High Pressure Flange Design

Kermit Frederick Allerman

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HIGH PRESSURE FLANGE DESIGN

BY

KERMIT FREDERICK ALLERMAN

A Research Report Presented in Partial Fulfillment of the Requirements for the Degree Master of Science in Engineering

FLORIDA TECHNOLOGICAL UNIVERSITY

August 1972
Abstract of research report presented to Florida Technological University in partial fulfillment of the requirements for the degree of Master of Science

HIGH PRESSURE FLANGE DESIGN

By
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Assistant Design Project Engineer
Pratt & Whitney Aircraft
August, 1972

This research report summarizes high pressure flange design techniques and considerations utilized by Pratt & Whitney Aircraft, Florida Research & Development Center, during its development programs for high pressure liquid rocket engines. The report covers:

1) Aspects of cooling, heating, pressure, and external loading with design safety factors
2) Cantilever type flanges optimized for weight
3) Seals and fastener considerations for 6000 psi environment
4) An example high pressure, cryogenic cantilever flange design
ACKNOWLEDGEMENT

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K. F. Allerman
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<tr>
<td>$A$</td>
<td>Area, in.$^2$</td>
</tr>
<tr>
<td>$a$</td>
<td>Acceleration, ft. / sec.$^2$</td>
</tr>
<tr>
<td>$c$</td>
<td>Extreme outer fiber, in.</td>
</tr>
<tr>
<td>$D$</td>
<td>Diameter, in.</td>
</tr>
<tr>
<td>$E$</td>
<td>Material modulus of elasticity, psi</td>
</tr>
<tr>
<td>$F$</td>
<td>Load, force, lbs.</td>
</tr>
<tr>
<td>$I$</td>
<td>Area moment of inertia, in.$^4$</td>
</tr>
<tr>
<td>$i$</td>
<td>Individual effect of several similar terms</td>
</tr>
<tr>
<td>$K$</td>
<td>Spring rate, lbs. / in.</td>
</tr>
<tr>
<td>$L$</td>
<td>Length, in.</td>
</tr>
<tr>
<td>$M$</td>
<td>Moment, lb. in.</td>
</tr>
<tr>
<td>$N$</td>
<td>Number, quantity of</td>
</tr>
<tr>
<td>$P$</td>
<td>Force, lb., due to pressure and / or load</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure, psi</td>
</tr>
<tr>
<td>$R$</td>
<td>Radius, in.</td>
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<tr>
<td>$S$</td>
<td>Stress, psi</td>
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<tr>
<td>$T$</td>
<td>Temperature, °Rankin</td>
</tr>
<tr>
<td>$B$</td>
<td>Subscript signifying bolt</td>
</tr>
<tr>
<td>$b$</td>
<td>Subscript signifying bending</td>
</tr>
<tr>
<td>$C$</td>
<td>Subscript signifying cooling</td>
</tr>
<tr>
<td>$e$</td>
<td>Subscript signifying effective stress</td>
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f = Subscript signifying flange
H = Subscript signifying heating effect
h = Subscript signifying hoop direction
l = Subscript signifying limiting condition
s = Subscript signifying shear force
t = Subscript signifying tension
v = Subscript signifying vibration effect
x = Subscript signifying axial direction
BC = Subscript signifying bolt circle
brg = Subscript signifying bearing
ch = Subscript signifying chordal direction
mm = Subscript signifying minimum minor thread diameter
\Delta \varepsilon = Material mean coefficient of thermal expansion
\Delta = A change in quantity
\Sigma = Summation of like parameters
\Theta = Angular rotation, radians
DP1 = Design point number one
DP2 = Design point number two
FIR = Full indicator reading
max = Maximum condition
min = Minimum condition
SP = Seal effective contact point, location
Y.S. = Material 0.2% yield strength, psi
U.T.S. = Material ultimate tensile strength, psi
( )' = Parameter evaluated on a per inch length basis
( )° = Degrees, temperature or rotational
ccc / sec = Standard cubic centimeters per second
0, 1, 2, etc = Time or operating points

Special nomenclature unique to computer programming is listed in the Appendix A.6, Calculation 12, and is depicted in Figure 2.6.
SECTION 1

FUNDAMENTAL FLANGE DESIGN CONSIDERATIONS

1.1 Loading of the Flange

The bolt loading must be sufficient to prevent flange separation, i.e., excessive deflection at the seal contact circle. The weight optimized flange utilizes the minimum number of the smallest size bolts that will support all loads without flange separation. The real loads which must be resisted are:

1) Thermal transient effects due to heating and cooling
2) Internal pressure effects
3) Externally imposed loads resulting from
   a) Supported structure under acceleration and vibration
   b) Mating structure assembly mismatch
4) Seal load
5) Bolt preload variation effects at installation

Imaginary loads established by desired safety factors or special load requirements must also be considered.

Explanation of the fundamental considerations in flange design can be expanded around a Typical Flange Load Map (Fig. 1.1). Figure 1.1 shows representative load curves for both conventional "L" type (unprimed points) and cantilever type (primed points). A conventional
Typical Flange Load Map

Showing Effects of Temperature, Pressure, and External Loading on Bolt Load Vs. Points of Operation Occuring With Time

\[ \Delta F_{BH} \text{ gain, bolts lag flange, heating} \]

\[ \Delta F_{BP} = \Delta F_{BP} + \Delta F_{BF} \]

Both flange and bolts at cooldown temperature

Gain due to pressure and external loading,

\[ \Delta F_{BC} \text{ loss, bolts lag flange cooling} \]

Partial to full recovery of \( \Delta F_{BC} \)

Design points (DP2) for max bolt load during operation, \( F_{BDP2} \)

Partial to full relaxation of \( \Delta F_{BH} \)

"L" Type Flange

Cantilever Flange

Design points (DP1) for no flange separation, \( F_{BDP1} \)

Loss due to flange rotation from pressure and external loading,

\[ \Delta F_{BP} = \Delta F_{B\theta} + \Delta F_{B\theta} = \Delta F_{B\theta} \]

Cooldown

Soak Equilibrium

Pressure Build-Up

Warm-Up

Steady State

Operational Time Points

FIGURE 1.1
type flange is shown in Figure 1.2 and Figure 1.5 shows a cantilever type.

1.2 Flange Cooling

For the time spans from 0 - 1 and 0 - 1' (Fig. 1.1), typical cooling situations applicable to the "L" type and cantilever type flanges respectively are shown. The flange may be in direct contact with a cryogenic fluid which thermally contracts the flange (Fig. 1.2). The bolts, shielded by the flanges, experience a time lag in sensing the change in temperature and therefore lag the flange in contraction. This condition causes a thermal relaxation loss in the bolt load.

Consider a situation in which the forces tending to separate an "L" type flange occur at time point 1 and the bolts lag the flange thermally. At installation, both the flange and the bolts are tight at length $L_0$ (Fig. 1.2). During cooling, the bolt tends to contract to length $L'_B$ and the flange similarly to $L'_f$. For the bolt to retain a desired load ($F'_B$) with the flange after cooling at some common length $L_1$, the room temperature installation bolt load ($F_{B_{RT}}$) must differ from $F'_B$ by an amount $F_{B_{0-1}}$ in order to compensate for the effects of thermal change (Fig. 1.3).

Bolt Change:

$$L_0 - \frac{F_{B_{0-1}} L_B}{A_B E_B} - \alpha_B L_B (T_1 - T_0) = L_1$$
FIGURE 1.2 - Conventional "L" Type Flange in a Cooling Situation
Typical Bolt Load Effect

$$\frac{F_{B0-1}}{A_f E_f} L_f$$

$$L_f = L_0$$

$$L_1$$

$$(\Delta T)_f$$

$$(\Delta T)_B$$

Typical Thermal Effect

$$L_B = L_0$$

$$L_1 = L_B$$

$$F_{B0-1} L_B$$

$$A_B E_B$$

FIGURE 1.3 - Dimensional Changes for Cooled Flanges
Flange Change:

\[ L_0 + \frac{F_{B_{0-1}L_f}}{A_t E_f} - \gamma L_f(T_1-T_0)_f = L_f \]

Equating \( L_1 \):

\[ \frac{F_{B_{0-1}L_B}}{A_{BE_B}} + \frac{F_{B_{0-1}L_f}}{A_t E_f} = \gamma L_f(T_1-T_0)_f - \gamma L_B(T_1-T_0)_B \]

Defining spring rates \( K_f \equiv \frac{A_t E_f}{L_f} \) and \( K_B \equiv \frac{A_{BE_B}}{L_B} \) for the flange and bolts respectively as discussed in Section 6.1:

\[ F_{B_{0-1}} = \left[ \gamma L_f(T_1-T_0)_f - \gamma L_B(T_1-T_0)_B \right] \frac{K_f K_B}{K_f + K_B} \]

The equation for bolt load change due to flange cooling effect during time span \( 0 - 1 \) can therefore be written as:

\[ \Delta F_{BC} = \frac{K_f K_B}{K_f + K_B} \]

1.2.1

\[ \Delta F_{BC} = \left[ \gamma L_f(T_1-T_0)_f - \gamma L_B(T_1-T_0)_B \right] \frac{K_f K_B}{K_f + K_B} \]

Where \( \gamma \) is the mean coefficient of expansion between \( T_1 \) and the room temperature \( (T_0) \), and where \( \Delta L_f \) and \( \Delta L_B \) are the length changes of the flange and bolts respectively due to temperature change.

Thus, to maintain a desired load \( F_{B_{1}} \) for no flange separation, the bolts must be tightened initially at room temperature to a load of:
Depending upon the equilibrium temperatures and the relative coefficients of expansion, the bolts may or may not regain their room temperature initial preload \( F_{BRT} \) during the time span 1-2 (Fig. 1.1). For this case, the flange has reached its coldest temperature while the bolts continue to cool down and tend to regain the load lost during the cooling from time span 0-1. As for Equation 1.2.1 except for time span 0-2:

\[
\Delta F_{BC}^{0-2} = \left[ \Delta L_f - \Delta L_B \right]^{0-2} \frac{K_f K_B}{K_f + K_B}
\]

Combining with Equation 1.2.1 to determine the net thermal bolt load change for time span 1-2 gives:

\[
\Delta F_{BC}^{1-2} = \Delta F_{BC}^{0-2} - \Delta F_{BC}^{0-1}
\]

Equations 1.2.1 and 1.2.2 also apply to cantilever flanges for the respective time spans 0-1' and 1'-2'.

1.3 Effects of Pressure

The effects of internal pressure applied to a line flange is considered during the time span 3-4 shown in Figure 1.1. The pressure, upon application, increases or decreases bolt load, the change being dependent upon the type of flange used. A bolt load gain is experienced
in the conventional "L" type flanges; in the cantilever type flanges, there is a bolt load loss due to flange rotation. Detailed explanations of these effects are given in Section 2.3 and the Appendix.

For "L" type flanges, the bolt load increase resulting when the application of internal pressure occurs is given by (1)*:

\[
\Delta F_{BP} = \frac{pA_{SP}}{N_B} \left[ \frac{K_B}{K_B + K_f} \right] \quad \text{(See Appendix A.1.1)}
\]

Where \( p \) = Internal pressure within the flange
\( A_{SP} \) = Area of the seal point diameter to which the \( p \) acts
\( N_B \) = Number of bolts
\( K_B, K_f \) = Bolt and flange spring rates respectively as discussed in Sections 2.8 and 6.1

\( \Delta F_{BP} \) is included in the unprimed points of time span 3-4 shown in Figure 1.1 and is part of the combined load term \( \Delta F_{BP} \) of Section 1.5 which considers the bolt load change due to applied pressure and applied external loads.

For cantilever type flanges, the bolt load decrease resulting from the application of internal pressure is given by:

\[
\Delta F_{BP} = \Delta F_{BP\theta} \quad \text{(See Appendix A.2.1)}
\]

\[
\Delta F_{BP\theta} = \left[ \Delta L_{f\theta} - \Delta L_{B\theta} \right]_p \frac{K_B K_f}{K_B + K_f}
\]

* Indicates reference number listed in Section 9.
Where the designation $\Delta F_{B\theta \rho}$ is used to avoid confusion with the similar term for "L" type flanges.

$\Delta F_{B\theta \rho}$ is included in the primed points of time span 3' - 4' shown in Figure 1.1 and is part of the combined load term $\Delta F_{B\theta}$ of Section 1.5 which considers the bolt load change due to applied pressure and applied external loads. The spring rates are as previously defined while the $\Delta L$ terms are the change in length of the flange and bolts due to flange rotation occurring when pressure is applied.

1.4 Flange Heating

For the time span 4-5 (Fig. 1.1) a typical heating situation is shown. In this case at time point 4, the flange of Figure 1.2 is assumed to be cryogenically cold on the inside with the surrounding environment getting hotter from a heating operation, radiation, etc. The bolts, shielded within the flange, thermally expand under a smaller thermal gradient than the flange, thus giving a thermal change effect reversed from a flange cooling situation. The same is true for cantilever flanges, time span 4'-5'.

Figure 1.4 shows typical dimensional changes occurring in heated flanges. Following a procedure similar to that which derived equation 1.2.1, thermal heating effects for the time span 0-4 give:

\[
\Delta F_{B_H}^{0-4} = \left[ \alpha_f L_f (T_4 - T_0) \right]_f - \alpha_B L_B (T_4 - T_0)_B \right] \frac{K_f K_B}{K_f + K_B}
\]
Typical Thermal Effect

\[ \alpha_f L_f (\Delta T)_f \]

\[ \frac{F_{B4-5}}{A_f E_f} \]

\[ L_f = L_4 \]

\[ L_5 \]

Flange

Typical Bolt Load Effect

\[ \alpha_B L_3 (\Delta T)_B \]

\[ \frac{F_{B4-5}}{A_B E_B} \]

\[ L_B = L_4 \]

\[ L_5 \]

Bolt

FIGURE 1.4 - Dimensional Changes for Heated Flanges
FIGURE 1.5 - A Typical Cantilever Type Flange
Or: \[ \Delta F_{B_{H}}^{0-4} = \left[ \Delta L_f - \Delta L_B \right]_{0-4} \frac{K_fK_B}{K_f + K_B} \]

Likewise: \[ \Delta F_{B_{H}}^{0-5} = \left[ \Delta L_f - \Delta L_B \right]_{0-5} \frac{K_fK_B}{K_f + K_B} \]

Where \( \alpha \) is the mean coefficient of expansion between a given temperature (as \( T_4 \) or \( T_5 \)) and room temperature (\( T_0 \)).

Combining terms, the equation for bolt load gain due to flange heating effect during the time span 4-5 is:

1.4.2 \[ \Delta F_{B_{H}} = \Delta F_{B_{H}}^{4-5} - \Delta F_{B_{H}}^{0-5} - \Delta F_{B_{H}}^{0-4} \]

The same equations apply to the cantilever flange, time span 4'-5' (Fig. 1.1).

