ARTICLE KD-6
DESIGN REQUIREMENTS FOR CLOSURES, INTEGRAL HEADS, THREADED FASTENERS, AND SEALS

KD-600 SCOPE

The requirements in this Article apply to integral heads, closures, threaded fasteners, and seals. These requirements are additional to the general requirements given in Articles KD-1 and KD-2.

KD-601 GENERAL

(a) Closures, integral heads, threaded fasteners, and seals shall have the capability to contain pressure with the same assurance against failure as the vessel for which it will be used.

(b) The Designer shall consider the influence of cross bores and other openings on the static strength integrity of the vessel.

(c) A complete stress analysis shall be made of all components that contribute to the strength and sealing capability of the closure.

(d) For applications involving cyclic loads, the requirements of Articles KD-3 or KD-4, as applicable, shall be met for all parts except the sealing element.

(e) Provisions shall be made to prevent separation of joints under all service loadings.

(f) The effects of the total load to be resisted, the number of threads, the number of threaded fasteners, the thread form, the relative stiffness of mating parts, and friction shall be considered in both the static and fatigue analyses.

(g) Vent passages shall be provided to prevent pressure buildup caused by accidental or incidental development of any secondary sealing areas exterior to the designated sealing surface (e.g., threads).

(h) Flared, flareless, and compression-type joints for tubing are not permitted. Proprietary fittings are addressed in KD-625.

KD-620 THREADED FASTENERS AND COMPONENTS

(a) Threaded fasteners are frequently described as bolts, studs, and tie rods.

(b) Straight threaded connections are permitted as provided for in this Article. Tapered pipe threads are not permitted.

(c) Where tapped holes are provided in pressure boundaries, the effect of such holes (e.g., stress riser, material loss) shall be considered in the vessel design.

(d) Thread load distribution shall be considered in design cyclic analysis in accordance with KD-622.

KD-621 ELASTIC–PLASTIC BASIS

In lieu of the requirements of KD-623(a) through KD-623(g), the Designer may use the elastic–plastic method and meet the applicable requirements of KD-230 for all threaded joints or fasteners of any thread form.

(a) The elastic–plastic rules of KD-231 are applied for all the loads and load cases to be considered as listed in Table KD-230.1 and defined in KD-231.2.

(b) The load combinations and load factors as listed in Table KD-230.4 are applied and the components are stable under the applied loads.

KD-622 FATIGUE AND FRACTURE MECHANICS ANALYSIS

(a) A fatigue analysis in accordance with Article KD-3 or a fracture mechanics analysis in accordance with Article KD-4 is required for all threaded connections.

(b) The fatigue evaluation of a threaded joint is made by the same methods as are applied to any other structure that is subjected to cyclic loading.

(c) ASME B18.2.2 Standard nuts of materials permitted by this Division do not require fatigue analysis. Internal threads mating with a stud or bolt do not require fatigue analysis for bolting loads. However, the effects of the internally threaded penetration on the nominal primary-plus-secondary stresses in the internally threaded member shall be considered.

KD-623 LINEAR ELASTIC BASIS

Linear elastic analysis may be used under the following conditions:

(a) The number and cross-sectional area of bolts required to resist primary loads shall be determined. The yield strength values to be used are the values given in Section II, Part D for bolting materials.
The relative strength of mating parts shall be considered.

The relative strength of mating parts shall be considered.

(b) The average primary stress intensity $S$ shall be based on the thread root diameter and shall not exceed the following limit:

$$S = \frac{1}{1.8}S_y$$

(c) For bolts with a reduced shank, which has a diameter less than 0.9 times the thread root diameter, the above equation shall be replaced by:

$$S = \frac{1}{1.5}S_y$$

provided the actual shank diameter is used.

(d) Primary-plus-secondary membrane stress intensity in bolts shall not exceed $0.75S_y$. Primary-plus-secondary membrane plus bending stress intensity in bolts shall not exceed $S_y$ due to the combination of both the design loads and preloads. Stress intensification due to the threads shall not be considered in the above analysis.

(e) If a standard bolt and nut pair conforming to material specifications in Section II, Part D is used and both members are of the same material, the thread shear and bearing capability need not be qualified further.

(f) The average shear stress in the threads, calculated by dividing the design load by the appropriate thread shear area, shall be limited to $0.25S_y$ at the design temperature.

(g) The average bearing stress in the threads due to the maximum design loading shall be limited to $0.75S_y$ at the design temperature.

(h) Relative radial displacement between mating threads shall be calculated considering the combination of applied loads and thermal effects. No credit shall be taken for thread friction. The results of this analysis shall demonstrate that the threads having relative radial displacement less than 10% of the minimum thread overlap meet the requirements of (f) and (g). No credit shall be taken for threads whose relative radial displacement exceeds 10%.

(i) The length of engagement is to be taken as the minimum which can occur within the drawing tolerances with no credit for partial threads.

(j) Connections which have imposed loads on threads in tapped holes shall comply with the requirements of (k). The vessel or an integral weld buildup shall have a flat surface machined on the shell to receive the connection.

(k) Where tapped holes are provided, the threads shall be full and clean and the engaged length shall not be less than the larger of $d_e$ or

$$0.75d_e \left( \frac{S_y \text{ of stud material at design temperature}}{S_y \text{ of tapped material at design temperature}} \right)$$

in which $d_e$ is the root diameter of the stud.
KD-624  THREADING AND MACHINING OF STUDS

Studs shall be threaded full length, or shall be machined down to the root diameter of the thread in the unthreaded portion. The threaded portions shall have a length of at least 1 1/2 times the nominal diameter, unless analysis (see KD-621) using the most unfavorable combination of tolerances at assembly demonstrates adequate thread engagement is achieved with a shorter thread length.

Studs greater than eight times the nominal diameter in length may have an unthreaded portion which has the nominal diameter of the stud, provided the following requirements are met.

(a) The stud shall be machined down to the root diameter of the thread for a minimum distance of 0.5 diameters adjacent to the threaded portion.

(b) A suitable transition shall be provided between the root diameter portion and the full diameter portion.

(c) Threads shall be of a "V" type, having a minimum thread root radius no smaller than 0.08 times the pitch.

(d) Fillet radii at the end of the shank shall be such that the ratio of fillet radius to shank diameter is not less than 0.06.

KD-625  SPECIAL THREADS AND PROPRIETARY JOINTS

Mechanical joints for which no standards exist and other proprietary joints may be used. A prototype of such a proprietary joint shall be subjected to performance tests to determine the safety of the joint under simulated service loadings in accordance with Article KD-12. When vibration, fatigue, cyclic conditions, low temperature, thermal expansion, or hydraulic shock is anticipated, the applicable loads shall be incorporated in the tests.

KD-630  LOAD-CARRYING SHELL WITH SINGLE THREADED END CLOSURES

Because of the many variables involved, and in order not to restrict innovative designs, detailed rules are kept to a minimum. The effects of the total load to be resisted, the number of threads, the thread form, the relative stiffness of mating parts, and friction shall be considered in both the static and fatigue analyses of the closure. Stresses can be minimized by providing generous undercuts ahead of the first threads and providing flexibility in mating parts to promote equalization of the thread loads.

