TD-500  DESIGN OF UNSTAYED FLAT HEADS AND COVERS

(a) The minimum thickness of unstayed flat heads, cover plates, and blind flanges shall conform to the requirements given in this paragraph. These requirements apply to both circular and noncircular heads and covers. Special consideration shall be given to the design of shells, nozzle necks, or flanges to which noncircular heads or covers are attached. Some acceptable types of flat heads and covers are shown in Figure TD-500. In this figure, the dimensions of the component parts and the dimensions of the welds are exclusive of extra metal required for corrosion allowance.

(b) The symbols used in this paragraph and in Figure TD-500 are defined as follows:

\[ C = \ \text{a factor depending upon the method of attachment of head, shell dimensions, and other items as listed in (d), dimensionless. The factors for welded covers also include a factor of 0.667 that effectively increases the allowable stress for such constructions to 1.5S} \]

\[ D = \ \text{long span of noncircular heads or covers measured perpendicular to short span} \]

\[ d = \ \text{diameter, or short span, measured as indicated in Figure TD-500} \]

\[ E = \ \text{joint efficiency, from Table TW-130.4, of any Category A weld as defined in TW-130.3} \]

\[ h_G = \ \text{gasket moment arm, equal to the radial distance from the centerline of the bolts to the line of the gasket reaction, as shown in Section VIII, Division 1, Mandatory Appendix 2, Table 2-5.2} \]

\[ L = \ \text{perimeter of noncircular bolted head measured along the centers of the bolt holes} \]

\[ M = \ \text{the ratio} \ t_r / t_w, \ \text{dimensionless} \]

\[ P = \ \text{internal design pressure (see TD-150)} \]

\[ r = \ \text{inside comer radius on a head formed by flanging or forging} \]

\[ S = \ \text{maximum allowable stress value in tension, psi, from applicable table of stress values referenced by TD-210} \]

\[ t = \ \text{minimum required design thickness of flat head or cover} \]

\[ t_{th} = \ \text{nominal thickness of flat head or cover} \]

\[ t_r = \ \text{required thickness of seamless shell, for pressure} \]

\[ t_s = \ \text{nominal thickness of shell} \]

\[ t_w = \ \text{thickness through the weld joining the edge of a head to the inside of a vessel, as indicated in Figure TD-500, sketch (g)} \]

\[ W = \ \text{total bolt load, given for circular heads for Section VIII, Division 1, Mandatory Appendix 2, 2-5, eqs. (3) and (4).} \]

\[ Y = \ \text{length of flange of flanged heads, measured from the tangent line of knuckle, as indicated in Figure TD-500, sketches (a) and (c)} \]

\[ Z = \ \text{a factor of noncircular heads and covers that depends on the ratio of short span to long span, as given in (c), dimensionless} \]

(c) The thickness of flat unstayed heads, covers, and blind flanges shall conform to one of the following three requirements. These equations provide adequate design margins against structural failure. However, no limit has been provided for deflection and rotation. If leakage at a threaded or gasketed joint is of concern, the thickness may have to be increased to provide adequate rotational stiffness.

\[ (1) \ \text{Circular blind flanges conforming to any of the flange standards listed in Table TG-130 and further limited in TD-100.5 shall be acceptable for the diameters and pressure–temperature ratings in the respective standard when the blind flange is of the types shown in Figure TD-500, sketches (j) and (k).} \]

\[ (2) \ \text{The minimum required thickness of flat unstayed circular heads, covers, and blind flanges shall be calculated by the following equation:} \]

\[ t = d \sqrt{CP / SE} \quad (1) \]

except when the head, cover, or blind flange is attached by bolts causing an edge moment [sketches (j) and (k)], in which case the thickness shall be calculated by

\[ t = \frac{CP}{SE} + \frac{1.9Wh_G}{SEd^3} \quad (2) \]

When using eq. (2), the thickness \( t \) shall be calculated for both operating conditions and gasket seating, and the greater of the two values shall be used. For operating conditions, the value of \( P \) shall be the design pressure, and the values of \( S \) at the design temperature and \( W \) from Section VIII, Division 1, Mandatory Appendix 2, 2-5, eq. (3) shall be used. For gasket seating, \( P \) equals...
zero, and the values of $S$ at atmospheric temperature and $W$ from Section VIII, Division 1, Mandatory Appendix 2-5, eq. (4) shall be used.

(3) Flat unstayed heads, covers, or blind flanges may be square, rectangular, elliptical, obround, segmental, or otherwise noncircular. Their required thickness shall be calculated by the following equation:

$$ t = d \sqrt{ZCP / SE} \quad (3) $$

where

$$ Z = 3.4 - \frac{2Ad}{D} \quad (4) $$

with the limitation that $Z$ need not be greater than 2.5.

Equation (3) does not apply to noncircular heads, covers, or blind flanges attached by bolts causing a bolt edge moment [sketches (j) and (k)]. For noncircular heads of this type, the required thickness shall be calculated by the following equation:

$$ t = d \sqrt{ZCP / SE} + \frac{6WbC}{SELd^2} \quad (5) $$

When using eq. (5), the thickness $t$ shall be calculated in the same way as specified above for eq. (2)(2).

(d) For the types of construction shown in Figure TD-500, the minimum values of $C$ to be used in eqs. (c)(2)(1), (c)(2)(2), (c)(3)(3), and (c)(3)(5) are:

(1) Sketch (a)

(-a) $C = 0.17$ for flanged circular and noncircular heads forged integral with or butt welded to the vessel with an inside corner radius not less than three times the required head thickness, with no special requirement with regard to length of flange, and where the welding meets all the requirements for circumferential joints given in Part TW.

(-b) $C = 0.10$ for circular heads, when the flange length for heads of the above design is not less than

$$ Y = \left( 1.1 - 0.8 \frac{L_s^2}{t_h^2} \right) \sqrt{dt_h} \quad (6) $$

(-c) $C = 0.10$ for circular heads, when the flange length $Y$ is less than the requirements in eq. (-b)(6) but the shell thickness is not less than

$$ t_s = 1.124t_h\sqrt{1.1 - Y/\sqrt{dt_h}} \quad (7) $$

for a length of at least $2\sqrt{dt_s}$.

When $C = 0.10$ is used, the taper shall be at least 1:3.

(2) Sketch (b-1). $C = 0.17$ for forged circular and noncircular heads integral with or butt welded to the vessel, where the flange thickness is not less than two times the shell thickness, the corner radius on the inside is not less than three times the flange thickness, and the welding meets all the requirements for circumferential joints given in Part TW.

(3) Sketch (b-2). $C = 0.33m$ but not less than 0.20 for forged circular and noncircular heads integral with or butt welded to the vessel, where the flange thickness is not less than the shell thickness, the corner radius on the inside is not less than the following:

(-a) $r_{\min} = 10$ mm (0.375 in.) for $t_s \leq 38$ mm ($1^{1/2}$ in.)

(-b) $r_{\min} = 0.25t$, for $t_s > 38$ mm ($1^{1/2}$ in.) but need not be greater than 19 mm ($3/4$ in.)

The welding shall meet all the requirements for circumferential joints given in Part TW.

(4) Sketch (c)

(-a) $C = 0.13$ for circular heads lap welded to the shell with corner radius not less than $3\pi$ and $Y$ not less than required by eq. (1)(-b)(6) and the requirements of TW-130.5 are met.

(-b) $C = 0.20$ for circular and noncircular lap-welded or brazed construction as above, but with no special requirement with regard to $Y$.

(-c) $C = 0.30$ for circular flanged plates screwed over the end of the vessel, with inside corner radius not less than $3\pi$, in which the design of the threaded joint against failure by shear, tension, or compression, resulting from the end force due to pressure, is based on a factor of safety of at least 3.5, and the threaded parts are at least as strong as the threads for standard piping of the same diameter. Seal welding may be used, if desired.

(5) Sketch (d). $C = 0.13$ for integral flat circular heads when the dimension $d$ does not exceed 610 mm (24 in.), the ratio of thickness of the head to the dimension $d$ is not less than 0.05 or greater than 0.25, the head thickness $t_h$ is not less than the shell thickness $t_s$, the inside corner radius is not less than 0.25$t$, and the construction is obtained by special techniques of upsetting and spinning the end of the shell, such as employed in closing header ends.

(6) Sketches (e), (f), and (g)

(-a) $C = 0.33m$ but not less than 0.20 for circular plates, welded to the inside of a vessel, and otherwise meeting the requirements for the respective types of welded vessels. If a value of $m < 1$ is used in calculating $t$, the shell thickness $t_s$ shall be maintained along a distance inwardly from the inside face of the head equal to at least $2\sqrt{dt_s}$. The throat thickness of the filling welds in sketches (e) and (f) shall be at least $0.7t_s$. The size of the weld $t_w$ in sketch (g) shall be not less than two times the required thickness of a seamless shell nor less than 1.25 times the nominal shell thickness but need not be greater than the head thickness; the weld shall be deposited in a welding groove with the root of the weld at the inner face of the head as shown in the sketch.
MANDATORY APPENDIX 2
RULES FOR BOLTED FLANGE CONNECTIONS WITH RING GASKETS

2-1 SCOPE

(a) The rules in Mandatory Appendix 2 apply specifically to the design of bolted flange connections with gaskets that are entirely within the circle enclosed by the bolt holes and with no contact outside this circle, and are to be used in conjunction with the applicable requirements in Subsections A, B, and C of this Division. The hub thickness of weld neck flanges designed to this Appendix shall also comply with the minimum thickness requirements in Parts TD, TF and TW of this Section. These rules are not to be used for the determination of the thickness of tubeshells integral with a bolting flange as illustrated in Figure UW-13.2, sketches (h) through (l) or Figure UW-13.3, sketch (c). Nonmandatory Appendix S provides discussion on Design Considerations for Bolted Flanged Connections.