1.5 Effects of External Loading

The summation of external axial loads \( F_x \) which may load the flange joint is given by:

\[ F_x = \sum F_{x_{\text{air}}} \quad \text{-- flange portion of supported structure under acceleration from maneuvering, ground handling, etc.} \]

\[ \sum F_{x_{\text{vib}}} \quad \text{-- flange portion of supported structure under vibration from maneuvering, ground handling, etc.} \]

\[ \sum F_{x_{\text{assy}}} \quad \text{-- flange portion of mating structure loads from mismatch of interfaces at assembly} \]
Likewise, the summation of external axial moments (which resolve into an $F_x$ effect) is given by:

$$M_x = \sum M_{x_{B_1}} + \sum M_{x_{V_1}} + \sum M_{x_{ass}}$$

Considering the maximum resultant combination of external loading and the associated spring rate effect, the bolt load changes for "L" type and cantilever type flanges are as follows.

For "L" type flanges externally loaded, the bolt load increase resulting when the application of external loading occurs is similar to equation 1.3.1 and is:

\[ 1.5.1 \quad \Delta F_{BP} = \left( \frac{F_x}{N_B} + \frac{2M_x}{NB_{BC}} \right) \frac{K_B}{K_B + K_F} \]  

(See Appendix A.1.2 and A.3)

Where $\Delta F_{BF}$ is the external loading portion of term $\Delta F_{BP}$ (Fig. 1.1, time span 3-4) which considers both pressure and external loading effects as one term $\Delta F_{BP} = \Delta F_{BP} + \Delta F_{BF}$.

For cantilever type flanges externally loaded, the bolt load loss ($\Delta F_{BF}$) resulting when the application of external loading occurs is similar to the $\Delta F_{BP}$ loss due to applied pressure (Equation 1.3.2). This results in $\Delta F_{BF} = f(F_x, M_x)$

\[ 1.5.2 \quad \Delta F_{BF} = \left[ \Delta L_{f\theta} - \Delta L_{B\theta} \right]_F \frac{K_B K_f}{K_B + K_f} \leq \Delta F_{B\theta} \]  

(See Appendix A.2.2)
Where the designation $\Delta F_{B\theta})_F$ for $\Delta F_{BF}$ is used to avoid confusion with the similar term for "L" type flanges. The spring rates are as previously defined while the $\Delta L$ terms are the changes in length of the flange and bolts due to flange rotation occurring when the external loads ($F_x$ and $M_x$) are applied.

Present computer application techniques (19) have not considered the external loads effect independently of the similar pressure effect. Experience has shown the magnitude of $\Delta F_{B\theta})_P$ approaches that of $\Delta F_{B\theta})_P$ in view of the $F_x$ and $M_x$ loading prevailing in present high pressure flange applications. The pressure and external loading effects on bolt load in cantilever flanges may be combined into a single term to consider, from Equations 1.3.2 and 1.5.2.

$$\Delta F_{B\theta} \equiv \Delta F_{B\theta})_P + F_{B\theta})_F$$

1.5.3 $\Delta F_{B\theta} = \left( \Delta L_{f\theta} - \Delta L_{B\theta} \right) \frac{K_P K_r}{K_B + K_f}$ (see Appendix A.2.3)

This can be approximated by:

1.5.3A $\Delta F_{B\theta} \approx \Delta L_{B_P} K_B$ (see Appendix A.2.3)

Where $\Delta L_{B_P}$ is the change in bolt length due to flange rotation from applied pressure and is equal in magnitude to $\Delta L_{f\theta}$ of Equation 1.3.2.

Calculations show that $\Delta F_{B\theta}$ calculated on a pressure basis only per Equation 1.5.3A results in a conservative error approaching 2% of the
total bolt load. This is considered an acceptable error in view of
the safety factors employed in high pressure flange designs. Figure 1.1
depicts the $\Delta F_{B_{0}}$ loss in the time span $3'-h'$.

1.6 **Effects of Seal Loading**

Seal loading is the force required to compress the seal in place
within the seal groove at installation. It is considered to load
the flange axially at the seal contact diameter and is a directly
added effect on bolts for both type flanges.

1.6.1

\[ F_{\text{seal}} = F'_{\text{seal}} D_{sp} \]

1.6.2

\[ F_{B_{\text{seal}}} = \frac{F_{\text{seal}}}{N_{B}} \]

Where $F'_{\text{seal}}$ = lb./in. circumference loading at the seal point
(SP) diameter.

1.7 **Bolt Preload Tolerance at Assembly**

The room temperature installation bolt preload must be such that its
value plus all loading changes incurred during operation will retain
the flange as desired; therefore:

1.7.1

\[ F_{B_{\text{RT \ min}}} + \sum (\Delta F_{B})_{C,P,H \ min} = F_{B_{\text{DP1}}} \]

This $F_{B_{\text{DP1}}}$ is the first design point, i.e., the minimum load required
to retain the flange against the separation loads (Fig. 1.1, time points 4 or 4'). At this point, the flange deflection at the seal point must not exceed $\Delta_{SP}$ allowable. $F_{BDPL}$ consists of the bolt loads from the maximum total direct loading expected on the flange plus the changes in loading incurred when these direct loadings are physically applied. $F_{BDPL}$ and the $\Delta F_B$ terms of Equation 1.7.1 establish $F_{BRT \min}$.

For the "L" type flanges, the direct loading and the effects of their application require:

$$1.7.2 \quad F_{BDPL} = \left[ \frac{PA_{SP}}{N_B} + \frac{F_X}{N_B} + \frac{2M_X}{NB_{BC}} + \frac{F_{seal}}{N_B} \right]_{\text{max}} +$$

$$+ \left[ \Delta F_{BP} + \Delta F_{BF} \right]_{0-4 \text{ max}}$$

Where the subscript max implies the most severe tendency towards flange separation.

Having established $F_{BDPL}$ and taking into consideration the effects of thermals, $F_{BRT \ min}$ can be calculated from Equation 1.7.1 as:

$$1.7.3 \quad F_{BRT \ min} = F_{BDPL} - \left[ \Delta F_{BC} + \Delta F_{BH} + \Delta F_{BP} + \Delta F_{BF} \right]_{0-4 \text{ min}}$$

Where the subscript min implies the least tendency towards flange retention.
For cantilever type flanges, only the direct loading terms establish $F_{BDP1}$ since the effects of their application do not cause an increase in bolt load, therefore:

$$1.7.4 \quad F_{BDP1} = \left[ \frac{F_{ASP}}{N_B} + \frac{F_X}{N_B} + \frac{2M_x}{NB_{RBC}} + \frac{F_{seal}}{N_B} \right]_{\text{max}}$$

From Equation 1.7.1 using the same procedure that established Equation 1.7.3:

$$1.7.5 \quad F_{BRT \min} = F_{BDP1} - \left[ \Delta F_{BC} + \Delta F_{BH} + \Delta F_{B0} \right]_{0-4', \min}$$

The room temperature installation bolt preload must also be such that its value plus all loading changes incurred during operation will not over-load the bolts, therefore:

$$1.7.6 \quad F_{BRT \max} + \sum (\Delta F_B)_{C,H,P} = F_{BDP2}$$

This $F_{BDP2}$ is the second design point, i.e., the maximum load imparted to the bolt during operation (Fig. 1.1, time points 5 or 5'). Bolt material strength criteria limits $F_{BDP2}$. If the bolt material is life limited from an elevated temperature application, the steady state condition (Fig. 1.1, time points 6 or 6') will become another design point to be considered. For cryogenic design, time point 5 (or 5') usually is limiting. However, in the case of cantilever flanges, $F_{BRT\max}$ may become a limiting design point for consideration since
the possibility exists that the highest bolt load occurs at installation. In this case the bolt material room temperature strength may set the design.

For "L" type flanges in a cryogenic application, the following criteria must be satisfied in regards to Equation 1.7.6:

1) The bolt stress at DP2 must not exceed the bolt yield strength at the temperature of time point 5.

\[ F_{BDP2} \leq F_{BY, S}. \]

As these terms become more equal, the design approaches optimization.

2) Since the room temperature bolt load at installation is a variable term dependent upon hardware limitations such as friction, method of loading, etc., as explained in Section 6; the higher load \( F_{BRT_{\text{max}}} \) is used to determine \( F_{BDP2} \). From Equation 1.7.6 rearranged:

\[ F_{BDP2} = F_{BRT_{\text{max}}} + \left[ \Delta F_{BC} + \Delta F_{Bp} + \Delta F_{BF} + \ldots + \Delta F_{BH} \right]_{0-5_{\text{max}}} \]

Where \( F_{BRT_{\text{max}}} = F_{BRT_{\text{min}}}/0.8 \) (Equation 6.1.5) and the subscript max implies the most severe tendency to load the bolt.
For cantilever flanges in a cryogenic application, the following criteria must be satisfied in regards to Equation 1.7.6:

1) The bolt stress at DP2 must not exceed the bolt yield strength at the temperature of time point 5'.

\[ F_{BDP2} \leq F_{BY,S}. \]  

2) The \( F_{BRT_{max}} \) preload is used to determine \( F_{BDP2} \) as:

\[ F_{BDP2} = F_{BRT_{max}} + \left[ \Delta F_{BC} + \Delta F_{B\theta} + \Delta F_{BH} \right]_{0-5'_{max}} \]

where \( F_{BRT_{max}} = F_{BRT_{min}} / 0.8 \) (Equation 6.1.5) and the subscript \( max \) implies the most severe tendency to load the bolt.

3) At room temperature:

\[ F_{BRT_{max}} \leq F_{BY,S}. \]

It should be noted that \( F_{BDP2} \) is a function of \( \Delta F_{B\theta} \) which is determined from trial values of \( F_{BDP2} \). Because of the interdependence of \( \Delta F_{B\theta} \) and \( F_{BDP2} \), iterative calculations are required to determine the bolt load \( F_{BDP2} \). These iterations can be performed by using the computer technique described in Section 2.5.
SECTION 2

FLANGE CONFIGURATION SELECTION

2.1 Candidate Flanges

A comparative study was made using the finite element technique to determine the optimum flange size and type for minimum weight in a high pressure zero leakage seal rig. Flange types evaluated were:

1) Flat Face ("L" type)
2) Cantilever
3) Undercut
4) Loose Ring

The initial design criteria of near zero axial deflection at the seal point resulted in excessive flange weights relative to design targets. An increase in the allowable deflection to 0.002 inch at the seal point indicated acceptable flange weights could be achieved and offered a reasonable design approach to maintain the seal tightness requirements of available seals.

2.2 0.002 Deflection Flanges

A weight and size comparison is shown (Fig. 2.1) for four flanges with a total axial deflection at the seal point of 0.002 inch. From the study, the cantilever flange proved to be the lightest while the undercut flange is the heaviest. The distance from the theoretical
FIGURE 2.1a - Seal Rig 0.002 Deflection Flanges
24 Bolts
0.500-20UNJF-3A
\( F_{RT} = 15,000 \) Lbs

Loose-Ring - 22.43 Lbs

16 Bolts
0.500-20UNJF-3A
\( F_{RT} = 15,000 \) Lbs

Cantilever with Bearing Surface
Inboard of Seal - 14.67 Lbs

FIGURE 2.lb - Seal Rig 0.002 Deflection Flanges
pivot point to the sealing point was found to set the flange weight. The cantilever flange with the bearing surface inboard of the seal proved to be lighter than the ordinary cantilever type. However, piloting requirements for smooth internal flow paths make the ordinary cantilever flange the more practical type and is the recommended choice.

Figure 2.2 shows the weight variation of a cantilever, undercut and loose ring flanges verses axial deflection at the seal point. The cantilever flange types show the sharpest decline in weight for any given deflection.

It can be seen from Figure 2.2 that for 0.002 inch seal point deflection, a cantilever flange with webs between the bolt holes is the lightest. However, for actual hardware, the standard spotfaces used will overlap and remove most of the web material; this is particularly true if the bolt spacing is close together. The spot-facing operation is used to square the flange surface under the bolt head with the flange bearing face since after welding and stress relieve, such surfaces are distorted.

Difficulties were experienced in the finite element method of analysis used for evaluation of webbed cantilever flanges. Apparently the analysis of web discontinuities was not properly written into the program and gave inconsistent results. It was also found that to
FIGURE 2.2 - Seal Rig Flange Weight Vs. Seal Point Deflection
physically have web space, fewer bolts were required. This in turn required larger bolts, a larger bolt circle, and an increased flange outer diameter which consequently resulted in a heavier flange. For these reasons, the use of webbed cantilever flanges is not recommended for high pressure weight optimized applications (2).

2.3 **Explanation of Cantilever Flange Rotational Effect**

The flange rotation effect occurs in cantilever flanges when internal pressure (or axial loading) is applied. The flange deflection occurs at both the seal point and the bolt circle. This deflection at the bolt circle reduces the stretch in the bolt with a corresponding loss in bolt load.

A simplified analysis of the effect considers the flanges as a ring with a uniformly distributed moment about its cross section (Fig. 2.3a). Assume the moment on the flange ring rotates the ring about the centroid of the cross section. The resulting rotation is a function of the cross sectional properties and the applied moment loading as (3):

\[
\theta = \frac{KMR^2}{EI} = M(\text{Constant})
\]

Where

- **M** = Total moment load
- **R** = Radius to the centroid of the ring cross section
- **\theta** = Angular rotation of ring in radians
- **K** = Proportionality constant between M and M'
Total Bolt Load $F_BN_B$

Deflection Due to Bolt Load

Taper Section

Seal Point Within Groove for Area of $p_{ASP}$ Load $F_BN_B$ Contact Face Load

FIGURE 2.3a - Cantilever Flange Rotational Effect

$p_{ASP}$ Pressure Load

Deflection Due to Bolt Load

Deflection Due to Bolt & Pressure Loads

$F_BN_B - p_{ASP}$ Contact Face Load

FIGURE 2.3b - Unpressureized Flange

FIGURE 2.3c - Pressureized Flange
Since the cross-sectional properties are the same whether pressurized or not, the rotation (and deflection) is a linear function of the applied moment.