KD-631  STRESSES IN VESSEL AT THREADS

The Designer shall identify the area of the threaded closure where the maximum stress intensity occurs. This is generally the area at the root of the most highly loaded thread, which is usually the first or second thread. Calculation of this stress intensity requires consideration of the actual thread load, stress concentration factor due to
thread form (in particular, the thread root radius), thread bending stress, and the membrane and bending stresses in the vessel at the thread.

**KD-631.1 Longitudinal Bending Stresses.** Unless it can be shown by analysis or test that a lower value is appropriate, the primary longitudinal bending stress in the vessel at the first thread shall be considered to be 3.0 times the primary longitudinal membrane stress.

**KD-631.2 Circumferential Stresses.** The circumferential stresses are significantly affected by the distance to the pressure seal. Unless shown by analysis or test that a lower value is appropriate, the circumferential stresses in the vessel at the first thread shall be considered to be those in the cylinder derived with the equations in 9-200. In addition, circumferential stresses due to resultant radial loading of the threads shall be included.

**KD-631.3 Thread Load Distribution.** In general, the threads do not carry the end load uniformly. The Designer shall determine thread load distribution. See E-200.

**KD-631.4 Fracture Mechanics Analysis.** Fracture mechanics analysis shall be made in accordance with Article KD-4. This analysis shall include as a minimum the combined effects of bending of the thread, and the shell membrane and bending stresses.

**KD-631.5 Progressive Distortion.** Screwed-on caps and screwed-in plugs are examples of nonintegral connections which are subject to failure by bellmouthing or other types of progressive deformation. Such joints may be subject to ratcheting, causing the mating members to progressively disengage. See KD-210(e)(9).

**KD-631.6 Interrupted Threads.** Closures utilizing interrupted threads may be analyzed as closures with continuous threads provided that a multiplier is applied to the resultant stresses. The multiplier is the ratio of the continuous thread circumferential length to that of the interrupted thread. The contact length used when calculating the stress distribution for an interrupted thread may be less than the thread length because of the profiling of the thread ends.

**KD-634 SPECIAL CLOSURES AND MATERIALS**

(a) Threaded closures for which no standards exist may be used, provided the closure is analyzed in accordance with the rules of Articles KD-2, KD-3, and KD-4, or a prototype has been evaluated in accordance with the rules of Article KD-12.

(b) For parts for which it is impossible or impractical to measure the yield strength after final processing, the maximum allowable tensile stress at design pressure shall be one-third the ultimate strength at design temperature, so long as the final processing does not adversely affect the ultimate strength.

**KD-640 INTEGRAL HEADS**

Integral heads shall be designed in accordance with KD-230 or Mandatory Appendix 9. The designer may use Nonmandatory Appendix E instead of KD-230 or Mandatory Appendix 9 if the conditions in Nonmandatory Appendix E are satisfied.

**KD-650 QUICK-ACTUATING CLOSURES**

**KD-651 GENERAL DESIGN REQUIREMENTS**

Quick-acting closures shall be so designed and installed that it can be determined by visual external observation that the holding elements are in good condition and that their locking elements, when the closure is in the closed position, are in full engagement. Alternatively, other means may be provided to ensure full engagement.

**KD-652 SPECIFIC DESIGN REQUIREMENTS**

Quick-acting closures that are held in position by positive locking devices and that are fully released by partial rotation or limited movement of the closure itself or the locking mechanism, and any automated closure, shall be designed to meet the following conditions:

(a) The closure and its holding elements are fully engaged in their intended operating position before the vessel can be pressurized.

(b) Pressure tending to open the closure shall be released before the locking mechanism is disengaged.

(c) A coefficient of friction less than or equal to 0.02 shall be used in the design analysis.

**KD-652.1 Permissible Design Deviations for Manually Operated Closures.** Quick-acting closures that are held in position by a locking device or mechanism that requires manual operation and are so designed that there shall be leakage of the contents of the vessel prior to disengagement of the locking elements and release of closure need not satisfy KD-652(a), KD-652(b), and KD-652(c). However, such closures shall be equipped with an audible or visible warning device that shall serve to warn the operator if pressure is applied to the vessel before the closure and its holding elements are fully engaged in their intended position and, further, will serve to warn the operator if an attempt is made to operate the locking mechanism or device before the pressure within the vessel is released.

**KD-652.2 Yokes.** Yokes or frames are quick-acting closures that shall comply with all the requirements of this Division.

**KD-653 REQUIRED PRESSURE-INDICATING DEVICES**

All vessels having quick-acting closures shall be provided with a pressure-indicating device visible from the operating station.
KD-660  REQUIREMENTS FOR CLOSURES AND SEALS

The requirement of a leak-tight seal is of primary importance in closures for high pressure vessels. This is because even small leaks produce a damaging (cutting) effect through the sealing surfaces, which may progress rapidly to increasingly hazardous conditions.

KD-661  REQUIREMENTS FOR CLOSURES

(a) Adequate venting shall be provided in the closure design in the event of seal failure.

(b) The effects of dilation, distortion, or both on the closure components under all expected conditions of pressure and temperature shall not result in an increase in the seal clearances greater than the values required to retain the sealing element.

KD-662  REQUIREMENTS FOR SEALING ELEMENTS

The material selected shall be compatible with all normally expected process and environmental conditions, such as pressure, temperature, corrosion, solubility, chemical reaction, etc., as specified in the User's Design Specification.

KD-662.1  Contained Sealing Elements. The materials of construction for sealing elements are generally not covered in Part KM. The User's Design Specification shall either specify the required material or furnish enough information to enable the Designer to make an appropriate selection.

KD-662.2  Unsupported Metallic Sealing Elements. Sealing elements which themselves provide the strength required to contain the pressure (i.e., cone joint, lapped joint, etc.) shall satisfy the requirements of this Division.
MANDATORY APPENDIX 1
NOMENCLATURE