These rules provide only for hydrostatic end loads and gasket seating. The flange design methods outlined in 2-4 through 2-8 are applicable to circular flanges under internal pressure. Modifications of these methods are outlined in 2-9 and 2-10 for the design of split and noncircular flanges. See 2-11 for flanges with ring type gaskets subject to external pressure, 2-12 for flanges with nut-stops, and 2-13 for reverse flanges. Rules for calculating rigidity factors for flanges are provided in 2-14. Recommendations for qualification of assembly procedures and assemblers are in 2-15. Proper allowance shall be made if connections are subject to external loads other than external pressure.

(b) The design of a flange involves the selection of the gasket (material, type, and dimensions), flange facing, bolting, hub proportions, flange width, and flange thickness. See Note in 2-5(c)(1). Flange dimensions shall be such that the stresses in the flange, calculated in accordance with 2-7, do not exceed the allowable flange stresses specified in 2-8. Except as provided for in 2-14(a), flanges designed to the rules of this Appendix shall also meet the rigidity requirements of 2-14. All calculations shall be made on dimensions in the corroded condition.

(c) It is recommended that bolted flange connections conforming to the standards listed in [TD-100.5] be used for connections to external piping. These standards may be used for other bolted flange connections and dished covers within the limits of size in the standards and the pressure–temperature ratings permitted in [TD-100.5]. The ratings in these standards are based on the hub dimensions given or on the minimum specified thickness of flanged fittings of integral construction. Flanges fabricated from rings may be used in place of the hub flanges in these standards provided that their strength, calculated by the rules in this Appendix, is not less than that calculated for the corresponding size of hub flange.

(d) Except as otherwise provided in (c) above, bolted flange connections for unfired pressure vessels shall satisfy the requirements in this Appendix.

(e) The rules of this Appendix should not be construed to prohibit the use of other types of flanged connections, provided they are designed in accordance with good engineering practice and method of design is acceptable to the Inspector. Some examples of flanged connections which might fall in this category are as follows:

1. flanged covers as shown in Figure 1-6;
2. bolted flanges using full-face gaskets;
3. flanges using means other than bolting to restrain the flange assembly against pressure and other applied loads.

2-2 MATERIALS

(a) Materials used in the construction of bolted flange connections shall comply with the requirements given in [TM-110 and TM-120].

(b) Flanges made from ferritic steel and designed in accordance with this Appendix shall be full-annealed, normalized, normalized and tempered, or quenched and tempered when the thickness of the flange, t (see Figure 2-4), exceeds [75 mm (3 in.)].

(c) Material on which welding is to be performed shall be proved of good weldable quality. Satisfactory qualification of the welding procedure under Section IX is considered as proof. Welding shall not be performed on steel that has a carbon content greater than 0.35%. All welding on flange connections shall comply with the requirements for postweld heat treatment given in this Section.
(d) Flanges with hubs that are machined from plate, bar stock, or billet shall not be machined from plate or bar material (except as permitted in UC-14(b)) unless the material has been formed into a ring and the following additional conditions are met:

(1) In a ring formed from plate, the original plate surfaces are parallel to the axis of the finished flange. (This is not intended to imply that the original plate surface should be present in the finished flange.)

(2) The joints in the ring are welded butt joints that conform to the requirements of this Section. Thickness to be used to determine postweld heat treatment and radiography requirements shall be the lesser of

\[
\frac{1}{2} (A - B)
\]

where these symbols are as defined in 2-3.

(3) The back of the flange and the outer surface of the hub are examined by either the magnetic particle method as per Mandatory Appendix V or the liquid penetrant method as per Mandatory Appendix VI.

(e) Bolts, studs, nuts, and washers shall comply with the requirements of this Section. It is recommended that bolts and studs have a nominal diameter of not less than \(13 \text{ mm (1/2 in.)}\). If bolts or studs smaller than \(13 \text{ mm (1/2 in.)}\) are used, ferrous bolting material shall be of alloy steel. Precautions shall be taken to avoid overstressing small-diameter bolts.

### 2-3 NOTATION

The symbols described below are used in the equations for the design of flanges (see also Figure 2-4):

- \(A\) = outside diameter of flange or, where slotted holes extend to the outside of the flange, the diameter to the bottom of the slots
- \(a\) = nominal bolt diameter
- \(A_b\) = cross-sectional area of the bolts using the root diameter of the thread or least diameter of unthreaded position, if less
- \(A_m\) = total required cross-sectional area of bolts, taken as the greater of \(A_{m1}\) and \(A_{m2}\)
- \(A_{m1}\) = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for the operating conditions
  \[= W_{m1} / S_b\]
- \(A_{m2}\) = total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket seating
  \[= W_{m2} / S_b\]
- \(B\) = inside diameter of flange. When \(B\) is less than \(20g_1\), it will be optional for the designer to substitute \(B_1\) for \(B\) in the formula for longitudinal stress \(S_L\)
- \(b\) = effective gasket or joint-contact-surface seating width [see Note in 2-5(c)(1)]
- \(B_1 = B + g_1\) for loose type flanges and for integral type flanges that have calculated values \(h / h_o\) and \(g_1 / g_o\) which would indicate an \(f\) value of less than 1.0, although the minimum value of \(f\) permitted is 1.0.
- \(B_f\) = bolt spacing. The bolt spacing may be taken as the bolt circle circumference divided by the number of bolts or as the chord length between adjacent bolt locations.
- \(B_{s\text{max}}\) = maximum bolt spacing
- \(C\) = bolt-circle diameter
- \(c\) = basic dimension used for the minimum sizing of welds equal to \(t_o\) or(468,831),(868,848), whichever is less
- \(C_o\) = conversion factor
  \[= 0.5\text{ for U.S. Customary calculations; 2.5 for SI calculations}\]
- \(d\) = factor
  \[= \frac{\mu}{V} h_o d_0^2\] for integral type flanges
  \[= \frac{\mu}{V_L} h_0 d_0^2\] for loose type flanges
- \(e\) = factor
  \[= \frac{F}{h_o}\] for integral type flanges
  \[= \frac{F_L}{h_o}\] for loose type flanges
- \(F\) = factor for integral type flanges (from Figure 2-7.2)
- \(f\) = hub stress correction factor for integral flanges from Figure 2-7.6 (When greater than one, this is the ratio of the stress in the small end of hub to the stress in the large end.) (For values below limit of figure, use \(f = 1\)).
- \(F_{L1}\) = factor for loose type flanges (from Figure 2-7.4)
- \(G\) = diameter at location of gasket load reaction. Except as noted in sketch (1) of Figure 2-4, \(G\) is defined as follows (see Table 2-5.2):
  \(a\) when \(b_o \leq 6 \text{ mm (1/4 in.)}, G = \text{ mean diameter of gasket contact face}\)
  \(b\) when \(b_o > 6 \text{ mm (1/4 in.)}, G = \text{ outside diameter of gasket contact face less } 2b\)
- \(g_1\) = thickness of hub at back of flange
- \(g_o\) = thickness of hub at small end
  \(a\) for optional type flanges calculated as integral and for integral type flanges per Figure 2-4, sketch (7), \(g_o = t_n\)
  \(b\) for other integral type flanges, \(g_o = \text{ the smaller of } t_n\) or the thickness of the hub at the small end
- \(H\) = total hydrostatic end force
  \[= 0.785G^2P\]
- \(h\) = hub length
\[ H_D = \text{hydrostatic end force on area inside of flange} = 0.785 B^2 \rho \]
\[ h_D = \text{radial distance from the bolt circle, to the circle on which } H_D \text{ acts, as prescribed in Table 2-6} \]
\[ H_G = \text{gasket load for the operating condition} = W_m - H \]
\[ h_G = \text{radial distance from gasket load reaction to the bolt circle} = (C - G)/2 \]
\[ h_o = \text{factor} = \sqrt{B \delta_p} \]
\[ H_p = \text{total joint-contact surface compression load} = 2b \times 3.14 GmP \]
\[ H_T = \text{difference between total hydrostatic end force and the hydrostatic end force on area inside of flange} = H - H_D \]
\[ h_T = \text{radial distance from the bolt circle to the circle on which } H_T \text{ acts as prescribed in Table 2-6} \]
\[ K = \text{ratio of outside diameter of flange to inside diameter of flange} = A/B \]
\[ L = \text{factor} = \frac{te + 1}{T} + \frac{t^3}{d} \]
\[ m = \text{gasket factor, obtain from Table 2-5.1 [see Note in 2-5(c)(1)]} \]
\[ M_D = \text{component of moment due to } H_D, = H_D h_D \]
\[ M_G = \text{component of moment due to } H_G, = H_G h_G \]
\[ M_o = \text{total moment acting upon the flange, for the operating conditions or gasket seating as may apply (see 2-6)} \]
\[ M_T = \text{component of moment due to } H_T, = H_T h_T \]
\[ N = \text{width used to determine the basic gasket seating with } h_o, \text{ based upon the possible contact width of the gasket (see Table 2-5.2)} \]
\[ P = \text{internal design pressure (see TD-150). For flanges subject to external design pressure, see 2-11.} \]
\[ R = \text{radial distance from bolt circle to point of intersection of hub and back of flange. For integral and hub flanges,} = \frac{C - B}{2} - \delta_1 \]
\[ S_a = \text{allowable bolt stress at atmospheric temperature (see TD-210).} \]
\[ S_b = \text{allowable bolt stress at design temperature (see TD-210).} \]
\[ S_f = \text{allowable design stress for material of flange at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply (see TD-210).} \]
\[ S_H = \text{calculated longitudinal stress in hub} \]
\[ S_A = \text{allowable design stress for material of nozzle neck, vessel or pipe wall, at design temperature (operating condition) or atmospheric temperature (gasket seating), as may apply (see TD-210).} \]
\[ S_R = \text{calculated radial stress in flange} \]
\[ S_T = \text{calculated tangential stress in flange} \]
\[ T = \text{factor involving } K \text{ (from Figure 2-7.1)} \]
\[ t = \text{flange thickness} \]
\[ t_n = \text{nominal thickness of shell or nozzle wall to which flange or lap is attached} \]
\[ t_x = \text{two times the thickness } g_o \text{, when the design is calculated as an integral flange or two times the thickness of shell nozzle wall required for internal pressure, when the design is calculated as a loose flange, but not less than } 6\text{ mm (1/4 in.)} \]
\[ U = \text{factor involving } K \text{ (from Figure 2-7.1)} \]
\[ V = \text{factor for integral type flanges (from Figure 2-7.3)} \]
\[ V_L = \text{factor for loose type flanges (from Figure 2-7.5)} \]
\[ W = \text{flange design bolt load, for the operating conditions or gasket seating, as may apply [see 2-5(e)]} \]
\[ w = \text{width used to determine the basic gasket seating width } b_o, \text{ based upon the contact width between the flange facing and the gasket (see Table 2-5.2)} \]
\[ W_{m1} = \text{minimum required bolt load for the operating conditions [see 2-5(c)]. For flange pairs used to contain a tubesheet for a floating head or a U-tube type of heat exchangers, or for any other similar design, } W_{m1} \text{ shall be the larger of the values as individually calculated for each flange, and that value shall be used for both flanges.} \]
\[ W_{m2} = \text{minimum required bolt load for gasket seating [see 2-5(c)]. For flange pairs used to contain a tubesheet for a floating head or U-tube type of heat exchanger, or for any other similar design where the flanges or gaskets are not the same, } W_{m2} \text{ shall be the larger of the values calculated for each flange and that value shall be used for both flanges.} \]
\[ Y = \text{factor involving } K \text{ (from Figure 2-7.1)} \]
\[ z = \text{gasket or joint-contact-surface unit seating load, [see Note 1, 2-5(c)]} \]
\[ Z = \text{factor involving } K \text{ (from Figure 2-7.1)} \]

