1) Hatched areas represent available reinforcement area A_a.

(2) Metal area within the zone boundary, in excess of the area formed by the intersection of the basic shells, shall be considered as contributing to the required area A_r. The basic shells are defined as having inside diameter D, thickness t_r, inside diameter of the nozzle d, and thickness t_{rn}.

(3) The available reinforcement ares A_a shall be at least equal to $A_r/2$ on each side of the nozzle center line and in every plane containing the nozzle axis.

FIG. NB-3339.4-1 LIMITS OF REINFORCING ZONE

TABLE NB-3339.7(c)-1
STRESS INDICES FOR INTERNAL PRESSURE LOADING

Nozzles in Spherical Shells and Spherical Heads

Stress	Inside	Outside
σ_n	$2.0 - d/D$	$2.0 - d/D$
σ_t	-0.2	$2.0 - d/D$
σ_r	$-4t/(D + t)$	0
S	larger of: $2.2 - d/D$ or $2.0 + [4t/(D + t)] - d/D$	$2.0 - d/D$

Nozzles in Cylindrical Shells

	Longitudinal Plane		Transverse Plane	
Stress	Inside	Outside	Inside	Outside
σ_n	3.1	1.2	1.0	2.1
σ_t	-0.2	1.0	-0.2	2.6
σ_r	$-2t/(D + t)$	0	$-2t/(D + t)$	0
S	3.3	1.2	1.2	2.6

the maximum stresses shall be considered as acting at the same point and added algebraically. If the stresses are otherwise determined by more accurate analytical techniques or by the experimental stress analysis procedure of Appendix II, the stresses are also to be added algebraically.

NB-3340 ANALYSIS OF VESSELS

The provisions of NB-3214 apply.

NB-3350 DESIGN OF WELDED CONSTRUCTION

NB-3351 Welded Joint Category

The term *Category* defines the location of a joint in a vessel, but not the type of joint. The categories established are for use in specifying special requirements regarding joint type and degree of examination for certain welded 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 apply to joints of a given category only when specifically stated. The joints included in each category are designated as joints of Categories A, B, C, and D. Figure NB-3351-1 illustrates typical joint locations included in each category.

NB-3351.1 Category A. Category A comprises longitudinal welded joints within the main shell, communicating chambers,[12] transitions in diameter, or nozzles; any welded joint within a sphere, within a formed or flat head, or within the side plates[13] of a flat sided vessel; and circumferential welded joints connecting hemispherical heads to main shells, to transitions in diameters, to nozzles, or to communicating chambers.

NB-3351.2 Category B. Category B comprises circumferential welded joints within the main shell, communicating chambers, nozzles, or transitions in diameter, including joints between the transition and a cylinder at either the large or small end; and circumferential welded joints connecting formed heads other than hemispherical to main shells, to transitions in diameter, to nozzles, or to communicating chambers.

NB-3351.3 Category C. Category C comprises 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;[12] any welded joint connecting one side plate[13] to another side plate of a flat sided vessel.

NB-3351.4 Category D. Category D comprises welded joints connecting communicating chambers 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. For nozzles at the small end of a transition in diameter, see Category B.

NB-3352 Permissible Types of Welded Joints

The design of the vessel shall meet the requirements for each category of joint. Butt joints are full penetration joints between plates or other elements that lie approximately in the same plane. Category B angle joints between plates or other elements that have an offset angle α not exceeding 30 deg. are considered as

[12]*Communicating chambers* are defined as portions of the vessel which intersect the shell or heads of a vessel and form an integral part of the pressure retaining closure, e.g., sumps.

[13]*Side plates of a flat sided vessel* are defined as any of the flat plates forming an integral part of the pressure retaining enclosure.

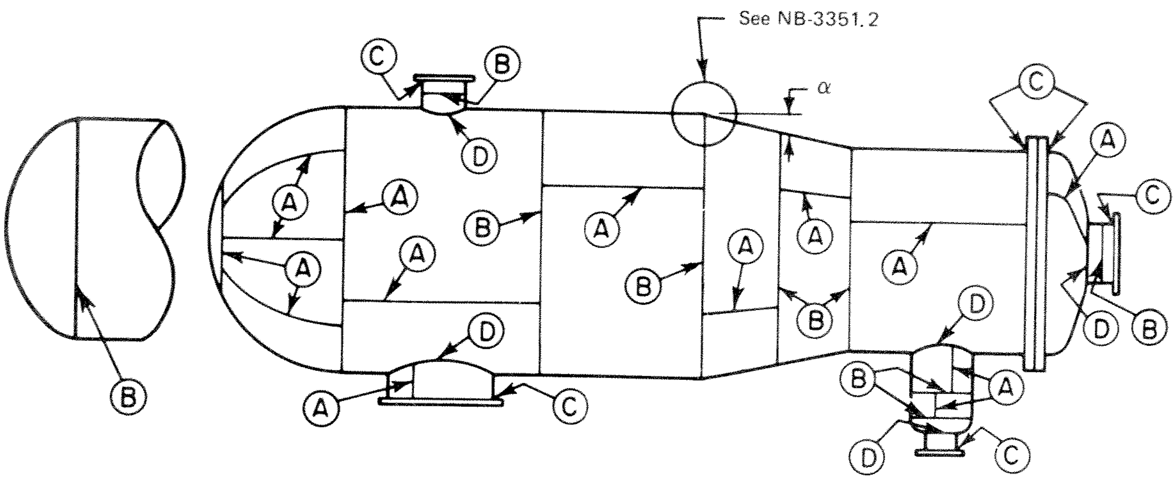


FIG. NB-3351-1 WELDED JOINT LOCATIONS TYPICAL OF CATEGORIES A, B, C, AND D

meeting the requirements for butt joints. Figure NB-3352-1 shows typical butt welds for each category joint.

NB-3352.1 Joints of Category A. All welded joints of Category A as defined in NB-3351 shall meet the fabrication requirements of NB-4241 and shall be capable of being examined in accordance with NB-5210.

NB-3352.2 Joints of Category B. All welded joints of Category B as defined in NB-3351 shall meet the fabrication requirements of NB-4242 and shall be capable of being examined in accordance with NB-5220. When joints with opposing lips to form an integral backing strip or joints with backing strips not later removed are used, the suitability for cyclic service shall be analyzed by the method of NB-3222.4 using a fatigue strength reduction factor of not less than 2.

NB-3352.3 Joints of Category C. All welded joints of Category C as defined in NB-3351 shall meet the fabrication requirements of NB-4243 and shall be capable of being examined in accordance with NB-5230. Minimum dimensions of the welds and throat thickness shall be as shown in Fig. NB-4243-1 where:

(a) for forged tubesheets, forged flat heads, and forged flanges with the weld preparation bevel angle not greater than 45 deg. measured from the face:

$t_c = 0.7t_n$ or $\frac{1}{4}$ in., whichever is less
$t_w = t_n/2$ or $t/4$, whichever is less
t, t_n = nominal thicknesses of welded parts, in.

(b) for all other material forms and for forged tubesheets, forged flat heads, and forged flanges with the weld preparation bevel angle greater than 45 deg. measured from the face:

$t_c = 0.7t_n$ or $\frac{1}{4}$ in., whichever is less
$t_w = t_n$ or $t/2$, whichever is less
t, t_n = nominal thicknesses of welded parts, in.

NB-3352.4 Joints of Category D. All welded joints of Category D as defined in NB-3351 shall be in accordance with the requirements of one of (a) through (e) below.

(a) Butt Welded Nozzles. Nozzles shall meet the fabrication requirements of NB-4244(a) and shall be capable of being examined in accordance with NB-5242. The minimum dimensions and geometrical requirements of Fig. NB-4244(a)-1 shall be met, where

t = nominal thickness of part penetrated, in.
t_n = nominal thickness of penetrating part, in.
$r_1 = \frac{1}{4}t$ or $\frac{3}{4}$ in., whichever is less
$r_2 = \frac{1}{4}$ in. minimum

(b) Full Penetration Corner Welded Nozzles. Nozzles shall meet the fabrication requirements of NB-4244(b) and shall be capable of being examined as required in NB-5243. The minimum dimensions of Fig. NB-4244(b)-1 shall be met, where

t = nominal thickness of part penetrated, in.
t_n = nominal thickness of penetrating part, in.
$t_c = 0.7t_n$ or $\frac{1}{4}$ in., whichever is less
$r_1 = \frac{1}{4}t$ or $\frac{3}{4}$ in., whichever is less
$r_2 = \frac{1}{4}$ in. minimum

(c) Use of Deposited Weld Metal for Openings and Nozzles

(1) Nozzles shall meet the fabrication require-

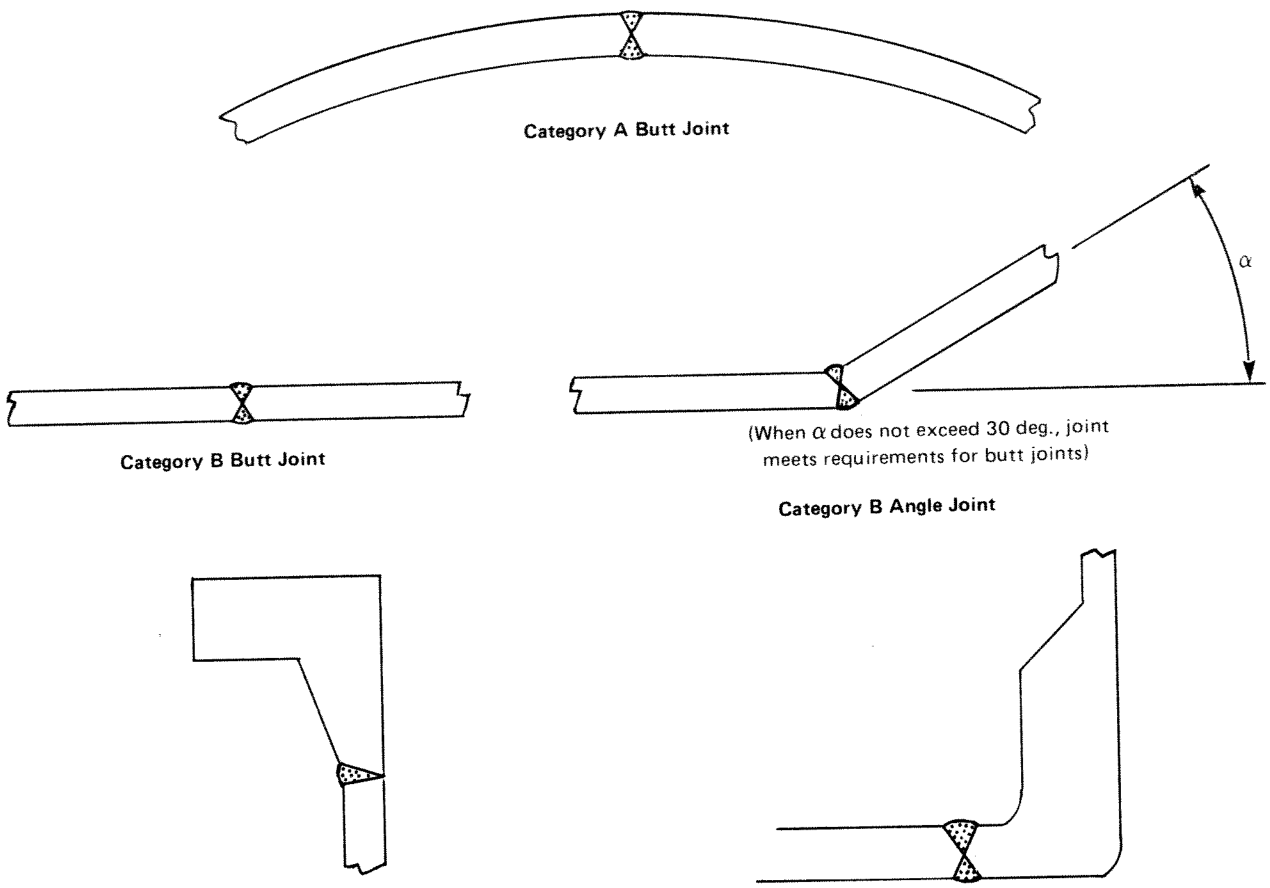


FIG. NB-3352-1 TYPICAL BUTT JOINTS

ments of NB-4244(c) and shall be capable of being examined in accordance with NB-5244.

(2) When the deposited weld metal is used as reinforcement, the coefficients of thermal expansion of the base metal, the weld metal, and the nozzle shall not differ by more than 15% of the lowest coefficient involved.

(3) The minimum dimensions of Fig. NB-4244(c)-1 shall be met, where

t = nominal thickness of part penetrated, in.
t_n = nominal thickness of penetrating part, in.
$t_c = 0.7t_n$ or $\frac{1}{4}$ in., whichever is less
$r_1 = \frac{1}{4}t$ or $\frac{3}{4}$ in., whichever is less

(4) The corners of the end of each nozzle neck extending less than $\sqrt{dt_n}$ beyond the inner surface of the part penetrated shall be rounded to a radius of one-half the thickness t_n of the nozzle neck or $\frac{3}{4}$ in., whichever is smaller.

(d) Attachment of Nozzles Using Partial Penetration Welds

(1) Partial penetration welds used to connect nozzles as permitted in NB-3337.3 shall meet the fabrication requirements of NB-4244(d) and shall be capable of being examined in accordance with the requirements of NB-5245.

(2) The minimum dimensions of Figs. NB-4244(d)-1 and NB-4244(d)-2 shall be met, where

t = nominal thickness of part penetrated, in.
t_n = nominal thickness of penetrating part or the lesser of t_{n1} or t_{n2} in Fig. NB-4244(d)-2, in.
$t_c = 0.7t_n$ or $\frac{1}{4}$ in., whichever is less
$r_1 = \frac{1}{4}t_n$ or $\frac{3}{4}$ in., whichever is less
d = outside diameter of nozzle or of the inner cylinder as shown in NB-4244(d)-2
$r_2 = \frac{1}{16}$ in. minimum
$r_3 = r_2$ or equivalent chamber minimum

$\lambda = \frac{1}{16}$ in. minimum
$\lambda = t_n$ maximum, in.
$r_4 = \frac{1}{2}t_n$ or $\frac{3}{4}$ in., whichever is less

(3) The corners of the end of each nozzle, extending less than $\sqrt{dt_n}$ beyond the inner surface of the part penetrated, shall be rounded to a radius of one-half of the thickness t_n of the penetrating part or $\frac{3}{4}$ in., whichever is smaller.

(4) Weld groove design for partial penetration joints attaching nozzles may require special consideration to achieve the $1.25t_n$ minimum depth of weld and adequate access for welding examination. The welds shown in the sketches of Figs. NB-4244(d)-1 and NB-4244(d)-2 may be on either the inside or the outside of the vessel shell. Weld preparation may be J-groove as shown in the figures or straight bevel.

(5) A fatigue strength reduction factor of not less than four shall be used when fatigue analysis is required.

(e) Oblique Full Penetration Nozzles. Internal or external nozzles shall meet the fabrication requirements of NB-4244(e) and shall be capable of being examined in accordance with NB-5246. Radiography of the nozzle weld may be waived by NB-5246, provided the requirements of (1) through (6) below are met.

(1) The inside nozzle diameter shall not exceed 6 in.

(2) The angle which the nozzle axis makes with the vessel wall at the point of attachment shall not be smaller than 40 deg.

(3) The opening shall be completely reinforced, with the reinforcement located in the shell or head of the vessel.

(4) The nozzle shall be subjected to essentially no pipe reactions and no thermal stresses greater than in the vessel itself.

(5) The nozzle wall and the weld shall develop the full strength of the nozzle.

(6) The minimum dimensions of Fig. NB-4244(e)-1 shall be met, where

t = nominal thickness of part penetrated, in.
t_n = nominal thickness of penetrating part, in.
$t_c = 0.7t_n$ or $\frac{1}{4}$ in., whichever is less
$r_1 = \frac{1}{2}t$ or $\frac{3}{4}$ in., whichever is less

NB-3354 Structural Attachment Welds

Welds for structural attachments shall meet the requirements of NB-4430.

NB-3355 Welding Grooves

The dimensions and shape of the edges to be joined shall be such as to permit complete fusion and complete joint penetration, except as otherwise permitted in NB-3352.4.

NB-3357 Thermal Treatment

All vessels and vessel parts shall be given the appropriate postweld heat treatment prescribed in NB-4620.

NB-3360 SPECIAL VESSEL REQUIREMENTS

NB-3361 Category A or B Joints Between Sections of Unequal Thickness

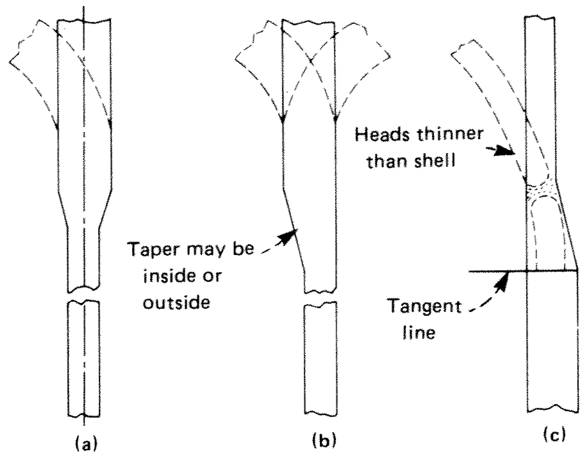
In general, a tapered transition section as shown in Fig. NB-3361-1 which is a type of gross structural discontinuity (NB-3213.2) shall be provided at joints of Categories A and B between sections that differ in thickness by more than one-fourth the thickness of the thinner section. The transition section may be formed by any process that will provide a uniform taper. An ellipsoidal or hemispherical head which has a greater thickness than a cylinder of the same inside diameter may be machined to the outside diameter of the cylinder provided the remaining thickness is at least as great as that required for a shell of the same diameter. A uniform taper is not required for flanged hubs. The adequacy of the transition shall be evaluated by stress analysis. Stress intensity limitations are given in NB-3220. The requirements of this paragraph do not apply to flange hubs.

NB-3362 Bolted Flange Connections

It is recommended that the dimensional requirements of bolted flange connections to external piping conform to ANSI B16.5, Steel Pipe Flanges and Flanged Fittings.

NB-3363 Access Openings

Access openings, where provided, shall consist of handhole or manhole openings having removable covers. These may be located on either the inside or outside of the shell or head openings and may be attached by studs or bolts in combination with gaskets or welded membrane seals or strength welds. Plugs using pipe threads are not permitted.



GENERAL NOTE:
Length of taper may include the width of the weld.

FIG. NB-3361-1 CATEGORY A AND B JOINTS BETWEEN SECTIONS OF UNEQUAL THICKNESS

NB-3364 Attachments

Attachments used to transmit support loads shall meet the requirements of NB-3135.

NB-3365 Supports

All vessels shall be so supported and the supporting members shall be arranged and attached to the vessel wall in such a way as to provide for the maximum imposed loadings. The stresses produced in the vessel by such loadings and by steady state and transient thermal conditions shall be subjected to the stress limits of this Subsection. Additional requirements are given in NCA-3240 and Subsection NF.

NB-3400 PUMP DESIGN

NB-3410 GENERAL REQUIREMENTS FOR CENTRIFUGAL PUMPS

NB-3411 Scope

NB-3411.1 Applicability. The rules of NB-3400 apply to (a) through (j) below.

(a) pump casings
(b) pump inlets and outlets
(c) pump covers
(d) clamping rings
(e) seal housing and seal glands
(f) related bolting
(g) pump internal heat exchanger piping
(h) pump auxiliary nozzle connections up to the face of the first flange or circumferential joint in welded connections, excluding the connecting weld
(i) piping identified with the pump and external to and forming part of the pressure retaining boundary and supplied with the pump
(j) mounting feet or pedestal supports when integrally attached to the pump pressure retaining boundary and supplied with the pump

NB-3411.2 Exemptions. The rules of NB-3400 do not apply to (a) through (c) below.

(a) pump shafts and impellers
(b) nonstructural internals
(c) seal packages

NB-3412 Acceptability

NB-3412.1 Acceptability of Large Pumps. The requirements for the design acceptability of pumps having an inlet connection greater than 4 in. nominal pipe size diameter are given in (a) and (b) below.

(a) The design shall be such that the requirements of NB-3100 and of NB-3200 or Appendix II (provided the requirements of NB-3414 and the minimum wall thicknesses of NB-3430 are met) are satisfied.

(b) The rules of this Subarticle shall be met. In cases of conflict between NB-3100 and NB-3200 or Appendix II and NB-3400, the requirements of NB-3400 apply.

NB-3412.2 Acceptability of Small Pumps. The requirements for the design acceptability of pumps having an inlet connection 4 in. nominal pipe size diameter or smaller are given in (a) and (b) below.

(a) The design shall be such that the requirements of NB-3100 or Appendix II are satisfied.

(b) The rules of this Subarticle shall be met. In cases of conflict between NB-3100 or Appendix II and NB-3400, the requirements of this Article shall apply.

NB-3414 Design and Service Conditions

The general design considerations, including definitions of NB-3100 plus the requirements of NB-3320, NB-3330, NB-3361, and NB-3362 are applicable to pumps.

NB-3415 Loads From Connected Piping

(a) Loads imposed on pump inlets and outlets by connected piping shall be considered in the pump casing design. The forces and moments produced by the

connected piping on each pump inlet and outlet shall be provided by the Owner in the Design Specifications.

(b) Stresses generated in the pump casing by the connected piping shall be combined with the pressure stresses in accordance with the requirements of NB-3200.

NB-3417 Earthquake Loadings

(a) The rules of this Subarticle consider that under earthquake loadings the integrity of the pump pressure retaining body is considered adequate when the requirements of NB-3100 and NB-3200 are satisfied.

(b) Where pumps are provided with drivers on extended supporting structures and these structures are essential to maintaining pressure integrity, an analysis, when required by the Design Specifications, may be performed based on static forces resulting from equivalent earthquake accelerations acting at the centers of gravity of the extended masses.

NB-3418 Corrosion

The requirements of NB-3121 apply.

NB-3419 Cladding

Cladding dimensions used in the design of pumps shall be required as in NB-3122.

NB-3420 DEFINITIONS

NB-3421 Radially Split Casing

A radially split casing shall be interpreted as one in which the primary sealing joint is radially disposed around the shaft.

NB-3422 Axially Split Casing

An axially split casing shall be interpreted as one in which the primary sealing joint is axially disposed with respect to the shaft.

NB-3423 Single and Double Volute Casings

Figures NB-3423-1 and NB-3423-2 show typical single and double volute casings, respectively.

NB-3424 Seal Housing

Seal housing is defined as that portion of the pump cover or casing assembly which contains the seal and forms a part of the primary pressure boundary.

NB-3425 Typical Examples of Pump Types

Figures NB-3441.1-1 through NB-3441.6(a)-1 are typical examples to aid in the determination of pump type and are not to be considered as limiting. Bearing locations and inlet and outlet orientations are optional.

NB-3430 DESIGN REQUIREMENTS FOR CENTRIFUGAL PUMPS

NB-3431 Design of Welding

(a) The design of welded construction shall be in accordance with NB-3350.

(b) Partial penetration welds are permitted for piping connections 2 in. nominal pipe size and less when the requirements of NB-3337.3 and NB-3352.4(d) are met.

NB-3432 Cutwater Tip Stresses

(a) It is recognized that localized high stresses may occur at the cutwater tips of volute casings (Fig. NB-3441.3-2). Adequacy of the design in this area shall be demonstrated either by an investigation through experimental stress analysis in accordance with Appendix II or by detailing satisfactory service performance of other pumps under similar operating conditions.

(b) Where experimental stress analysis is used, stress intensity at this point shall meet the requirements of NB-3222.

NB-3433 Reinforcement of Pump Casing Inlets and Outlets

NB-3433.1 Axially Oriented Inlets and Outlets

(a) An axially oriented pump casing inlet or outlet shall be considered similar to an opening in a vessel and shall be reinforced. It shall be treated as required in NB-3331 through NB-3336.

(b) To avoid stress concentrations, the outside radius r_2 in Fig. NB-3441.3-2 shall not be less than one-half the thickness of the inlets and outlets as reinforced.

NB-3433.2 Radially Oriented Inlets and Outlets. Reinforcement of radially oriented inlets and outlets in accordance with the rules of NB-3331 through NB-3336 is required.

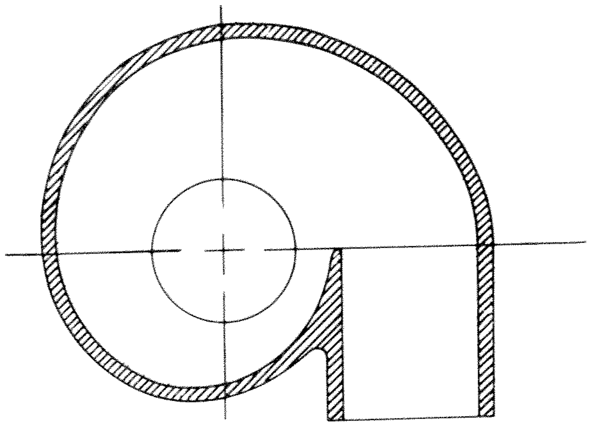


FIG. NB-3423-1
TYPICAL SINGLE VOLUTE CASING

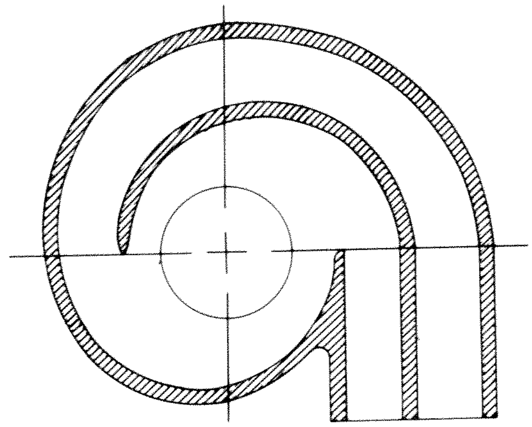


FIG. NB-3423-2
TYPICAL DOUBLE VOLUTE CASING

NB-3433.3 Tangential Inlets and Outlets. Except as modified in NB-3433.4, any design which has been demonstrated to be satisfactory for the specified Design Loadings may be used.

NB-3433.4 Minimum Inlet and Outlet Wall Thicknesses. The wall thickness of the inlet or outlet shall not be less than the minimum wall thickness of the casing for a distance l as shown in Fig. NB-3433.4-1. The wall thickness beyond the distance l may be reduced to the minimum wall thickness of the connected piping. The change in wall thickness shall be gradual and have a maximum slope as indicated in Fig. NB-4250-1. The distance l in Fig. NB-3433.4-1 is the limit of reinforcement. The value of l, in., shall be determined from the relationship:

$$l = 0.5\sqrt{r_m t_m}$$

where

$r_m = r_i + 0.5t_m$, in.

r_i = inlet or outlet inside radius, in.

$= d_i/2$

t_m = mean inlet or outlet wall thickness, in., taken between section x–x and a parallel section y–y

NB-3434 Bolting

Bolting in axisymmetric arrangements involving the pressure boundary shall be designed in accordance with NB-3230.

NB-3435 Piping

NB-3435.1 Piping Under External Pressure. Piping located within the pressure retaining boundary of the pump shall be designed in accordance with NB-3133.

NB-3435.2 Piping Under Internal Pressure. Piping identified with the pump and external to or forming a part of the pressure retaining boundary, such as auxiliary water connections, shall be designed in accordance with NB-3600.

NB-3436 Attachments

(a) External and internal attachments to pumps shall be designed so as not to cause excessive localized bending stresses or harmful thermal gradients in the pump as determined by the rules of NB-3200. Such attachments shall be designed to minimize stress concentrations in applications where the number of stress cycles, due either to pressure or thermal effect, is relatively large for the expected life of the equipment.

(b) Attachments shall meet the requirements of NB-3135.

NB-3437 Pump Covers

Pump covers shall be designed in accordance with NB-3200.

NB-3438 Supports

Pump supports shall be designed in accordance with the requirements of Subsection NF unless included under the rules of NB-3411.1(j).

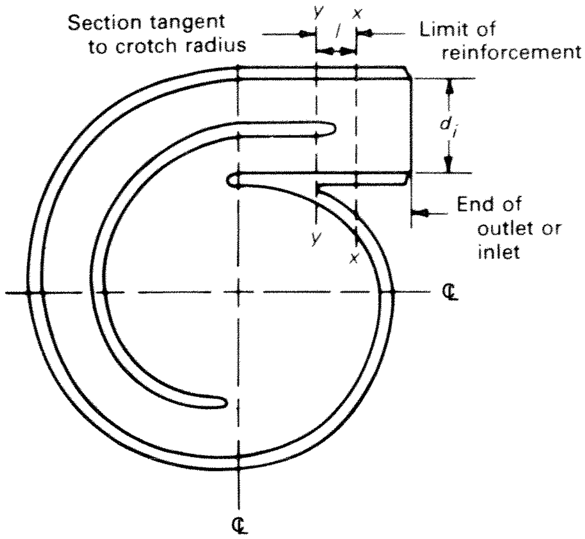


FIG. NB-3433.4-1 MINIMUM TANGENTIAL INLET AND OUTLET WALL THICKNESS

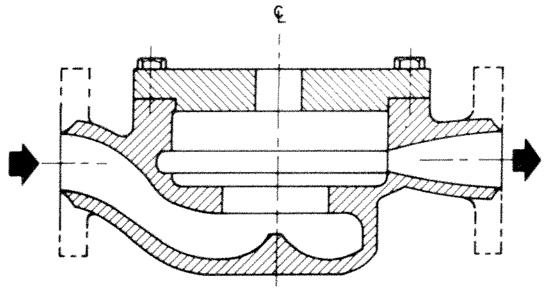


FIG. NB-3441.1-1 TYPE A PUMP

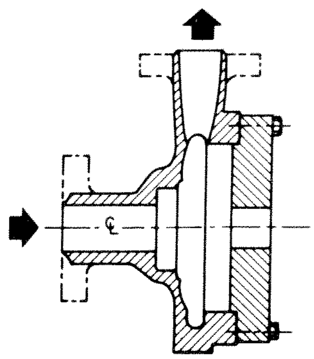


FIG. NB-3441.1-2 TYPE A PUMP

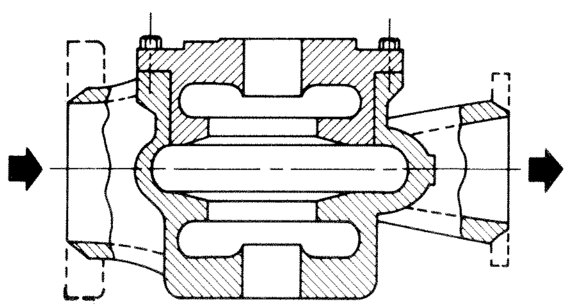


FIG. NB-3441.2-1 TYPE B PUMP

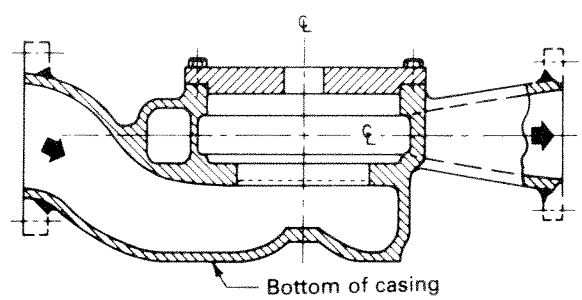


FIG. NB-3441.3-1 TYPE C PUMP

NB-3440 DESIGN OF SPECIFIC PUMP TYPES

NB-3441 Standard Pump Types

NB-3441.1 Design of Type A Pumps. Type A pumps are those having single volutes and radially split casings with single suction, as illustrated in Figs. NB-3441.1-1 and NB-3441.1-2. Their design shall be in accordance with the requirements of this Subarticle.

NB-3441.2 Design of Type B Pumps. Type B pumps are those having single volutes and radially split casings with double suction, as illustrated in Fig. NB-3441.2-1. Their design shall be in accordance with the requirements of this Subarticle.

NB-3441.3 Design of Type C Pumps. Type C pumps are those having double volutes and radially split casings with single suction, as illustrated in Figs. NB-3441.3-1 and NB-3441.3-2. The splitter is considered a structural part of the casing. Casing design shall be in accordance with the requirements of this Subarticle and with those given in (a) through (d) below.

(a) Casing Wall Thickness. Except where specifically indicated in these rules, no portion of the casing wall shall be thinner than the value of t determined as follows:

$$t = (0.63 \times P \times A)/S_m$$

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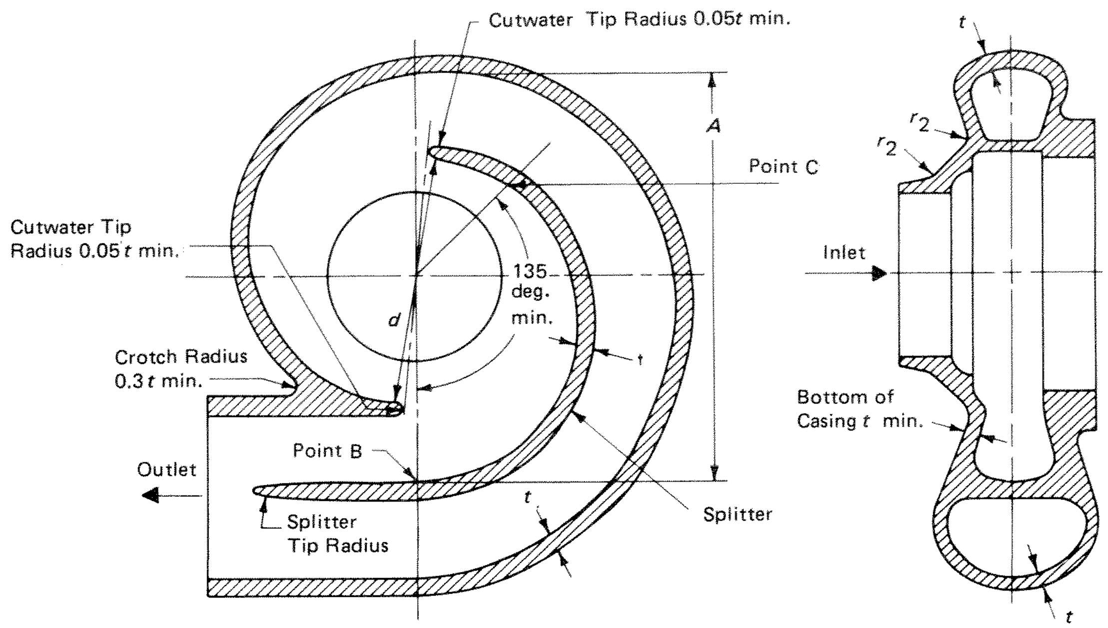

FIG. NB-3441.3-2 TYPE C PUMP

where

t = minimum allowable wall thickness, in.