\[ A = \text{interference pressure factor (KD-811)} \]
\[ = \text{actual discharge area of relief device (KR-531)} \]
\[ = \text{nozzle or opening reinforcement area (H-120)} \]
\[ A_B = \text{cross-sectional area of a vessel normal to the vessel axis through female threads (E-210)} \]
\[ A_k = \text{total cross-sectional area of the bolts per clamp lug (G-300)} \]
\[ A_C = \text{cross-sectional area of a closure normal to the vessel axis through female threads (E-210)} \]
\[ A_e = \text{cross-sectional area of closure (E-210)} \]
\[ A_{c1} = \text{partial clamp area (G-300)} \]
\[ A_{c2} = \text{partial clamp area (G-300)} \]
\[ A_{c3} = \text{cross-sectional area (KD-502)} \]
\[ A_g = \text{gap area (Figure KD-826)} \]
\[ A_m = \text{required cross-sectional area of the bolts per clamp (G-300)} \]
\[ A_{m1} = \text{cross-sectional area of the bolts, gasket seating (G-300)} \]
\[ A_{m2} = \text{cross-sectional area of the bolts, operating (G-300)} \]
\[ A_o = \text{outside diameter of the hub (G-300)} \]
\[ A_{or} = \text{outside bearing diameter of the hub (G-300)} \]
\[ A_1 = \text{curve fitting constant for the elastic region of the stress–strain curve (KM-620)} \]
\[ A_2 = \text{curve fitting constant for the plastic region of the stress–strain curve (KM-620)} \]
\[ A_3 = \text{lesser of } A_{3a} \text{ and } A_{3b} \text{ (G-300)} \]
\[ A_{3a} = \text{hub longitudinal shear area based on straight shear (G-300)} \]
\[ A_{3b} = \text{hub longitudinal shear area based on 45 deg conical (G-300)} \]
\[ A_5 = \text{minimum clamp cross-sectional area, radial-tangential (G-300)} \]
\[ A_{5b} = \text{maximum clamp bolt hole cutout area (G-300)} \]
\[ A_{5i} = \text{individual clamp bolt hole cutout area (G-300)} \]
\[ A_6 = \text{minimum clamp cross-sectional area, tangential-longitudinal (G-300)} \]
\[ A_{6b} = \text{maximum clamp bolt hole cutout area (G-300)} \]
\[ A_{6i} = \text{individual clamp bolt hole cutout area (G-300)} \]
\[ A_7 = \text{clamp lip longitudinal shear area, lesser of } A_{7a} \text{ and } A_{7b} \text{ (G-300)} \]
\[ A_{7a} = \text{clamp lip longitudinal shear area, straight shear surface (G-300)} \]
\[ A_{7b} = \text{clamp lip longitudinal shear area, 45 deg conical (G-300)} \]
\[ a = \text{crack depth (D-401)} \]
\[ = \text{radius of hot spot or heated area within a plate or the depth of a flaw at a weld toe, as applicable} \]
\[ B = \text{inside diameter of hub (G-300)} \]
\[ B_e = \text{radial distance from connection centerline to center of bolts (G-300)} \]
\[ b = \text{length of gap between layers (KF-826)} \]
\[ C = \text{fatigue crack growth coefficient (Table KD-430)} \]
\[ = \text{relief valve factor for specific heat ratio (KR-531)} \]
\[ = \text{coefficient based on geometry of a blind end (E-120)} \]
\[ = \text{diameter of effective clamp–hub reaction circle (G-300)} \]
\[ C_{depth} = \text{cavity depth (KE-211)} \]
\[ C_g = \text{effective clamp gap (G-300)} \]
\[ C_i = \text{inside bearing diameter of clamp (G-300)} \]
\[ C_{ir} = \text{inside bearing diameter with corner radius (G-300)} \]
\[ C_n = \text{inside diameter of neck of the clamp (G-300)} \]
\[ C_r = \text{thread factor (E-210)} \]
\[ C_t = \text{effective clamp thickness (G-300)} \]
\[ C_{us} = \text{conversion factor, } C_{us} = 1.0 \text{ for units of stress in ksi and } C_{us} = 6.894757 \text{ for units of stress in MPa} \]
\[ C_w = \text{clamp width (G-300)} \]
\[ C_o = \text{tangential bending stress moment arm (G-300)} \]
\[ COD = \text{repair cavity diameter (KE-211)} \]
\[ CP = \text{collapse pressure (KD-1212, KD-1222, KD-1241, KD-1243, KD-1253)} \]
CTOD = crack tip opening displacement [KM-250, D-600(b)(2)]

CVN = Charpy V-notch impact strength (D-600),

\[ F = \text{peak stress (KD-210, 9-200, Figure 9-200-1)} \]

\[ F_b = \text{correction factor for the Bauschinger effect (KD-502, KD-522, KD-523)} \]

\[ F_c = \text{permissible layer gap factor (KD-802)} \]

\[ F_i = \text{nodal force resultant for element location position } i \]

\[ F_L = \text{load on thread } n \] (E-210)

\[ F_T = \text{total load on threads (E-210)} \]

\[ F_1 = \text{magnification factor for calculating stress intensity of internal radial–circumferential cracks (D-403)} \]

\[ F_2 = \text{magnification factor for calculating stress intensity of internal radial–circumferential cracks (D-403)} \]

\[ F_3 = \text{magnification factor for calculating stress intensity of internal radial–circumferential cracks (D-403)} \]

\[ F_4 = \text{magnification factor for calculating stress intensity of internal radial–circumferential cracks (D-403)} \]

\[ F(\delta) = \text{a fatigue modification factor based on the out-of-phase angle between } \Delta \sigma_k \text{ and } \Delta \tau_k \]

\[ f = \text{hub stress correction factor (G-300)} \]

\[ f_f = \text{environmental correction factor to the welded joint fatigue curve} \]

\[ f_i = \text{fatigue improvement method correction factor the welded joint fatigue curve} \]

\[ f_{M,k} = \text{mean stress correction factor for the } k \text{th cycle} \]

\[ f_{MT} = \text{material and temperature correction factor to the welded joint fatigue curve} \]

\[ G = \text{material property (KD-430)} \]

\[ g_2 = \text{height of hub shoulder (G-300)} \]

\[ H = \text{specific stiffness (D-405)} \]

\[ H_{b} = \text{hydrostatic end force on bore area (G-300)} \]

\[ H_{c} = \text{total hydrostatic end force (G-300)} \]

\[ H_{G} = \text{difference between hub preload and required forces (G-300)} \]

\[ H_m = \text{gasket seating requirements (G-300)} \]

\[ H_p = \text{joint-contact surface compression load (G-300)} \]

\[ H_T = \text{difference between total hydrostatic end force and on bore (G-300)} \]
$H_2 = \text{correction factor for bending stress (D-401)}$

$h = \text{distance from flange face to end of skirt [Figure KD-830.3, sketch (c)]}$

$= \text{gap between two layers (KF-826)}$

$h_D = \text{radial distance clamp–hub reaction circle to } H_D \text{ (G-300)}$

$h_G = \text{radial distance clamp–hub reaction circle to } H_G \text{ (G-300)}$

$h_n = \text{hub neck length (G-300)}$

$h_T = \text{radial distance clamp–hub reaction circle to } H_T \text{ (G-300)}$

$h_2 = \text{average thickness of hub shoulder (G-300)}$

$I = \text{material property (KD-430)}$

$= \text{correction factor used in the structural stress evaluation}$

$I_c = \text{moment of inertia of clamp (G-300)}$

$I_h = \text{moment of inertia of hub shoulder (G-300)}$

$I_S = \text{minimum clamp moment of inertia in any radial–tangential plane (G-300)}$

$I_{5b} = \text{maximum reduction clamp moment of inertia bolt holes (G-300)}$

$I_6 = \text{minimum clamp moment of inertia in any tangential–longitudinal plane (G-300)}$