2-4 CIRCULAR FLANGE TYPES

For purposes of computation, there are three types:

(a) Loose Type Flanges. This type covers those designs in which the flange has no direct connection to the nozzle neck, vessel, or pipe wall, and designs where the method of attachment is not considered to give the mechanical strength equivalent of integral attachment. See Figure 2-4, sketches (1), (1a), (2), (2a), (3), (3a), (4), (4a), (4b), and (4c) for typical loose type flanges and the location of the loads and moments. Welds and other details of

<table>
<thead>
<tr>
<th>Table 2-5.2</th>
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<td>[see TD-150].</td>
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<td>[see TD-210].</td>
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<td>[see TD-210].</td>
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2-5 BOLT LOADS

(a) General Requirements

(1) In the design of a bolted flange connection, calculations shall be made for each of the two design conditions of operating and gasket seating, and the more severe shall control.

(2) In the design of flange pairs used to contain a tubesheet of a heat exchanger or any similar design where the flanges and/or gaskets may not be the same, loads must be determined for the most severe condition of operating and/or gasket seating loads applied to each side at the same time. This most severe condition may be gasket seating on one flange with operating on the other, gasket seating on each flange at the same time, or operating on each flange at the same time. Although no specific rules are given for the design of the flange pairs, after the loads for the most severe conditions are determined, calculations shall be made for each flange following the rules of Mandatory Appendix.

(b) Integral Type Flanges. This type covers designs where the flange is cast or forged integrally with the nozzle neck, vessel or pipe wall, butt welded thereto, or attached by other forms of welding of such a nature that the flange and nozzle neck, vessel or pipe wall is considered to be the equivalent of an integral structure. In welded construction, the nozzle neck, vessel, or pipe wall is considered to act as a hub. See Figure 2-4, sketches (5), (6), (6a), (6b), and (7) for typical integral type flanges and the location of the loads and moments. Welds and other details of construction shall satisfy the dimensional requirements given in Figure 2-4, sketches (5), (6), (6a), (6b), and (7).

(c) Optional Type Flanges. This type covers designs where the attachment of the flange to the nozzle neck, vessel, or pipe wall is such that the assembly is considered to act as a unit, which shall be calculated as an integral flange, except that for simplicity the designer may calculate the construction as a loose type flange, provided none of the following values is exceeded:

\[
g_b = \frac{16 \text{ mm (5/8 in.)}}{B/\delta_b = 300} \\
P = \frac{2 \text{ MPa (300 psi)}}{\text{operating temperature } = 370^\circ C (700^\circ F)}
\]

See Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11) for typical optional type flanges. Welds and other details of construction shall satisfy the dimensional requirements given in Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11).

(3) Recommended minimum gasket contact widths for sheet and composite gaskets are provided in Table 2-4.

(b) Design Conditions

(1) Operating Conditions. The conditions required to resist the hydrostatic end force of the design pressure tending to part the joint, and to maintain on the gasket or joint-contact surface sufficient compression to assure a tight joint, all at the design temperature. The minimum load is a function of the design pressure, the gasket material, and the effective gasket or contact area to be kept tight under pressure, per eq. (c)(1)(1) below, and determines one of the two requirements for the amount of the bolting \( A_{m1} \). This load is also used for the design of the flange, per eq. (d)(3) below.

(2) Gasket Seating. The conditions existing when the gasket or joint-contact surface is seated by applying an initial load with the bolts when assembling the joint, at atmospheric temperature and pressure. The minimum initial load considered to be adequate for proper seating is a function of the gasket material, and the effective gasket or contact area to be seated, per eq. (c)(2)(2) below, and determines the other of the two requirements for the amount of bolting \( A_{m2} \). For the design of the flange, this load is modified per eq. (e)(4) below to take account of the operating conditions, when these govern the amount of bolting required \( A_{m} \), as well as the amount of bolting actually provided \( A_{b} \).

(c) Required Bolt Loads. The flange bolt loads used in calculating the required cross-sectional area of bolts shall be determined as follows.

(1) The required bolt load for the operating conditions \( W_{m1} \) shall be sufficient to resist the hydrostatic end force \( H \) exerted by the maximum allowable working pressure on the area bounded by the diameter of gasket reaction, and, in addition, to maintain on the gasket or joint-contact surface a compression load \( H_p \), which experience has shown to be sufficient to ensure a tight joint. (This compression load is expressed as a multiple \( m \) of the internal pressure. Its value is a function of the gasket material and construction.)

NOTE: Tables 2-5.1 and 2-5.2 give a list of many commonly used gasket materials and contact facings, with suggested values of \( m, b, \) and \( y \) that have proved satisfactory in actual service. These values are suggested only and are not mandatory.

The required bolt load for the operating conditions \( W_{m1} \) is determined in accordance with eq. (1).

\[
W_{m1} = H + H_p = 0.785G^2p + \left(2b \times 3.14Gmp\right)
\]

(2) Before a tight joint can be obtained, it is necessary to seat the gasket or joint-contact surface properly by applying a minimum initial load (under atmospheric temperature conditions without the presence of internal pressure), which is a function of the gasket material and
Figure 2-4
Types of Flanges

(1) Screwed Flange With Hub

(1a) Screwed Flange

(2) Screwed Flange With Hub

(2a) Screwed Flange [Note (2)]

(3) See Note (1)

[Note (2)]

(3a) [Note (2)]

(4) See Note (1)

[Note (2)]

(4a) [Note (2)]

(4b) [Note (2)]

(4c) [Note (2)]

Loose-Type Flanges [Notes (3) and (4)]

To be taken at midpoint of contact between flange and lap independent of gasket location. (Optional hub is shown by dotted line.)

Full penetration weld, single or double. The full penetration weld may be through the lap \( t_l \) or through the wall \( t_n \).

This weld may be machined to a corner radius to suit standard lap joint flanges.