P = Design Pressure, psig

A = scroll dimension inside casing as shown in Fig. NB-3441.3-2, in.

S_m = allowable stress intensity for casing material at Design Temperature, psi

(b) Splitter Wall Thickness

(1) The splitter shall have a minimum wall thickness of t as determined above for the casing wall and shall extend from point B in Fig. 3441.3-2 through a minimum angle of 135 deg. to point C. Beyond point C, the splitter wall may be reduced in thickness and tapered to blend with the cutwater tip radius.

(2) Cutwater tip and splitter tip radii shall not be less than 0.05t.

(3) All cutwater and splitter fillets, including the tips, where they meet the casing wall, shall have a minimum radius of 0.10t or 0.25 in., whichever is greater.

(c) Crotch Radius (Fig. NB-3441.3-2). The crotch radius shall not be less than 0.3t.

(d) Bottom of Casing

(1) That section of the pump casing within the diameter defined by dimension A in Fig. NB-3441.3-2 on the inlet side of the casing, normally referred to as the bottom of the casing (Fig. NB-3441.3-1), shall have a wall thickness no less than the value of t determined in (a) above.

(2) The casing surface shall be analyzed in accordance with an acceptable procedure, such as that shown for flat heads in A-5000, or by an experimental stress technique, such as described in Appendix II.

(3) The minimum permissible thickness of the bottom of the casing shall be the lesser of the value determined by the analysis in (2) above and the value obtained from the calculation shown in (a) above.

NB-3441.4 Design of Type D Pumps

(a) Type D pumps are those having double volutes and radially split casings with double suction as illustrated in Fig. NB-3441.4(a)-1. The design shall be in accordance with this Subarticle.

(b) The requirements of NB-3441.3(a), (b), and (c), governing casing wall thickness, splitter wall thickness, and crotch radius, apply.

(c) In the casing portion between the cover and the casing wall, a wall thickness in excess of t may be required.

NB-3441.5 Design of Type E Pumps. Type E pumps are those having volute type radially split casings and multivane diffusers which form structural parts of the

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casing as illustrated in Fig. NB-3441.5-1. The design shall be in accordance with this Subarticle.

NB-3441.6 Design of Type F Pumps

(a) Type F pumps are those having radially split, axisymmetric casings with either tangential or radial outlets as illustrated in Fig. NB-3441.6(a)-1. The basic configuration of a Type F pump casing is a shell with a dished head attached at one end and a bolting flange at the other. The inlet enters through the dished head, and the outlet may be either tangent to the side or normal to the center line of the casing. Variations of these inlet and outlet locations are permitted.

(b) The design of Type F pumps shall be in accordance with this Subarticle.

NB-3442 Special Pump Types — Type J Pumps

(a) Type J pumps are those that cannot logically be classified with any of the preceding types.

(b) Any design method which has been demonstrated to be satisfactory for the specified Design Conditions may be used.

NB-3500 VALVE DESIGN

NB-3510 ACCEPTABILITY[14]

NB-3511 General Requirements[15]

The requirements for design acceptability for valves shall be those given in this Subarticle. In all cases, pressure–temperature rating shall be as given in NB-3530 and, except for NB-3512.2(d) and in local regions (NB-3221.2), the wall thickness of the valve body shall not be less than that given by NB-3541. The requirements for prevention of nonductile failure [NB-3211(d)] may be considered to have been met under all service loadings at temperatures equal to or higher than the lowest service temperatures, if all materials meet the requirements of NB-2332 and if all other

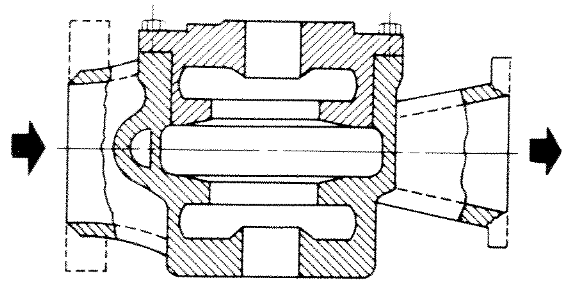

FIG. NB-3441.4(a)-1 TYPE D PUMP

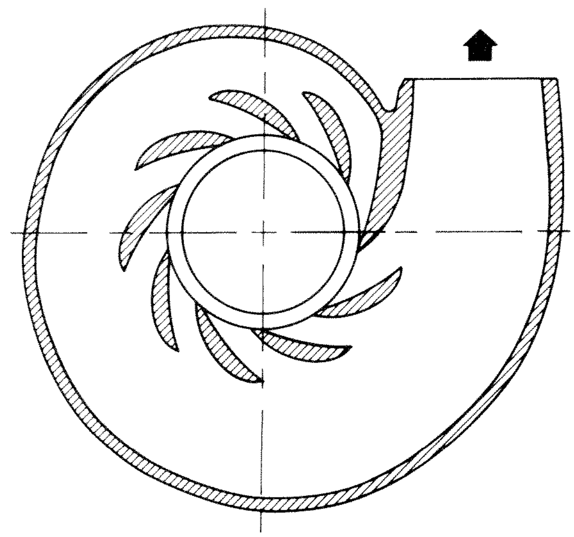

FIG. NB-3441.5-1 TYPE E PUMP

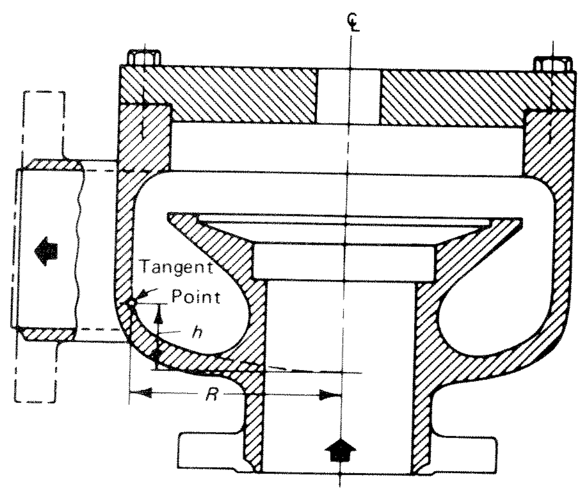

FIG. NB-3441.6(a)-1 TYPE F PUMP

[14]These requirements for the acceptability of a valve design are not intended to ensure the functional adequacy of the valve. However, for pressure relief valves the Designer is cautioned that the requirements of NB-7000 relative to set pressure, lift, blowdown, and closure shall be met.

[15]CAUTIONARY NOTE: Certain types of double seated valves have the capability of trapping liquid in the body or bonnet cavity in the closed position. If such a cavity accumulates liquid and is in the closed position at a time when adjacent system piping is increasing in temperature, a substantial and uncontrolled increase in pressure in the body or bonnet cavity may result. Where such a condition is possible, it is the responsibility of the Owner or the Owner's designee to provide, or require to be provided, protection against harmful overpressure in such valves.

applicable requirements of this Subsection are met.[16] The requirements of NCA-3254(a) for specifying the location of valve boundary jurisdiction may be considered to have been met by employing the minimum limits of NB-1131, unless the Design Specification extends the boundary of jurisdiction beyond these minimum limits. The requirements of NCA-3254(b) for specifying the boundary conditions are not applicable to valve end connections.

NB-3512 Acceptability of Large Valves

Valve designs having an inlet piping connection larger than 4 in. nominal pipe size are acceptable when they satisfy either the standard design rules or one of the alternative design rules.

NB-3512.1 Standard Design Rules. The design shall be such that requirements of this Subarticle are met. The requirements of NB-3530 through NB-3550 apply to valves of conventional shape having generally cylindrical or spherical bodies with a single neck of a diameter commensurate with that of the main body portion, such as having a neck inside diameter less than twice the main run inside diameter in the neck region.

NB-3512.2 Alternative Design Rules. A valve design may not satisfy all of the requirements of NB-3512.1. A design may be accepted provided it meets one of the alternatives listed in (a), (b), (c), or (d) below.

(a) When the valve design satisfies the rules of NB-3530 through NB-3546.2 with thermal stresses neglected, the rules of NB-3200 relative to accounting for thermal secondary stresses and fatigue analysis [NB-3222.4(e) and NB-3232.3] shall also be satisfied.

(b) When a valve is exempted from fatigue analysis by the rules of NB-3222.4(d), the design is acceptable, provided that the requirements of (1) or (2) below are met.

(1) The rules of NB-3530 through NB-3546 shall be met. The rules of NB-3200 may be substituted for those of NB-3545.2 for evaluating secondary stresses, and NB-3545.3 need not be considered.

(2) The rules of NB-3530 and NB-3541 shall be met. An experimental stress analysis is performed in accordance with Appendix II, and the rules of NB-3200 with respect to primary and secondary stresses resulting from pressure and mechanical loads shall be met. Unless otherwise specified in the Design Specifications, the pipe reactions shall be taken as those loads which produce a stress [NB-3545.2(b)] of 0.5 times the yield strength of the piping in tension for the direct or axial load and a stress of 1.0 times the yield strength of the piping in bending and torsion. Thermal secondary stresses shall be accounted for by either the rules of NB-3200 or NB-3545.

(c) When a valve design satisfies the rules of NB-3530 and NB-3541, and when an experimental stress analysis has been performed upon a similar valve in accordance with Appendix II, and an acceptable analytic method has been established, the results may be used in conjunction with the requirements of NB-3200 for pressure and mechanical loads to establish design acceptability. Accommodation of thermal secondary stresses and pipe reactions shall be as given in NB-3512.2(b)(2). Requirements for fatigue analysis of either NB-3200 or NB-3550 shall be met.

(d) When permitted by the Design Specification, a weld end valve that does not meet all of the requirements of NB-3540 may be designed so that it meets the requirements of NB-3200 for all pressure retaining parts and those parts defined by NB-3546.3(a), and shall also meet all of the following requirements.

(1) Pressure, thermal, and mechanical effects, such as those resulting from earthquake, maximum stem force, closure force, assembly forces, and others that may be defined in the Design Specification, shall be included in the design analysis. For Level A Service Limits, the pipe reaction effects are to be determined by considering that the maximum fiber stress in the connected pipe is at one-half of its yield strength in direct tension and at its yield strength in torsion and in bending in the plane of the neck and run, and also in the plane of the run perpendicular to the neck, each considered separately. The individual pipe reaction effects that result in the maximum stress intensity at all points, including all other effects, shall be used for the analysis to satisfy the rules of NB-3200. The valve Design Specification shall provide the loadings and operating requirements to be considered under Level B, C, and D Service Limits [NCA-3252(a)(6)] for which a design analysis is to be included in the Design Report.

(2) In place of using the values of S_m to satisfy the rules of NB-3200, the allowable stress intensity values for ferritic valve body and bonnet materials shall be those allowable stress values given in Table I-7.1. For materials in Table I-1.2, a reduced allowable stress intensity based on applying a factor of 0.67 to the yield strengths listed in Table I-2.2 shall be used.

[16]The severity and frequency of specified fluid temperature variations may be such that the period of calculated pressure integrity is less than plant design life. In such cases it is the responsibility of the Certificate Holder to state these conditions in the Design Report (NB-3560).

(3) The adequacy of the stress analysis of the body and bonnet shall be verified by experimental stress analysis conducted in accordance with the requirements of II-1100 through II-1400. Individual tests shall be made to verify the adequacy of the stress analysis of internal pressure effects and pipe reaction effects. Tests shall be made on at least one valve model of a given configuration, but a verified analytical procedure may then be applied to other valves of the same configuration, although they may be of different size or pressure rating. The geometrical differences shall be accounted for in the extrapolation stress analysis. The analytical procedure shall have verified capability of providing this extrapolation.

(4) A Design Report shall be prepared in sufficient detail to show that the valve satisfies all applicable requirements.

(5) Prior to installation, the valve shall be hydrostatically tested in accordance with NB-3531.2. For this purpose, the primary pressure rating shall be determined by interpolation in accordance with NB-3543(c).

NB-3513 Acceptability of Small Valves

Valve designs having an inlet piping connection 4 in. nominal pipe size or less are acceptable when they satisfy either the standard design rules or the alternative design rules.

NB-3513.1 Standard Design Rules. The design shall be such that the requirements of NB-3530 and NB-3541 shall be met for wall thicknesses corresponding to the applicable pressure–temperature rating. When the Special Class Ratings of ANSI B16.34 apply, the NDE exemptions of NB-2510 shall not be used.

NB-3513.2 Alternative Design Rules. A valve design shall satisfy the requirements of NB-3512.2.

NB-3515 Acceptability of Metal Bellows and Metal Diaphragm Stem Sealed Valves

Valves using metal bellows or metal diaphragm stem seals shall be constructed in accordance with the rules of this Subarticle, based on the assumption that the bellows or diaphragms do not retain pressure, and Design Pressure is imposed on a required backup stem seal such as packing. The bellows or diaphragms need not be constructed in accordance with the requirements of this Section.

NB-3520 DESIGN CONSIDERATIONS

NB-3521 Design and Service Loadings

The general design considerations of NB-3100 are applicable to valves. In case of conflict between NB-3100 and NB-3500, the requirements of NB-3500 shall apply.

NB-3524 Earthquake

The rules of this Subarticle consider that under earthquake loadings the piping system, not the valve, will be limiting and that the integrity of the valve pressure retaining body is adequately considered under the piping requirements of NB-3600. Where valves are provided with operators having extended structures and these structures are essential to maintaining pressure integrity, an analysis, when required by the Design Specifications, may be performed based on static forces resulting from equivalent earthquake accelerations acting at the centers of gravity of the extended masses.

NB-3525 Level A and B Service Limits

The design rules of NB-3512 and NB-3513 apply to loadings for which Level A or B Limits are designated except that when evaluating Level B Limits during operation of relief or safety valves (a) and (b) below shall be met.

(a) The service pressure may exceed the Design Pressures defined by the pressure-temperature ratings of ANSI B16.34 by no more than 10%.

(b) The rules of NB-3540 apply using allowable stress intensity values of 110% of those listed in Appendix I.

NB-3526 Level C Service Limits

If the Design Specifications specify any loadings for which Level C Limits are designated, the rules used in evaluating these loadings shall be those of NB-3512 and NB-3513, except as modified by the following subparagraphs.

NB-3526.1 Pressure–Temperature Ratings. The pressure permissible for loadings for which Level C Limits are designated shall not exceed 120% of that permitted for Level A Limits.

NB-3526.2 Pipe Reaction Stress. Pipe reaction stresses shall be computed in accordance with the equations of NB-3545.2(b)(1), and the allowable value considered individually is $1.8S_m$ for the valve body material at 500°F. In performing these calculations, the value of S shall be taken as 1.2 times the yield strength at 500°F of the material of the connected pipe, or 36.0 ksi when the pipe material is not defined in the Design Specifications.

NB-3526.3 Primary Stress and Secondary Stress. The equation of NB-3545.2 shall be satisfied using C_p equal to 1.5, P_{ed} computed in accordance with NB-3526.2, and Q_T equal to 0, and the calculated value shall be limited to $2.25S_m$.

NB-3526.4 Secondary and Peak Stresses. The requirements of NB-3545 and NB-3550 need not be met.

NB-3527 Level D Service Limits

If the Design Specifications specify any loadings for which Level D Limits are designated, the guidelines of Appendix F may be used in evaluating those loadings independently of other loadings.

NB-3530 GENERAL RULES

NB-3531 Pressure–Temperature Ratings and Hydrostatic Tests

NB-3531.1 Pressure–Temperature Ratings. A valve designed in accordance with NB-3541 may be used in accordance with the pressure–temperature ratings in ANSI B16.34, Tables 2-1.1A to 2-2.7A (Standard Class) for flanged end or welding end (including socket welding end) valves, and ANSI B16.34, Tables 2-1.1B to 2-2.7B (Special Class) for welding end (including socket welding end) valves, provided the Design Pressure and Design Temperature are used. When a single valve has a flanged and a welding end, the flanged end requirements shall be used. The materials[17] listed in ANSI B16.34, Table 1, may be used if listed in Tables I-1.0, subject to the temperature limitations therein, and as defined in NCA-1220.

NB-3531.2 Hydrostatic Tests

(a) Valves designed in accordance with NB-3541 shall be subjected to the shell hydrostatic test pressures required by ANSI B16.34 and in accordance with other appropriate rules of NB-6000. Valves with a primary pressure rating less than Class 150 shall be subjected to the required test pressure for Class 150 rated valves.

(b) The shell hydrostatic test shall be made with the valve in the partially open position. Stem leakage during this test is permissible. End closure seals for retaining fluid at test pressure in welding end valves may be positioned in the welding end transitions, as defined in NB-3544.8(b), in reasonable proximity to the end plane of the valve so as to ensure safe application of the test pressure.

(c) After the shell hydrostatic test, a valve closure test shall also be performed with the valve in the fully closed position with a test pressure across the valve disk no less than 110% of the 100°F pressure rating. For valves that are designed for Service Conditions that have the pressure differential across the closure member limited to values less than the 100°F pressure rating, and have closure members or actuating devices (direct, mechanical, fluid, or electrical), or both, that would be subject to damage at high differential pressures, the test pressure may be reduced to 110% of the maximum specified differential pressure in the closed position. This exception shall be identified in the Design Specification, and this maximum specified differential pressure shall be noted on the valve nameplate and N Certificate Holder's Data Report Form. During this test, seat leakage is permitted unless a limiting leakage value is defined by the Design Specifications. The duration of this test shall be 1 min/in. of minimum wall thickness t_m with a minimum duration of 1 min unless otherwise defined in the Design Specifications.

(d) For valves designed for nonisolation service, whose primary function is to modulate flow, and by their design are prevented from providing full closure, the valve closure test defined in (c) above is not required. This exception shall be identified in the Design Specification and noted on the valve nameplate and the N Certificate Holder's Data Report Form.

(e) Hydrostatic tests for metal bellows or metal diaphragm stem sealed valves shall include hydrostatic testing of the valve body, bonnet, body-to-bonnet joint, and either the bellows or diaphragm or the required backup stem seal.

(f) The inlet (primary pressure containing) portion of pressure relief valves shall be hydrostatically tested at a pressure at least 1.5 times the set pressure marked on the valve. For closed system application, the outlet portion of the pressure relief valves shall be hydrostatically tested to 1.5 times the design secondary pressure (NB-7111).

NB-3531.3 Allowance for Variation From Design Loadings. Under the conditions of relief or safety valve operation for valves designed in accordance with NB-3541, the service pressure may exceed the Design Pressure as defined by the pressure–temperature ratings of ANSI B16.34 by no more than 10%.

NB-3532 Design Stress Intensity Values

Design stress intensity values to be used in the design of valves are given in Tables I-1.0.[17]

NB-3533 Marking

Each valve shall be marked as required by ANSI B16.34 and NCA-8220.

NB-3534 Nomenclature

A_f = effective fluid pressure area based on fully corroded interior contour for calculating crotch primary membrane stress [NB-3545.1(a)], sq in.

A_m = metal area based on fully corroded interior contour effective in resisting fluid force acting on A_f [NB-3545.1(a)], sq in.

C_a = stress index for oblique bonnets [NB-3545.2(a)]

C_b = stress index for body bending secondary stress resulting from moment in connected pipe [NB-3545.2(b)]

C_p = stress index for body primary plus secondary stress, inside surface, resulting from internal pressure [NB-3545.2(a)]

C_2 = stress index for thermal secondary membrane stress resulting from structural discontinuity

C_3 = stress index for maximum secondary membrane plus bending stress resulting from structural discontinuity

C_4 = maximum magnitude of the difference in average wall temperatures for wall thicknesses T_{e1} and t_e (resulting from a step change in fluid temperature ΔT_f) divided by ΔT_f

C_5 = stress index for thermal fatigue stress component resulting from through-wall temperature gradient caused by step change in fluid temperature (NB-3550)

$C_6 = E\alpha$ = product of Young's modulus and the coefficient of linear thermal expansion at 500°F, psi/°F (NB-3550)

C_7 = stress index for thermal stress resulting from through-wall temperature gradient associated with 100°F/hr fluid temperature change rate, psi/in.

d = inside diameter used as a basis for crotch reinforcement [NB-3545.1(a)], in.

d_m = inside diameter used as basis for determining body minimum wall thickness (NB-3541), in.

F_b = bending modulus of standard connected pipe

G_b = valve body section bending modulus at crotch region [NB-3545.2(b)], in.^{3}

G_d = valve body section area at crotch region [NB-3545.2(b)], sq in.

G_t = valve body section torsional modulus at crotch region [NB-3545.2(b)], in.^{3}

I = moment of inertia, in.^{4}, used in calculating G_b [NB-3545.2(b)(5)]

I_t = fatigue usage factor for step changes in fluid temperature

K_e = strain distribution factor used in elastic–plastic fatigue calculation (NB-3550)

L_A, L_N = effective distances used to determine A_f, A_m [NB-3545.1(a)(3)]

m, n = material parameters for determining K_e (NB-3554)

N_a = permissible number of complete startup/shutdown cycles at 100°F/hr fluid temperature change rate (NB-3545.3)

N_i = permissible number of step changes in fluid temperature from Figs. I-9.0

N_{ri} = required number of fluid step temperature changes ΔT_{fi} (NB-3553)

p_d = Design Pressure, psi

p_r = Pressure Rating Class Index, psi

p_s = standard calculation pressure from NB-3545.1, psi

p_1, p_2 = rated pressures from tables of ANSI B16.34 corresponding to Pressure Rating Class Indices p_{r1} and p_{r2}, psi

P_{eb} = secondary stress due to pipe reaction [NB-3545.2(b)], psi

ΔP_{fi} = full range of pressure fluctuation associated with ΔT_{fi}, psi

ΔP_i = specified range of pressure fluctuation associated with ΔT_i, psi

P_m = general primary membrane stress intensity at crotch region, calculated according to NB-3545.1(a), psi

Q_p = sum of primary plus secondary stresses at crotch resulting from internal pressure [NB-3545.2(a)], psi

Q_{T1} = maximum thermal stress component caused by through-wall temperature gradient associated with 100°F/hr fluid temperature change rate [NB-3545.2(c)], psi

Q_{T3} = maximum thermal secondary membrane plus bending stress resulting from structural discontinuity and 100°F/hr fluid temperature change rate, psi

r_2 = fillet radius of external surface at crotch

[17]Special features such as wear surfaces or seating surfaces may demand special alloys or proprietary treatments. The absence of such materials from Tables I-1.0 shall not be construed to prohibit their use and such materials do not require approval under Appendix IV (NB-2121).

[NB-3545.1(a)], in.

r_i = inside radius of body at crotch region for calculating Q_p [NB-3545.2(a)], in.

r = mean radius of body wall at crotch region [Fig. NB-3545.2(c)-1], in.

S = assumed maximum stress in connected pipe for calculating P_e [NB-3545.2(b)], psi

S_m = design stress intensity (NB-3532), psi

S_n = sum of primary plus secondary stress intensities at crotch region resulting from 100°F/hr temperature change rate (NB-3545.2), psi

$S_{n(\max)}$ = maximum range of sum of primary plus secondary stress, psi

S_{p1} = fatigue stress intensity at inside surface in crotch region resulting from 100°F/hr fluid temperature change rate (NB-3545.3), psi

S_{p2} = fatigue stress intensity at outside surface in crotch region resulting from 100°F/hr fluid temperature change rate (NB-3545.3), psi

S_i = fatigue stress intensity range at crotch region resulting from step change in fluid temperature ΔT_{fi} and pressure ΔP_{fi} (NB-3550), psi

t_e = minimum body wall thickness adjacent to crotch for calculating thermal stresses [Fig. NB-3545.2(c)-1], in.

t_m = minimum body wall thickness as determined by NB-3541, in.

t_1, t_2 = minimum wall thicknesses from ANSI B16.34 corresponding to Listed Pressure Rating Class Indices p_{r1} and p_{r2} and inside diameter d_m, in.

T_b = thickness of valve wall adjacent to crotch region for calculating L_A and L_N [Fig. NB-3545.1(a)-1], in.

T_e = maximum effective metal thickness in crotch region for calculating thermal stresses [Fig. NB-3545.2(c)-1], in.

ΔT_i = specified range of fluid temperature, °F, where i = 1, 2, 3, . . ., n; used to evaluate normal valve usage (NB-3553)

T_r = thickness of body (run) wall adjacent to crotch for calculating L_A and L_N [Fig. NB-3545.1(a)-1], in.

$\Delta T'$ = maximum magnitude of the difference in average wall temperatures for walls of thicknesses t_e and T_e, resulting from 100°F/hr fluid temperature change rate, °F

ΔT_{fi} = a specified step change in fluid temperature, °F, where i=1, 2, 3, . . ., n; used to determine the fatigue acceptability of a valve body (NB-3554)

NB-3540 DESIGN OF PRESSURE RETAINING PARTS

NB-3541 General Requirements for Body Wall Thickness

The minimum wall thickness of a valve body is to be determined by the rules of NB-3542 or NB-3543.

NB-3542 Minimum Wall Thickness of Listed Pressure Rated Valves[18]

The wall thickness requirements for listed pressure rated valves apply also to integral body venturi valves. For a valve designed to a listed pressure rating of ANSI B16.34, the minimum thickness of its body wall, including the neck, is to be determined from ANSI B16.34, except that the inside diameter d_m shall be the larger of the basic valve body inside diameters in the region near the welding ends. Highly localized variations of inside diameter associated with weld preparation [NB-3544.8(a) and (b)] need not be considered for establishing minimum wall thickness t_m. In all such cases, however, the requirements of NB-3545.2(b)(6) shall be satisfied.

NB-3543 Minimum Wall Thickness of Valves of Nonlisted Pressure Rating[18]

To design a valve for Design Pressure and Design Temperature corresponding to other than one of the pressure ratings listed in the tables of ANSI B16.34, the procedure is the same as that of NB-3542 except that interpolation is required as follows.

(a) Based on the Design Temperature, linear interpolation between the tabulated temperature intervals shall be used to determine the listed pressure rating p_1, next below, and p_2, next above, the Design Pressure p_d corresponding to listed Pressure Rating Class Indices,[19] p_{r1} and p_{r2}, respectively.

(b) Determine the minimum wall thickness t_m corresponding to Design Loadings by:

$$t_m = t_1 + \left(\frac{p_d - p_1}{p_2 - p_1}\right) \times (t_2 - t_1)$$

[18]A listed pressure rated valve is one listed in the tables of ANSI B16.34. A nonlisted pressure rated valve is one whose Design Pressure and Temperature do not specifically appear in those tables (NB-3543).

[19]For all listed pressure ratings except Class 150, the Pressure Rating Class Index is the same as the pressure rating class designation. For Class 150 use 115 psi for the Pressure Rating Class Index.

(c) Determine the interpolated Pressure Rating Class Index p_r, corresponding to Design Loadings, by:

$$p_r = p_{r_1} + \left(\frac{p_d - p_1}{p_2 - p_1}\right) \times (p_{r_2} - p_{r_1})$$

NB-3544 Body Shape Rules

The rules of this paragraph constitute minimum requirements intended to limit the fatigue strength reduction factor, associated with local structural discontinuities in critical regions, to 2.0 or less. When smaller values of the fatigue strength reduction factor can be justified, it is permissible to use them.

NB-3544.1 Fillets for External and Internal Intersections and Surfaces

(a) Intersections of the surfaces of the pressure retaining boundary at the neck to body junction shall be provided with fillets of radius $r_2 \geq 0.3t_m$. Figure NB-3544.1(a)-1 illustrates such fillets.

(b) Corner radii on internal surfaces with $r_4 < r_2$ are permissible.

(c) Sharp fillets shall be avoided. When sharp discontinuities are convenient for ring grooves and similar configuration details, they shall be isolated from the major body primary and secondary stresses or modified as illustrated by Fig. NB-3544.1(c)-1.

NB-3544.2 Penetrations of Pressure Retaining Boundary. Penetrations of the pressure retaining boundary, other than the neck intersection, such as holes required for check valve shafts and drain or sensing lines, shall be located to minimize the compounding of normal body stresses.

NB-3544.3 Attachments. Attachments, such as lugs and similar protuberances, on the pressure retaining boundary shall be tapered to minimize discontinuity stresses (Fig. NB-3544.3-1). Reentrant angles shall be avoided. Attachments shall meet the requirements of NB-3135.

NB-3544.4 Body Internal Contours. Body internal contours in sections normal to the run or neck center lines shall be generally smooth in curvature, or so proportioned that the removal of unavoidable discontinuities, such as the valve seat, will leave generally smooth curvature.

NB-3544.5 Out-of-Roundness. Out-of-roundness in excess of 5% for sections of essentially uniform thickness shall be such that:

$$\frac{b}{t_b} + \frac{3}{4}\left(\frac{3b^2 - 2ab - a^2}{t_b^2}\right) + 1 \leq 1.5\left(\frac{S_m}{p_s}\right)$$

where

$2a$ = minor inside diameter, in.
$2b$ = major inside diameter, in.
t_b = thickness, in.

The ovality criterion can be satisfied by increasing the thickness locally, provided that the thickness variation is smoothly distributed. Out-of-roundness in excess of this limitation must be compensated for by providing reinforcement.

NB-3544.6 Doubly Curved Sections. Sections curved longitudinally with radius r_{Long}, as well as laterally with radius r_{Lat}, must be such that:

$$\frac{1}{r_{\text{Long}}} + \frac{1}{r_{\text{Lat}}} \geq \frac{4}{3d_m}$$

where d_m is the diameter used to establish the local wall thickness by NB-3541.

NB-3544.7 Flat Sections. Flat sections shall be sufficiently limited in extent so that arcuate sections having the same radius–thickness ratio as required by NB-3542 may be inscribed (Fig. NB-3544.7-1). The inscribed section may be less thick than the minimum thickness required by NB-3542, provided that its radius is proportionally smaller than the value used to determine the minimum required thickness. The method of NB-3544.6 above may be used to show additive support, but the denominator of the right side term must be reduced in the ratio of the thickness of the inscribed arcuate section to the minimum required thickness (NB-3542). If adequacy cannot be shown by the above rules, it is necessary to determine the stresses in the flat region experimentally to demonstrate adequacy for pressure induced stresses only, with internal pressure equal to the standard calculation pressure p_s.

NB-3544.8 Body End Dimensions

(a) Valve body contours at valve weld ends shall be in accordance with Fig. NB-4250-1, and, unless otherwise stated in the Design Specifications, with ANSI B16.34.

(b) Valve body transitions leading to valve weld ends shall be in accordance with ANSI B16.34.

(c) Flanged ends shall be in accordance with ANSI B16.34.

(d) Alignment tolerances given in Fig. NB-4233-1 shall apply to all auxiliary piping, such as drain lines, which begin or terminate at the valve.

