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HUNTSVILLE, ALABAMA
DESIGN CRITERIA FOR LOW PROFILE FLANGE CALCULATIONS. (FINAL REPORT)

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Huntsville, AL

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DESIGN CRITERIA
FOR LOW PROFILE FLANGE CALCULATIONS

FINAL REPORT

March 1973

Contract NAS8-28614

Prepared for National Aeronautics and Space Administration
Marshall Space Flight Center, Alabama 35812

by

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NOTICE

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FOREWORD

This document is the final report for Contract NAS8-28614, "Design Criteria for Low Profile Flange Calculations," needed for the establishment of a Low Profile Flange Standard. Low Profile Flanges are characterized by featuring a small width but large height. Testing of Low Profile Flanges showed their superiority in performance, weight, and envelope volume in comparison to commonly used flanges for space application.

The work was initiated by the Layout and Assembly Engineering Branch, Engineering Division, Astronautics Laboratory of the NASA, Marshall Space Flight Center, in a joint effort with the Lockheed Missiles and Space Company, Inc., Huntsville Research and Engineering Center.

The primary objective of this effort was to evaluate the existing design procedure shown in the publication "Application of Low Profile Flange Design for Space Vehicles," and other flange design literature to establish a standard for Low Profile Flange calculations.

The period of performance of this study was from May 18, 1972, to March 22, 1973.

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**Appendixes**

- **A** Summary of the Design Procedure A-1
- **B** Summary of the Analysis Method B-1
- **C** Input Instructions for Design and Analysis Program C-1
Section 1
INTRODUCTION

The purpose of this study was to develop an analytical method and a design procedure to design flanged separable pipe connectors based on the previously established algorism for calculating low profile flanges. These flanges demonstrated their superiority with respect to leak-tightness and weight savings in comparison with other flanges used for space application.

When the low profile flange was first considered for space vehicle and launch application no design procedure was established and conventional flange design methods were used for the basic analysis. To remedy the situation Prastrofer (Ref. 1) devised a simple but effective design procedure considering the strength of the flange ring cross section as the design criteria.

It has been shown by Schwaigerer (Ref. 2) and through experiments by Bühner et al. (Ref. 3) and Haenle (Ref. 4), that there is a major contribution of the adjacent tube wall to the strength of a flanged connection. If one plots the flange roll angle $\chi$ versus the applied moment one finds a gradual decline of the slope of the curve (Fig. 1-1). This points to the existence of a plastic hinge at the most highly stressed section of the tube. The location of the plastic hinge is close to the neck of the flange, depending, in part, on the variation of the wall thickness of the tube in the area of the neck (Fig. 1-2).

Bühner et al. (Ref. 3) present a large number of data relating to the performance of flanges after the formation of the plastic hinge. The comparison for flanges with identical cross-section (Fig. 1-3) reveals that the next best choice to a conventional design with conical hub is one with a fillet (c). This comparison is not realistic, though, since design (b) can be replaced by a much
Fig. 1-1 - Applied Moment vs Roll Angle of the Flange

Fig. 1-2 - Location of the Plastic Hinge
<table>
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| m_{F_1} | 0.474 | 1.0 | 0.587 | 0.332 |
| m_{F_2} | 0.387 | 1.0 | 0.635 | 0.403 |

* 0.2% permanent set at S
** 1 deg permanent roll χ

Fig. 1-3 - Comparison of Structural Performance Based on Identical Flange Cross Section (Ref. 3)
narrower one of type (c), thus reducing the applied moment and therefore not requiring the larger moment capacity available with type (b). This is one of the advantages of the low profile flange which leads to the attendant weight saving, weight being proportional to the cross-sectional area and the centroidal radius. It also should be remembered that the strength of a flange increases approximately linearly with the flange width, b, but quadratically with the flange height, h, while the stiffness (resistance against roll) increases even cubically with h. This explains the better performance of the low profile flange over conventional wide profile flanges.

Thus, the advantage of the low profile flange is seen as being twofold: first, to reduce the lever arm, e, between the gasket and bolt circle, and second, to have the material of the flange where it is most effective, i.e., have the height, h, larger than the width, b (see Fig. 1-2).

Most of the available data on flange performance in the plastic range has been devoted to designs in steel at moderate temperatures. Steel has, however, a distinct yield point in its stress-strain diagram as compared to aluminum or titanium. The development of a plastic hinge for the latter materials at different temperatures would be a most interesting subject for further experimental investigations since the flange design method in this report is partially based on the assumption of a plastic hinge.

The plastic design method has been made part of the German flange design code DIN 2505 (Ref. 5), whereas American practice is based on an elasticity approach (Refs. 6 and 7). The use of the plastic design method is valid when the material is capable of undergoing large strains without fracture. The plastic method assumes a ductile failure. If a brittle failure is the predominant mode such as for certain high strength steels then the elastic method is more suitable.
1.1 DESIGN CRITERIA

The condition for sufficient strength of a structural component requires that

$$\sigma_{e_{\text{max}}} = \frac{K}{(F.S.)}$$

where $\sigma_{e_{\text{max}}}$ at the maximum equivalent stress, $K$ at the reference strength of the material, and (F.S.) is the factor of safety (with or without subscripts). In this paragraph these three quantities are briefly reviewed.

(a) Maximum Equivalent Stress: The computation of the maximum equivalent stress, $\sigma_{e_{\text{max}}}$, to be compared with the uniaxial material strength is based on the type of expected failure. For a failure associated with plastic deformation (yielding) or fatigue, the hypothesis of the limit of the elastic distribution energy by Huber (Ref. 8) and von Mises (Ref. 9) is used. The equivalent stress is

$$\sigma_e = \frac{1}{\sqrt{2}} \sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}$$

where $\sigma_1$, $\sigma_2$, $\sigma_3$ are the principal stresses. For components subjected to high tensile stresses, i.e., if $\sigma_1 > 0$, the equivalent stress is

$$\sigma_e = \sigma_1$$

This failure mode is fracture.
A third equivalent stress occasionally considered is the one defined by Tresca (Ref. 10) and is used for shear failures,

\[ \sigma_e = \sigma_{\text{max}} - \sigma_{\text{min}} = 2\tau_{\text{max}}. \]  

In the development of the flange design procedure the Huber and von Mises hypothesis is used.

(b) **Material Strength:** The material strength \( K \) to be used in Eq. (1.1) depends on the type of failure envisioned and must correspond to the type of equivalent stress \( \sigma_{e_{\text{max}}} \) computed. The two most frequently encountered types of uniaxial stress-strain diagrams are shown on Fig. 1-4. Diagram (a) has a distinct yield point with the tensile yield strength \( F_{\text{ty}} \). The ultimate tensile strength is \( F_{\text{tu}} \). Diagram (b), on the other hand, has a gradual change in slope requiring the definition of a yield strength from permanent strain considerations. Typically, the yield strength is \( F_{\text{ty}} = F_{\text{y}} \) where \( F_{\text{y}} \) is the 0.2% stress at permanent set. If \( F_{\text{tu}} \) is much larger than \( F_{\text{y}} \), the definition of the yield strength may be based on \( F_{0.5} \) or \( F_{1.0} \). This is the case with highly ductile materials. For the subsequent use in a design formula the stress-strain diagram is replaced by an idealized diagram as shown on Fig. 1-5 which could be called elastic – ideally plastic. This diagram must specify, however, a limit of its validity by giving a maximum allowable strain, \( \varepsilon_{\text{max}} \). A component which has been designed according to a plastic design method, such as the one proposed in this report for flanges, needs to be checked for strains under the design conditions, i.e., the ultimate load. This load condition will be discussed later in more detail.

When temperature effects are to be considered, the appropriate strength values at the design temperature must be used. Similarly, fatigue strength and creep rupture strength can be the dominant strength values to be considered.
Fig. 1-4 - Typical Stress-Strain Diagrams

(a) Steel

(b) Aluminum

Fig. 1-5 - Idealized Stress-Strain Diagram
(c) **Safety Factors:** The proper use of the safety factors is important in a complex system such as a bolted connection, but it also leaves room for different design philosophies. For a pipe system three pressure levels are considered: operating pressure, proof pressure and burst pressure. Usually the proof pressure is $1\frac{1}{2}$ times the operating pressure, and the burst pressure two times the operating pressure. These factors are implied safety factors against uncertainties in the prediction of the operating pressure due to pressure surges at valve closure or vehicle vibrations accompanied by pressure oscillations. The design pressure is mostly chosen as to be proof pressure, that is, the structure is to be able to withstand the proof pressure without damage. That condition occurs at least once in the lifetime of the structure. If an elastic design method is employed and only stress peaks are checked, a small safety factor of say 1.2 against yielding at critical points is sufficient. For the plastic design method, however, instead of a safety factor an ultimate factor of at least 1.5 is used by which the load is multiplied. This magnified load is called the ultimate load. For example, the maximum applied moment on the flange due to the proof pressure condition is $m_F$ and the ultimate moment that is to be carried is

$$m_{Fu} = (F.S.) m_F$$  \hspace{1cm} (1.5)

A structural capacity has to be provided for $m_{Fu}$. This capacity can be expressed as

$$m_{Fu} = Z_F F_{ty}$$  \hspace{1cm} (1.6)

where $F_{ty}$ is the tensile yield strength of the material and $Z_F$ is the combined section modulus of the flange and the adjacent tube after a plastic hinge has formed in the neck.
The flange is then still in an elastic state of stress with only the extreme fibers yielded. Schwaigerer suggests in Ref. 2 that the flange cross section, too, should be considered being in a fully plastic state of stress. This condition is, however, somewhat exaggerated.

Other safety factors are needed to cover uncertainties such as in the computation of the stresses (using average values) and uncertainties in the material properties, i.e., the values of K chosen for the different materials. Possibly the material properties of the flange and the bolts are much more accurately known than those of the gasket, requiring a higher gasket safety factor.

The total safety factor may be defined as the product of the individual safety factors

\[(F.S.) = (F.S._1 \cdot (F.S._2 \cdot (F.S._3 \cdot \ldots \cdot (F.S._n)^1 (1.7)\]

In this study the values for the safety factors have been chosen more or less arbitrarily. Also, some design formulas such as the ones for the tube wall thickness contain implied safety factors. These are explained where they occur in Section 2.

1.2 PAST EXPERIENCE WITH LOW PROFILE FLANGES

The initial idea for the low profile flange concept came from Boon and Lok (Ref. 11) which was taken up later by Prasthofer (Ref. 12) for the design of launch vehicle pipe connections. Qualification testing reported by (Ref. 13) and experimental stress analysis of one photoelastic model configuration by Kubitza and Hearne (Ref. 14) showed the soundness of this flange concept. Design procedures for a similar type of flange have been established by Trainer et al. (Ref. 15), by Aerojet General (Ref. 16) and Pratt and Whitney (Ref. 17) although not in a usable form. The latter procedures are tailored to specific seal configurations and are therefore not generally applicable. In
addition most of the existing methods require an excessive amount of computations if carried out manually. An automated design study reported by Rathbun (Ref. 18) is set up to produce wide profile flanges with conical hubs, being undesirable in the context of the low profile concept.

Previous design methods were not definite on the minimum spacing requirements for the bolts. Bolt spacing is the driving parameter for the flange width. Minimum bolt spacing assures a low profile flange. The spacing requirements are discussed in more detail in Section 2. It should be noted here that the bolts should be as small as possible and be located as closely to the tube wall as can be accomplished within the constraint of wrench clearance requirements. This can be accomplished by providing countersunk spot faces for the bolts in Configuration (c) on Fig. 1-3 or by a machined groove as in Configuration (d). The latter sacrifices some ultimate moment capacity. Configuration (c) has been used exclusively, so far, in all past low profile flange designs. The introduction of stress peaks around the spot faces has been impressively demonstrated in Ref. 14. It is therefore suggested to supply contoured washers that would eliminate countersinking, possibly even with spherical glide surfaces to accommodate rotations of the bolt head or nut with respect to the flange. This brief summary may suffice to characterize past experience.

Seen in the light of the long history of flange design and analysis methods, beginning with Westphal's classical paper (Ref. 19) of 1897, the methods presented in this report constitute the logical extension of current ideas.
Section 2
DESIGN PROCEDURE

In this section the individual steps of the design procedure are derived. Some of the steps have several alternatives and the most suitable ones are selected. A summary of the design procedure to serve as a guideline for manual computations and as an outline for the design section of the computer program is given in Appendix A.

2.1 TUBE DESIGN

The computation of the required wall thickness for a cylindrical tube under internal pressure is based on the stresses. For a thick-walled tube these are

\[ \sigma_r = \frac{-(b/r)^2 + 1}{(b/a)^2 - 1} \cdot p, \quad (2.1) \]

\[ \sigma_x = \frac{1}{(b/a)^2 - 1} \cdot p, \quad (2.2) \]

\[ \sigma_\phi = \frac{(b/r)^2 + 1}{(b/a)^2 - 1} \cdot p. \quad (2.3) \]

The coordinate system is defined on Fig. 2-1. The equivalent stress is

\[ \sigma_e = \frac{1}{\sqrt{2}} \sqrt{(\sigma_r - \sigma_\phi)^2 + (\sigma_\phi - \sigma_x)^2 + (\sigma_x - \sigma_r)^2} = \frac{\sqrt{3} (b/a)^2}{(b/a)^2 - 1} \cdot p. \quad (2.4) \]
Fig. 2-1 - Coordinate System for a Thick-Walled Tube
with a maximum at $r = a$,

$$
\sigma_{\text{emax}} = \frac{\sqrt{3} (b/a)^2}{(b/a)^2 - 1} \ p.
$$

(2.5)

If $\sigma_x$ is larger than given by Eq. (2.2), say $\sigma' > \sigma_x$, then

$$
\sigma_e = \frac{\sqrt{3} (b/r)^4 + (1 - \sigma'/\sigma_x)^2}{(b/a)^2 - 1} \ p.
$$

(2.6)

The average equivalent stress is generally

$$
\sigma_{\text{av}} = \frac{(b/a)^2 + 1}{(b/a)^2 - 1} \ p.
$$

(2.7)

When the equivalent stress $\sigma_e$ according to Eq. (2.4) is equal to the yield strength $F_{ty}$, which is assumed to be a constant for mathematical simplicity, then

$$
\bar{\sigma}_r = -\frac{2}{\sqrt{3}} F_{ty} \ln \frac{b}{r}
$$

(2.8)

$$
\bar{\sigma}_x = \frac{2}{\sqrt{3}} F_{ty} \left( \frac{1}{2} - \ln \frac{b}{r} \right)
$$

(2.9)

$$
\bar{\sigma}_\phi = \frac{2}{\sqrt{3}} F_{ty} \left( 1 - \ln \frac{b}{r} \right)
$$

(2.10)

The equivalent stress in this case (Ref. 20, p. 106) is

$$
\bar{\sigma}_e = \frac{2}{\sqrt{3}} \ln \left( \frac{b}{r} \right)
$$

(2.11)
The fully elastic and fully plastic states of stress are shown on Fig. 2-2. The reversal of the stresses when going from the elastic to the plastic state of stress is obvious.

In Ref. 2, p. 29, it has been shown that up to a ratio

\[ \frac{b}{a} = 1.2 \]  \hspace{1cm} (2.12)

the average equivalent stress \( \sigma_{e_{\text{av}}} \) can be used for the design of a tube since it is almost equal to the maximum equivalent stress \( \sigma_{e_{\text{max}}} \) and to the equivalent stress of a fully plastic state, i.e.,

\[ \frac{P}{\sigma_{e_{\text{max}}}} \approx \frac{P}{\sigma_{e_{\text{av}}}} \approx \frac{P}{\sigma_{e}} \quad \text{if} \quad \frac{b}{a} < 1.2 \]  \hspace{1cm} (2.13)

where

\[ \frac{P}{\sigma_{e_{\text{max}}}} = \frac{(\frac{b}{r})^2 - 1}{\sqrt{3}(\frac{b}{a})^2} \]  \hspace{1cm} (2.14)

\[ \frac{P}{\sigma_{e}} = \frac{2}{\sqrt{3}} \ln(\frac{b}{r}) \]  \hspace{1cm} (2.15)

and

\[ \frac{P}{\sigma_{e_{\text{av}}}} = \frac{2t}{2a + t} \]  \hspace{1cm} (2.16)

The latter equation (2.16) was derived from Eq. (2.7) with \( t = b-a \), where \( t \) is the wall thickness. When the strength design criterion Eq. (1.1), is applied the relation becomes

\[ P = \frac{2t}{2a + t} \frac{K}{(F.S.)} \]  \hspace{1cm} (2.17)
Fig. 2-2 - States of Stress in a Thick-Walled Cylindrical Shell Under Internal Pressure
which can be solved for the wall thickness, giving

\[ t = \frac{p a}{K} \left(\frac{F.S.}{2} - \frac{P}{2}\right). \quad (2.18) \]

This formula is the basis for most pressure vessel design codes, for example, the American "ASME Pressure Vessel Code" (Ref. 21) or the German "Dampfkessel-Bestimmungen" (Ref. 22).

In order to accommodate wall thickness tolerances \( \Delta t \) and a factor \( \psi \) for weakening by welds the wall thickness \( t \) in Eq. (2.12) is replaced by

\[ \bar{t} = (t - \Delta t) \psi \quad (2.19) \]

which gives the formula for the thickness as

\[ t = \frac{p a}{K} \left(\frac{F.S.}{2} - \frac{P}{2}\right) \psi + \Delta t \quad (2.20) \]

This is the formula used in Ref. 22.

In Ref. 21 this equation has been modified by taking

\[ t = \frac{1.1 p a}{K} \left(\frac{F.S.}{2} - 0.4 p\right) \quad (2.21) \]

while a formula used by Pratt & Whitney (Ref. 17) is

\[ t = \frac{p a}{K} \left(\frac{F.S.}{2}\right) \psi + 2 \Delta t \quad (2.22) \]
The simplest formula, based only on circumferential stress, is

\[ t = \frac{pa}{K (F.S.)} \]  \hspace{1cm} (2.23)

The weakening factor \( \psi \) has been used in the order of

\[ 0.70 \leq \psi \leq 1.00 \]  \hspace{1cm} (2.24)

For the weakening by a weld the stress component perpendicular to the weld is most important. If a weld is at an angle \( \gamma \) with the cylinder axis then the stress perpendicular (normal) to the weld is given by

\[ \sigma_n = \sigma_x \sin^2 \gamma + \sigma_\psi \cos^2 \gamma \]  \hspace{1cm} (2.25)

Therefore in Eqs. (2.19) and (2.20) the weakening factor is generally

\[ \psi = \frac{2\psi'}{1 + \cos^2 \gamma} \]  \hspace{1cm} (2.26)

The factor \( \psi' \) has to be determined by test.

Considerations other than internal pressure, such as creep, vibrations, bending and shear, may influence tube design. These are briefly reviewed in the following paragraphs.

**Creep:** To simplify the derivation, only steady state creep is considered. This problem was studied in depth by several investigators (Refs. 23 through 27). Let the material law be given by

\[ \dot{\varepsilon}_e = B \sigma_e^n \]  \hspace{1cm} (2.27)

where \( \varepsilon_e \) is the equivalent strain defined by
\[ \varepsilon_e^2 = \frac{\sqrt{2}}{3} \sqrt{(\varepsilon_r - \varepsilon_\theta)^2 + (\varepsilon_\phi - \sigma_x)^2 + (\varepsilon_x - \varepsilon_r)^2} \]  

(2.28)

and the creep constant \( B \) is

\[ B = \beta e^{\alpha T} \]  

(2.29)

where \( \alpha \) and \( \beta \) are independent of temperature, \( T \) is the absolute temperature and \( e \) is the base of the natural logarithm. The stresses in this case, corresponding to Eqs. (2.1) through (2.3) are given by

\[ \sigma_r = \frac{\left( \frac{b}{r} \right)^{2/n} - 1}{p} \]  

(2.30)

\[ \sigma_x = \frac{(1-n) \left( \frac{b}{r} \right)^{2/n} + 1}{n \left( \frac{b}{a} \right)^{2/n} - 1} \]  

(2.31)

\[ \sigma_\phi = \frac{(2-n) \left( \frac{b}{r} \right)^{2/n} + 1}{n \left( \frac{b}{a} \right)^{2/n} - 1} \]  

(2.32)

It can be seen that for \( n = 1 \) the elastic case of Eqs. (2.1) through (2.3) is obtained. Using the foregoing relations the accumulation of strains can be computed for the lifetime of the tube, thus serving as a design criterion for the selection of the tube thickness.

From Eqs. (2.30) through (2.32) the maximum equivalent stress at the inside of the tube is, similar to Eq. (2.5),
To design a tube for a given lifetime until creep rupture occurs, the ultimate equivalent strength is computed by

\[ \sigma_{u_{\text{max}}} = (F.S.) \sigma_{e_{\text{max}}} \]  

(2.34)

and from a plot of the equivalent stress versus the creep parameter, \( P \), the value of \( P \) for \( \sigma_{u_{\text{max}}} \) of Eq. (2.34) is obtained.

The creep parameter, if for example, Larson and Miller's (Ref. 28) formulation is used, is defined by

\[ P = c_1 T \left[ \log (t_{\text{rupt}}) + c_2 \right] \]  

(2.35)

which can be solved for \( t_{\text{rupt}} \) giving

\[ t_{\text{rupt}} = \text{antilog} \left( \frac{P}{c_1 T} - c_2 \right) . \]  

(2.36)

This excursion into creep analysis methods may suffice.