Gasket

Gasket

Gasket

Gasket

---

\( h_G \) or \( h_l \)

\( H_G + H_T \)

\( t \)

\( h \)

\( W \)

\( t_l \geq t_n \)

\( G \)

\( h_D \)

\( H_D \)

\( g_1 \)

\( g_0 \)

\( H \)

\( h_D \)

\( h_D \)

\( g_1 \)

\( g_0 \)

\( C \)

\( A \)

\( B \)

\( C \)

\( t \)

\( h \)

\( W \)

\( t_l \geq t_n \)

\( G \)

\( h_D \)

\( H_D \)

\( g_1 \)

\( g_0 \)

\( C \)

\( A \)

\( B \)

\( C \)

\( t \)

\( h \)

\( W \)

\( t_l \geq t_n \)

\( G \)

\( h_D \)

\( H_D \)

\( g_1 \)

\( g_0 \)

\( C \)

\( A \)

\( B \)

\( C \)

\( t \)

\( h \)

\( W \)

\( t_l \geq t_n \)

\( G \)

\( h_D \)

\( H_D \)

\( g_1 \)

\( g_0 \)

\( C \)

\( A \)

\( B \)

\( C \)

\( t \)

\( h \)

\( W \)

\( t_l \geq t_n \)

\( G \)

\( h_D \)

\( H_D \)

\( g_1 \)

\( g_0 \)

\( C \)

\( A \)

\( B \)

\( C \)

\( t \)

\( h \)

\( W \)

\( t_l \geq t_n \)

\( G \)

\( h_D \)

\( H_D \)

\( g_1 \)

\( g_0 \)

\( C \)

\( A \)

\( B \)

\( C \)

\( t \)

\( h \)

\( W \)

\( t_l \geq t_n \)

\( G \)

\( h_D \)

\( H_D \)

\( g_1 \)

\( g_0 \)

\( C \)

\( A \)

\( B \)

\( C \)

\( t \)

\( h \)

\( W \)

\( t_l \geq t_n \)

\( G \)

\( h_D \)

\( H_D \)

\( g_1 \)

\( g_0 \)

\( C \)

\( A \)

\( B \)

\( C \)

\( t \)

\( h \)

\( W \)

\( t_l \geq t_n \)

\( G \)

\( h_D \)

\( H_D \)

\( g_1 \)

\( g_0 \)

\( C \)

\( A \)

\( B \)

\( C \)

\( t \)

\( h \)

\( W \)

\( t_l \geq t_n \)

\( G \)

\( h_D \)

\( H_D \)

\( g_1 \)

\( g_0 \)

\( C \)

\( A \)

\( B \)

\( C \)

\( t \)

\( h \)

\( W \)

\( t_l \geq t_n \)
Figure 2-4
Types of Flanges (Cont’d)

Gasket

(5)

Integral-Type Flanges [Notes (3) and (4)]

(6)

Where hub slope adjacent to flange exceeds 1:3, use sketches (6a) or (6b)

Where hub slope adjacent to flange exceeds 1:3, use sketches (6a) or (6b)

(7)

0.25$g_o$, but not less than 6 mm (1/4 in.), the minimum for either leg. This weld may be machined to a corner radius as permitted in sketch (5), in which case $g_1 = g_o$. 

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Optional-Type Flanges [Notes (5), (6), and (7)]

(12) For Flanged Nozzles 460 mm (18 in.) and Smaller Nominal Size
(12a) For Flanged Nozzles Over 460 mm (18 in.) Nominal Size

Flanges With Nut Stops [Note (8)]

NOTES:
(1) For hub tapers 6 deg or less, use \( g_o = g_1 \).
(2) Loading and dimensions for sketches (2a), (3), (3a), (4), (4a), (4b), and (4c) not shown are the same as for sketch (2).
(3) Fillet radius \( r \) to be at least 0.25 \( g_1 \) but not less than 5 mm (3/16 in.).
the effective gasket area to be seated. The minimum initial bolt load required for this purpose \( W_{m2} \) shall be determined in accordance with eq. (2).

\[
W_{m2} = 3.14b_0g_y 
\]

(2)

The need for providing sufficient bolt load to seat the gasket or joint-contact surfaces in accordance with eq. (2) will prevail on many low-pressure designs and with facings and materials that require a high seating load, and where the bolt load computed by eq. (1)(1) for the operating conditions is insufficient to seat the joint. Accordingly, it is necessary to furnish bolting and to pretighten the bolts to provide a bolt load sufficient to satisfy both of these requirements, each one being individually investigated. When eq. (2) governs, flange proportions will be a function of the bolting instead of internal pressure.

(3) Bolt loads for flanges using gaskets of the self-energizing type differ from those shown above.

(a) The required bolt load for the operating conditions \( W_{m1} \) shall be sufficient to resist the hydrostatic end force \( H \) exerted by the maximum allowable working pressure on the area bounded by the outside diameter of the gasket. \( H_p \) is to be considered as 0 for all self-energizing gaskets except certain seal configurations which generate axial loads which must be considered.

(b) \( W_{m2} = 0 \).

Self-energizing gaskets may be considered to require an inconsequential amount of bolting force to produce a seal. Bolting, however, must be pretightened to provide a bolt load sufficient to withstand the hydrostatic end force \( H \).

(d) Total Required and Actual Bolt Areas, \( A_m \) and \( A_b \). The total cross-sectional area of bolts \( A_m \) required for both the operating conditions and gasket seating is the greater of the values for \( A_{m1} \) and \( A_{m2} \), where \( A_{m1} = W_{m1}/S_b \) and \( A_{m2} = W_{m2}/S_p \). A selection of bolts to be used shall be made such that the actual total cross-sectional area of bolts \( A_b \) will not be less than \( A_m \). For vessels in lethal service or when specified by the user or his designated agent, the maximum bolt spacing shall not exceed the value calculated in accordance with eq. (3).

\[
B_{b_{\text{max}}} = 2a + \frac{6t}{m + 0.5} 
\]

(3)

(e) Flange Design Bolt Load \( W \). The bolt loads used in the design of the flange shall be the values obtained from eqs. (4) and (5). For operating conditions,

\[
W = W_{m1} 
\]

(4)

For gasket seating,

\[
W = \frac{(A_m + A_b)S_u}{2} 
\]

(5)

\( S_u \) used in eq. (5) shall be not less than that tabulated in the stress tables (see TD-210). In addition to the minimum requirements for safety, eq. (5) provides a margin against abuse of the flange from overbolting. Since the margin against such abuse is needed primarily for the initial, bolting-up operation which is done at atmospheric temperature and before application of internal pressure, the flange design is required to satisfy this loading only under such conditions.
Table 2-5.1
Gasket Materials and Contact Facings
Gasket Factors $m$ for Operating Conditions and Minimum Design Seating Stress $y$