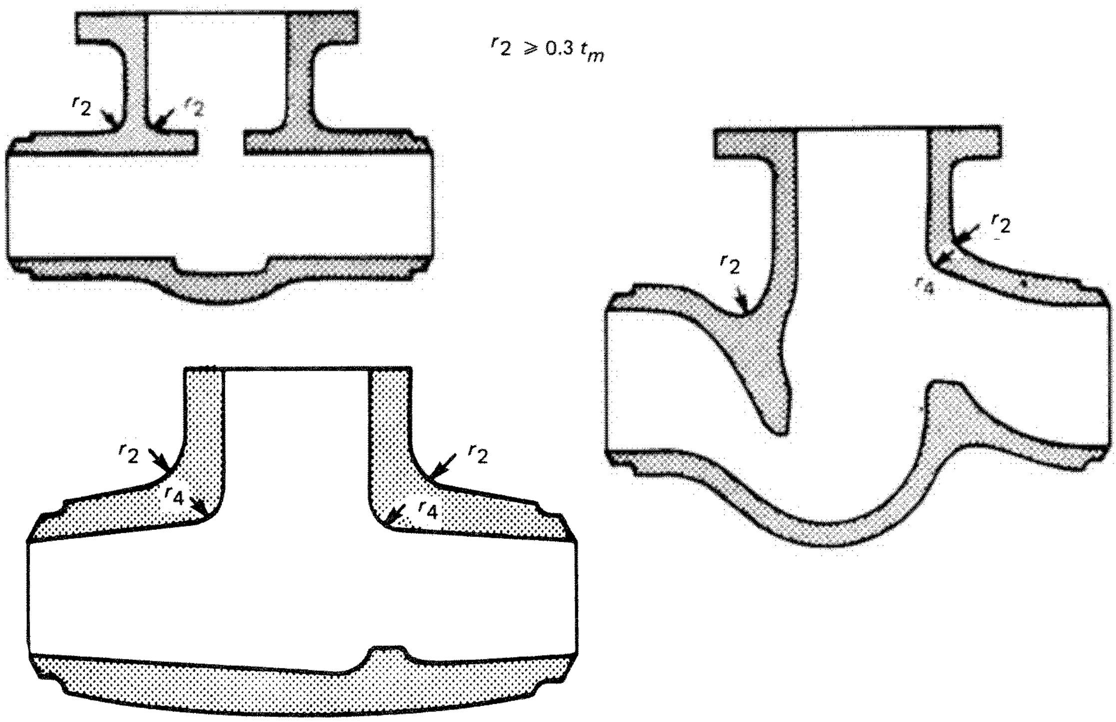

FIG. NB-3544.1(a)-1 FILLETS AND CORNERS

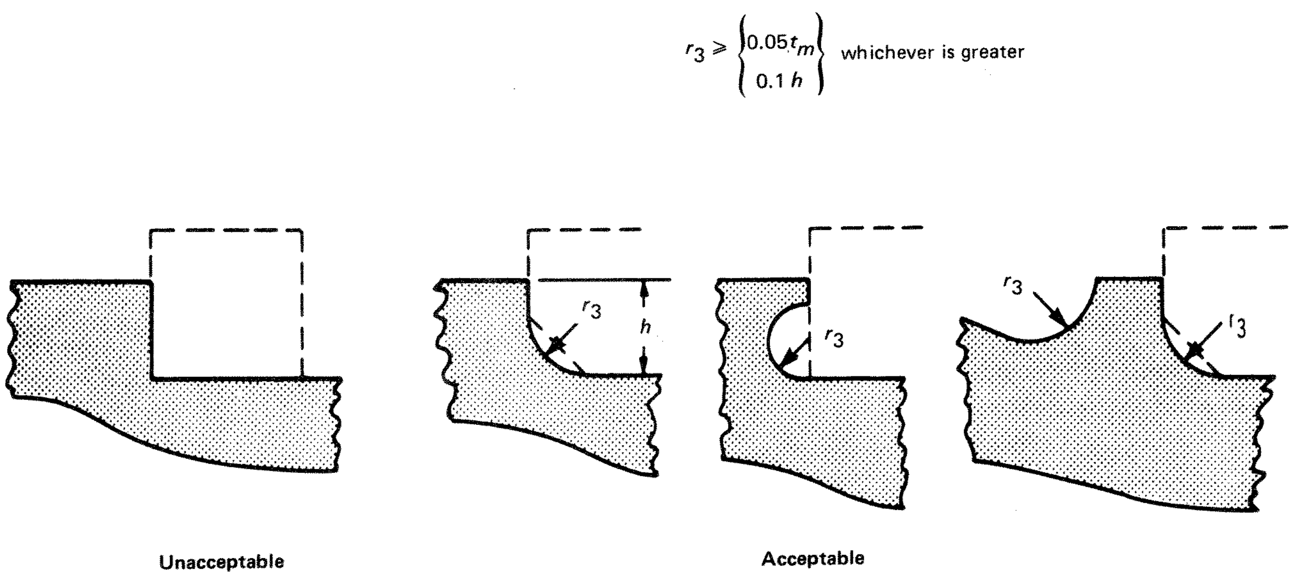

FIG. NB-3544.1(c)-1 RING GROOVES

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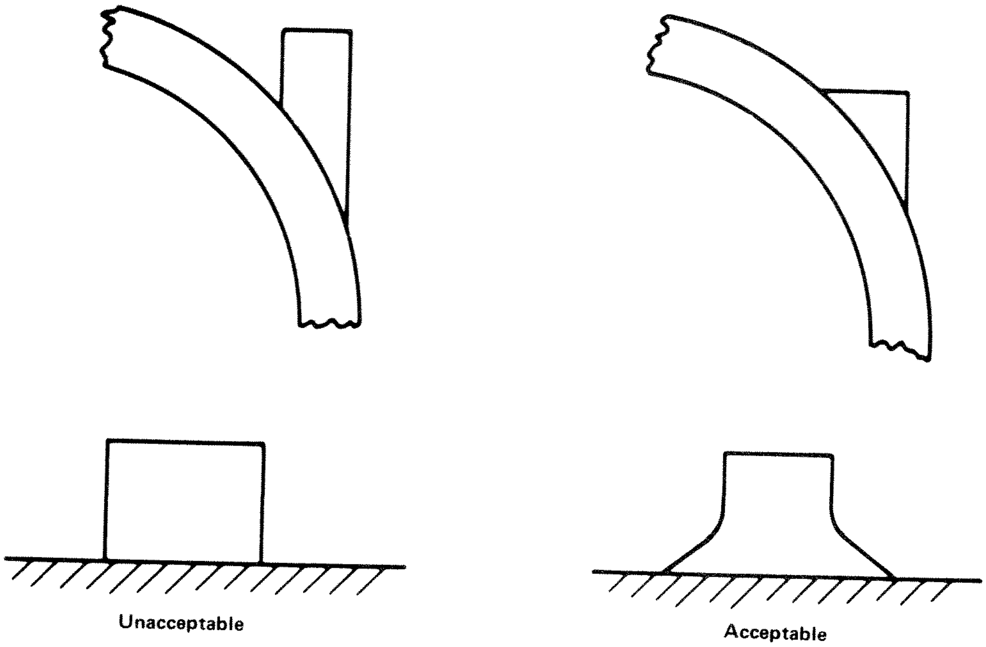

FIG. NB-3544.3-1 LUGS AND PROTUBERANCES

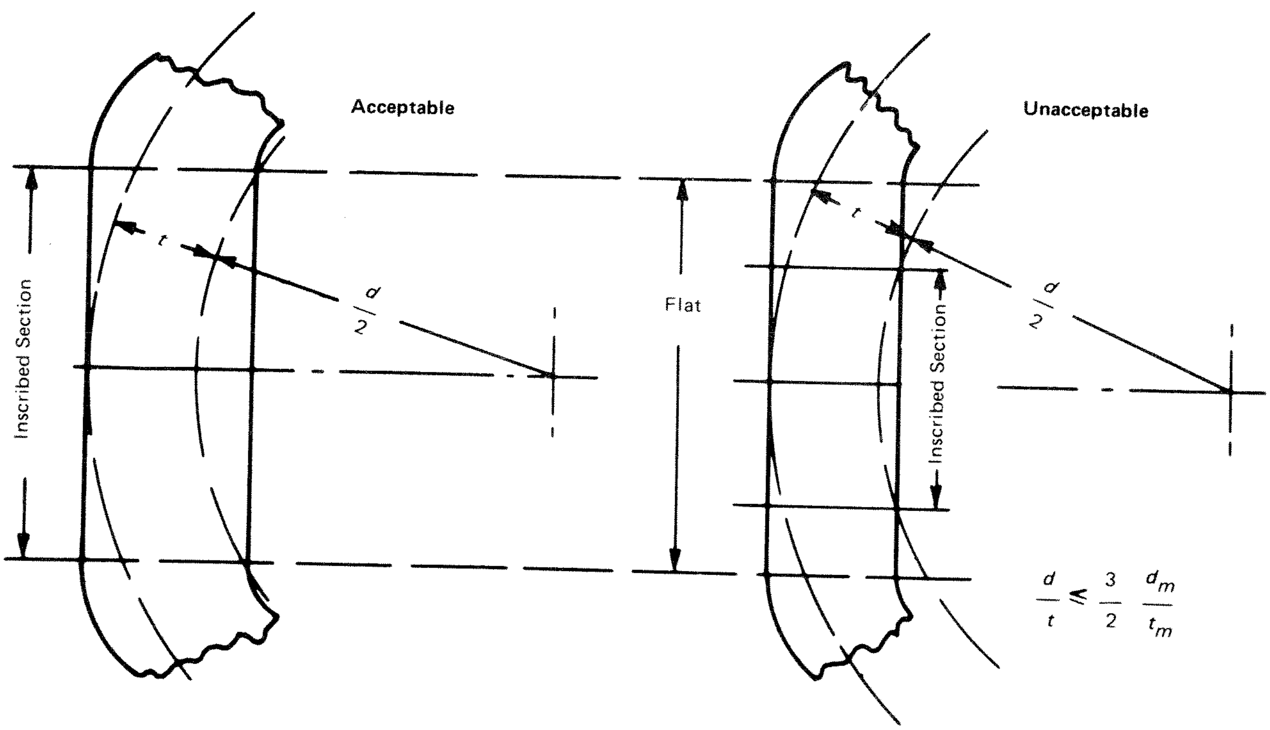

FIG. NB-3544.7-1 FLAT WALL LIMITATION

(e) For socket welding ends, valves of nominal size 2 in. and smaller for which the body cavity consists of cylindrically bored sections shall meet all of the following.

(1) d_m shall be the port drill diameter.

(2) The requirements of NB-3542 shall be satisfied.

(3) Socket welding end valves of nominal size greater than 2 in. shall not be used.

NB-3544.9 Openings for Auxiliary Connections. Openings for auxiliary connections, such as for drains, bypasses, and vents, shall meet the requirements of ANSI B16.34 and the applicable reinforcement requirements of NB-3330.

NB-3545 Body Primary and Secondary Stress Limits

The limits of primary and secondary stresses are established in the following subparagraphs.

NB-3545.1 Primary Membrane Stress Due to Internal Pressure. For valves meeting all requirements of this Subarticle, the most highly stressed portion of the body under internal pressure is at the neck to flow passage junction and is characterized by circumferential tension normal to the plane of center lines, with the maximum value at the inside surface. The rules of this paragraph are intended to control the general primary membrane stress in this crotch region. The Standard Calculation Pressure p_s to be used for satisfying the requirements of NB-3545 is found either directly or by interpolation from the tables in ANSI B16.34 as the pressure at 500°F for the given Pressure Rating Class Index p_r.

(a) In the crotch region, the maximum primary membrane stress is to be determined by the pressure area method in accordance with the rules of (1) through (6) below using Fig. NB-3545.1(a)-1.

(1) From an accurately drawn layout of the valve body, depicting the finished section of the crotch region in the mutual plane of the bonnet and flow passage center lines, determine the fluid area A_f and metal area A_m. A_f and A_m are to be based on the internal surface after complete loss of metal assigned to corrosion allowance.

(2) Calculate the crotch general primary membrane stress intensity:

$$P_m = \left(\frac{A_f}{A_m} + 0.5\right) p_s$$

The allowable value of this stress intensity is S_m for the valve body material at 500°F as given in Tables I-1.0.

(3) The distances L_A and L_N which provide bounds on the fluid and metal areas are determined as follows. Use the larger value of:

$$L_A = 0.5d - T_b$$

or

$$L_A = T_r$$

and use

$$L_N = 0.5r_2 + 0.354\sqrt{T_b(d + T_b)}$$

where the dimensions are as shown in Fig. NB-3545.1(a)-1.

In establishing appropriate values for the above parameters, some judgment may be required if the valve body is irregular as it is for globe valves and others with nonsymmetric shapes. In such cases, the internal boundaries of A_f shall be the lines that trace the greatest width of internal wetted surfaces perpendicular to the plane of the stem and pipe ends [Fig. NB-3545.1(a)-1 sketches (b), (d), and (e)].

(4) If the calculated boundaries for A_f and A_m, as defined by L_A and L_N, fall beyond the valve body [Fig. NB-3545.1(a)-1 sketch (b)], the body surface becomes the proper boundary for establishing A_f and A_m. No credit is to be taken for any area of connected piping which may be included within the limits of L_A and L_N. If the flange is included with A_m, the area of one bolt hole is to be subtracted for determining the net value of A_m.

(5) Except as modified below, web or fin-like extensions of the valve body are to be credited to A_m only to an effective length from the wall equal to the average thickness of the credited portion. The remaining web area is to be added to A_f [Fig. NB-3545.1(a)-1 sketch (b)]. However, to the extent that additional area will pass the following test, it may also be included in A_m. A line perpendicular to the plane of the stem and pipe ends from any points in A_m does not break out of the wetted surface but passes through a continuum of metal until it breaks through the outer surface of the body.

(6) In most cases, it is expected that the portions defined by A_m in the several illustrations of Fig. NB-3545.1(a)-1 will be most highly stressed. However, in the case of highly irregular valve bodies, it is recom-

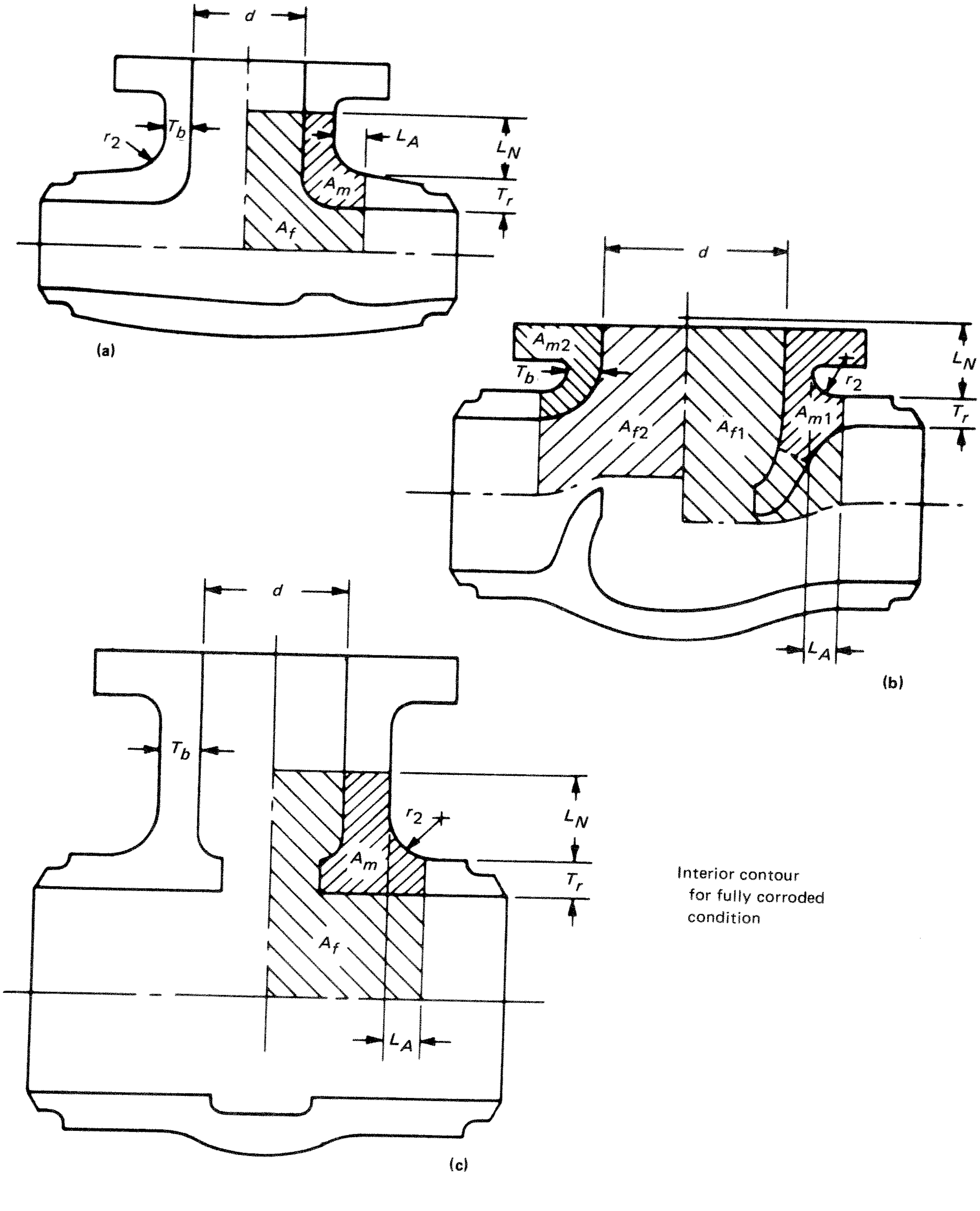

FIG. NB-3545.1(a)-1 PRESSURE AREA METHOD

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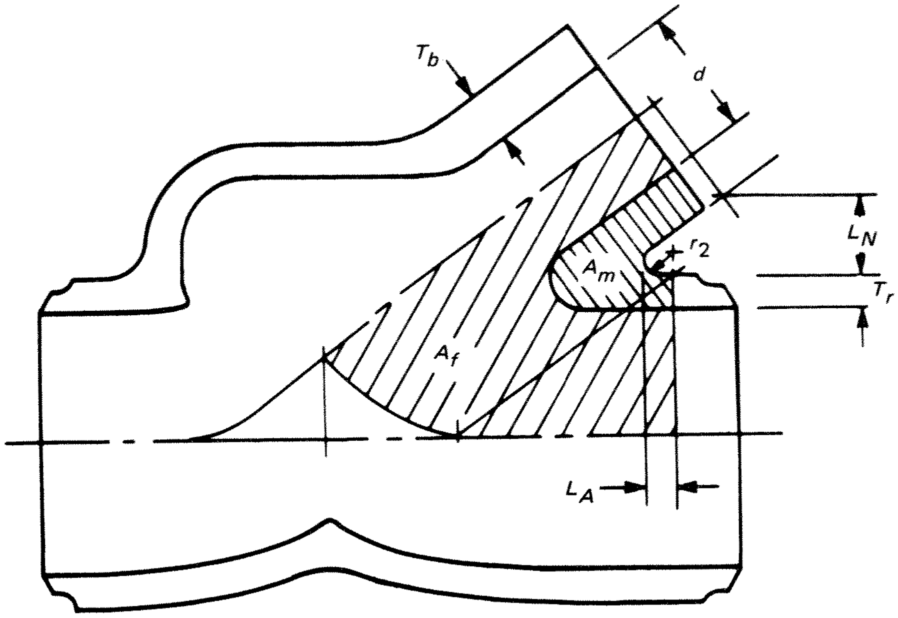
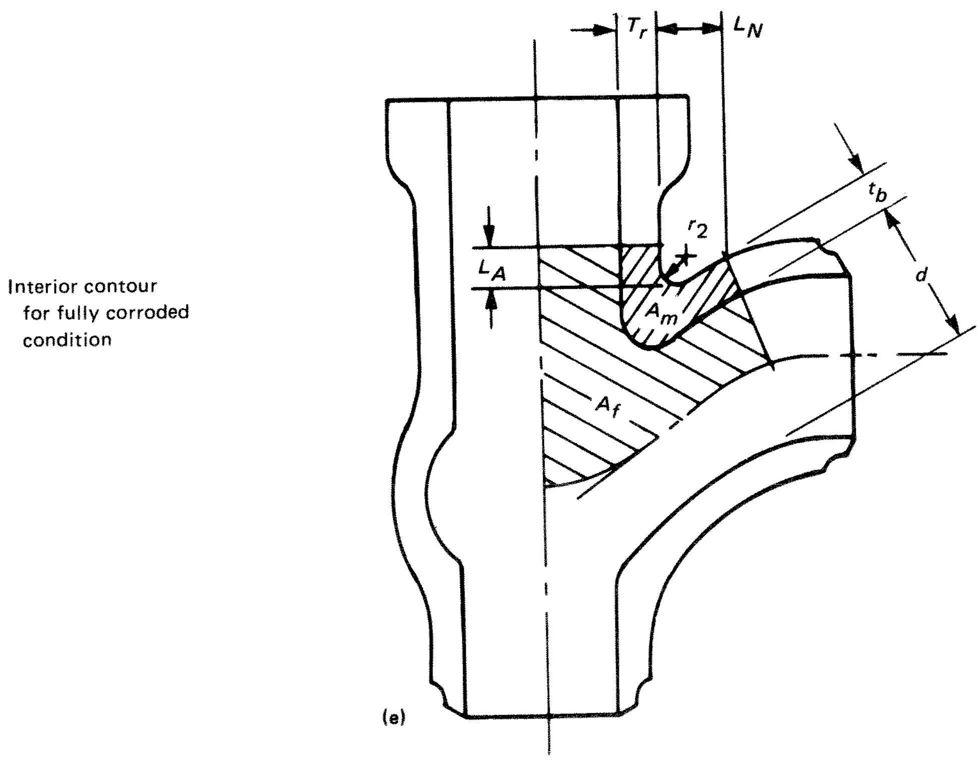


(d)



(e)

FIG. NB-3545.1(a)-1 PRESSURE AREA METHOD (CONT'D) mended that all sections of the crotch be checked to ensure that the largest value of P_m has been established considering both open and fully closed conditions.

(b) In regions other than the crotch, while the value of P_m calculated by NB-3545.1(a) will be the highest value of body general primary membrane stress for all normal valve types with typical wall proportioning, the designer is cautioned to review unusual body configurations for possible higher stresser regions. Suspected regions are to be checked by the pressure area method applied to the particular local body contours. The allowable value of this stress intensity is S_m for the valve body material at 500°F as given in Tables I-1.0.

NB-3545.2 Secondary Stresses. In addition to satisfying the criteria of NB-3541 through NB-3545.1, a valve body shall also satisfy the criterion that the range of primary plus secondary stresses S_n due to internal pressure, pipe reaction, and thermal effects shall not exceed $3S_m$ for the body material at 500°F, where Q_p, P_{eb}, and Q_{T3} are determined by the rules of this paragraph. That is:

$$S_n = Q_p + P_{eb} + 2\,Q_{T_3} \leq 3\,S_m$$

(a)(1) The body primary plus secondary stress Q_p due to internal pressure is to be determined by:

$$Q_p = C_p\left(\frac{r_i}{t_e} + 0.5\right)p_s$$

where the primary plus secondary pressure stress index C_p is equal to 3 and

p_s = Standard Calculation Pressure defined by NB-3545.1, psi

r_i = radius of a circle which circumscribes the inside wall contour in the crotch region, in.

t_e = an effective wall thickness at that location, in. (typically $t_e = T_r$) [Fig. NB-3545.1(a)-1]

In choosing an appropriate value for t_e, credit may be taken for general reinforcement material at the critical section but not for local fillets. Protuberances or ribs are not to be considered in determining r_i and t_e. Guidance is provided by Fig. NB-3545.2(a)-1 in which the illustrations correspond to the critical sections of the valve bodies of Fig. NB-3545.1(a)-1. The parameters r_i and t_e are intended to be representative of a tee, reinforced or unreinforced, with the general configuration of the valve body for which minor shape details associated with the valve function are ignored.

(2) For valve bodies with bonnet center lines other than perpendicular to the flow passage, the body stress Q_p due to internal pressure defined above must be multiplied by the factor C_a:

$$C_a = 0.2 + \frac{0.8}{\sin\alpha}$$

where

α = acute angle between the bonnet and flow passage center lines, deg.

(b) The secondary stress due to pipe reaction shall meet the criteria of (1) through (6) below to ensure the adequacy of the valve body for safely transmitting forces and moments imposed by the connected piping system.

(1) Based on the critical section A–A at the crotch, as illustrated by Fig. NB-3545.2(a)-1, calculate the value of P_{eb} where

(Bending load effect)

$$P_{eb} = \frac{C_b F_b S}{G_b}$$

The allowable value of P_{eb} is $1.5S_m$ for the valve body material at 500°F. Determination of S, F_b, C_b, and G_b required to calculate P_{eb} is to be in accordance with the requirements of (2) through (5) below.

(2) When the valve designer knows the material of the connected pipe, S may be calculated as the yield strength for the pipe material at 500°F. When the designer does not know the piping material or is designing a valve independently of a particular application, the value of S shall be taken as 30.0 ksi.

(3) Calculate F_b where

$$F_b = \frac{0.393\,d_e^{3}P_s}{20{,}000 - p_s}$$

but not less than:

(a) the section modulus of Schedule 40 pipe with the next larger inside diameter than d_e for $d_e < 10.02$ in.

(b) $0.295\,d_e^{2}$ for $d_e > 10.02$ in.

where d_e equals the inside diameter of the larger end of the valve body.

(4) Calculate the factor C_b:

$$C_b = 0.335\left(\frac{r}{t_e}\right)^{2/3}$$

When the results are less than 1.0, use $C_b = 1.0$.

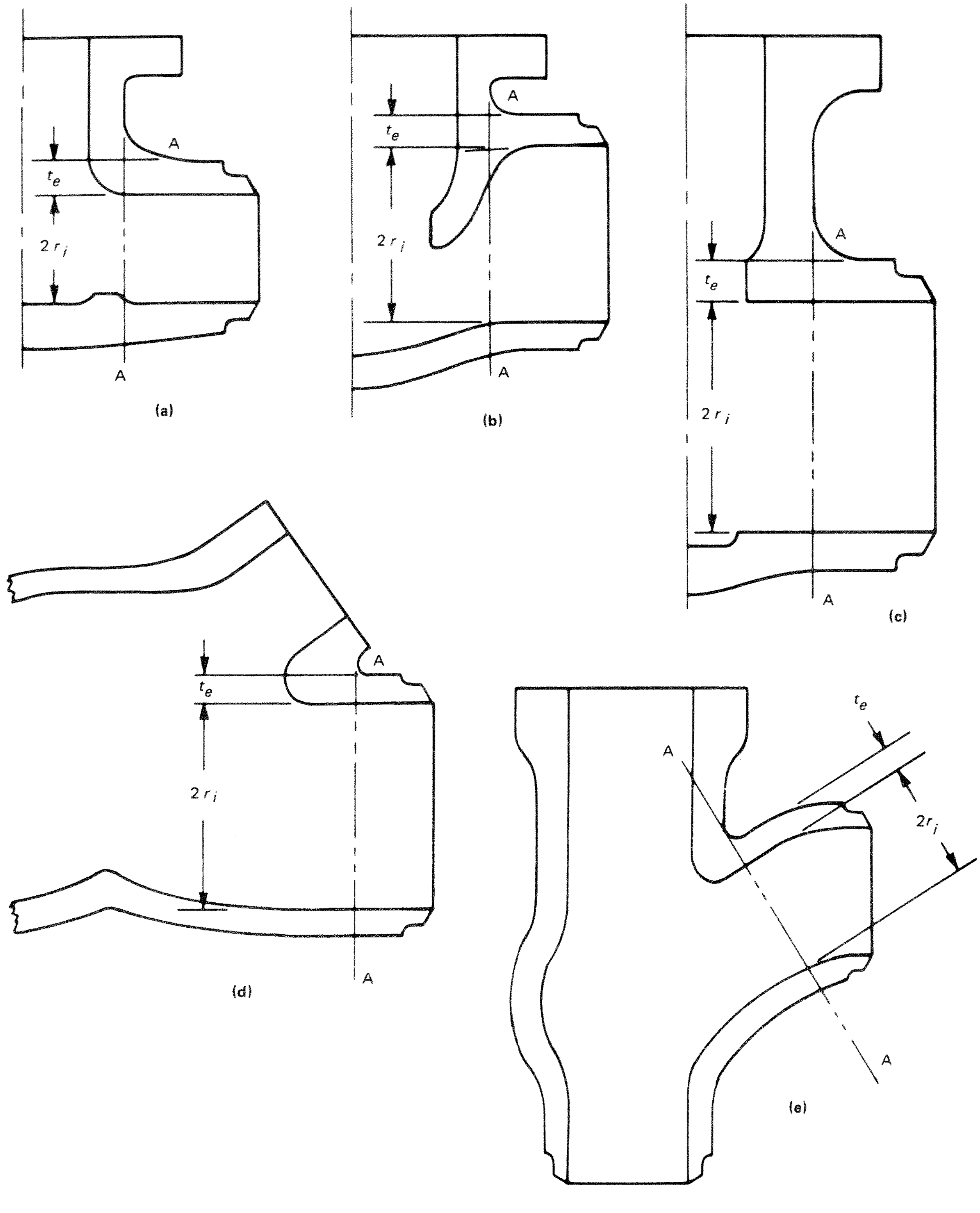
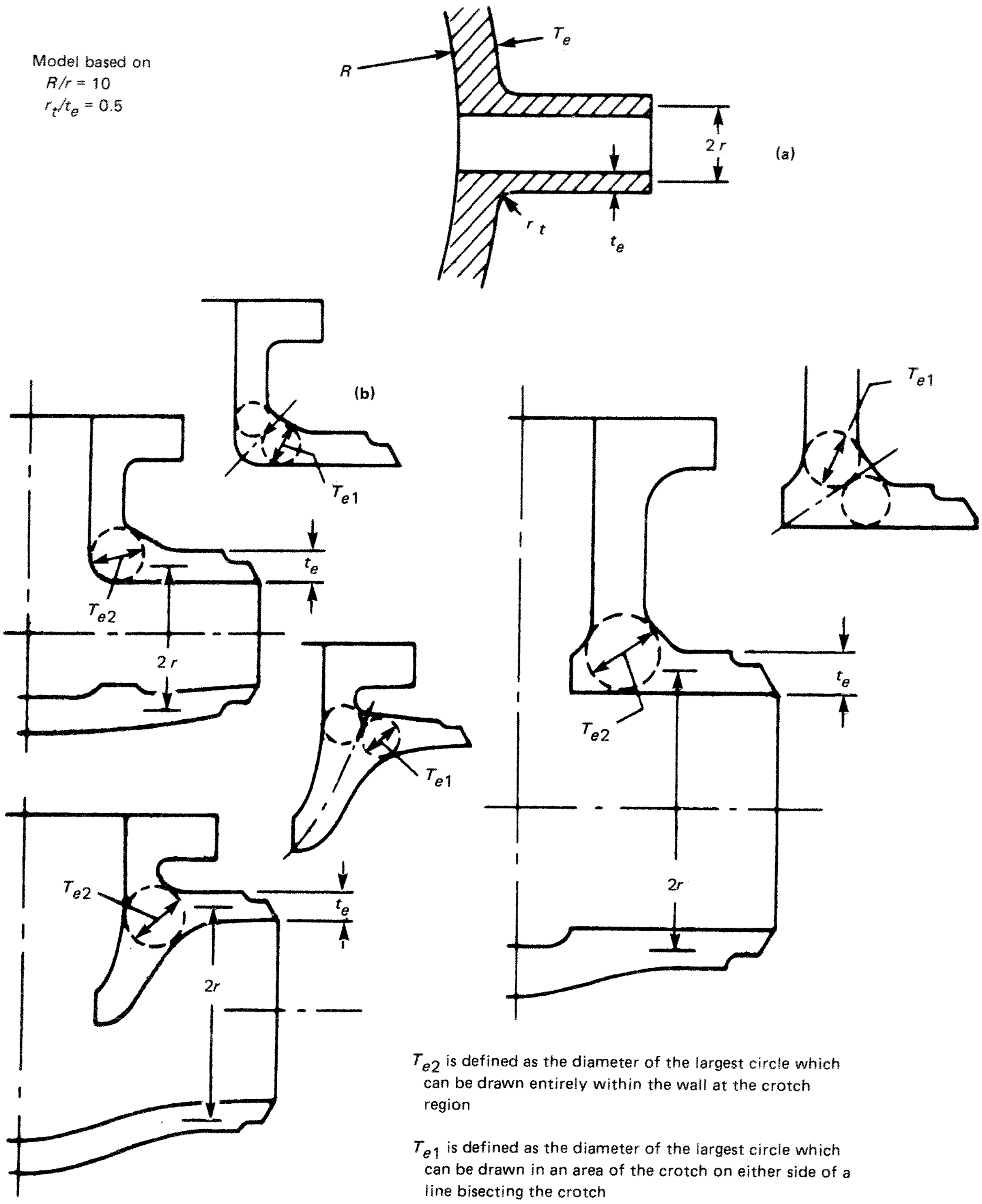


FIG. NB-3545.2(a)-1 CRITICAL SECTIONS OF VALVE BODIES



T_{e2} is defined as the diameter of the largest circle which can be drawn entirely within the wall at the crotch region

T_{e1} is defined as the diameter of the largest circle which can be drawn in an area of the crotch on either side of a line bisecting the crotch