\underline{Vibration:} The oscillations involving propellant feedlines, engines and longitudinal structural modes of a launch vehicle are described in a paper by Ryan et al. (Ref. 29). To cope with the problem from a design point of view the following approach may be taken

\[ t = \frac{(\Delta p)a}{K \frac{F}{(F.S.)} - \frac{\Delta p}{2}} + c \]  

(2.37)
where $K_F$ is the appropriate fatigue strength of the tube material for the stress ratio $R = 0$ assuming a maximum internal pressure of

$$\Delta p = p_{\text{max}} - p_{\text{min}}$$  \hspace{1cm} (2.38)

The maximum pressure $p_{\text{max}}$ would be determined from a vibration analysis such as the one cited.

**Bending and Shear:** The presence of a bending moment and a shear force introduce stresses into the tube wall which may control the design. The expressions for the stresses in terms of a bending moment $M_1$ and a shear force $S_1$ are

$$\sigma_x = \frac{M_1}{\pi R^2 t} \cos \phi$$  \hspace{1cm} (2.39)

and

$$\tau_{x\phi} = \frac{S_1}{\pi R t} \sin \phi$$  \hspace{1cm} (2.40)

where $\phi$ is the circumferential coordinate. From Eq. (2.39) an equivalent axial force of

$$n_x = \frac{M_1}{\pi R^2}$$  \hspace{1cm} (2.41)

being the maximum, should be used in addition to the axial stresses resulting from internal pressure.

### 2.2 BOLT SIZE

The essential idea of the low profile flange concept is to have a large number of small diameter bolts. Therefore, the design calculation is started by selecting a bolt diameter and then trying to accommodate the number of bolts required to keep a leakproof connection. To find a basis for selecting the bolt diameter it is assumed that the wall thickness computed previously can
support an internal pressure of

\[ p = \frac{t}{a} \frac{K_T}{(F.S.)_T} \]  

(2.42)

which is obtained from the simplified formula given by Eq. (2.23), \( K_T \) being the tube strength. On the other hand the entire pressure has also to be carried by the bolts, i.e., the bolt force

\[ P_B = \pi a^2 p \]  

(2.43)

has to be equal to

\[ P_B = \frac{2 \pi r_B}{s} \frac{\pi d_B^2}{4} \frac{K_B}{(F.S.)_B} \]  

(2.44)

where \( r_B \) is the bolt circle radius, \( s \) is the spacing of the bolts, \( d_B \) is the nominal bolt diameter and \( K_B \) the bolt strength. The bolt strength is usually chosen to be the ultimate tensile strength, together with the appropriate safety factor. When the pressure value from Eq. (2.42) is substituted into Eq. (2.43),

\[ P_B = \pi a^2 \left[ \frac{t}{a} \frac{K_T}{(F.S.)_T} \right] \]  

(2.45)

and Eqs. (2.43) and (2.44) set equal, then

\[ \pi a^2 \left[ \frac{t}{a} \frac{K_T}{(F.S.)_T} \right] = \frac{2 \pi r_B}{s} \frac{\pi d_B^2}{4} \frac{K_B}{(F.S.)_B}, \]  

(2.46)

or, after rearranging,

\[ \frac{d_B}{t} = \frac{2}{\pi} \frac{s}{d_B} \frac{a}{r_B} \frac{K_T}{(F.S.)_T} = \frac{K_B}{(F.S.)_B} \]  

(2.47)

2-11
An estimate for the ratio $d_B/t$ is arrived at by assuming the following ratios

$$
\frac{K_B}{(F.S.)_B} / \frac{K_T}{(F.S.)_T} \approx 1.5 , \quad (2.48)
$$

$$
\frac{r_B}{a} \approx 1.1 , \quad (2.49)
$$

$$
\frac{s}{d_B} \approx 2.5 , \quad (2.50)
$$

so that

$$
\frac{d_B}{t} \approx \frac{2}{\pi} \left( \begin{array}{c}
(2.5) \\
(1.1) \\
1.5
\end{array} \right) \approx 1.0 . \quad (2.51)
$$

This is only an initial estimate to get the design calculations started. The final bolt diameter will be determined when checked against all the other requirements to be discussed later.

When the initial selection of the bolt diameter has been made the requirements for wrench clearance and adequate spacing of the bolts from each other and from the edge of the flange have to be considered. On Figs. 2-3 through 2-5, and Tables 2-1 through 2-3, which were taken from Ref. 18, non-dimensional values for

$$
\eta_0 = \frac{s}{d_B} \quad (2.52)
$$

$$
\eta_1 = \frac{e_1}{d_B} \quad (2.53)
$$

and

$$
\eta_2 = \frac{e_2}{d_B} \quad (2.54)
$$
Figure 2-3 - Design Parameters for Open-End Wrenching
Fig. 2-4 - Design Parameter for Socket Wrenching
Fig. 2-5 - Design Parameters for Internal Wrenching
Table 2-1
BOLT TABLE FOR OPEN-END WRENCHING (REF. 18)

<table>
<thead>
<tr>
<th>Size</th>
<th>( d_B ) (in.)</th>
<th>( \eta_0 )</th>
<th>( \eta_1 )</th>
<th>( \eta_2 )</th>
<th>( A_{OB} ) (in(^2))</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.2500</td>
<td>3.00</td>
<td>2.00</td>
<td>1.50</td>
<td>0.03182</td>
</tr>
<tr>
<td>2</td>
<td>0.3125</td>
<td>2.60</td>
<td>1.80</td>
<td>1.40</td>
<td>0.05243</td>
</tr>
<tr>
<td>3</td>
<td>0.3750</td>
<td>2.67</td>
<td>1.67</td>
<td>1.33</td>
<td>0.07749</td>
</tr>
<tr>
<td>4</td>
<td>0.4375</td>
<td>2.57</td>
<td>1.57</td>
<td>1.29</td>
<td>0.10631</td>
</tr>
<tr>
<td>5</td>
<td>0.5000</td>
<td>2.50</td>
<td>1.62</td>
<td>1.24</td>
<td>0.14190</td>
</tr>
<tr>
<td>6</td>
<td>0.5625</td>
<td>2.45</td>
<td>1.56</td>
<td>1.22</td>
<td>0.18194</td>
</tr>
<tr>
<td>7</td>
<td>0.6250</td>
<td>2.40</td>
<td>1.50</td>
<td>1.20</td>
<td>0.22600</td>
</tr>
<tr>
<td>8</td>
<td>0.7500</td>
<td>2.33</td>
<td>1.49</td>
<td>1.08</td>
<td>0.33446</td>
</tr>
<tr>
<td>9</td>
<td>0.8750</td>
<td>2.35</td>
<td>1.43</td>
<td>1.07</td>
<td>0.46173</td>
</tr>
<tr>
<td>10</td>
<td>1.0000</td>
<td>2.25</td>
<td>1.37</td>
<td>1.06</td>
<td>0.60574</td>
</tr>
<tr>
<td>11</td>
<td>1.1250</td>
<td>2.22</td>
<td>1.33</td>
<td>1.00</td>
<td>0.76327</td>
</tr>
<tr>
<td>12</td>
<td>1.2500</td>
<td>2.25</td>
<td>1.40</td>
<td>1.00</td>
<td>0.92905</td>
</tr>
<tr>
<td>13</td>
<td>1.3750</td>
<td>2.23</td>
<td>1.36</td>
<td>1.00</td>
<td>1.15488</td>
</tr>
<tr>
<td>14</td>
<td>1.5000</td>
<td>2.17</td>
<td>1.33</td>
<td>1.00</td>
<td>1.40525</td>
</tr>
</tbody>
</table>

Legend:

- Size = size number of the bolt
- \( d_B \) = nominal diameter of the bolt
- \( \eta_0 = \frac{s}{d_B} \)
- \( \eta_1 = \frac{e_1}{d_B} \) spacing parameter (dimensionless)
- \( \eta_2 = \frac{e_2}{d_B} \)
- \( A_{OB} \) = stress area of one bolt
- \( d_{hole} = d_B + 0.005 \) in.

2-16
Table 2-2
BOLT TABLE FOR SOCKET WRENCHING (REF. 18)

<table>
<thead>
<tr>
<th>Size</th>
<th>$d_B$ (in.)</th>
<th>$\eta_0$</th>
<th>$\eta_1$</th>
<th>$\eta_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.2500</td>
<td>2.76</td>
<td>1.60</td>
<td>1.40</td>
</tr>
<tr>
<td>2</td>
<td>0.3125</td>
<td>2.53</td>
<td>1.50</td>
<td>1.28</td>
</tr>
<tr>
<td>3</td>
<td>0.3750</td>
<td>2.37</td>
<td>1.33</td>
<td>1.20</td>
</tr>
<tr>
<td>4</td>
<td>0.4375</td>
<td>2.26</td>
<td>1.25</td>
<td>1.14</td>
</tr>
<tr>
<td>5</td>
<td>0.5000</td>
<td>2.18</td>
<td>1.20</td>
<td>1.10</td>
</tr>
<tr>
<td>6</td>
<td>0.5625</td>
<td>2.20</td>
<td>1.22</td>
<td>1.11</td>
</tr>
<tr>
<td>7</td>
<td>0.6250</td>
<td>2.22</td>
<td>1.25</td>
<td>1.12</td>
</tr>
<tr>
<td>8</td>
<td>0.7500</td>
<td>2.12</td>
<td>1.17</td>
<td>1.07</td>
</tr>
<tr>
<td>9</td>
<td>0.8750</td>
<td>2.28</td>
<td>1.31</td>
<td>1.14</td>
</tr>
<tr>
<td>10</td>
<td>1.0000</td>
<td>2.19</td>
<td>1.25</td>
<td>1.10</td>
</tr>
<tr>
<td>11</td>
<td>1.1250</td>
<td>2.14</td>
<td>1.22</td>
<td>1.07</td>
</tr>
<tr>
<td>12</td>
<td>1.2500</td>
<td>2.09</td>
<td>1.18</td>
<td>1.04</td>
</tr>
<tr>
<td>13</td>
<td>1.3750</td>
<td>2.00</td>
<td>1.16</td>
<td>1.02</td>
</tr>
<tr>
<td>14</td>
<td>1.5000</td>
<td>2.02</td>
<td>1.13</td>
<td>1.00</td>
</tr>
</tbody>
</table>

Legend:

Size  =  size number of the bolt

$d_B$  =  nominal diameter of the bolt

$\eta_0 = \frac{S}{d_B}$

$\eta_1 = \frac{e_1}{d_B}$  spacing parameter (dimensionless)

$\eta_2 = \frac{e_2}{d_B}$

$A_{OB}$  =  stress area of one bolt (see Table 2-1)

d$_{hole}$  =  $d_B + 0.005$ in.
Table 2-3
BOLT TABLE FOR INTERNAL WRENCHING (REF. 18)

<table>
<thead>
<tr>
<th>Size</th>
<th>(d_B) (in.)</th>
<th>(\eta_0)</th>
<th>(\eta_1)</th>
<th>(\eta_2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.2500</td>
<td>1.92</td>
<td>1.16</td>
<td>0.96</td>
</tr>
<tr>
<td>2</td>
<td>0.3125</td>
<td>1.86</td>
<td>1.09</td>
<td>0.93</td>
</tr>
<tr>
<td>3</td>
<td>0.3750</td>
<td>1.79</td>
<td>1.04</td>
<td>0.91</td>
</tr>
<tr>
<td>4</td>
<td>0.4375</td>
<td>1.80</td>
<td>1.03</td>
<td>0.91</td>
</tr>
<tr>
<td>5</td>
<td>0.5000</td>
<td>1.78</td>
<td>1.00</td>
<td>0.90</td>
</tr>
<tr>
<td>6</td>
<td>0.5625</td>
<td>1.76</td>
<td>0.98</td>
<td>0.89</td>
</tr>
<tr>
<td>7</td>
<td>0.6250</td>
<td>1.75</td>
<td>0.96</td>
<td>0.88</td>
</tr>
<tr>
<td>8</td>
<td>0.7500</td>
<td>1.68</td>
<td>0.91</td>
<td>0.84</td>
</tr>
<tr>
<td>9</td>
<td>0.8750</td>
<td>1.69</td>
<td>0.90</td>
<td>0.85</td>
</tr>
<tr>
<td>10</td>
<td>1.0000</td>
<td>1.67</td>
<td>0.89</td>
<td>0.84</td>
</tr>
<tr>
<td>11</td>
<td>1.1250</td>
<td>1.86</td>
<td>0.96</td>
<td>0.92</td>
</tr>
<tr>
<td>12</td>
<td>1.2500</td>
<td>1.67</td>
<td>0.87</td>
<td>0.83</td>
</tr>
<tr>
<td>13</td>
<td>1.3750</td>
<td>1.80</td>
<td>0.93</td>
<td>0.89</td>
</tr>
<tr>
<td>14</td>
<td>1.5000</td>
<td>1.65</td>
<td>0.85</td>
<td>0.82</td>
</tr>
</tbody>
</table>

Legend:

\(\text{Size}\) = size number of the bolt
\(d_B\) = nominal diameter of the bolt
\(\eta_0\) = \(\frac{S}{d_B}\)
\(\eta_1\) = \(\frac{e_1}{d_B}\) spacing parameter (dimensionless)
\(\eta_2\) = \(\frac{e_2}{d_B}\)
\(A_{oB}\) = stress area of one bolt (see Table 2-1)
\(d_{\text{hole}}\) = \(d_B + 0.005\) in.
are given. The distances $e_1$ and $e_2$ refer to a flange with machined spotfaces and are shown on Fig. 2-6. Also, the tables contain the stress area $A_{OB}$ of each bolt size. These were taken from Ref. 30 for the ISO-inch coarse thread series for $1/4 \leq d_B \leq 3/2$ inch. The corresponding recommended metric series is given in Tables 2-4 through 2-6, where $6.3 \leq d_B \leq 36.0$ mm ($0.2480 \leq d_B \leq 1.4173$ inch). Where 14 different sizes were used for the indicated diameter range in the ISO-inch series, only nine different sizes are given for the metric series. This series, however, is tentative and subject to further studies by the Industrial Fasteners Institute in Cleveland, Ohio.

The bolt tables given are not to be taken as definite data. They were merely used for the numerical example problems of this project. The design procedure and the corresponding program are configured to allow additional bolt data to be incorporated such as data for 8 or 12 point heads.

The diameter of the bolt hole is taken as $d_{\text{hole}} = d_B + 0.005$ inch (+0.1 mm). These clearances have been assumed to be able to compute numerical examples and are not to be taken as definite design data.

The spot face diameter is assumed as $d_{\text{spot}} = 2e_1$, where $e_1$ is given in the bolt tables. A fillet radius of $r_{\text{spot}} = 0.062$ inch (1.5 mm) is provided.

When a machined groove is selected both distances are $e_1 = e_2 = \eta_2 d_B$ as shown on Fig. 2-7.

The selection of the fillet radius on Fig. 2-6 and the groove radius on Fig. 2-7 is somewhat arbitrary. While the machined groove is intended to reduce stress concentrations due to notch effects at the neck, it cannot reduce the high stresses in the cylinder portion. The fillet on Fig. 2-6 is intended to do this. A basis for the size of the fillet radius can be found by considering the wavelength $L$ of the stress pattern along the shell meridian. This stress pattern alternates sinusoidally with exponentially decreasing amplitudes. The ratio of two successive amplitudes, considering only the edge disturbance

2-19
Table 2-4
METRIC BOLT TABLE FOR OPEN-END WRENCHING (REF. 18)

<table>
<thead>
<tr>
<th>i size</th>
<th>( d_B ) (mm)</th>
<th>( d_B ) (in.)</th>
<th>( \eta_0 )</th>
<th>( \eta_1 )</th>
<th>( \eta_2 )</th>
<th>( A_{oB_2} ) (mm²)</th>
<th>( A_{oB_2} ) (in.²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>6.300</td>
<td>0.2480</td>
<td>3.00</td>
<td>2.00</td>
<td>1.50</td>
<td>22.276</td>
<td>0.035</td>
</tr>
<tr>
<td>2</td>
<td>8.000</td>
<td>0.3150</td>
<td>2.85</td>
<td>1.90</td>
<td>1.43</td>
<td>36.126</td>
<td>0.055</td>
</tr>
<tr>
<td>3</td>
<td>10.000</td>
<td>0.3937</td>
<td>2.70</td>
<td>1.80</td>
<td>1.85</td>
<td>57.261</td>
<td>0.089</td>
</tr>
<tr>
<td>4</td>
<td>12.500</td>
<td>0.4921</td>
<td>2.59</td>
<td>1.68</td>
<td>1.28</td>
<td>91.524</td>
<td>0.142</td>
</tr>
<tr>
<td>5</td>
<td>16.000</td>
<td>0.6299</td>
<td>2.45</td>
<td>1.57</td>
<td>1.18</td>
<td>155.070</td>
<td>0.240</td>
</tr>
<tr>
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<td>20.000</td>
<td>0.7874</td>
<td>2.35</td>
<td>1.48</td>
<td>1.12</td>
<td>242.297</td>
<td>0.375</td>
</tr>
<tr>
<td>7</td>
<td>25.000</td>
<td>0.9843</td>
<td>2.28</td>
<td>1.40</td>
<td>1.07</td>
<td>382.801</td>
<td>0.593</td>
</tr>
<tr>
<td>8</td>
<td>30.000</td>
<td>1.1811</td>
<td>2.23</td>
<td>1.35</td>
<td>1.02</td>
<td>555.296</td>
<td>0.861</td>
</tr>
<tr>
<td>9</td>
<td>36.000</td>
<td>1.4173</td>
<td>2.19</td>
<td>1.33</td>
<td>1.01</td>
<td>809.423</td>
<td>1.255</td>
</tr>
</tbody>
</table>

Table 2-5
METRIC BOLT TABLE FOR SOCKET WRENCHING (REF. 18)

<table>
<thead>
<tr>
<th>i size</th>
<th>( d_B ) (mm)</th>
<th>( \eta_0 )</th>
<th>( \eta_1 )</th>
<th>( \eta_2 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>6.300</td>
<td>2.80</td>
<td>1.60</td>
<td>1.40</td>
</tr>
<tr>
<td>2</td>
<td>8.000</td>
<td>2.69</td>
<td>1.53</td>
<td>1.36</td>
</tr>
<tr>
<td>3</td>
<td>10.000</td>
<td>2.57</td>
<td>1.48</td>
<td>1.29</td>
</tr>
<tr>
<td>4</td>
<td>12.500</td>
<td>2.46</td>
<td>1.41</td>
<td>1.23</td>
</tr>
<tr>
<td>5</td>
<td>16.000</td>
<td>2.34</td>
<td>1.33</td>
<td>1.16</td>
</tr>
<tr>
<td>6</td>
<td>20.000</td>
<td>2.25</td>
<td>1.27</td>
<td>1.11</td>
</tr>
<tr>
<td>7</td>
<td>25.000</td>
<td>2.15</td>
<td>1.22</td>
<td>1.07</td>
</tr>
<tr>
<td>8</td>
<td>30.000</td>
<td>2.08</td>
<td>1.17</td>
<td>1.03</td>
</tr>
<tr>
<td>9</td>
<td>36.000</td>
<td>2.02</td>
<td>1.14</td>
<td>1.01</td>
</tr>
</tbody>
</table>
### Table 2-6
METRIC BOLT TABLE FOR INTERNAL WRENCHING (REF. 18)

<table>
<thead>
<tr>
<th>i_{size}</th>
<th>d_B (mm)</th>
<th>η_0</th>
<th>η_1</th>
<th>η_2</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>6.300</td>
<td>1.92</td>
<td>1.16</td>
<td>0.96</td>
</tr>
<tr>
<td>2</td>
<td>8.00</td>
<td>1.88</td>
<td>1.12</td>
<td>0.94</td>
</tr>
<tr>
<td>3</td>
<td>10.000</td>
<td>1.85</td>
<td>1.08</td>
<td>0.92</td>
</tr>
<tr>
<td>4</td>
<td>12.500</td>
<td>1.81</td>
<td>1.03</td>
<td>0.90</td>
</tr>
<tr>
<td>5</td>
<td>16.000</td>
<td>1.77</td>
<td>0.99</td>
<td>0.87</td>
</tr>
<tr>
<td>6</td>
<td>20.000</td>
<td>1.73</td>
<td>0.94</td>
<td>0.85</td>
</tr>
<tr>
<td>7</td>
<td>25.000</td>
<td>1.70</td>
<td>0.91</td>
<td>0.84</td>
</tr>
<tr>
<td>8</td>
<td>30.000</td>
<td>1.68</td>
<td>0.88</td>
<td>0.83</td>
</tr>
<tr>
<td>9</td>
<td>36.000</td>
<td>1.66</td>
<td>0.86</td>
<td>0.82</td>
</tr>
</tbody>
</table>
Fig. 2-6 - Low Profile Flange with Machined Spot Faces
Fig. 2-7 - Low Profile Flange with Machined Groove
introduced by the flange, is

\[
\frac{A_1}{A_2} = e^{2\pi} = e^{\frac{\rho L}{r_o}}, \quad (2.55)
\]

where

\[
\rho = \frac{4}{\sqrt{3(1-\nu^2)}} \frac{r_o^2}{t^2}. \quad (2.56)
\]

The radius of the cylinder middle surface, \( r_o \), is

\[
r_o = a + t/2 . \quad (2.57)
\]

From the logarithmic decrement

\[
\rho L/r_o = 2\pi \quad (2.58)
\]

the wavelength is

\[
L = 2\pi \frac{r_o}{\rho} \quad (2.59)
\]

The fillet radius should cover approximately one-eighth of this wavelength in order to reduce the shell stresses at the neck. Equations (2.55) through (2.58) are illustrated on Fig. 2-8. To simplify the design procedure, approximate fillet and groove radii are listed in Tables 2-7 and 2-8, respectively.