$I_{6b} = \text{maximum reduction clamp moment of inertia bolt holes (G-300)}$

$I_r = \text{correction factor used in the structural shear stress evaluation}$

$I_{ic} = \text{critical stress intensity for plane stress (KM-250, D-600)}$

$K = \text{stress concentration factor (Figure 9-200-1)}$

$= 0.73 \text{times transition radius } r_2 \text{ (H-142)}$

$K_{css} = \text{material parameter for the cyclic stress-strain curve model}$

$K_D = \text{relief device discharge coefficient (KR-523, Figure KR-523.3, KR-531)}$

$K_f = \text{fatigue strength reduction factor used to compute the cyclic stress amplitude or range}$

$K_I = \text{stress intensity factors in a crack (KD-420, KD-440)}$

$K_{I_{max}} = \text{maximum stress intensity factor in a crack (KD-430)}$

$K_{I_{min}} = \text{minimum stress intensity factor in a crack (KD-430)}$

$K_{Ires} = \text{residual stress intensity in a crack (KD-420, KD-430)}$

$K_{IC} = \text{critical stress intensity factor for crack (KM-250)}$

$= \text{fracture toughness (KD-401)}$

$K_{I_{REF}} = \text{crack tip stress intensity factor (D-405)}$

$K_N = \text{wire factor (KD-932)}$

$K_n = \text{greater of 2.6 or } (K_s)^{0.3} \text{ (KD-1262)}$

$K_r = \text{surface roughness factor (KD-322)}$

$K_r = \text{wire factor (KD-932)}$

$K_{SL} = \text{wire factor (KD-932)}$

$K_{SS} = \text{wire factor (KD-932)}$

$K_s = \text{greater of 1.25 or } K_n K_s K_a K_{sa} \text{ (KD-1262)}$

$K_{sa} = \text{factor for size of highly stressed fatigue area (KD-1262)}$

$K_{sc} = \text{factor for fatigue curves at varying temperatures (KD-1262)}$

$K_{sf} = \text{factor for fatigue surface finish (KD-1262)}$

$K_{st} = \text{factor for statistical variation in fatigue tests (KD-1262)}$

$K_{st} = \text{factor for fatigue test temperature (KD-1262)}$

$K_{tb} = \text{threshold stress intensity (KD-430)}$

$K_T = \text{hoop stress concentration factor for cross-bored holes (J-110)}$

$K_{TN} = \text{test life ratio (KD-1262)}$

$K_{TS} = \text{test stress ratio (KD-1262)}$

$= \text{length along nozzle with thickness } t_n \text{ plus transition (H-142)}$

$K_{ut} = \text{factor of upper limit of hydrostatic test pressure (KD-221, KT-312)}$

$K_{uteq} = \text{equivalent factor of upper limit of hydrostatic test pressure for layered construction (KD-231.2)}$

$K_{u_{ef}} = \text{factor of upper limit of hydrostatic test pressure for each individual layer (KD-221, KT-312)}$

$k = C_p/C_v \text{ ratio of specific heats}$

$L = \text{appurtenance live loading (KD-230)}$

$L_A = \text{floating transporter acceleration loads due to spectral motion response determined in KD-237}$

$L_a = \text{distance from clamp bolt centerline to where clamp lug joins body (G-300)}$