For $T_{e1} < t_e$ as determined above, use $T_{e1} = t_e$

FIG. NB-3545.2(c)-1 MODEL FOR DETERMINING SECONDARY STRESS IN VALVE CROTCH REGION

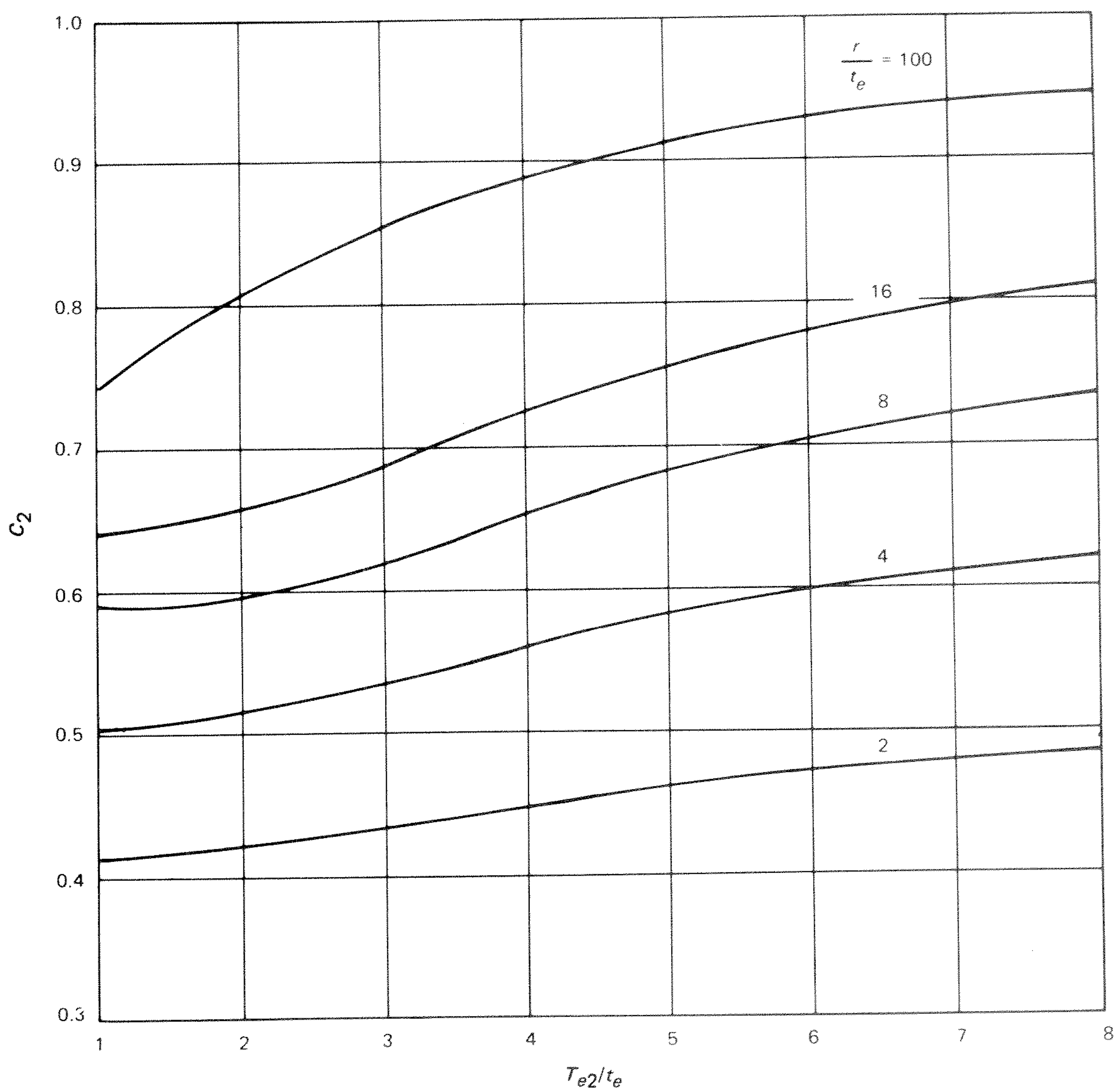


FIG. NB-3545.2(c)-3 THERMAL STRESS INDEX VS THICKNESS CONTINUITY RUN OR BRANCH

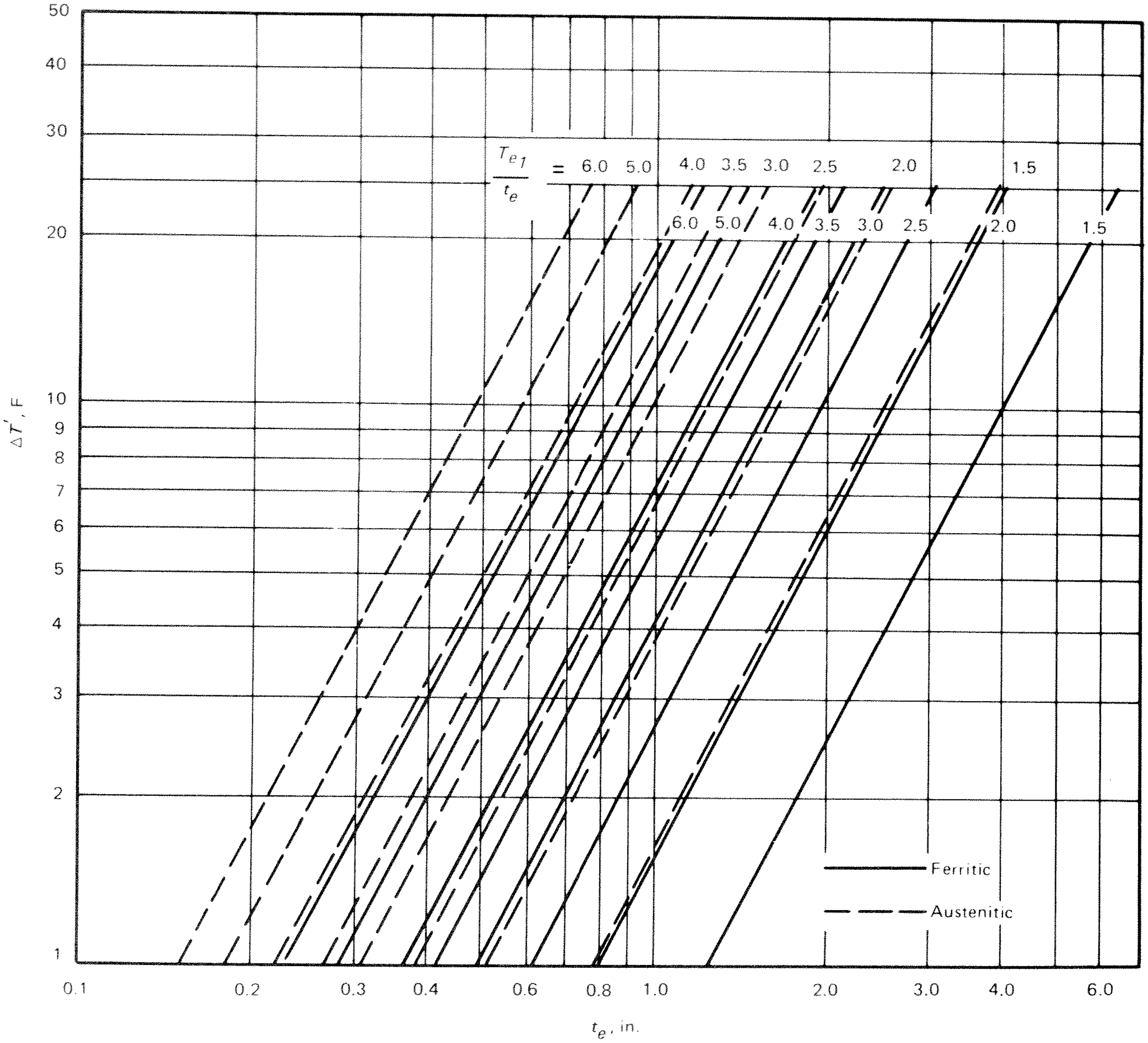


FIG. NB-3545.2(c)-4 MAXIMUM TEMPERATURE DIFFERENCE BETWEEN AVERAGE TEMPERATURE OF THE THICK WALL T_{e1} AND AVERAGE TEMPERATURE OF THE THIN WALL t_e AT 100°F/hr FOR FERRITIC AND AUSTENITIC MATERIALS

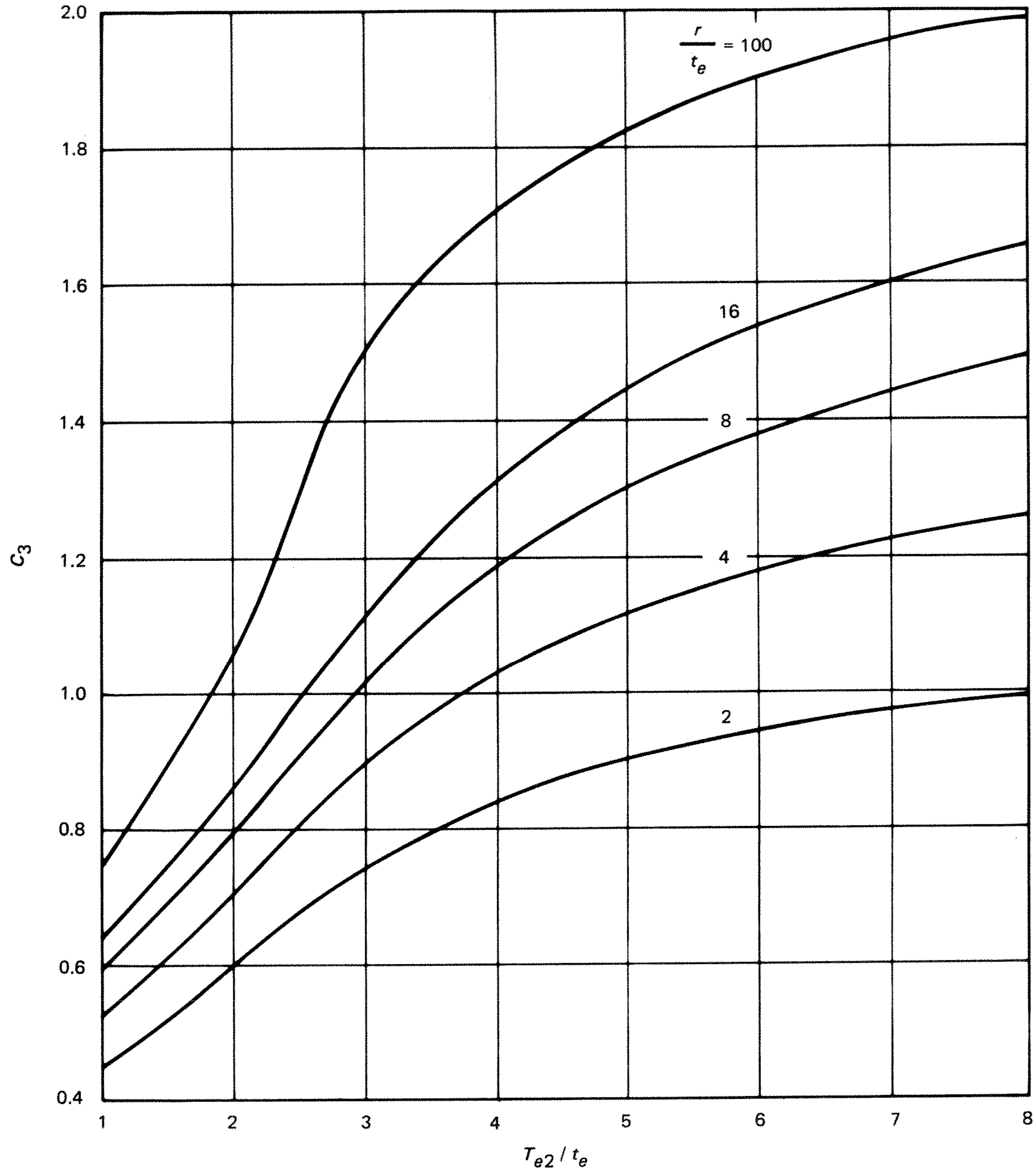


FIG. NB-3545.2(c)-5 SECONDARY STRESS INDEX VS THICKNESS CONTINUITY RUN OR BRANCH

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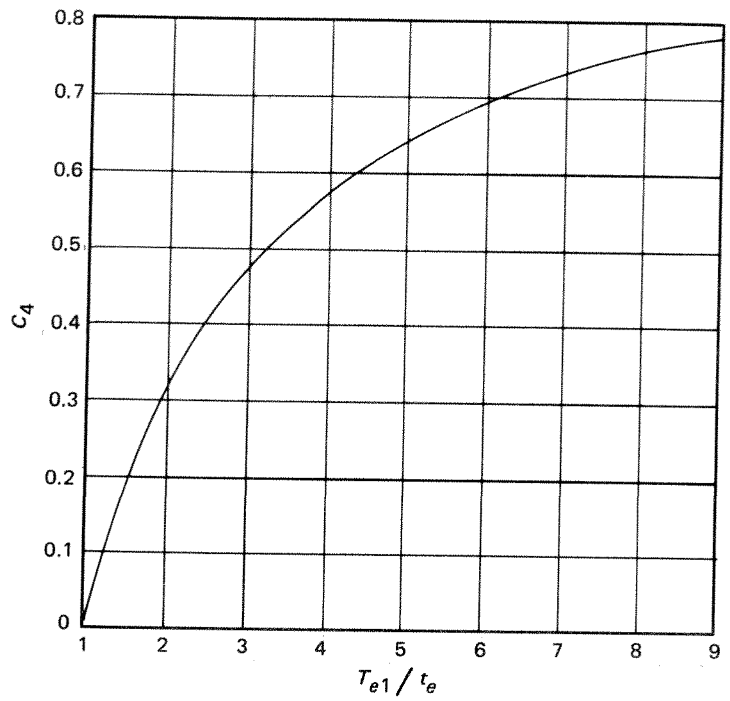


FIG. NB-3545.2(c)-6 C_4 VS T_{e1}/t_e

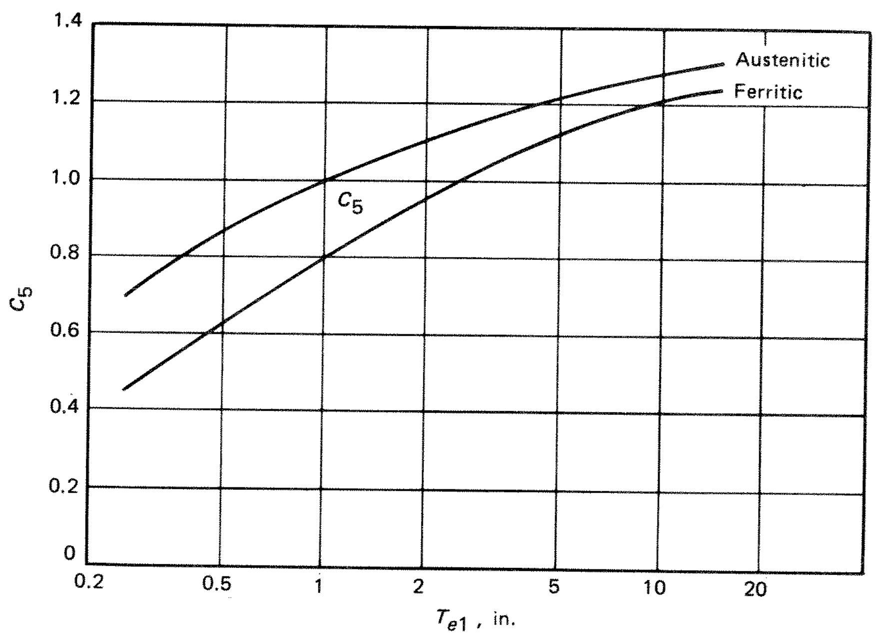


FIG. NB-3545.2(c)-7 STRESS INDEX FOR THERMAL FATIGUE

(5) The factor G_b is the section modulus $I/(r_i + t_e)$, in.^{3}, for bending at the plane through A–A about the axis perpendicular to the mutual plane of bonnet and body center lines, such as that axis which produces maximum bending stress at the corner of the crotch. The fiber stress at the outside surface is to be considered as governing in calculating G_b.

(6) When valves are to be applied in a venturi arrangement such that the connected pipe may be larger than that corresponding to the nominal size of the valve, it is necessary to base P_{eb} on the actual larger connected pipe. Such cases must be treated individually to ensure compliance with the secondary and fatigue stress criteria of this Subarticle. When the venturi arrangement is not fabricated by the N Certificate Holder, the Design Specifications shall include sufficient information to permit the N Certificate Holder to make this check.

(c) Thermal secondary stresses in the valve crotch region, resulting from through-wall temperature gradient and thickness variation (average temperature difference), are to be calculated on the basis of a continuous ramp change in fluid temperature at 100°F/hr using the model of Fig. NB-3545.2(c)-1 sketch (a). Figure NB-3545.2(c)-1 sketch (b) illustrates how r, T_{e_1}, T_{e_2}, and t_e are to be determined for the typically irregular crotch shape of valves. The thermal secondary stress components are to be determined in accordance with the following:

(1) Stress component Q_{T_1} which is the result of a through-wall temperature gradient is defined as:

$$Q_{T_1} = C_7(T_{e_1})^2$$

where

$C_7 = 100$ psi/in.^{2} for ferritic steels, or

$C_7 = 380$ psi/in.^{2} for austenitic steels

T_{e_1} is illustrated in Fig. NB-3545.2(c)-1.

(2) Stress component Q_{T_3} which is the membrane plus bending stress as a result of wall thickness variation is defined as:

$$Q_{T_3} = C_6 C_3 \Delta T'$$

where C_3 is found from Fig. NB-3545.2(c)-5.

NB-3545.3 Fatigue Requirements. The fatigue analysis requirements are satisfied provided the rules of this subparagraph and the rules of NB-3550 are met.

The calculated allowable number of cycles is $N_a \geq$ 2000 cycles, where N_a is determined from Figs. I-9.0 by entering the appropriate curve with S_a, with S_a defined as the larger value of S_{p_1} and S_{p_2} defined as follows:

$$S_{p_1} = \frac{2}{3} Q_p + \frac{P_{eb}}{2} + Q_{T_3} + 1.3 Q_{T_1}$$

$$S_{p_2} = 0.4 Q_p + \frac{K}{2}(P_{eb} + 2Q_{T_3})$$

The values of S_{p_1} and S_{p_2} are based on the values for Q_p, P_{eb}, Q_{T_1}, and Q_{T_3} found in accordance with the rules of NB-3545.2. K is the fatigue strength reduction factor associated with the external fillet at the crotch and is to be considered as 2.0 unless the designer can justify use of a smaller value.

NB-3546 Design Requirements for Valve Parts Other Than Bodies

NB-3546.1 Body-to-Bonnet Joints

(a) Bolted body-to-bonnet joints shall be designed in accordance with the pressure design rules of XI-3000, including the use of the appropriate allowable stress given in Tables I-7.0, or by the procedures of NB-3200, except fatigue analysis of bolts is not required.

(b) Body-to-bonnet joints other than bolted connections including joints of special or patented types for which specific standards do not exist may be used provided that the N Certificate Holder shall use methods of design and construction that will be as safe as otherwise required by the rules of this Subarticle for the following design conditions:

(1) Design Pressure equal to Standard Calculation Pressure p_s (NB-3545.1);

(2) calculation temperature of 500°F;

(3) thermal stresses based on most severe conditions resulting from continuous fluid temperature ramp increase or decrease at 100°F/hr;

(4) fatigue life at least 2000 cycles of startup/shutdown based on the above conditions with simultaneous increase or decrease of pressure and temperature.

NB-3546.2 Valve Disk. The valve disk shall be considered a part of the pressure retaining boundary. The primary membrane stress intensity shall not exceed S_m, and the primary bending stress intensity shall not exceed $1.5S_m$.

NB-3546.3 Other Valve Parts

(a) Valve stems, stem retaining structures, and other significantly stressed valve parts whose failure can lead to gross violation of the pressure retaining boundary shall be designed so that their primary stresses, based on pressure equal to the 100°F pressure rating and conservatively estimated or calculated additional loadings, where applicable, do not exceed S_m as tabulated in Tables I-1.0, or for materials not listed in Tables I-1.0, do not exceed two-thirds of the minimum specified yield strength or one-fourth of the minimum specified tensile strength, whichever is lower.

(b) Bypass piping shall be designed in accordance with the requirements of NB-3600. Unless otherwise stated in the valve Design Specifications, bypass piping design shall be the responsibility of the piping system designer.

(c) Valve designs, requiring solenoid plunger core tubes or electromagnetic position indicator core tubes, may substitute the rules of NB-3641.1 for the requirements of NB-3541, NB-3542, or NB-3543 for minimum wall thickness of the extension, provided that detailed calculations are prepared in accordance with NB-3200 at the 100°F valve pressure rating conditions, and covering all discontinuities in the core tube, including the cap end and attachment end, and all welds, including any dissimilar metal welds. These calculations shall be included in the Design Report (NB-3560). The calculations shall include the design loadings given in NB-3546.1(b)(1) through NB-3546.1(b)(4) along with any additional requirements given in the Design Specifications.

NB-3546.4 Fatigue Evaluation. When the Design Specifications include such service loadings that the valve is not exempted from fatigue analysis by the rules of NB-3222.4(d), it is recommended that consideration be given to the cyclic stress duty of the portions considered by NB-3546.

NB-3550 CYCLIC LOADING REQUIREMENTS

NB-3551 Verification of Adequacy for Cyclic Conditions

The adequacy of a valve for cyclic loading shall be verified in accordance with this Subsubarticle. Nonintegral seat rings attached to the valve body by partial penetration or fillet welds (NB-4433) are exempt from the fatigue analysis requirements of NB-3123.2, provided the seat rings are shouldered against the valve body; see Fig. NB-3544.1(c)-1.

NB-3552 Excluded Cycles

In satisfying the cyclic loading requirements, the following variations need not be considered:

(a) pressure variations less than $p_d/3$ for carbon and low alloy steels and less than $p_d/2$ for austenitic stainless steels;

(b) temperature variations less than 30°F;

(c) accident or maloperation cycles expected to occur less than five times (total) during the expected valve life;

(d) startup, shutdown cycles with temperature change rates of 100°F/hr or less, not in excess of 2000.

NB-3553 Fatigue Usage

The application of a valve conforming to NB-3512.1 is acceptable for cyclic loading conditions provided its fatigue usage I_t is not greater than 1.0 as evaluated in (a), (b), and (c) below.

(a) Consider fluid temperature changes not excluded by NB-3552 to occur instantaneously. Provided that these changes occur in one direction and recovery is at temperature change rates not in excess of 100°F/hr, the fatigue usage factor may be found by:

$$I_t = \sum \frac{N_{ri}}{N_i}$$

where N_{ri} is the required or estimated number of fluid temperature step changes ΔT_{fi} and N_i is found from Figs. I-9.1 and I-9.2.

(b) If both heating and cooling effects are expected at change rates exceeding 100°F/hr, the number of cycles are to be associated by temperature ranges ΔT_i. For example, assuming the following variations are specified:

$$\begin{aligned} &20 \text{ variations: } \Delta T_1 = 250 \text{ heating} \\ &10 \text{ variations: } \Delta T_2 = 150 \text{ cooling} \\ &100 \text{ variations: } \Delta T_3 = 100 \text{ cooling} \end{aligned}$$

lump the ranges of variation so as to produce the greatest effects as follows:

$$\begin{aligned} &10 \text{ cycles } \Delta T_{f1} = 250 + 150 = 400 \\ &10 \text{ cycles } \Delta T_{f2} = 250 + 100 = 350 \\ &90 \text{ cycles } \Delta T_{f3} = 100 \end{aligned}$$

(c) Pressure fluctuations not excluded by NB-3552 are to be included in the cyclic load calculations. The full range of pressure fluctuation from the normal condition to the condition under consideration shall be represented by Δp_i in NB-3554.

NB-3554 Cyclic Stress Calculations

A valve conforming to NB-3512.1 shall be qualified by the procedure of (a) through (d) below.

(a) The following criterion shall be met by the greatest temperature range:

$$Q_p[\Delta p_{f(\max)}/p_s] + C_6 C_2 C_4 \Delta T_{f(\max)} < 3S_m$$

where $\Delta T_f(\max)$ is the largest lumped temperature range obtained using the methods of NB-3553(b), and $\Delta p_f(\max)$ is the largest range of pressure fluctuation associated with $\Delta T_f(\max)$.

(b) Calculate:

$$S_{n(\max)} = Q_p[\Delta p_{f(\max)}/p_s] + C_6 C_3 C_4 \Delta T_{f(\max)}$$

Provided that $S_n(\max) \leq 3S_m$, calculate the fatigue stresses for each cyclic loading condition as follows:

$$S_i = \tfrac{4}{3} Q_p(\Delta p_{fi}/p_s) + C_6(C_3 C_4 + C_5)\Delta T_{fi}$$

Determine the allowable number of cycles N_i for each loading condition by entering Figs. I-9.1 and I-9.2 with $S_i/2$, and determine the fatigue usage by NB-3553(a).

(c) If $S_n(\max)$ is greater than $3S_m$ but less than $3mS_m$, the value of $S_i/2$ to be used for entering the design fatigue curve is to be found by multiplying S_i by K_e, where:

$$K_e = 1.0 + \frac{(1-n)}{n(m-1)}\left(\frac{S_n}{3S_m} - 1\right)$$

and where the values of the material parameters m and n are as given in Table NB-3228.3(b)-1.

(d) If $S_n(\max)$ is greater than $3mS_m$, use $K_e = 1/n$.

NB-3560 DESIGN REPORTS

NB-3561 General Requirements

The certified Design Reports listed in this paragraph meet the requirements of NCA-3550 for the Design Report.

NB-3562 Design Report for Valves Larger Than 4 in. Nominal Pipe Size

A Design Report shall be prepared in sufficient detail to show that the valve satisfies the requirements of NB-3512. For a valve designed in accordance with NB-3512.1, the Design Report shall show that the applicable requirements of NB-3530, NB-3541 through NB-3546.2, and NB-3550 have been met. It is not necessary to write a special Design Report based on specified Design Pressure and Design Temperature when they are within the pressure–temperature rating and when supplementary information or calculations are also provided, as necessary, to complete the report for a specific application, such as the thermal cyclic duty evaluation of NB-3550. A report submitted demonstrating a design for loadings more severe than the specified loadings is also acceptable.

NB-3563 Design Report Requirements for 4 in. and Smaller Nominal Pipe Size Valves

For valves whose inlet piping connection is nominally 4 in. or smaller, the Design Report shall include details to show that the requirements of NB-3513 have been met.

NB-3590 PRESSURE RELIEF VALVE DESIGN

NB-3591 Acceptability

NB-3591.1 General. The rules of this Subsubarticle constitute the requirements for the design acceptability of spring-loaded pressure relief valves. The design rules for pilot operated and power actuated pressure relief valves are covered by NB-3500. The rules of this Subsubarticle cover the pressure retaining integrity of the valve inlet and outlet connections, nozzle, disk, body structure, bonnet (yoke), and body-to-bonnet (yoke) bolting. The rules of this Subsubarticle also cover other items such as the spring, spindle (stem), spring washers, and set pressure adjusting screw. The rules of this Subsubarticle do not apply to guides, control ring, bearings, set screws, and other nonpressure-retaining items. Figures NB-3591.1-1 and NB-3591.1-2 are illustrations of typical pressure relief valves.

NB-3591.2 Definitions. The definitions for pressure relief valve terms used in this Subsubarticle are given in ANSI B95.1, Terminology for Pressure Relief Devices, and also in NB-7000. Pressure relief valves characteristically have multipressure zones within the valve, that is, a primary pressure zone and a secondary pressure zone as illustrated by Figs. NB-3591.1-1 and NB-3591.1-2.

NB-3591.3 Acceptability of Small Liquid Relief Valves. Liquid pressure relief valves meeting the requirements of NB-7000 and having an inlet piping connection 2 in. nominal pipe size and under shall comply with the minimum wall thickness requirements of NB-3542 or NB-3543 for the applicable pressure zone. Flange end ratings of NB-3531.1 shall be used regardless of end connection. The applicable design requirements of this Subsubarticle covering the nozzle, disk, and bonnet shall apply. The analyses of NB-3544, NB-3545, and NB-3550 do not apply.

NB-3591.4 Acceptability of Safety and Safety Relief Valves. The design shall be such that the requirements of this Subsubarticle are met.

NB-3592 Design Considerations

NB-3592.1 Design Conditions. The general design requirements of NB-3100 are applicable, with consideration for the design conditions of the primary and secondary pressure zones. The design pressure of the Design Specification shall be used for the applicable zones.

In case of conflict between NB-3100 and NB-3590, the requirements of NB-3590 shall apply. Mechanical loads for both the closed and the open (full discharge) positions shall be considered in conjunction with the service conditions. In addition, the requirements of NB-7000 shall be met.

NB-3592.2 Stress Limits for Specified Service Loadings

(a) Stress limits for Level A and B Service Loadings shall be as follows:

(1) the primary membrane stress intensity shall not exceed S_m;

(2) the primary membrane stress intensity plus primary bending stress intensity shall not exceed $1.5S_m$;

(3) substantiation by analysis of localized stresses associated with contact loading of bearing or seating surfaces is not required;

(4) the values of S_m shall be in accordance with Tables I-1.1, I-1.2, and I-1.3.

(b) Stress limits for Level C Service Loadings shall be as follows:

(1) the primary membrane stress intensity shall not exceed $1.5S_m$ (NB-3600);

(2) the primary membrane stress intensity plus primary bending stress intensity shall not exceed $1.8S_m$ (NB-3526.2);

(3) the rules of NB-3526.3 must be satisfied.

(c) Stress limits for Level D Service Loadings shall be as follows:

(1) the guidelines of Appendix F may be used in evaluating these conditions.

(d) These requirements for the acceptability of valve design are not intended to ensure the functional adequacy of the valve. However, the Designer is cautioned that the requirements of NB-7000 relative to set pressure, lift, blowdown, and closure shall be met.

NB-3592.3 Earthquake. The rules of this Subsubarticle consider that under earthquake loadings the piping system or vessel nozzle, rather than the valve body, will be limiting. Pressure relief valves have extended structures and these structures are essential to maintaining pressure integrity. An analysis, when required by the Design Specification, shall be performed based on static forces resulting from equivalent earthquake acceleration acting at the centers of gravity of the extended masses. Classical bending and direct stress equations, where free body diagrams determine a simple stress distribution that is in equilibrium with the applied loads, may be used.

NB-3593 Special Design Rules

NB-3593.1 Hydrostatic Test. Hydrostatic testing shall be performed in accordance with NB-3531.2(e).

NB-3593.2 Marking. In addition to the marking required by NCA-8220 and NB-7000, the secondary Design Pressure shall be marked on the valve or valve nameplate.

NB-3594 Design of Pressure Relief Valve Parts

NB-3594.1 Body. The valve body shall be analyzed with consideration for the specific configuration of the body and the applicable pressure zone and loadings. The design shall take into consideration the adequacy of the inlet flange connection, the outer flange connection, and the body structural configuration. In valve designs where the outlet flange is an extension of the bonnet, the bonnet design shall conform to all rules of body design. The body shall be designed in accordance with the rules of NB-3540 through NB-3550. The design adequacy of the inlet and outlet flanges shall be determined using the rules of NB-3658. Flanges shall conform to the applicable pressure–temperature ratings of NB-3531.1 and shall meet the interface dimensions of ANSI B16.5.

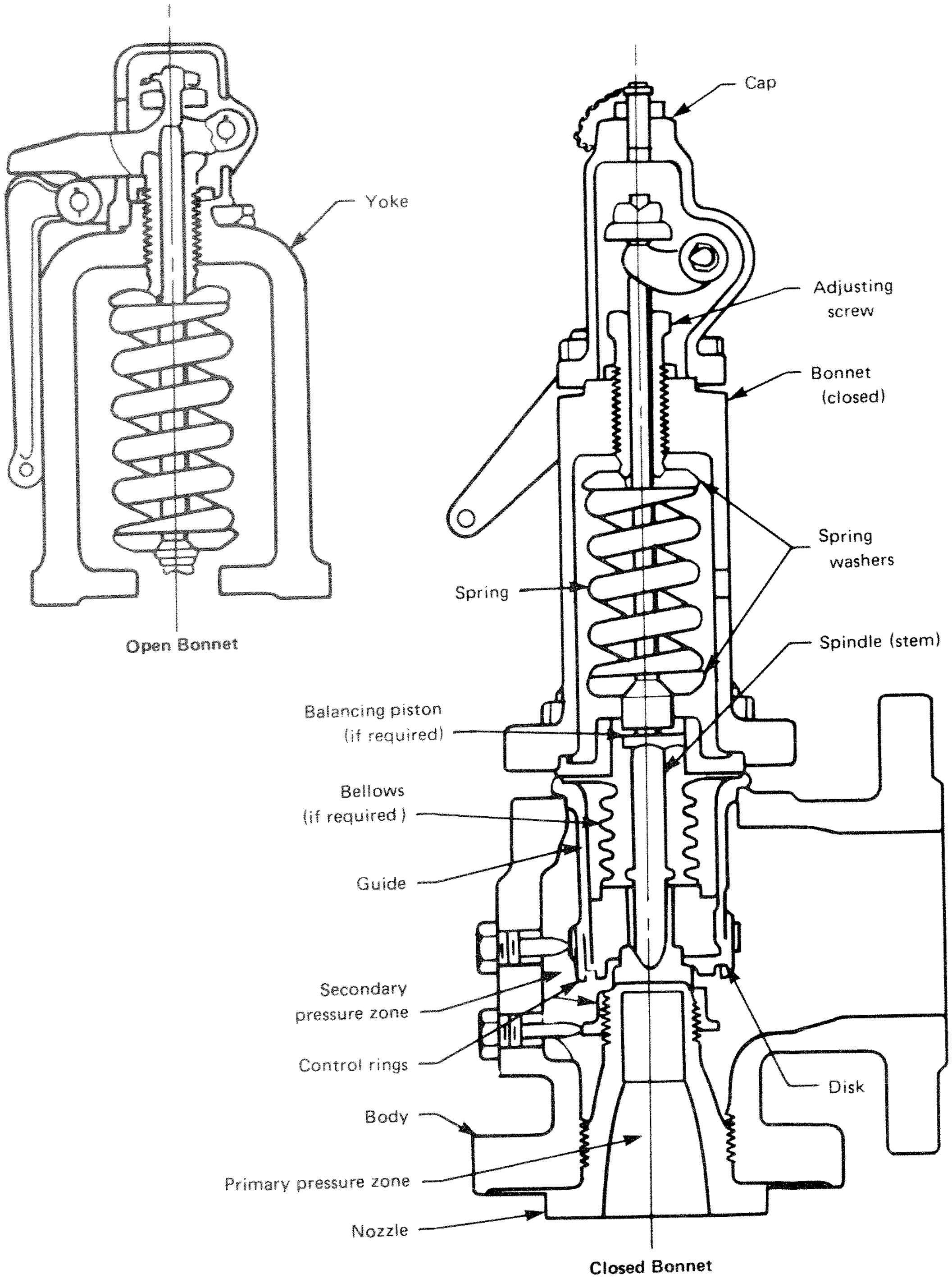
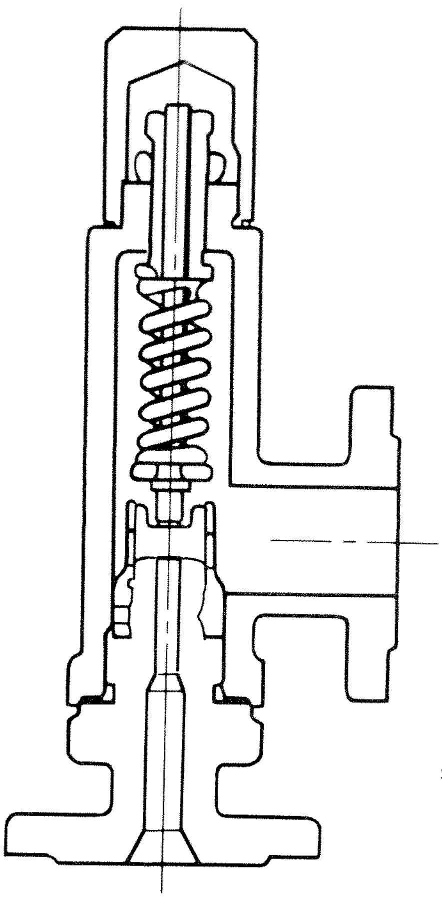


FIG. NB-3591.1-1 TYPICAL PRESSURE RELIEF DEVICES



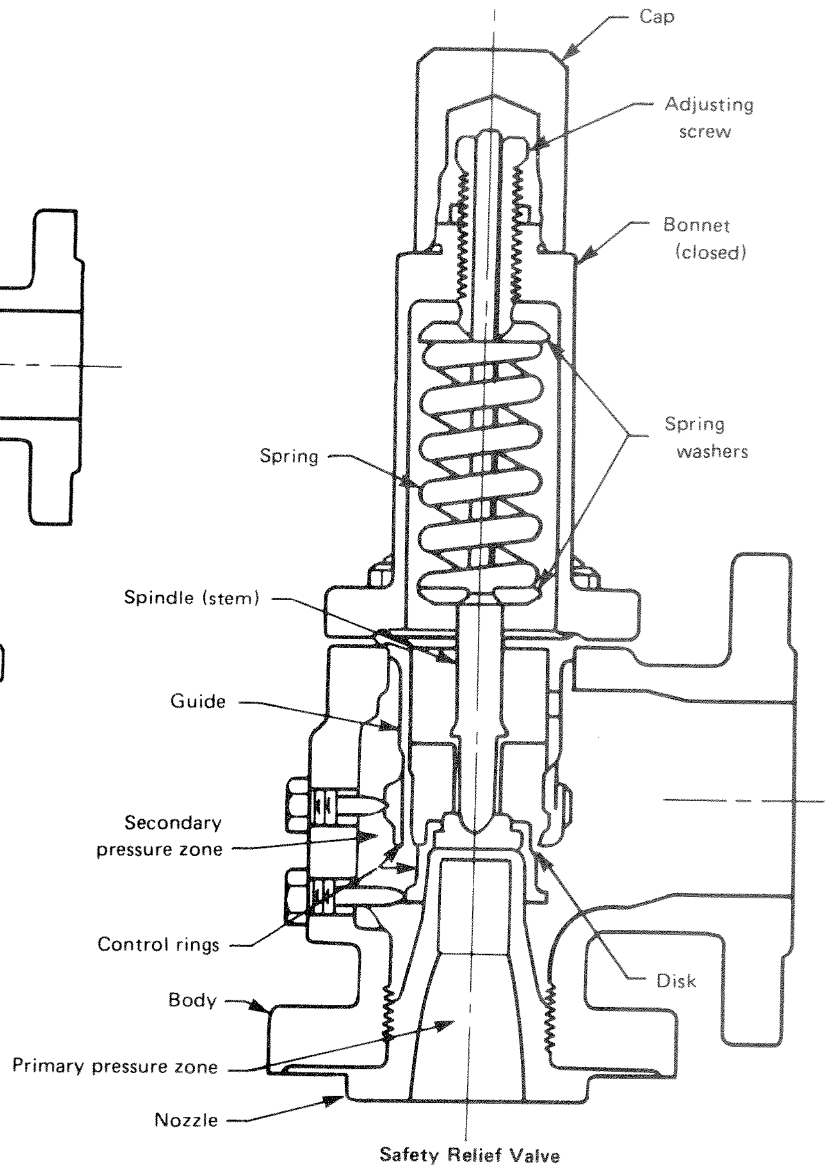


FIG. NB-3591.1-2 TYPICAL PRESSURE RELIEF AND SAFETY RELIEF DEVICES

NB-3594.2 Bonnet (Yoke). The bonnet (yoke) may be analyzed using classic bending and direct stress formulae, with appropriate free body diagrams. The general primary membrane stress intensity and the general primary membrane plus primary bending stress intensity shall be determined and shall not exceed the stress limits of NB-3592.2.