The rotations and deflections in the pressurized state (subscript 2) can be shown to be greater than those in the unpressurized state (subscript 1) by comparing the moment loading applied to each condition. The sum of the moments about the ring centroid is (Fig. 2.3):

\[ \sum M_1 = (F_{B}N_B)b + (F_{B}N_B)a \]
\[ \sum M_2 = (F_{B}N_B)b + (F_{B}N_B-pA)a + (pA)a \]

If \( M_2 \) is greater than \( M_1 \), then \( M_2 - M_1 > 0 \)

\[\left[ (F_{B}N_B)b + (F_{B}N_B-pA)a + (pA)c \right] - \left[ (F_{B}N_B)b + (F_{B}N_B)a \right] > 0 \]
\[ - (pA)a + (pA)c > 0 \]
\[ pA(c-a) > 0 \]

(where \( pA = pASp \), Fig. 2.3b)

Cantilever type flanges always exhibit pilot, seal groove, and contact face features that result in the geometric quantity (c-a) being greater than zero. Therefore, \( M_2 \) is greater than \( M_1 \) and \( \theta_2 \) will be greater than \( \theta_1 \). \( \theta_2 - \theta_1 \) is the flange rotational effect due to pressure which results in \( \Delta F_{B\theta} \). The same type of rotational effect occurs due to application of external loading which results in \( \Delta F_{B\theta} \).
2.4 Other Configuration Factors That Influence Weight Optimization

1) The minimum seal diameter ($D_{SP}$) and the smallest cross section seal should be used. The smallest seal reduces the pressure-area load, the seal load, and permits use of a smaller bolt circle, all of which help to minimize weight.

2) Flange weight decreases as the flange factor ($N_B F_B / p_{Asp}$) decreases. As the bolt load decreases, the flange ring rolling couple produced by the bolt load ($F_B N_B$) and the contact face load ($F_B N_B - p_{Asp}$) is decreased (Fig. 2.3). Therefore, the minimum number of small bolts consistent with the required flange factor should be used (Fig. 2.4).

3) Weight decreases as bolt size decreases (Fig. 2.5). The flange ring rolling moment is decreased because the distance between the lines of action is diminished, allowing the use of a thinner flange; this effect is small. The major gain is in the reduction in flange outer diameter (O.D.) since smaller bolts can be located closer to the pipe wall.

4) Flange pilot thickness should be as small as possible for it directly influences the size seal and remainder of the flange diameters. A pilot thickness of 0.050 inch is a practical minimum (19).

5) The transition taper between the flange ring and tube or neck section should also be minimum size for weight reasons (Fig. 2.6).
FIGURE 2.4 - Flange Factor Vs. % Weight Penalty

FIGURE 2.5 - Increase in Minimum Bolt Diameter Vs. % Weight Penalty
FIGURE 2.6 - Ordinary Cantilever Flange Cross Section
Equations 2.4.1 and 2.4.2 are recommended for design of the taper height and length.

2.4.1 \[ h_T = (1.35 \text{ to } 1.50) \, t_W \]
\[ = (t_W + 0.020) \text{ min} \]

2.4.2 \[ L_T = \frac{h_T - t_W}{\tan 60^\circ} \]

6) Other recommended geometric relationships (Fig. 2.6) to aid the finite element analysis have been established and include (2):

2.4.3 \[ \frac{2h_T}{D_{BC} - D_{BH} - D_I} \leq 0.90 \quad \text{-- controls } h_T \text{ relative to bolt hole} \]

2.4.4 \[ t_S = \frac{t_f}{8} \quad \text{-- prevents excessive seal groove depth relative to flange thickness} \]

2.4.5 \[ \frac{1}{2} (D_{BC} - D_{BH} - D_R) \geq 0.020 \text{ in.} \quad \text{-- keeps rocking point inside of bolt holes} \]

2.4.6 \[ L_C \geq \frac{\pi}{1.285} \left( D_I + t_W \right) \frac{t_W}{2} \quad \text{-- provides sufficient } L_C \text{ to eliminate the effect of tube bends} \]

2.4.7 \[ 4t_W \leq L_C \leq 16 \, t_W \quad \text{-- alternate for Equation 2.4.6} \]

These limits are required to prevent excessive distortion of the finite elements in the standard element break-up procedure (Fig. 2.7).
FIGURE 2.7 - Standard Finite Element Break-Up, Ordinary Cantilever Flange
2.5 Configuration Analysis Using Finite Element Method (6)

High pressure flange computer analysis and synthesis can be performed using the finite element method technique. This technique models the structure to be analyzed into small subdivisions of two dimensional triangular elements from which a stiffness matrix can be calculated from an assumed deflection matrix. These elements, interconnected at nodal points, are sized sufficiently small so the stresses can be assumed constant throughout the element plane and without appreciably affecting accuracy. Stresses in the elastic realm are calculated by Hooke's law and in the plastic realm by an iterative method on the stress - strain curve for the applicable material (15).

A standard finite element break-up for an ordinary cantilever flange is shown (Fig. 2.7). Output nodes and sections for ordinary cantilever flanges are shown (Fig. 2.8). The Appendix A.6 Calculation 12 details the typical finite element input and output data used for the analysis of cantilever flanges.

2.6 Welded Flanges

Lightweight welded high pressure flanges (Fig. 2.9) are sometimes necessary but have inherent limitations:

1) Stress relieve of welded joints may not be compatible with materials or required dimensional controls of other integral hardware

2) Loss of smooth flow path from internal weld bead
FIGURE 2.8 - Output Nodes and Sections for Ordinary Cantilever Flanges
Weld Joint Strength 85% of Parent Material, Added Thickness Required

Flow Disturbance Possible From Weld Bead

Electron Beam Weld Requires Back-Up Ring

Initial Welds of Materials "A" to "B" With Stress Relieve

"Buttering Strips"

Final Weld of Material "B" to "B" With No Stress Relieve

FIGURE 2.9 - Considerations for Welded Flange Configurations
3) More difficult to repair or assemble in system application.

2.7 **Action Line Flanges**

Action line flanges (Fig. 2.10) offer a good design approach for major component interfacing. With proper design, the load line of action passes through the flange ring centroid thereby imparting minimal deflection at the seal point.

2.8 **Flange Spring Rates**

The equations of Section 1 give sufficient accuracy in determining the bolt loads if the proper flange spring rate parameters $A_f$ and $L_f$ are used. $A_f$ has been found to be the average area of the bolt head and nut outer diameters minus the area of the bolt hole in the flange. $L_f$ is considered to be the same as the effective working length ($L_p$) of the bolt (19).

The $K_f$ established using the above $A_f$ and $L_f$ terms applies only to the deflections at the flange bolt circle. This $K_f$ is not to be confused with other so-called spring rates that could be associated with deflections elsewhere in the flange. In order to closely establish flange deflections, particularly at the seal point and the bolt circle, the finite element method of analysis should be employed for accurate values. This degree of accuracy is necessary if flange optimization is to be achieved in regards to bolt load and seal leakage.
Action Lines (AL) Designed to Pass Through Flange Ring Centroid Reduce Seal Point Deflection and Component Stresses.

FIGURE 2.10 - A Typical Action Line Flange Configuration
3.1 Structural Design Criteria

In order to meet performance targets for a typical system application, the design approach to achieve the minimum weight and leakage requirements necessary for high pressure couplings is based upon clearly defined structural design criteria. This criteria establishes customer and in-house requirements that insure a safe, low risk, configuration is achieved.

3.2 Customer Requirements

Customer requirements such as for NASA are usually specified in a Contract End Item (C.E.I.) and may include considerations for man-rating of the system hardware. Typical requirements applicable to high pressure flanges establish the allowable design criteria for strength levels, testing of pressurized hardware, and the acceptable leakage rates (7).

1) General Structural Criteria

3.2.1 \( S_e = \frac{Y.S.}{1.1} \) or \( S_e = \frac{U.T.S.}{1.4} \) which ever governs, at limit load.
Where limit load includes the maximum load point considering transient overshoots, cycle tolerance, vibration, thermals, acoustics, assembly and acceleration loads.

Local yielding is allowed provided that:

a) The extent of strain is limited by redistribution of stress.

b) It is not detrimental to system operation.

2) Special Structural Criteria (Proof and burst tests)

3.2.2 Proof pressure = Proof factor X limit pressure at design temperature.

The proof factor shall be the larger of 1.2 or the factor determined by fracture mechanics.

3.2.3 Burst pressure = 1.5 X limit pressure at design temperature.

Where limit pressure = maximum component operating pressure including effects of transient overshoots and system acceleration effects on the fluid within the pressure vessel.

3) Maximum Leakage Rates

a) 1 X 10^{-4} scc / sec of He at operating or leak check pressure for separable connectors.

b) 1 X 10^{-6} scc / sec of He at operating or leak check pressure for welded, brazed, or hermetically sealed enclosures.
3.3 In-House Requirements

The design agency structural criteria is established from testing, experience, or analytical studies to insure that a safe, systematic, and competitive development program is achieved.

1) General Structural Criteria

Re-definition or upgrading of customer general and specific structural requirements as applicable to fluid interfaces (8).

Definition of system application requirements for:

a) Service life
b) Number of starts (cycles)
c) Normal power level operation (NPL)
d) Emergency power level operation (EPL)

2) Specific Structural Criteria

Pressure Vessels - A material fracture of mechanics analysis shall be performed to establish the proof test pressure factor required for safe design. This is to determine the maximum permissible flaw size the material can have while insuring service life capability with respect to sustained and cyclic loading (8).

Fatigue Criteria - A detailed design life cycle history curve shall be developed in sufficient detail so that accumulative cycle fatigue damage assessment can be
analytically verified. Components shall be designed to the following safety factors (SF) based on guaranteed minimum life (7):

a) Low cycle fatigue (LCF), use (SF) on no. of cycles

b) High cycle fatigue (HCF), use (2.5 SF) on no. of cycles

A life-fraction analysis is to be used to determine the safety factors (24):

a) Allowable major cycles (starts)

3.3.1

\[
\frac{N_s \times SF}{N_c} \Big|_{NPL} + \frac{N_s \times SF}{N_c} \Big|_{EPL} = 1
\]

b) Allowable minor cycles (excursions)

3.3.2

\[
\frac{N_s \times SF}{N_c} \Big|_{NPL} + \frac{N_s \times SF}{N_c} \Big|_{EPL} + \frac{M_c \times SF}{N_m} = 1
\]

Where

- \( N_s \) = Number of starts at normal (NPL) or emergency (EPL) power levels
- \( N_c \) = Allowable number of major cycles at NPL or EPL for material LCF strength
- \( N_m \) = Allowable number of minor cycles for material LCF strength for a given transient
- \( M_c \) = Number of allowable minor stress cycles with \( N_{SEPL} + N_{SNPL} \) cycles
Temperature Limited Structure - Cryogenic components are not limited by creep or stress rupture criteria. In a high temperature situation, the allowable component stress levels shall be determined by a life-fraction method of analysis.

Acceleration And Gimbal Loads - All components shall be designed to withstand the most severe maneuver and gimbal g-loads, axial and transverse. Criteria defining the motion for flight hardware shall include the applicable vehicle pitch and roll rates (°/sec). Additional motion within the vehicle shall also be considered; this includes the rate (°/sec) and acceleration (radians/sec²) as typified by moving an engine for thrust vector control.

Environmental Requirements - The following environmental parameters shall be determined for system application:

a) Temperature exposure
b) Pressure exposure
c) Relative humidity and salt exposure
d) Vibration, shock, and acoustic exposure
e) Sand, dust, and contamination exposure
f) Ground storage and handling
g) Space environment, time, and orbits
Stress Allowables – The effective stress shall not exceed the 0.2% yield strength, or some legislative factor thereof, unless a plastic strain analysis shows that LCF life and deflection limits are met (8).

Effective stress:

\[ S_e = \sqrt{S_1^2 + S_2^2 + S_3^3 - S_1 S_2 - S_2 S_3 - S_1 S_3} \]

Where

- \( S_1 = \) Principle plane stress (max)
- \( S_2 = \) Principle place stress (min for biaxial)
- \( S_3 = \) Principle plane stress (min for triaxial)

Shear stress:

\[ S_s = 0.57 (0.2\% \text{ Y.S.}) \]

Combined stress, bending and tension interaction:

\[ \frac{S_t}{C_1 \times 0.2\% \text{ Y.S.}} + \frac{S_b}{C_2 \times 0.2\% \text{ Y.S.}} \leq 1 \]

Where

- \( C_1 = \) Legislated value, this usually ranges from 0.85 to 1.00 depending upon material
- \( C_2 = f(\text{Material, material cross sectional shape, and temperature}) \)
- \( = (C_3) cQ, \) this value ranges from 1.3 to 1.55 for most nickel base alloys
- \( C_3 = f(\text{Material, determined by bending-tensile-temperature test}) \)
\begin{align*}
c & = \text{Geometric shape distance to neutral axis} \\
I & = \int x^2 \, dA, \text{ total area moment of inertia} \\
Q & = \int c \cdot x \, dA, \text{ first (static) moment}
\end{align*}

\textbf{Welded Joints} - Stresses will be limited to 85\% of parent material properties. Creep and stress rupture criteria shall be established for high temperature applications.

\textbf{Bolted Flanges} - Specific in-house requirements for bolted flanges shall meet the following (9):

1) Flanges

a) Flanges shall be of the ordinary cantilever type with 0.002 inch deflection at the seal point excluding "V" type seals for which 0.004 inch deflection shall apply.

b) Seal load shall not exceed 350 lb./in. of seal circumference.

c) Calculated flange effective stress cannot exceed 85\% of the 0.2\% Y.S. except flange contact face average bearing stress shall not exceed 1.1 X 0.2\% Y.S.

d) The \textbf{pressure load} shall equal the limit pressure load at design temperature calculated using the area at the seal contact point diameter.

e) In the absence of known plumbing loads, flanges
and their supporting structure shall support 75% of a moment load which will stress to yield the mating pressurized pipe.

2) Bolts

a) Use high strength bolts, necked per design agency standards.

b) At installation or working temperature, the bolt bending and tension shall be combined by the applicable interaction criteria.

c) A minimum preload bolt factor of $1.8 (p_{Asp}) + F_{seal}$ shall be used when external loadings (and effects) are not finalized. The preload bolt factor may be reduced to $1.5 (p_{Asp}) + F_{seal}$ when the thermal lag in the bolts and the relaxation of the bolts due to flange rotation under pressure are accounted for.

d) Bolts shall be designed for installation free of torsion with a spread of 20% between minimum and maximum preload.

3.4 Special Additional Considerations

Flange design requirements are subjective to system / component general design requirements. Typical considerations include (10):
Failure Mode, Effect, and Criticality Analysis

Overhaul

Protective Treatment - Silver plate, hard face, lubrication
Materials - Properties, suitability, conformance to specifications
Design - Where possible, use standardized design procedures and hardware parts
Maintainability - Modular design, assembly ease
Interchangeability - From system to system except for selective fits
Cleanliness - Conformance with fuel, oxidizer
Protective Covers - Wrapping, packaging
Welding Process - Fusion, arc, gas, electron beam, per latest technology
Heat Treatment - Solution, stabilization, and precipitation
Inspection - X-ray, zyglo, quality control
4.1 Material Selection
Flange materials are usually set by mating hardware material requirements. The most compatible high pressure coupling system is achieved if the interface components, plumbing, flanges, bolts, and housings are of the same material; welding and thermal change compatibility are thus achieved. Inconel 718 material has been successfully fabricated and used in high pressure flange and line configurations. Figures 4.1 and 4.2 depict Inconel 718 material properties (11).