$L_d = \text{design fatigue life (KD-330)}$

$L_h = \text{clamp lug height (G-300)}$

$L_T = \text{length of wire pieces in fatigue test (KD-932.3)}$

$L_W = \text{average distance between wire cracks (KD-932.3)}$

$L_w = \text{clamp lug width (G-300)}$

$L_x = \text{measured length of vessel at test pressure}$

$= \text{length of required taper (Figure KD-1121)}$

$= \text{surface length of crack (D-401)}$

$\ell_c = \text{effective clamp lip length (G-300)}$

$= \text{circumferential separation of nozzle centerlines (H-101)}$

$\ell_\ell = \text{longitudinal separation of nozzle centerlines (H-101)}$

$\ell_m = \text{effective clamp lip moment arm (G-300)}$

$M = \text{molecular weight (KR-523.3, KR-531)}$

$= \text{correction factor for membrane stress (D-401)}$

$= \text{mean flexibility of vessel body and end closure (E-210)}$

$M_D = \text{moment due to } H_D \text{ (G-300)}$

$M_F = \text{offset moment (G-300)}$

$M_G = \text{moment due to } H_G \text{ (G-300)}$
\[ M_H = \text{reaction moment at hub neck (G-300)} \]
\[ M_i = \text{nodal moment resultant for element location position } i \]
\[ M_o = \text{total rotational moment on hub (G-300)} \]
\[ M_p = \text{pressure moment (G-300)} \]
\[ M_R = \text{radial clamp equilibrating moment (G-300)} \]
\[ M_T = \text{moment due to } H_T \text{ (G-300)} \]
\[ M_s = \text{clamp longitudinal stress bending moment (G-300)} \]
\[ M_6 = \text{clamp tangential stress bending moment (G-300)} \]
\[ m = \text{crack growth rate exponent (KD-430, Table KD-430)} \]
\[ m_{i} = \text{line moment at element location position } i \]
\[ m_{ss} = \text{exponent used in a fatigue analysis based on structural stress} \]
\[ mN = \text{specified number of repetitions of the event associated with time point } ^{m}t \]
\[ mP_k = \text{component crack face pressure at time point } ^{m}t \text{ for the } k\text{th cycle. The crack face pressure should be specified if the maximum value of the membrane plus bending stress used in the analysis occurs on a surface that is exposed to the fluid pressure. A conservative approach is to always specify the pressure defined in the loading time history.} \]
\[ ^{m}t = \text{time point under consideration with the highest peak or lowest valley} \]
\[ ^{m}n\Delta \sigma_{\text{range}} = \text{von Mises equivalent stress range between time points } ^{m}t \text{ and } ^{n}t \]
\[ ^{m}n\Delta \sigma_{ij} = \text{stress component range between time points } ^{m}t \text{ and } ^{n}t \]
\[ ^{m}n\Delta \sigma_{11} = \text{stress range associated with the normal stress component in the 1-direction between time points } ^{m}t \text{ and } ^{n}t \]
\[ ^{m}n\Delta \sigma_{12} = \text{stress range associated with the shear stress component in the 1-direction between time points } ^{m}t \text{ and } ^{n}t \]
\[ ^{m}n\Delta \sigma_{13} = \text{stress range associated with the shear stress component in the 2-direction between time points } ^{m}t \text{ and } ^{n}t \]
\[ ^{m}n\Delta \sigma_{22} = \text{stress range associated with the normal stress component in the 2-direction between time points } ^{m}t \text{ and } ^{n}t \]
\[ ^{m}n\Delta \sigma_{23} = \text{stress range associated with the shear stress component in the 3-direction between time points } ^{m}t \text{ and } ^{n}t \]
\[ ^{m}n\Delta \sigma_{33} = \text{stress range associated with the normal stress component in the 3-direction between time points } ^{m}t \text{ and } ^{n}t \]
\[ ^{m}\sigma_{e, bk} = \text{elastically calculated bending stress at the point under evaluation for the } k\text{th cycle at the } m \text{ point} \]
\[ ^{m}\sigma_{e, m, k} = \text{elastically calculated membrane stress at the point under evaluation for the } k\text{th cycle at the } m \text{ point} \]
\[ ^{m}\sigma_{ij} = \text{stress tensor at the point under evaluation at time point } ^{m}t \]
\[ ^{m}\epsilon_{b, k} = \text{elastically calculated bending component of shear stress distribution at the point under evaluation for the } k\text{th cycle at the } m \text{ point} \]
\[ ^{m}\epsilon_{m, k} = \text{elastically calculated membrane component of shear stress distribution at the point under evaluation for the } k\text{th cycle at the } m \text{ point} \]
\[ N = \text{number of layers (KD-802)} \]
\[ N_D = \text{design number of alternating stress cycles (KD-932, KD-1262)} \]
\[ N_f = \text{design limit number of alternating stress cycles (KD-320)} \]
\[ N_H = \text{outside diameter of hub neck (G-300)} \]
\[ N_i = \text{allowable number of cycles for vessel service life (KD-330)} \]
\[ N_k = \text{permissible number of cycles for the } k\text{th cycle} \]
\[ N_p = \text{design operating cycles to one-fourth critical crack depth (KD-440)} \]
\[ N_T = \text{number of test cycles (KD-1262)} \]
\[ N_{T_{min}} = \text{minimum number of test cycles (KD-1262)} \]
\[ N_{F_{ij}} = \text{nodal force at node } j \text{, normal to the section, for element location position } i \]
\[ N_{F_j} = \text{nodal force at node } j \text{, normal to the section} \]
\[ N_{M_{ij}} = \text{in-plane nodal moment at node } j \text{, normal to the section, for a shell element} \]
\[ n = \text{number of layer (KD-802)} \]
\[ n_F = \text{number of fully engaged threads (E-210)} \]
\[ n_i = \text{number of applied alternating stress cycles (KD-330)} \]
\[ n_k = \text{actual number of repetitions for the } k\text{th cycle} \]
\[ ^{n}N = \text{specified number of repetitions of the event associated with time point } ^{n}t \]
\[ ^{n}P_k = \text{component crack face pressure at time point } ^{n}t \text{ for the } k\text{th cycle. The crack face pressure should be specified if the maximum value of the membrane plus bending stress used in the analysis occurs on a surface that is exposed to the fluid pressure. A conservative approach is to always specify the crack face pressure. The crack face pressure is based on the actual or operating pressure defined in the loading time history.} \]
\[ n_{css} = \text{material parameter for the cyclic stress-strain curve model} \]
\[ n_i = \text{number of applied alternating stress cycles (KD-330)} \]
\[ n_k = \text{actual number of repetitions for the } k\text{th cycle} \]
\[ ^{n}N = \text{specified number of repetitions of the event associated with time point } ^{n}t \]
\[ ^{n}P_k = \text{component crack face pressure at time point } ^{n}t \text{ for the } k\text{th cycle. The crack face pressure should be specified if the maximum value of the membrane plus bending stress used in the analysis occurs on a surface that is exposed to the fluid pressure. A conservative approach is to always specify the crack face pressure. The crack face pressure is based on the actual or operating pressure defined in the loading time history.} \]
\[ 261 \]
\[ n \tau_{b,k} = \text{elastically calculated bending component of shear stress distribution at the point under evaluation for the } k \text{th cycle at the } n \text{ point} \]

\[ n \tau_{m,k} = \text{elastically calculated membrane component of shear stress distribution at the point under evaluation for the } k \text{th cycle at the } n \text{ point} \]

\[ n \sigma_{ij} = \text{stress tensor at the point under evaluation at time point } n \]

\[ n t = \text{time point under consideration that forms a range with time point } n \]

\[ n \sigma_{m,k} = \text{elastically calculated membrane stress at the point under evaluation for the } k \text{th cycle at the } n \text{ point} \]

\[ n \sigma_{b,k} = \text{elastically calculated bending stress at the point under evaluation for the } k \text{th cycle at the } n \text{ point} \]

\[ n \sigma_{b,k} = \text{stress tensor at the point under evaluation at time point } n \]

\[ n P = \text{design pressure (KD-802, G-300)} \]

\[ n P_a = \text{maximum autofrettage pressure (KD-502)} \]

\[ n P_b = \text{back pressure on relief device (KD-523)} \]

\[ n P_t = \text{cyclic test loading (KD-1262)} \]

\[ n P_t = \text{hydrostatic test pressure (KD-231, KD-824, KT-312)} \]

\[ n Q = \text{secondary membrane plus bending stress (9-200, Figure 9-200-1)} \]

\[ n Q = \text{reaction shear force at hub neck (G-300)} \]

\[ n Q_c = \text{percent of theoretical circumferential growth measured on outside (KD-802)} \]

\[ n r_{ij} = \text{radial coordinate of node } j \text{ for an axisymmetric element} \]

\[ n r_{2} = \text{transition radius inside nozzle to vessel wall (H-142)} \]

\[ n S_{eq} = \text{equivalent calculated alternating stress intensity (KD-312.4)} \]

\[ n S_a = \text{allowable fatigue strength (KD-120)} \]

\[ n S_a = \text{allowable bolt stress, room temperature (G-300)} \]

\[ n S_a = \text{stress intensity at design load (KD-1262)} \]

\[ n S_a = \text{allowable amplitude of alternating stress (KD-312.4)} \]

\[ n S_{alt} = \text{alternative stress intensity (KD-302)} \]

\[ n S_{alt} = \text{allowable bolt stress, design temperature (G-300)} \]

\[ n S_{act} = \text{actual yield stress (KD-1254)} \]

\[ n S_{act} = \text{actual yield strength per material specification (KD-1254)} \]

\[ n S_{act} = \text{yield stress for clamp material, room temperature (G-300)} \]

\[ n S_{act} = \text{yield stress for clamp material, design temperature (G-300)} \]

\[ n S_{act} = \text{yield stress for hub material, room temperature (G-300)} \]

\[ n S_{act} = \text{yield stress for hub material, design temperature (G-300)} \]

\[ n S_{act} = \text{hub longitudinal stress on outside hub neck (G-300)} \]
\( S_2 = \) maximum Lamé hoop stress at hub bore (G-300)
\( S_3 = \) hub shear stress at shoulder (G-300)
\( S_4 = \) hub radial stress in hub neck (G-300)
\( S_5 = \) clamp longitudinal stress at clamp body inner diameter (G-300)
\( S_6 = \) clamp tangential stress at clamp body outer diameter (G-300)
\( S_7 = \) maximum shear stress in clamp lips (G-300)
\( S_8 = \) clamp lip bending stress (G-300)
\( S_9 = \) clamp lug bending stress (G-300)
\( S_{10} = \) maximum clamp lug shear stress (G-300)
\( S_{11} = \) effective bearing stress between clamp and hub (G-300)