2.3 BOLT CIRCLE RADIUS AND FLANGE WIDTH

The magnitude of the bolt circle radius and the flange width are determined by the space required on the upper surface as shown on Figs. 2-6 and 2-7. In the case of machined spot faces a minimum distance \( c_1 \) from the tube wall is maintained to accommodate the tool for making the spot face. The formulas for the bolt circle radius can be written for the machined spot face,
Fig. 2-8 - Relation of Fillet Radius to Shell Stresses
Table 2-7
FILLET RADIUS FOR FLANGE WITH MACHINED SPOT FACES

<table>
<thead>
<tr>
<th>t (in.)</th>
<th>r_{fil} (in.)</th>
<th>r_{fil} (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>≥ 0.2</td>
<td>0.3750</td>
<td>10</td>
</tr>
<tr>
<td>&lt; 0.2</td>
<td>0.3125</td>
<td>8</td>
</tr>
<tr>
<td>&lt; 0.15</td>
<td>0.2500</td>
<td>6</td>
</tr>
<tr>
<td>&lt; 0.10</td>
<td>0.1875</td>
<td>4</td>
</tr>
<tr>
<td>&lt; 0.05</td>
<td>0.1250</td>
<td>3</td>
</tr>
</tbody>
</table>

Legend:

\( t \) = tube thickness

\( r_{fil} \) = fillet radius

Table 2-8
GROOVE RADIUS FOR FLANGE WITH MACHINED GROOVE

<table>
<thead>
<tr>
<th>t (in.)</th>
<th>r_{fil} (in.)</th>
<th>r_{fil} (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>≥ 0.2</td>
<td>0.1250</td>
<td>3</td>
</tr>
<tr>
<td>&lt; 0.2</td>
<td>0.1000</td>
<td>2.5</td>
</tr>
<tr>
<td>&lt; 0.15</td>
<td>0.0750</td>
<td>2</td>
</tr>
<tr>
<td>&lt; 0.10</td>
<td>0.0500</td>
<td>1</td>
</tr>
<tr>
<td>&lt; 0.05</td>
<td>0.0250</td>
<td>0.5</td>
</tr>
</tbody>
</table>

Legend:

\( t \) = tube thickness

\( r_{fil} \) = groove radius
Fig. 2-6, as

\[ r_B = r_i + t + c_1 + e_1 \]  

(2.60)

where \( c_1 = 0.0625 \) inch or 1.5 mm was assumed, and for a machined groove, Fig. 2-7, as

\[ r_B = r_i + t + 2r_{fil} + e_1 . \]  

(2.61)

The flange width is then

\[ b = r_B + e_z - r_i . \]  

(2.62)

The inside radius of the tube, \( a \), is denoted by \( r_i \) in this and the following sections.

2.4 GASKET

In selecting the gasket and computing the required contact force, two phases must be considered. The first phase is the initial precompression phase for which a total flange force of

\[ P_G^{(1)} = 2\pi r_G S_G^{(1)} \]  

(2.63)

is required, where \( S_G^{(1)} \) is the corresponding line load per unit length of the centerline of the contact surface. The radius of this centerline is \( r_G \). The second phase is the operational phase in which a certain minimum contact force is to be maintained to have zero leakage. This contact force is written as

\[ P_G^{(2)} = 2\pi r_G S_G^{(2)} \]  

(2.64)

2-27
The total required flange force \( P_F \) is the sum of the force caused by internal pressure, \( P_p \),

\[
P_p = \pi r_G^2 p
\]  \hspace{1cm} (2.65)

and the contact force \( P_G^{(2)} \),

\[
P_G^{(2)} = \pi r_G^2 p + 2\pi r_G S_G^{(2)}
\]  \hspace{1cm} (2.66)

It is important to understand what \( S_G^{(1)} \) and \( S_G^{(2)} \) are for various gaskets and how they are related to the interface leakage rate. Starting out from a macroscopic view, i.e., looking at the whole flange assembly, the relations between the flange force and the internal pressure at which a given leakage occurs are shown on Fig. 2-9. In the low pressure range a nonlinear relation exists between the initial precompression force \( P_G^{(0)} \) and the corresponding pressure. When the force \( P_G^{(1)} \) has been reached, as given by Eq. (2.63), this relation becomes linear as the pressure increases. As the pressure is reduced from above \( p_1 \) the relation remains linear all the way to \( p = 0 \). Under renewed pressurization the relation remains linear. Thus \( P_G^{(1)} \) has been established as the minimum load for precompression of the material. The initial precompression force \( P_G^{(0)} \) in terms of \( P_G^{(1)} \) and \( P_F^{(2)} \) is approximately

\[
P_G^{(0)} = \alpha P_G^{(1)} + (1 - \alpha) \sqrt{P_F^{(2)} P_G^{(1)}}
\]  \hspace{1cm} (2.67)

The coefficient \( \alpha \) should be selected to match the test data.

To characterize a gasket material two numbers are needed. First, a number characterizing \( P_G^{(1)} \), and second, a number characterizing the slope of the straight line.
Fig. 2-9 - Required Gasket Forces
For flat gaskets the line loads are

\[ S^{(1)}_G = b^{(1)}_{\text{eff}} \sigma_G \]  

and

\[ S^{(2)}_G = b^{(2)}_{\text{eff}} k_p p \]  

For other than flat gaskets the line load \( S^{(1)}_G \) is given directly while

\[ S^{(2)}_G = K_p p \]  

The quantity \( \sigma_G \) has the units of stress and \( k_p \) is dimensionless. The quantity \( K_1 \) has the dimensions of a length. The effective widths \( b^{(1)}_{\text{eff}} \) and \( b^{(2)}_{\text{eff}} \) depend on the shape of the interface, such as tongue and groove, etc., and the type of gasket, such as serrated, etc.

Data for conventional applications can be found in Ref. 6. For cryogenic or storable propellants in liquid or gaseous form these data are not readily available and will have to be obtained from testing or be established from data not currently in this format.

One source of information is the comprehensive study by Bauer et al. (Ref. 31), which contains data on interface leakage as related to material hardness, contact surface topography (surface finish) and contact stress, expressed as

\[ h^3 = \frac{K_e}{\sigma^m} \]  

where \( h^3 \) is the "conductance parameter", \( K_e \) is a constant and \( \sigma \) is the contact stress. The exponent \( m \) depends on both the material hardness and the surface topography. This relation of Eq. (2.71) is obtained from graphs of the form shown on Fig. 2-10. There are four regimes identified. The fourth one indicates the hysteresis effect shown more clearly on Fig. 2-11. To use the graphs, the anticipated leak rate, either gas (volume) or liquid
Fig. 2-10 - Relation Between Conductance Parameter and Interface Contact Stress (Ref. 31)
$\Omega$ (Leak-rate of the Connection)

$P_F$ (Interface Total Force of the Connection)

Fig. 2-11 - Hysteresis Effect (Ref. 31)
(weight) leak rate, is the basis for computing the required conductance parameter. For laminar, isothermal, compressible (gas) flow the volume leak rate is

\[ Q = \frac{w (p_2^2 - p_1^2)}{24\mu p_o L} \quad h^3 \]  

(2.72)

where \( w \) is the width and \( L \) is the length of the leak path, \( \mu \) is the viscosity of the medium and \( p_o \) the standard atmospheric pressure. The inlet pressure is \( p_2 \) and the exit pressure is \( p_1 \). Other effects such as inertia, transition flow with molecular correction and adiabatic frictionless flow are presented in Ref. 31. These are, however, less important than the one given by Eq. (2.72). The volume leak rate \( Q \) of a gas can be converted into a mass leak rate by the relation

\[ W = \frac{pm}{RT} Q \]  

(2.73)

where \( R \) is the gas constant, \( T \) is the absolute temperature, \( p \) is the pressure and \( m \) is the molecular weight. For laminar, incompressible (liquid) flow the mass leak rate is

\[ W = \frac{\rho w(p_2 - p_1)}{12\mu L} \quad h^3 \]  

(2.74)

where \( \rho \) is the mass density of the medium. The width and the length of the leak path for a gasket are, respectively

\[ w \approx 2\pi r_G \]

and

\[ L \approx b_G \]  

(2.75)

Usually the definition of zero-leak is defined in terms of volume per unit time of helium at standard temperature and pressure. This leak rate can be converted into an equivalent liquid leak rate by using conversion graphs. One such graph is described in a report by Weiner (Ref. 32), which
represents the Poiseuille equation for gas and liquid flow, as given by Eqs. (2.72) and (2.74). The procedure is illustrated on Fig. 2-12.

The design procedure for finding the width of a flat gasket is based on the condition that

$$P_G^{(1)} = P_F^{(2)} = P_p + P_G^{(2)} \quad (2.76)$$

This leads to

$$2\pi b_{\text{eff}}^G r_G \sigma_G = \pi r_G^2 P + 2\pi b_{\text{eff}}^G r_G k_p P \quad (2.77)$$

which can be solved for $b_G$ after substituting

$$b_{\text{eff}}^{(1)} = \gamma_1 b_G \quad (2.78)$$
$$b_{\text{eff}}^{(2)} = \gamma_2 b_G \quad (2.79)$$

resulting in

$$b_G = \frac{r_G P}{2(\gamma_1 \sigma_G - \gamma_2 k_p P)} \quad (2.80)$$

The safety factor should be attached to $P_G^{(2)}$, taking it as (F.S.) = 1 for the proof pressure condition and (F.S.) = 1.5 for the operating pressure condition.

As a numerical example consider ALLPAX flat gaskets of thickness 1/8; 1/16 and 1/32 inch, having a yield strength of $K_G = 10.0$ ksi. The minimum stresses for precompression based on experience in conventional applications are $\sigma_G = 1.6; 3.7; 6.5$ ksi and the slope of the straight line as shown on Fig. 2-9 is $k_p = 2.0; 2.75; 3.5$. If the gaskets are precompressed to yield stress then the following gasket widths are obtained, as given in Table 2-9.
Fig. 2-12 - Fluid Flow Conversion Graph (Ref. 32)
Table 2-9
COMPARISON OF GASKET WIDTHS

<table>
<thead>
<tr>
<th>Pressure</th>
<th>Gasket Thickness, ( h_G )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( p ) (psi)</td>
<td>( 1/8 )</td>
</tr>
<tr>
<td>100</td>
<td>0.0102 ( r_G )</td>
</tr>
<tr>
<td>1000</td>
<td>0.125 ( r_G )</td>
</tr>
</tbody>
</table>

Since it is not necessary to precompress to the yield stress an alternate procedure would be to start with the available gasket width after the design has progressed to this point. It is

\[
b_{G,\text{avail}} = r_B - \frac{d_{\text{hole}}}{2} - r_i - 2\Delta r
\]  

(2.81)

A tolerance of \( \Delta r \) is provided in this formula. Then from Eq. (2.77) the required flange force under operating condition is

\[
P_{F}^{(2)} = \pi r_G^2 p + 2\pi b_{G,\text{avail}} \gamma_2 r_G k_p p \text{ (F.S.)}
\]  

(2.82)

where (F.S.) is the appropriate factor of safety. The condition of Eq. (2.76), making the initial flange force equal to the one under operating condition, gives the required initial contact stress as

\[
\sigma_G = \frac{P_{F}^{(2)}}{2\pi b_{G,\text{avail}} \gamma_1 r_G}
\]  

(2.83)

If \( \sigma_G \) is less than the minimum precompression stress required to seat the gasket, the initial flange load should be increased to achieve this minimum precompression stress. For example, if for a 1/10 inch ALLPAX gasket \( \sigma_G \) as computed with Eq. (2.83) is less than 3.7 ksi, the initial flange load should be \( P_{F}^{(2)} = 2\pi b_{G,\text{avail}} \gamma_1 r_G \) (3.7 ksi).
The safety factor in Eq. (2.82) will compensate for reduction of gasket stress due to the elastic deformation of the connection. The analysis of this deformation is described in Section 3.

2.5 PRESSURE ENERGIZED SEAL

The application of a pressure energized seal in both a cantilever flange and a flange with metal-to-metal contact is illustrated on Figs. 2-13 and 2-14, respectively.

The basic difference of the two flange configurations can be seen in the accompanying calculations of bolt forces. The bolt force required for the cantilever flange is simply

\[ P_B = (F.S.) P_p \]  \hspace{1cm} (2.84)

where

\[ P_p = \pi r_s^2 p \]  \hspace{1cm} (2.85)

For the metal-to-metal flange the pivot point \( A \) is outside the bolt circle, while previously it was in line with the middle surface of the tube wall. Taking the bending moment about point \( A \), i.e., assuming a situation where differential axial motion exists at the seal-to-flange interface, the required bolt force is approximately

\[ P_B = (F.S.) P_p \left( 1 + \frac{e}{e_2} \right). \]  \hspace{1cm} (2.86)

In any case the required bolt force is by the factor of \( 1 + e/e_2 \) higher than the corresponding cantilever flange.

The size of the seal gland depends on the type and size of seal to be used. These dimensions, \( h_s \) and \( b_s \), are supplied by the seal manufacturers' catalogs. The height of the recess, \( h_r \), for the cantilever flange is to be
Fig. 2-13 - Cantilever Flange with Pressure Energized Seal
Fig. 2-14 - Metal-to-Metal Flange with Pressure Energized Seal
determined from the roll angle and corresponding differential axial displacement at the outer edge of the flange cross section. If the recess is not high enough, this will result in the same situation as for the metal-to-metal flange and is therefore undesirable.

The width $b_G$, carrying the same label for convenience of notation as the width of a flat gasket, is assumed as being equal to the tube wall thickness, $t$. The distance of the seal gland from the bolt holes was assumed as being the same as the width of the seal gland itself. These dimensional relations cannot be readily defined and will have to be determined by a developmental test program.

2.6 BOLT FORCE, NUMBER OF BOLTS AND BOLT SPACING

The required bolt force is determined by the gasket initial stress and minimum stress during operation, or in the case of a pressure energized seal, by the force required to prevent separation near the seal-flange interface. The maximum force of the ones determined by different criteria is used to compute the number of bolts required. Two design criteria are used. Under proof pressure the bolts should not yield and under burst pressure they should not break. These two criteria can be formulated as

$$n_{B1} = \frac{P_B}{F_{ty} A_{oB}}$$

(2.87)

where $P_B$ is the bolt force under proof pressure, including the safety factor, and

$$n_{B2} = \frac{P_B^{(burst)}}{F_{tu} \Delta_{oB}}$$

(2.88)

where $P_B^{(burst)}$ is the bolt force at burst pressure. The minimum number of bolts was assumed as six. This will give an even stress distribution for
flanges with low internal pressures and small inner diameter, for which less than six bolts would be computed according to Eqs. (2.87) and (2.88).

The bolt spacing is simply

\[ s = \frac{2\pi r_B}{n_B} \]  

(2.89)

where \( n_B \) is the maximum of the numbers of bolts, \( n_{B1} \) or \( n_{B2} \), computed previously. The spacing should not increase beyond a certain level, which has been arbitrarily fixed at \( s = 8d_B \), where \( d_B \) is the nominal bolt diameter. This maximum spacing depends on the thickness of the flange, too, since bending out of the plane of the flange would introduce a reduction in interface stress in the space between the bolt holes. For the low profile flanges, however, this situation is not critical since the aspect ratio of \( h/b \) for the flange cross section is usually greater than \( 3/4 \), mostly being around 1. Therefore it acts quite differently from a flat plate assumed in previous flange design methods. A minimum spacing is provided by the value of \( s = \eta_o d_B \), where \( \eta_o \) is tabulated for various types of bolt heads as a function of nominal diameter.

2.7 FLANGE HEIGHT

The computation of the flange height is based on the capacity to carry the ultimate applied moment

\[ m_{Fu} = \frac{(F.S.) P_B e}{2\pi r_o} \]  

(2.90)

where \( P_B \) is the maximum bolt force considered for the design and \( e \) the internal lever arm between the bolt circle and the gasket circle,

\[ e = r_B - r_G \]  

(2.91)
The radius $r_o$ is the one of the middle surface of the tube wall,

$$r_o = r_1 + t/2 \quad (2.92)$$

The width of the flange cross section has already been determined, either from considerations to accommodate the boltheads or to accommodate the gasket. The effect of the bolt holes, however, has to be taken into account. A simple rule has been suggested by Schwaigerer (Ref. 2) based on experience, by computing an effective width, $\overline{b}$, from the bolt hole diameter $d_{hole}$ and the bolt spacing, $s$,

$$\overline{b} = b - d_{hole} \sqrt{\frac{d_{hole}}{s}} \quad (2.93)$$

Previous design methods have suggested to subtract the entire hole diameter. This would be unduly conservative as proven by tests (Ref. 3).

The computation of the flange height assumes a linear stress distribution in the flange and the development of a plastic hinge in the neck. This procedure of designing a statically indeterminate structure by introducing plastic hinges to reduce redundancies was first used for steel frames (Ref. 33) and resulted in more efficient designs. The state of stress in the neck of the flange is three-dimensional, however, and the method used in frame design is, therefore, not rigorously applicable.

The derivation of the concept of a plastic section modulus, $Z_T$, for the tube wall is described in detail in Section 3. The result is

$$Z_T = \frac{t^2}{4} - \frac{t_N^2}{4} + \zeta_2 (t - t_N) h \quad (2.94)$$

where $\zeta_1$ and $\zeta_2$ are coefficients determined by the state of stress in the flange neck. Since this state of stress is unknown at this point they are assumed to be
\[ \xi_1 = 0.8, \quad \xi_2 = 0.18 \]  

(2.95)

In Section 3 the computation of \( \xi_1 \) and \( \xi_2 \) for a given flange under given loading conditions is shown in detail.

The elastic section modulus of the flange cross section is given by

\[ S_F = \frac{b}{6} \frac{h^2}{r_o} \]  

(2.96)

The design formula for the flange is now derived by requiring

\[ m_{Fu} = F_{ty} \left[ \frac{F_{ty}}{F_{ty}} + \frac{F_{ty}}{F_{ty}} \frac{Z_T}{S_F} \right] \]  

(2.97)

Usually the flange and tube wall are made of the same material so that \( F_{ty}/F_{ty} = 1 \). When the expressions for \( Z_T \) and \( S_F \) are substituted into Eq. (2.97) a quadratic equation for \( h \) results,

\[ Ah^2 + Bh + C = 0, \]  

(2.98)

where

\[ A = \frac{F_{ty}^{(F)}}{F_{ty}} \frac{b}{6} \frac{r_o}{}, \]  

(2.99)

\[ B = F_{ty}^{(F)} \xi_2 (t - t_N)/2, \]  

(2.100)

\[ C = F_{ty}^{(F)} \xi_1 (t^2 - t_N^2)/4 - m_{Fu}. \]  

(2.101)

The solution for \( h \) is

\[ h = \frac{-B \pm \sqrt{B^2 - 4AC}}{2A} \]  

(2.102)
If the contribution of the plastic hinge in the neck is neglected in the design of the flange, i.e., when $\xi_1$ and $\xi_2$ are assumed to be zero, then

$$h = \sqrt{\frac{6}{\pi} \frac{r_m}{F_u}} \left( \frac{F}{F_{ty}} \right) b.$$  (2.103)

This design formula has been used previously for computational convenience but may result in overly conservative designs.

Finally, a check is made of the flange height versus the bolt spacing, $s$,

$$\text{if } s/h < 3 \Rightarrow h = s/3$$  (2.104)

2.8 FLANGE WEIGHT

The weight added to the tube by the flange is given by computing the volume of the material having the cross sectional area

$$A_w = (b - t) h$$  (2.105)

and the centroidal radius

$$r_w = r_i + \frac{(t+b)}{2}$$  (2.106)

so that

$$\text{vol} = 2\pi r_w A_w$$  (2.107)

the actual weight is

$$\Delta W = \rho_F \text{ vol}$$  (2.108)
2.9 MATERIAL DATA

To facilitate the computation of numerical examples the properties for aluminum and steel commonly used for rocket propulsion systems are given in Tables 2-10 and 2-12. These data were taken from Ref. 15. Data for gaskets were compiled for some materials used in some earlier MSFC computations (Ref. 34) and are listed in Tables 2-11 and 2-13.