4.2 Properties of Inconel 718

1) Weldability
Welding is accomplished with the same general techniques used for Inconel X and Waspaloy, but with less susceptibility to strain cracking. Fusion welding is done in the solution heat treated condition with parent metal filler followed by full heat treatment to achieve optimum properties (11). Weld strength is assumed to be 85% of parent material strength.

2) Ductility
Good transverse ductility on materials three inches or thicker
FIGURE 4.1 - Y.S. and U.T.S. Vs. Temperature for Cold Inconel 718
Density - 0.297 lb/in$^3$
Elongation - 20%

FIGURE 4.2 - $\alpha$ and $E$ Vs. Temperature for Cold Inconel 718
is attainable, but difficult. Average cryogenic elongation is 20%.

3) **Forgeability**

Fair to good, more readily forgeable than Waspaloy and Astroloy (11).

4) **Application**

For parts requiring high strength, good weldability, good corrosion and oxidation resistance at temperatures from -423°F to 1100°F; non magnetic. Alloy exhibits moderate increase in strength to -423°F with slight increase in elongation (11). With four times the strength and three times the modulus of elasticity of aluminum, Inconel 718 is potentially the lighter material.

5) **Machinability**

Difficult - machined with same general techniques and degree of difficulty as Inconel X and Incoloy 901. Machinable in all conditions; fully heat treated condition is preferred for finish machining (11).

6) **Inconel 718 For Plumbing**

For high pressure fluid lines, Inconel 718 affords relatively thin walls with lower weight compared to Inconel 625, 347 SST, and A - 286 SST. Modern forming techniques have been developed for Inconel 718 lines and tubes (12).
7) Limitations

Heat treating can reduce ductility to 50% of parent properties. Susceptibility to hydrogen embrittlement presently under study; the phenomena appears to be a function of temperature.

8) Toughness

Good at cryogenic temperatures, Figure 4.3 gives a comparison of fracture toughness vs. temperature for Inconel 718, aluminum and titanium (13).

9) Bending – Tension Interaction

Figure 4.4 shows the interaction behavior of Inconel 718 vs. various shape factors at room temperature (14). Computer programs have been written which enables an interaction curve to be drawn for any material at any temperature providing the stress strain diagram is known. These type curves are based on 0.2% strain in the outer fiber for a given loading situation.

Theoretical curves so produced are conservative compared to test data in pure bending; however, their use is recommended rather than actual test data interaction curves which apparently are sensitive to material structure inconsistency. Indications are that the compressive side of the beam does not behave identically to the tension side when loaded. The
Numbers are not accurate, use for material screening only.

FIGURE 4.3 - Fracture Toughness Vs. Temperature for Inconel 718, Aluminum, and Titanium
Shape Factor $SF = \frac{2QC}{I}$

Note:
Curves cannot be interpolated as a function of temperature.
For Symmetrical Geometry Only

FIGURE 4.4 - Inconel 718 Interaction Curves Vs. Shape Factor
result is a higher gain in pure bending allowable than theory predicts.

The interaction curves indicate that for higher yield strength materials, the allowable gain decreases. This is dependent on the ratio of plastic strain to total strain. As the temperature increases and the yield point decreases, there is a point where this ratio increases - at this point the allowable gain begins to gradually rise (15).

4.3 MP 35 N Bolt Material Shows Very High Strength

Recently available as Aerospace Material Specification (AMS) 5758, MP 35 N material is a good candidate material for high pressure flange bolt application. Figures 4.5 and 4.6 depict the cold material properties of this multiphase cobalt base material (11). Compared with cold Inconel 718, MP 35 N exhibits:

a) 56% to 75% greater 0.2% Y.S.
b) 33% to 44% greater U.T.S.
c) 80% to 101% change range for χ
d) 14% to 15% greater E

e) Approximately 50% decrease in % elongation

High pressure flanges of Inconel 718 material using MP 35 N bolts have been recently proposed for high pressure rocket engine applications.
The high strength properties of MP 35 N compensate the mismatch in \( \mathcal{L}'s \); however, the reduced percentage of elongation is the apparent limiting characteristic of MP 35 N.
FIGURE 4.5 - Y.S. and U.T.S. Vs. Temperature for Cold MP 35 N
FIGURE 4.6 - $\alpha$ and $E$ Vs. Temperature for Cold MP 35 N

Density - 0.304 lb/in$^3$
Elongation - 10%

Temperature - F$^\circ$

Coeficient of Expansion - $\alpha \times 10^{-6}$ in/in F$^\circ$

Modulus of Elasticity - $E \times 10^6$ psi
SECTION 5

SEAL CONSIDERATIONS

Candidate Seals

Proprietary seal and cavity designs are often required to achieve stringent leakage rates. Maximum leakage rates of $1 \times 10^{-4}$ sec I sec He per inch of seal circumference have been achieved; candidate seals for achieving this performance together with seal loading for each are listed in Table 5.1:

**TABLE 5.1**

**CANDIDATE SEALS - SOURCE AND LOADING**

<table>
<thead>
<tr>
<th>Seal Name</th>
<th>Manufacturer</th>
<th>Seal Loading (F' seal), lbs. / in.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1) &quot;V&quot; Seal</td>
<td>Parker Seal Co.</td>
<td>250</td>
</tr>
<tr>
<td>2) Metal &quot;O&quot; Ring</td>
<td>United Aircraft Products Co.</td>
<td>575</td>
</tr>
<tr>
<td>3) DelTau &quot;C&quot; Ring</td>
<td>Pressure Science Inc.</td>
<td>225</td>
</tr>
<tr>
<td>4) Bi - Metallic Gasket</td>
<td>Del Manufacturing Co.</td>
<td>300</td>
</tr>
<tr>
<td>5) Omega Seal</td>
<td>Servotronics Inc.</td>
<td>150</td>
</tr>
<tr>
<td>6) Apex Seal</td>
<td>Servotronics Inc.</td>
<td>150</td>
</tr>
</tbody>
</table>

If $F'$ seal is unknown, experience has shown that 350 lbs. / in. is a reasonable assumption to use for preliminary designs (19).
All seals are designed for pressure assist at the seal contact point. Inside-out seals to accommodate high external pressures are available or can be developed.

**Seal Space Allowance**

The geometry space required (Fig. 5.1) to accommodate the candidate seals may have to be accounted for prior to final seal selection. Since these seals are competitive in performance and size, a given flange design can be pre-sized with minimal machining at finalization (16).

Double seal configurations vented inbetween the inner and outer seals for leakage collection are not recommended for use in ordinary cantilever flange designs. The distance from the inner seal point to the rocking point becomes excessively large.

**Seal Point Deflection**

Testing has demonstrated that limiting the seal point deflection ($\Delta_{SP}$) to 0.002 inch in high pressure cantilever flanges is an acceptable design criteria (4). The "V" seal requires a thicker flange than an "O" ring for the same 0.002 deflection (17). This is because the "O" ring has a contact point closer to the cantilever rocking point than does the "V" seal (Fig. 5.2). It is therefore recommended that a deflection of 0.004 inch maximum be allowed for "V" seals because of their inherent axial flexibility (17).
FIGURE 5.1 - Seal Allowance Space for Candidate Seals
FIGURE 5.2 - Seal Contact Point Comparison

"V" Seal Contact Point

"0" Ring Contact Point
As the cross sectional size of a candidate seal diminishes, machining accuracy (tolerances) become more critical since the axial deflections are smaller with smaller seals.

Seal Contact Surfaces Consideration

Seal surfaces should be either corrosion resistant or silver plated. Seals coated with fluorocarbon for improved contact surfaces interaction perform well initially but exhibit cold flowing with time, and therefore usually have poor shelf-life.

The seal cavity in the flange should be closely machined. Surfaces at the sealing points having a finish of 16 microinch arithmetical average roughness, 0.005 inch waviness, and with a circular tool lay pattern, have proven to be acceptable (Fig. 5.3) (18).

Other Seal Cavity Considerations

The seal cavity may be shared in part by each of the flange halves, but gives a more expensive configuration to manufacture (Fig. 5.3).

Placing the entire seal cavity and the male pilot in the least expensive component is prudent for repair or replacement reasons in case of damage (Fig. 5.1). Protruding surfaces such as male pilots are susceptible to handling damage. If minor damage such as scratches occur to a seal surface that is flush with the flange face, a shallow clean-up operation salvages the more expensive hardware.
0.005 R Max, Typical

0.010 Max, Typical

Typical for Back-Up Shoulders

0.0005

Typical for All Type Seals

Parallel within 0.002 FIR

FIGURE 5.3 - "V" Seal Cavity Design
Good design practice places the male pilot of the flange joint on the component that is assembled in the upwards direction. This aids in retaining the seal in a proper position during installation (Fig. 5.1).

For "V" seals, the seal cavity should provide backup shoulders to reinforce the seal against collapse (Fig. 5.3).
6.1 Bolt Considerations

Necked Bolts - A good design practice is to neck down the shank diameter of highly loaded bolts. Necked bolt diameters \(D_B\) (Fig. 6.1) are sized to give a shank cross sectional area equal to 93% of the area at the minimum minor thread diameter \(D_{mm}\). Therefore:

\[
D_B = \sqrt{0.93} \cdot D_{mm} = 0.9644 \cdot D_{mm}
\]

For necked bolts under load, the encountered strain is confined primarily to the smooth shank region. This tends to reduce the possibility of failure in the stress concentration areas of the threads. The length of the necked portion of the shank shall not be less than the major diameter of the thread. Where shorter bolts are necessary, bolts should be necked only to the thread pitch diameter (19).

Spring Rates - Figure 6.1 depicts the spring rate parameters \(L_B, A_B, L_f, A_f\) recommended for "through bolt" designs. Figure 6.2 shows the length parameters \(L_B\) and \(L_f\) to be used in spring rate calculations for bolts in tapped (threaded) flanges. These geometric
Self-Locking Nut

Bolt Neck Dia \( D_B \) Determines Bolt Spring Rate

Area \( A_B \)

Two Threads Max

Radius \( R \) Per Table 6.1

UNJF-3 Threads

Two Threads Max

\( L_B, L_f \)

Cylindrical Area \( A_f \) Used to Approximate Flange Spring Rate

Wrench Clearance Radius \( L_W \) Nut or Bolt Per Table 6.2

Hex End With Lockwire Hole

Bolt Hole Dia

\( D_{BH} = D_T + 0.031 \pm 0.005 \)

0.094 - 0.141 R

FIGURE 6.1 - A Typical Through-Bolt Installation
Finish Machine Critical Surfaces After Heat Treatment for Squarness Control

Allow Space for Threaded Insert Repair (Optional)

Two Full Threads + 0.020 Engagement To Prevent Surface A Deformation by F_B

FIGURE 6.2 - A Typical Tapped-Flange Bolt Installation
definitions used for estimating the spring rates $K_r$ and $K_B$ apply to the equations of Section 1.

**Bolt Threads** - Use UNJF-3 threads, rolled per Military Specification MIL-S-8879. Such threads are preferred for external thread applications and where desired quantities are relatively large, because they are less expensive to manufacture and because of improved physical properties. Due to the grain flow and cold working effect of rolling threads, a part fabricated in this manner will have a greatly increased fatigue life, and an extended stress rupture life at high temperature. Rolling of threads also produces a very smooth surface finish (less than 16 microinches roughness) which makes assembly easier and reduces the tendency for thread seizure (19).

The J-thread form, of the Unified National Fine specification, features a controlled radius root external thread and an increased internal thread minor diameter. J-threads reduce the tendency for thread galling, particularly if the bolt and nut (or tapped hole) are of the same material.

Fine threads exhibit higher load capacity than coarse threads and are recommended for use except in aluminum applications where thread shear is more critical.

The minimum full thread length shall be sufficient to accommodate:

1) The maximum flange assembly stackup that establishes $L_{f_{max}}$. 
2) The maximum length of the largest candidate nut.
3) The approximate length of two exposed threads.

For bolts installed in tapped holes, the full thread engagement length should be a minimum of the bolt major diameter. The first thread of the bolt (nearest the flange interface) should be buried into the flange approximately two threads plus 0.020 inch; this permits a longer bolt for stretch plus eliminates the possibility of having raised dimples on the flange surface due to high bolt loading (Surface A, Fig. 6.2).

In the event of damage to tapped threads, it is prudent to provide repair space to accommodate helicoil type inserts. In the case of cantilever flanges, this criteria may set the bolt circle further from the rocking point. For "L" type flanges, this criteria reduces the flange contact face effective bearing area.

Bolt Hole Clearance - The bolt hole diameter ($D_{BH}$) shall be equal to the bolt thread nominal diameter plus 0.031 ± 0.005 inch (Fig. 6.1). Locate each flange hole within 0.005 radius of true position relative to the flange pilot diameters; this insures interference free assembly of the flange halves and bolts. One bolt location may be offset 0.060 or less in order to maintain a required flange rotational alignment position (19).
Bolt Head Radius  - The bolt shank-to-head radius (R, Fig. 6.1) is usually established by the bolt manufacturer's standards. Table 6.1 depicts typical values of R for standard and tensilized bolts (20).

TABLE 6.1

BOLT HEAD RADIUS R (FIG. 6.1)

<table>
<thead>
<tr>
<th>Bolt Size</th>
<th>Standard</th>
<th>Tensilized</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.250 &amp; under</td>
<td>0.015 - 0.025</td>
<td>0.031 - 0.041</td>
</tr>
<tr>
<td>0.3215 - 0.375</td>
<td>0.020 - 0.030</td>
<td>0.037 - 0.047</td>
</tr>
<tr>
<td>0.4375 - 0.500</td>
<td>0.025 - 0.035</td>
<td>0.047 - 0.057</td>
</tr>
<tr>
<td>0.625 - 0.750</td>
<td>0.030 - 0.040</td>
<td>0.063 - 0.073</td>
</tr>
<tr>
<td>0.875</td>
<td>0.035 - 0.045</td>
<td>0.063 - 0.073</td>
</tr>
<tr>
<td>1.000</td>
<td>0.040 - 0.055</td>
<td>0.063 - 0.073</td>
</tr>
<tr>
<td>1.000 +</td>
<td>0.050 - 0.070</td>
<td>0.077 - 0.089</td>
</tr>
</tbody>
</table>

Bolt Spacing  - The maximum number of bolts possible on a given bolt circle is governed by the minimum possible chordal space (L<sub>ch</sub>) between the bolts from either Equation 6.1.2 or 6.1.3:

6.1.2 Washer face limited  \[ L_{ch} = D_{w,\text{min}} \text{ O.D.} + 0.020'' \]

6.1.3 Wrench clearance limited  \[ L_{ch} = \frac{1}{2} L_H + 0.010'' + L_W \]
Where

\[ L_H = \text{Hex head cross corner length} \]
\[ L_W = \text{Wrench clearance radius (Fig. 6.3)} \]

Using the larger term of Equation 6.1.2 or 6.1.3, the maximum possible number of bolts is determined from geometry as:

\[ N_{B_{\text{max}}} = \frac{360^\circ}{2 \sin^{-1} \left( \frac{L_{\text{ch min}}}{D_{BC}} \right)} \]

Wrench clearance \( L_W \) for socket and box wrenches is shown in Figures 6.1 and 6.3, and is tabulated in Table 6.2.