\( s_j = \) local coordinate, parallel to the stress classification line, that defines the location of nodal force \( NF_j \) relative to the mid-thickness of the section

\( T = \) absolute temperature
\( T = \) thickness (KM-201, KM-210, KM-211.1)
\( T = \) maximum heat-treated thickness (KM-211)
\( T = \) self-restraining load case (KD-230)
\( T = \) thickness of hub shoulder (G-300)
\( T_c = \) critical temperature (KD-1262)
\( T = \) clamp lip thickness below outside edge hub (G-300)
\( T_h = \) hub shoulder thickness below inside edge hub (G-300)
\( T_t = \) test temperature (KD-1262)
\( t = \) distance between highly stressed surface and the nearest quenched surface (KM-211.2)
\( t = \) wall thickness, nominal vessel thickness (9-220, KD-802, H-142)
\( t = \) minimum wall thickness in the region under consideration, or the thickness of the vessel, as applicable

\( t_b = \) thickness of blind end (E-110)
\( t_{ess} = \) structural stress effective thickness
\( t_H = \) thickness of head at joint (Figure KD-830.2)
\( t_j = \) thickness of each layer (KD-221)
\( t_L = \) thickness of layer at joint (Figure KD-830.2)
\( t_n = \) thickness of layer \( n \) (KD-802)
\( t_n = \) nominal thickness of nozzle wall less corrosion allowance (Figure KD-830.6)
\( t_n = \) thickness of nozzle wall (Figure KD-1130)
\( t_p = \) thickness of attached pipe wall (H-142)
\( t_r = \) minimum wall thickness without opening (H-120)
\( t_{rn} = \) required thickness of seamless nozzle wall (Figure KD-1122)
\( t_S = \) shell thickness (Figure KD-830.2)
\( t_w = \) thickness of vessel wall (E-110)

\( U = \) cumulative usage factor (KD-330)
\( V_{REF} = \) longitudinal crack displacement (D-405)
\( W = \) wind load (KD-230)
\( W = \) mass flow of any gas or vapor (KR-523.3, KR-531)
\( W = \) total design bolt load (G-300)
\( W_A = \) assembly loads (e.g., shrink fit, wire winding, sealing preload) (KD-230)
\( W_o = \) rated air flow for relief device (KR-531)
\( W_{e} = \) total effective clamping preload on one lip (G-300)
\( W_{m1} = \) minimum operating bolt load (G-300)
\( W_{m2} = \) minimum gasket seating bolt load (G-300)
\( W_{pt} = \) pressure test wind load case (KD-230)
\( W_T = \) theoretical mass flow (KR-523.3)
\( w = \) width of the element to determine structural stresses from Finite Element Analysis
\( X = \) absolute value of the range (load or stress) under consideration using the Rainflow Cycle Counting Method
\( X_b = \) basic clamp dimension to neutral axis (G-300)
\( X_g = \) global \( X \) axis
\( X_i = \) average radial distance from bolt cutout area (G-300)
\( X_L = \) local \( X \) axis, oriented parallel to the stress classification line
\( X_S = \) modified clamp dimension to neutral axis (G-300)
\( X_J = \) modified clamp dimension to neutral axis (G-300)
\( x = \) diameter at any point (KD-911)
\( x = \) through-wall thickness coordinate
\( x_1 = \) any diameter of cylinder (KD-911)
\( x_2 = \) any diameter of winding (KD-911)
\( Y = \) wall ratio or \( D_o/D_i \) of a shell (KD-220, KD-502)
\( Y = \) weld offset (Figure KD-830.2)
\( Y = \) absolute value of the adjacent range (load or stress) to previous \( X \) using the Rainflow Cycle Counting Method
\( Y_g = \) global \( Y \) axis
\( Y_i = \) ratio of outside diameter to inside diameter of inner layer (KD-802)
\( Y_j = \) ratio of outside diameter to inside diameter of each layer (KD-220, KT-312)
\( Y_L = \) local \( Y \) axis, oriented normal to the stress classification line
\( Y_o = \) ratio of outside diameter to inside diameter of outer layer (KD-802)
\( y = \) radial offset in buttwelding of unequal section thicknesses (Figure KD-1121)
\( Z = \) \( D_o/D \), where \( D \) can be any point in the wall (KD-220)
\( y = \) compressibility factor (KR-531)
\( y = \) clamp-hub taper angle (G-300)
\( \alpha = \) shape factor \((KD-210, \text{Figure 9-200-1, 9-100})\)

= angle, maximum angle \((Figure \text{KE-321})\)

= maximum rake angle \((E-110)\)

\(\alpha_r = \) thermal expansion of reinforcing metal \((H-150)\)

\(\alpha_v = \) thermal expansion of vessel wall \((H-150)\)

\(\beta = \) factor in equivalent alternating stress intensity \((KD-312)\)

= factor = 0.2 \((KD-932)\)

\(\gamma_1 = \) true strain in the micro-strain region of the stress–strain curve \((KM-620)\)

\(\gamma_2 = \) true strain in the macro-strain region of the stress–strain curve \((KM-620)\)

\(\Delta = \) difference, increment

\(\Delta K = \) range of stress intensity factor \((KD-430)\)

\(\Delta S_{ess,k} = \) equivalent structural stress range parameter for the \(k\)th cycle

= computed equivalent structural stress range parameter from Part 5

\(\overline{\Delta \epsilon} = \) average relative standard deviation of fatigue strength \((KD-932.3)\)

\(\Delta T = \) operating temperature range \((H-150)\)

\(\Delta \epsilon_k = \) local nonlinear structural strain range at the point under evaluation for the \(k\)th cycle

\(\Delta \epsilon^e_k = \) elastically calculated structural strain range at the point under evaluation for the \(k\)th cycle

\(\Delta \epsilon_{p,ij} = \) the range of plastic strain component, \(ij\), at the point under evaluation for the cycle under evaluation. Note that the shear strains are the engineering strain values that are typically output from a finite element analysis (i.e., not tensor strains) \((KD-323)\)

\(\Delta \epsilon_{p,ij,k} = \) the range of plastic strain component, \(ij\), at the point under evaluation for the \(k\)th loading condition or cycle. Note that the shear strains are the engineering strain values that are typically output from a finite element analysis (i.e., not tensor strains) \((KD-323)\)

\(\Delta \sigma_k = \) local nonlinear structural stress range at the point under evaluation for the \(k\)th cycle

\(\Delta \sigma_{x} = \) structural stress range

\(\Delta \sigma^e_{b,k} = \) elastically calculated structural bending stress range at the point under evaluation for the \(k\)th cycle

\(\Delta \sigma^e_k = \) elastically calculated structural stress range at the point under evaluation for the \(k\)th cycle

\(\Delta \sigma^e_{m,k} = \) elastically calculated structural membrane stress range at the point under evaluation for the \(k\)th cycle

\(\Delta \tau_k = \) structural shear stress range at the point under evaluation for the \(k\)th cycle