Both data tables are incorporated in the computer program. They can be enlarged easily by including a larger variety of data. It was not the purpose of this study to compile all available data.
Table 2-10
PROPERTIES OF METALIC MATERIALS FOR TUBES, FLANGES AND BOLTS

<table>
<thead>
<tr>
<th>No.</th>
<th>Material</th>
<th>(E) (psi)</th>
<th>(\nu)</th>
<th>(\rho) (lb/in.(^3))</th>
<th>(\alpha) (in./in./°F)</th>
<th>(F_{ty}) (psi)</th>
<th>(F_{tu}) (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Al 6061-T6 @ RT</td>
<td>(9.9 \times 10^6)</td>
<td>.33</td>
<td>.098</td>
<td>(12.5 \times 10^{-6})</td>
<td>(35.0 \times 10^3)</td>
<td>(42.2 \times 10^3)</td>
</tr>
<tr>
<td>2</td>
<td>Al 6061-T6 @ 200°F</td>
<td>(9.9 \times 10^6)</td>
<td>.33</td>
<td>.098</td>
<td>(12.5 \times 10^{-6})</td>
<td>(32.2 \times 10^3)</td>
<td>(38.1 \times 10^3)</td>
</tr>
<tr>
<td>3</td>
<td>Al 2024-T3 @ RT</td>
<td>(9.9 \times 10^6)</td>
<td>.33</td>
<td>.098</td>
<td>(12.5 \times 10^{-6})</td>
<td>(50.0 \times 10^3)</td>
<td>(62.0 \times 10^3)</td>
</tr>
<tr>
<td>4</td>
<td>Al 2024-T3 @ 200°F</td>
<td>(9.9 \times 10^6)</td>
<td>.33</td>
<td>.098</td>
<td>(12.5 \times 10^{-6})</td>
<td>(47.0 \times 10^3)</td>
<td>(59.0 \times 10^3)</td>
</tr>
<tr>
<td>5</td>
<td>347 SS @ RT</td>
<td>(28.0 \times 10^6)</td>
<td>.30</td>
<td>.288</td>
<td>(9.5 \times 10^{-6})</td>
<td>(35.0 \times 10^3)</td>
<td>(90.0 \times 10^3)</td>
</tr>
<tr>
<td>6</td>
<td>347 SS @ 200°F</td>
<td>(28.0 \times 10^6)</td>
<td>.30</td>
<td>.288</td>
<td>(9.5 \times 10^{-6})</td>
<td>(30.0 \times 10^3)</td>
<td>(76.0 \times 10^3)</td>
</tr>
<tr>
<td>7</td>
<td>347 SS @ 600°F</td>
<td>(28.0 \times 10^6)</td>
<td>.30</td>
<td>.288</td>
<td>(9.5 \times 10^{-6})</td>
<td>(25.0 \times 10^3)</td>
<td>(68.0 \times 10^3)</td>
</tr>
<tr>
<td>8</td>
<td>A286 @ RT</td>
<td>(28.0 \times 10^6)</td>
<td>.30</td>
<td>.288</td>
<td>(9.5 \times 10^{-6})</td>
<td>(131.0 \times 10^3)</td>
<td>(200.0 \times 10^3)</td>
</tr>
<tr>
<td>9</td>
<td>A286 @ 200°F</td>
<td>(28.0 \times 10^6)</td>
<td>.30</td>
<td>.288</td>
<td>(9.5 \times 10^{-6})</td>
<td>(128.0 \times 10^3)</td>
<td>(196.0 \times 10^3)</td>
</tr>
<tr>
<td>10</td>
<td>A286 @ 600°F</td>
<td>(28.0 \times 10^6)</td>
<td>.30</td>
<td>.288</td>
<td>(9.5 \times 10^{-6})</td>
<td>(120.0 \times 10^3)</td>
<td>(180.0 \times 10^3)</td>
</tr>
</tbody>
</table>

Legend: \(E\) = elastic modulus
\(\nu\) = Poisson's ratio
\(\rho\) = weight density
\(\alpha\) = linear thermal expansion coefficient
\(F_{ty}\) = tensile yield strength
\(F_{tu}\) = ultimate tensile strength
Table 2-11
PROPERTIES OF GASKET MATERIALS

<table>
<thead>
<tr>
<th>No.</th>
<th>Material</th>
<th>E (psi)</th>
<th>KG (psi)</th>
<th>σG (psi)</th>
<th>α (in/in/°F)</th>
<th>μ (-)</th>
<th>hG (in.)</th>
<th>k_p</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Asbestos 1/32 in.</td>
<td>44.0 x 10^3</td>
<td>10.0 x 10^3</td>
<td>6.5 x 10^3</td>
<td>1.3 x 10^-3</td>
<td>.5</td>
<td>.03125</td>
<td>3.50</td>
</tr>
<tr>
<td>2</td>
<td>Asbestos 1/16 in.</td>
<td>44.0 x 10^3</td>
<td>10.0 x 10^3</td>
<td>3.7 x 10^3</td>
<td>1.3 x 10^-3</td>
<td>.5</td>
<td>.06250</td>
<td>2.75</td>
</tr>
<tr>
<td>3</td>
<td>Asbestos 1/8 in.</td>
<td>44.0 x 10^3</td>
<td>10.0 x 10^3</td>
<td>1.6 x 10^3</td>
<td>1.3 x 10^-3</td>
<td>.5</td>
<td>.12500</td>
<td>2.00</td>
</tr>
<tr>
<td>4</td>
<td>KEL-F81</td>
<td>180.0 x 10^3</td>
<td>8.0 x 10^3</td>
<td>4.0 x 10^3</td>
<td>3.8 x 10^-5</td>
<td>.12</td>
<td>.06250</td>
<td>3.00</td>
</tr>
<tr>
<td>5</td>
<td>CRES 321-A</td>
<td>28.0 x 10^6</td>
<td>40.0 x 10^3</td>
<td>18.9 x 10^3</td>
<td>9.5 x 10^-6</td>
<td>.30</td>
<td>.02500</td>
<td>5.50</td>
</tr>
</tbody>
</table>

Legend:

E = elastic modulus
KG = yield (crushing) strength
σG = minimum seating stress
α = linear thermal expansion coefficient
μ = friction coefficient
hG = thickness of the gasket

Assumed:

γ_1 = γ_2 = 0.5 for No's 1, 2 and 3
γ_1 = γ_2 = 1.0 for No's 4 and 5
Table 2-12
METRIC PROPERTIES OF METALLIC MATERIALS FOR TUBES, FLANGES AND BOLTS

<table>
<thead>
<tr>
<th>No.</th>
<th>Material</th>
<th>$E$ (N/mm$^2$)</th>
<th>$\nu$</th>
<th>$\rho$ (g/mm$^3$)</th>
<th>$\alpha$ (mm/mm$^\circ$C)</th>
<th>$F_{ty}$ (N/mm$^2$)</th>
<th>$F_{tu}$ (N/mm$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Al 6061-T6</td>
<td>68 x 10$^3$</td>
<td>.33</td>
<td>.271 x 10$^{-2}$</td>
<td>22.5 x 10$^{-6}$</td>
<td>242</td>
<td>290</td>
</tr>
<tr>
<td>2</td>
<td>Al 6061-T6</td>
<td>68 x 10$^3$</td>
<td>.33</td>
<td>.271 x 10$^{-2}$</td>
<td>22.5 x 10$^{-6}$</td>
<td>222</td>
<td>263</td>
</tr>
<tr>
<td>3</td>
<td>Al 2024-T3</td>
<td>68 x 10$^3$</td>
<td>.33</td>
<td>.271 x 10$^{-2}$</td>
<td>22.5 x 10$^{-6}$</td>
<td>345</td>
<td>428</td>
</tr>
<tr>
<td>4</td>
<td>Al 2024-T3</td>
<td>68 x 10$^3$</td>
<td>.33</td>
<td>.271 x 10$^{-2}$</td>
<td>22.5 x 10$^{-6}$</td>
<td>324</td>
<td>407</td>
</tr>
<tr>
<td>5</td>
<td>347 SS</td>
<td>193 x 10$^3$</td>
<td>.30</td>
<td>.798 x 10$^{-2}$</td>
<td>17.1 x 10$^{-6}$</td>
<td>242</td>
<td>621</td>
</tr>
<tr>
<td>6</td>
<td>347 SS</td>
<td>193 x 10$^3$</td>
<td>.30</td>
<td>.798 x 10$^{-2}$</td>
<td>17.1 x 10$^{-6}$</td>
<td>207</td>
<td>528</td>
</tr>
<tr>
<td>7</td>
<td>347 SS</td>
<td>193 x 10$^3$</td>
<td>.30</td>
<td>.798 x 10$^{-2}$</td>
<td>17.1 x 10$^{-6}$</td>
<td>173</td>
<td>469</td>
</tr>
<tr>
<td>8</td>
<td>A286</td>
<td>193 x 10$^3$</td>
<td>.30</td>
<td>.798 x 10$^{-2}$</td>
<td>17.1 x 10$^{-6}$</td>
<td>904</td>
<td>1380</td>
</tr>
<tr>
<td>9</td>
<td>A286</td>
<td>193 x 10$^3$</td>
<td>.30</td>
<td>.798 x 10$^{-2}$</td>
<td>17.1 x 10$^{-6}$</td>
<td>883</td>
<td>1352</td>
</tr>
<tr>
<td>10</td>
<td>A286</td>
<td>193 x 10$^3$</td>
<td>.30</td>
<td>.798 x 10$^{-2}$</td>
<td>17.1 x 10$^{-6}$</td>
<td>828</td>
<td>1242</td>
</tr>
</tbody>
</table>
Table 2-13

METRIC PROPERTIES OF GASKET MATERIALS

<table>
<thead>
<tr>
<th>No.</th>
<th>Material</th>
<th>E (N/mm²)</th>
<th>k_G (N/mm²)</th>
<th>σ_G (N/mm²)</th>
<th>α (mm/mm°C)</th>
<th>μ</th>
<th>h_G (mm)</th>
<th>k_p</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Asbestos, 0.8 mm</td>
<td>304.0</td>
<td>69.0</td>
<td>45.0</td>
<td>2.3 x 10^{-3}</td>
<td>.5</td>
<td>.8</td>
<td>3.5</td>
</tr>
<tr>
<td>2</td>
<td>Asbestos, 1.6 mm</td>
<td>304.0</td>
<td>69.0</td>
<td>26.0</td>
<td>2.3 x 10^{-3}</td>
<td>.5</td>
<td>1.6</td>
<td>2.75</td>
</tr>
<tr>
<td>3</td>
<td>Asbestos, 3.2 mm</td>
<td>304.0</td>
<td>69.0</td>
<td>11.0</td>
<td>2.3 x 10^{-3}</td>
<td>.5</td>
<td>3.2</td>
<td>2.00</td>
</tr>
<tr>
<td>4</td>
<td>KEL-F81 1.6 mm</td>
<td>1242.0</td>
<td>55.0</td>
<td>28.0</td>
<td>7.0 x 10^{-5}</td>
<td>.12</td>
<td>1.6</td>
<td>3.00</td>
</tr>
<tr>
<td>5</td>
<td>CRES 321-A, 6 mm</td>
<td>193 x 10³</td>
<td>276.0</td>
<td>130.0</td>
<td>17.1 x 10^{-6}</td>
<td>.30</td>
<td>.6</td>
<td>5.50</td>
</tr>
</tbody>
</table>
Section 3
ANALYSIS METHOD

The analysis method described in this section is based on thin shell theory and simple ring theory. These theories are not too involved algebraically to be used for hand computations. Also the approximate state of stress in the plastic hinge near the flange used is described. A summary of the formulas used in the analysis is given in Appendix B.

3.1 SHELL THEORY

The membrane solution for a cylindrical shell under an internal pressure \( p \) and a temperature differential \( \Delta T \) is characterized by the stress resultants.

\[
\begin{align*}
    n_x &= \frac{p r_o}{2} \\
    n_\phi &= p r_o
\end{align*}
\]  

(3.1)

and

\[
    n_\phi = p r_o 
\]  

(3.2)

where \( n_x \), \( n_\phi \) are the axial and circumferential stress resultants, respectively measured as a force per unit length. The radial expansion of the shell under this loading condition is

\[
    W = \frac{p r^2}{E t} (1 - \frac{\nu}{2}) + r_o \alpha \Delta T
\]

(3.3)

where \( \alpha \) is the linear thermal expansion coefficient and \( E \) and \( \nu \) are the elastic modulus and Poisson's ratio, respectively.
In addition to this solution the edge disturbance of the cylinder, introduced by the flange, must be considered. It can be shown (Ref. 35) that the linear differential equation

\[ \frac{d^4 w}{dx^4} + k^4 w = 0 \]  

(3.4)

where

\[ k^4 = \frac{12(1-\nu^2)}{r_o^2 t^2} \]  

(3.5)

describes this behavior. This differential equation for the range of parameters considered, assuming the shell to be infinitely long, has the solution

\[ w = e^{-kx} (C_1 \cos kx + C_2 \sin kx) \]  

(3.6)

The integration constants are found from the edge conditions. The flange usually introduces an edge moment \( m_o \) and an edge shear \( q_o \) into the shell. Both are measured per unit length (Fig. 3-1). Knowing that

\[ m_x \bigg|_{x=0} = -B \left( \frac{d^2 w}{dx^2} \right)_{x=0} = m_o \]  

(3.7)

\[ q_x \bigg|_{x=0} = -B \left( \frac{d^3 w}{dx^3} \right)_{x=0} = -q_o \]  

(3.8)

where

\[ B = \frac{Et^3}{12(1-\nu^2)} \]  

(3.9)
Fig. 3-1 - Edge-Loaded Cylindrical Shell
The constants are derived using

\[
\frac{dw}{dx} = -ke^{-kx} \left[ (C_1 - C_2) \cos kx + (C_1 + C_2) \sin kx \right], \tag{3.10}
\]

\[
\frac{d^2w}{dx^2} = 2k^2 e^{-kx} (C_1 \sin kx - C_2 \cos kx), \tag{3.11}
\]

\[
\frac{d^3w}{dx^3} = -2k^3 e^{-kx} \left[ (C_1 - C_2) \sin kx - (C_1 + C_2) \cos kx \right] \tag{3.12}
\]

It follows then that

\[
C_1 = \frac{m_o}{2k^2B}; \quad C_2 = \frac{q_o - km_o}{2k^3B} \tag{3.13}
\]

With these constants the radial displacement is

\[
w = \frac{1}{2k^3B} e^{-kx} \left[ q_o \cos kx - km_o (\cos kx - \sin kx) \right] \tag{3.14}
\]

and the rotation (rolling) of the shell wall is

\[
\chi = \frac{dw}{dx} = \frac{1}{2k^2B} e^{-kx} \left[ -q_o (\cos kx + \sin kx) + 2km_o \cos kx \right] \tag{3.15}
\]

For the edge where \(x=0\) the flexibility matrix is seen to be

\[
\begin{bmatrix}
w \\
\chi \\
\end{bmatrix} = \frac{1}{2k^3B} \begin{bmatrix} 1 & -k \\ -k & 2k^2 \end{bmatrix} \begin{bmatrix} q_o \\ m_o \end{bmatrix} \tag{3.16}
\]
The meridional bending moment along the shell wall is

\[ m_x = e^{-kx} \left[ m_o (\cos kx + \sin kx) - \frac{q_o}{k} \sin kx \right] \]  

(3.17)

and the meridional shear is

\[ q_x = e^{-kx} \left[ q_o (\sin kx - \cos kx) + 2k m_o \sin kx \right] \]  

(3.18)

The circumferential bending moment is

\[ m_\varphi = \nu m_x \]  

(3.19)

and the circumferential stress resultant is

\[ n_\varphi = \frac{E t}{r_o} w \]  

(3.20)

This concludes the description of the analysis of the edge disturbance.

It remains to be shown how the stresses are computed in the elastic range and how the plastic state of stress is described. Three stresses exist in the shell, the axial stress

\[ \sigma_x = \frac{n_x}{t} + \frac{m_x}{t^3} z \]  

(3.21)

the circumferential stress

\[ \sigma_\varphi = \frac{n_\varphi}{t} + \frac{m_\varphi}{t^3} z, \]  

(3.22)
and the shear stress

\[ \tau_{xz} = \frac{q_x}{t} \left( \frac{t}{6} \right)^2 - z^2 \]  \hspace{1cm} (3.23)

The coordinate \( z \) is measured from the shell middle surface outward in the normal direction.

To arrive at an expression for the development of a plastic hinge in the shell it is assumed that a core (Fig. 3-2) of thickness,

\[ t_n = \frac{n_X}{Y_0}, \]  \hspace{1cm} (3.24)

is required to carry the axial force, where \( Y_0 \) is the uniaxial tensile yield strength of the material. This leaves for the plastic moment, \( m_x^P \),

\[ \sigma_x^P = \frac{m_x^P}{t_2^2 - t_n^2}/4 \]  \hspace{1cm} (3.25)

and the plastic shear force, \( q_x^P \),

\[ \tau_{xz}^P = \frac{q_x^P}{t_2 - t_n} \]  \hspace{1cm} (3.26)

In order to relate the three-dimensional state of stress to the uniaxial tensile yield strength \( Y_0 \) the yield condition of von Mises is used.

\[ \sigma = \sqrt{\frac{1}{2} \left[ (\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 \right]} \leq Y_0 \]  \hspace{1cm} (3.27)

where \( \sigma_1, \sigma_2, \sigma_3 \) are the principal stresses.
Fig. 3-2 - Assumed Stress Distribution in the Plastic Hinge
The principal stresses for the problem at hand are

\[
\sigma_1 = \frac{\sigma_{x}^{p}}{2} + \sqrt{\left(\frac{\sigma_{x}^{p}}{2}\right)^{2} + \left(\tau_{xz}^{p}\right)^{2}} \tag{3.28}
\]

\[
\sigma_2 = \frac{\sigma_{x}^{p}}{2} - \sqrt{\left(\frac{\sigma_{x}^{p}}{2}\right)^{2} + \left(\tau_{xz}^{p}\right)^{2}} \tag{3.29}
\]

\[
\sigma_3 = \sigma_{\phi}^{p} \tag{3.30}
\]

The expressions for the principal stresses are simplified by introducing \(\sigma_{x}^{p}\) as a reference stress, where

\[
\sigma_{\phi}^{p} = \alpha_1 \sigma_{x}^{p} \tag{3.31}
\]

and

\[
\tau_{xz}^{p} = \alpha_2 \sigma_{x}^{p} \tag{3.32}
\]

then

\[
\sigma_{1,2} = \sigma_{x}^{p} \left(\frac{1}{2} \pm \sqrt{\frac{1}{4} + \alpha_2^2}\right) \tag{3.33}
\]

\[
\sigma_3 = \alpha_1 \sigma_{x}^{p} \tag{3.34}
\]
and the equivalent stress $\bar{\sigma}$ of Eq. (3.27) is

$$\bar{\sigma} = \sigma_x^p \sqrt{1 + \alpha_1 + \alpha_2^2 + 3\alpha_2^2} \quad (3.35)$$

the computation of $\alpha_1, \alpha_2$ and $\sigma_x^p$ for a given loading condition will be shown later.

3.2 FLANGE THEORY

Adding the flange to the shell requires finding the interface moment $m_o$ and interface shear $q_o$ (Fig. 3-3) in terms of given loading conditions.

In the analysis of the flange deformations it is useful to derive an equivalent rotational spring constant per unit length of the flange (Ref. 36). Starting with the equation for the radial displacement $w$ and rotation $\chi$ of the interface point (A) (Fig. 3-3) caused by an applied moment $m_F$ per unit length,

$$\begin{bmatrix} w \\ \chi \end{bmatrix} = - \frac{r_o r_c}{EI} \begin{bmatrix} c^2 + \frac{I}{A} & c \\ c & 1 \end{bmatrix} \begin{bmatrix} q_o \\ m_o \end{bmatrix} + \frac{r_o r_c}{EI} \begin{bmatrix} c \\ 1 \end{bmatrix} m_F \quad (3.36)$$

where $A$ is the cross sectional area and $I$ is the moment of inertia of the ring cross section, an equation for $m_o$ and $q_o$ can be constructed by requiring compatibility of the displacements of point (A) on the ring and on the shell. Using Eq. (3.16) for the shell displacements it follows that

$$\begin{bmatrix} \frac{1}{2k^3 B} \begin{bmatrix} 1 & -k \\ -k & 2k^2 \end{bmatrix} \begin{bmatrix} q_o \\ m_o \end{bmatrix} = - \frac{r_o r_c}{EI} \begin{bmatrix} c^2 + \frac{I}{A} & c \\ c & 1 \end{bmatrix} \begin{bmatrix} q_o \\ m_o \end{bmatrix} + \frac{r_o r_c}{EI} \begin{bmatrix} c \\ 1 \end{bmatrix} m_F \quad (3.37)$$

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Fig. 3-3 - Ring-Shell Interface
which can be combined as

\[
\begin{bmatrix}
\frac{1}{2k^2} + \beta \left( c^2 + \frac{1}{A} \right) & -\frac{1}{2k} + c\beta \\
-\frac{1}{2k} + c\beta & 1 + \beta
\end{bmatrix}
\begin{bmatrix}
q_o \\
m_o
\end{bmatrix}
= \beta
\begin{bmatrix}
c \\
m_F
\end{bmatrix}
\] (3.38)

where

\[
\beta = \frac{Bk}{E\Gamma_o} 
\] (3.39)

The determinant of this equation is

\[
D = (1+\beta) \left[ \frac{1}{2k^2} + \beta \left( c^2 + \frac{1}{A} \right) \right] - (\frac{1}{2k} + c\beta)^2 
\] (3.40)

To find \( q_o \) and \( m_o \), Cramer's rule is used,

\[
q_o = \frac{\beta}{D} \left( c + \frac{1}{2k} \right) m_F 
\] (3.41)

and

\[
m_o = \frac{\beta}{D} \left( \frac{1}{2k^2} + \beta \frac{1}{A} + \frac{c}{2k} \right) m_F 
\] (3.42)

The rotation of the cross section is then
The rotational spring constant is obtained from Eq. (3.43) by dividing \( m_F \) by the rotation \( \chi \),

\[
\chi = \frac{1}{2k^3 \beta} (2k^2 m_o - kq_o)
\]

\[
= \frac{\beta}{2k^3 BD} (2k^2 \beta \frac{I}{A} + \frac{1}{2}) m_F
\]

(3.43)

The second loading condition to be considered in this paragraph is a differential radial displacement \( \Delta w \) between the ring and the shell where

\[
\Delta w = w_{\text{shell}} - w_{\text{ring}}
\]

(3.45)

the corresponding equation for \( m_o \) and \( q_o \) to be solved is obtained by replacing the right hand side of Eq. (3.38) by

\[
- \frac{\beta EI}{r_o r_c D} \begin{bmatrix}
1 \\
0
\end{bmatrix} \Delta w.
\]

(3.46)

The solution is given by

\[
q_o = \frac{-\beta EI}{r_o r_c D} \frac{1 + \beta}{} \Delta w,
\]

(3.47)
\[
m_o = -\frac{\beta EI}{r_o r_c D} \left( \frac{1}{2k} - c\beta \right) \Delta w , \quad (3.48)
\]

and the rotation is in accordance with Eq. (3.43).

\[
\chi = \frac{\beta}{D} \left( c + \frac{1}{2k} \right) \Delta w \quad (3.49)
\]

The same rotation can be produced by an applied moment of

\[
m_F = c_F \chi = \frac{B (c + \frac{1}{2k})}{\left( \frac{\beta}{K} + \frac{1}{A} + \frac{1}{4k^2} \right)} \Delta w \quad (3.50)
\]

The stresses in the ring at points \(A\) and \(B\) are

\[
\sigma^A_{\varphi} = \frac{E}{r_o} w \quad (3.51)
\]

and

\[
\sigma^B_{\varphi} = \frac{E}{r_o} (w - h\chi) \quad (3.52)
\]

where \(w\) and \(\chi\) are the radial displacement and rotation, respectively, of point \(A\).