NB-3594.3 Nozzle. The nozzle shall be analyzed in accordance with the applicable rules of NB-3540 and NB-3550, with a basic analytical model configuration as shown in Fig. NB-3594.3-1. The sections of the nozzle where dimensions are limited by the flow capacity and the operational control requirements may be considered as simple cylindrical sections. The minimum wall thickness of these sections shall be determined in accordance with NB-3324.1. These requirements are not applicable to the transition region to the seat contacting area of the nozzle, defined by L in Fig. NB-3594.3-1, provided dimension L is less than the nominal wall thickness t_1.

NB-3594.4 Body-to-Bonnet Joint. The body-to-bonnet joint shall be analyzed in accordance with NB-3546.1.

NB-3594.5 Disk. The valve disk shall satisfy the requirements of NB-3546.2.

NB-3594.6 Spring Washer. The average shear stress shall not exceed $0.6S_m$. The primary bending stress intensity shall not exceed the stress limits of NB-3592.2.

NB-3594.7 Spindle (Stem). The general primary membrane stress intensity shall not exceed the stress limits of NB-3592.2.

NB-3594.8 Adjusting Screw. The adjusting screw shall be analyzed for thread shear stress in accordance with the method of ANSI B1.1 and this stress shall not exceed $0.6S_m$. The general primary membrane stress intensity of the adjusting screw shall not exceed the stress limits of NB-3592.2, based on the root diameter of the thread.

NB-3594.9 Spring. The valve spring shall be designed so that the full lift spring compression shall be no greater than 80% of the nominal solid deflection. The permanent set of the spring (defined as the difference between the free height and height measured a minimum of 10 min after the spring has been compressed solid three additional times after presetting at room temperature) shall not exceed 0.5% of the free height.

NB-3595 Design Report

NB-3595.1 General Requirements. A Design Report shall be prepared in sufficient detail to show that the valve satisfies the rules of this Subsubarticle and NCA-3550.

NB-3600 PIPING DESIGN

NB-3610 GENERAL REQUIREMENTS

NB-3611 Acceptability

The requirements for acceptability of a piping system are given in the following subparagraphs.

NB-3611.1 Stress Limits. The design shall be such that the stresses will not exceed the limits described in NB-3630 except as provided in NB-3611.2.

NB-3611.2 Acceptability When Stresses Exceed Stress Limits. When the stresses as determined by the methods given in NB-3630 exceed the limits thereof, the design can be accepted, provided it meets the requirements of NB-3200. The rules of NB-3630 meet all the requirements of NB-3200.

NB-3611.3 Conformance to NB-3600. In cases of conflict between NB-3100 and NB-3600, the requirements of NB-3600 shall apply.

NB-3611.4 Dimensional Standards. For the applicable year of issue of all dimensional standards referred to in NB-3600, see Table NB-3132-1.

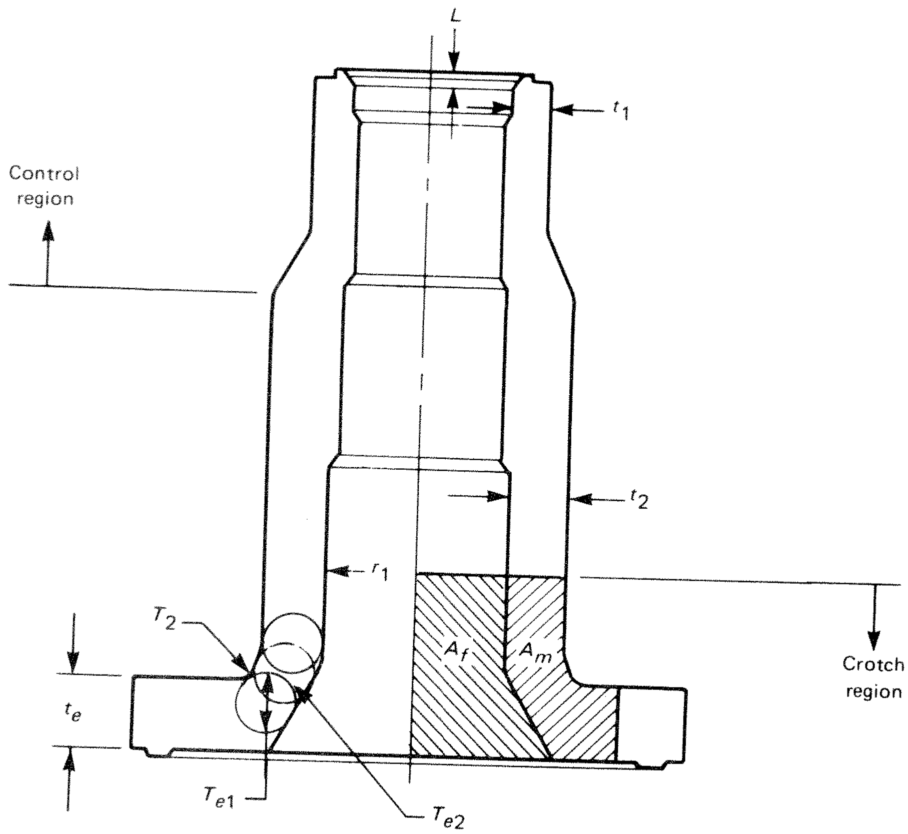
NB-3611.5 Prevention of Nonductile Failure. The requirements for prevention of nonductile failure [NB-3211(d)] may be considered to have been met under all service loadings at temperatures equal to or higher than the lowest service temperatures, if materials meet the requirements of NB-2300.

NB-3612 Pressure–Temperature Ratings

NB-3612.1 Standard Piping Products

(a) When standard piping products are used, the pressure ratings given as functions of temperature in the appropriate standards listed in Table NB-3132-1 shall not be exceeded. In addition, the requirements of NB-3625 shall be met. When established pressure ratings of standard products do not extend to the upper temperature limits for the material, the ratings between those established and the upper temperature limit may be determined in accordance with NB-3649.

(b) When the adequacy of the pressure design of a standard product is established by burst tests as per-



t_1, t_2 = nozzle wall thickness, in.
t_e = minimum body wall thickness adjacent to crotch, in.
T_{e1} = maximum effective thickness in crotch region, in.
T_{e2} = effective crotch wall thickness, in.
r_1 = inside radius at crotch region, in.
r_2 = outside fillet radius at crotch
L = length of seat transition region, in.
A_f = fluid area, in.^{2}
A_m = metal area, in.^{2}

FIG. NB-3594.3-1 VALVE NOZZLE

mitted in NB-3649 (ANSI B16.9, Section 8), the manufacturer of the product shall maintain a record of burst tests conducted to ensure adequacy of product and shall so certify. Such records shall be available to the purchaser.

NB-3612.2 Piping Products Without Specific Ratings. If piping products are used for which methods of construction are not covered by this Subsection, the manufacturer of the product shall use methods of construction that will be as safe as otherwise provided by the rules of this Subsection. When products are used for which pressure–temperature ratings have not been established by the standards listed in Table NB-3132-1, the products shall be designed and tested in accordance with NB-3640. The manufacturer's recommended pressure–temperature ratings shall not be exceeded.

NB-3612.4 Considerations for Local Conditions and Transients

(a) When piping systems operating at different pressures are connected by a valve or valves, the valve or valves shall be designed for the higher pressure system requirements of pressure and temperature. The lower pressure system shall be designed in accordance with (1), (2), or (3) below

(1) The requirements of the pressure system shall be met.

(2) Pressure relief devices or safety valves shall be included to protect the lower pressure system in accordance with NB-7311.

(3) Ensure compliance with all the conditions of (a) through (e) below.

(a) Redundant check or remote actuated valves shall be used in series at the interconnection, or a check in series with a remote actuated valve.

(b) When mechanical or electrical controls are provided, redundant and diverse controls shall be installed which will prevent the interconnecting valves from opening when the pressure in the high pressure system exceeds the Design Pressure of the low pressure system.

(c) Means shall be provided such that operability of all components, controls, and interlocks can be verified by test.

(d) Means shall be provided to ensure that the leakage rate of the interconnecting valves does not exceed the relieving capacity of the relief devices on the low pressure system.

(e) Adequate consideration shall be given to the control of fluid pressure caused by heating of the fluid trapped between two valves.

The low pressure system relieving capacity may be determined in accordance with NB-7311, on the basis of interconnecting valve being closed but leaking at a specified rate, when (3)(a) to (3)(e) above are met. The pressure relief devices or safety valves shall adjoin or be as close as possible to the interconnecting valve and shall relieve preferably to a system where the relieved effluent may be contained. The design of the overpressure protection system shall be based on pressure transients that are specified in the Design Specification, and all other applicable requirements of NB-7000 shall be met.

(b) When pressure reducing valves are used and one or more pressure relief devices or safety valves are provided, bypass valves may be provided around the pressure reducing valves. The combined relieving capacity of the pressure relief devices, safety valves, and relief piping shall be such that the lower pressure system service pressure will not exceed the lower pressure system Design Pressure by more than 10% if the pressure reducing valve fails in the open position and the bypass valve is open at the same time. If the pressure reducing valve and its bypass valve are mechanically or electrically interlocked so that only one may be open at any time the high pressure system is at a pressure higher than the Design Pressure of the low pressure system, then the relieving capacity of the pressure relief devices, safety valves, and relief piping shall be at least equal to the maximum capacity of the larger of the two valves. The interlocks shall be redundant and diverse.

(c) Exhaust and pump suction lines for any service and pressure shall have relief valves of a suitable size unless the lines and attached equipment are designed for the maximum pressure and temperature to which they may be accidentally or otherwise subjected.

(d) The effluent from relief devices may be discharged outside the containment only if provisions are made for the disposal of the effluent.

(e) Drip lines from steam headers, mains, separators, or other equipment operating at different pressures shall not discharge through the same trap. Where several traps discharge into a single header that is or may be under pressure, a stop valve and a check valve shall be provided in the discharge line from each trap. The Design Pressure of trap discharge piping shall not be less than the maximum discharge pressure to which it may be subjected. Trap discharge piping shall be designed for the same pressure as the trap inlet piping unless the discharge piping is vented to a system operated under lower pressure and has no intervening stop valves.

(f) Blowdown, dump and drain piping from water

spaces of a steam generation system shall be designed for saturated steam at the pressures and temperatures given below.

Vessel Pressure, psi	Design Pressure, psi	Design Temperature, °F
600 and below	250	410
601 to 900	400	450
901 to 1500	600	490
1501 and above	900	535

These requirements for blowdown, dump, and drain piping apply to the entire system beyond the blowdown valves to the blowdown tank or other points where the pressure is reduced to approximately atmospheric and cannot be increased by closing a valve. When pressures can be increased because of calculated pressure drop or otherwise, this shall be taken into account in the design. Such piping shall be designed for the maximum pressure to which it may be subjected.

(g) Pump discharge piping shall be designed for the maximum pressure exerted by the pump at any load and for the highest corresponding temperature actually existing.

(h) Where a fluid passes through heat exchangers in series, the design temperature of the piping in each section of the system shall conform to the most severe temperature condition expected to be produced by heat exchangers in that section.

NB-3613 Allowances

NB-3613.1 Corrosion or Erosion. When corrosion or erosion is expected, the wall thickness of the piping shall be increased over that required by other design requirements. This allowance shall be consistent with the specified design life of the piping.

NB-3613.2 Threading and Grooving. The calculated minimum thickness of piping that is to be threaded or grooved shall be increased by an allowance equal to the depth of the cut.

NB-3613.3 Mechanical Strength. When necessary to prevent damage, collapse, or buckling of pipe due to superimposed loads from supports or other causes, the wall thickness of the pipe shall be increased, or, if this is impractical or would cause excessive local stresses, the superimposed loads or other causes shall be reduced or eliminated by other design methods.

NB-3620 DESIGN CONSIDERATIONS

NB-3621 Design and Service Loadings

The provisions of NB-3110 apply.

NB-3622 Dynamic Effects

NB-3622.1 Impact. Impact forces caused by either external or internal loads shall be considered in the piping design.

NB-3622.2 Earthquake. The effects of earthquake shall be considered in the design of piping, piping supports, and restraints. The loadings, movements (earthquake anchor movements), and number of cycles to be used in the analysis shall be part of the Design Specifications. The stresses resulting from these earthquake effects must be included with weight, pressure, or other applied loads when making the required analysis.

NB-3622.3 Vibration. Piping shall be arranged and supported so that vibration will be minimized. The designer shall be responsible, by design and by observation under startup or initial service conditions, for ensuring that vibration of piping systems is within acceptable levels.

NB-3622.4 Relief and Safety Valve Thrust. The effects of thrusts from relief and safety valve loads from pressure and flow transients shall be considered in the design of piping, pipe supports, and restraints. See Appendix O.

NB-3623 Weight Effects

Piping systems shall be supported to provide for the effects of live and dead weights, as defined in the following subparagraphs, and they shall be arranged or properly restrained to prevent undue strains on equipment.

NB-3623.1 Live Weight. The live weight shall consist of the weight of the fluid being handled or of the fluid used for testing or cleaning, whichever is greater.

NB-3623.2 Dead Weight. The dead weight shall consist of the weight of the piping, insulation, and other loads permanently imposed upon the piping.

NB-3624 Thermal Expansion and Contraction Loads

NB-3624.1 Loadings, Displacements, and Restraints. The design of piping systems shall take into account the forces and moments resulting from thermal expansion and contraction, equipment displacements and rotations, and the restraining effects of hangers, supports, and other localized loadings.

NB-3624.2 Analysis of Thermal Expansion and Contraction Effects. The analysis of the effects of thermal expansion and contraction is covered in NB-3672.

NB-3624.3 Provision for Rapid Temperature Fluctuation Effects. The Designer shall provide for unusual thermal expansion and contraction loads caused by rapid temperature fluctuations.

NB-3625 Stress Analysis

A stress analysis shall be prepared in sufficient detail to show that each of the stress limitations of NB-3640 and NB-3650 is satisfied when the piping is subjected to the loadings required to be considered by this Subarticle.

NB-3630 PIPING DESIGN AND ANALYSIS CRITERIA

(a) The design and analysis of piping when subjected to the individual or combined effects of the loadings defined in NB-3100 and NB-3620 may be performed in accordance with this Subarticle. Design for pressure loading shall be performed in accordance with the rules of NB-3640. Standard piping products that meet the requirements of ANSI B16.9 or NB-3649 satisfy the requirements of NB-3640, and only the analysis required by NB-3650 need be performed.

(b) Within a given piping system, the stress and fatigue analysis shall be performed in accordance with one of the methods given in NB-3650, NB-3200, or Appendix II. Stress indices are given in NB-3680 for standard piping products, for some fabricated joints, and for some fabricated piping products. Some piping products designed for pressure by applying the rules of NB-3649 may not be listed in NB-3680. For such products, the designer shall determine the stress indices as required in NB-3650.

(c) When a design does not satisfy the requirements of NB-3640 and NB-3650, the more detailed alternative analysis given in NB-3200 or the experimental stress analysis of Appendix II may be used to obtain stress values for comparison with the criteria of NB-3200.

(d) The requirements of this Subarticle shall apply to all Class 1 piping except as exempted under (1) or (2) below.

(1) Piping of 1 in. nominal pipe size or less which has been classified as Class 1 in the Design Specification may be designed in accordance with the design requirements of Subsection NC.

(2) Class 1 piping may be analyzed in accordance with the Class 2 analysis of piping systems in Subsection NC, using the allowable Class 2 stresses and stress limits, provided the specified service loads for which Level A and B Service Limits are designated meet all of the requirements stipulated in (a) through (e) below.

(a) Atmospheric to Service Pressure Cycle. The specified number of times (including startup and shutdown) that the pressure will be cycled from atmospheric pressure to service pressure and back to atmospheric pressure during normal service does not exceed the number of cycles on the applicable fatigue curve of Figs. I-9.0 corresponding to an S_a value of three times the S_m value for the material at service temperature.

(b) Normal Service Pressure Fluctuation. The specified full range of pressure fluctuations during normal service does not exceed the quantity $\frac{1}{3} \times$ Design Pressure $\times$ (S_a/S_m), where S_a is the value obtained from the applicable design fatigue curve for the total specified number of significant pressure fluctuations and S_m is the allowable stress intensity for the material at service temperature. If the total specified number of significant pressure fluctuations exceeds the maximum number of cycles defined on the applicable design fatigue curve, the S_a value corresponding to the maximum number of cycles defined on the curve may be used. Significant pressure fluctuations are those for which the total excursion exceeds the quantity: Design Pressure $\times \frac{1}{3} \times (S/S_m)$, where S is defined as follows:

(1) If the total specified number of service cycles is 10^6 cycles or less, S is the value of S_a obtained from the applicable design fatigue curve for 10^6 cycles.

(2) If the total specified number of service cycles exceeds 10^6 cycles, S is the value of S_a obtained from the applicable design fatigue curve for the maximum number of cycles defined on the curve.

(c) Temperature Difference — Startup and Shutdown. The temperature difference, °F, between any two adjacent points[20] of the component during normal service does not exceed $S_a/2E\alpha$, where S_a is the value

[20] *Adjacent points* are defined as points which are spaced less than the distance $2\sqrt{Rt}$ from each other, where R and t are the mean radius and thickness, respectively, of the vessel, nozzle, flange, or other component in which the points are located.

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obtained from the applicable design fatigue curves for the specified number of startup–shutdown cycles, α is the value of the instantaneous coefficient of thermal expansion at the mean value of the temperatures at the two points as given by Table I-5.0, and E is taken from Table I-6.0 at the mean value of the temperature at the two points.

(d) Temperature Difference — Normal Service.[8] The temperature difference, °F, between any two adjacent points[20] does not change[21] during normal service by more than the quantity $S_a/2E\alpha$, where S_a is the value obtained from the applicable design fatigue curve of Figs. I-9.0 for the total specified number of significant temperature difference fluctuations. A temperature difference fluctuation shall be considered to be significant if its total algebraic range exceeds the quantity $S/2E\alpha$, where S is defined as follows:

(1) If the total specified number of service cycles is 10^6 cycles or less, S is the value of S_a obtained from the applicable design fatigue curve for 10^6 cycles.

(2) If the total specified number of service cycles exceeds 10^6 cycles, S is the value of S_a obtained from the applicable design fatigue curve for the maximum number of cycles defined on the curve.

(e) Temperature Difference — Dissimilar Materials. For components fabricated from materials of differing moduli of elasticity or coefficients of thermal expansion, the total algebraic range of temperature fluctuation, °F, experienced by the component during normal service does not exceed the magnitude $S_a/2(E_1\alpha_1 - E_2\alpha_2)$, where S_a is the value obtained from the applicable design fatigue curve for the total specified number of significant temperature fluctuations, E_1 and E_2 are the moduli of elasticity (Table I-6.0), and α_1 and α_2 are the values of the instantaneous coefficients of thermal expansion (Table I-5.0) at the mean temperature value involved for the two materials of construction. A temperature fluctuation shall be considered to be significant if its total excursion exceeds the quantity $S/2(E_1\alpha_1 - E_2\alpha_2)$, where S is defined as follows.

(1) If the total specified number of service cycles is 10^6 cycles or less, S is the value of S_a obtained from the applicable design fatigue curve for 10^6 cycles.

(2) If the total specified number of service cycles exceeds 10^6 cycles, S is the value of S_a obtained from the applicable design fatigue curve for the maximum number of cycles defined on the curve. If the two materials used have different applicable design fatigue curves, the lower value of S_a shall be used in applying the rules of this paragraph.

NB-3640 PRESSURE DESIGN

NB-3641 Straight Pipe

NB-3641.1 Straight Pipe Under Internal Pressure. The minimum thickness of a pipe wall required for Design Pressure shall be determined from one of the following equations:

$$t_m = \frac{PD_o}{2(S_m + Py)} + A \tag{1}$$

$$t_m = \frac{Pd + 2A(S_m + Py)}{2(S_m + Py - P)} \tag{2}$$

where

t_m = the minimum required wall thickness, in. [Eq. (2) is valid only if $d = D_o - 2t_m$. If pipe is ordered by its nominal wall thickness, the manufacturer's tolerance on wall thickness must be taken into account.]

P = internal Design Pressure, psi

D_o = outside diameter of the pipe, in. (For design calculations, the specified outside diameter of pipe disregarding outside tolerances shall be used to obtain the value of t_m.)

S_m = maximum allowable stress intensity for the material at the Design Temperature taken from Tables I-1.0, psi

d = inside diameter, in.

A = an additional thickness to provide for material removed in threading, corrosion or erosion allowance, and material required for structural strength of the pipe during erection, as appropriate, in.

y = 0.4

The allowable working pressure of pipe may be determined from the following equation:

$$P_a = \frac{2S_m t}{D_o - 2yt} \tag{3}$$

where

t = the specified or actual wall thickness as appropriate, minus, as appropriate, material removed in threading, corrosion or erosion allowance, material manufacturing

[8] *Normal service* is defined as service, other than startup and shutdown, resulting in specified Service Loadings for which Level A Limits, Level B Limits, or Testing Limits are designated.

[21] The algebraic range of the difference shall be used.

TABLE NB-3642.1(b)-1
BEND RADIUS VS THICKNESS

Radius of Bends	Minimum Thickness Recommended Prior to Bending [Note (1)]
6 pipe diameters or greater	$1.06t_m$
5 pipe diameters	$1.08t_m$
4 pipe diameters	$1.16t_m$
3 pipe diameters	$1.25t_m$

NOTE:
(1) t_m is determined by Eq. (1) or (3) of NB-3641.1.

tolerances, bending allowance (NB-3642.1), or material to be removed by counterboring, in.

P_a = the calculated maximum allowable internal pressure for a straight pipe which shall at least equal the Design Pressure, psi. It may be used for piping products with pressure ratings equal to that of straight pipe (see ANSI B16.9). For piping products where the pressure rating may be less than that of the pipe [for example, flange joints and reinforced branch connections (NB-3643), where part of the required reinforcement is in the run pipe], the Design Pressure shall be used instead of P_a. P_a may be rounded out to the next higher unit of 10.

NB-3641.2 Straight Pipe Under External Pressure. The rules of NB-3133 shall be used.

NB-3642 Curved Segments of Pipe

NB-3642.1 Pipe Bends. The wall thickness for pipe bends shall be determined in the same manner as determined for straight pipe in accordance with NB-3641, subject to the limitations given in (a), (b), and (c) below.

(a) The wall thickness after bending shall not be less than the minimum wall thickness required for straight pipe.

(b) The information in Table NB-3642.1(b)-1 is given to guide the designer when ordering pipe.

(c) For the effects of ovality on stress levels, see NB-3680.

NB-3642.2 Elbows. Elbows, manufactured in accordance with the standards listed in Table NB-3132-1 as limited by NB-3612.1, shall be considered as meeting the requirements of NB-3640, except that the minimum thickness in the crotch region of short radius welding elbows in accordance with ANSI B16.28 shall be 20% greater than the minimum thickness required for the straight pipe by Eq. (1) (NB-3641.1). The crotch region is defined as that portion of the elbow between ϕ = 210 deg. and 330 deg., where ϕ is defined in Fig. NB-3685.2-1.

NB-3643 Intersections

NB-3643.1 General Requirements

(a) The rules contained in this paragraph meet the requirements of NB-3640 in the vicinity of branch connections.

(b) Openings shall be circular, elliptical, or of any other shape that results from the intersection of a circular or elliptical cylinder with a cylindrical shape. Additional restrictions affecting stress indices are given in NB-3680.

(c) Openings are not limited in size except to the extent provided for in connection with the stress indices listed in NB-3680.

(d) All references to dimensions apply to the finished dimensions, excluding material added for corrosion allowance.

(e) Any type of opening permitted in these rules may be located in a welded joint.

(f) Where intersecting pipes are joined by welding a branch to a run pipe as shown in Fig. NB-3643.3(a)-2, the angle α between axes of the intersecting pipes shall not be less than 60 deg. or more than 120 deg. For angles outside this range, use fittings as specified in NB-3643.2(a) or (b).

NB-3643.2 Branch Connections. Branch connections in piping may be made by using one of the products or methods set forth in (a) through (c) below.

(a) Flanged, butt welding, or socket welding fittings meeting the applicable ANSI standards listed in Table NB-3132-1, subject to the limitations or requirements of this Subsection, are acceptable. Fittings that comply with the test requirements of ANSI B16.9 or of NB-3649 are not required to meet requirements for reinforcement given in NB-3643.3.

(b) Welded outlet fittings, cast or forged branches, pipe adapters, couplings, or similar products with butt welding, socket welding, or flanged ends are acceptable for attachment to the run pipe when limited to types that have integral reinforcement and are attached to the main run by welding [Figs. NB-4244(a)-1 and NB-4244(b)-1].

(c) An extruded outlet at right angles to the run pipe is acceptable.

NB-3643.3 Reinforcement for Openings

(a) Nomenclature

(1) The following terms are as shown in Fig. NB-3643.3(a)-1.

r_2 = transition radius between branch or extruded lip and run pipe, in.

T_r = nominal thickness of the run pipe, not including corrosion allowance or mill tolerance, in.

T_b = nominal thickness of the branch, not including corrosion allowance or mill tolerance, in.

r_n = nominal radius [sketch (c) only], in.
$= r_i + 0.5T'_b + 0.5y \cos \theta$

d_o = outside diameter of branch, in.

h = height of the extruded lip, equal to or greater than r_2, in.

T_o = corroded finished thickness or extruded outlet measured at a height of r_2 above the outside surface of the run pipe, in.

T'_b = nominal thickness of the branch pipe exclusive of corrosion allowance or mill tolerance, in.

r_i = inside radius of branch pipe, in.

y = slope offset distance, in.

θ = angle between vertical and slope, deg.

r_m = mean radius of the branch pipe, in.
$= r_i + 0.5T_b$

r'_m = mean radius of the branch pipe, in.
$= r_i + 0.5T'_b$

R_m = mean radius of the run pipe, in.

d_o = outside diameter of the branch pipe, in.

(2) The following terms are as shown on Fig. NB-3643.3(a)-2.

A_1 = metal area available for reinforcement, sq in.

A_2 = metal area available for reinforcement, sq in.

A_3 = metal area available for reinforcement, sq in.

α = angle between axes of branch and run (90 deg. $\geq \alpha \geq$ 60 deg.), deg.

d = diameter in the given plane of the finished opening in its corroded condition, in.

L_A = half-width of reinforcement zone measured along the midsurface of the run pipe, in.

L'_A = half-width of zone in which two-thirds of compensation must be placed, in.

L_N = limit of reinforcement measured normal to run pipe wall, in.

r = radius of the finished opening in the corroded condition, in.

t_r = minimum required thickness of the run pipe, not including corrosion allowance, according to NB-3641.1, in.
$= t_m - A$

t_b = minimum required thickness of the branch pipe, not including corrosion allowance, according to NB-3641.1, in.
$= t_m - A$

T_r = nominal thickness of the run pipe, not including corrosion allowance or mill tolerance, in.

(3) The following terms are as shown in Fig. NB-3643.3(a)-3.

T_o = finished thickness of the extruded outlet in the corroded condition measured at a height equal to r_2 above the outside surface of the main run, in.

D_o = outside diameter of the run pipe, in.

T'_r = minimum thickness of the run pipe after extrusion of the opening, not including corrosion allowance or mill tolerance, in. Allowance shall be made for thinning of the run pipe wall by the extrusion of the opening, if it occurs.

(b) Requirements

(1) Reinforcement shall be provided in amount and distribution so that the requirements for the area of reinforcement are satisfied for all planes through the center of the opening and normal to the surface of the run pipe, except that openings need not be provided with reinforcement if all of the requirements of (a), (b), and (c) below are met.

(a) A single opening has a diameter not exceeding $0.2\sqrt{R_m T_r}$ or, if there are two or more openings within any circle of diameter, $2.5\sqrt{R_m T_r}$ but the sum of the diameters of such unreinforced openings shall not exceed $0.25\sqrt{R_m T_r}$.

(b) No two unreinforced openings shall have their centers closer to each other, measured on the inside wall of the run pipe, than the sum of their diameters.

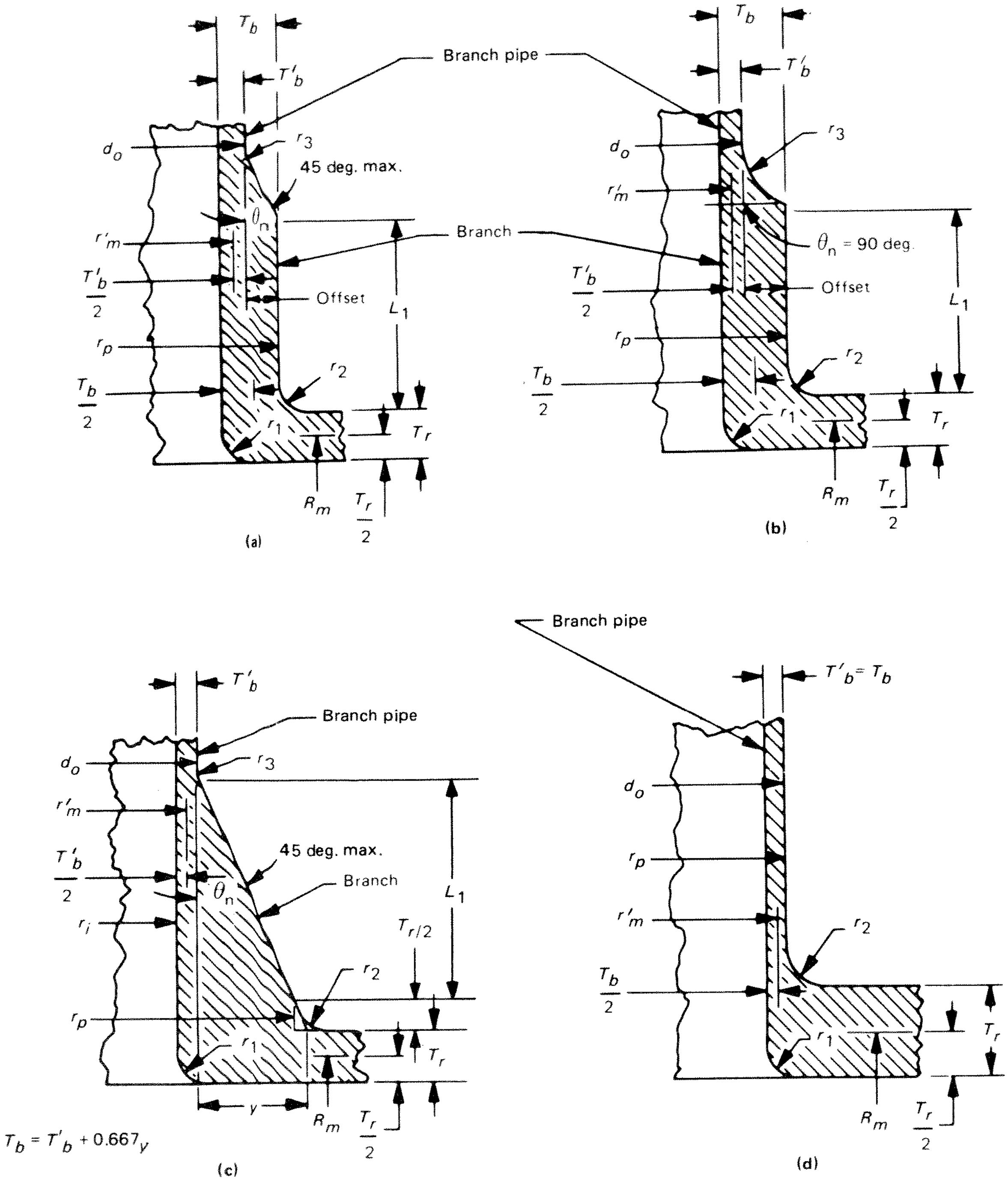
(c) No unreinforced opening shall have its center closer than $2.5\sqrt{R_m T_r}$ to the edge of any other locally stressed area.

(2) The total cross-sectional area of reinforcement A required in any given plane for a pipe under internal pressure shall not be less than:

$$A = dt_r(2 - \sin \alpha)$$

(3) The required reinforcing material shall be uniformly distributed around the periphery of the branch except that, in the case of branches not at right angles, the designer may elect to provide additional reinforcement in the area of the crotch.

(c) Limits of Reinforcement. The boundaries of the cross-sectional area in any plane passing through the



GENERAL NOTE:
If L_1 equals or exceeds $0.5\sqrt{r_i T_b}$, then r'_m can be taken as the radius to the center of T_b.