<table>
<thead>
<tr>
<th>Gasket Material</th>
<th>Gasket Factor $m$</th>
<th>Min. Design Seating Stress $y$, psi (MPa)</th>
<th>Sketches</th>
<th>Facing Sketch and Column in Table 2-5.2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Self-energizing types (O-rings, metallic, elastomer, other gasket types considered as self-sealing)</td>
<td>0</td>
<td>0 (0)</td>
<td>...</td>
<td>...</td>
</tr>
<tr>
<td>Elastomers without fabric or high percent of mineral fiber:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Below 75A Shore Durometer</td>
<td>0.50</td>
<td>0 (0)</td>
<td>(1a), (1b), (1c), (1d), (4), (5): Column II</td>
<td></td>
</tr>
<tr>
<td>75A or higher Shore Durometer</td>
<td>1.00</td>
<td>200 (1.4)</td>
<td>(1a), (1b), (1c), (1d), (4), (5): Column II</td>
<td></td>
</tr>
<tr>
<td>Mineral fiber with suitable binder for operating conditions:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.2 mm (1/8 in.) thick</td>
<td>2.00</td>
<td>1,600 (11)</td>
<td>(1a), (1b), (1c), (1d), (4), (5): Column II</td>
<td></td>
</tr>
<tr>
<td>1.6 mm (1/16 in.) thick</td>
<td>2.75</td>
<td>3,700 (26)</td>
<td>(1a), (1b), (1c), (1d), (4), (5): Column II</td>
<td></td>
</tr>
<tr>
<td>0.8 mm (1/32 in.) thick</td>
<td>3.50</td>
<td>6,500 (45)</td>
<td>(1a), (1b), (1c), (1d), (4), (5): Column II</td>
<td></td>
</tr>
<tr>
<td>Elastomers with cotton fabric insertion</td>
<td>1.25</td>
<td>400 (2.8)</td>
<td>(1a), (1b), (1c), (1d), (4), (5): Column II</td>
<td></td>
</tr>
<tr>
<td>Elastomers with mineral fiber fabric insertion (with or without wire reinforcement):</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3-ply</td>
<td>2.25</td>
<td>2,200 (15)</td>
<td>(1a), (1b), (1c), (1d), (4), (5): Column II</td>
<td></td>
</tr>
<tr>
<td>2-ply</td>
<td>2.50</td>
<td>2,900 (20)</td>
<td>(1a), (1b), (1c), (1d), (4), (5): Column II</td>
<td></td>
</tr>
<tr>
<td>1-ply</td>
<td>2.75</td>
<td>3,700 (26)</td>
<td>(1a), (1b), (1c), (1d), (4), (5): Column II</td>
<td></td>
</tr>
<tr>
<td>Vegetable fiber</td>
<td>1.75</td>
<td>1,100 (7.6)</td>
<td>(1a), (1b), (1c), (1d), (4), (5): Column II</td>
<td></td>
</tr>
<tr>
<td>Spiral-wound metal, mineral fiber filled:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Carbon</td>
<td>2.50</td>
<td>10,000 (69)</td>
<td>(1a), (1b): Column II</td>
<td></td>
</tr>
<tr>
<td>Stainless, Monel, and nickel-base alloys</td>
<td>3.00</td>
<td>10,000 (69)</td>
<td>(1a), (1b): Column II</td>
<td></td>
</tr>
<tr>
<td>Corrugated metal, mineral fiber inserted, or corrugated metal, jacketed mineral fiber filled:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Soft aluminum</td>
<td>2.50</td>
<td>2,900 (20)</td>
<td>(1a), (1b): Column II</td>
<td></td>
</tr>
<tr>
<td>Soft copper or brass</td>
<td>2.75</td>
<td>3,700 (26)</td>
<td>(1a), (1b): Column II</td>
<td></td>
</tr>
<tr>
<td>Iron or soft steel</td>
<td>3.00</td>
<td>4,500 (31)</td>
<td>(1a), (1b): Column II</td>
<td></td>
</tr>
<tr>
<td>Monel or 4–6% chrome</td>
<td>3.25</td>
<td>5,500 (38)</td>
<td>(1a), (1b): Column II</td>
<td></td>
</tr>
<tr>
<td>Stainless steels and nickel-base alloys</td>
<td>3.50</td>
<td>6,500 (45)</td>
<td>(1a), (1b): Column II</td>
<td></td>
</tr>
<tr>
<td>Corrugated metal</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Soft aluminum</td>
<td>2.75</td>
<td>3,700 (26)</td>
<td>(1a), (1b), (1c), (1d): Column II</td>
<td></td>
</tr>
<tr>
<td>Soft copper or brass</td>
<td>3.00</td>
<td>4,500 (31)</td>
<td>(1a), (1b), (1c), (1d): Column II</td>
<td></td>
</tr>
<tr>
<td>Iron or soft steel</td>
<td>3.25</td>
<td>6,500 (38)</td>
<td>(1a), (1b), (1c), (1d): Column II</td>
<td></td>
</tr>
<tr>
<td>Monel or 4–6% chrome</td>
<td>3.50</td>
<td>6,500 (45)</td>
<td>(1a), (1b), (1c), (1d): Column II</td>
<td></td>
</tr>
<tr>
<td>Stainless steels and nickel-base alloys</td>
<td>3.75</td>
<td>7,600 (52)</td>
<td>(1a), (1b), (1c), (1d): Column II</td>
<td></td>
</tr>
<tr>
<td>Flat metal, jacketed mineral fiber filled:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Soft aluminum</td>
<td>3.25</td>
<td>5,500 (38)</td>
<td>(1a), (1b): Column II</td>
<td></td>
</tr>
<tr>
<td>Soft copper or brass</td>
<td>3.50</td>
<td>6,500 (45)</td>
<td>(1a) [Note (1)], (1c) [Note (1)]</td>
<td></td>
</tr>
<tr>
<td>Iron or soft steel</td>
<td>3.75</td>
<td>7,600 (52)</td>
<td>(1a) [Note (1)], (1d) [Note (1)]</td>
<td></td>
</tr>
<tr>
<td>Monel</td>
<td>3.50</td>
<td>8,000 (55)</td>
<td>(2) [Note (1)]</td>
<td></td>
</tr>
<tr>
<td>4–6% chrome</td>
<td>3.75</td>
<td>9,000 (62)</td>
<td>(1a), (1b), (1c), (1d): Column II</td>
<td></td>
</tr>
<tr>
<td>Stainless steels and nickel-base alloys</td>
<td>3.75</td>
<td>9,000 (62)</td>
<td>(1a), (1b), (1c), (1d): Column II</td>
<td></td>
</tr>
</tbody>
</table>

Swap units to MPa (psi) for entire column.
NOTE: Where additional safety against abuse is desired, or where it is necessary that the flange be suitable to withstand the full available bolt load \( A_b S_a \), the flange may be designed on the basis of this latter quantity.

### Table 2-5.1

<table>
<thead>
<tr>
<th>Gasket Material</th>
<th>Gasket Factor ( m )</th>
<th>Min. Design Seating Stress ( y ), psi (MPa)</th>
<th>Facing Sketch and Column in Table 2-5.2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solid flat metal:</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Soft aluminum</td>
<td>3.25</td>
<td>5,500 (38)</td>
<td>(1a), (1b), (1c), (3), (4), (5), Column II</td>
</tr>
<tr>
<td>Soft copper or brass</td>
<td>3.50</td>
<td>6,500 (45)</td>
<td>(1d), (2), (3), (4), (5)</td>
</tr>
<tr>
<td>Iron or soft metal</td>
<td>3.75</td>
<td>7,600 (52)</td>
<td>Column I</td>
</tr>
<tr>
<td>Monel or 4–6% chrome</td>
<td>3.75</td>
<td>9,000 (62)</td>
<td>(5), Column I</td>
</tr>
<tr>
<td>Stainless steels and nickel-base alloys</td>
<td>4.25</td>
<td>10,100 (70)</td>
<td></td>
</tr>
<tr>
<td>Solid flat metal:</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Soft aluminum</td>
<td>4.00</td>
<td>8,800 (61)</td>
<td>(1a), (1b), (1c), (3), (4), (5)</td>
</tr>
<tr>
<td>Soft copper or brass</td>
<td>4.75</td>
<td>13,000 (90)</td>
<td>(1d), (2), (3), (4), (5)</td>
</tr>
<tr>
<td>Iron or soft steel</td>
<td>5.50</td>
<td>18,000 (124)</td>
<td>(5), Column I</td>
</tr>
<tr>
<td>Monel or 4–6% chrome</td>
<td>6.00</td>
<td>21,800 (150)</td>
<td>(5), Column I</td>
</tr>
<tr>
<td>Stainless steels and nickel-base alloys</td>
<td>6.50</td>
<td>26,000 (180)</td>
<td></td>
</tr>
</tbody>
</table>

**GENERAL NOTE:** This Table gives a list of many commonly used gasket materials and contact facings with suggested design values of \( m \) and \( y \) that have generally proved satisfactory in actual service when using effective gasket seating width \( b \) given in Table 2-5.2. The design values and other details given in this Table are suggested only and are not mandatory.

**NOTE:**
(1) The surface of a gasket having a lap should not be against the nubbin.

For gasket seating, the total flange moment \( M_o \) is based on the flange design bolt load of eq. 2-5(e)(5), which is opposed only by the gasket load, in which case

\[
M_o = W \left[ \frac{(C - G)}{2} \right]
\]

For vessels in lethal service or when specified by the user or his designated agent, the bolt spacing correction shall be applied in calculating the flange stress in 2-7, 2-13(c), and 2-13(d). The flange moment \( M_o \) without correction for bolt spacing is used for the calculation of the rigidity index in 2-14.

When the bolt spacing exceeds \( 2a + t \), multiply \( M_o \) by the bolt spacing correction factor \( B_{SC} \) for calculating flange stress, where

\[
B_{SC} = \sqrt{\frac{R_s}{2a + t}}
\]
Table 2-5.2  
**Effective Gasket Width**

<table>
<thead>
<tr>
<th>Facing Sketch (Exaggerated)</th>
<th>Basic Gasket Seating Width, $b_o$</th>
</tr>
</thead>
<tbody>
<tr>
<td>(1a)</td>
<td></td>
</tr>
<tr>
<td><img src="image" alt="Facing Sketch (Exaggerated)" /></td>
<td>$b = b_o$, when $b_o \leq \frac{1}{4}$ in. (6 mm); $b = 2b_o$, when $b_o &gt; \frac{1}{4}$ in. (6 mm)</td>
</tr>
<tr>
<td>(1b)</td>
<td></td>
</tr>
<tr>
<td><img src="image" alt="Facing Sketch (Exaggerated)" /></td>
<td>See Note (1)</td>
</tr>
</tbody>
</table>

| (1c)                        | $\frac{w + T}{2} \left(\frac{w + N}{4} \max\right)$ |
| w ≤ N                      | $\frac{w + T}{2} \left(\frac{w + N}{4} \max\right)$ |
| ![Facing Sketch (Exaggerated)](image) |                                   |

| (1d)                        | $\frac{w + T}{2} \left(\frac{w + N}{4} \max\right)$ |
| w ≤ N                      | $\frac{w + T}{2} \left(\frac{w + N}{4} \max\right)$ |
| ![Facing Sketch (Exaggerated)](image) |                                   |

| (2)                         | $\frac{w + N}{4}$ |
| 0.4 mm (1/64 in.) nubbin | $\frac{w + 3N}{8}$ |
| ![Facing Sketch (Exaggerated)](image) | w ≤ N/2 |

| (3)                         | $\frac{N}{4}$ |
| 0.4 mm (1/64 in.) nubbin | $\frac{3N}{8}$ |
| ![Facing Sketch (Exaggerated)](image) | w ≤ N/2 |

| (4)                         | $\frac{3N}{8}$ |
| ![Facing Sketch (Exaggerated)](image) | See Note (1) |

| (5)                         | $\frac{N}{4}$ |
| ![Facing Sketch (Exaggerated)](image) | See Note (1) |

| (6)                         | $\frac{w}{8}$ |
| ![Facing Sketch (Exaggerated)](image) |                                   |

Effective Gasket Seating Width, $b$

- $b = b_o$, when $b_o \leq \frac{6}{4}$ mm (1/4 in.)
- $b = 2b_o$, when $b_o > \frac{6}{4}$ mm (1/4 in.)
Table 2-5.2
Effective Gasket Width (Cont'd)

Location of Gasket Load Reaction

GENERAL NOTE: The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.

