NONMANDATORY APPENDIX S
DESIGN CONSIDERATIONS FOR BOLTED FLANGE CONNECTIONS

S-1 BOLTING

The primary purpose of the rules for bolted flange connections in Mandatory Appendix 2 and Nonmandatory Appendix Y is to ensure safety, but there are certain practical matters to be taken into consideration in order to obtain a serviceable design. One of the most important of these is the proportioning of the bolting, i.e., determining the number and size of the bolts.

In the great majority of designs the practice that has been used in the past should be adequate, viz., to follow the design rules in Mandatory Appendix 2 and Nonmandatory Appendix Y and tighten the bolts sufficiently to withstand the test pressure without leakage. The considerations presented in the following discussion will be important only when some unusual feature exists, such as a very large diameter, a high design pressure, a high temperature, severe temperature gradients, an unusual gasket arrangement, and so on.

The maximum allowable stress values for bolting given in Section II, Part D, Subpart 1, Table 3 are design values to be used in determining the minimum amount of bolting required under the rules. However, a distinction must be kept carefully in mind between the design value and the bolt stress that might actually exist or that might be needed for conditions other than the design pressure. The initial tightening of the bolts is a prestressing operation, and the amount of bolt stress developed must be within proper limits, to ensure, on the one hand, that it is adequate to provide against all conditions that tend to produce a leaking joint, and on the other hand, that it is not so excessive that yielding of the bolts and/or flanges can produce relaxation that also can result in leakage. Remember that the torque to overcome thread friction plus the bolt-head or nut-turning friction constitutes approximately 90% of the total applied torque, leaving only 10% to stretch the bolt. This is true even under the desired contact surface and lubrication conditions; thus the need to be mindful of these considerations during joint assembly (see ASME PCC-1, sections 4 and 7). The use of through-hardened washers may be appropriate (see ASME PCC-1, Appendix M).

The first important consideration is the need for the joint to be tight in the hydrostatic test. Therefore, an initial bolt stress of some magnitude greater than the design value must be provided. If it is not, further bolt strain develops during the test, which tends to part the joint and thereby to decompress the gasket enough to allow leakage. It may be thought that 50% extra bolt stress above the design value will be sufficient. However, this is an oversimplification because, on the one hand, the safety factor against leakage under test conditions in general need not be as great as under operating conditions. On the other hand, if a stress-strain analysis of the joint is made, it may indicate that an initial bolt stress still higher than $1\frac{1}{2}$ times the design value is needed. Such an analysis is one that considers the changes in bolt elongation, flange deflection, and gasket load that take place with the application of internal pressure, starting from the prestressed condition. In any event, it is evident that an initial bolt stress higher than the design value may, and in most cases, must, be developed in the tightening operation, and it is the intent of this Division that such a practice is permissible, provided it includes necessary and appropriate provision to ensure against excessive flange distortion and gross crushing of the gasket. See ASME PCC-1, section 12, Target Torque Determination. For guidance on troubleshooting flange joint leakage incidents, see ASME PCC-1, Appendix P.

It is possible for the bolt stress to decrease after initial tightening, because of initial slow creep or relaxation of the gasket, particularly in the case of the “soft” gasket materials (see ASME PCC-1, Appendix D) and/or when the effective stretching length of the bolt is short (see ASME PCC-1, footnote 6). This may be the cause of leakage in the hydrostatic test, in which case it may suffice merely to retighten the bolts. A decrease in bolt stress can also occur in service at elevated temperatures, as a result of creep in the bolt and/or flange or gasket material, with consequent relaxation. When this results in leakage under service conditions, it is common practice to retighten the bolts, and sometimes a single such operation, or perhaps several repeated at long intervals, is sufficient to correct the condition. To avoid chronic difficulties of this nature, however, it is advisable when designing a joint for high temperature service to give attention to the relaxation properties of the materials involved, especially for temperatures where creep is the controlling factor in design. This prestress should not be the controlling factor in design. This prestress should not be confused with initial bolt stress used in the design of Nonmandatory Appendix Y flanges (i.e., flat-face flanges with metal-to-metal contact outside the bolt circle).

In the other direction, excessive initial bolt stress can present a problem in the form of yielding in the bolting itself, and may occur in the tightening operation to the
Radial Stress in Flange II at Bolt Circle

\[ S_{RII} = \frac{6(M_p + M_{eII})}{t_{II}^2(\pi C - nD)} \]  

(35)

Radial Stress in Flange II at Diameter B1

\[ S_{RII} = \frac{6M_{eII}}{\pi B_1 t_{II}^2} \]  

(36)

Tangential Stress in Flange II at Diameter B1

\[ S_{TII} = \frac{t_{II} E_{II} \theta_{BII}}{B_1} - \frac{1.8M_{eII}}{\pi B_1 t_{II}^2} \]  

(37)

Radial and Tangential Stress at Center of Flange II

\[ S_{RII} = S_{TII} = \frac{0.3094PB_1^2}{t_{II}^2} - \frac{6M_{eII}}{\pi B_1 t_{II}^2} \]  

(38)

(b) The thickness of Flange II of a Class 3 assembly determined by the above rules shall be used in lieu of the thickness that is determined by UG-34. However, any centrally located opening in Flange II shall be reinforced to meet the rules of UG-39(b).