FIG. NB-3643.3(a)-1 BRANCH CONNECTION NOMENCLATURE

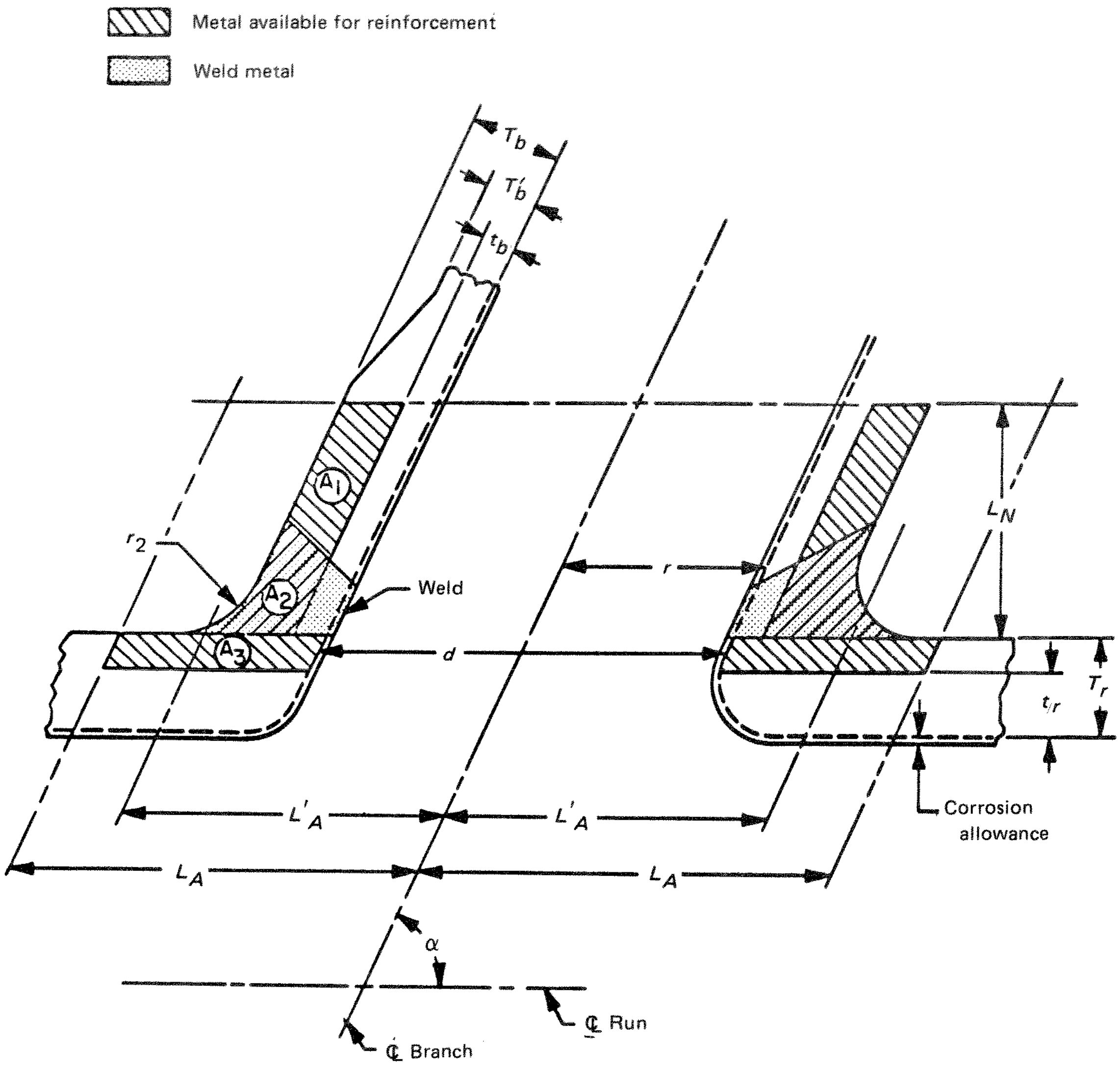
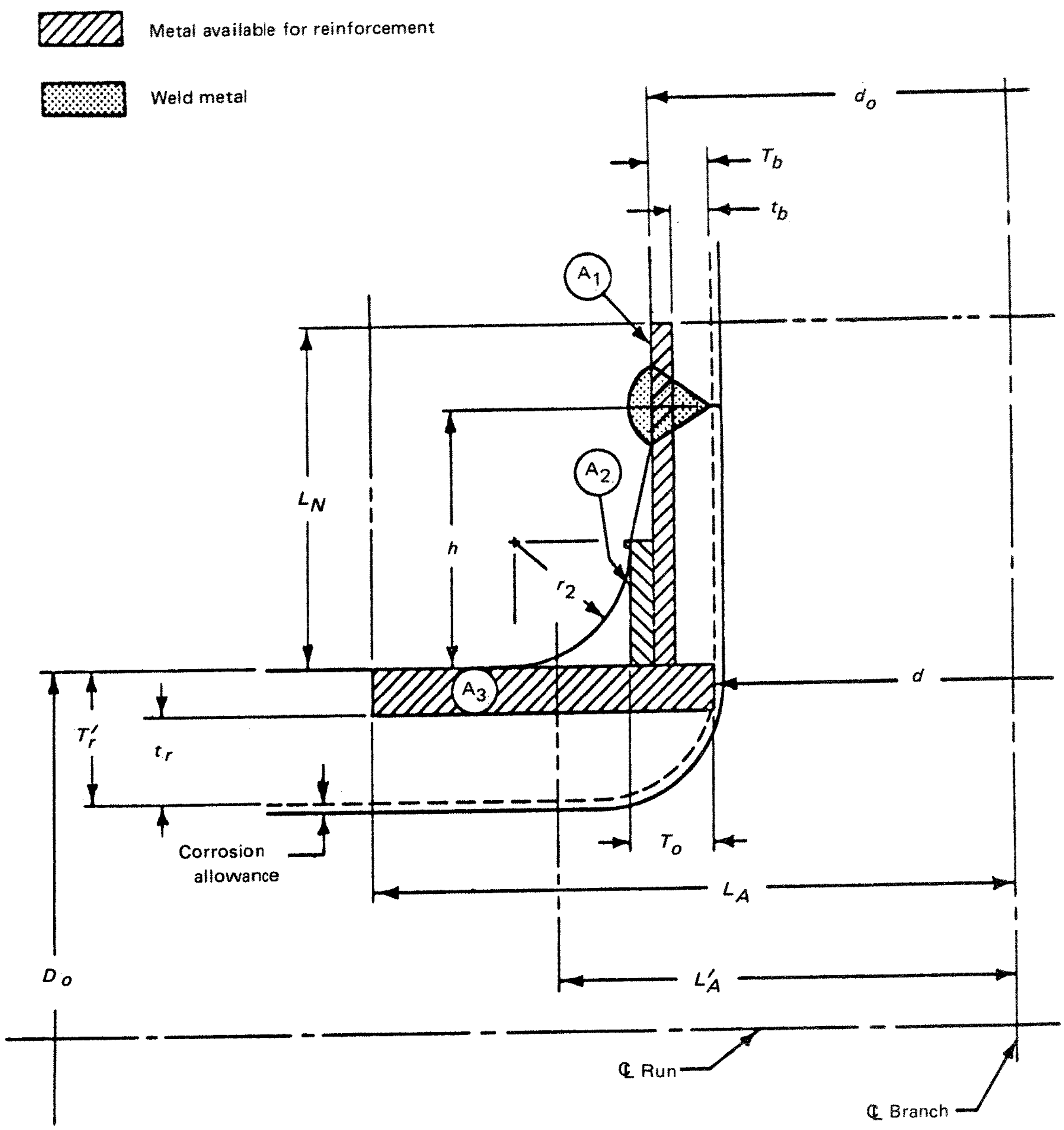


FIG. NB-3643.3(a)-2 TYPICAL REINFORCEMENT OF OPENINGS



GENERAL NOTE:
The areas A_1, A_2, and A_3, shown are only one-half of those defined in NB-3643.3(f)(3).

FIG. NB-3643.3(a)-3 TYPICAL REINFORCED EXTRUDED OUTLET axis of the opening within which metal may be located to have value as reinforcement are designated as the limits of reinforcement for that plane, and they are given in (1) and (2) below.

(1) The limits of reinforcement, measured along the midsurface of the nominal wall thickness of the run pipe, L_A shall be at a distance on each side of the axis of the opening that is equal to the greater of (a) or (b) below:

(a) the diameter of the finished opening in the corroded condition; or

(b) the radius of the finished opening in the corroded condition r, plus the nominal thickness of the run pipe T_r, plus the nominal thickness of the branch wall T_b.

(c) In addition, two-thirds of the required reinforcement shall be provided within the greater of the limit given in (b) above and the limit L'_A which is the greater of either

$$r + 0.5\sqrt{R_m T_r}$$

or

$$r + T_b/\sin\alpha + T_r$$

(2) The limits of reinforcement measured normal to the wall of the run pipe L_N shall conform to the contour of the surface of the branch at a distance from each surface equal to the limits given in (a) and (b) below and as shown in Fig. NB-3643.3(a)-1.

(a) For nozzle types of Fig. NB-3643.3(a)-1 sketches (a), (b), and (d):

$$L_N = 0.5\sqrt{r_m T_b} + 0.5r_2$$

(b) For Fig. NB-3643.3(a)-1 sketch (c):

$$L_N = 0.5\sqrt{r_n T_b}$$

(d) Metal Available for Reinforcement

(1) Metal may be counted as contributing to the area of reinforcement called for in NB-3643.3(b) if it lies within the area of reinforcement specified in NB-3643.3(c), and it shall be limited to material that meets the requirements of (a), (b), and (c) below:

(a) metal forming a part of the run wall that is in excess of that required on the basis of NB-3641.1 and is exclusive of corrosion allowance shown in Fig. NB-3643.3(a)-2;

(b) similar excess metal in the branch wall, if the branch is integral with the run wall or is joined to it by a full penetration weld, as denoted by A_1 in Fig. NB-3643.3(a)-2;

(c) weld metal that is fully continuous with the wall of the run pipe, as denoted by area A_2 in Fig. NB-3643.3(a)-2.

(2) The mean coefficient of thermal expansion of the metal to be included as reinforcement under (1)(b) and (1)(c) above shall be within 15% of the value for the metal in the wall of the run pipe.

(3) Metal available for reinforcement shall not be considered as applying to more than one opening.

(4) Metal not fully continuous with the run pipe, as that in branches attached by partial penetration welds, shall not be counted as reinforcement.

(e) Strength of Metal. Material used for reinforcement shall preferably be the same as that of the wall of the run pipe. If material with a lower design stress intensity S_m is used, the area provided by such material shall not be counted at full value but shall be multiplied by the ratio (less than unity) of the design stress intensity values S_m of the reinforcement material and of the run pipe material before being counted as reinforcement. No reduction in the reinforcement requirement may be taken for the increased strength of either the branch material or weld metal having a higher design stress intensity value than that of the material of the run pipe wall. The strength of the material at the point under consideration shall be used in the fatigue analysis.

(f) Requirements for Extruded Outlets. Extruded outlets shall meet all of the requirements of NB-3643.3(a) and NB-3643.3(b), and these rules apply only where the axis of the outlet intersects and is perpendicular to the axis of the run pipe.

(1) Geometric Requirements

(a) An extruded outlet is one in which the extruded lip at the outlet has a height h above the surface of the run pipe that is equal to or greater than the transition radius between the extruded lip and the run pipe r_2.

(b) The minimum value of the transition radius r_2 shall not be less than $0.05d_o$, except that on branch pipe sizes larger than 30 in. the transition radius need not exceed 1.5 in. The maximum value of the transition radius r_2 shall be limited as follows: for branch pipes nominally 8 in. and larger, the dimension of the transition radius shall not exceed $0.10d_o + 0.50$ in.; for branch pipe sizes nominally less than 8 in., r_2 shall not be greater than 1.25 in.

(c) When the external contour contains more than one radius, the radius of any arc sector of approximately 45 deg. shall meet the requirements given in (b) above.

(d) Machining shall not be employed to meet the requirements of (b) and (c) above.

(2) Limits of Reinforcement

(a) The height of the reinforcement zone shall be limited as shown in Fig. NB-3643.3(a)-3:

$$L_N = 0.5\sqrt{d_o T_o}$$

(b) The half width of the reinforcement zone shall be limited as shown in Fig. NB-3643.3(a)-3:

$$L_A = d$$

(3) Metal Available for Reinforcement. The reinforcement area shall be the sum of areas $A_1 + A_2 + A_3$ defined in (a), (b), and (c) below and shown in Fig. NB-3643.3(a)-3. Metal counted as reinforcement shall not be applied to more than one opening.

(a) Area A_1 is the area lying within the reinforcement zone that results from any excess thickness available in the wall of the branch pipe:

$$A_1 = 2L_N(T'_b - t_b)$$

(b) Area A_2 is the area lying within the reinforcement zone that results from excess thickness available in the lip of the extruded outlet:

$$A_2 = 2r_2(T_o - T'_b)$$

(c) Area A_3 is the area lying within the reinforcement zone that results from any excess thickness in the run pipe wall:

$$A_3 = d\,(T'_r - t_r)$$

NB-3644 Miters

Mitered joints may be used in piping systems under the conditions stipulated in (a) through (d) below.

(a) The minimum thickness of a segment of a miter shall be determined in accordance with NB-3641. The minimum thickness thus determined does not allow for the discontinuity stresses that exist at the junction between segments. The discontinuity stresses are reduced for a given miter as the number of segments is increased.

(b) The angle θ in Fig. NB-3644(b)-1 shall not be more than 22½ deg.

(c) The center line distance S between adjacent miters shall be in accordance with Fig. NB-3644(b)-1.

(d) Stress indices and flexibility factors shall be determined in accordance with the requirements of Appendix II.

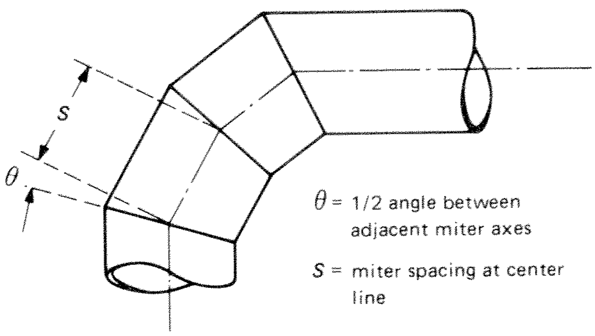


FIG. NB-3644(b)-1 MITER JOINT GEOMETRY

NB-3646 Closures

(a) Closures in piping systems may be made by use of closure fittings, such as blind flanges or threaded or welded plugs or caps, either manufactured in accordance with standards listed in Table NB-3132-1 and used within the specified pressure–temperature ratings, or made in accordance with (b) below.

(b) Closures not manufactured in accordance with the standards listed in Table NB-3132-1 may be made in accordance with the rules for Class 2 vessels using the equation:

$$t_m = t + A$$

where

- t_m = minimum required thickness, in.
- t = pressure design thickness, in., calculated for the given closure shape and direction of loading using appropriate equations and procedures for Class 2 Vessels, except that the symbols used to determine t shall be defined as:
  - P = Design Pressure, psi
  - S = applicable design stress intensity value S_m from Tables I-1.0, psi
- A = sum of mechanical allowances (NB-3613), in.

(c) Connections to closures may be made by welding or extruding. Connections to the closure shall be in accordance with the limitations provided for such connections in NB-3643 and Figs. NB-4243-1, NB-4244(a)-1, NB-4244(b)-1, and NB-4244(c)-1 for branch connections. If the size of the opening is greater than

one-half the inside diameter of the closure, the opening should be considered as a reducer in accordance with NB-3648.

(d) Openings in closures may be reinforced in accordance with the requirements of NB-3643.

(e) Flat heads that have an opening with a diameter that does not exceed one-half of the head diameter shall have a total cross-sectional area of reinforcement not less than $dt/2$, where

d = the diameter of the finished opening, in.
t = the design thickness for the closure, in.

NB-3647 Pressure Design of Flanged Joints and Blanks

NB-3647.1 Flanged Joints

(a) Flanged joints manufactured in accordance with the standards listed in Table NB-3132-1, as limited by NB-3612.1, shall be considered as meeting the requirements of NB-3640.

(b) Flanged joints not included in Table NB-3132-1 shall be designed in accordance with XI-3000, including the use of the appropriate allowable stress given in Tables I-7.0.

NB-3647.2 Permanent Blanks. The minimum required thickness of permanent blanks (Fig. NB-3647.2-1) shall be calculated from the following equations:

$$t_m = t + A \qquad (7)$$

where

t_m = minimum required thickness, in.
t = pressure design thickness, in., calculated from Eq. (8)
A = sum of the mechanical allowances, in. (NB-3613)

$$t = d_6 \left(\frac{3P}{16S_m}\right)^{1/2} \qquad (8)$$

where

d_6 = inside diameter of the gasket for raised or flat face flanges or the pitch diameter of the gasket for retained gasketed flanges, in.
P = Design Pressure, psi
S_m = the design stress intensity value in accordance with Tables I-1.0, psi

NB-3647.3 Temporary Blanks. Blanks to be used for test purposes only shall have a minimum thickness not less than the Design Pressure thickness t, calculated from Eq. (8) above, except that P shall not be less than the test pressure and the design stress intensity value S_m may be taken as 95% of the specified minimum yield strength of the blank material (Tables I-2.0).

NB-3648 Reducers

Reducer fittings manufactured in accordance with the standards listed in Table NB-3132-1 shall be considered suitable for use. Where butt welding reducers are made to a nominal pipe thickness, the reducers shall be considered suitable for use with pipe of the same nominal thickness.

NB-3649 Pressure Design of Other Piping Products

Other piping products manufactured in accordance with the standards listed in Table NB-3132-1 shall be considered suitable for use provided the design is consistent with the design philosophy of this Subsection. Piping products not included in Table NB-3132-1 may be used if they satisfy the requirements of NB-3200. The pressure design shall be based on an analysis consistent with this Subsection, or experimental stress analysis as described in Appendix II, or an ANSI B16.9 type burst test. The bursting pressure in a B16.9 type burst test shall be equal to or greater than that of the weakest pipe to be attached to the piping product, where the burst pressure of the weakest pipe is calculated by the equation:

$$P = 2St/D_o$$

where

S = specified minimum tensile strength of pipe material, psi
t = minimum specified wall thickness of pipe, in.
D_o = outside diameter of pipe, in.

NB-3649.1 Expansion Joints. Rules are currently under development for the application of expansion joints in piping systems. Until these rules are available, expansion joints shall not be used in piping.

NB-3650 ANALYSIS OF PIPING PRODUCTS

NB-3651 General Requirements

NB-3651.1 Piping Products for Which Stress Indices Are Given. Piping products, for which values of stress indices B, C, and K are given in NB-3683.2 and which meet the requirements of NB-3640, satisfy the design criteria of NB-3611 provided they comply with

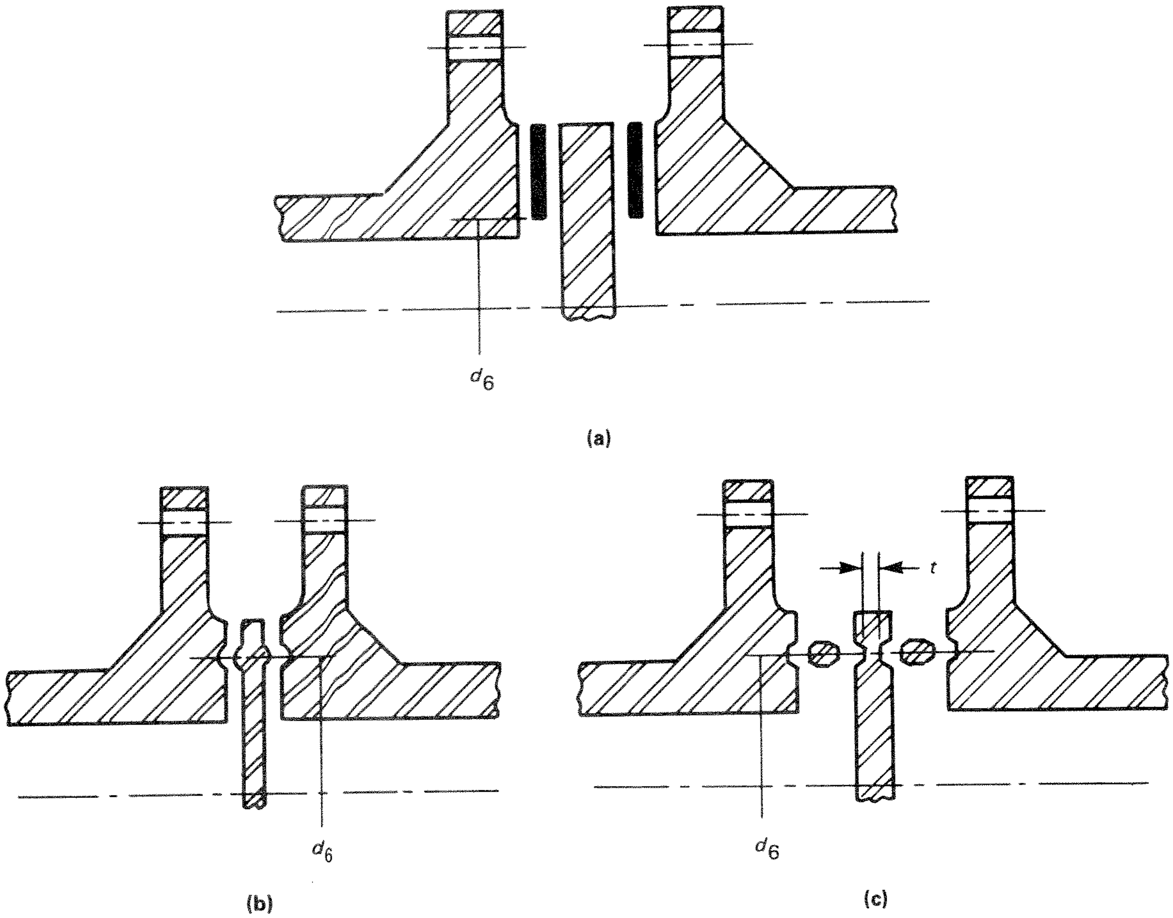


FIG. NB-3647.2-1 TYPES OF PERMANENT BLANKS

these rules. To validate a design in accordance with these rules, it is necessary to perform several flexibility analyses in accordance with the requirements of NB-3672 and to use the moments and forces obtained from these analyses as required in NB-3650.

NB-3651.2 Piping Products for Which Stress Indices Are Not Available. For analysis of flanged joints, see NB-3658. For other piping products for which stress indices are not available, see NB-3680.

NB-3651.3 Attachments

(a) Lugs, brackets, stiffeners, and other attachments may be welded, bolted, and studded to, or bear upon the outside or inside of piping. The interaction effects of attachments on the pressure boundary, producing thermal gradients, localized bending stresses, stress concentrations, or restraint of the pressure boundary shall be considered by the piping designer. Standard clamps generally have a negligible effect on the pressure boundary. However, the effects of clamps on thin-wall piping may need to be evaluated.

(b) Attachments shall meet the requirements of NB-3135.

(c) Figure NB-4433-1 shows some typical types of attachment welds (NB-4430).

NB-3652 Consideration of Design Conditions

The primary stress intensity limit is satisfied if the requirement of Eq. (9) is met:

$$B_1 \frac{PD_o}{2t} + B_2 \frac{D_o}{2I} M_i \leq 1.5 S_m \qquad (9)^{22}$$

[22] For piping products, such as tees and branch connections, the second term of Eqs. (9), (10), and (11), namely that containing M_b is to be calculated as referred to in NB-3683.1(d).

where

B_1, B_2 = primary stress indices for the specific product under investigation (NB-3680)

P = Design Pressure, psi

D_o = outside diameter of pipe, in. (NB-3683)

t = nominal wall thickness of product, in. (NB-3683)

I = moment of inertia, in.[4] (NB-3683)

M_i = resultant moment due to a combination of Design Mechanical Loads, in.-lb. All Design Mechanical Loads, and combinations thereof shall be provided in the Design Specification. In the combination of loads, all directional moment components in the same direction shall be combined before determining the resultant moment (i.e., resultant moments from different load sets shall not be used in calculating the moment M_i). If the method of analysis for earthquake or other dynamic loads is such that only magnitudes without relative algebraic signs are obtained, the most conservative combination shall be assumed.

S_m = allowable design stress intensity value, psi (Tables I-1.0)

NB-3653 Consideration of Level A Service Limits

NB-3653.1 Satisfaction of Primary Plus Secondary Stress Intensity Range

(a) This calculation is based upon the effect of changes which occur in mechanical or thermal loadings which take place as the system goes from one load set, such as pressure, temperature, moment, and force loading, to any other load set which follows it in time. It is the range of pressure, temperature, and moment between two load sets which is to be used in the calculations. For example, one of the load sets to be included is that corresponding to zero pressure, zero moment, and room temperature. Equation (10) shall be satisfied for all pairs of load sets:

$$S_n = C_1 \frac{P_o D_o}{2t} + C_2 \frac{D_o}{2I} M_i + C_3 E_{ab} \times |\alpha_a T_a - \alpha_b T_b| \leq 3S_m \qquad (10)$$

(b) If for one or more pairs of load sets Eq. (10) is not met, the piping product may still be satisfactory, provided that the conditions of NB-3653.6 are met or provided that the requirements of NB-3200 are satisfied.

(c) The nomenclature used in Eq. (10) is defined as follows:

C_1, C_2, C_3 = secondary stress indices for the specific component under investigation (NB-3680)

D_o, t, I, S_m = as defined for Eq. (9)

M_i = resultant range of moment which occurs when the system goes from one service load set to another, in.-lb. Service loads and combinations thereof shall be provided in the Design Specification. In the combination of moments from load sets, all directional moment components in the same direction shall be combined before determining the resultant moment (i.e., resultant moments from different load sets shall not be used in calculating the moment range M_i). Weight effects need not be considered in determining the loading range since they are noncyclic in character. If the method of analysis is such that only magnitudes without relative algebraic signs are obtained, the most conservative combination shall be assumed. If a combination includes earthquake effects, M_i shall be either: (1) the resultant range of moment due to the combination of all loads considering one-half the range of the earthquake; or (2) the resultant range of moment due to the full range of the earthquake alone, whichever is greater.

$T_a(T_b)$ = range of average temperature on side *a(b)* of gross structural discontinuity or material discontinuity, °F. For generally cylindrical shapes, the averaging of T (NB-3653.2) shall be over a distance of $\sqrt{d_a t_a}$ for T_a and over a distance of $\sqrt{d_b t_b}$ for T_b.

$d_a(d_b)$ = inside diameter on side *a(b)* of a gross structural discontinuity or material discontinuity, in.

$t_a(t_b)$ = average wall thickness through the length $\sqrt{d_a t_a}$ ($\sqrt{d_b t_b}$), in. A trial and error solution for t_a and t_b may be necessary.

$\alpha_a(\alpha_b)$ = coefficient of thermal expansion on side *a(b)* of a gross structural discontinuity or material discontinuity, at room temperature, 1/°F (Table I-5.0)

E_{ab} = average modulus of elasticity of the two sides of a gross structural discontinuity or material discontinuity at room temperature, psi (Table I-6.0)

P_o = range of service pressure, psi

NB-3653.2 Satisfaction of Peak Stress Intensity Range

(a) For every pair of load sets (NB-3653), calculate S_p values using Eq. (11):

$$S_p = K_1 C_1 \frac{P_o D_o}{2t} + K_2 C_2 \frac{D_o}{2I} M_i$$

$$+ \frac{1}{2(1-\nu)} K_3 E\alpha \,|\Delta T_1| + K_3 C_3 E_{ab}$$

$$\times |\alpha_a T_a - \alpha_b T_b| + \frac{1}{1-\nu} E\alpha \,|\Delta T_2| \quad (11)$$

NOTE: This simplified analysis is intended to provide a value of S_p that conservatively estimates the sum of $P_L + P_b + P_e + Q + F$ as required in Fig. NB-3222-1.

The nomenclature used in Eq. (11) is defined as follows:

K_1, K_2, K_3 = local stress indices for the specific component under investigation (NB-3680)

$E\alpha$ = modulus of elasticity (E) times the mean coefficient of thermal expansion (α) both at room temperature, psi/°F

$|\Delta T_2|$ = absolute value of the range for that portion of the nonlinear thermal gradient through the wall thickness not included in ΔT_1 as shown below, °F

$|\Delta T_1|$ = absolute value of the range of the temperature difference between the temperature of the outside surface T_o and the temperature of the inside surface T_i of the piping product assuming moment generating equivalent linear temperature distribution, °F

For a quantitative definition of $|\Delta T_1|$ and $|\Delta T_2|$, see NB-3653.2(b) below. All other terms are as defined for Eq. (10).

(b) Quantitative Definitions of $|\Delta T_1|$ and $|\Delta T_2|$. The following nomenclature is used:

t = thickness of the wall of the pipe or element, in.

y = radial position in the wall, measured positive outward from the midthickness position ($-t/2 \leq y \leq t/2$), in.

$T_j(y)$, $T_k(y)$ = temperature, as a function of radial position, for load set j and load set k, respectively, °F

$T(y)$ = temperature distribution range from condition j to condition k, °F

$$= T_k(y) - T_j(y)$$

T_o = value of $T(y)$ at outside surface, °F

$$= T(t/2)$$

T_i = value of $T(y)$ at inside surface, °F

$$= T(-t/2)$$

Then the temperature distribution range $T(y)$ may be thought of as being composed of three parts:

(1) a constant value:

$$T = (1/t) \int_{-t/2}^{t/2} T(y)dy$$

which is the average value through the thickness. T may be used in determining free thermal expansions. Also, the values of T determined (for the same pair of load sets) or two locations a and b on either side of a gross continuity may be used for T_a and T_b in Eqs. (10) and (11).

(2) a linear portion, with zero average value, having variation given by:

$$V = (12/t^2) \int_{-t/2}^{t/2} yT(y)dy$$

(3) a nonlinear portion with a zero average value and a zero first moment with respect to the mid-thickness. This decomposition of $T(y)$ into three parts is illustrated in Fig. NB-3653.2(b)-1. The value of ΔT_1 to be used in Eq. (11) is the variation V of the linear portion:

$$\Delta T_1 = V$$

The value of ΔT_2 to be used in Eq. (11) is as follows:

$$\Delta T_2 = \max.\,(|T_o - T| - \tfrac{1}{2}|\Delta T_1|,\ |T_i - T| - \tfrac{1}{2}|\Delta T_1|,\ 0)$$

NB-3653.3 Alternating Stress Intensity. The alternating stress intensity S_{alt} is equal to one-half the value of S_p ($S_{alt} = S_p/2$) calculated in Eq. (11) above.

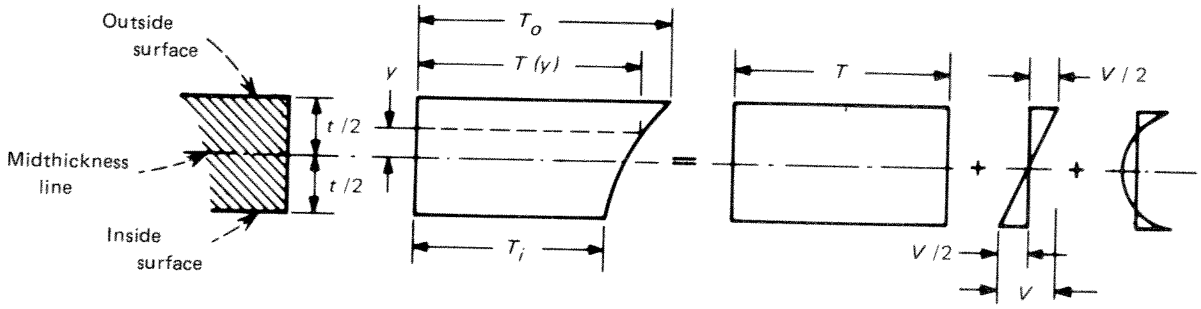


FIG. NB-3653.2(b)-1 DECOMPOSITION OF TEMPERATURE DISTRIBUTION RANGE

NB-3653.4 Use of Design Fatigue Curve. Enter the applicable design fatigue curve, Figs. I-9.0, on the ordinate using $S_a = S_{alt}$, and find the corresponding number of cycles on the abscissa. If the service cycle being considered is the only one that produces significant fluctuating stresses, this is the allowable number of cycles.

NB-3653.5 Cumulative Damage. The cumulative damage shall be evaluated in accordance with NB-3222.4(e)(5). If N_i is greater than the maximum number of cycles defined on the applicable design fatigue curve, the value of n_i/N_i may be taken as zero.

NB-3653.6 Simplified Elastic–Plastic Discontinuity Analysis. If Eq. (10) cannot be satisfied for all pairs of load sets, the alternative analysis described below may still permit qualifying the component under NB-3650. Only those pairs of load sets which do not satisfy Eq. (10) need be considered.

(a) Equation (12) shall be met:

$$S_e = C_2 \frac{D_o}{2I} M_i^* \leq 3S_m \qquad (12)$$

where

S_e = nominal value of expansion stress, psi

M_i^* = same as M_i in Eq. (10), except that it includes only moments due to thermal expansion and thermal anchor movements, in.-lb

(b) The requirements of NB-3653.7 shall be met, and, having satisfied those requirements, the primary plus secondary membrane plus bending stress intensity, excluding thermal bending and thermal expansion stresses, shall be $< 3S_m$. This requirement is satisfied by meeting Eq. (13) below:

$$C_1 \frac{P_o D_o}{2t} + C_2 \frac{D_o M_i}{2I} + C'_3 E_{ab} \times |\alpha_a T_a - \alpha_b T_b| \leq 3S_m \qquad (13)$$

where

M_i = as defined in NB-3652 and all other variables are as defined in NB-3653

C'_3 = values in Table NB-3681(a)-1

(c) If these conditions are met, the value of S_{alt} shall be calculated by Eq. (14):

$$S_{alt} = K_e \frac{S_p}{2} \qquad (14)$$

where

S_{alt} = alternating stress intensity, psi

S_p = peak stress intensity value calculated by Eq. (11), (NB-3653.2), psi

K_e = 1.0 for $S_n \leq 3S_m$

= $1.0 + [(1 - n)/n(m - 1)](S_n/3S_m - 1)$, for $3S_m < S_n < 3mS_m$

= $1/n$, for $S_n \geq 3mS_m$

S_n = primary plus secondary stress intensity value calculated in Eq. (10), NB-3653.1, psi

m, n = material parameters given in Table NB-3228.3(b)-1

S_{alt} for all load sets shall be calculated in accordance with NB-3653.3 or Eq. (14). Using the alternating stress intensity values calculated by the above procedures, determine the cumulative usage factor in accordance with NB-3653.4 and NB-3653.5. The cumulative usage factor shall not exceed 1.0.