NOTE:
(1) Where serrations do not exceed 0.4 mm (1/64 in.) depth and 0.8 mm (1/32 in.) width spacing, sketches (1b) and (1d) shall be used.

Table 2-6
Moment Arms for Flange Loads Under Operating Conditions

<table>
<thead>
<tr>
<th></th>
<th>$h_D$</th>
<th>$h_T$</th>
<th>$h_L$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Integral-type flanges [see Figure 2-4, sketches (5), (6), (6a), (6b), and (7)] and optional type flanges calculated as integral type [see Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)]</td>
<td>$R + 0.5g_1$</td>
<td>$R + g_1 + h_G$</td>
<td>$C - G$</td>
</tr>
<tr>
<td>Loose type, except lap-joint flanges [see Figure 2-4, sketches (2), (2a), (3), (3a), (4), and (4a)]; and optional type flanges calculated as loose type [see Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)]</td>
<td>$C - B$</td>
<td>$h_D + h_G$</td>
<td>$C - G$</td>
</tr>
<tr>
<td>Lap-type flanges [see Figure 2-4, sketches (1) and (1a)]</td>
<td>$C - B$</td>
<td>$C - G$</td>
<td>$C - G$</td>
</tr>
</tbody>
</table>
2-7 CALCULATION OF FLANGE STRESSES

The stresses in the flange shall be determined for both the operating conditions and gasket seating condition, whichever controls, in accordance with the following equations:

(a) for integral type flanges [Figure 2-4, sketches (5), (6), (6a), (6b), and (7)], for optional type flanges calculated as integral type [Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)], and for loose type flanges with a hub which is considered [Figure 2-4, sketches (1), (1a), (2), (2a), (3), (3a), (4), (4a), (4b), and (4c)]:

Longitudinal hub stress

\[ S_H = \frac{fM_o}{Lg^2B} \]  
(8)

Radial flange stress

\[ S_R = \frac{(1.33\pi + 1)M_o}{L\pi^2B} \]  
(9)

Tangential flange stress

\[ S_T = \frac{YM_o}{t^2B} - ZS_R \]  
(10)

(b) for loose type flanges without hubs and loose type flanges with hubs which the designer chooses to calculate without considering the hub [Figure 2-4, sketches (1), (1a), (2), (2a), (3a), (4), (4a), (4b), and (4c)] and optional type flanges calculated as loose type [Figure 2-4, sketches (8), (8a), (9), (9a), (10), (10a), and (11)]:

\[ S_T = \frac{YM_o}{t^2B} \]

\[ S_R = 0 \]

\[ S_H = 0 \]  
(11)

2-8 ALLOWABLE FLANGE DESIGN STRESSES

(a) The flange stresses calculated by the equations in 2-7 shall not exceed the following values:

(1) longitudinal hub stress \( S_H \) not greater than \( S_f \) for cast iron and, except as otherwise limited by (a) and (b) below, not greater than 1.5\( S_f \) for materials other than cast iron:

\[ (-a) \text{ longitudinal hub stress } S_H \text{ not greater than } S_f \]

\[ \text{and, except as otherwise limited by } (-a) \text{ and } (-b) \text{ below, not greater than } 1.5 S_f \text{ for materials other than } S_f. \]

\[ (-b) \text{ longitudinal hub stress } S_H \text{ not greater than } S_f; \]

(2) radial flange stress \( S_R \) not greater than \( S_f \);

(3) tangential flange stress \( S_T \) not greater than \( S_f \);

(4) also \( (S_H + S_R)/2 \) not greater than \( S_f \) and \( (S_H + S_T)/2 \) not greater than \( S_f \).

(b) For hub flanges attached as shown in Figure 2-4, sketches (2), (2a), (3), (3a), (4), (4a), (4b), and (4c), the nozzle neck, vessel or pipe wall shall not be considered to have any value as a hub.

(c) In the case of loose type flanges with laps, as shown in Figure 2-4, sketches (1) and (1a), where the gasket is so located that the lap is subjected to shear, the shearing stress shall not exceed 0.8 \( S_n \) for the material of the lap, as defined in 2-3.

2-9 SPLIT LOOSE FLANGES

Loose flanges split across a diameter and designed under the rules given in this Appendix may be used under the following provisions.

(a) When the flange consists of a single split flange or flange ring, it shall be designed as if it were a solid flange (without splits), using 200% of the total moment \( M_o \) as defined in 12-4.

(b) When the flange consists of two split rings each ring shall be designed as if it were a solid flange (without splits), using 75% of the total moment \( M_o \) as defined in 12-4. The pair of rings shall be assembled so that the splits in one ring shall be 90 deg from the splits in the other ring.

(c) The splits should preferably be midway between bolt holes.

(d) It is not a requirement that the flange rigidity rules of 2-14 be applied to split loose flanges.

Add Endnote:

When the flange material is cast iron, particular care should be taken when tightening the bolts to avoid excessive stress that may break the flange. The longitudinal hub stress has been limited to 5\( S_f \) in order to minimize any cracking of flanges. An attempt should be made to apply no greater torque than is needed to assure tightness during the hydrostatic test.

Add Endnote:

Loose flanges of the type shown in Figure 2-4, sketch (1) are of the split design when it is necessary to install them after heat treatment of a stainless steel vessel, or when for any reason it is desired to have them completely removable from the nozzle neck or vessel.
Figure 2-7.1
Values of $T$, $U$, $Y$, and $Z$ (Terms Involving $K$)

\[
T = \frac{K^2 (1 + 8.55246 \log_{10} K)^{-1}}{(1.04720 \times 1.9448 K^2) (K-1)}
\]
\[
U = \frac{K^2 (1 + 8.55246 \log_{10} K)^{-1}}{1.36136 (K^2 - 1) (K-1)}
\]
\[
Y = \frac{1}{K-1} \left[ 0.66845 + 5.71690 \left( \frac{K^2 \log_{10} K}{K^2 - 1} \right) \right]
\]
\[
Z = \frac{K^2 + 1}{K^2 - 1}
\]

Poisson’s ratio assumed = 0.3

$K = \frac{A}{B}$
Figure 2-7.2
Values of $F$ (Integral Flange Factors)

GENERAL NOTE: See Table 2-7.1 for equations.
Figure 2-7.3
Values of V (Integral Flange Factors)

GENERAL NOTE: See Table 2-7.1 for equations.
Figure 2-7.4
Values of $F_L$ (Loose Hub Flange Factors)

GENERAL NOTE: See Table 2-7.1 for equations.

Figure 2-7.5
Values of $V_L$ (Loose Hub Flange Factors)

GENERAL NOTE: See Table 2-7.1 for equations.
Figure 2-7.6
Values of \( f \) (Hub Stress Correction Factor)

\[ f = 1 \text{ (minimum)} \]
\[ = 1 \text{ for hubs of uniform thickness } (g_1 / g_0 = 1) \]
\[ = 1 \text{ for loose hubbed flanges} \]

GENERAL NOTE: See Table 2-7.1 for equations.
Table 2-7.1

<table>
<thead>
<tr>
<th>Integral Flange</th>
<th>Loose Hub Flange</th>
</tr>
</thead>
<tbody>
<tr>
<td>Factor $F$ per Figure 2-7.2 is then solved by</td>
<td>Factor $F_L$ per Figure 2-7.4 is solved by</td>
</tr>
<tr>
<td>$F = \frac{E_6}{C} \left(1 + A\right)^3$</td>
<td>$F_L = C_{18}\left(\frac{1}{2} + \frac{A}{6}\right) + C_{21}\left(\frac{1}{4} + \frac{11A}{84}\right) + C_{24}\left(\frac{1}{70} + \frac{A}{105}\right) - \left(\frac{1}{40} + \frac{A}{72}\right)$</td>
</tr>
<tr>
<td>Factor $V$ per Figure 2-7.3 is then solved by</td>
<td>Factor $V_L$ per Figure 2-7.5 is solved by</td>
</tr>
<tr>
<td>$V = \frac{E_4}{C} \left(1 + A\right)^3$</td>
<td>$V_L = \frac{1}{4} \cdot \frac{C_{24} - \frac{3C_{21}}{2}}{\left(1 + A\right)^3}$</td>
</tr>
<tr>
<td>Factor $f$ per Figure 2-7.6 is then solved by</td>
<td>Factor $f$ per Figure 2-7.6 is set equal to 1.</td>
</tr>
<tr>
<td>$f = C_{26} / \left(1 + A\right)$</td>
<td>$f = 1$</td>
</tr>
</tbody>
</table>

The values used in the above equations are solved using eqs. (1) through (45) below based on the values $g_1, g_2, h, h$, and $h_1$, as defined by 2-3. When $g_1 = g_2$, $F = 0.908920$, $V = 0.550103$, and $f = 1$; thus eqs. (1) through (45) need not be solved. The values used in the above equations are solved using eqs. (1) through (5), (7), (9), (10), (12), (14), (16), (18), (20), (23), and (26) below based on the values $g_1, g_2, h, h$, and $h_1$, as defined by 2-3.