Y-9 ESTIMATING FLANGE THICKNESSES AND BOLTING

(a) The following simple equations are offered for calculating approximate values of \( t, t_0, t_{II}, \) and \( A_b \) before applying the rules in Y-4 through Y-8. The equations are not intended to replace the rules; however, they should significantly reduce the amount of work required to achieve a suitable design. Since the flanges are in metal-to-metal contact and interact, the stresses in one flange are influenced by the stiffness of the mating flange and theoretically an unlimited number of designs can be found which satisfy the rules. In practice, however, economics, engineering judgment, and dimensional constraints will show which is the "best" design. It should be noted that the equations in Table Y-9.1 assume that both flanges comprising an assembly have essentially the same modulus of elasticity and allowable stress.

(b) Equations for Trial Flange Thickness and Bolting

\[ t_a = 2.45 \sqrt{\frac{M_p}{(\pi C - nD)S_f}} \]  

(39)

\[ t_b = 0.56B_1 \sqrt{P/S_f} \]  

(40)

\[ t_c = \text{greater of } t_a \text{ or } t_b \]
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S-1 BOLTING

The primary purpose of the rules for bolted flange connections in Appendices 2 and Y is to ensure safety, but there are certain practical matters to be taken into consideration in order to obtain a serviceable design. One of the most important of these is the proportioning of the bolting, i.e., determining the number and size of the bolts.

In the great majority of designs the practice that has been used in the past should be adequate, viz., to follow the design rules in Appendices 2 and Y and tighten the bolts sufficiently to withstand the test pressure without leakage. The considerations presented in the following discussion will be important only when some unusual feature exists, such as a very large diameter, a high design pressure, a high temperature, severe temperature gradients, an unusual gasket arrangement, and so on.

The maximum allowable stress values for bolting given in Table 3 of Section II, Part D are design values to be used in determining the minimum amount of bolting required under the rules. However, a distinction must be kept carefully in mind between the design value and the bolt stress that might actually exist or that might be needed for conditions other than the design pressure. The initial tightening of the bolts is a prestressing operation, and the amount of bolt stress developed must be within proper limits, to insure, on the one hand, that it is adequate to provide against all conditions that tend to produce a leaking joint, and on the other hand, that it is not so excessive that yielding of the bolts and/or flanges can produce relaxation that also can result in leakage.

The first important consideration is the need for the joint to be tight in the hydrostatic test. An initial bolt stress of some magnitude greater than the design value therefore must be provided. If it is not, further bolt strain develops during the test, which tends to part the joint and thereby decompress the gasket enough to allow leakage. The test pressure is usually 1.5 times the design pressure, and on this basis it may be thought that 50% extra bolt stress above the design value will be sufficient. However, this is an oversimplification because, on the one hand, the safety factor against leakage under test conditions in general need not be as great as under operating conditions. On the other hand, if a stress-strain analysis of the joint is made, it may indicate that an initial bolt stress still higher than 1.5 times the design value is needed. Such an analysis is one that considers the changes in bolt elongation, flange deflection, and gasket load that take place with the application of internal pressure, starting from the prestressed condition. In any event, it is evident that an initial bolt stress higher than the design value may and, in some cases, must be developed in the tightening operation, and it is the intent of this Division that such a practice is permissible, provided it includes necessary and appropriate provision to insure against excessive flange distortion and gross crushing of the gasket.

It is possible for the bolt stress to decrease after initial tightening, because of slow creep or relaxation of the gasket, particularly in the case of the "softer" gasket materials. This may be the cause of leakage in the hydrostatic test, in which case it may suffice merely to retighten the bolts. A decrease in bolt stress can also occur in service at elevated temperatures, as a result of creep in the bolt and/or flange or gasket material, with consequent relaxation. When this results in leakage under service conditions, it is common practice to retighten the bolts, and sometimes a single such operation, or perhaps several repeated at long intervals, is sufficient to correct the condition. To avoid chronic difficulties of this nature, however, it is advisable when designing a joint for high temperature service to give attention to the relaxation properties of the materials involved, especially for temperatures where creep is the controlling factor in design. This prestress should not be the controlling factor in design. This prestress should not be confused with initial bolt stress $S_i$ used in the design of Appendix Y flanges.