3.3 EFFECTS OF BOLTS AND GASKET

The bolts and the gasket contribute to the elastic properties of the flanged connection. Both can be thought of as elastic springs (Fig. 3-4) whose spring constants can be combined with the equivalent spring of the flange. The gasket spring constant, \(k_G\), is
Fig. 3-4 - Gasket and Bolts Modeled as Springs
\[ k_G = \frac{A_G E_G}{2\pi r_0 t_G} \]  

where the gasket area, \( A_G \), is

\[ A_G = 2\pi r_G b_G \]  

\( E_G \) is the elastic modulus and \( t_G \) is the thickness of the gasket. Similarly, the bolt spring constant is

\[ k_B = \frac{A_B E_B}{2\pi r_0 l_B} \]  

where the total bolt area, \( A_B \), for \( n_B \) bolts is

\[ A_B = n_B A_{oB} \]  

\( A_{oB} \) is the stress area of a single bolt, and \( l_B \) is the stress portion of the bolt shaft. With the radial distance

\[ a = r_B - r_G \]  

between the bolt circle and the gasket the equivalent rotational spring is

\[ c_E = \frac{k_G k_B}{k_G + k_B} a^2 \]  

The centroid of both springs is given by the radius

\[ r_a = r_G + \frac{k_B}{k_B + k_G} a = r_B - \frac{k_G}{k_B + k_G} a \]  

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Finally, the displacements in gasket and bolts are, respectively,

$$\delta_G = u + \chi \frac{K_B}{K_G + K_B} a$$  \hspace{1cm} (3.60)$$

and

$$\delta_B = u - \chi \frac{K_G}{K_B + K_G} a$$  \hspace{1cm} (3.61)$$

where $u$ is the axial displacement at the centroid of both springs, and the corresponding changes in gasket and bolt stresses are, respectively,

$$\Delta \sigma_G = \frac{E_G \delta_G}{t_G},$$  \hspace{1cm} (3.62)$$

and

$$\Delta \sigma_B = \frac{E_B \delta_B}{\ell_B}. \hspace{1cm} (3.63)$$

3.4 SEQUENCE OF LOADING CONDITIONS

In the preceding three paragraphs the mathematical apparatus for the analysis of the deformations and stresses of a flange have been presented. It will now be used in the step-by-step analysis of the loading conditions.

The initial loading of the flange occurs when the bolts are torqued to achieve a tight seat of the gasket. This force was computed in Section 2 based on the gasket design requirements. This initial bolt force may be related to the bolt torque applied when the connection is assembled, for
which torquing charts are available. It would go beyond the scope of this report to go into these torquing requirements. For the further discussion a bolt force, \( f_B \), per unit length of circle \( r_o \),

\[
f_B^{(0)} = \frac{n_B \sigma_B^{(0)} A_{oB}}{2\pi r_o} \quad (3.64)
\]

is considered, where \( \sigma_B^{(0)} \) is the stress in the bolts at initial torquing. For a cantilever flange the corresponding applied flange moment is

\[
m^o_F = a f_B^{(0)} \quad (3.65)
\]

It is evident that from Eq. (3.43) the rotation \( \chi \) of the cross section and from Eqs. (3.41) and (3.42) the interface shear \( q_o \) and interface moment \( m_o \) can be computed and the remainder of the analysis of shell and flange be performed as described in paragraphs 3.1 and 3.2.

When the separation of the fluid system is started the internal fluid pressure causes an axial force in the tube

\[
f_T = \frac{p r_o}{2} \quad (3.66)
\]

and a force

\[
f_F = \frac{r^2 - r_i^2}{2r_o} p \quad (3.67)
\]

on the face of the flange. The latter force acts at a radius

\[
r_F = \frac{2}{3} \frac{r_G^2 + r_i r_G + r_i^2}{r_G + r_i} \quad (3.68)
\]
The corresponding applied flange moment is

\[ m_F^{(1)} = f_T (r_a - r_o) + f_F (r_a - r_F) \]  

(3.69)

Another flange moment \( m_F^{(2)} \) is caused by the differential radial displacement, according to Eq. (3.45) and Eq. (3.50). The radial displacement of the shell for the most general case was given by Eq. (3.3). The term attributed to the temperature differential \( \Delta T \) is probably unrealistic for cryogenic applications when assumed that the ring could not experience the same differential, i.e., both ring and shell probably experience simultaneously the same \( \Delta T \) and therefore this term does not produce a \( \Delta w \). The radial expansion of the flange ring due to internal pressure is

\[ w_{ring} = \left( \frac{r_o r_i}{EA} \right) \left( \frac{ph r_i}{r_o} \right) \]  

(3.70)

It is now possible to compute the rotations of the flange. Initially a rotation

\[ \chi^{(o)} = \frac{m^{(o)}}{c_F} \]  

(3.71)

occurs. The corresponding axial displacement is the reference position and taken as

\[ u^{(o)} = 0. \]  

(3.72)

When \( m_F^{(1)} \) and \( m_F^{(2)} \) are applied the rotation is

\[ \chi^{(p)} = \frac{m_F^{(1)} + m_F^{(2)}}{c_F} \]  

(3.73)
and the axial displacement is

\[ u(p) = \frac{f_T + f_F}{K_G + K_B} \]

(3.74)

The gasket and bolt deformations according to Eq. (3.60) and (3.61) are evaluated with \( u(p) \) and \( \chi(p) \). The final stresses in the flange, however, are computed with

\[ \chi(T) = \chi(o) + \chi(p) \]

(3.75)

for which a corresponding \( m_F^T \) can be computed with Eq. (3.49). It is not necessary to repeat here how the stresses in the shell and the flange are computed from the moment \( m_F^T \) and the rotation \( \chi(T) \). In summary, an interface moment \( m_o^T \) and an interface shear \( q_o^T \) are arrived at. Also a radial displacement at point \( A \) (Fig. 3-3) of \( w_o^T \) is computed.

The plastic stresses at the flange neck are generated by increasing the pressure until \( m_o^T \) and \( q_o^T \) become \( m_x^p \) and \( q_x^p \) as in Eqs. (3.25) and (3.26). At the same time \( \chi(T) \) increases to \( \chi^p \) and \( w_o^T \) increases to \( w^p \). The stresses are then

\[ c_x^p = \frac{B}{t^2 - t_n^2} \left( \frac{1}{2k} + \beta \frac{I}{A} + \frac{c}{2k} \right) \chi^p \]

(3.76)

\[ \tau_{xz}^p = \frac{B}{(t-t_n)} \left( \frac{c + \frac{1}{2k}}{\beta \frac{I}{k} + \frac{1}{4k^3}} \right) \chi^p \]

(3.77)
\[ \sigma_p = \frac{E}{r_o} w^p + \nu \sigma_x \]  

so that

\[ \alpha_1 = \frac{t^2 - \frac{t^2}{4} E \frac{1}{r_o} \frac{1}{B} \left( \frac{1}{2k^2} + \beta \frac{1}{A} + \frac{c}{2k} \right) w^p + \nu \]  

and

\[ \alpha_2 = \frac{t + \frac{t}{4}}{4} \frac{c + \frac{1}{2k}}{\frac{1}{2k^2} + \beta \frac{1}{A} + \frac{c}{2k}} \]  

according to Eqs. (3.31) and (3.32).

3.5 ESTIMATE OF THE MOMENT CAPACITY OF THE FLANGE

The capacity of the flange to carry an applied moment of \( m_F \) is exhausted when

\[ m_{Fu} = Y_o \left[ Z_F + Z_T \right] \]  

where \( Y_o \) is the tensile yield strength of the material and \( Z_F \) and \( Z_T \) are the equivalent plastic section moduli of the flange ring and the tube, respectively. This is the same equation as Eq. (2.97). A more conservative assumption would be to let the stresses in the ring just reach the yield stress in the extreme fibers so that the elastic modulus \( S_F \) should be used instead of \( Z_F \). For a rectangular ring cross section with the reduced width \( b \) the two section moduli are

\[ Z_F = \frac{bh^2}{4r_o} \]
The equivalent plastic section modulus of the tube wall, $Z_T$, can be expressed in terms of the expressions derived in Eqs. (3.76) through (3.80) and Eqs. (3.31), (3.32) and (3.35) when

$$
\bar{\sigma} = Y. 
$$

then

$$
Z_T = \frac{t^2 - t_n^2}{4} + \alpha_2 (t - t_n) \frac{h}{2} \sqrt{1 + \alpha_1 + \alpha_1^2 + 3\alpha_2^2} \tag{3.85}
$$

or simply

$$
Z_T = \xi_1 \frac{t^2 - t_n^2}{4} + \xi_2 (t - t_n) \frac{f}{2} \tag{3.86}
$$

The two dimensionless parameters are

$$
\xi_1 = \frac{1}{\sqrt{1 + \alpha_1 + \alpha_1^2 + 3\alpha_2^2}} \tag{3.87}
$$

and

$$
\xi_2 = \frac{\alpha_2}{\sqrt{1 + \alpha_1 + \alpha_1^2 + 3\alpha_2^2}} \tag{3.88}
$$
Section 4
COMPUTER PROGRAM

This section describes the computer program which was developed to implement the design standard and verify the stresses and deformations of the flange. The program is written in FORTRAN IV language for use on the Univac 1108 Exec 8 system. The algorithms of these computer programs are based on the design procedure and the analysis method outlined in the previous two sections. A listing of the code is included. Input instructions for the computer program are given in Appendix C. Example problems are presented in Section 5.

PROGRAM OUTLINE

The program consists of a main program which reads the input data, and four major subroutines in addition to two output routines. These major routines are DESIGN and ANALYS, corresponding to the design and analysis part of the program and PLOTF1 and PLOTF2, which are the two plot routines for the Stromberg-Carlson 4020 plotter. The organization of the entire program is shown on Chart 4-1 and the individual routines are briefly described in Table 4-1. The two routines DESIGN and ANALYS follow principally the sequence of formulas given in Appendixes A and B. The individual variables are easily recognizable and are therefore not explained here in detail.

The program allows the design and analysis of cantilever flanges with flat gaskets and pressure energized seals. The machining of the upper flange surface may be with machined spot faces or with a machined groove. These different options can be turned on by specifying the appropriate values of the variable KOPT(I), as described in the User's instructions in Appendix C.
Chart 4-1 - Organization of the Flange Design and Analysis Program
### Table 4-1

#### PROGRAM DESCRIPTION

<table>
<thead>
<tr>
<th>No.</th>
<th>Symbol</th>
<th>Name</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>DMAIN</td>
<td></td>
<td>Design program for low profile flanges</td>
</tr>
<tr>
<td>2</td>
<td>D021</td>
<td>METALS</td>
<td>Table of metallic materials design properties for tubes, flanges and bolts</td>
</tr>
<tr>
<td>3</td>
<td>D022</td>
<td>GASKET</td>
<td>Table of gasket materials design properties</td>
</tr>
<tr>
<td>4</td>
<td>D001</td>
<td>DESIGN</td>
<td>Design routine for low profile flanges</td>
</tr>
<tr>
<td>5</td>
<td>D010</td>
<td>BOLT</td>
<td>Bolt data handling</td>
</tr>
<tr>
<td>6</td>
<td>D011</td>
<td>BTABL1</td>
<td>Bolt table for open wrenching</td>
</tr>
<tr>
<td>7</td>
<td>D012</td>
<td>BTABL2</td>
<td>Bolt table for socket wrenching</td>
</tr>
<tr>
<td>8</td>
<td>D013</td>
<td>BTABL3</td>
<td>Bolt table for internal wrenching</td>
</tr>
<tr>
<td>9</td>
<td>P001</td>
<td>PLOTF1</td>
<td>Plot routine for low profile flanges with flat gasket and machined spotfaces</td>
</tr>
<tr>
<td>10</td>
<td>P002</td>
<td>PLOTHC</td>
<td>Plot a half circle from IA to IB</td>
</tr>
<tr>
<td>11</td>
<td>P003</td>
<td>PLOTLN</td>
<td>Plot a line</td>
</tr>
<tr>
<td>12</td>
<td>P004</td>
<td>PLOTLB</td>
<td>Plot label</td>
</tr>
<tr>
<td>13</td>
<td>P005</td>
<td>PLOTQC</td>
<td>Plot a quarter circle from IA to IB</td>
</tr>
<tr>
<td>14</td>
<td>P006</td>
<td>PLOTAR</td>
<td>Plot an arrow head for different orientations</td>
</tr>
<tr>
<td>15</td>
<td>P007</td>
<td>PLOTTX</td>
<td>Plot text</td>
</tr>
<tr>
<td>16</td>
<td>P008</td>
<td>DASHLN</td>
<td>Dashed-dotted line</td>
</tr>
<tr>
<td>17</td>
<td>D100</td>
<td>OUTDES</td>
<td>Output of the design routine</td>
</tr>
<tr>
<td>18</td>
<td>A001</td>
<td>ANALYS</td>
<td>Analysis routine</td>
</tr>
<tr>
<td>19</td>
<td>P010</td>
<td>PLOTF2</td>
<td>Plot of the analysis results</td>
</tr>
<tr>
<td>20</td>
<td>D200</td>
<td>OUTAN</td>
<td>Output of the analysis results</td>
</tr>
</tbody>
</table>
The plot routines PLOTF1 and PLOTF2 summarize the design and analysis. The first one plots the geometry of the flange cross section in 1:1 scale. The second one summarizes the stresses and deformations of the tube wall and the flange, using a 1:2 scale for the flange geometry. The layout of the graphs is given on Figs. 4-1 and 4-2. The small numbers refer to x and y coordinate points in the code and are given here to facilitate future modifications in the program.

A list of the entire code is given in this section. The limited scope of this contract did not allow inclusion of all possible flange configurations to be considered in this program with the corresponding plot option. At this point, however, it would be possible to automate the design process further by combining the computer code with a different type of plotting equipment, allowing larger size plots. The SC 4020 plot area is limited to 7 1/2 by 7 1/2 inch.

Sample computer output, printed and plotted, is presented in Section 5.
Fig. 4-1 - Layout of Design Summary SC 4020 Plot
Fig. 4-2 - Layout of Analysis Summary SC 4020 Plot
PROGRAM LISTING
LIST OF ROUTINES IN FLANGE DESIGN AND ANALYSIS PROGRAM

WHDG DL21 (2) (METALS)
WHDG DL22 (3) (GASKET)
WHDG D222 (4) (DESIGN)
WHDG D221 (5) (BOLT)
WHDG D211 (6) (TABLE1)
WHDG D211 (7) (TABLE2)
WHDG D212 (8) (TABLE3)
WHDG D213 (9) (PLOTF1)
WHDG D213 (10) (PLOTHC)
WHDG P213 (11) (PLOTLN)
WHDG P213 (12) (PLOTLC)
WHDG P213 (13) (PLOTWC)
WHDG P213 (14) (PLOTTAR)
WHDG P213 (15) (PLOTIX)
WHDG P213 (16) (DASHLN)
WHDG D116 (17) (OUTDES)
WHDG AG21 (18) (ANALYSIS)
WHDG A212 (19) (PLOTF2)
WHDG D220 (20) (OUTAN)
DESIGN PROGRAM FOR LOW PROFILE FLANGES

DIMENSION HEAD(12), AD(22), KOPT(10)
DIMENSION A(9,4), SRES(15,4), STR(5,4), AP(8)
DATA (AD(1), I = 1, 22) / H /
READ (15, 1) (AD(I), I = 1, 12)
CALL IDENT(9, AD)
READ (15, 3) NCASES
CASE = 1
READ (15, 1) (HEAD(I), I = 1, 12)
READ (15, 2) P, DI, DELT, HT
READ (15, 2) PF, BF, FS, F
READ (15, 3) IT, IF, I8, IG
IF (IT .EQ. 6) READ (15, 2) ET, ANUT, KHOT, ALFAT, FYT, FIUT
IF (IF .EQ. 6) READ (15, 2) EF, ANUF, KMUF, ALFAF, FYF, FIUF
IF (IF .EQ. 6) READ (15, 2) EB, ANUB, RHOB, ALFAB, FYB, FIB
IF (IF .EQ. 6) READ (15, 2) EG, AKG, SG, ALFAG, AMUG, GAMU, GAM5, HG, SP
IF (IG .GT. 6) READ (5, 2) MS, BS, HR
IF (IG .GT. 6) CALL METALS (IT, ET, ANUT, KHOT, ALFAT, FYT, FIUT)
IF (IF .GT. 6) CALL METALS (IF, EF, ANUF, KMUF, ALFAF, FYF, FIUF)
IF (IF .GT. 6) CALL METALS (IF, EB, ANUB, RHOB, ALFAB, FYB, FIB)
IF (IG .GT. 6) CALL GASKET (IG, EG, AKG, SG, ALFAG, AMUG, GAMU, GAM5, HG, SP)
READ (15, 3) KOPT(I), I = 1, 10
DIMENSION TUBMTL(2), FLAMTL(2), BOLMTL(2), GASMTL(2)
READ (15, 1) (TUBMTL(I), I = 1, 2), (FLAMTL(I), I = 1, 2)
READ (15, 3) NPHASE
READ (15, 2) DELTAT
101 FORMAT (111)
102 FORMAT (* NOMINAL PRESSURE P = * * F10.3, + * PSI */
* NOMINAL DIAMETER DI = * F16.3, + INCH */
* TUBE THICKNESS T = * F16.3, + INCH */
* TUBE THICKN TOLK DT = * F16.3, + INCH */
* HEIGHT TO WELD HT = * F16.3, + INCH /// */
103 FORMAT (* PROOF FACTOR PF = * F10.3, 3/*
* BURST FACTOR BF = * F10.3, 3/*
* SAFETY FACTOR FS = * F10.3, 3/*
* GASKET FACTOR GF = * F10.3, 3/*
104 FORMAT (* PROPERTIES OF TUBE MATERIAL */)
105 FORMAT (* PROPERTIES OF FLANGE MATERIAL */)
106 FORMAT (* PROPERTIES OF BOLT MATERIAL */)
107 FORMAT (* PROPERTIES OF GASKET MATERIAL */)
108 FORMAT (* MATERIAL TABLE NO. I = 15 */
* ELASTIC MODULUS E = * E16.8, * PSI */
* POISSON'S RATIO NRU = * F10.3, 3/*
* DENSITY RHO = * F10.4, 9, LB/CUBIC- INCH */
LMSI-HREC TR D306492

MAIN

1 (MAIN PROGM)

50  THERM EXP COEFF
51  TENSILE YIELD STR
52  ULTIMATE TENS STR
53  MATERIAL TABLE NO.
54  ELASTIC MODULUS
55  YIELD STRENGTH
56  SEATING STRESS
57  THERM EXP COEFF
58  COEFF OF FRICTION
59  WIDTH COEFFICIENT
60  THERM COEFFICIENT
61  GASKET THICKNESS
62  SEATING STRESS RATE

110 FORMAT(* OPTIONS*)
111 FORMAT(* PICTURE*)
119 FORMAT(* PRESSURE ACTIVATED SEAL*)
120 FORMAT(* NUMBER OF PHASES TO BE CONSIDERED IN THE ANALYSIS = *)
121 FORMAT(* COMPUTED THICKNESS = *)

C

WRITE(6,101)
WRITE(6,102) P,DI,T,DELT,HT
WRITE(6,103) PF,BF,FS,GF
 WRITE(6,104)
WRITE(6,108) IT,ET,ANUT,RHOT,ALFA,FY,FTUT
WRITE(6,105)
WRITE(6,108) IF,EF,ANUF,RHOF,ALFA,FY,FTUF
WRITE(6,106)
WRITE(6,108) IB,EB,ANUB,RHUB,ALFA,FY,FTUB
IF(IG.LT.1) GO TO 20
WRITE(6,107)
WRITE(6,109) IG,EG,AKG,SG,ALFA,AMUG,GAMU,GAMS,HG,SP
GO TO 25
WRITE(6,119) HS,BS,HR
WRITE(6,110)
WRITE(6,111) I=KOPT(I),I=1,10)
WRITE(6,120) NPHASE,DELTAT

C

CALL DESIGN(P,DI,T,DELT,PF,BF,FS,GF)

106  ET,ANUT,RHOT,ALFA,FY,FTUT
107  EF,ANUF,RHOF,ALFA,FY,FTUF
108  EB,ANUB,RHUB,ALFA,FY,FTUB
109  EG,AKG,SG,ALFA,AMUG,GAMU,GAMS,HG,SP,HS,BS
110  KOPT,ADD,WEIGHT,PB
111  M,RI,KG,RB,RFIL,SPOT,DHOLE,DSPO,N,BG,HT)
WRITE(6,121) T