\(\Delta \tau^e_{b,k} = \) elastically calculated bending component of the structural shear stress range at the point under evaluation for the \(k\)th cycle

\(\Delta \tau^e_{m,k} = \) elastically calculated membrane component of the structural shear stress range at the point under evaluation for the \(k\)th cycle

\(\delta = \) any difference, diametral interference \((KD-802)\)

= out-of-phase angle between \(\Delta \sigma_k\) and \(\Delta \tau_k\) for the \(k\)th cycle

\(\epsilon_m = \) average tangential strain, autofrettaged outside diameter \((KD-502)\)

\(\epsilon_p = \) average tangential strain, autofrettaged bore \((KD-502)\)

= stress–strain curve fitting parameter \((KM-620)\)

\(\epsilon_{ta} = \) true total strain amplitude

\(\epsilon_{tr} = \) true total strain range

\(\epsilon_{ts} = \) true total strain \((KM-620)\)

\(\epsilon_{ys} = 0.2\%\) engineering offset strain \((KM-620)\)

\(\epsilon_x = \) true plastic strain in the micro-strain region of the stress–strain curve \((KM-620)\)

\(\epsilon_z = \) true plastic strain in the macro-strain region of the stress–strain curve \((KM-620)\)

\(\mu = \) viscosity

= friction angle \((G-300)\)

\(\nu = \) Poisson’s ratio \((KD-802)\); also sometimes viscosity

\(\nu_i = \) inner layer Poisson’s ratio \((KD-802)\)

\(\nu_o = \) outer layer Poisson’s ratio \((KD-802)\)

\(\pi = \) constant = 3.14159

\(\rho = \) mass density \((KR-523)\)

\(\Sigma = \) summation

\(\sigma = \) normal or principal stresses, with various subscripts

\(\sigma_{AD} = \) value of \(\sigma_{iR,AD}\) at \(D = D_i\) \((KD-502)\)

\(\sigma_a = \) total stress amplitude

\(\sigma_b = \) bending stress

\(\sigma_{bi} = \) bending stress for element location position \(i\)

\(\sigma_{CD} = \) residual tangential stress at bore, including the Bauschinger effect \((KD-502)\)

\(\sigma_e = \) von Mises stress

\(\sigma_{ij} = \) stress tensor at the point under evaluation

\(\sigma_{ij,b} = \) bending stress tensor at the point under evaluation

\(\sigma_{ij,F} = \) peak stress component

\(\sigma_{ij,in} = \) stress tensor on the inside surface of the shell

\(\sigma_{ij,m} = \) membrane stress tensor at the point under evaluation

\(\sigma_{ij, out} = \) stress tensor on the outside surface of the shell

\(\sigma_{m} = \) membrane stress

\(\sigma_{mi} = \) membrane stress for element location position \(i\)
\[ \sigma_{\text{max},k} = \text{maximum stress in the } k\text{-th cycle} \]
\[ \sigma_{\text{mean},k} = \text{mean stress in the } k\text{-th cycle} \]
\[ \sigma_{\text{min},k} = \text{minimum stress in the } k\text{-th cycle} \]
\[ \sigma_{\text{am}} = \text{associated mean stress (KD-302.2)} \]
\[ \sigma_R = \text{radial stress component at radius } r \text{ (KD-802)} \]
\[ \sigma_r = \text{total stress range} \]
\[ \sigma_{RR} = \text{radial stress at bore, including the Bauschinger effect (KD-522)} \]
\[ \sigma_{rr} = \text{residual radial stress (KD-802)} \]
\[ \sigma_{RRA} = \text{first approximation of residual radial stress after autofrettage (KD-522)} \]
\[ \sigma_s = \text{structural stress} \]
\[ \sigma_t = \text{tangential stress component at radius } r \text{ (KD-802)} \]
\[ \sigma_{TR} = \text{true stress at which the true strain will be evaluated (KM-620)} \]
\[ \sigma_{tR} = \text{residual tangential stress, including the Bauschinger effect (KD-522)} \]
\[ \sigma_{tRA} = \text{first approximation of residual tangential stress after autofrettage (KD-522)} \]
\[ \sigma_{tR} = \text{residual tangential stress (KD-802)} \]
\[ \sigma_{uts} = \text{engineering ultimate tensile stress evaluated at the temperature of interest (KM-620)} \]
\[ \sigma_{uts, t} = \text{true ultimate tensile stress evaluated at the true ultimate tensile strain} \]
\[ \sigma_1 = \text{principal stress in the 1-direction} \]
\[ \sigma_2 = \text{principal stress in the 2-direction} \]
\[ \sigma_3 = \text{principal stress in the 3-direction} \]
\[ \tau = \text{shear stresses, with various subscripts} \]
\[ \Phi = \text{flaw shape parameter (D-401)} \]
\[ \phi = \text{clamp shoulder angle (G-300)} \]
NONMANDATORY APPENDIX E
CONSTRUCTION DETAILS

E-100 INTEGRAL HEADS (BLIND ENDS)

The thickness and proportions of blind ends of cylindrical vessels may conform to the recommendations given in this Appendix.

E-110 THICK WALL PROPORTIONS

(a) For initial sizing, if the $D_0/D_1$ ratio of the vessel is 1.25 or more, the proportions of blind pressure vessel ends shall be kept within the following limits (see Figure E-110).

(1) The minimum inside corner radius $R_c$ shall be 25% of the design thickness of the vessel wall.

(2) The thickness of the blind end at the tangent of the inside corner ($t_b$ in Figure E-110) shall be no less than the design thickness of the vessel wall and no greater than twice the design thickness, $t_w$, of the vessel wall. For $Y = 1.5$ to $2.25$ and $R_c = 0.25t_w$, the minimum blind end thickness shall be calculated by the following equation:

$$t_b = t_w - 1.0677Y^3 + 6.80Y^2 - 15.433Y + 13.45$$

(E.1)

For $Y > 2.25$ and $R_c \geq 0.25t_w$, the minimum thickness, $t_b$, shall be calculated by the following equation:

$$t_b = t_w$$

(E.2)

(3) The maximum angle $\alpha$ from the tangent of the inside corner to the vessel centerline shall be 10 deg from the plane perpendicular to the vessel axes (see Figure E-110).

(4) The diameter $D_{op}$ of any opening in the blind end shall not exceed 15% of the vessel inside diameter and shall be located on the vessel centerline.