**FIGURE 6.3 - Wrench Clearance Parameters**
### TABLE 6.2

**WRENCH CLEARANCE FOR BOLTS AND NUTS**

<table>
<thead>
<tr>
<th>$L_{\text{flat}}$</th>
<th>$L_{NW} \text{ min}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal</td>
<td>Socket</td>
</tr>
<tr>
<td>0.188</td>
<td>0.181</td>
</tr>
<tr>
<td>0.219</td>
<td>0.201</td>
</tr>
<tr>
<td>0.250</td>
<td>0.221</td>
</tr>
<tr>
<td>0.312</td>
<td>0.250</td>
</tr>
<tr>
<td>0.375</td>
<td>0.297</td>
</tr>
<tr>
<td>0.438</td>
<td>0.344</td>
</tr>
<tr>
<td>0.500</td>
<td>0.375</td>
</tr>
<tr>
<td>0.562</td>
<td>0.422</td>
</tr>
<tr>
<td>0.625</td>
<td>0.456</td>
</tr>
<tr>
<td>0.688</td>
<td>0.500</td>
</tr>
<tr>
<td>0.750</td>
<td>0.531</td>
</tr>
<tr>
<td>0.875</td>
<td>0.625</td>
</tr>
<tr>
<td>0.938</td>
<td>0.703</td>
</tr>
<tr>
<td>1.062</td>
<td>0.781</td>
</tr>
<tr>
<td>1.250</td>
<td>0.938</td>
</tr>
<tr>
<td>1.438</td>
<td>1.062</td>
</tr>
</tbody>
</table>

**Bolt Loading** - Obtaining an exact desired bolt load at assembly is virtually impossible to achieve by conventional torque calibration. Depending upon the coefficient of friction, the torque energy is partially consumed overcoming friction. Since the coefficient of friction is highly variable according to test data (21) conventional torquing of high pressure bolts is not recommended.
Alternate methods for loading bolts are (22):

1) Angle of turn
2) Hydraulic ram
3) Measured stretch
4) Shank-held bolt

Method 4) combined with 3) is a satisfactory approach. By preventing the bolt from twisting, the measured axial strain is an accurate indicator of the pure tensile load. Figure 6.1 shows a typical hex-end bolt feature used for this purpose.

Prior to installation, all external threads should be treated with a dry film lubricant such as Microseal 100-1 to minimize frictional effects.

A reasonable target for room temperature assembly bolt load \( F_{BRT} \) is (2):

\[
F_{BRT\ max} = \frac{F_{BRT\ min}}{0.80}
\]

Bolt Stress - The bolt head and flange bearing stresses should always be checked. Determination of the bearing area includes consideration for all chamfers and rounds of the nut, bolt head, and flange features (Fig. 6.2).

The bolt stress should be calculated taking into account the effect of axial loading and bending:
1) Conventional theory (23)

\[ S_B = S_t + S_b \]

\[ S_B = P/A + Mc/I \]

\[
6.1.6 \quad S_B = \frac{4F_B}{\pi D_B^2} + \frac{4}{\sqrt{\pi}} \left( \frac{\theta \sqrt{E_B}}{F_B} \right) D_B \sinh \left[ \frac{4}{\sqrt{\pi}} \frac{L_{Be}}{D_B^2} \sqrt{\frac{F_B}{E_B}} \right]
\]

(See Appendix A.4)

Where

\[ L_{Be} = L_B \text{ of Figure 6.1 for through-bolts} \]
\[ = L_B \text{ of Figure 6.2 for tapped flange configurations} \]
\[ = \frac{1}{2} L_B \text{ of Figure 6.2 if tapped flange is fixed (no rotation)} \]

2) Interaction theory (preferred), as given by Equation 3.3.5

\[
\frac{S_t}{(C_1) 0.2\% \text{ Y.S.}} + \frac{S_b}{(C_2) 0.2\% \text{ Y.S.}} \leq 1
\]

Where

\[
6.1.7 \quad S_b = \frac{\theta E_B D_B}{L_B} \quad \text{(See Appendix A.5)}
\]

The bending stress imposed on the bolts can be kept lower if closely controlled squareness and parallelism of the flange critical surfaces are maintained. Therefore, finish machine the critical flange surfaces after heat treating; maintain these surfaces parallel within 0.001 FIR
and keep the bolt holes square with the flange surfaces within 0.002
FIR (Fig. 6.2).

6.2 Nut Considerations

Nut recommendations include:

1) Self-locking feature required
2) Plating or lubrication desirable
   a) Silver plate
   b) Solid (dry) film
3) 12 - point (double hex) head configuration preferred
4) UNJF-3B threads preferred
5) 160 KSI minimum strength at room temperature required.

Nuts and bolts are usually vendor-supplied items that can be special
ordered to meet the requirements for a given application.
7.1 Task

The design problem consists of providing a flanged interface between mating high pressure lines. The flange joint is to be used on initial-build development hardware. To insure a low-risk design, the following criteria is to be used:

1) A factor 1.5 on the limit pressure
2) The seal point deflection shall not exceed 0.004 inch for "V" type seals
3) In addition to known loading, the flange joint, for development purposes, shall be capable of supporting 75% of the moment load which will yield the mating pressurized pipe; the 0.004 seal point deflection need not apply under this loading
4) The flange shall be of the ordinary cantilever type.

7.2 Given Data:

1) Fluid Media - hydrogen at:

\[ P_l = 6900 \text{ psi limit pressure} \]
\[ T^\circ = 137^\circ \text{ Rankine} \]
2) **Known Geometry:** (See Fig. 2.7)

- $t_w = 0.190$ inch each line
- $D_I = 1.966$ inches internal flowpath diameter

3) **Known Loading:**

- $F'_\text{seal} = 350$ lb./in. circumference
- $F_x = 310$ lb. axial load due to suspended mass
- $M_x = 11,700$ lb./in. moment

### 7.3 Structural Criteria

1) **Customer Requirements:**

- $S_e = \text{Y.S.} / 1.1$ or
  - $= \text{U.T.S.} / 1.4$, effective stress (Eqn. 3.2.1)
- $P_p = 1.2 \ p_1$, proof pressure (Eqn. 3.2.2)
- $P_b = 1.5 \ p_1$, burst pressure (Eqn. 3.2.3)

2) **In House Requirements:** (See Figs. 4.1, 4.2)

Material shall be Inconel 718 for bolts and flanges, selection is based upon:

- a) Material adjacent to each flange is Inconel 718
- b) Welded joints of like materials preferable
- c) Compatibility with environment is adequate

The fracture mechanics factor for Inconel 718 is 1.34.
Bending and tension interaction for Inconel 718 material requires:

\[
\frac{S_t}{(0.85) \text{Y.S.}} + \frac{S_b}{(1.3) \text{U.T.S.}} \leq 1 \quad \text{(Eqn. 3.3.5)}
\]

Welded joints are to be limited to 85% of the allowable stress permissible for the parent material.

In lieu of unestablished vibrational data and noting that creep and stress rupture criteria do not apply to cryogenic applications, allowable stress levels are:

\[
S_e \leq 0.85 \text{ (0.2\% Y.S.) (Section 3.3, Bolted Flanges Criteria lc)}
\]

\[
S_{brg} \leq 1.1 \text{ (0.2\% Y.S.) (Section 3.3, Bolted Flanges Criteria lc)}
\]

Bolts will be necked and designed for room temperature installation free of torsion with a spread of 20% between minimum and maximum preload.

7.4 Summary Of Problem

The example flange design problem is summarized in Figures 7.1 and 7.2. Figure 7.1 depicts the flange load map which shows the effects of temperature and pressure change on bolt load. Figure 7.2 is a full scale cross-section of the example flange with a summary of the more important loads, stresses, and deflections considered in the analysis.
**EXAMPLE PROBLEM CANTILEVER FLANGE LOAD MAP**

<table>
<thead>
<tr>
<th>Room Temp</th>
<th>( T_f ) = 50(^\circ)R</th>
<th>( T_f ) = 50(^\circ)R</th>
<th>( T_f ) = 50(^\circ)R</th>
<th>( T_f ) = 50(^\circ)R</th>
<th>( T_f ) = 200(^\circ)R</th>
<th>( T_f ) = 200(^\circ)R</th>
</tr>
</thead>
<tbody>
<tr>
<td>( T_B ) = 100(^\circ)R</td>
<td>( T_B ) = 60(^\circ)R</td>
<td>( T_B ) = 60(^\circ)R</td>
<td>( T_B ) = 60(^\circ)R</td>
<td>( T_B ) = 100(^\circ)R</td>
<td>( T_B ) = 190(^\circ)R</td>
<td></td>
</tr>
</tbody>
</table>

- \( F_{BRT \ max} = 8307 \)
- \( F_{BRT \ min} = 6646 \)
- \( F_{BDP1} = 5200 \)
- \( F_{BDP2} = 8250 \)

\( \Delta F_{BC} \):
- 0-1: -284
- 1-2: +243
- 2-3: 0
- 3-4: -1200
- 4-5: +1184
- 5-6: -1014

Operational Time Points

**FIGURE 7.1**
FIGURE 7.2 - Summary of Example Problem

Legend for Figure 7.2

(1) 6646 - 8307 lbs. at installation ($F_{BRT}$).

8250 lbs. design point 2 load ($F_{BDP2}$).

15,520 lbs. max possible bolt load ($F_{BDP2+M}$).*

(2) 57,090 lbs. separation load due to given $F_x$, $M_x$, $F'_\text{seal}$, & 1.5 p.

(3) 69,000 psi bending stress plus 114,000 psi vs. 122,000 psi allowable tensile stress at 100° R.*

(4) 98,700 psi bearing stress vs. 183,700 psi allowable at 200° R.*

(5) 62,600 psi effective stress vs. 142,000 psi allowable at 200° R.

(6) 46,800 psi effective stress vs. 142,000 psi allowable at 200° R.

(7) 123,500 psi bearing stress vs. 183,700 psi allowable at 200° R.*
(8) 97,500 psi bearing stress vs. 183,700 psi allowable at 200°F.

(9) Length reduces 0.0004884 inch at bolt circle due to flange rotation, gives bolt load relaxation of 1200 lbs. ($\Delta F_{bg}$).

(10) Flange opens 0.0025176 inch due to flange rotation.

(11) Moment required to yield pressurized line, an additional safety requirement to (2).

(12) Total weight of flange and bolts is 6.6004 lbs.

* Values given include the effect of moment (11).
8.1 Conclusion
The high pressure flange design approach as described in this report has been substantiated by flight quality hardware developed by Pratt & Whitney Aircraft, Florida Research and Development Center.

8.2 Recommendations
1) The finite element method of analysis described in Section 2.5 does not have the capability of directly inputting the loading effects of $F_x$ and $M_x$. It is recommended that the computer program be revised to consider these effects. This would enable the analysis to provide the deflections for the $\Delta L_p$ and $\Delta L_f$ terms of Equations 1.3.2 and 1.5.2 which can then be used for determining $\Delta F_B$ (Equation 1.5.3) more accurately.

2) Further hardware evaluation of MP35N as a bolt material is recommended, particularly in regards to elongation properties (see Section 4.3).

3) As shown by Figure 2.1b, the cantilever type flange with the bearing surface inboard of the seal is potentially the lightest weight
design. Investigation of this type of flange for an acceptable piloting feature and evaluation of its performance without webs should be pursued (See Section 2.2).
REFERENCES


15. H. E. Johnson, P&WA FRDC, Job # 82821, 1969.*
18. M. Gershon, P&WA FRDC, Memo, 10/30/1968.*

25. V. S. Fehr, P&WA FRDC, FTDM 256, 1968.*

* Pratt & Whitney Aircraft, Florida Research and Development Center, unpublished data
APPENDIX

A.1. The equations describing the change in bolt preload when operating loads are applied to an "L" type flange are given by derivations A.1.1 and A.1.2.

A.1.1 Derivation of Equation 1.3.1 (1)

\[ \Delta F_{Bp} = \frac{pA_{SP}}{N_B} \left[ \frac{K_B}{F_B + K_f} \right] \]

Consider Figure A.1 which has the bolt passing through the flange center of gravity. The nut is only snug against the flange, thereby giving no initial bolt load.

1) When a separating force (F) is applied, the flange and bolt stretch.
2) F is shared partially by bolt (FB) and by flange (Ff) as:
   \[ F = F_B + F_f \]
3) Since the deflection of the bolt is the same as for the flange,
   \[ \frac{F_B}{K_B} = \frac{F_f}{K_f} \]
   \[ \text{where } K = \frac{AE}{L} \]
4) Noting \( F_f = F - F_B \), the equation of 3) becomes
   \[ \frac{F_B}{K_B} = \frac{F - F_B}{K_f} \]
   \[ = \frac{F}{K_f} - \frac{F_B}{K_f} \]
Rearranging,

\[ F_B \left( \frac{1}{K_B} + \frac{1}{K_f} \right) = \frac{F}{K_f} \]

\[ F_B = \frac{F}{K_f} \left[ \frac{K_B K_f}{K_B + K_f} \right] \]

\[ = F \left[ \frac{K_B}{K_B + K_f} \right] \]

With an initial preload \( (F_{B_0}) \) in the bolt

\[ F_B = F \left[ \frac{K_B}{K_B + K_f} \right] + F_{B_0} \]

The change in bolt load is

\[ F_B - F_{B_0} = \Delta F_B = F \left[ \frac{K_B}{K_B + K_f} \right] \]

5) For Equation 1.3.1, \( F \) is the pressure load \( \frac{p^{ASP}}{N_B} \)

\[ \therefore \Delta F_B = \frac{p^{ASP}}{N_B} \left[ \frac{K_B}{K_B + K_f} \right] \]

when pressure is applied to the flange.