C

CALL PLTFIL(B,H,T,R1,KG,RFIL,SPOT,DHOLE,DSPO,N,BG,HS,HT)

4-10
DMAIN (MAIN PROGRAM)

112 C
113 CALL OUTDES(HEAD, AO, WEIGHT, KOPT, T
114 * B, H, R, K, RB, RFIL, RSPOT, DHOLE, DSPOT, N, DU, HT)
115 C
116 CALL ANALYSIS(DI, T, DELT, PF, BF, FS, GF
117 * ET, ANUT, KHOT, ALFA, FT YT, FT UF
118 * EF, ANUF, RHOF, ALFAF, FYF, FT UF
119 * RB, ANUB, RHOB, ALFAB, FYB, FTUB
120 * EG, AK, SG, ALFAG, AMUG, GAMUG, GAMS, HG, HS, BS
121 * KOPT, AO, NPHASE, DELTAT, PB
122 * B, H, RI, KG, RB, RFIL, RSPOT, DHOLE, DSPOT, N, DU, HT
123 * A, SRES, STR, AP, HEAD)
124 C
125 CALL OUTAN(HEAD, A, SRES, STR)
126 C
127 ICASE = ICASE + 1
128 IF (ICASE .LE. NCASES) GO TO 10
129 C
130 CALL ENDJOB
131 STOP
132 END

WHOG D021 (2) (METALS)

WPRT.C D021
FURPUN 2411-03/10-14:53
SUBROUTINE METALS(N,E,ANU,RHO,ALFA,FTY,FTU)

TABLE OF METALLIC MATERIALS DESIGN PROPERTIES FOR
TUBES, FLANGES AND BOLTS

K.M. LEIMBACH, 28 NOVEMBER 1972

COMMUN/PROMTX/P(10,10)

DATA((P(I,J),J=1,6),I=1,10)/

- 9.9 E+6, 0.33 - 0.098 - 12.5 E-6, 35.0 E+3, 42.0 E+3
- 9.9 E+6, 0.33 - 0.098 - 12.5 E-6, 32.2 E+3, 30.1 E+3
- 9.9 E+6, 0.33 - 0.098 - 12.5 E-6, 47.0 E+3, 39.0 E+3
- 28.0 E+6, 0.30 - 0.288 - 9.5 E-6, 35.0 E+3, 70.0 E+3
- 28.0 E+6, 0.30 - 0.288 - 9.5 E-6, 30.0 E+3, 65.0 E+3
- 28.0 E+6, 0.30 - 0.288 - 9.5 E-6, 25.0 E+3, 55.0 E+3
- 28.0 E+6, 0.30 - 0.288 - 9.5 E-6, 20.0 E+3, 45.0 E+3
- 28.0 E+6, 0.30 - 0.288 - 9.5 E-6, 15.0 E+3, 35.0 E+3
- 28.0 E+6, 0.30 - 0.288 - 9.5 E-6, 10.0 E+3, 25.0 E+3

LEGEND= AL 6061-T6 (1) AT RT; (2) AT 200F
AL 2024-T3 (3) AT RT; (4) AT 200F
304 SS (5) AT RT; (6) AT 200F; (7) AT 600F
A286 (8) AT RT; (9) AT 200F; (10) AT 600F

E=P(N,1)
ANU=P(N,2)
RHO=P(N,3)
ALFA=P(N,4)
FTY=P(N,5)
FTU=P(N,6)

RETURN
END

WMG D022 31 (GASKET)

WPR1,C D022
FURPUK 24H-03/10-14:53
TABLE OF GASKET MATERIALS DESIGN PROPERTIES

COMMON/PRUGSK/P(5,9)

DATA((P(I,J), j=1,9),i=1,5)/
GASKET(INK,ALK,SG,ALFA,AMU,GAMS,HG,SP)

E=P(N,1)
AKG=P(N,2)
SG=P(N,3)
ALFA=P(N,4)
AMU=P(N,5)
GAMS=P(N,6)
AMU=P(N,7)
HG=P(N,8)
SP=P(N,9)

RETURN
END
SUBROUTINE DESIGN(P,DI,T,DELT,PF,BF,FS,GF
*  .ET,ANUT,RMOF,ALFA,FTY,T,FTUF
*  .ET,ANUF,RMOF,ALFA,FTY,F,FTUB
*  .EF,ANUB,RMB,ALFA,FTY,F,FTUB
*  .EG,AKG,SG,ALFAG,AMUG,GAMG,HG,SP,HS,BS
*  .KUPT,AQB,WEIGHT,PB
*  .B,H,RI,RG,RB,RFIL,RSPOT,DHOLE,DSPUT,NO,BG,MT)

C C DESIGN ROUTINE FOR LOW PROFILE FLANGES

C K.H.LEIBNACH, 28 NOVEMBER 1972

C DIMENSION KOPT(10)

C TUBE THICKNESS

JBOLT=KOPT(4)

KI=DI/2.

IF(KOPT(1),EQ,0) GO TO 40

IF(KOPT(1),EQ,1) GO TO 10

IF(KOPT(1),EQ,2) GO TO 20

IF(KOPT(1),EQ,3) GO TO 30

10 T=FS* P*KI/FRTY

go to 40

20 ALAMB=.75

T1=BF*P*KI/(FTUT*ALAMB)+2.*DELT

T2=PF*P*KI/(FTUT*ALAMB)+2.*DELT

T=AMAX(T1,T2)

go to 40

29 T1=1.1*PF*P*KI/(FTUT+.4*PF*P)

T2=1.1*BF*P*KI/(FTUT+.4*BF*P)

T=AMAX(T1,T2)

30 CONTINUE

C BOLT SIZE

DB=T

IU=0

45 CALL BOLT(DB,ETA1,ETA1,ETA2,AB,DHOLE,DSPUT,RSPOT,SIZE,JBOLT,10)

47 E1=ETA1*DB

E2=ETA2*DB

IF(T*G1.0) RFIL=0.375

IF(T*LT.0.25) RFIL=0.3125

IF(T*LT.0.15) RFIL=0.25

IF(T*LT.0.10) RFIL=0.1875

IF(T*LT.0.05) RFIL=0.125

GO TO 70

E1=ETA2*DB

E2=E1

IF(T*G1.0) RFIL=0.125

IF(T*LT.0.25) RFIL=0.0625

IF(T*LT.0.15) RFIL=0.0375

IF(T*LT.0.10) RFIL=0.0156

IF(T*LT.0.05) RFIL=0.0062

70 CONTINUE
C BOLT CIRCLE RADIUS
C #6=4.625
C #2=4.75
IF (KOPT(2).EQ.1) KB=RI+T+C1+E1
IF (KOPT(2).EQ.2) KB=RI+T+2.*HOLE+E1
C FLANGE WIDTH
B=KB+E2-R1
C GASKET WIDTH
IF (KOPT(3).GT.0) GO TO 60
RG=4.35+(KB-.5*DAMB+R1)
IF (KOPT(5).EQ.0) BG=PF*PG/12*(GAMU*AKG-GAMS*SG*GF))
IF (KOPT(5).EQ.1) BG=PF*PG/12*(GAMU*AKG-GAMS*SP*PF*GF))
IF (KOPT(5).EQ.2) BG=5.*HOLE-R1-2.*C2
IF (KOPT(6).EQ.0) GO TO 76
KG=KB-.5*DAMB=.5*BU-C2
R1=KG-.5*BG+C2
R2=RI
IF (R1.GT.R2) GO TO 75
RG=K1+.5*dG+C2
74 RB=KG+.5*BG+C2
75 B=RB+E2-R1
79 GO TO 90
GO TO 100
IF (R1.GT.R2) GO TO 75
R2=R1
GO TO 74
78 GO TO 74
GO TO 74
77 RB=KG+.5*BG+.5*DAMB-C2
78 75 B=RB+E2-R1
90 CONTINUE
C BOLT FORCES
PI=3.141593
95 IF (KOPT(3).GT.0) GO TO 100
96 IF (KOPT(5).EQ.2) GO TO 98
97 PB1=2.*PI*RG*BG*GAMU*AKG
98 IF (KOPT(5).EQ.1) PB2=2.*PI*RG*BG*GAMS*SG*GF+PI*RG*2*PF*P
99 IF (KOPT(5).EQ.1) PB2=2.*PI*RG*BG*GAMS*SP*PF*GR+PI*KG**2*PF*P
100 PB=AMAX1(PB1,PB2)
101 GO TO 102
GO TO 100
102 PB2=2.*PI*RG*BG*GAMS*SP*PF*GR+PI*RG*2*PF*P
103 PB1=PB2
104 SG2=PB2/12.*PI*BG*GAMU*RG)
105 IF (SG2.LT.SG1) P01=2.*PI*RG*BG*GAMU*BG
106 PB=AMAX1(PB1,PB2)
107 GO TO 110
108 PB=PI*KG**2*PF*P
109 CONTINUE
110 C NUMBER OF BOLTS
111
ENB1 = PB / (FTYB * AOB)
ENB2 = (BF / PF) * PB / (FTYB * AOB)
ENB = MAX1 (ENB1, ENB2)
NB = ENB
IF (NB .GE. 6) GO TO 115
NB = 6
PB = 6 * FTYB * AOB
CONTINUE

C BOLT SPACING
ENB = NB
S = 2 * PI * KB / ENB
ID = 1
SOVD = S / DB
IF (SOVD .GT. B + U) GO TO 120
IF (SOVD .LT. ETAJ) GO TO 133
GO TO 140
ISIZE = ISIZE + 1
IF (ISIZE .LT. LT + 1) GO TO 140
GO TO 45
ISIZE = ISIZE + 1
IF (ISIZE .GT. 14) GO TO 140
GO TO 45
CONTINUE

C FLANGE HEIGHT
E = KB - RG
KU = RT + 5*T
TN = T / 2*
EMFU = FS * PB * E / (2 * PI * RO)
BBAN = B - DHOLE * SQRT (DHOLE / S)
ZETA1 = W * BU
ZETA2 = W * 18
CAPA = FTYF * BBAN / (6 * KU)
CAPC = FTYF * ZETA2 * (T - IN) / 2*
RTSW = CAPB * (2 - 4 * CAPA * CAPC)
RT = SQRT (RTSW)
H = (RT - CAPB) / (2 * CAPA)

C CHECK FLANGE HEIGHT
SOVH = S / H
IF (SOVH .GT. 3) H = S / 3

C WEIGHT COMPUTATION
RW = .5 * (2 * RT + 8)
AW = (B - T) * H
VOL = 2 * PI * RT * AW
WEIGHT = RHOF * VOL
RETURN
END
DO10 (5) (JBLT)

34600*TPFS.DO10

1 SUBROUTINE BOLT(D,ETA0,ETA1,ETA2,AUG,OHOLE,U5POT,K5POT,ISIZE
2     JBLT,1D)
3     COMMON/BOLTDT/DX(14,5)
4     C
5     IF(JBLT.EQ.1) CALL BTABL1
6     IF(JBLT.EQ.2) CALL BTABL2
7     IF(JBLT.EQ.3) CALL BTABL3
8     IF(JBLT.LE.0.OR.JBLT.GE.4) STOP
9     C
10     IF(I0.GT.0) GO TO 10
11     I=1
12     DO=1=DI
13     IF(D0.LE.IO) GO TO 20
14     I=I+1
15     GO TO 15
16     I=ISIZE
17     C
18     DO=1=DI
19     ETA0=DX(1,1)
20     ETA1=DX(1,2)
21     ETA2=DX(1,3)
22     AUG=DX(1,5)
23     OHOLE=D++O0S
24     U5POT=Z*ETA1*0
25     K5POT=O62
26     IF(I0.EQ.0) ISIZE=I
27     C
28     RETURN
29     EN0

)HDG DO11 (6) (BTABL1)

)PRT,C DO11

:UPUR 24HI-03/16-14:53

4-18
SUBROUTINE TABL1  
BOLT TABLE FOR OPEN WRENCHING  
14 SIZES FROM .25 IN TO 1.5 IN NOMINAL DIAMETER  
COMMON/BOLTO1/DX(14,5)  
COMMON/BOLTO2/DX(14,5)  
DATA((DX(I,J),J=1,5),I=1,14)  
10  *     .2500  3.00  2.00  1.50  *  .63182  
11  *     .3125  2.60  1.80  1.40  *  .05293  
12  *     .3750  2.67  1.67  1.33  *  .07749  
13  *     .4375  2.57  1.57  1.29  *  .10631  
14  *     .5000  2.50  1.62  1.24  *  .14197  
15  *     .5625  2.45  1.56  1.22  *  .18194  
16  *     .6250  2.40  1.50  1.20  *  .22680  
17  *     .7500  2.33  1.49  1.19  *  .33390  
18  *     .8750  2.35  1.43  1.07  *  .46173  
19  *     .9375  2.25  1.37  1.00  *  .49574  
20  *     .9875  2.22  1.33  1.00  *  .78327  
21  *     1.0625  2.25  1.40  1.00  *  .92995  
22  *     1.1250  2.23  1.36  1.00  *  1.15480  
23  *     1.5000  2.17  1.33  1.00  *  1.40525  
24  DO 10  J=1,5  
25  DO 10  I=1,14  
26  10  U(I,J)=DX(I,J)  
27    RETURN  
28    END  

DO 12  I=1,14  
12   DO 10  J=1,5  
10   U(I,J)=DX(I,J)  
28   RETURN  
29   END
SUBROUTINE BTABL2

BOLT TABLE FOR SOCKET WRENCHING

14 SIZES FROM .25 IN TO 1.5 IN NOMINAL DIAMETER

COMMON/BOLTU2/0X(14,5)
COMMON/BOLTD2/0X(14,5)

DATA(10X(I,J),J=1,5),I=1,14)/

10  2560.  2.76  1.60  1.40  0.3102,
11  3125.  2.53  1.50  1.20  0.5243,
12  3750.  2.37  1.33  1.14  0.7749,
13  4375.  2.26  1.25  1.14  1.0631,
14  5000.  2.18  1.20  1.10  1.4194,
15  5625.  2.08  1.12  1.11  1.8194,
16  6250.  2.00  1.12  1.12  2.2664,
17  7500.  1.92  1.17  1.17  3.4460,
18  8750.  1.84  1.14  1.14  4.6173,
19  10000.  1.79  1.25  1.10  6.0574,
20  11250.  1.74  1.22  1.07  7.6327,
21  12500.  1.69  1.19  1.04  9.2955,
22  13750.  1.63  1.16  1.02  1.16418,
23  15000.  1.58  1.13  1.00  1.40525/

DO 10 I=1,14
DO 10 J=1,5
10 DO(1,1)=0X(I,J)
RETURN
END

WHDG D013 (6) (BTABL3)

APRT,C D013
FURPUR 24H1-03/10-14:53

4-20
SUBROUTINE BTL3

BOLT TABLE FOR INTERNAL WRENCHING

14 SIZES FROM .25 IN TO 1.5 IN NOMINAL DIAMETER

COMMON/BOLTD3/DX(14,5)

COMMON/BOLDT3/D(14,5)

DATA((DX(I,J),J=1,5),I=1,14)/

DO 1 J=1,5

RETURN

END

SUBROUTINE BTL3

COMMON/BOLTD3/DX(14,5)

COMMON/BOLDT3/D(14,5)

DATA((DX(I,J),J=1,5),I=1,14)/

DO 1 J=1,5

RETURN

END
SUBROUTINE PLOTF1(B,H,T,R,RI,RF,RG,RB,RF,LHOLE,DSH,SPOT,N,BY,MG)

C PLOT ROUTINE FOR LOW PROFILE FLANGE WITH FLAT GASKET AND
C MACHINED SPOTFACES FOR THE BOLTS
C K*R*LEIMBACH, 7 NOVEMBER 1972

C DIMENSION FLAMTL(2),BOLMTL(2),GASMTL(2),HEAD(12)
C & DATA1(11),DATA2(11),DATA3(3),DATA4(3),DATA5(3)
C & DATA6(4),DATA7(11),DATA8(11),DATA9(11),DATA10(11)
C & DATA11(5),DATA12(5)
C & X(100),Y(100),IX(100),Y(100)
C & KOPT(10)

C CALL FRAMEV(0)
CALL SCRECT(31,31,991,991)
CALL PRINTV(72,HEAD,41,1003)
A=3.*75
CALL XSCALV(-A,A,0.,0.)
CALL YSCALV(-A,A,0.,0.)

C I16=0
D=DHOLE
X(1)=B/2*
Y(1)=H/2*
X(2)=X(1)
Y(2)=-Y(1)
IF(KOPT(2) .EQ.2) GO TO 210
X(3)=-X(1)+T+RFIL
Y(3)=Y(2)
X(4)=X(3)-RFIL
Y(4)=Y(3)+RFIL
GO TO 215

E1=RI+R-B
X(3)=X(2)=2.*E1
Y(3)=Y(2)
X(4)=X(3)-2.*RFIL
Y(4)=Y(2)
CONTINUE
X(5)=X(4)
Y(6)=Y(5)+HT
X(6)=X(5)-T
Y(6)=Y(5)
X(7)=X(6)
Y(7)=Y(11)
X(8)=X(7)+RB-RI+D/2*
Y(8)=Y(1)
X(9)=X(8)
Y(9)=Y(2)
X(10)=X(8)=D
Y(10)=Y(1)
X(11)=X(10)
Y(11)=Y(2)
(9) (PLOTFL)

```
56  X(12)=X(8)+D/2.
57  Y(12)=Y(1)+.25
58  X(13)=X(12)
59  Y(13)=Y(2)+.25
60  Y(14)=Y(2)
61  X(14)=X(13)-D*SPOT/2.*X*SPOT
62  X(15)=X(14)*KSPOT
63  Y(15)=Y(2)+KSPOT
64  X(16)=X(15)
65  IF(X(16)*LT*X(3)) GO TO 95
66  X(14)=X(2)
67  X(15)=X(3)
68  X(16)=X(3)
69  Y(14)=Y(3)
70  Y(15)=Y(3)
71  Y(16)=Y(3)
72  110=1
73  GO TO 102
74  95 CONTINUE
75
76  C
77  DX=X(3)-X(16)
78  IF(DX*GE.*RFIL) GO TO 101
79  DY=SQR(J(RFIL**2+DX**2))
80  DYBAR=RFIL-DY
81  Y(16)=Y(2)+DYBAR
82  GO TO 102
83  101 X(16)=X(4)
84  Y(16)=Y(4)
85  102 CONTINUE
86
87  C
88  X(17)=X(7)+RG-RG-1+BG/2.
89  Y(17)=Y(11)
90  X(18)=X(17)-BG
91  Y(18)=Y(1)
92  X(19)=X(17)
93  Y(19)=Y(1)-HG
94  X(20)=X(18)
95  Y(20)=Y(19)
96
97  C
98  X(21)=X(18)
99  Y(21)=Y(20)+.125
100 X(22)=X(18)
101 Y(22)=Y(21)+.375
102 X(23)=X(18)
103 Y(23)=Y(22)+.125
104 X(24)=X(6)+.2*0
105 Y(24)=Y(23)
106 X(25)=X(17)
107 Y(25)=Y(24)+.25
108 X(26)=X(17)
109 Y(26)=Y(25)+.750
110 X(27)=X(17)
111 Y(27)=Y(26)+.125
112 X(28)=X(24)
113 Y(28)=Y(27)
```

4-23
\( x(29) = x(1) + 1.25 \)
\( y(29) = y(1) \)
\( x(30) = x(29) + 0.75 \)
\( y(30) = y(1) \)
\( x(31) = x(29) + 0.375 \)
\( y(31) = y(1) \)
\( x(32) = x(31) \)
\( y(32) = y(1) + H/2 = 1.25 \)
\( x(33) = x(31) \)
\( y(33) = y(32) + 0.25 \)
\( x(34) = x(31) \)
\( y(34) = y(2) \)
\( x(35) = x(31) + 1.25 \)
\( y(35) = y(2) \)
\( x(36) = x(2) + 1.25 \)
\( y(36) = y(2) \)
\( x(37) = x(30) - 0.125 \)
\( y(37) = y(1) \)
\( x(38) = x(37) \)
\( y(38) = y(1) + (H+HT)/2 = 1.25 \)
\( x(39) = x(37) \)
\( y(39) = y(38) + 0.25 \)
\( x(40) = x(37) \)
\( y(40) = y(5) \)
\( x(41) = x(30) \)
\( y(41) = y(6) \)
\( x(42) = x(5) + 1.25 \)
\( y(42) = y(5) \)
\( x(43) = x(31) + 1.25 \)
\( y(43) = y(19) \)
\( x(44) = x(19) + 1.25 \)
\( y(44) = y(19) \)
\( x(45) = x(31) \)
\( y(45) = y(19) \)
\( x(46) = x(31) \)
\( y(46) = y(19) + 0.375 \)
\( x(47) = x(24) \)
\( y(47) = y(1) + 1.25 \)
\( x(48) = x(1) \)
\( y(48) = y(47) \)
\( x(49) = x(24) \)
\( y(49) = y(2) + 1.25 \)
\( x(50) = x(12) \)
\( y(50) = y(49) \)
\( x(51) = x(24) \)
\( y(51) = y(5) + 1.25 \)
\( x(52) = x(6) \)
\( y(52) = y(51) \)
\( x(53) = x(51) \)
\( y(53) = y(51) \)
\( x(54) = x(5) + 0.375 \)
\( y(54) = y(51) \)
\( y(55) = y(2) + DR \)