(b) The corner radius principal stresses used to perform the fatigue evaluation in accordance with Article KD-3 shall be calculated by the following equations for $Y = 1.25$ to 4.0 and $R_c = 0.25t_w$:

$$\sigma_1 = P \left( 0.6320 - 1.5160Y + 4.5731 \frac{Y}{\ln Y} - 19.1428 \frac{1}{\sqrt{Y}} + 31.0567 \frac{\ln Y}{Y^2} \right)$$

(E.3)

$$\sigma_2 = P \left( -0.5718 + 0.1141Y + 1.0208 \frac{1}{\ln Y} + 1.6096 \frac{\ln Y}{Y^2} - 1.3667 \frac{1}{Y^2} \right)$$

(E.4)

$$\sigma_3 = -P$$

(E.5)
E-120 THIN WALL PROPORTIONS

(a) If the $D_0/D_1$ ratio of the vessel is less than 1.25, the minimum thickness of blind ends (as shown in Figure E-120) shall be calculated by the following equation without detailed stress analysis:

$$ t_b = D_0 \left( \frac{1.5CP}{S_y} \right)^{0.5} \tag{E.6} $$

(b) The minimum multiplier to be used with the $C$ values below when performing fatigue and fracture mechanics evaluations using Articles KD-3 and KD-4, respectively, is 1.8. The stress intensity is:

$$ S = 1.8C \left( \frac{D_0}{t_b} \right)^2 P \tag{E.7} $$

(c) The value of $C$ to be used in eqs. (a)(E.6) and (b)(E.7) shall be based on the following:

1. $C = 0.22$ if the inside corner radius is at least three times the minimum required end thickness.

2. $C = 0.44$ if the inside corner radius is less than three times the minimum required end thickness.

3. The inside corner radius shall be greater than or equal to 25% of the design thickness of the vessel wall $t_w$.

(d) There is no special requirement with regard to cylindrical lengths; however, the taper between such thicknesses shall be a minimum of 3:1.

E-200 THREADED END CLOSURES

Specific requirements for threads and threaded closures are given in Article KD-6. In the following, one thread is understood to mean one 360 deg turn of a single start thread with a full cross section. The number of threads should be less than 20 but at least 4. The helix angle of the thread should not exceed 2 deg. The internal thread should have a generous undercut. The axial length of the external threaded portion should be at least one thread pitch longer than the mating internal threaded portion to ensure full engagement of all internal threads.

The threads do not carry the axial load uniformly. The internal thread closest to the undercut carries generally the largest portion of the load. The following method may be used to determine the load distribution. The threads are numbered from the undercut.

For vessels where the outside diameter is not uniform along the whole length of the body, the methods given in this Appendix for calculating thread load distribution may be nonconservative due to the effects of the vessel outer diameter on the flexibility factors calculated in E-210 for such vessels. See KD-100(b).

E-210 NOMENCLATURE (SEE FIGURES E-210.1, E-210.2, AND E-210.3)

- $A_B$ = cross-sectional area of the vessel normal to the vessel axis through the internal threads
- $A_C$ = cross-sectional area of the closure normal to the vessel axis through the external threads
- $A_T$ = flexibility factor of the threads
- $C_M$ = combined flexibility factor of the body and closure
- $D_0$ = outside diameter of the vessel
- $D_p$ = pitch diameter of the threads
- $F_1$ = load on the first thread
- $F_2$ = load on the second thread

$$ A_B = \pi \left( \frac{D_0^2}{4} - \frac{D_p^2}{4} \right) $$

$$ A_C = \pi \frac{D_p^2}{4} $$

$$ A_T = \left( \frac{1}{D_p} + \frac{1}{D_C} \right) P $$

$$ C_M = \left( \frac{1}{E_B} + \frac{1}{E_C} \right) $$

$$ E_B = \text{modulus of elasticity of the body} $$

$$ E_C = \text{modulus of elasticity of the closure} $$
E-220 THREAD LOAD DISTRIBUTION

E-221 CONTINUOUS THREADS

The thread load distribution may be obtained by:

\[ F_i = F_i + 1 + \frac{C_M}{C_T} \sum_{j=1}^{n} F_j \]  \hspace{1cm} (E.8)

where

\[ F_{\text{sum}} = \sum_{j=1}^{n} F_j \]  \hspace{1cm} (E.9)

\[ F_n = \text{load on the last thread} \]

\[ F_T = \text{total load on all threads} \]

\[ P_T = \text{thread pitch} \]

\[ S = \text{number of loaded segments in one pitch} \]

\[ S_i = \text{segment load} \]

\[ S_M = \text{combined flexibility factor of the body and the closure} \]

\[ = \frac{C_M}{S} \]

\[ S_T = \text{flexibility factor of the threads} \]

\[ = C_T S \]

\[ n = \text{total number of threads} \]

E-222 INTERRUPTED (BREECH) THREAD

A similar equation for segments to those for full threads is:

\[ S_i = S_i + 1 + \frac{S_M}{S_T} \left( S_{\text{sum}} \right) \]  \hspace{1cm} (E.10)

(a) Example for Continuous Thread. Assuming the load on the last thread is unity, given the values below, we obtain the values in Table E-222.1.

\[ A_B = 398.197 \]
\[ A_C = 132.732 \]
\[ C_M = 0.01 \]
\[ C_T = 0.154 \]
\[ D_o = 26 \]
\[ D_p = 13 \]
\[ n = 10 \]
\[ P_T = 1 \]

(b) Example for Interrupted Thread. For \( S = 4 \) (one-eighth turn to open), we obtain the values in Table E-222.2.

where the mating parts are the same material
Figure E-210.1
Typical Threaded End Closure
<table>
<thead>
<tr>
<th>Thread</th>
<th>( F_i ) [Note (1)]</th>
<th>( F_{\text{sum}} )</th>
<th>( C_M/C_T \times F_{\text{sum}} )</th>
<th>( F_i ), %</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>1.000</td>
<td>1.000</td>
<td>0.065</td>
<td>4.1</td>
</tr>
<tr>
<td>9</td>
<td>1.065</td>
<td>2.065</td>
<td>0.135</td>
<td>4.3</td>
</tr>
<tr>
<td>8</td>
<td>1.200</td>
<td>3.265</td>
<td>0.213</td>
<td>4.9</td>
</tr>
<tr>
<td>7</td>
<td>1.413</td>
<td>4.679</td>
<td>0.305</td>
<td>5.7</td>
</tr>
<tr>
<td>6</td>
<td>1.719</td>
<td>6.398</td>
<td>0.418</td>
<td>7.0</td>
</tr>
<tr>
<td>5</td>
<td>2.137</td>
<td>8.534</td>
<td>0.557</td>
<td>8.7</td>
</tr>
<tr>
<td>4</td>
<td>2.694</td>
<td>11.228</td>
<td>0.733</td>
<td>10.9</td>
</tr>
<tr>
<td>3</td>
<td>3.427</td>
<td>14.655</td>
<td>0.957</td>
<td>13.9</td>
</tr>
<tr>
<td>2</td>
<td>4.384</td>
<td>19.039</td>
<td>1.243</td>
<td>17.8</td>
</tr>
<tr>
<td>1</td>
<td>5.627</td>
<td>24.666</td>
<td>1.611</td>
<td>22.8</td>
</tr>
</tbody>
</table>

**NOTE:**
(1) \( F_T = 24.666 \) (obtained by adding the ten \( F_i \) values).
## Table E-222.2
### Interrupted Thread Example

<table>
<thead>
<tr>
<th>Thread No.</th>
<th>Segment No.</th>
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