A.1.2 Derivation of Equation 1.5.1

\[ \Delta F_{BF} = \left[ \frac{F_X}{N_B} + \frac{2M_X}{N_B R_{BC}} \right] \frac{K_B}{K_B + K_f} \]
The derivation is the same as given in A.1.1 except in this case F of Figure A.1 equals the external loads applied to the "L" type flange \[ \frac{F_x}{N_B} + \frac{2M_x}{N_BR_BC} \].

A.2 The equations describing the change in bolt preload when operating loads are applied to a cantilever type flange are given by derivations A.2.1, A.2.2, and A.2.3.

A.2.1 Derivation of Equation 1.3.2

\[ \Delta F_{B_0} = \left[ \Delta L_{f_0} - \Delta L_{B_0} \right]_D \frac{K_B K_f}{K_B + K_f} \]

Consider Figure A.2 which has a flange and bolt at a length \( L_3 \) corresponding to the length at an operational time point 3 such as shown in Figure 7.1.

1) When a separating force (P) consisting of pressure and external loading is applied to the flange, the resulting moments cause the flange half to rotate such that \( L_3 \) is shortened to some length \( L_4 \).

2) This \( L_4 \) corresponds to one-half the entire bolt length at an operational time point 4 such as shown in Figure 7.1.

3) Since \( L_4 \) is less than \( L_3 \), the strain in the bolt has been relaxed during the flange rotation with a corresponding loss in bolt load.

4) At the bolt circle, flange face A has changed position by \( \Delta L_{f_0} \),
\[ \Delta L_{f\theta} = \Delta L_{fP} - \frac{\Delta F_{Bf} E_f}{A_f} \]

\[ \Delta L_{BP} = \Delta L_{B\theta} + \frac{\Delta F_{Bf} L_B}{A_B E_B} \]

**Figura A.2**
an amount equal to $\Delta L_{f\theta}$ due to rotation minus the spring-back of the face $\Delta F_{B\theta} L_f/A_f E_f$ due to bolt load relaxation.

5) Similarly, the bolt length has changed by $\Delta L_{Bp}$, an amount equal to $\Delta L_{B\theta}$ due to bolt rotation plus the spring-back of the bolt $\Delta F_{B\theta} L_B/A_B E_B$.

6) Noting from Figure A.2

$$L_3 - \Delta L_{fp} = L_4 = L_3 - \Delta L_{Bp}$$

$$\Delta L_{fp} = \Delta L_{Bp}$$

$$\therefore \quad - \frac{\Delta F_{B\theta} L_f}{A_f E_f} + \Delta L_{f\theta} = \frac{\Delta F_{B\theta} L_B}{A_B E_B} + \Delta L_{B\theta}$$

$$\Delta F_{B\theta} \left[ \frac{1}{K_B} + \frac{1}{K_f} \right] = \Delta L_{f\theta} - \Delta L_{B\theta}$$

or $\Delta F_{B\theta} = \left[ \Delta L_{f\theta} - \Delta L_{B\theta} \right] \frac{K_B K_f}{K_B + K_f}$

which is Equation 1.5.3 of the text where both flange halves are considered in determining $\Delta F_{B\theta}$ of the entire bolt.

7) If the separating force ($P$) of Figure A.2 consists only of pressure effects $\left[ \frac{P\lambda_{SP}}{N_B} \right]$ the subscript "p" is added to Equation 1.5.3 to give Equation 1.3.2 as

$$\Delta F_{B\theta}p = \left[ \Delta L_{f\theta} - \Delta L_{B\theta} \right]_p \frac{K_B K_f}{K_B + K_f}$$
A.2.2 Derivation of Equation 1.5.2

If the separating force \( P \) of Figure A.2 consists only of external loading effects \( \left[ \frac{F_x}{N_B} + \frac{2M_x}{N_B R_{BC}} \right] \), the subscript \( F \) is added to Equation 1.5.3 in the same manner as \( p \) was added in 7) of derivation A.2.1.

Therefore Equation 1.5.2 is obtained as

\[
\Delta F_{B\theta})_F = \left[ \Delta L_{f\theta} - \Delta L_{B\theta} \right] p \quad \frac{K_B K_f}{K_B + K_f}
\]

A.2.3 Justification for Equation 1.5.3A

\[
\Delta F_{B\theta} \approx \Delta L_{B\theta} p K_B
\]

This approximation for Equation 1.5.3 can best be justified using actual values from the example problem given in Appendix A.6, calculation 13c. For this case, \( \Delta F_{B\theta} = -1200 \text{ lbs.} \) using Equation 1.5.3A. This value compares with \( \Delta F_{B\theta} = -1040 \text{ lbs.} \) as calculated by the summation of Equations 1.3.2 and 1.5.2 as follows:

\[
\Delta F_{B\theta})_p = \left[ \Delta L_{f\theta} + \Delta L_{B\theta} \right] p \quad \frac{K_B K_f}{K_B + K_f} \quad (\text{Eqn. 1.3.2})
\]

\[
= 688 \text{ lbs.}
\]

Where

\[
\Delta L_{f\theta} = 0.0004884 \text{ in.}
\]

\[
= \Delta L_{B\theta} \text{ per calculation 13c}
\]
\[ \Delta L_{B_\theta} = \Delta R_{62_p}(\theta), \text{ where } \Delta R_{62_p} \text{ is calculated from the computer output data as (see Figure A.3)} \]

\[ \Delta R_{62_p} = \Delta R_{62_p + F_B} - \Delta R_{62_F} \]

\[ = 0.0020336 - 0.0011144 \quad \text{(See Appendix A.6, calculation 12b)} \]

\[ = 0.0009192 \text{ in.} \]

and \( \theta = 0.00488 \text{ radians} \quad \text{(See Appendix A.6, calculation 16)} \)

\[ \Delta L_{B_\theta} = 0.00000896 \text{ in.} = 2\Delta R_{62_p} \text{ for entire bolt} \]

\[ K_B = 2.457 \times 10^6 \text{ lbs./in. per Table A.1} \]

\[ K_f = 3.450 \times 10^6 \text{ lbs./in. per Table A.1} \]

\[ \Delta F_{B_\theta} = \left[ \Delta L_{f_\theta} - \Delta L_{B_\theta} \right]_F \frac{K_BK_f}{K_B + K_f} \quad \text{(Eqn. 1.5.2)} \]

\[ \approx 352 \text{ lbs.} \]

Since the computer program of Section 2.5 doesn't provide the deflections \( \Delta L_{f_\theta} \) and \( \Delta L_{B_\theta} \) due to external loading, the value of \( \Delta F_{B_\theta} \) may be obtained by ratioing the external load to the pressure load as follows:

a) Pressure loaded the bolts to

\[ P_{ASP} = \frac{6900}{11} \frac{t}{4} (2.250)^2 = 2400 \text{ lbs/bolt} \]

which gave \( \Delta F_{B_\theta} = 688 \text{ lbs. (relaxation)} \)

b) External loading loads the bolts to

\[ \frac{F_X}{N_B} + \frac{2M_X}{N_B^R_{BC}} = \frac{310}{11} + \frac{2(11,700)}{11(1.77)} = 1228 \text{ lbs/bolt} \]
\[ X = \Delta R_{62_p}(e) \]
\[ L_{Be} \leq X \]
\[ L_{Be} \geq \Delta R_{62_p}(e) \]

FIGURE A.3
c) Ratio of a) and b)

$$\Delta F_{B\theta} = \Delta F_{B\theta}^a + \Delta F_{B\theta}^b$$

From Equations 1.3.2 and 1.5.2:

$$\Delta F_{B\theta} = \frac{1228}{2400} (688) = 352 \text{ lbs (relaxation)}$$

Therefore Equation 1.5.3A is conservative by 1200 - 1040 = 160 lbs vs. the more accurate value given by Equation 1.5.3. This represents an error of $\frac{160}{8250} = 1.94\%$ in the design bolt load.

A.3 Derivation of $F_B = \frac{2M_x}{R_{BC} N_B}$ as found in Equations 1.5.1, 1.7.2, and 1.7.4.

Consider Figure A.4 which represents a cross-section through the bolts of a flange. The flange may be of either the "L" type or cantilever type with the section taken at mid-length of the equal size bolts.

As a moment due to external loading is applied around some axis $X - X$, the bolts above the axis $X - X$ receive tensile loading proportional to their distance from axis $X - X$. The bolts below the axis relax in tensile loading.
FIGURE A.4
Denoting the angular distance between the bolts as $\theta = \frac{2\pi}{N_B}$, and neglecting the moment of inertia effect of each individual bolt, the bending stress of the bolts considered as a system is

$$S_B = \frac{Mc}{I}$$

The maximum bending stress of the system occurs in bolt "A" as a tensile stress, therefore

$$S_B^{\text{max}} = S_{tA} = \frac{MR_{BC}}{I_{xx}}$$

For the bolt system, the significant terms of the parallel axis theorem give

$$I_{xx} = A_B(R_{BC})^2 + A_B(R_{BC} \cos \theta)^2 + A_B(R_{BC} \cos 2\theta)^2 - \cdots - A_B[R_{BC} \cos (N_B-1)\theta]^2$$

$$= A_B R_{BC}^2 \left[ 1 + \sum_{K=1}^{N_B-1} \cos^2 K\theta \right]$$

Using the identity *

$$\sum_{K=1}^{n-1} \cos^2 K\theta = \frac{n-1}{2} + \frac{\cos n\theta \sin(n-1)\theta}{2 \sin \theta}$$

\[ I_{xx} = A_B R_{BC}^2 \left[ 1 + \frac{N_B - 1}{2} + \frac{\cos N_B \theta \sin(N_B - 1) \theta}{2 \sin \theta} \right] \]
\[ = A_B R_{BC}^2 \left[ 1 + \frac{N_B - 1}{2} + \frac{\cos 2\pi \sin(N_B - 1) \frac{2\pi}{N_B}}{2 \sin 2\pi \frac{N_B}{N_B}} \right] \]
\[ = A_B R_{BC}^2 \left[ 1 + \frac{N_B}{2} - \frac{1}{2} \right] \]
\[ = A_B R_{BC}^2 \frac{N_B}{2} \]

\[ \therefore S_t^A = \frac{M R_{BC}}{A_B R_{BC}^2 \frac{N_B}{2}} = \frac{2M}{A_B R_{BC} N_B} \]

Since \( F_B = A_B S_t \)
\[ F_{B\text{max}} = A_B S_{t\text{max}} = A_B S_t^A = \frac{2M}{R_{BC} N_B} \]

or \[ F_B = \frac{2M_x}{R_{BC} N_B} \]
as used in the text.

A.4 Derivation of text Equation 6.1.6 for Bolt Stress from Tensile Plus Bending Loading (23)
Consider a bolt loaded as shown in Figure A.5

\[ S_B = \frac{P}{A} + \frac{M}{I} \]
bolt stress from combined tension and bending

\[ M_{\text{max}} = M \sech \frac{U}{2} \quad \text{at} \quad X = \frac{L}{2} \]
Figure A.5

Figure A.6

G. Bolt Shank

G 1/2 Bolt Shank
Where

\[ U = \frac{L}{J} = \frac{L}{\sqrt{EI/P}} \]

\[ \theta = \frac{M}{PJ} \tanh \frac{U}{2} \text{ at } X = 0, L \]

Or

\[ M = \frac{PJ\theta}{\tanh \frac{U}{2}} \]

\[ \therefore M_{\text{max}} = \frac{PJ\theta}{\tanh \frac{U}{2}} \text{ sech} \frac{U}{2} = \frac{PJ\theta}{\sinh \frac{U}{2}} \]

\[ \frac{Mc}{I} = \frac{PJ\theta c}{I \sinh \frac{U}{2}} = \frac{P\theta c \sqrt{EI/P}}{I \sinh \left[ \frac{1}{2} \frac{L}{\sqrt{EI/P}} \right]} \]

Using the terms of the text,

\[ c = \frac{D_B}{2}, \quad I = \frac{\pi}{64} (D_B)^4, \quad P = F_B, \quad L = L_B, \quad E = E_B \]

\[ \therefore \frac{Mc}{I} = \frac{F_B(\theta)}{64} \frac{D_B}{\sqrt{64 F_B}} \left[ \frac{L_B}{2 \sqrt{E}} \right] \sinh \left[ \frac{D_B^2}{64} \sqrt{\frac{1}{4} \frac{D_B^4}{F_B}} \right] \]

\[ = \frac{1}{D_B} \sqrt{\pi \theta} \left( \frac{\sqrt{E F_B}}{F_B} \right) \sinh \left[ \frac{L_B}{\sqrt{D_B^2}} \sqrt{\frac{F_B}{E_B}} \right] \]
\[
\frac{P}{A} = \frac{F_B}{A_B} = \frac{4F_B}{\pi D_B^2}
\]

\[
S_B = \frac{4F_B}{\pi D_B^2} \pm \frac{4}{\sqrt{\pi}} (\theta) \sqrt{EBF_B}
\]

\[
D_B \sinh \left[ \frac{4L_B}{\sqrt{\pi} D_B^2} \sqrt{\frac{F_B}{E_B}} \right]
\]

A.5 Derivation of Equation 6.1.7

\[
S_B = \frac{EBDB}{LB}
\]

Consider a bolt loaded as shown in Figure A.6,

\[
\theta = \frac{ML}{EI} \text{ at } A
\]

\[
M = \frac{\theta EI}{L}
\]

\[
S_B = \frac{Mc}{I}
\]

\[
= \frac{\theta EI}{L} \times \frac{c}{I}
\]

\[
= \frac{\theta Ec}{L}
\]

Using the terms of the text,

\[
S_B = \frac{2\theta EB}{LB} \times \frac{DB}{2}
\]

\[
\therefore S_B = \frac{EBDB}{LB}
\]
A.6 Example Problem Calculations in Detail:

1) From an initial design sketch, a thermal analysis predicts the operating time point temperatures as shown in the flange load map of Figure 7.1.