\( DR = RFIL * (1 - SQRT(2 * H / 2) \)
\( x(55) = x(5) + DR \)
\( y(55) = y(2) + DR \)
IF (KOPT(2).EQ.2)  Y(55)=Y(2)*DR
X(56)=X(55)*.25
Y(56)=Y(55)*.25
X(57)=X(4)*.1
Y(57)=Y(56)
X(58)=Y(2)
X(59)=X(12)*.1
Y(59)=Y(58)
X(60)=X(12)
Y(60)=Y(59)
X(61)=X(17)
Y(61)=Y(17)-HS
X(62)=X(17)+BS
Y(62)=Y(61)
X(63)=X(62)
Y(63)=Y(17)

310 CONTINUE
C
NP=63
C
DO 10 I=1,NP
CALL XSCF1(X(I),IX(I),IERR)
CALL YSCF1(Y(I),IY(I),IERR)
10 CONTINUE
C
CALL PLOT1N(1,2,IX,1Y)
CALL PLOT1N(2,3,IX,1Y)
IF (KOPT(2).EQ.1) CALL PLOTUC(3,4,IX,1Y)
IF (KOPT(2).EQ.2) CALL PLOTHC(3,4,IX,1Y)
CALL PLOT1N(4,5,IX,1Y)
CALL PLOT1N(5,6,IX,1Y)
CALL PLOT1N(6,7,IX,1Y)
CALL PLOT1N(7,1,IX,1Y)
GO TO 320
320 CALL PLOT1N(18,20,IX,1Y)
CALL PLOT1N(20,19,IX,1Y)
CALL PLOT1N(19,17,IX,1Y)
CALL PLOT1N(17,61,IX,1Y)
CALL PLOT1N(61,62,IX,1Y)
CALL PLOT1N(62,63,IX,1Y)
CALL PLOT1N(63,4,IX,1Y)
325 CONTINUE
C
CALL PLOT1N(8,9,IX,1Y)
CALL PLOT1N(10,11,IX,1Y)
IF (KOPT(2).EQ.2) GO TO 220
CALL PLOTUC(3,14,IX,1Y)
IF (16*EQ.1) GO TO 96
CALL PLOTUC(14,15,IX,1Y)
96 CONTINUE
C
CALL PLOT1N(15,16,IX,1Y)
220 CONTINUE
C
CALL DASHLN(12,13,IX,1Y)
C IF(KOPF(3).EQ.1) GO TO 330
CALL PLOTLN(17,19,IX,IY)
CALL PLOTLN(18,20,IX,IY)
CALL PLOTLN(19,20,IX,IY)
CONTINUE
CALL PLOTLN(21,22,IX,IY)
CALL PLOTLN(23,24,IX,IY)
CALL PLOTLN(25,26,IX,IY)
CALL PLOTLN(27,28,IX,IY)
CALL PLOTLN(29,30,IX,IY)
CALL PLOTLN(31,32,IX,IY)
CALL PLOTLN(33,34,IX,IY)
CALL PLOTLN(35,36,IX,IY)
CALL PLOTLN(37,38,IX,IY)
CALL PLOTLN(39,40,IX,IY)
CALL PLOTLN(41,42,IX,IY)
CALL PLOTLN(43,44,IX,IY)
CALL PLOTLN(45,46,IX,IY)
CALL PLOTLN(47,48,IX,IY)
CALL PLOTLN(49,50,IX,IY)
CALL PLOTLN(51,52,IX,IY)
CALL PLOTLN(53,54,IX,IY)
CALL PLOTLN(55,56,IX,IY)
CALL PLOTLN(56,57,IX,IY)
CALL PLOTLN(58,59,IX,IY)
CALL PLOTLN(59,60,IX,IY)
CALL PLOTAR(1,23,IX,IY)
CALL PLOTAR(1,24,IX,IY)
CALL PLOTAR(1,27,IX,IY)
CALL PLOTAR(1,31,IX,IY)
CALL PLOTAR(1,3,IX,IY)
CALL PLOTAR(1,27,IX,IY)
CALL PLOTAR(1,27,IX,IY)
CALL PLOTAR(1,31,IX,IY)
CALL PLOTAR(1,3,IX,IY)
CALL PLOTAR(1,31,IX,IY)
DATA(DATA(I),I=1,1)/6H DIA/
DATA(DATA2(I),I=1,1)/6H R /
DATA(DATA3(I),I=1,1)/18H DIA, HOLES/
DATA(DATA4(I),I=1,1)/18H DIA SPOTFACE /
DATA(DATA5(I),I=1,1)/18H R FILLET /
DATA(DATA6(I),I=1,1)/24H PRESSURE PSIG /
DATA(DATA7(I),I=1,1)/66H FLANGE MATERIAL FTY= KSI, FTU= KS /
DATA(DATA8(I),I=1,1)/66H BOLT MATERIAL FTY= KSI, FTU= KS /
DATA(DATA9(I),I=1,1)/66H GASKET MATERIAL SFA /
DATA(DATA10(I),I=1,1)/66H STRESS KSI /
DATA(DATA11(I),I=1,1)/66H WEIGHT OF FLANGE LB /
DATA\(\text{DATA12(1)}\), \(i=1,5\)/3UH PRESSURE ENERGIZED SEAL

C
DGO=2.*RG+BG
DGI=DGO=2.*BG
DI=2.*RI
DB=2.*HR
DFO=DI+2.*B
HWELE=D+HT

CALL PLOTLB(28, 24, 5, IX, IY, DGO)
CALL PLOTTX(28, 56, 5, IX, IY, DATA1, 6)
CALL PLOTLB(24, 24, 5, IX, IY, DGI)
CALL PLOTTX(24, 56, 5, IX, IY, DATA1, 6)
CALL PLOTLB(47, 24, 5, IX, IY, DFO)
CALL PLOTTX(47, 56, 5, IX, IY, DATA1, 6)
CALL PLOTLB(49, 24, 5, IX, IY, DB)
CALL PLOTTX(49, 56, 5, IX, IY, DATA1, 6)
CALL PLOTLB(51, 24, 5, IX, IY, DI)
CALL PLOTTX(51, 56, 5, IX, IY, DATA1, 6)
CALL PLOTLB(53, 24, 5, IX, IY, T)
CALL PLOTLB(57, 20, 5, IX, IY, FFL)
CALL PLOTTX(57, 52, 5, IX, IY, DATA2, 6)
CALL PLOTLB(60, 29, 5, IX, IY, DMOL)
CALL PLOTTX(60, 56, 5, IX, IY, DATA3, 18)
1PL=IX(160)+120
JPL=IY(60)+5
ON=N

IF(KOPT(2), EQ, 2) GO TO 346
CALL PLOTLB(60, 24, -15, IX, IY, DSpot)
CALL PLOTTX(60, 56, -15, IX, IY, DATA4, 18)
CALL PLOTLB(60, 24, -35, IX, IY, RSpot)
CALL PLOTTX(60, 56, -35, IX, IY, DATA5, 18)
CONTINUE 340

CALL PLOTLB(32, -24, 10, IX, IY, H)
CALL PLOTLB(38, -24, 10, IX, IY, HWELE)
CALL PLOTLB(43, 24, -20, IX, IY, G)
CALL PLOTTX(26, -300, -50, IX, IY, DATA7, 66)
CALL PLOTTX(26, -300, -70, IX, IY, DATA8, 66)
IF(KOPT(3), GT, 0) GO TO 50
CALL PLOTTX(26, -300, -90, IX, IY, DATA9, 66)
CALL PLOTTX(26, -300, -100, IX, IY, DATA10, 66)
GO TO 51

50 CALL PLOTTX(26, -300, -120, IX, IY, DATA12, 30)
CONTINUE 51

CALL PLOTTX(26, -156, -50, IX, IY, FLMTL, 12)
CALL PLOTTX(26, -156, -70, IX, IY, BULMTL, 12)
IF(KOPT(3), GT, 0) GO TO 60
CALL PLOTTX(26, -156, -90, IX, IY, GASMTL, 12)
CONTINUE 60

C
FTYF=FTYF/1000*
FTUF=FTUF/1000*
FTYB=FTYB/1000*
FTUB=FTUB/1000*
SG=SG/1000*
FYG=FYG/1000*
336      CALL PLOTLB(26,12,-50,IX,1Y,FTYF)
337      CALL PLOTLB(26,116,-50,IX,1Y,FITUF)
338      CALL PLOTLB(26,12,-70,IX,1Y,FTYB)
339      CALL PLOTLB(26,116,-70,IX,1Y,FTUB)
340      IF (IKOPT(3).GT.0) GO TO 70
341      CALL PLOTLB(26,116,-90,IX,1Y,5G)
342      CALL PLOTLB(26,116,-109,IX,1Y,FYG)
343      70 CONTINUE
344      CALL PLOTTX(6,26U,50,IX,1Y,DATA6,24)
345      IXP=IX(6)+120
346      IYP=1Y(6)+50
347      CALL LBLV(P,IXP,1YP,4,1,4)
348      CALL LBLV(DN,1PL,JPL,3,1,3)
349      CALL PLOTTX(6,200,30,IX,1Y,DATA11,30)
350      IWX=IX(6)-56
351      IYW=1Y(6)+30
352      CALL LBLV(HEIGHT,IXW,1YW,6,1,2)
353      RETURN
354      END

WHDG P002 (10) (PLOTHC)

WPNTC P002
FURPUR 24H1-03/10-14:53
SUBROUTINE PLTHC(IA, IB, IX, IY)

C**** PLOT A HALF CIRCLE FROM IA TO IB
C**** K. R. LEIMBACH, 8 NOVEMBER 1972

C

DIMENSION IX(1), IY(1)

N=20
AN=N
AP=3.1415926/AN
IR=(IX(IA)-IX(IB))/2
II=IX(IA)
JJ=IY(IA)
NP1=N+1
DO 10 J=1, NP1
AJ=J
APJ=(AJ-1.)*AP
CPJ=COS(APJ)
SPJ=SIN(APJ)
H=IR
DX=H*(1.-CPJ)
DY=H*SPJ
IDX=DX
IDY=DY
IZ=IX(IA)-IDX
JZ=IY(IA)-IDY
CALL LINEV(II, JJ, IZ, JZ)
10 II=II+
11 J1=J2
12 CONTINUE

C
RETURN
END

@HDG P003 (111) (PLT LN)

@PRRT.C P003
FURPUR 24M1-03/16-14:53
SUBROUTINE PLOTLN(IA,IB,IX,IY)
C
C    PLOT A LINE
C
K.R.LEIMBACH, 8 NOVEMBER 1972
C
DIMENSION IX(I),IY(I)
CALL LINEV(IX(IA),IY(IA),IX(IB),IY(IB))
C
RETURN
END

4-30
SUBROUTINE PLOTLB(I,NX,NY,IX,IY,Z)
C*** PLOT LABEL
C*** K. K. LEIMBACH, 8 NOVEMBER 1972
DIMENSION IX(1),IY(1)
IXP=IX(I)+NX
IYP=IY(I)+NY
CALL LABELV(Z,IXP,IYP,7,1,3)
RETURN
END

PHOG PO05 (11) (PLOTGC)
PRT,C PO05
FURPUN 24H1-03/10-14:53
SUBROUTINE PLOTWC(IA, IB, IX, IY)

**C**** PLOT A QUARTER CIRCLE FROM IA TO IB

C**** K.R. LEIMBACH, 8 NOVEMBER 1972

C

DIMENSION IX(1), IY(1)

N = 10

AN = N

AP = 3.1415926/(2.*AN)

IR = IX(IA) - IX(IB)

J1 = IY(IA)

NP1 = N + 1

DO 10 J = 1, NP1

AJ = J

APJ = (AJ - 1.) * AP

CPJ = COS(APJ)

SPJ = SIN(APJ)

R = IR

DX = R * SPJ

DY = R * (1. - CPJ)

10 CONTINUE

CONTINUE

RETURN

END

SUBROUTINE PLOTAR

CALL LINEV(11, J1, 12, J2)

11 = 12

J1 = J2

10 CONTINUE

CALL LINEV(11, J1, 12, J2)

11 = 12

J1 = J2

10 CONTINUE

RETURN

END

SUBROUTINE PLOTAR

CALL LINEV(11, J1, 12, J2)

11 = 12

J1 = J2

10 CONTINUE

RETURN

END

SUBROUTINE PLOTAR

CALL LINEV(11, J1, 12, J2)

11 = 12

J1 = J2

10 CONTINUE

RETURN

END
SUBROUTINE PLOTAR(IORNT, IX, IY)

C Plot an Arrow Head for Different Orientations
C K.R. Leimbach, 7 November 1972

C

C IORNT Pointing
C 1 Right
C 2 Left
C 3 Up
C 4 Down
C 5 Down-Left 45 Degrees

DIMENSION IX(1), IY(1)

10 IF (IORNT .NE. 1) GO TO 10
11 I2 = I1 + 10
12 J2 = J1 + 5
13 J3 = J1 - 5
14 GO TO 100
15
20 IF (IORNT .NE. 2) GO TO 20
21 I2 = I1 + 10
22 J2 = J1 + 5
23 J3 = J1 - 5
24 GO TO 100
25
30 IF (IORNT .NE. 3) GO TO 30
31 I2 = I1 + 10
32 J2 = J1 - 10
33 J3 = J2
34 GO TO 100
35
40 IF (IORNT .NE. 4) GO TO 40
41 I2 = I1 + 10
42 J2 = J1 + 3
43 J3 = J1 + 11
44 GO TO 100
45
50 RETURN
51 CALL LINEV(I1, J1, I2, J2)
52 CALL LINEV(I1, J1, I3, J3)
53 CALL LINEV(I2, J2, I3, J3)
54
C RETURN
C END
PO06  (14) (PLOTAR)

HDG PO07  (15) (PLOTTX)

PRTC PO07
FURPUR 2441-03/10-14:53
SUBROUTINE PLOTTX(I,NX,NY,IX,IY,AK,NTEXT)

C***** PLUT TEXT
C***** K.R. LEIMBACH, 8 NOVEMBER 1972
C
DIMENSION IX(1),IY(1),AK(1)

IPLT=IA(I)+NX
JPLT=IY(I)+NY

CALL PRINTV(NTXT,AK,IPLT,JPLT)

RETURN
END
SUBROUTINE DASHLN(IA, IB, IX, IY)

K*K*L = 11/30/72

DIMENSION IX(1), IY(1)

II = IX(IA)
J1 = IY(IA)
J2 = J1 + 25

CALL LINEV(II, J1, II, J2)
J2 = J1 + 25
J1 = J2 + 25
IF(J2 .LT. IY(IB)) GO TO 10
IF(J1 .LT. IY(IB)) GO TO 15
GO TO 20
J2 = IY(IB)
GO TO 10

CONTINUE
J1 = IY(IA) + 35
J2 = J1 + 5
CALL LINEV(II, J1, II, J2)
J1 = J2 + 45
J2 = J1 + 5
IF(J2 .LT. IY(IB)) GO TO 30
IF(J1 .LT. IY(IB)) GO TO 35
GO TO 40
J2 = IY(IB)
GO TO 30
CONTINUE
GO TO 35
RETURN
END

PRINT D100 (17) (OUTUES)

FURPUR 24H1-03/10-14:53
SUBROUTINE OUTDES(HEAD,AUB,WEIGHT,KOPT,T)

C*** OUTPUT OF DESIGN PARAMETERS
C*** K R L 1/8/73

DIMENSION HEAD(12),KOPT(10)

I01 FORMAT(12A6)
I02 FORMAT(* AUB=1,F10.4,* SQ=IN*/
  * WEIGHT=1,F10.4,* LB=1/
  * B=1,F10.4,* IN=1/
  * H=1,F10.4,* IN=1/
  * R=1,F10.4,* IN=1/
  * RG=1,F10.4,* IN=1/
  * RB=1,F10.4,* IN=1/
  * RFIL=1,F10.4,* IN=1/
  * KSPOT=1,F10.4,* IN=1/
  * DUSPUT=1,F10.4,* IN=1/
  * NB=1,11D/
  * BG=1,F10.4,* IN=1/
  * HT=1,F10.4,* IN=1/)

I03 FORMAT(8F10.4)

C
D1=2.*RI
DG1=2.*RG-BG
DG0=2.*RG+BG
DB=2.*KB

C
WRITE(6,101) (HEAD(I),I=1,12)
IF (KOPT(3),GT,0) GO TO 50
WRITE(6,102) AUB,WEIGHT,B,H,RI,UB,RG,DGI,DGU,RB,DB
  * RFIL,RSPOT,DHOLE,DSPUT,NB,BG,HT
C
RETURN
50 WRITE(6,103) AGB,WEIGHT,T,B,H,KI,DI,RB,IRFIL,KSPOT,DHOLE,DSPUT
  * NB,BG,HT
RETURN
END

FORMAT A01 (18) (ANALYS)

FORMAT C A001

FURPUN 24H1-63/10-14:53
SUBROUTINE ANALYS(P,DI,T,DELT,PF,BF,FS,GF
  ,ET,ANUT,RHOT,ALFAT,FTYT,FTUT
  ,EF,ANUF,RHOF,ALFAF,FTYF,FTUF
  ,EB,ANUB,RMOB,ALFAB,FTYB,FTUB
  ,EG,ZKG,SG,ALFAG,AMUG,AMUS,GMS,HG,HS,SB
  ,KUPT,AOB,NPHASE,DELTAT,PE
  ,B,H,R,RG,RB,RFIL,RSPOT,DHOLE,DSPUT,NB,BG,HT
  ,A,SRES,STR,AP,HEAD)

C
C    STRESS AND DEFORMATION ANALYSIS
C    K*R*LEIMBACH, 5 JANUARY 1973

C    DIMENSION A(9,4),SR(S(5,4),ST(5,4),AP(8),HEAD(12),KUPT(10)
PI = 3.14159
RU = HI + T/2.
FX=PF*RU/2.
FR=PF*HI/2.
RS=RU*RI/2.
FP=PF*(RS*2-R1**2)/(2*RI)
RP=(RS**2+RI*RS+R1**2)/(15*RS*RI)
AG=2*PI*H*GB
IF(KUPT(3)) GO TO 50
EKG=AG*EF/(2*PI*RO*HG)
GO TO 51
SG EKG=AG*EF/(2*PI*RO*(HG+HS))
CONTINUE
ELB=H
ENB=NB
A6=ENB+A6B
EKB=AB*EB/(2*PI*CO*ELB)
E=NB+RG
RA=RG+EKB*E/(EKB+EK)
CE=EKB+EKG*E2/(EK+EK)
AF=EA-H
AIF=AF*H**2/12.
C=H/2.
RC=RI+6/2.
BEN=ET*T**3/(12*(1.-ANUT**2))
AK=12*(1.-ANUT**2/(RU**2*T**2)
AK2=SQRT(AK4)
AK=SQRT(AK2)
BETA=BEND*AK*HU+RC/(EF*AIF)
DX=(1.*BETA)**2/2+2*AXK+2*AXIF/AF)/(2+BETA**2/2+2*AXK+2*AXIF/AF)*2
CF=BEND*DX/(BETA*1+AK+AXIF/AIF+5*AK)/AK2)
CWF=1.5/(AK2*AK2+BEND)
BETADX=BETA/DX
CMF= BETAUX *(1.5/AK2+BETA+1+5*AK)/AK)
CWF= BETAUX *(1.5/AK)

C    101 J=1
BFC=PB
EAM=EF*EF/(2*PI*RO)
ADFL=U.
CHUB=EMH/CF
FRUT=CHUB
4-38
A061

(ANALYS)

112 201 WTOP=RDFL
113 WBOT = RDFL - H * FROT
114 SFTOP = EF * WTOP / RG
115 SFBOT = EF * WBOT / RG
116 IF (KOPT(2), EQ. 1) GO TO 202
117 SX1 = -6 * EMX / T * 2 + ENX / T
118 SXO = -6 * EMX / T * 2 + ENX / T
119 SY1 = -6 * EMY / T * 2 + ENY / T
120 SYO = -6 * EMY / T * 2 + ENY / T
121 TXZ = 1.5 * EWX / T
122 GO TO 203
123 202 TX = T + 0.5 * RFIL
124 SX1 = -6 * EMX / TX * 2 + ENX / TX
125 SXO = -6 * EMX / TX * 2 + ENX / TX
126 SY1 = -6 * EMY / TX * 2 + ENY / TX
127 SYO = -6 * EMY / TX * 2 + ENY / TX
128 TXZ = 1.5 * EQX / TX
129 203 CONTINUE
130 C
131 A(1, J) = BFC
132 A(2, J) = EAM
133 A(3, J) = ADFL
134 A(4, J) = RDFL
135 A(5, J) = FROT
136 A(6, J) = BSTRS
137 A(7, J) = GSTRS
138 A(8, J) = SFTOP
139 A(9, J) = SFBOT
140 C
141 SRES(1, J) = ENX
142 SRES(2, J) = ENY
143 SRES(3, J) = EMX
144 SRES(4, J) = EMY
145 SRES(5, J) = EQX
146 C
147 STLH(1, J) = SX1
148 STLH(2, J) = SY1
149 STLH(3, J) = SXO
150 STLH(4, J) = SYO
151 STLH(5, J) = TXZ
152 C
153 AP(1) = BFC
154 AP(2) = LAM
155 AP(3) = ADFL
156 AP(4) = WP
157 AP(5) = FROT
158 AP(6) = LNA
159 AP(7) = EMX
160 AP(8) = EQX
161 IPHASE = J
162 CALL PLOTF2(I, J, B, H, M, B, RFIL, T, HT, ALFAT, DELTAT
163 IF(I, IPHASE) GO TO 300
164 IF(J, EQ. 2) GO TO 102

4-40
166 IF(J.EQ.3) GO TO 103
169 IF(J.EQ.4) GO TO 104
170 C
171 C END OF STRESS AND DEFORMATION ANALYSIS
172 C
173 C ULTIMATE MOMENT CAPACITY
174 300 TN = 0.5*T
175 ALFA1 = (T**2 -TN**2)/4.
176 ALFA1 = ALFA1 / (BEND*(0.25/AK2 + BETA*ALFA/AF + 0.5*C/AK))
178 ALFA1 = ALFA1 + ANU1
179 ALFA2 = (0.25*(T+TN)*(C+0.5/AK))/0.5/AK2 + BETA * ALFA/AF + 0.5
180 ALFA2 = ALFA2 * (BETA*AIF/(AK*AF) * O025/(AK2*AIF)*F*ROFL/IW)
181 ALFA2 = ALFA2 + ANU2
182 ALFA2 = ALFA2 * (BETA*AIF/(AK*AF) * O025/(AK2*AIF)*F*ROFL/IW)
183 ZETA2 = ALFA2 * ZETA1
184 SXX = B*H**2/4.o
185 EMFU = FTYT*(SXX/RO + 0.25*ZETA1*(T**2 - TN**2) + ZETA2*(T-TN)*C
186 1000 FORMAT(*, MFU= *,E16.8, * IN-LB/IN / * ZETA1= *,E16.8/
187 1 * ZETA2= *,E16.8/)
188 WRITE(6,1000) EMFU, ZETA1, ZETA2
189 RETURN
190 END

196 PO10  (19) (PLOTF2)

3PKT.C PO10
"URPUR 24HI-G3/10-14:53"
SUBROUTINE PLOTF2(DI,B,H,RB,RFIL,T,HT,ALFAT,DELAT,HEAD,KOPT,P,PH)

C***** PLOT ROUTINE FOR SUMMARY OF STRESS AND DEFORMATION ANALYSIS
C***** FOR LOW PROFILE FLANGES
C***** K.R.LEIMBACH, 19 DECEMBER 1972

DIMENSION X(100),Y(100),IX(100),IY(100)

DIMENSION AP(10)

CALL FRAMEV(O)
CALL SCRLCT(31,31,991,991)
CALL PRINTV(72,HEAD,41,1003)
A1 = 3.5
A2 = 4.0
A3=3.75
CALL XSCALV(-A3,A3,0,0)
CALL YSCALV(-A1,A2,0,0)
SCALE=2.0
BS=B/SCALE
HS=H/SCALE
RFILS=RFIL/SCALE
TS=T/SCALE
HTS = 2.0
R=U1/2.