NB-3653.7 Thermal Stress Ratchet. When the limits of Eq. (10) are exceeded and before the rules of Eq. (13) of NB-3653.6 can be utilized, the value of the range of ΔT_1 cannot exceed that calculated as follows:

$$\Delta T_1 \text{ range} \leq \frac{y' S_y}{0.7\, E\alpha} C_4$$

where
- $y' = 3.33$, 2.00, 1.20, and 0.80 for $x = 0.3$, 0.5, 0.7, and 0.8, respectively
- $x = (PD_o/2t)\,(1/S_y)$
- $P =$ maximum pressure for the set of conditions under consideration, psi
- $C_4 = 1.1$ for ferritic material
  $= 1.3$ for austenitic material
- $E\alpha =$ as defined for Eq. (11), psi/°F
- $S_y =$ yield strength value, psi, taken at average fluid temperature of the transient under consideration

NB-3654 Consideration of Level B Service Limits

NB-3654.1 Permissible Pressure. For Level B Service Limits [NCA-2142.4(b)(2)], the permissible pressure shall not exceed the pressure P_a, calculated in accordance with Eq. (3) of NB-3641.1, by more than 10%.

NB-3654.2 Analysis of Piping Components. For Service Loadings for which Level B Service Limits are designated, the conditions of Eq. (9) shall be met using Service Level B coincident pressure P and moments M_i which result in the maximum calculated stress. The allowable stress to be used for this condition is $1.8S_m$, but not greater than $1.5S_y$. In addition, the procedures for analyzing Service Loadings for which Level B Service Limits are designated are the same as those given in NB-3653 for Level A Service Limits.

$S_y =$ yield strength value, psi, taken at average fluid temperature of the transient under consideration

NB-3655 Consideration of Level C Service Limits

NB-3655.1 Permissible Pressure. When Level C Service Limits [NCA-2142.4(b)(3)] are specified, the permissible pressure shall not exceed the pressure P_a, calculated in accordance with Eq. (3) of NB-3641.1, by more than 50%.

NB-3655.2 Analysis of Piping Components. Under any Service Loadings for which Level C Service Limits are designated [NCA-2142.4(b)(3)], the conditions of Eq. (9) of NB-3652 shall be met using Service Level C coincident pressure P and moments M_i which result in the maximum calculated stress. The allowable stress to be used for this condition is $2.25S_m$ but not greater than $1.8S_y$.

NB-3655.3 Deformation Limits. Any deformation limits prescribed by the Design Specifications shall be considered with respect to Level C Service Limits.

NB-3656 Consideration of Level D Service Limits

(a) If the Design Specifications specify any Service Loadings for which Level D Service Limits are designated [NCA-2142.4(b)(4)], the rules contained in Appendix F may be used in evaluating these Service Loadings independently of all other Design and Service Loadings.

(b) As an alternative, the conditions of Eq. (9) shall be met. The allowable stress to be used for this condition is 3.0 S_m, but not greater than 2.0 S_y. The permissible pressure shall not exceed 2.0 times the pressure P_a calculated in accordance with Eq. (3) of NB-3641.1.

NB-3657 Test Loadings

The evaluation of Test Loadings shall be carried out in accordance with NB-3226.

NB-3658 Analysis of Flanged Joints

Flanged joints using flanges, bolting, and gaskets as specified in ANSI B16.5 and using a bolt material having an S_m value at 100°F not less than 20.0 ksi may be analyzed in accordance with the following rules or in accordance with NB-3200. Other flanged joints shall be analyzed in accordance with NB-3200.

NB-3658.1 Design Limits, Levels A and B Service Limits

(a) Bolting. The bolting shall meet the requirements of NB-3232. In addition, the limitations given by Eqs. (15) and (16) shall be met:

$$M_{fs} \leq 3125\,(S_y/36)\,CA_b \qquad (15)$$

where
- $M_{fs} =$ bending or torsional moment (considered separately) applied to the joint due to weight, thermal expansion of the piping,

sustained anchor movements, relief valve steady-state thrust, and other sustained mechanical loads, in.-lb. If cold springing is used, the moment may be reduced to the extent permitted by NB-3672.8.

S_y = yield strength of flange material at Design Temperature (Table I-2.2), ksi. The value of $S_y/36$ shall not be taken as greater than unity.

C = diameter of bolt circle, in.

A_b = total cross-sectional area of bolts at root of thread or section of least diameter under stress, sq in.

$$M_{fd} \leq (6250 S_y/36)\, CA_b \qquad (16)$$

where

M_{fd} = bending or torsional moment (considered separately) as defined for M_{fs}, but including dynamic loadings, in.-lb

(b) Flanges. Flanges of ANSI B16.5 flanged joints meeting the requirements of NB-3612.1 are not required to be analyzed under NB-3650. However, the pipe-to-flange welds shall meet the requirements of NB-3652, NB-3653, and NB-3654, using appropriate stress indices from Table NB-3681(a)-1.

NB-3658.2 Level C Service Limits

(a) The pressure shall not exceed 1.5 times the Design Pressure.

(b) The limitation given by Eq. (17) shall be met:

$$M_{fd} \leq [11{,}250 A_b - (\pi/16) D_f^2 P_{fd}]\, C(S_y/36) \qquad (17)$$

where

D_f = outside diameter of raised face, in.

P_{fd} = pressure concurrent with M_{fd}, psi

M_{fd}, C, S_y, the limitation on $S_y/36$, and A_b are defined in NB-3658.1(a).

(c) Pipe-to-flange welds shall be evaluated by Eq. (9) of NB-3652, using a stress limit of $2.25S_m$.

NB-3658.3 Level D Service Limits

(a) The pressure shall not exceed 2.0 times the Design Pressure.

(b) The limitation given by Eq. (17) of NB-3658.2(b) shall be met, where P_{fd} and M_{fd} are pressures, psi, and moments, in.-lb, occurring concurrently.

(c) Pipe-to-flange welds shall be evaluated by Eq. (9) of NB-3652, using a stress limit of $3.0S_m$.

NB-3658.4 Test Loadings. Analysis for Test Loadings is not required.

NB-3660 DESIGN OF WELDS

NB-3661 Welded Joints

NB-3661.1 General Requirements. Welded joints shall be made in accordance with NB-4200.

NB-3661.2 Socket Welds

(a) Socket welded piping joints shall be limited to pipe sizes of 2 in. and less.

(b) Socket welded piping joints shall conform to the requirements specified in ANSI B16.11, the applicable standards listed in Table NB-3132-1, and Fig. NB-4427-1. A gap of approximately $\frac{1}{16}$ in. shall be provided between the end of the pipe and the bottom of the socket before welding.

(c) Socket welds shall not be used where the existence of crevices could accelerate corrosion.

NB-3661.3 Partial Penetration Welds for Branch Connections

(a) Partial penetration welds are allowed for branch connections in which there are substantially no piping reactions transmitted from the branch, such as openings for instrumentation. The ratio of nominal pipe size of the main pipe to that of the branch shall not be less than 10. Maximum branch size shall not exceed 2 in. nominal pipe size. For such branch connections, all reinforcing shall be an integral part of the pipe penetrated. Partial penetration welds shall be of sufficient size to develop the full strength of the branch. Reinforcing requirements of NB-3643 shall be met.

(b) Partial penetration branch connections shall be groove welds as shown in Fig. NB-4244(d)-1. These welds shall be capable of being examined in accordance with the requirements of NB-5245.

(c) The inner corners of finished openings, in which the branch does not extend beyond the inner surface of the pipe penetrated, shall be rounded to a minimum radius of one-fourth the thickness t_n of the penetrating part or $\frac{3}{4}$ in., whichever is smaller. The corners of the end of each branch extending less than $\sqrt{dt_n}$ beyond the inner surface of the pipe penetrated shall be rounded to radius of one-half the thickness t_n of the penetrating part or $\frac{3}{4}$ in., whichever is smaller.

NB-3670 SPECIAL PIPING REQUIREMENTS

NB-3671 Selection and Limitation of Nonwelded Piping Joints

The type of piping joint used shall be suitable for the Design Loadings and shall be selected with consideration of joint tightness, mechanical strength, and the nature of the fluid handled. Piping joints shall conform to the requirements of this Subsection with leak tightness being a consideration in selection and design of joints for piping systems to satisfy the requirements of the Design Specifications.

NB-3671.1 Flanged Joints. Flanged joints are permitted.

NB-3671.2 Expanded Joints. Expanded joints shall not be used.

NB-3671.3 Threaded Joints. Threaded joints in which the threads provide the only seal shall not be used. If a seal weld is employed as the sealing medium, the stress analysis of the joint must include the stresses in the weld resulting from the relative deflections of the mated parts.

NB-3671.4 Flared, Flareless, and Compression Joints. Flared, flareless, and compression type tubing fittings may be used for tubing sizes not exceeding 1 in. O.D. within the limitations of applicable standards and specifications listed in Table NB-3132-1 and requirements (b) and (c) below. In the absence of such standards or specifications, the Designer shall determine that the type of fitting selected is adequate and safe for the Design Loadings in accordance with the requirements of (a), (b), and (c) below.

(a) The pressure design shall meet the requirements of NB-3649.

(b) Fittings and their joints shall be suitable for the tubing with which they are to be used in accordance with the minimum wall thickness of the tubing and method of assembly recommended by the manufacturer.

(c) Fittings shall not be used in services that exceed the manufacturer's maximum pressure–temperature recommendations.

NB-3671.5 Caulked Joints. Caulked or leaded joints shall not be used.

NB-3671.6 Brazed and Soldered Joints

(a) Brazed Joints

(1) Brazed joints of a maximum nominal pipe size of 1 in. may be used only at dead end instrument connections and in special applications where space and geometry conditions prevent the use of joints permitted under NB-3661.2, NB-3661.3, and NB-3671.4. The depth of socket shall be at least equal to that required for socket welding fittings and shall be of sufficient depth to develop a rupture strength equal to that of the pipe at Design Temperature (NB-4500).

(2) Brazed joints that depend upon a fillet rather than a capillary type filler addition are not acceptable.

(3) Brazed joints shall not be used in systems containing flammable fluids or in areas where fire hazards are involved.

(b) Soldered Joints. Soldered joints shall not be used.

NB-3671.7 Sleeve Coupled and Other Patented Joints. Mechanical joints, for which no standards exist, and other patented joints may be used provided the requirements of (a), (b), and (c) below are met.

(a) Provision is made to prevent separation of the joints under all Service Loadings.

(b) They are accessible for maintenance, removal, and replacement after service.

(c) Either of the following two criteria are met.

(1) A prototype joint has been subjected to performance tests to determine the safety of the joint under simulated service conditions. When vibration, fatigue, cyclic conditions, low temperature, thermal expansion, or hydraulic shock is anticipated, the applicable conditions shall be incorporated in the tests. The mechanical joints shall be sufficiently leak tight to satisfy the requirements of the Design Specifications.

(2) Joints are designed in accordance with the rules of NB-3200.

NB-3672 Expansion and Flexibility

(a) In addition to meeting the design requirements for pressure, weight, and other loadings, piping systems shall be designed to absorb or resist thermal expansion or contraction or similar movements imposed by other sources and shall meet the criteria for allowable stress intensity as specified in NB-3611. Piping systems shall be designed to have sufficient flexibility to prevent the movements from causing:

(1) failure of piping or anchors from overstress or overstrain;

(2) leakage at joints;

(3) detrimental distortion of connected equipment resulting from excessive thrusts and moments.

(b) The effects of stresses, caused by pressure, thermal expansion, and other loads and their stress intensification factors, shall be considered cumulatively.

NB-3672.1 Properties. Thermal expansion data and moduli of elasticity shall be determined from Tables I-5.0 and I-6.0, which cover more commonly used piping materials. For materials not included in these tables, reference shall be to authoritative source data, such as publications of the National Bureau of Standards.

NB-3672.2 Unit Thermal Expansion Range. The unit thermal expansion range in in./100 ft, used in calculating the expansion range, shall be determined from Table I-5.0 as the algebraic difference between the unit expansion shown for the highest metal temperature and that for the lowest metal temperature resulting from service or shutdown conditions.

NB-3672.3 Moduli of Elasticity. The moduli of elasticity for ferrous and nonferrous materials shall be as given in Table I-6.0.

NB-3672.4 Poisson's Ratio. When required for flexibility calculations, Poisson's ratio shall be taken as 0.3 for all metals at all temperatures.

NB-3672.5 Stresses. Flexibility calculations of the moments and forces in the piping system due to thermal expansion and end motions shall be based on the hot modulus E_h. Calculations for the expansion stresses shall be based on the least cross-sectional area of the pipe or fitting, using nominal dimensions. The expansion stress computed from the forces and moments shall be multiplied by the ratio E_c/E_h. The effect of expansion stresses in combination with stresses from other causes shall be evaluated in accordance with NB-3611 or NB-3630.

NB-3672.6 Method of Analysis. All systems shall be analyzed for adequate flexibility by a rigorous structural analysis unless they can be judged technically adequate by an engineering comparison with previously analyzed systems.

NB-3672.7 Basic Assumptions and Requirements

(a) When calculating the flexibility of a piping system between anchor points, the system between the anchor points shall be treated as a whole. The significance of all parts of the line and of all restraints, such as supports or guides, including intermediate restraints introduced for the purpose of reducing moments and forces on equipment or small branch lines, shall be considered.

(b) Comprehensive calculations shall take into account the flexibility factors and stress indices found to exist in piping products other than straight pipe. Credit may be taken where extra flexibility exists in the piping system. Flexibility factors and stress indices are given in NB-3680.

(c) The total expansion range shall be used in all calculations whether or not the piping is cold sprung. Not only the expansion of the line itself, but also linear and angular movements of the equipment and supports to which it is attached, shall be considered.

(d) Where assumptions are used in calculations or model tests, the likelihood of underestimates of forces, moments, and stresses, including the effects of stress intensification, shall be evaluated.

NB-3672.8 Cold Springing. Cold springing provides a beneficial effect in assisting a system to attain its most favorable position sooner. The effect of cold springing shall be analyzed as any other movement in the system is analyzed. The maximum stress allowed due to cold springing is $2.0S_m$ at the cold spring temperature. Since the usual erection procedures may not permit accurate determination of cold spring in a piping system, the allowable reduction of forces and moments at anchors or equipment caused by cold springing shall be limited to no more than two-thirds of the calculated reduction.

NB-3674 Design of Pipe Supporting Elements

Supporting elements, including hangers, anchors, and sliding supports, shall be designed in accordance with the requirements of Subsection NF.

NB-3677 Pressure Relief Piping

NB-3677.1 General Requirements. Pressure relief piping within the scope of this Subsection shall be supported to sustain reaction forces and shall conform to the requirements of the following subparagraphs.

NB-3677.2 Piping to Pressure Relieving Safety Devices

(a) Piping that connects a pressure relieving safety device to a piping system shall comply with all the requirements of the class of piping of the system which it is designated to relieve.

(b) There shall be no intervening stop valves between systems being protected and their protective device or devices except as provided for in NB-7142.

NB-3677.3 Discharge Piping From Pressure Relieving Safety Devices

(a) Discharge piping from pressure relieving safety devices shall comply with the requirements applicable to the conditions under which it operates.

(b) There shall be no intervening stop valve between the protective device or devices and the point of discharge except as provided for in NB-7142.

(c) The effluent from relief devices may be discharged outside the containment only if adequate provisions are made for the safe disposal of the effluent. It shall not impinge on other piping or structure or

equipment and shall be directed away from platforms and other areas which might be used by personnel.

(d) It is recommended that individual discharge lines be used, but, if two or more reliefs are combined, the discharge piping shall be designed with sufficient flow area to prevent undue back pressure.

(e) When the umbrella or drip pan type of connection between the pressure relieving safety device and the discharge piping is used, the discharge piping shall be so designed as to prevent binding due to expansion movements and shall be so dimensioned as to prevent the possibility of blow back of the effluent. Individual discharge lines shall be used in this application. Drainage shall be provided to remove water collected above the safety valve seat.

(f) Discharge lines from pressure relieving safety devices within the scope of this Subsection shall be designed to facilitate drainage if there is any possibility that the effluent can contain liquid.

NB-3680 STRESS INDICES AND FLEXIBILITY FACTORS

NB-3681 Scope

(a) There are two types of analyses allowed by the rules of this Subarticle. The applicable B, C, and K indices to be used with Eqs. (9), (10), and (11) of NB-3650 are given in Table NB-3681(a)-1. The applicable indices to be used with the detailed analysis of NB-3200 are given in NB-3685 and NB-3338.

(b) Methods of determining flexibility factors for some commonly used piping products are given in NB-3686.

(c) Values of stress indices are tabulated for commonly used piping products and joints. Unless specific data, which shall be referenced in the Design Report, exist that would warrant lower stress indices than those tabulated or higher flexibility factors than those calculated by the methods of NB-3686, the stress indices given shall be used as minimums and the flexibility factors shall be used as maximums.

(d) For piping products not covered by NB-3680, stress indices and flexibility factors shall be established by experimental analysis (Appendix II) or theoretical analysis. Such test data or theoretical analysis shall be included in the Design Report.

(e) When determining stress indices by experimental methods, the nominal stress at the point under consideration (crack site, point of maximum stress intensity, etc.) shall be used.

NB-3682 Definitions of Stress Indices and Flexibility Factors

(a) The general definition of a stress index for mechanical loads is:

$$B,\ C,\ K,\ \text{or}\ i = \frac{\sigma}{S}$$

where

σ = elastic stress, psi, due to load L

S = nominal stress, psi, due to load L

For B indices, σ represents the stress magnitude corresponding to a limit load. For C or K indices, σ represents the maximum stress intensity due to load L. For i factors, σ represents the principal stress at a particular point, surface, and direction due to load L. The nominal stress S is defined in detail in the tables of stress indices.

(b) The general definition of a stress index for thermal loads is:

$$C\ \text{or}\ K = \frac{\sigma}{E\alpha\Delta T}$$

where

σ = maximum stress intensity, psi, due to thermal difference ΔT

E = modulus of elasticity, psi

α = coefficient of thermal expansion

ΔT = thermal difference, °F

The values of E, α, and ΔT are defined in detail in NB-3650.

(c) Flexibility factors are identified herein by k with appropriate subscripts. The general definition of a flexibility factor is:

$$k = \theta_{ab}/\theta_{\text{nom}}$$

where

θ_{ab} = rotation of end a, with respect to end b, due to a moment load M and in the direction of the moment M

θ_{nom} = nominal rotation due to moment load M

The flexibility factor k and nominal rotation θ_{nom} are defined in detail for specific components in NB-3686.

NB-3683 Stress Indices for Use With NB-3650

The stress indices given herein and in Table NB-3681(a)-1 and subject to the additional restrictions specified herein are to be used with the analysis

TABLE NB-3681(a)-1
STRESS INDICES FOR USE WITH EQUATIONS IN NB-3650

Piping Products and Joints [Note (2)]	Applicable for $D_o/t \leq 100$ for C or K Indices and $D_o/t \leq 50$ for B Indices									Notes
	Internal Pressure [Note (1)]			Moment Loading [Note (1)]			Thermal Loading			
	B_1	C_1 [Note (3)]	K_1 [Note (3)]	B_2	C_2 [Note (3)]	K_2 [Note (3)]	C_3	C'_3	K_3 [Note (3)]	
Straight pipe, remote from welds or other discontinuities	0.5	1.0	1.0	1.0	1.0	1.0	1.0	. . .	1.0	(4)
Longitudinal butt welds in straight pipe										
(a) flush	0.5	1.0	1.1	1.0	1.0	1.1	1.0	. . .	1.1	(5)
(b) as-welded $t > 3/16$ in.	0.5	1.1	1.2	1.0	1.2	1.3	1.0	. . .	1.2	(5)
(c) as-welded $t \leq 3/16$ in.	0.5	1.4	2.5	1.0	1.2	1.3	1.0	. . .	1.2	(5)
Girth butt welds between nominally identical wall thickness items										
(a) flush	0.5	1.0	1.1	1.0	1.0	1.1	0.60	0.50	1.1	(6)
(b) as-welded	0.5	1.0	1.2	1.0	1.0	1.8	0.60	0.50	1.7	(6)
Girth fillet weld to socket weld, fittings, socket weld valves, slip-on or socket welding flanges	0.75	1.8	3.0	1.5	2.1	2.0	2.0	1.0	3.0	(7)
NB-4250 Transitions										
(a) flush	0.5	. . .	1.1	1.0	. . .	1.1	. . .	1.0	1.1	(8)
(b) as-welded	0.5	. . .	1.2	1.0	. . .	1.8	. . .	1.0	1.7	(8)
Transitions within a 1:3 slope envelope										
(a) flush	0.5	. . .	1.2	1.0	. . .	1.1	. . .	0.60	1.1	(9)
(b) as-welded	0.5	. . .	1.2	1.0	. . .	1.8	. . .	0.60	1.7	(9)
Butt welding reducers per ANSI B16.9 or MSS SP-87	1.0	. . .	. . .	1.0	. . .	. . .	1.0	0.5	1.0	(10)
Curved pipe or butt welding elbows	. . .	. . .	1.0	. . .	. . .	1.0	1.0	0.5	1.0	(11)
Branch connections per NB-3643	0.5	. . .	2.0	. . .	. . .	. . .	1.8	1.0	1.7	(12)
Butt welding tees	0.5	1.5	4.0	. . .	. . .	. . .	1.0	0.5	1.0	(13)

NOTES:
(1) For the calculation of pressure and moment loads and special instructions regarding Eqs. (9) through (13), see NB-3683.1(d).
(2) For definitions, applicability, and specific restrictions, see NB-3683.
(3) For special instructions regarding the use of these indices for welded products, intersecting welds, abutting products, or out-of-round products, see NB-3683.2.
(4) See NB-3683.3, Straight Pipe Remote From Welds.
(5) See NB-3683.4(a), Longitudinal Butt Welds.
(6) See NB-3683.4(b), Girth Butt Welds.
(7) See NB-3683.4(c), Girth Fillet Welds.
(8) See NB-3683.5(a), NB-4250 Transitions.
(9) See NB-3683.5(b), Transitions Within a 1:3 Slope.
(10) See NB-3683.6, Concentric and Eccentric Reducers.
(11) See NB-3683.7, Curved Pipe or Butt Welding Elbows. See also NB-3683.2(a) and NB-3683.2(b).
(12) See NB-3683.8, Branch Connections per NB-3643. See also NB-3683.1(d).
(13) See NB-3683.9, Butt Welding Tees. See also NB-3683.1(d).

methods of NB-3650. For piping products outside the applicable range, stress indices shall be established in accordance with NB-3681.

NB-3683.1 Nomenclature

(a) Dimensions. Nominal dimensions as specified in the dimensional standards of Table NB-3132-1 shall be used for calculating the numerical values of the stress indices given herein and in Table NB-3681(a)-1, and for evaluating Eqs. (9) through (14) of NB-3650. For ANSI B16.9, ANSI B16.28, or MSS SP-87 piping products, the nominal dimensions of the equivalent pipe (for example, Schedule 40) as certified by the manufacturer shall be used. Not more than one equivalent pipe size shall be certified for given product items of the same size, shape, and weight.

For piping products such as reducers and tapered-wall transitions which have different dimensions at either end, the nominal dimensions of the large or small end, whichever gives the larger value of D_o/t, shall be used. Dimensional terms are defined as follows:

D_o = nominal outside diameter of pipe, in.

D_i = nominal inside diameter of pipe, in.

D_m = mean diameter of designated run pipe, in. [see NB-3683.8(c) and Fig. NB-3643.3(a)-1]
$= 2R_m = (D_o - T_r)$

D_{max} = maximum outside diameter of cross section, in.

D_{min} = minimum outside diameter of cross section, in.

D_1 = nominal outside diameter at large end of concentric and eccentric reducers, in. (see NB-3683.6)

D_2 = nominal outside diameter at small end of concentric and eccentric reducers, in. (see NB-3683.6)

d_o = nominal outside diameter of attached branch pipe, in.

d_i = nominal inside diameter of branch, in.

d_m = nominal mean diameter of reinforced or unreinforced branch, in. [see NB-3683.8(c)]
$= (d_i + t_n)$

h = characteristic bend parameter of a curved pipe or butt welding elbow
$= tR/r_m^2$

I = moment of inertia of pipe, in.^{4}
$= 0.0491\ (D_o^4 - D_i^4)$

L_1 = height of nozzle reinforcement for branch connections, in. [see Fig. NB-3643.3(a)-1]

L_1, L_2 = length of cylindrical portion at the large end and small end of a reducer, respectively (see NB-3683.6)

R = nominal bend radius of curved pipe or elbow, in.

R_m = mean radius of designated run pipe, in. [see NB-3683.8 and Fig. NB-3643.3(a)-1]
$= (D_o - T_r)/2$

r_i = inside radius of branch, in. [see Fig. NB-3643.3(a)-1]
$= d_i/2$

r_m = mean pipe radius, in.
$= (D_o - t)/2$

r'_m = mean radius of attached branch pipe, in. [see Fig. NB-3643.3(a)-1]
$= (d_o - T'_b)/2$

r_p = outside radius of reinforced nozzle or branch connection, in. [see Fig. NB-3643.3(a)-1]

r_1, r_2, r_3 = designated radii for reinforced branch connections, concentric and eccentric reducers, in. [see NB-3683.6, NB-3683.8, and Fig. NB-3643.3(a)-1]

T'_b = nominal wall thickness of attached branch pipe, in. [see Fig. NB-3643.3(a)-1]

T_b = wall thickness of branch connection reinforcement, in. [see Fig. NB-3643.3(a)-1]

T_r = nominal wall thickness of designated run pipe, in. [see Fig. NB-3643.3(a)-1]

t = nominal wall thickness of pipe, in. For piping products purchased to a minimum wall specification, the nominal wall thickness shall be taken as 1.14 times the minimum wall.

t_n = wall thickness of nozzle or branch connection reinforcement, in. (see NB-3683.8; also used for concentric and eccentric reducers, see NB-3683.6)

t_{max} = maximum wall thickness of a welding transition within a distance of $\sqrt{D_o t}$ from the welding end [see NB-3683.5(b)]

t_1 = nominal wall thickness at large end of concentric and eccentric reducers, in. (see NB-3683.6)

t_2 = nominal wall thickness at small end of concentric and eccentric reducers, in. (see NB-3683.6)

t_{1m}, t_{2m} = minimum wall thickness at the large end and small end of a reducer, respectively, that is required to resist the Design Pressure P in accordance with Eq. (1), NB-3641.1

Z = section modulus of pipe, in.^{3}
$= 2I/D_o$

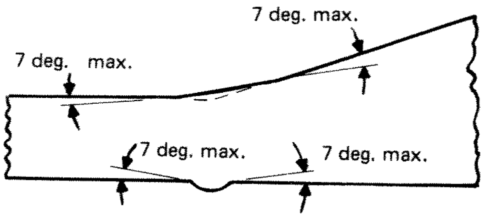


FIG. NB-3683.1(c)-1

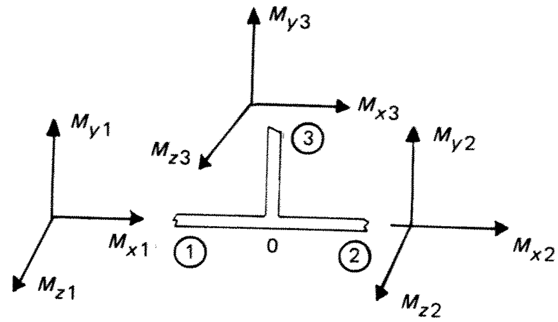


FIG. NB-3683.1(d)-1

Z_b = approximate section modulus of attached branch pipe, in.^{3}
$= \pi(r'_m)^2 T'_b$

Z_r = approximate section modulus of designated run pipe, in.^{3}
$= \pi(R_m)^2 T_r$

α = cone angle of concentric and eccentric reducers, deg. (see NB-3683.6)

δ = average permissible mismatch at girth butt welds as shown in Fig. NB-4233-1. A value of δ less than $\frac{1}{32}$ in. may be used provided the smaller mismatch is specified for fabrication. For *flush* welds as defined in NB-3683.1(c) and for $t > 0.237$ in., δ may be taken as zero.

Δ = radial weld shrinkage measured from the nominal outside surface, in.

θ_n = slope of nozzle-to-pipe transition for branch connections, deg. [see Fig. NB-3643.3(a)-1]

(b) Material Properties. Unless otherwise specified, material properties at the appropriate temperature, as given in Appendix I, shall be used. Terms are defined as follows:

E = modulus of elasticity for the material at room temperature, psi, taken from Table I-6.0

M = materials constant
= 2, for ferritic steels and nonferrous materials except nickel–chrome–iron alloys and nickel–iron–chrome alloys
= 2.7, for austenitic steel, nickel–chrome–iron alloys and nickel–iron–chrome alloys [see NB-3683.2(b)]

S_y = yield strength of the material at the Design Temperature, psi, taken from Table I-2.0

ν = Poisson's ratio
= 0.3

(c) Connecting Welds. Connecting welds in accordance with the requirements of this Subsection are defined as either *flush* or *as-welded* welds.

(1) Flush welds are those welds with contours as defined in Fig. NB-3683.1(c)-1. The total thickness (both inside and outside) of the weld reinforcement shall not exceed $0.1t$. There shall be no concavity on either the interior or exterior surfaces and the finished contour shall not have any slope greater than 7 deg. where the angle is measured from a tangent to the surface of the pipe or on the tapered transition side of the weld to the nominal transition surface.

(2) As-welded welds are those welds not meeting the special requirements of flush welds.

(d) Loadings. Loadings for which stress indices are given include internal pressure, bending and torsional moments, and temperature differences. The indices are intended to be sufficiently conservative to account also for the effects of transverse shear forces normally encountered in flexible piping systems. If, however, thrust or shear forces account for a significant portion of the loading on a given piping product, the effect of these forces shall be included in the design analysis. The values of the moments and forces shall be obtained from an analysis of the piping system in accordance with NB-3672. Loading terms are defined as follows:

P = Design Pressure, psi

P_o = range of service pressure, psi

P^* = maximum value of pressure in the load cycle under consideration, psi

$M_1, M_2,$
M_3 = orthogonal moment loading components at a given position in a piping system, in.-lb

M_i = resultant moment loading applied during the specified operating cycle for straight-through products such as straight pipe, curved pipe or elbows, and concentric reducers

$= \sqrt{M_1^2 + M_2^2 + M_3^2}$

M_{ij} = orthogonal moment components of a tee or branch connection as shown in Fig. NB-3683.1(d)-1 where $i = x, y, z$ and $j = 1,2,3$

The moment components M_{x1}, M_{x2}, M_{y1}, M_{y2}, M_{z1}, and M_{z2} for the run are calculated at the intersection of the run and branch center lines. The moment components M_{x3}, M_{y3}, and M_{z3} for a branch connection where $d_o/D_o \leq 0.5$ may be calculated for a point on the branch center line at a distance $D_o/2$ from the intersection of the run and branch center lines. Otherwise, M_{x3}, M_{y3}, and M_{z3} are calculated at the intersection of the run and branch center lines.

M_{xr}, M_{yr},
M_{zr} = run moment components for use with the stress indices of NB-3683.8 and NB-3683.9. Their numerical values are calculated as follows. If M_{i1} and M_{i2} (where $i = x,y,z$) have the same algebraic sign (+/−), then M_{ir} equals zero. If M_{i1} and M_{i2} have opposite algebraic signs, then M_{ir} equals the smaller of M_{i1} or M_{i2}. If M_{i1} and M_{i2} are unsigned, then M_{ir} may be taken as the smaller of M_{i1} or M_{i2}. Combination of signed and unsigned moments from different load sources shall be done after determination of M_{ir}.

M_b = resultant moment on the branch for branch connections or tees, in.-lb

$= \sqrt{M^2_{x3} + M^2_{y3} + M^2_{z3}}$

M_b^* = same as M_b, except it includes only moments due to thermal expansion and thermal anchor movements

M_r = resultant moment on the run for branch connections or tees, in.-lb

$= \sqrt{M^2_{xr} + M^2_{yr} + M^2_{zr}}$

M_r^* = same as M_r except it includes only moments due to thermal expansion and thermal anchor movements

For branch connections or tees, the pressure term of Eqs. (9), (10), (11), and (13) shall be replaced by the following terms:

For Eq. (9): $B_1\,(PD_o/2T_r)$

For Eqs. (10) and (13): $C_1\,(P_oD_o/2T_r)$

For Eq. (11): $K_1\,C_1\,(P_o\,D_o/2T_r)$

For branch connections or tees, the moment term of Eqs. (9) through (13) shall be replaced by the following pairs of terms:

For Eq. (9): $B_{2b}\,(M_b/Z_b) + B_{2r}\,(M_r/Z_r)$

For Eqs. (10) and (13): $C_{2b}\,(M_b/Z_b) + C_{2r}\,(M_r/Z_r)$

For Eq. (11): $C_{2b}K_{2b}\,(M_b/Z_b) + C_{2r}K_{2r}\,(M_r/Z_r)$

For Eq. (12): $C_{2b}\,(M_b^*/Z_b) + C_{2r}\,(M_r^*/Z_r)$

where the approximate section moduli are:

$$Z_b = \pi(r'_m)^2\,T'_b$$

$$Z_r = \pi(R_m)^2\,T_r$$

NB-3683.2 Applicability of Indices — General. The B, C, and K stress indices given herein and in Table NB-3681(a)-1 predict stresses at a weld joint or within the body of a particular product. The stress indices given for ANSI B16.9, ANSI B16.28, MSS SP-48, and MSS SP-87 piping products apply only to seamless products with no connections, attachments, or other extraneous stress raisers on the body thereof. The stress indices for welds are not applicable if the radial weld shrinkage Δ is greater than $0.25t$.