**Equations**

1. $A = (g_1/g_2) - 1$
2. $C = 43.68(h/h_0)\sqrt{h}$
3. $C_1 = 1/3 + A/12$
4. $C_2 = 5/4 + 17A/336$
5. $C_3 = 1/20 + A/360$
6. $C_4 = 11/360 + 59A/5040 + (1 + 3A)/C$
7. $C_5 = 1/100 + 54/1008 - (1 + A)^2/C$
8. $C_6 = 1/120 + 17A/5040 + 1/C$
9. $C_7 = 215/2772 + 51A/1232 + (60/7 + 225A/14 + 75A^2/7 + 5A^3/2)/C$
10. $C_8 = 31/6930 + 128A/45,045 + (6/7 + 15A/7 + 12A^2/7 + 5A^3/11)/C$
12. $C_{10} = 29/3780 + 3A/704 - (1/2 + 33A/14 + 81A^2/28 + 13A^3/12)/C$
13. $C_{11} = 31/6048 + 176A/665,280 + (1/2 + 2A/7 + 15A^2/28 + 5A^3)/C$
14. $C_{12} = 1/2925 + 71A/300,300 + (8/35 + 18A/35 + 156A^2/865 + 9A^3)/C$
15. $C_{13} = 215/2,865,600 + 937A/1,663,200 + (1/35 + 13A + 11A^3)/C$
16. $C_{14} = 197/4,15,800 + 103A/332,640 - (1/35 + 6A/15 + 17A^2/70 + A^3)/C$
17. $C_{15} = 233/9,83,359,000 + 97A/554,400 + (1/35 + 3A/35 + A^2/14 + 2A^3)/C$
18. $C_{16} = C_{16}C_{12} + C_{16}C_{12} + C_{16}C_{12} - C_{16}C_{12} - C_{16}C_{12} - C_{16}C_{12} - C_{16}C_{12} - C_{16}C_{12} - C_{16}C_{12}$
19. $C_{17} = C_{17}C_{12} + C_{17}C_{12} + C_{17}C_{12} - C_{17}C_{12} - C_{17}C_{12} - C_{17}C_{12} - C_{17}C_{12} - C_{17}C_{12} - C_{17}C_{12}$
20. $C_{18} = C_{18}C_{12} + C_{18}C_{12} + C_{18}C_{12} - C_{18}C_{12} - C_{18}C_{12} - C_{18}C_{12} - C_{18}C_{12} - C_{18}C_{12} - C_{18}C_{12}$
21. $C_{19} = C_{19}C_{12} + C_{19}C_{12} + C_{19}C_{12} - C_{19}C_{12} - C_{19}C_{12} - C_{19}C_{12} - C_{19}C_{12} - C_{19}C_{12} - C_{19}C_{12}$
22. $C_{20} = C_{20}C_{12} + C_{20}C_{12} + C_{20}C_{12} - C_{20}C_{12} - C_{20}C_{12} - C_{20}C_{12} - C_{20}C_{12} - C_{20}C_{12} - C_{20}C_{12}$
23. $C_{21} = C_{21}C_{12} + C_{21}C_{12} + C_{21}C_{12} - C_{21}C_{12} - C_{21}C_{12} - C_{21}C_{12} - C_{21}C_{12} - C_{21}C_{12} - C_{21}C_{12}$
24. $C_{22} = C_{22}C_{12} + C_{22}C_{12} + C_{22}C_{12} - C_{22}C_{12} - C_{22}C_{12} - C_{22}C_{12} - C_{22}C_{12} - C_{22}C_{12} - C_{22}C_{12}$
25. $C_{23} = C_{23}C_{12} + C_{23}C_{12} + C_{23}C_{12} - C_{23}C_{12} - C_{23}C_{12} - C_{23}C_{12} - C_{23}C_{12} - C_{23}C_{12} - C_{23}C_{12}$
26. $C_{24} = C_{24}C_{12} + C_{24}C_{12} + C_{24}C_{12} - C_{24}C_{12} - C_{24}C_{12} - C_{24}C_{12} - C_{24}C_{12} - C_{24}C_{12} - C_{24}C_{12}$
27. $C_{25} = C_{25}C_{12} + C_{25}C_{12} + C_{25}C_{12} - C_{25}C_{12} - C_{25}C_{12} - C_{25}C_{12} - C_{25}C_{12} - C_{25}C_{12} - C_{25}C_{12}$
28. $C_{26} = - (C/4)^{1/4}$
29. $C_{27} = C_{27} - C_{17} = 5/12 + C_{17}C_{26}$
30. $C_{28} = C_{28} - C_{19} = 1/12 + C_{19}C_{26}$
31. $C_{29} = - (C/4)^{1/4}$
32. $C_{30} = - (C/4)^{1/4}$
33. $C_{31} = 3A/2 - C_{19}C_{30}$
34. $C_{32} = 1/2 - C_{19}C_{30}$
35. $C_{33} = 0.5C_{20}C_{32} + 2C_{20}C_{32} - (0.5C_{20}C_{26} + C_{20}C_{26})$
2-10 NONCIRCULAR SHAPED FLANGES WITH CIRCULAR BORE

The outside diameter $A$ for a noncircular flange with a circular bore shall be taken as the diameter of the largest circle, concentric with the bore, inscribed entirely within the outside edges of the flange. Bolt loads and moments, as well as stresses, are then calculated as for circular flanges, using a bolt circle drawn through the centers of the outermost bolt holes.

2-11 FLANGES SUBJECT TO EXTERNAL PRESSURES

(a) The design of flanges for external pressure only shall be based on the equations given in 2-7 for internal pressure except that for operating conditions:

$$M_o = H_D(h_D - h_G) + H_T(h_T - h_G)$$  \hspace{1cm} (10)

For gasket seating,

$$M_o = W h_G$$  \hspace{1cm} (11)

where

$$W = \frac{A_m 2 + A_b S_g}{2}$$  \hspace{1cm} (11a)

$$H_D = 0.785B^2 P_e$$  \hspace{1cm} (11b)

$$H_T = H - H_D$$  \hspace{1cm} (11c)

$$H = 0.785G^2 P_e$$  \hspace{1cm} (11d)

$P_e$ = external design pressure

See 2-3 for definitions of other symbols. $S_g$ used in eq. (11a) shall be not less than that tabulated in the stress tables (see TD-210).

(b) When flanges are subject at different times during operation to external or internal pressure, the design shall satisfy the external pressure design requirements given in (a) above and the internal pressure design requirements given elsewhere in this Appendix.

The requirements of 2-4(c) apply to Figure 2-4, sketches (a) and (b). The requirements of 2-4(c) apply to Figure 2-4, sketches (a) and (b).

NOTE: The combined force of external pressure and bolt loading may plastically deform certain gaskets to result in loss of gasket contact pressure when the connection is depressurized. To maintain a tight joint when the unit is repressurized, consideration should be given to gasket and facing details so that excessive deformation of the gasket will not occur. Joints subject to pressure reversals, such as in heat exchanger floating heads, are in this type of service.

2-12 FLANGES WITH NUT-STOPs

(a) When flanges are designed per this Appendix, or are fabricated to the dimensions of ASME B16.5 or other acceptable standards [see TD-100.5], except that the dimension $R$ is decreased to provide a nut-stop, the fillet radius relief shall be as shown in Figure 2-4, sketches (12) and (12a) except that:

(1) for flanges designed to this Appendix, the minimum dimension $g_1$ must be the lesser of $2t$ (t from TD-300) or $4r$, but in no case less than $13$ mm (1/2 in.).

where $r$ = the radius of the undercut

(2) for ASME B16.5 or other standard flanges, the dimension of the hub $g_o$ shall be increased as necessary to provide a nut-stop.

2-13 REVERSE FLANGES

(a) Flanges with the configuration as indicated in Figure 2-13.1 shall be designed as integral reverse flanges and those in Figure 2-13.2 shall be designed as loose ring type reverse flanges. These flanges shall be designed in conformance with the rules in 2-3 through 2-8, but with the modifications as described in the following. Mandatory use of these rules is limited to $K \leq 2$. When $K > 2$, results become increasingly conservative and U-2(g) may be used.

(1) Integral Type Reverse Flange. The shell-to-flange attachment of integral type reverse flanges may be attached as shown in Figure 2-4, sketches (5) through (11), as well as Figure UW-13.2 sketched (a) and (b). The requirements of 2-4(c) apply to Figure 2-4, sketches (8) through (11) as well as Figure UW-13.2 sketches (a) and (b).