In the other direction, excessive initial bolt stress can present a problem in the form of yielding in the bolting itself, and may occur in the tightening operation to the extent of damage or even breakage. This is especially likely with bolts of small diameter and with bolt materials having a relatively low yield strength. The yield strength of mild carbon steel, annealed austenitic stainless steel, and certain of the nonferrous bolting materials can easily be exceeded.
(or flat circular head) of a Class 3 assembly:

Rigid Body Rotation of Flanges Times \( E^* \)

\[
E^* \theta_{b1} = \frac{X(C_1 - C_2)}{1.206 \log \left( \frac{A}{B_1} \right) - XC_3 - (1 - X)C_3}
\]

\[
E^* \theta_{bII} = -E^* \theta_{bII}(E^*/E^*)
\]

Total Flange Moment at Diameter \( B_1 \)

\[
M_{s1} = C_1(E^* \theta_{b1}) + C_4
\]

\[
M_{sII} = C_1(E^* \theta_{bII}) + C_2
\]

Unbalanced Flange Moment at Diameter \( B_1 \)

\[
M_{s1} = 1.206 E^* \theta_{b1} \log \left( \frac{A}{B_1} \right)
\]

\[
M_{sII} = 1.206 E^* \theta_{bII} \log \left( \frac{A}{B_1} \right)
\]

Balanced Flange Moment at Diameter \( B_1 \)

\[
M_{b1} = M_{s1} - M_{sII}
\]

\[
M_{bII} = M_{sII} - M_{s1}
\]

Slope of Flange at Diameter \( B_1 \) Times \( E \)

\[
E_1 \theta_{b1} = \frac{5.46}{\pi a^2} (J_3 M_{b1} + J_p M_p) + E^* \theta_{b1}/t_1^3
\]

\[
E_1 \theta_{bII} = -1.337(M_{sII} - \pi PB_1^2/32)/t_1^3
\]

Contact Force Between Flanges at \( h_C \)

\[
H_C = (M_p + M_{b1})/h_C
\]

Bolt Load at Operating Conditions

\[
W_{nl} = H + H_G + H_C
\]

Operating Bolt Stress

\[
\sigma_p = W_{nl}/A_b
\]

Design Prestress in Bolts

\[
S_1 = \sigma_p - 1.159 h_C^2 (M_p + M_{b1})/2(1 - X)a t_1^2 l r_{b1} B_1
\]

Radial Stress in Flange I at Bolt Circle

\[
S_{R1} = \frac{6(M_p + M_{s1})}{t_1^2 (\pi C - nD)}
\]

Radial Stress in Flange I at Inside Diameter

\[
S_{RI} = \left( \frac{2F_{T1}}{h_0 + F_{TL}} + 6 \right) \frac{M_{s1}}{\pi B_1 t_1^2}
\]

\[
S_{RII} = \left( \frac{2F_{T1} l_1}{h_0 + F_{TL}} + 6 \right) \frac{M_{sII}}{\pi B_1 t_1^2}
\]

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Tangential Stress in Flange I at Inside Diameter

\[
S_{TI} = \frac{t_1 E_1 \theta_{b1}}{B_1} + \left( \frac{2F_{T1} Z}{h_0 + F_{TL} - 1.8} \right) \frac{M_{sII}}{\pi B_1 t_1^2}
\]

Longitudinal Hub Stress in Flange I

\[
S_{HI} = \frac{h_0 E_1 \theta_{bII}}{0.91(g_1/g_0)^3 B_1 V}
\]

Radial Stress in Flange II at Bolt Circle

\[
S_{RII} = \frac{6(M_p + M_{sII})}{t_1^2 (\pi C - nD)}
\]

Radial Stress in Flange II at Diameter \( B_1 \)

\[
S_{RII} = \frac{6 M_{sII}}{\pi B_1 t_1^2}
\]

Tangential Stress in Flange II at Diameter \( B_1 \)

\[
S_{TII} = \frac{t_1 E_1 \theta_{bII}}{B_1} + \left( \frac{1.8 M_{sII}}{\pi B_1 t_1^2} \right)
\]

Radial and Tangential Stress at Center of Flange II

\[
S_{RII} = S_{TII} = \frac{0.3094 PB_1^2}{t_1^2} - \frac{2 M_{sII}}{\pi B_1 t_1^2}
\]

(b) The thickness of Flange II of a Class 3 assembly determined by the above rules shall be used in lieu of the thickness that is determined by UG-34. However, any centrally located opening in Flange II shall be reinforced to meet the rules of UG-39(b).

Y-7 ALLOWABLE FLANGE DESIGN STRESSES

The stresses calculated by the above equations, whether tensile or compressive (–), shall not exceed the following values for all classes of assemblies:

(a) operating bolt stress \( \sigma_p \) not greater than \( S_b \) for the design value of \( S_1 \);  

(b) The symbols for the various stresses in the case of a Class 3 assembly also carry the subscript I or II. For example, \( S_{RI} \) represents the longitudinal hub stress in Flange I of the Class 3 assembly.