2) Utilizing a "V" type seal, the following geometry is required (Fig. 2.6):

\[
\begin{align*}
D_{SP} &= 2.250 \text{ inches} \\
t_s &= 0.078 \text{ inches} \\
D_s &= 2.452 \text{ inches} \\
h_s &= 0.150 \text{ inches} \\
F_{\text{seal}} &= \pi D_{SP} F'_{\text{seal}} \quad (\text{Eqn. 1.6.1}) \\
&= \pi \times 2.250 \times 350 \\
&= 2480 \text{ lbs.}
\end{align*}
\]

3) Design Load (first approximation using Section 3.3 Bolted Flanges criteria 2c):

\[
\begin{align*}
N_B F_B &= 1.8 \left[ \Pi A_{SP} \right] + F_{\text{seal}} \\
&= 1.8 \left[ 6900 \times \frac{\pi}{4} (2.250)^2 \right] + 2480 \\
&= 49,300 + 2480 \\
&= 51,780 \text{ lbs.}
\end{align*}
\]

4) Bolts - first trial, try 0.500-20 UNJF-3A bolts:

\[
D_B = 0.9644 D_{\text{mm}} \quad (\text{Eqn. 6.1.1})
\]
\[ \text{DB} = 0.9644 (0.4360) \]
\[ = 0.420 \text{ max to } 0.416 \text{ min, in.} \]

\[ \text{AB} = \frac{\pi}{4} (0.416)^2 = 0.136 \text{ in.}^2 \text{ min.} \]

\[ S_t = 0.85 \text{ Y.S. at } 200^\circ R, \text{ allowable tensile} \]
\[ = 0.85 (167,000) \]
\[ = 142,000 \text{ psi (limiting)} \]

or \[ S_t = \text{U.T.S. / 1.4 at } 200^\circ R \]
\[ = 218,000 / 1.4 \]
\[ = 154,000 \text{ psi (not limiting this case)} \]

\[ F_{B_{\text{max}}} = \text{AB} \cdot S_{t_{\text{max}}} \]
\[ = (0.136) \cdot 142,000 \]
\[ = 19,300 \text{ psi} \]

\[ F_{B_{\text{min}}} = 0.80 (19,300) \quad (\text{Eqn. 6.1.5}) \]
\[ = 15,400 \text{ lbs} \]

\[ N_B = 51,780 / 15,400 = 3.36 \]
\[ = 4 \text{ bolts, not considering bending stress, flange rotation, thermals, and external loads.} \]

5) Using parametric data based upon experience (2), the 51,780 lbs. load plus considerations for bending stress, flange rotation, thermals, and external loads give estimated bolt options as:
a) 0.625 bolts, \( N_B = 6 \) bolts

b) 0.562 bolts, \( N_B = 8 \) bolts

c) 0.500 bolts, \( N_B = 10 \) bolts

6) From several iterative trials and considering the requirements of Task 7.1 criteria 3, the following parameters were established as applicable to the design problem:

a) Bolts, 0.500-20 UNJF-3A

\[
N_B = 11 \text{ required}
\]

\[
D_{mm} = 0.436 \text{ in.}
\]

\[
D_B = 0.416 \text{ in. min.}
\]

\[
A_B = 0.136 \text{ in.}^2 \text{ min.}
\]

\[
L_B = 2 \left( t_f + t_U \right)
\]

\[
= 2 \left( 0.830 + 0.025 \right)
\]

\[
= 1.710 \text{ in.}
\]

\[
D_{brg} = 0.710 \text{ in.}, \text{ bolt head bearing diameter}
\]

\[
D_{BC} = 3.540 \text{ in.}
\]

\[
R_{BC} = 1.770 \text{ in.}
\]

b) Nuts, 0.500-20 UNJF-3B

Nut contact face bearing diameter = 0.740 in.

c) Flange

Flange bolt hole diameter and chamfer = 0.576 in. max.

Nut / flange bearing area:
\[ A_{brg} = \frac{\pi}{4} (0.740^2 - 0.576^2) = 0.170 \text{ in.}^2 \text{ min.} \]
\[ F_{fbrg} = 0.170 \times (1.1)(0.85) \times (Y.S. = 0.159 \times (Y.S.) \]
\[ \text{limited} = 0.159 \times (150,000) = 23,900 \text{ lbs. at } 530^\circ R \]
\[ = 0.159 \times (167,000) = 26,600 \text{ lbs. at } 200^\circ R \]

Bolt / flange bearing area:

\[ A_{brg} = \frac{\pi}{4} (0.710^2 - 0.576^2) = 0.135 \text{ in.}^2 \text{ min.} \]
\[ F_{fbrg} = 0.135 \times (1.1)(0.85) \times (Y.S. = 0.126 \times (Y.S.) \]
\[ = 0.126 \times (150,000) = 18,900 \text{ lbs. at } 530^\circ R \]
\[ = 0.126 \times (167,000) = 21,000 \text{ lbs. at } 200^\circ R \]

Flange area for spring rate (\(A_F\)):

Average diameter of bolt and nut = \(\frac{0.740 + 0.710}{2}\)

\[ = 0.725 \text{ in.} \]

\[ A_F = \frac{\pi}{4} (0.725^2 - 0.531^2) = 0.191 \text{ in.}^2 \]

Flange length, \(L_F = L_B = 1.710 \text{ in.} \]

7) Tabulated data for Figure 7.1 time points is shown in Table A.1
### TABLE A.1

**TABULATED DATA**

<table>
<thead>
<tr>
<th>Inconel 718</th>
<th>$T = 530^\circ R$</th>
<th>$T = 50^\circ R$</th>
<th>$T = 60^\circ R$</th>
<th>$T = 100^\circ R$</th>
<th>$T = 200^\circ R$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E, \text{ psi}$</td>
<td>$29 \times 10^6$</td>
<td>$31.6 \times 10^6$</td>
<td>$31.55 \times 10^6$</td>
<td>$31.4 \times 10^6$</td>
<td>$30.9 \times 10^6$</td>
</tr>
<tr>
<td>$\alpha, \text{ in.}/\text{in.}/^\circ R$</td>
<td>$6.80 \times 10^{-6}$</td>
<td>$5.50 \times 10^{-6}$</td>
<td>$5.58 \times 10^{-6}$</td>
<td>$5.88 \times 10^{-6}$</td>
<td>$6.25 \times 10^{-6}$</td>
</tr>
<tr>
<td>$\text{Y.S., psi}$</td>
<td>$150,000$</td>
<td>$180,000$</td>
<td>$178,000$</td>
<td>$175,000$</td>
<td>$167,000$</td>
</tr>
<tr>
<td>$\text{St}_{\text{max}} = 0.85 \text{ (Y.S.)}$</td>
<td>$127,500$</td>
<td>$153,000$</td>
<td>$151,000$</td>
<td>$149,000$</td>
<td>$142,000$</td>
</tr>
<tr>
<td>$\text{Y.S.} / 1.1$</td>
<td>$135,000$</td>
<td>$163,700$</td>
<td>$162,000$</td>
<td>$159,000$</td>
<td>$152,000$</td>
</tr>
<tr>
<td>$\text{U.T.S., psi}$</td>
<td>$180,000$</td>
<td>$238,000$</td>
<td>$236,000$</td>
<td>$232,000$</td>
<td>$218,000$</td>
</tr>
<tr>
<td>$\text{U.T.S.} / 1.4$</td>
<td>$128,500$</td>
<td>$170,000$</td>
<td>$168,500$</td>
<td>$166,000$</td>
<td>$154,000$</td>
</tr>
<tr>
<td>$K_B = \frac{A_B E_B}{L_B}, \text{ lb./in.}$</td>
<td>$2.305 \times 10^6$</td>
<td>$2.515 \times 10^6$</td>
<td>$2.505 \times 10^6$</td>
<td>$2.496 \times 10^6$</td>
<td>$2.457 \times 10^6$</td>
</tr>
<tr>
<td>$K_f = \frac{A_f E_f}{L_f}, \text{ lb./in.}$</td>
<td>$3.240 \times 10^6$</td>
<td>$3.530 \times 10^6$</td>
<td>$3.522 \times 10^6$</td>
<td>$3.510 \times 10^6$</td>
<td>$3.450 \times 10^6$</td>
</tr>
<tr>
<td>$\Delta L = \alpha L(T-530), \text{ in.}$</td>
<td>$0$</td>
<td>$0.004515$</td>
<td>$0.004485$</td>
<td>$0.004320$</td>
<td>$0.003530$</td>
</tr>
</tbody>
</table>
8) Bolt load changes due to thermals at time points of Figure 7.1 with data from Table A.1.

a) At time point 1, \( T_f = 50^\circ R \) \( T_B = 100^\circ R \)

\[
\Delta L_f = \alpha L (T_f - T_0)
\]

\[
0-1 = 5.5 \times 10^{-6} (1.710) (50 - 530) = -0.004515 \text{ in.}
\]

\[
\Delta L_B = \alpha L (T_B - T_0)
\]

\[
0-1 = 5.58 \times 10^{-6} (1.710) (100 - 530) = -0.004320 \text{ in.}
\]

\[
K_{B100} = \frac{A_B E_B}{L_B} = (0.136) 31.4 \times 10^6 / 1.710
\]

\[
= 2.496 \times 10^6 \text{ lbs./in.}
\]

\[
K_{f50} = \frac{A_f E_f}{L_f} = (0.191) 31.6 \times 10^6 / 1.710
\]

\[
= 3.530 \times 10^6 \text{ lbs./in.}
\]

\[
\Delta F_{BC} = (\Delta L_f - \Delta L_B) \frac{K_fK_B}{K_f + K_B} \quad \text{(Eqn. 1.2.1)}
\]

\[
0-1 = -284 \text{ lbs.}
\]

b) Likewise at time points 2, 3, and 4:

\[
T_f = 50^\circ R \quad T_B = 60^\circ R
\]

\[
\Delta L_f = -0.004515 \quad K_f = 3.530 \times 10^6
\]

\[
\Delta L_B = -0.004485 \quad K_B = 2.505 \times 10^6
\]

\[
\Delta F_{BC} = (\Delta L_f - \Delta L_B) \frac{K_fK_B}{K_f + K_B}
\]

\[
0-2 = 41 \text{ lbs.}
\]

\[
\Delta F_{BC} = \Delta F_{BC} = \Delta F_{BH}
\]

\[
0-3 \quad 0-4 \quad 0-4
\]
\[ \Delta F_{BC} = \Delta F_{BC} - \Delta F_{BC} \quad \text{(Eqn. 1.2.2)} \]
\[ = -41 - (-284) \]
\[ = 243 \text{ lbs.} \]

\[ \Delta F_{BC} = \Delta F_{BC} - \Delta F_{BC} = 0 \]
\[ 2-3 \quad 0-3 \quad 0-2 \]

\[ \Delta F_B = \Delta F_B - \Delta F_B = 0 \]
\[ 3-4 \quad 0-4 \quad 0-3 \]

\(c)\) At time point 5, \(T_f = 200^\circ R\) \(T_B = 100^\circ R\)

\[ \Delta L_f = -0.003530 \quad K_f = 3.450 \times 10^6 \]

\[ \Delta L_B = -0.004320 \quad K_B = 2.496 \times 10^6 \]

\[ \Delta F_{BH} = (\Delta L_f - \Delta L_B) \frac{K_f K_B}{K_f + K_B} \quad \text{(Eqn. 1.4.1)} \]
\[ = +1143 \]

\[ \Delta F_{BH} = \Delta F_{BH} - \Delta F_{BH} = +1143 - (-41) \]
\[ = 1184 \text{ lbs.} \]

9) Corresponding to time point 4, the first design point (DPl) requires the bolts to have sufficient load to prevent flange separation from the operational loads imposed. From Eqn. 1.7.4,

\[ N_{BF_{BDPL}} = N_{BF_{Bl}} = 1.5 F_{A_{SP}} + F_X + \frac{2M_X}{R_{BC}} + F_{seal} \]

\[ \text{min req'd} \]
1.5 = a safety factor on limit pressure ($p_1$)

$$p_1A_{SP} = \frac{(6900) \gamma}{4} (2.250)^2 = 27,400 \text{ lbs.}$$

$$F_x = 310 \text{ lbs. due to supported mass}$$

$$M_x = 11,700 \text{ lb. in. moment on one of the lines}$$

$$F_{seal} = 2480 \text{ lbs.}$$

$$N_B = 11$$

$$11F_{BDP1} = 41,100 + 310 + 13,200 + 2,480$$

$$F_{BDP1} = \frac{57,090}{11} = 5,200 \text{ lbs. per bolt}$$

(Note: The 57,090 lb. separation load compares favorably with the 51,780 lbs. approximation of calculation 3. The 5200 lbs. bolt load is considerably less than the approximated 15,400 lbs. ability and the flange could function with less bolts. However, the requirement to support 75% of the moment that fails the mating line under pressure justifies the eleven bolts as will be shown.)

10) Estimating room temperature minimum preload ($F_{BR T \min}$)

using $\Delta F_B = -1300 \text{ lbs.}$ as determined from iterative load-deflection calculations, Equation 1.7.5 gives:

$$F_{BR T} + \Delta F_B + \Delta F_C + \Delta F_B + \Delta F_{BO} = F_{BDP1}$$

$$F_{BR T} + (-284) + 243 + 0 + (-1300) = 5200 \text{ lbs.}$$
11) Estimating DP2 bolt load, Equation 6.1.5 gives:

\[ F_{B_{RT\ max}}^{est.} = F_{B_{RT\ min}}^{est.} / 0.8 = \frac{6541}{0.8} = 8200 \text{ lbs.} \]

From Equation 1.7.10:

\[ F_{B_{RT\ max}}^{est.} + \Delta F_{BC} + \Delta F_{BC} + \Delta F_B + \Delta F_{B\theta} + \Delta F_{BH} = F_{B_{DP2}} \]

\[ 8200 - 284 + 243 + 0 - 1300 - 1184 = F_{B_{DP2}} \]

\[ F_{B_{DP2}} = 8043 \text{ lbs., estimated} \]

At this point, a finite element structural analysis is performed to determine the bolt circle deflection \( \Delta L_B \), then \( \Delta F_{B\theta} \) is calculated from Equation 1.5.3 as shown by calculation 13c. The \( \Delta F_{B\theta} \) term plus the known terms of Equation 1.7.10 must be equal to the bolt load input \( F_B = F_{B_{DP2}} \) for the design to be balanced. Iterative trials using \( F_{B_{DP2}} \) as the variable are performed as necessary to balance the loads. For this problem, \( F_{B_{DP2}} = 8250 \text{ lbs.} \) gives a balanced design as shown by calculations 12, 13, and 14.