(2) Flanges with the configuration as shown in Figure 2-13.2 shall be designed as loose ring reverse flanges and those in Figure 2-13.3 shall be designed as loose ring reverse flanges. These flanges shall be designed in conformance with the rules in 2-3 through 2-8 and the modifications as described in the following. Mandatory use of these rules is limited to $K \leq 2$. When $K > 2$, results become increasingly conservative and U-2(g) may be used.

(3) Shell-reversing flanges shall be designed as integral reverse flanges and those in Figure 2-13.4 shall be designed as loose ring reverse flanges. These flanges shall be designed in conformance with the rules in 2-3 through 2-8, but with the modifications as described in the following. Mandatory use of these rules is limited to $K \leq 2$. When $K > 2$, results become increasingly conservative and U-2(g) may be used.

(4) When flanges are subject to pressure reversals, the design shall satisfy the requirements given in 2-11(a) for external pressure and bolt loading may plastically deform certain gaskets to result in loss of gasket contact pressure when the connection is depressurized. To maintain a tight joint when the unit is repressurized, consideration should be given to gasket and facing details so that excessive deformation of the gasket will not occur. Joints subject to pressure reversals, such as in heat exchanger floating heads, are in this type of service.

Add Endnote:
When internal pressure occurs only during the required pressure test, the design may be based on external pressure, and auxiliary devices such as clamps may be used during the application of the required test pressure.

Table 2-7.1
Flange Factors in Formula Form (Cont’d)

<table>
<thead>
<tr>
<th>Equations (Cont’d)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(41) $E_2 = C_{26} C_{36} + C_{27} + C_{25} C_{37}$</td>
</tr>
<tr>
<td>(42) $E_2 = C_{26} C_{36} + C_{24} + C_{25} C_{37}$</td>
</tr>
<tr>
<td>(43) $E_2 = 1/4 + C_{35}/12 + C_{36}/4 - E_{2}/5 - 3E_{2}/2 - E_1$</td>
</tr>
<tr>
<td>(44) $E_2 = E_{2}(1/2 + A/6) + E_{2}(1/2 + 11A/84) + E_{2}(1/70 + A/105)$</td>
</tr>
<tr>
<td>(45) $E_2 = E_{2} - C_{36}/120 + A/36 + 3A/C) - 1/40 - A/60 + A/120 + 1/C$</td>
</tr>
</tbody>
</table>

NOTE: The combined force of external pressure and bolt loading may plastically deform certain gaskets to result in loss of gasket contact pressure when the connection is depressurized. To maintain a tight joint when the unit is repressurized, consideration should be given to gasket and facing details so that excessive deformation of the gasket will not occur. Joints subject to pressure reversals, such as in heat exchanger floating heads, are in this type of service.

other recognized and generally accepted methods, such as those found in other ASME, EN, ISO, national and industry standards or codes may be used. This option shall provide details of design consistent with the allowable stress criteria provided in TD-210.
(2) Loose Ring Type Reverse Flange. The shell-to-flange attachment of loose ring type reverse flanges may be attached as shown in Figure 2-4, sketches (3a), (4a), (8), (9), (10), and (11) as well as Figure TW-130.5-2, sketches (c) and (d). When Figure TW-130.5-2, sketches (c) and (d) are used, the maximum wall thickness of the shell shall not exceed 10 mm (3/8 in.), and the maximum design metal temperature shall not exceed 340°C (650°F).

The symbols and definitions in this paragraph pertain specifically to reverse flanges. Except as noted in (b) below, the symbols used in the equations of this paragraph are defined in 2-3.

The equations for $S_R$, $S_H$, and $S_T$ correspond, respectively, to eqs. 2-7(a)(8), 2-7(a)(9), and 2-7(a)(10), in direction, but are located at the flange outside diameter. The sole stress at the flange inside diameter is a tangential stress and is given by the formula for $S_{T2}$.

(b) Notation

- $B = \text{inside diameter of shell}$
- $B' = \text{inside diameter of reverse flange}$
- $d_r = U_r h_{or} g_o / V$
- $e_r = F / h_{or}$
- $F = \text{factor (use } h_{or} \text{ for } h_o \text{ in Figure 2-7.2)}$
- $f = \text{factor (use } h_{or} \text{ for } h_o \text{ in Figure 2-7.6)}$
- $H = \text{total hydrostatic end force on attached component} = 0.785 G^2 P$
- $H_D = \text{hydrostatic end force on area inside of flange} = 0.785 B^2 P$
- $H_T = \text{difference between hydrostatic end force on attached component and hydrostatic end force on area inside of flange} = H - H_D$
- $h_D = \text{radial distance from the bolt circle to the circle on which } H_D \text{ acts} = (C + g_1 - 2g_o - B) / 2$ for integral type reverse flanges
- $h_{or} = \text{factor} = \sqrt[4]{A g_o}$
\( h_T \) = radial distance from the bolt circle, to the circle on which \( H_T \) acts
\[
\frac{1}{2} \left( C - \frac{B + G}{2} \right)
\]
\( K \) = ratio of outside diameter of flange to inside diameter of flange
\[
A/B'
\]
\( L_r \) = factor
\[
\frac{te_r + 1}{T_r} + \frac{t^3}{d_r}
\]
\( M_o \) = total moment acting on the flange, for the operating conditions or gasket seating as may apply
\( M_D, M_T, \) and \( M_G \): Values of load \( H_T \) and moment arm \( h_o \) are negative; value of moment arm \( h_T \) may be positive as in Figure 2-13.1, or negative. If \( M_o \) is negative, use its absolute value in calculating stresses to obtain positive stresses for comparison with allowable stresses.
\[
\alpha_r = \frac{0.668(K + 1)}{Y} / K^2
\]

(c) For Integral Type Reverse Flanges

(1) Stresses at the Outside Diameter

\[
S_H = f M_o / L_r g_1 B'
\]
\[
S_R = (1.33te_r + 1) M_o / L_r t^2 B'
\]
\[
S_T1 = (Y, M_o / t^2 B') - Zsp[0.67te_r + 1] / (1.33te_r + 1)
\]

(2) Stress at Inside Diameter \( B' \)

\[
S_T2 = \left( M_o / t^2 B' \right) \left( Y - \frac{2K^2(1 + \frac{1}{2}te_r)}{(K^2 - 1)T_r} \right)
\]

(d) For Loose Ring Type Reverse Flanges

\[
S_T = YM_o / t^2 B'
\]
\[
S_R = 0
\]
\[
S_H = 0
\]
2-14 FLANGE RIGIDITY

(a) Flanges that have been designed based on allowable stress limits alone may not be sufficiently rigid to control leakage. This paragraph provides a method of checking flange rigidity. The rigidity factors provided in Table 2-14 have been proven through extensive user experience for a wide variety of joint design and service conditions. The use of the rigidity index does not guarantee a leakage rate within established limits. The use of the factors must be considered as only part of the system of joint design and assembly requirements to ensure leak tightness. Successful service experience may be used as an alternative to the flange rigidity rules for fluid services that are non-lethal and nonflammable and designed within the temperature range of −20°F (−29°C) to 366°F (186°C) without exceeding design pressures of 150 psi (1035 kPa).

(b) The notation is as follows:

- $E$ = modulus of elasticity for the flange material at design temperature (operating condition) or at atmospheric temperature (gasket seating condition), psi
- $J$ = rigidity index $\leq 1$
- $K_I$ = rigidity factor for integral or optional flange types $= 0.3$
- $K_L$ = rigidity factor for loose-type flanges $= 0.2$

Experience has indicated that $K_I$ and $K_L$ provided above are sufficient for most services; other values may be used with the User’s agreement.

Other notation is defined in 2-3 for flanges and 2-13 for reverse flanges.

(c) The rigidity criterion for an integral type flange and for a loose type flange without a hub is applicable to the reverse flanges in Figures 2-13.1 and 2-13.2, respectively. The values of $h_o$ or shall be substituted for $h_o$, and the value $L_0$ shall be substituted for the value $L$ in the rigidity equation for integral type flanges. Also substitute $h_o$ for $h_o$, in determining the factor $V$ in the equation for integral type flanges.

(d) If the value of $J$, when calculated by the appropriate formula above, is greater than 1.0, the thickness of the flange, $t$, shall be increased and $J$ recalculated until $J \leq 1$ for both gasket seating and operating conditions.

2-15 QUALIFICATION OF ASSEMBLY PROCEDURES AND ASSEMBLERS

It is recommended that flange joints designed to this Appendix be assembled by qualified procedures and by qualified assemblers. ASME PCC-1 may be used as a guide.

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Table 2-14
Flange Rigidity Factors

<table>
<thead>
<tr>
<th>Flange Type</th>
<th>Rigidity Criterion</th>
</tr>
</thead>
<tbody>
<tr>
<td>Integral-type flanges and optional type flanges designed as integral-type flanges</td>
<td>$J = \frac{52.14VM_0}{LEb_0^kK_h_0} \leq 1.0$</td>
</tr>
<tr>
<td>Loose-type flanges with hubs</td>
<td>$J = \frac{52.14VM_0}{LEb_0^kK_Lh_0} \leq 1.0$</td>
</tr>
<tr>
<td>Loose-type flanges without hubs and optional flanges designed as loose-type flanges</td>
<td>$J = \frac{109.4M_0}{Et^2K_L\ln K} \leq 1.0$</td>
</tr>
</tbody>
</table>