For products with longitudinal butt welds, the K_1, K_2, and K_3 indices shown shall be multiplied by 1.1 for flush welds or by 1.3 for as-welded welds. At the intersection of a longitudinal butt weld in straight pipe with a girth butt weld or girth fillet weld, the C_1, K_1, C_2, K_2, and K_3 indices shall be taken as the product of the respective indices.

(a) Abutting Products. In general and unless otherwise specified, it is not required to take the product of stress indices for two piping products, such as a tee and a reducer when welded together, or a tee and a girth butt weld. The piping product and the weld shall be qualified separately.

For curved pipe or butt welding elbows welded together or joined by a piece of straight pipe less than one pipe diameter long, the stress indices shall be taken as the product of the indices for the elbow or curved pipe and the indices for the girth butt weld, except for B_1 and C'_3 which are exempted.

(b) Out-of-Round Products. The stress indices given in Table NB-3681(a)-1 are applicable for products and welds with out-of-roundness not greater than $0.08t$ where out-of-roundness is defined as $D_{max} - D_{min}$. For straight pipe, curved pipe, longitudinal butt welds in straight pipe, girth butt welds, NB-4250 transitions,

and 1:3 transitions not meeting this requirement, the stress indices shall be modified as specified below.

(1) If the cross section is out-of-round but with no discontinuity in radius, e.g., an elliptical cross section, an acceptable value of K_1 may be obtained by multiplying the tabulated values of K_1 by the factor F_{1a}:

$$F_{1a} = 1 + \frac{D_{max} - D_{min}}{t}\left[\frac{1.5}{1 + 0.455(D_o/t)^3(p/E)}\right]$$

where

D_o = nominal outside diameter, in.

p = internal pressure (use maximum value of pressure in the load cycle under consideration), psi

E = modulus of elasticity of material at room temperature, psi

Other symbols are defined in (b) above.

(2) If there are discontinuities in radius, e.g., a flat spot, and if $D_{max} - D_{min}$ is not greater than $0.08D_o$, an acceptable value of K_1 may be obtained by multiplying the tabulated values of K_1 by the factor F_{1b}:

$$F_{1b} = 1 + MS_y/(PD_o/2t)$$

where

M = 2, for ferritic steels and nonferrous materials except nickel–chromium–iron alloys and nickel–iron–chromium alloys

= 2.7, for austenitic steel, nickel–chromium–iron alloys, and nickel–iron–chromium alloys

S_y = yield strength at Design Temperature (Tables I-2.0), psi

P = Design Pressure, psi

D_o and t are defined in (a) and (b) above.

NB-3683.3 Straight Pipe Remote From Welds. The stress indices given in Table NB-3681(a)-1 apply for straight pipe remote from welds or other discontinuities, except as modified by NB-3683.2.

NB-3683.4 Connecting Welds. The stress indices given in Table NB-3681(a)-1 are applicable for longitudinal butt welds in straight pipe, girth butt welds joining items with identical nominal wall thicknesses, and girth fillet welds used to attach socket weld fittings, socket weld valves, slip-on flanges, or socket welding flanges, except as modified herein and by NB-3683.2.

(a) Longitudinal Butt Welds. The stress indices shown in Table NB-3681(a)-1 are applicable for longitudinal butt welds in straight pipe, except as modified by NB-3683.2.

(b) Girth Butt Welds. The stress indices shown in Table NB-3681(a)-1, except as modified herein and in NB-3683.2, are applicable to girth butt welds between two items for which the wall thickness is between $0.875t$ and $1.1t$ for an axial distance of $\sqrt{D_o t}$ from the welding ends. Girth welds may also exhibit a reduction in diameter due to shrinkage of the weld material during cooling. The indices are not applicable if Δ/t is greater than 0.25 where Δ is the radial shrinkage measured from the nominal outside surface.

For as-welded girth butt welds joining items with nominal wall thicknesses $t < 0.237$, the C_2 index shall be taken as:

$$C_2 = 1.0 + 3(\delta/t) \quad \text{but not} \quad > 2.1$$

(c) Girth Fillet Welds. The stress indices shown in Table NB-3681(a)-1 are applicable to girth fillet welds used to attach socket weld fittings, socket weld valves, slip-on flanges, or socket welding flanges, except as modified in NB-3683.2.

NB-3683.5 Welded Transitions. The stress indices given in Table NB-3681(a)-1, except as modified herein and in NB-3683.2, are applicable for NB-4250 welded transitions as defined under NB-3683.5(a) and for 1:3 welded transitions as defined under NB-3683.5(b). Girth butt welds may also exhibit a reduction in diameter due to shrinkage of the weld material during cooling. The indices are not applicable if Δ/t is greater than 0.25.

(a) NB-4250 Transitions. The stress indices given in Table NB-3681(a)-1, except as modified herein and in NB-3683.2, are applicable to girth butt welds between an item for which the wall thickness is between $0.875t$ and $1.1t$ for an axial distance of $\sqrt{D_o t}$ from the welding end and another item for which the welding end is within the envelope of Fig. NB-4250-1, but with inside and outside surfaces that do not slope in the same direction. For transitions meeting these requirements, the C_1, C_2, and C_3 indices shall be taken as:

$$C_1 = 0.5 + 0.33\,(D_o/t)^{0.3} + 1.5\,(\delta/t) \quad \text{but not} \quad > 1.8$$

$$C_2 = 1.7 + 3.0\,(\delta/t) \quad \text{but not} \quad > 2.1$$

$$C_3 = 1.0 + 0.03\,(D_o/t) \quad \text{but not} \quad > 2.0$$

For flush welds and for as-welded joints between items with $t > 0.237$, δ may be assumed to be zero.

(b) Transitions Within a 1:3 Slope. The stress indices given in Table NB-3681(a)-1, except as modified herein

and in NB-3683.2, are applicable to girth butt welds between an item for which the wall thickness is between $0.875t$ and $1.1t$ for an axial distance of $\sqrt{D_o t}$ from the welding end and another item for which the welding end is within an envelope defined by a 1:3 slope on the inside, outside, or both surfaces for an axial distance of $\sqrt{D_o t}$, but with inside and outside surfaces that do not slope in the same direction. For transitions meeting these requirements, the C_1, C_2, and C_3 indices shall be taken as:

$$C_1 = 1.0 + 1.5\,(\delta/t) \quad \text{but not} \quad > 1.8$$

$$C_2 = t_{max}/t + 3(\delta/t) \quad \text{but not} \quad > \text{the smaller of } [1.33 + 0.04\sqrt{D_o/t} + 3\,(\delta/t)] \text{ or } 2.1$$

$$C_3 = 0.35\,(t_{max}/t) + 0.25 \quad \text{but not} \quad > 2.0$$

where t_{max} is the maximum wall thickness within the transition zone. If $(t_{max}/t) \leq 1.10$, the stress indices given in NB-3683.4(b) for girth butt welds may be used. For flush welds and for as-welded joints between items with $t > 0.237$, δ may be assumed to be zero.

NB-3683.6 Concentric and Eccentric Reducers. The stress indices given in Table NB-3681(a)-1, except as added to and modified herein and in NB-3683.2, are applicable to butt welding reducers manufactured to the requirements of ANSI B16.9 or MSS SP-87 if the cone angle α defined in Fig. NB-3683.6-1 is less than 60 deg. and if the wall thickness is not less than t_{1m} throughout the body of the reducer, except in and immediately adjacent to the cylindrical portion on the small end where the thickness shall not be less than t_{2m}. The wall thicknesses t_{1m} and t_{2m} are the minimum thicknesses required to resist the Design Pressure P at the large end and small end, respectively, in accordance with Eq. (1) of NB-3641.1. For eccentric reducers, the dimensions shown in Fig. NB-3683.6-1 are to be taken at the location on the circumference where α is the maximum.

(a) Primary Stress Indices. The B_1 stress indices given in (1) or (2) below shall be used depending on the cone angle α.

(1) $B_1 = 0.5$ for $\alpha \leq 30$ deg.

(2) $B_1 = 1.0$ for 30 deg. $< \alpha \leq 60$ deg.

(b) Primary Plus Secondary Stress Indices. The C_1 and C_2 stress indices given in (1) or (2) below shall be used depending on the dimensions of the transition radii r_1 and r_2.

(1) For reducers with r_1 and $r_2 \geq 0.1\,D_1$:

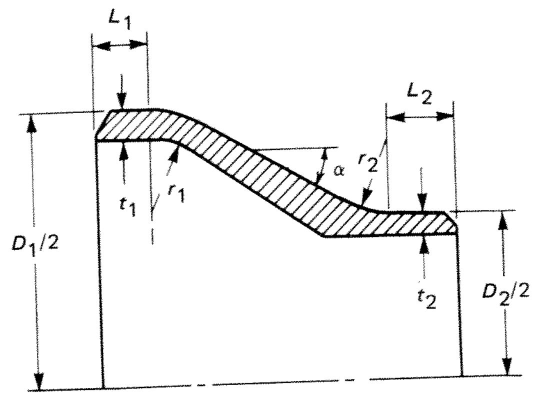


FIG. NB-3683.6-1

$$C_1 = 1.0 + 0.0058\,\alpha\sqrt{D_n/t_n}$$

$$C_2 = 1.0 + 0.36\,\alpha^{0.4}\,(D_n/t_n)^{0.4(D_2/D_1 - 0.5)}$$

where D_n/t_n is the larger of D_1/t_1 and D_2/t_2.

(2) For reducers with r_1 and/or $r_2 < 0.1D_1$:

$$C_1 = 1.0 + 0.00465\,\alpha^{1.285}\,(D_n/t_n)^{0.39}$$

$$C_2 = 1.0 + 0.0185\,\alpha\sqrt{D_n/t_n}$$

where D_n/t_n is the larger of D_1/t_1 and D_2/t_2.

(c) Peak Stress Indices. The K_1 and K_2 indices given in (1), (2), or (3) below shall be used depending on the type of connecting weld, amount of mismatch, and thickness dimensions.

(1) For reducers connected to pipe with flush girth butt welds:

$$K_1 = 1.1 - 0.1\,L_m/\sqrt{D_m t_m} \quad \text{but not} \quad < 1.0$$

$$K_2 = 1.1 - 0.1\,L_m/\sqrt{D_m t_m} \quad \text{but not} \quad < 1.0$$

where $L_m/\sqrt{D_m t_m}$ is the smaller of $L_1/\sqrt{D_1 t_1}$ and $L_2/\sqrt{D_2 t_2}$.

(2) For reducers connected to pipe with as-welded girth butt welds where t_1 or $t_2 > 3/16$ in. and δ_1/t_1 or $\delta_2/t_2 \leq 0.1$:

$$K_1 = 1.2 - 0.2\,L_m/\sqrt{D_m t_m} \quad \text{but not} \quad < 1.0$$

$$K_2 = 1.8 - 0.8\,L/\sqrt{D_m t_m} \quad \text{but not} \quad < 1.0$$

where $L_m/\sqrt{D_m t_m}$ is the smaller of $L_1/\sqrt{D_1 t_1}$ and $L_2/\sqrt{D_2 t_2}$.

(3) For reducers connected to pipe with as-welded girth butt welds where t_1 or $t_2 \leq 3/16$ in. or δ_1/t_1 or $\delta_2/t_2 > 0.1$:

$$K_1 = 1.2 - 0.2\, L_m/\sqrt{D_m t_m} \quad \text{but not} \quad < 1.0$$

$$K_2 = 2.5 - 1.5\, L_m/\sqrt{D_m t_m} \quad \text{but not} \quad < 1.0$$

where $L_m/\sqrt{D_m t_m}$ is the smaller of $L_1/\sqrt{D_1 t_1}$ and $L_2/\sqrt{D_2 t_2}$.

NB-3683.7 Curved Pipe or Butt Welding Elbows. The stress indices given in Table NB-3681(a)-1, except as added to and modified herein and in NB-3683.2, are applicable to curved pipe or butt welding elbows manufactured to the requirements of ANSI B16.9, ANSI B16.28, MSS SP-48, or MSS SP-87.

(a) Primary Stress Index. The B_1 and B_2 indices shall be taken as:

$$B_1 = -0.1 + 0.4h \quad \text{but not} \quad < 0 \quad \text{nor} \quad > 0.5$$

$$B_2 = 1.30/h^{2/3} \quad \text{but not} \quad < 1.0$$

where

$$h = tR/r_m^2$$

(b) Primary Plus Secondary Stress Indices. The C_1 and C_2 indices shall be taken as:

$$C_1 = (2R - r_m)/2(R - r_m)$$

$$C_2 = 1.95/h^{2/3} \quad \text{but not} \quad < 1.5$$

where

$$h = tR/r_m^2$$

NB-3683.8 Branch Connections per NB-3643. The stress indices given in Table NB-3681(a)-1, except as added to and modified herein and in NB-3683.2, are applicable to reinforced or unreinforced branch connections meeting the general requirements of NB-3643 and the additional requirements of NB-3683.8(a). Symbols are defined in NB-3683.1 and in Fig. NB-3643.3(a)-1.

(a) Applicability. The stress indices are applicable, provided the following limitations are met.

(1) For branch connections in a pipe, the arc distance measured between the centers of adjacent branches along the outside surface of the run pipe is not less than three times the sum of the two adjacent branch inside radii in the longitudinal direction, or is not less than two times the sum of the two adjacent branch radii along the circumference of the run pipe.

(2) The axis of the branch connection is normal to the run pipe surface.

(3) The run pipe radius-to-thickness ratio R_m/T_r is less than 50, and the branch-to-run radius ratio r'_m/R_m is less than 0.50.

(4) The inside corner radius r_1 [Fig. NB-3643.3(a)-1] for nominal pipe sizes greater than 4 in. NPS shall be between 10% and 50% of T_r. The radius r_1 is not required for branch pipe sizes smaller than 4 in. NPS.

(5) The branch-to-run fillet radius r_2 is not less than the larger of $T_b/2$, $T_r/2$, or $(T'_b + y)/2$ [Fig. NB-3643.3(a)-1(c)].

(6) The branch-to-pipe fillet radius r_3 is not less than the larger of 0.002 $\theta\, d_o$ or $2(\sin\theta)^3$ times offset [Fig. NB-3643.3(a)-1], where θ is expressed in deg.

(7) If L_1 equals or exceeds $0.5\sqrt{r_i T_b}$, then r'_m can be taken as the radius to the center of T_b.

(b) Primary Stress Indices. The primary stress indices B_{2b} and B_{2r} shall be taken as:

$$B_{2b} = 0.5C_{2b} \quad \text{but not} \quad < 1.0$$

$$B_{2r} = 0.75C_{2r} \quad \text{but not} \quad < 1.0$$

(c) Primary Plus Secondary Stress Indices. The C_1, C_{2b}, and C_{2r} indices [for moment loadings, see NB-3683.1(d)] shall be taken as:

$$C_1 = 1.4\left(\frac{D_m}{T_r}\right)^{0.182}\left(\frac{d_m}{D_m}\right)^{0.367}\left(\frac{T_r}{t_n}\right)^{0.382}\left(\frac{t_n}{r_2}\right)^{0.148}$$

$$\text{but not} < 1.2$$

If $r_2/t_n > 12$, use $r_2/t_n = 12$ for computing C_1.

$$C_{2b} = 3\left(\frac{R_m}{T_r}\right)^{2/3}\left(\frac{r'_m}{R_m}\right)^{1/2}\left(\frac{T'_b}{T_r}\right)\left(\frac{r'_m}{r_p}\right)$$

$$\text{but not} < 1.5$$

$$C_{2r} = 1.15\left(\frac{r'_m}{t_n}\right)^{1/4}$$

$$\text{but not} < 1.5$$

where

For Figs. NB-3643.3(a)-1(a) and (b):

$$t_n = T_b \text{ if } L_1 \geq 0.5\ (d_m T_b)^{1/2}$$

$$= T'_b \text{ if } L_1 < 0.5\ (d_m T_b)^{1/2}$$

For Fig. NB-3643.3(a)-1(c):

$$t_n = T'_b + (2/3)y \quad \text{if} \quad \theta \leq 30°$$

$$= T'_b + 0.385\ L_1 \quad \text{if} \quad \theta > 30°$$

For Fig. NB-3643.3(a)-1(d):

$$t_n = T'_b = T_b$$

(d) Peak Stress Indices. The peak stress indices K_{2b} and K_{2r} for moment loadings [see NB-3683.1(d)] shall be taken as:

$$K_{2b} = 1.0$$

$$K_{2r} = 1.75$$

and $K_{2r}C'_{2r}$ shall be a minimum of 2.65.

NB-3683.9 Butt Welding Tees. The stress indices given in Table NB-3681(a)-1, except as added to and modified herein and in NB-3683.2, are applicable to butt welding tees manufactured to the requirements of ANSI B16.9, MSS SP-48, or MSS SP-87.

(a) Primary Stress Indices. The primary stress indices B_{2b} and B_{2r} shall be taken as:

$$B_{2b} = 0.4\ (R_m/T_r)^{2/3} \quad \text{but not} \quad < 1.0$$

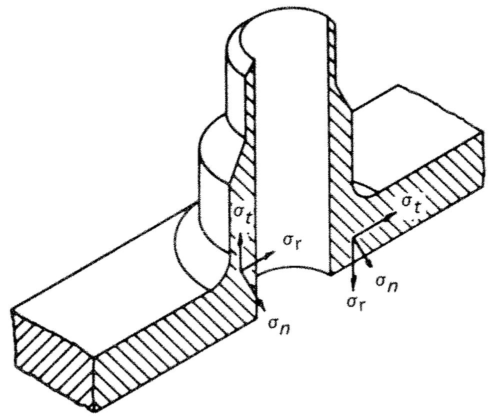
$$B_{2r} = 0.50\ (R_m/T_r)^{2/3} \quad \text{but not} \quad < 1.0$$

(b) Primary Plus Secondary Stress Indices. The C_{2b} and C_{2r} stress indices for moment loadings [see NB-3683.1(d)] shall be taken as:

$$C_{2b} = 0.67\ (R_m/T_r)^{2/3} \quad \text{but not} \quad < 2.0$$

$$C_{2r} = 0.67\ (R_m/T_r)^{2/3} \quad \text{but not} \quad < 2.0$$

(c) Peak Stress Indices. The peak stress indices K_{2b} and K_{2r} for moment loadings [see NB-3683.1(d)] shall be taken as:



σ_t = stress component in the plane of the section under consideration and parallel to the boundary of the section
σ_n = stress component normal to the plane of the section
σ_r = stress component normal to the boundary of the section
σ = stress intensity (combined stress) at the point under consideration

FIG. NB-3684-1 DIRECTION OF STRESS COMPONENTS

$$K_{2b} = 1.0$$

$$K_{2r} = 1.0$$

NB-3684 Stress Indices for Detailed Analysis

The symbols for the stress components and their definitions are given in Fig. NB-3684-1. These definitions are applicable to all piping products, and the stress indices given in the tables in NB-3685 and NB-3338 are so defined.

NB-3685 Curved Pipe or Welding Elbows

NB-3685.1 Applicability of Indices. The indices given in Tables NB-3685.1-1 and NB-3685.1-2 give stresses in curved pipe or elbows at points remote from girth or longitudinal welds or other local discontinuities. Stresses in elbows with local discontinuities, such as longitudinal welds, support lugs, and branch connections in the elbow, shall be obtained by appropriate theoretical analysis or by experimental analysis in accordance with Appendix II.

TABLE NB-3685.1-1
CURVED PIPE OR WELDING END ELBOWS, INTERNAL PRESSURE

Location	Surface	Stress Direction	Stress Index [Note (1)]
		Round Cross Section	
ϕ	Inside	σ_n	$\left[\dfrac{D_o - 0.8(t_m - A)}{2(t_m - A)}\right]\left[\dfrac{0.5(2R + r\sin\phi)}{R + r\sin\phi}\right] = i_1$
ϕ	Mid	σ_n	
ϕ	Outside	σ_n	
ϕ	Inside	σ_t	$\dfrac{D_i}{4(t_m - A)} = i_2$
ϕ	Mid	σ_t	
ϕ	Outside	σ_t	
		Out-of-Round Cross Section [Note (2)]	
α	Inside	σ_n	$i_1 - i_3$
α	Mid	σ_n	i_1
α	Outside	σ_n	$i_1 - i_3$
α	Inside	σ_t	$i_2 - 0.3i_3$
α	Mid	σ_t	i_2
α	Outside	σ_t	$i_2 - 0.3i_3$

NOTES:
(1) The radial stress σ_r is equal to $-P$ on the inside surface, to $-P/2$ on the midsurface, and to 0 on the outside surface.
(2) For out-of-round cross section:

$$i_3 = \left[\frac{D_o(D_1 - D_2)}{2t_m^2}\right]\left[\frac{1.5}{1 + 0.455(D_o/t_m)^3(P/E)}\right]\cos 2\alpha$$

NB-3685.2 Nomenclature (Fig. NB-3685.2-1)

P = internal pressure, psi
D_o = nominal outside diameter of cross section, in.
$D_i = D_o - 2(t_m - A)$, in.
t_m = minimum specified wall thickness, in.
A = an additional thickness, in. (NB-3641.1)

R = bend radius, in.
r = mean cross section radius, in.
$\lambda = t_m R/r^2\sqrt{1 - \nu^2}$ (Table NB-3685.1-2 limited to $\lambda \geq 0.2$)
$D_1(D_2)$ = maximum (minimum) outside diameter of elbow with out-of-round cross section essentially describable as an ellipse or oval shape (Fig. NB-3685.2-1), in.
Z = section modulus of cross section, in.^{3}
$= 0.0982\,(D_o^4 - D_i^4)/D_o$
E = modulus of elasticity, psi (Table I-5.0)

NB-3685.3 Stress From Stress Indices. To obtain stresses from stress index:

Load	Multiply Stress Index by
Internal Pressure	P
M_x	$M_x/2Z$
M_y	M_y/Z
M_z	M_z/Z

NB-3685.4 Classification of Stresses. For analysis of a curved pipe or welding elbow to NB-3210, the following rules shall apply to the classification of stresses developed under a load controlled in-plane or out-of-plane moment as distinguished from a displacement controlled loading.

(a) The entire membrane portion of the axial, circumferential, and torsional stresses shall be considered as primary (P_L).

(b) Seventy-five percent of the through-wall bending stresses in both the axial and the circumferential di-

TABLE NB-3685.1-2
CURVED PIPE OR WELDING END ELBOWS, MOMENT LOADING ($\lambda \geq 0.2$)

Location	Surface	Stress Direction	Stress Index [Note (1)]
		Torsional Moment M_x	
All	All	τ_{nt} [Note (2)]	1.0
		In-Plane or Out-of-Plane Moments M_y or M_z [Note (3)]	
ϕ	Outside	σ_n	$\nu\sigma_{tm} + \sigma_{nb}$
ϕ	Mid	σ_n	$\nu\sigma_{tm}$
ϕ	Inside	σ_n	$\nu\sigma_{tm} - \sigma_{nb}$
ϕ	Outside	σ_t	$\sigma_{tm} + \nu\sigma_{nb}$
ϕ	Mid	σ_t	σ_{tm}
ϕ	Inside	σ_t	$\sigma_{tm} - \nu\sigma_{nb}$

NOTES:
(1) The radial stress σ_r is zero for all surfaces.
(2) τ_{nt} is a shear stress in the n–t plane and must be appropriately combined with the principal stresses σ_n and σ_t to obtain principal stresses due to combinations of M_x with M_y or M_z.
(3) Nomenclature for stress indices:

ν = Poisson's ratio

$$\left.\begin{aligned}\sigma_{tm} &= \sin\phi + [(1.5X_2 - 18.75)\sin 3\phi + 11.25\sin 5\phi]/X_4 \\ \sigma_{nb} &= \lambda\,(9X_2\cos 2\phi + 225\cos 4\phi)/X_4\end{aligned}\right\}\ \text{In-plane } M_z$$

$$\left.\begin{aligned}\sigma_{tm} &= \cos\phi + [(1.5X_2 - 18.75)\cos 3\phi + 11.25\cos 5\phi]/X_4 \\ \sigma_{nb} &= \lambda\,(9X_2\sin 2\phi + 225\sin 4\phi)/X_4\end{aligned}\right\}\ \text{Out-of-plane } M_y$$

$X_1 = 5 + 6\lambda^2 + 24\Psi$

$X_2 = 17 + 600\lambda^2 + 480\Psi$

$X_3 = X_1X_2 - 6.25$

$X_4 = (1 - \nu^2)(X_3 - 4.5X_2)$

$\lambda = t_mR/(r^2\sqrt{1 - \nu^2})$ (Equations are valid for $\lambda \geq 0.2$ only.)

$\Psi = PR^2 / Ert_m$

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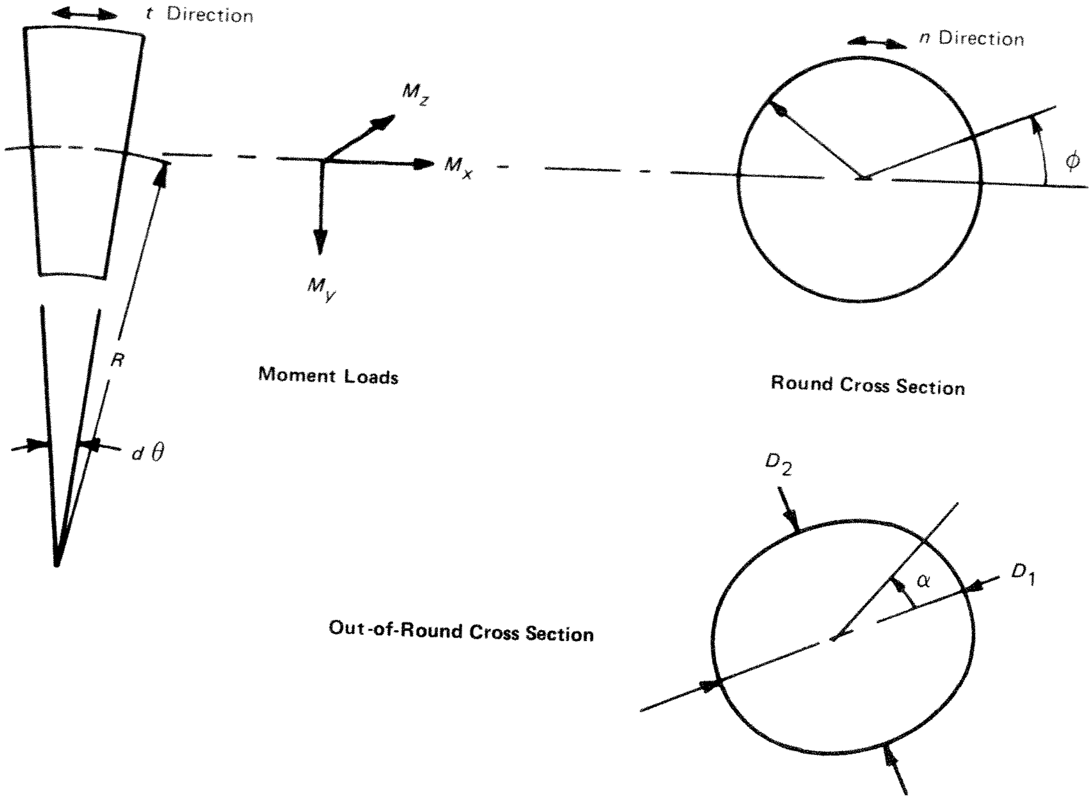

FIG. NB-3685.2-1 ELBOW NOMENCLATURE

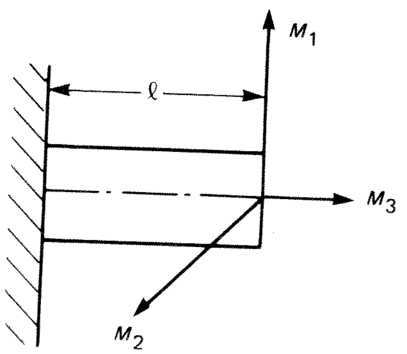

FIG. NB-3686.1-1

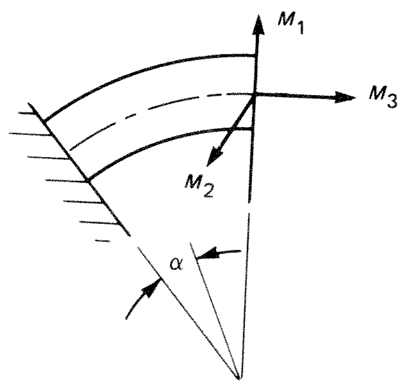

FIG. NB-3686.2-1

rections shall be classified as primary (P_b). The remaining 25% shall be classified as secondary (Q). The stresses induced by displacement controlled in-plane or out-of-plane moments shall be classified as secondary (Q).

NB-3686 Flexibility Factors

NB-3686.1 Straight Pipe. For $M = M_1$ or M_2 (see Fig. NB-3686.1-1):

$$k = 1.0 \quad \theta_{nom} = \frac{Ml}{EI}$$

For $M = M_3$ (see Fig. NB-3686.1-1):

$$k = 1.0 \quad \theta_{nom} = \frac{Ml}{GJ}$$

In both cases
- l = one pipe diameter
- I = plane moment of inertia, in.^{4}
- J = polar moment of inertia, in.^{4}
- E = modulus of elasticity, psi
- G = shear modulus, psi

NB-3686.2 Curved Pipe and Welding Elbows. The flexibility factors may be calculated by the equations given below for k, provided[23] that:

(a) R/r is not less than 1.7;

(b) center line length $R\alpha$ is greater than $2r$;

(c) there are no flanges or other similar stiffeners within a distance r from either end of the curved section of pipe or from the ends of welding elbows.

For M_1 or M_2 (see Fig. NB-3686.2-1):

$$k = \frac{1.65}{h}\left[\frac{1}{1 + (Pr/tE)X_k}\right]$$

but not less than 1.0, and

$$\theta_{nom} = \frac{R}{EI}\int_o^a M(d\alpha)$$

For M_3 (see Fig. NB-3686.2-1):

$$k = 1.0$$

$$\theta_{nom} = \frac{R}{GJ}\int_o^a M(d\alpha)$$

In both cases
- $h = tR/r^2$
- R = bend radius, in.
- P = internal pressure, psi
- r = pipe or elbow mean radius, in.
- t = pipe or elbow nominal wall thickness, in.
- $X_k = 6(r/t)^{4/3}(R/r)^{1/3}$
- I = plane moment of inertia of cross section, in.^{4}
- J = polar moment of inertia of cross section, in.^{4}
- E = modulus of elasticity, psi
- G = shear modulus of elasticity, psi
- α = arc angle, rad

NB-3686.3 Miter Bends. The requirements of NB-3681(d) apply.

NB-3686.4 Welding Tee or Branch Connections. For welding tees (ANSI B16.9) or branch connections (NB-3643) not included in NB-3686.5, the load displacement relationships shall be obtained by assuming that the run pipe and branch pipe extend to the intersection of the run pipe center line with the branch pipe center line. The imaginary juncture is to be assumed rigid, and the imaginary length of branch pipe from the juncture to the run pipe surface is also to be assumed rigid.

NB-3686.5 Branch Connections in Straight Pipe. (For branch connections in straight pipe meeting the dimensional limitations of NB-3338.) The load displacement relationships may be obtained by modeling the branch connections in the piping system analysis (NB-3672) as shown in (a) through (d) below. (See Fig. NB-3686.5-1.)

(a) The values of k are given below.

For M_{x3}:

$$k = 0.1\,(D/T_r)^{1.5}\,[(T_r/t_n)(d/D)]^{1/2}\,(T'_b/T_r)$$

For M_{z3}:

$$k = 0.2\,(D/T_r)[(T_r/t_n)(d/D)]^{1/2}\,(T'_b/T_r)$$

where
- $M = M_{x3}$ or M_{z3}, as defined in NB-3683.1(d)
- D = run pipe outside diameter, in.

[23]The flexibility of a curved pipe or welding elbow is reduced by *end effects,* provided either by the adjacent straight pipe or by the proximity of other relatively stiff members which inhibit ovalization of the cross section. In certain cases, these end effects may also reduce the stress. Additional work is underway to provide guidance for both flexibility factors and stress indices where end effects are significant.

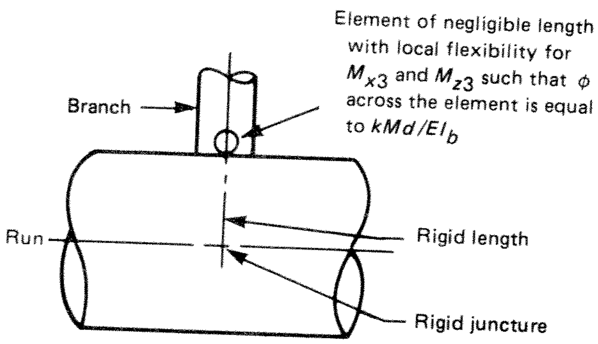

FIG. NB-3686.5-1 BRANCH CONNECTIONS IN STRAIGHT PIPE

d = branch pipe outside diameter, in.
I_b = moment of inertia of branch pipe, in.[4] (to be calculated using d and T'_b)
E = modulus of elasticity, psi
T_r = run pipe wall thickness, in.
ϕ = rotation in direction of moment, rad

(b) For branch connections per Fig. NB-3643.3(a)-1 sketches (a) and (b):

$$t_n = T_b \quad \text{if} \quad L_1 \geq 0.5\,[(2r_i + T_b)T_b]^{1/2}$$

$$= T'_b \quad \text{if} \quad L_1 < 0.5\,[(2r_i + T_b)T_b]^{1/2}$$

(c) For branch connections per Fig. NB-3643.3(a)-1 sketch (c):

$$t_n = T'_b + (2/3)y \quad \text{if} \quad \theta \leq 30 \text{ deg.}$$

$$= T'_b + 0.385L_1 \quad \text{if} \quad \theta > 30 \text{ deg.}$$

(d) For branch connections per Fig. NB-3643.3(a)-1 sketch (d):

$$T'_b = T_b$$

NB-3690 DIMENSIONAL REQUIREMENTS FOR PIPING PRODUCTS

NB-3691 Standard Piping Products

Dimensions of standard piping products shall comply with the standards and specifications listed in Table NB-3132-1. However, compliance with these standards does not replace or eliminate the requirements of NB-3625.

NB-3692 Nonstandard Piping Products

The dimensions of nonstandard piping products shall be such as to provide strength and performance as required by this Subsection. Nonstandard piping products shall be designed in accordance with NB-3640.