12) Substituting \( F_B = F_{B_{DP2}} = 8250 \text{ lbs.} \) into the finite element structures analysis program, the final iteration reads as follows:

a) Computer program input (Fig. 2.6):
\[ D_B = 0.500 \text{ in., bolt shank diameter} \]
\[ D_{BC} = 3.540 \text{ in., bolt circle diameter} \]
\[ D_{BH} = 0.531 \text{ in., bolt hole diameter} \]
\[ F_B = 8250 \text{ lbs., bolt load, lb./bolt} \]
\[ \sigma_f = 0.297 \text{ lbs./in.}^3, \text{ flange weight density} \]
\[ D_{SP} = 2.250 \text{ in., seal point diameter} \]
\[ h_S = 0.153 \text{ in., seal slot height} \]
\[ t_S = 0.078 \text{ in., seal slot width} \]
\[ h_f = 0.970 \text{ in., flange height} \]
\[ t_f = 0.830 \text{ in., flange thickness} \]
\[ p = 6900 \text{ psi, internal pressure} \]
\[ F_S = 2480 \text{ lbs., seal load} \]
\[ t_U = 0.025 \text{ in., undercut depth} \]
\[ D_R = 2.975 \text{ in., contact face outside diameter} \]
\[ D_S = 2.452 \text{ in., seal slot outside diameter} \]
\[ N_B = 11, \text{ number of bolts} \]
\[ \sigma_B = 0.297 \text{ lbs./in.}^3, \text{ bolt weight density} \]
\[ h_T = 0.260 \text{ in., taper height} \]
\[ L_T = 0.430 \text{ in., taper length} \]
\[ D_T = 1.966 \text{ in., flange or tube inside diameter} \]
\[ t_W = 0.190 \text{ in., tube wall thickness} \]
\[ L_C = 0.820 \text{ in., tube length} \]
\[ E = 30.9 \times 10^6 \text{ psi, elastic modulus} \]
\[ \Delta_{SP} = 0.004 \text{ in., seal point total axial deflection} \]
b) Computer program prime output for one flange half:

$\triangle_{SP}$ due to $F_B = 0.00035841$ in.

$\triangle_{SP}$ due to $F_B+p = 0.0012588$ in.

t_F = 0.830 in.

Contact face stresses (Fig. 2.8):

Node 8 = 51,979 psi axial compression

Node 9 = 16,608 psi axial compression

Stresses at Section A (Fig. 2.8):

$S_t = 36,233$ psi tensile

$S_b = 23,976$ psi bending

$S_t + S_b = 60,209$ psi combined axial = $S_x$

$S_h = 64,912$ psi hoop

Stresses at Section B (Fig. 2.9):

$S_t = 13,704$ psi tensile

$S_b = 18,172$ psi bending

$S_t + S_b = 31,876$ psi combined axial = $S_x$

$S_h = 53,689$ psi hoop

Weights:

Weight of (2) flange halves = 4.3595 lbs.

Weight of bolts and nuts = 2.2409 lbs.

Total weight = 6.6004 lbs.
Deflections at bolt circle (Node 62):

\[ \Delta \beta_{FB} = -0.0022612 \text{ in. due to } F_B, \text{axial} \]

\[ \Delta \beta_{FB+p} = -0.0025054 \text{ in. due to } F_B+p, \text{axial} \]

\[ \Delta \rho_B = 0.0011444 \text{ in. due to } F_B, \text{radial} \]

\[ \Delta \rho_{FB+p} = 0.0020336 \text{ in. due to } F_B+p, \text{radial} \]

Deflections at Nodes 58 and 66:

\[ \Delta \beta_{66} = -0.0038872 \text{ in., axial} \]

\[ \Delta \beta_{58} = 0.0008513 \text{ in., axial} \]

13) Review of output data:

a) All flange stresses are less than the allowable stress of (0.85) Y.S. = 142,000 psi at 200°F. At sections A and B, the effective stress is calculated by Equation 3.3.3 as:

\[ S_e = \sqrt{S_h^2 + S_x^2 - S_hS_x} \]

\[ S_{eA} = 62,600 \text{ psi} \]

\[ S_{eB} = 46,800 \text{ psi} \]

b) The \( \Delta \text{SP} = 2(0.0012588) = 0.0025176 \) for both flange halves vs. 0.004 maximum permissible for "V" seal designs.

c) From the deflection of one flange half at the bolt circle (Node 62) and Equation 1.5.3A, \( \Delta F_{B0} \) can be calculated.

\[ \Delta \beta_{PC} = \Delta \beta_{PC+p} - \Delta \beta_{PC} = -0.0002442 = \frac{1}{2} \Delta L_{BP} \]
\[ S_t = \frac{E \Delta L_{Bp}}{L_B} = -30.9 \times 10^6 \times 488.4 \times 10^{-6} \]
\[ = -8825 \text{ psi} \]

\[ \Delta F_{B0} = \Delta L_{Bp} K_B = \Delta S_{tA} B = -8825 (0.136) \]
\[ = -1200 \text{ lbs.}, \text{ vs. the} \ -1300 \text{ lbs. estimate of} \]


calculation 10.

14) Finalizing loads \( F_{BRT \ max} \), \( F_{BRT \ min} \), \( F_{B6 \ max} \), and \( F_{B6 \ min} \):

a) \[ F_{BRT \ max} = F_{BRT \ max} \]
\[ = 284 + 243 + 0 - 1200 + 1184 = 8250 \]

\[ F_{BRT \ max} = 8307 \]

b) \[ F_{BRT \ min} = (0.8) 8307 = 6646 \text{ lbs.} \ (\text{Eqn. 6.1.5}) \]

Checking if \( F_{BRT \ min} \) gives 5200 lb. load at DPl,

\[ \Delta F_{B0} \leq -1200 \text{ lbs.} \]

Since \( F_{BRT} = 6646 \text{ lbs.} \) will give less flange rotation

than \( F_{BRT} = 8307 \text{ lbs.} \)

\[ F_{BRT \ min} + \Delta F_{BC} + \Delta F_{BC} + \Delta F_{B} + \Delta F_{B0} \geq F_{BDPL} \]
\[ = 5200 \text{ required by calculation 9} \]
\[ 6646 - 284 + 243 + 0 - 1200 = 5405 \]

\[ F_{BDP1} \geq 5200 \text{ lbs. min criteria has been met and the iteration of calculation 12 is considered to be sufficiently accurate for the final analysis.} \]

c) At time point 6, \( T_f = 200^\circ R \) \( T_B = 190^\circ R \)
\[ \Delta L_f = -0.003530 \quad K_f = 3.450 \times 10^6 \]
\[ \Delta L_B = -0.003620 \quad K_B = 2.462 \times 10^6 \]
\[ \Delta F_{BH}^{0-6} = (\Delta L_f - \Delta L_B) \frac{K_f K_B}{K_f + K_B} \text{ (Eqn. 1.4.1)} \]
\[ = +129 \text{ lbs.} \]
\[ \Delta F_{BH}^{0-5} = +1143 \text{ lbs. (See calculation 8c)} \]
\[ \Delta F_{BH}^{5-6} = \Delta F_{BH}^{0-6} - \Delta F_{BH}^{0-5} = +129 - 1143 \]
\[ = -1014 \text{ lbs.} \]

d) \( F_{B6_{\text{max}}} = F_{BDP2} + \Delta F_{BH}^{5-6} \)
\[ = 8250 - 1014 = 7236 \text{ lbs.} \]

e) \( F_{B6_{\text{min}}} = F_{B4_{\text{min}}} + \Delta F_{BH}^{4-5} + \Delta F_{BH}^{5-6} \)
\[ = 5405 + 1184 - 1014 \]
\[ = 5575 \text{ lbs.} \]
15) Moment required to yield mating line

a) At 200°R with pressure:

\[ M_{200} = \frac{I}{c} \left[ Y.S. - \frac{P_l R_l}{2 t_W} \right] \]

Where:

\[ I = \pi (R)^3 t_W = \pi (0.983 + 0.095)^3 0.190 \]
\[ = 0.744 \text{ in.}^4 \]
\[ c = 0.983 + 0.190 = 1.173 \]
\[ Y.S. = 0.2\% \text{ Y.S. at } T_F = 200^\circ R \]

\[ M_{200} = \left( \frac{0.744}{1.173} \right) \left[ 167,000 - \frac{6900 (0.983)}{2 (0.190)} \right] \]
\[ = 94,500 \text{ lb. in.} \]

The bolts must accommodate 75% of this effect,

\[ F_{BM_{200}} = (0.75) \frac{2 M_{200}}{R_l N_B} = (0.75) \frac{2 (94,500)}{1.770 (11)} \]
\[ = 7270 \text{ lbs.} \]

b) At room temperature (530°R) without pressure:

\[ M_{RT} = \frac{I}{c} \left[ Y.S. \right] \]
\[ = \left( \frac{0.744}{1.173} \right) \left[ 150,000 \right] \]
\[ = 95,000 \text{ lb. in.} \]

The bolts must accommodate 75% of this effect,
\[ F_{BM_{RT}} = (0.075) \frac{2M_{RT}}{R_{I}N_{B}} = (0.75) \frac{2 \times (95,000)}{1.770(11)} = 7320 \text{ lbs.} \]

16) For task 7.1 criteria 3, determine the bolt load \( (F_{BDP2+M}) \) at DP2 loading plus 75% of the moment loading required to yield the mating pressurized line. From calculation 15a,

\[ T_B = 100^\circ R \quad T_f = 200^\circ R \]

\[ F_{BDP2+M} = F_{BDP2} + F_{BM200} \]

\[ = 8250 + 7270 = 15,520 \text{ lbs.} \]

From the computer output, the slope of the bolt head face is found from the change in axial deflections at nodes 58 and 66 for \( F_{BDP2} = 8250 \text{ lbs.} \):

\[ \Delta x_{66} = -0.0038872 \]

\[ \Delta x_{58} = +0.0008513 \]

\[ \Delta x_{\text{net}} = \Delta x_{66} - \Delta x_{58} = -0.0047385 \text{ in.} \]

\[ \theta \approx \frac{\Delta x_{\text{net}}}{h_f} = -0.0047385 / 0.970 = 0.00488 \]

For \( F_{BDP2+M} = 15,520 \text{ lbs.} \):

\[ \theta \approx (0.00488) \frac{15,520}{8250} = 0.00918 \]
\[ S_b = \frac{2 \theta E c_B}{L_B} \]  
\[ = \frac{2 (0.00914) 30.9 \times 10^6 (0.208)}{1.710} \]  
\[ = 69,000 \text{ psi} \]

Bending and tension interaction requires:

\[ \frac{S_t}{Y.S.} + \frac{S_b}{1.3 \text{ Y.S.}} \leq 1 \]  
(Eqn. 3.3.5 without 0.85% Y.S. requirement)

\[ \frac{S_{t_{\text{max}}}}{175,000} + \frac{69,000}{1.3 (175,000)} = 1 \]

\[ S_{t_{\text{max}}} = 122,000 \text{ psi} \]

\[ F_{B_{\text{DP2+M}}} = S_{t_{\text{max}}} A_B \]
\[ = 122,000 (0.136) \]
\[ = 16,600 \text{ lbs. permissible} \]

Have \( F_{B_{\text{DP2+M}}} = 15,520 \text{ lbs. imposed} \)

If 10 bolts were used:

\[ F_{B_{\text{DP2+M}}} \approx \frac{11}{10} (15,520) = 17,100 \text{ lbs.} \]

\[ N_B = 11 \text{ preferable} \]

17) In addition to task 7.1 criteria 3, a check is made to determine the room temperature absolute maximum bolt load (\( F_{B_{\text{RT,abs max}}} \))
from $F_{BRT_{\text{max}}}$ loading plus 75% of the moment loading required to yield the unpressurized mating line. From calculation 15b,

$$T_B = T_f = 530^\circ R$$

$$F_{BRTabs_{\text{max}}} = 8307 + 7320 = 15,627 \text{ lbs.}$$

$$\theta \approx (0.00488) \frac{15,627}{8250} = 0.00925$$

$$S_b = \frac{2 \theta E c_B}{L_B} = \frac{2 (0.00925) 30.9 \times 10^6 (0.208)}{1.710} = 69,400 \text{ psi}$$

$$\frac{S_t}{\text{Y.S.}} + \frac{S_b}{1.3 \text{ Y.S.}} \leq 1$$

$$\frac{S_{tRT \text{ max}}}{150,000} + \frac{69,400}{1.3 (150,000)} = 1$$

$$S_{tRT_{\text{max}}} = 96,600 \text{ psi}$$

$$F_{BRTabs_{\text{max}}} = 96,600 (0.136) = 13,130 \text{ lbs. permissible}$$

$$\neq 15,627 \text{ lbs. imposed}$$

$$N_E = \frac{15,627}{13,130} (11) \approx 13 \text{ required for calculation 17 criteria (see 19, observation a)}$$

18) Burst and proof check of mating plumbing:

a) Line limited by burst criteria:
\[ t_{W_{\text{min}}} = \frac{P_b R_I}{(U.T.S.)} \]

Where:

- \( P_b \) = (1.5) \( p_1 \), burst pressure
- \( U.T.S. \) = tube material ultimate tensile strength at 200\(^\circ\)R
- \( R_I \) = \( D_I / 2 = 1.966 / 2 = 0.983 \)
- \( t_{W_{\text{min}}} = \frac{1.5 (6900) (0.983)}{(210,000)} \)
  = 0.0485, have 0.190, OK

b) Line limited by proof criteria:

\[ t_{W_{\text{min}}} = \frac{P_p R_I}{(Y.S.)} \]

Where:

- \( P_p \) = 1.34 \( p_1 \), fracture mechanics proof pressure
- \( Y.S. \) = tube material 0.2\% yield strength at 200\(^\circ\)R
- \( t_{W_{\text{min}}} = \frac{1.34 (6900) (0.983)}{167,000} \)
  = 0.0545, have 0.190, OK

19) Observations:

a) The bolt strength shortage of calculation 17) can be considered acceptable in view of the following:
1. High safety factors were previously incorporated into the design

2. The added cost and weight of additional bolts cannot be justified

3. The extreme moment loading can be reduced by selectively fitting the end points for a given line installation.

b) After development efforts, in all probability the line wall thickness \( t_w \) will be reduced to approximately 0.060 inch (see calculation 18b). This reduces the moment required to yield the line with a subsequent reduction in design bolt load. The flange and bolts can then be finalized to a lighter flight-weight configuration.

c) For comparison, the same flange of the conventional "L" type configuration requires the following bolt loads not considering the additional moment loading criteria:

\[
\begin{align*}
F_{BDP1} &= 6340 \text{ lbs.} \\
F_{BR_{\text{min}}} &= 5350 \text{ lbs.} \\
F_{BR_{\text{max}}} &= 6700 \text{ lbs.} \\
F_{BDP2} &= 8920 \text{ lbs.}
\end{align*}
\]

d) For comparison, the same cantilever flange as the problem except for minor geometry changes with:
Ds = 2.391 in.
Dsp = 2.284 in.
h_s = 0.1215 in.
t_s = 0.082 in.
F_s = 2520 in.
Δ_sp = 0.0010073 in.

gave the following results:
for F_B = 10,000 lbs.,
    \[ t_F = 0.896 \text{ in.} \quad ΔF_B = -860 \text{ lbs.} \]
for F_B = 12,532 lbs.,
    \[ t_F = 0.908 \text{ in.} \quad ΔF_B = -795 \text{ lbs.} \]