X(1)=BS/2.
Y(1)=-HS/2.
X(2)=X(1)
Y(2)=-Y(1)
IF(KOPT(2).EQ.2) GO TO 210
X(3)=X(1)+TS+RFILS
Y(3)=Y(2)
X(4)=X(3)-RFILS
Y(4)=Y(2)
GO TO 215

210 CONTINUE
E1=RI+R=B
EIS=E1/SCALE
X(3)=X(2)-2.*EIS
Y(3)=Y(2)
X(4)=X(3)-2.*RFILS
Y(4)=Y(2)

215 CONTINUE
X(5)=X(4)
Y(5)=Y(2)+HTS
X(6)=X(5)-TS
Y(6)=Y(5)
X(7)=X(6)
Y(7)=Y(1)
X(8)=X(7)

Y(8)=Y(7)+1
P010 (19) (PLOT2)

50  X(9) = X(7)
51  Y(9) = Y(7) + 0.6
52  X(10) = X(11)
53  Y(10) = Y(8)
54  X(11) = X(11)
55  Y(11) = Y(9)
56  X(12) = X(7)
57  Y(12) = Y(7) + 0.5
58  X(13) = X(1)
59  Y(13) = Y(12)
60  X(14) = X(1) + 1
61  Y(14) = Y(1)
62  X(15) = X(1) + 0.6
63  Y(15) = Y(1)
64  X(16) = X(14)
65  Y(16) = Y(2)
66  X(17) = X(15)
67  Y(17) = Y(2)
68  X(18) = X(1) + 0.5
69  Y(18) = Y(1)
70  X(19) = X(18)
71  Y(19) = Y(2)
72  X(20) = X(5)
73  Y(20) = Y(5) + 0.1
74  X(21) = X(5)
75  Y(21) = Y(5) + 0.6
76  X(22) = X(6)
77  Y(22) = Y(20)
78  X(23) = X(6)
79  Y(23) = Y(21)
80  X(24) = X(5)
81  Y(24) = Y(5) + 0.6
82  X(25) = X(5) + 0.2
83  Y(25) = Y(24)
84  X(26) = X(6)
85  Y(26) = Y(24)
86  X(27) = X(6) + 0.2
87  Y(27) = Y(24)
88  X(28) = X(7) - 1.5
89  Y(28) = Y(12)
90  X(29) = X(28) + 0.5
91  Y(29) = Y(12)
92  X(30) = X(29) + 0.5
93  Y(30) = Y(12)
94  X(31) = X(29) + 0.25
95  Y(31) = Y(12) + 1.0
96  X(32) = X(31)
97  Y(32) = Y(12) - 0.5
98  X(33) = X(21) + 1
99  Y(33) = Y(5) + 1
100  X(34) = X(33) + 2
101  Y(34) = Y(33)
102  X(35) = X(33) + 1.0
103  Y(35) = Y(33)
104  X(36) = X(35)
105  Y(36) = Y(2)
112 X(37) = X(6) - 0.1
113 Y(37) = Y(5) + 0.1
114 X(38) = X(37) - 2.
115 Y(38) = Y(37)
116 X(39) = X(37) - 1.
117 Y(39) = Y(37)
118 X(40) = X(39)
119 Y(40) = Y(2)
120 X(41) = X(38)
121 Y(41) = Y(1) - 1.
122 X(42) = X(41) + 2.6
123 Y(42) = Y(41)
124 X(43) = X(41) + 1.0
125 Y(43) = Y(1)
126 X(44) = X(43)
127 Y(44) = Y(2)
128 C NP = 44
130 C DO 10 I = 1, NP
132 CALL XSCLVI(X(I), IX(I), IERR)
133 CALL YSCLVI(Y(I), IY(I), IERR)
134 10 CONTINUE
136 C CALL PLOTLN(1, 2, IX, IY)
137 CALL PLOTLN(2, 3, IX, IY)
138 IF (KOPT(2) .EQ. 1) CALL PLOTHC(3, 4, IX, IY)
139 IF (KOPT(2) .EQ. 2) CALL PLOTGC(3, 4, IX, IY)
140 CALL PLOTLN(4, 5, IX, IY)
141 CALL PLOTLN(5, 6, IX, IY)
142 CALL PLOTLN(6, 7, IX, IY)
143 CALL PLOTLN(7, 8, IX, IY)
144 CALL PLOTLN(8, 9, IX, IY)
145 CALL PLOTLN(10, 11, IX, IY)
146 CALL PLOTLN(12, 13, IX, IY)
147 CALL PLOTLN(14, 15, IX, IY)
148 CALL PLOTLN(16, 17, IX, IY)
149 CALL PLOTLN(18, 19, IX, IY)
150 CALL PLOTLN(20, 21, IX, IY)
151 CALL PLOTLN(22, 23, IX, IY)
152 CALL PLOTLN(24, 25, IX, IY)
153 CALL PLOTLN(26, 27, IX, IY)
154 CALL PLOTLN(28, 29, IX, IY)
155 CALL PLOTLN(30, 31, IX, IY)
156 CALL PLOTLN(32, 33, IX, IY)
157 CALL PLOTLN(34, 35, IX, IY)
158 CALL PLOTLN(36, 37, IX, IY)
159 CALL PLOTLN(38, 39, IX, IY)
160 CALL PLOTLN(40, 41, IX, IY)
161 CALL PLOTLN(42, 43, IX, IY)
162 CALL DSHLNV(IX(28), IY(28), IX(29), IY(29), 8, 8)
163 CALL PLOTLN(29, 30, IX, IY)
164 CALL DASHLN(31, 32, IX, IY)
165 CALL PLOTLN(33, 34, IX, IY)
166 CALL PLOTLN(35, 36, IX, IY)
167 CALL PLOTLN(37, 38, IX, IY)
168 CALL PLOTLN(39,40,IX,1Y)
169 CALL PLOTLN(41,42,IX,1Y)
170 CALL PLOTLN(43,44,IX,1Y)
171
172 ISCALE=1
173 X1=X(34)
174 Y1=Y(34)
175 Y2=Y1+05
176 IDY=5
177 CALL YSCLV1(Y1,1Y1,IERR)
178 CALL YSCLV1(Y2,1Y2,IERR)
179 STRESS=90*
180 IYS=Y1+IDY
181 DO 21 I=1,9
182 X2=X1
183 CALL XSCLV1(X1,IX1,IERR)
184 CALL XSCLV1(X2,IX2,IERR)
185 CALL LINEV(IX1,1Y1,IX2,1Y2)
186 IX=IX1-8
187 CALL LABELV(STRESS,1AS,1YS,3,12)
188 XI=X1+25
189 STRESS=STRESS-10*
190 21 CONTINUE
191 ISCALE=ISCALE+1
192 IF(ISCALE.EQ.2) GO TO 22
193 IF(ISCALE.EQ.3) GO TO 23
194 IF(ISCALE.EQ.4) GO TO 24
195 22 XI=X(37)
196 Y1=Y(37)
197 Y2=Y1+05
198 IDY=5
199 GO TO 20
200 23 XI=X(42)
201 Y1=Y(42)
202 Y2=Y1+05
203 IDY=21
204 GO TO 20
205 24 CONTINUE
206 C
207 DIMENSION D1(3),D2(3),D3(3),D4(3),D5(3),D6(3)
208 * D7(3),D8(3),D9(3),D10(3),D11(3)
209 * D12(4),D13(4),D14(4),D15(4)
210 DIMENSION D1O1(3),D1O2(3),D1O3(3),D1O4(3)
211 C
212 DATA(D1(I),I=1,3)/18M PHASE
213 DATA(D2(I),I=1,3)/18M BOLT FORCE
214 DATA(D3(I),I=1,3)/18M (LB)
215 DATA(D4(I),I=1,3)/18M APPLIED MOMENT
216 DATA(D5(I),I=1,3)/18M (IN-LB)
217 DATA(D6(I),I=1,3)/18M AXIAL DISPLACEMET
218 DATA(D7(I),I=1,3)/18M (IN)
219 DATA(D8(I),I=1,3)/18M ROTATION
220 DATA(D9(I),I=1,3)/18M (RAD)
221 DATA(D10(I),I=1,3)/18M
222 DATA(D11(I),I=1,3)/18M
223 DATA(D12(I),I=1,4)/24M STRESSES ON INSIDE

4-45
DATA(D13(I),I=1,4)/24H STRESSES ON OUTSIDE /
DATA(D14(I),I=1,4)/24H BENDING STRESSES IN FLG/
DATA(D15(I),I=1,4)/24H SUMMARY OF ANALYSIS /

CALL PRINTV(18,E1,191,151)
CALL PRINTV(18,E2,350,151)
CALL PRINTV(18,E3,510,151)
CALL PRINTV(18,E4,510,151)
CALL PRINTV(18,E5,510,151)
CALL PRINTV(18,E6,510,151)
CALL PRINTV(18,E7,510,151)
CALL PRINTV(18,E8,510,151)

CALL PRINTV(18,01,151)
CALL PRINTV(18,02,301,151)
CALL PRINTV(18,03,501,311)
CALL PRINTV(18,04,510,151)
CALL PRINTV(18,05,510,151)
CALL PRINTV(18,06,510,151)
CALL PRINTV(18,07,510,151)
CALL PRINTV(18,08,510,151)

CALL LINEV(190,111,991,111)
CALL LINEV(350,161,350,31)
CALL LINEV(510,161,510,31)
CALL LINEV(670,161,670,31)
CALL LINEV(830,161,830,31)

CALL PRINTV(24,D1,11XP1IYP)
CALL PRINTV(24,D2,11XP1IYP)
CALL PRINTV(24,D3,11XP1IYP)
CALL PRINTV(24,D4,11XP1IYP)

CALL LINEV(400,951,400,997)
CALL LINEV(400,997,400,951)

DIMENSION DATA1(1)
DATA(DATA1(I),I=1,11)/68H DI/

CALL PLOTLB(28,05,1XIY,D1)
CALL PLOTTX(28,32,5,1XIY;DATA1,6)
CALL PLOTLB(18,10,25,1XIY,H)
CALL PLOTLB(9,10,20,1XIY,B)
CALL PLOTLB(21,16,1;1XIY,T)

BFC=AP(1)
EAM=AP(2)
ADFL=AP(3)
WPU=AP(4)
CHIU=AP(5)
ENQ=AP(6)
EMQ=AP(7)
EQO=AP(8)

SSCALE=00G025
#SCALE=10.
R=1+1/2.
AL=HTS*2.0
DL=U*US
NL=AL/QL
I=IPHASE
280    AK2=AK*AK
281    BEND=ET*T*3/(12*(1.-ANUT**2))
282    CMF=.5/(AK2*AK*BEND)
283
284    DO 100 K=1,NL
285    EK=K
286    AK=(EK-1.)*DL
287    SX=SIN(AK*XX)
288    CX=COS(AK*XX)
289    EX=EXP(-AK*AK)
290    WX=CMF*EX*(EQ0*CX*AK*EMG*(CX-SX))+WP0
291    EMX=EX*(EMG*(CX+SX)-EQ0*SX/AK)
292    EQX=EX*(EQ0*(SX-CX)+2*AK*EMG*SX)
293    IF(K.EQ.1) NTOP=WX
294    ENX=EWU
295    ENY=ET*T*NO/R0
296    IF(IPHASE.EQ.2) ENY=ENY-ET*T*ALFAT*DELTAT*P*RU*ANUT/2*
297    IF(IPHASE.EQ.3) ENY=ENY+P*RU*ANUT/2*
298    IF(IPHASE.EQ.4) ENY=ENY+PF*P*RU*ANUT/2*
299    EMY=ANUT*EMX
300
301    IF(KOPT(2).EQ.1) GO TO 202
302    SXI=-6.*EMX/T**2+ENX/T
303    SX0=6.*EMX/T**2+ENX/T
304    SYI=-6.*EMY/T**2+ETY/T
305    SYO=6.*EMY/T**2+ETY/T
306    TMAX=1.5*EQX/T
307    GO TO 205
308
309    202 IF(AK+LRFIL) GO TO 203
310    TX=T
311    GO TO 204
312
313    203 XP=RFIL-AK
314    YP=SQRT(RFIL**2-XP**2)
315    TX=T+RFIL-YP
316    CONTINUE
317
318    204 CONTINUE
319    SAI=-6.*EMX/TX**2+ENX/TX
320    SX0=6.*EMX/TX**2+ENX/TX
321    SYI=-6.*EMY/TX**2+ETY/TX
322    SYO=6.*EMY/TX**2+ETY/TX
323    TMAX/X(16)=SSCA:*TMAX
324    AXH=X(16)+WSCA*W
325
326    CALL YSCLVI(YS,1YS,IERR)
327    CALL XSCLVI(XSXI,IXSI,IERR)
328    CALL XSCLVI(XSI,IXSI,IERR)
329    CALL XSCLVI(XSXI,IXSI,IERR)
330    CALL XSCLVI(XSXI,IXSI,IERR)
331    CALL XSCLVI(XSXI,IXSI,IERR)
332    CALL XSCLVI(XSXI,IXSI,IERR)
333    CALL XSCLVI(XSXI,IXSI,IERR)
334    CALL XSCLVI(XSXI,IXSI,IERR)
335    CALL XSCLVI(XSXI,IXSI,IERR)

4-47
CALL XSCLVI(X*XIXW,*IY,*IY,1,EYER)

C

IF(K.EQ.1) GO TO 98
IF(K.EQ.6) GO TO 98
IF(K.EQ.11) GO TO 98
GO TO 99

CALL PLOTV(IXSX,1,1YS,29,0)
CALL PLOTV(IXSYI1,1YS,30,0)
CALL PLOTV(IXSXO1,1YS,29,0)
CALL PLOTV(IXSYO1,1YS,30,0)
CALL PLOTV(IXTMAO,IYS,25,0)
CALL PLOTV(IAXW,X,1Y,*28,0)

CALL PLOTV(IXSXI1,1YS,35,0)
CALL PLOTV(IXSYI1,1YS,35,0)
CALL PLOTV(IXSXO1,1YS,35,0)
CALL PLOTV(IXSBO1,1YS,35,0)
CALL PLOTV(IXTMAXO1,1YS,35,0)
CALL PLOTV(IXWSXI1,1YS,35,0)

CONTINUE

WBOU=H*TOP+H*CHU
XWBOT=X(7)+WBOU
XWTOP=X(7)+TOP
SIGTOP=EF*WTOP/RO
SIGBOT=EF*WBOU/RO
YSTOP=Y(2)
YSBOT=Y(1)
XSTOP=X(43)+SSCAL*SIGTOP
XSBOU=X(43)+SCAL*SIGBOT

CALL XSCLVI(XSTOP,1XSTOP,1YSTOP,1EYER)
CALL YSCLVI(YSTOP,1YSTOP,1YSTOP,1EYER)
CALL XSCLVI(XSBOT,1XSBOT,1YSBOT,1EYER)
CALL YSCLVI(YSBOT,1YSBOT,1YSBOT,1EYER)
CALL XSCLVI(XWBOT,1XWBOU,1XWBOU,1EYER)
CALL XSCLVI(XWTOPI,1XWTOP,1XWTOP,1EYER)

CALL PLOTV(IXSTOP,1YSTOP,1YSTOP,1YSTOP,30,0)
CALL PLOTV(IXSBOT,1YSBOT,1YSBOT,1YSBOT,30,0)
CALL LINEV(IX(44),1Y(44),IXSTOP,1YSTOP)
CALL LINEV(IX(43),1Y(1),IXSTOP,1YSTOP)
CALL PLOTV(IXWBOU,1Y(1),28,0)
CALL DSHLNVI1,AWBOU,1Y(1),1XWBOU,1Y(7),5,15

DATA(D1011),1=1,3/18H BOLT-UP /
DATA(D1021),1=1,3/18H START-UP /
DATA(D1031),1=1,3/18H OPERATION /
DATA(D1041),1=1,3/18H SHUT-DOWN /
IF(IPHASE.EQ.1) CALL PRINTV(18,D1011,1Y(7),190,76)
IF(IPHASE.EQ.2) CALL PRINTV(18,D1021,1Y(7),190,76)
IF(IPHASE.EQ.3) CALL PRINTV(18,D1031,1Y(7),190,76)
IF(IPHASE.EQ.4) CALL PRINTV(18,D1041,1Y(7),190,76)

CALL LAIVL(BFC,380,76,-6,1,1)
Calls LAIVL(TEAM,430,76,-6,1,1)
CALL LAIVL(ADFL,760,76,-6,1,1)
PO10 (19) (PLOTF2)

392 CALL LABLV(CH10.860:76,-6.1:1)
393 C
394 RETURN
395 END

SHDG D20D (20) (OUTAN)

OPRT,C D20D
FURPUR 24H1-03/10-14:53
SUBROUTINE OUTAN(HEAD, A, SR, S)

** PRINT-OUT ANALYSIS RESULTS

K. R. LEIBMACH, 4 JANUARY 1973

DIMENSION A(9,4), SR(15,4), S(5,4), HEAD(12)

101 FORMAT(1HI)
102 FORMAT(12A6)
103 FORMAT(* OUTPUT OF THE STRESS AND DEFORMATION ANALYSIS RESULTS */
104 FORMAT(* VARIABLE 1,30A,** BOLT-UP *)
105 * START-UP *
106 * OPERATION *
107 * SHUT-DOWN **/)
201 FORMAT(16H BOLT FORCE (LB), 24X, 9E16.8/
202 FORMAT(16H EQUV APPL MOMENT (IN-LB/IN), 11X, 9E16.8/
203 FORMAT(16H AXIAL DEFLACTION (IN), 18X, 9E16.8/
204 FORMAT(16H RADIAL DEFLACTION (IN), 17X, 9E16.8/
205 FORMAT(16H FLANGE ROTATION (Radian), 15X, 9E16.8/
206 FORMAT(16H BOLT STRESS (PSI), 22X, 9E16.8/
207 FORMAT(16H GASKET STRESS (PSI), 20X, 9E16.8/
208 FORMAT(16H STRESS IN FLANGE TOP (PSI), 13X, 9E16.8/
209 FORMAT(16H STRESS IN FLANGE BOTTOM (PSI), 10X, 9E16.8/
210 FORMAT(16H STRESS RESULTS (LB/IN) NA=9E16.8/
211 FORMAT(16H EQUV APPL MOMENT (IN-LB/IN) MX=9E16.8/
212 FORMAT(16H AXIAL DEFLACTION (IN) MY=9E16.8/
213 FORMAT(16H RADIAL DEFLACTION (IN) MZ=9E16.8/
214 FORMAT(16H FLANGE ROTATION (Radian) M=9E16.8/
215 FORMAT(16H BOLT STRESS (PSI) WK=9E16.8/
216 FORMAT(16H GASKET STRESS (PSI) OK=9E16.8/
217 FORMAT(16H STRESS IN FLANGE TOP (PSI) IX=9E16.8/
218 FORMAT(16H STRESS IN FLANGE BOTTOM (PSI) IY=9E16.8/
219 FORMAT(16H STRESSES AT NECK (PSI) INNER SIGX=9E16.8/
220 FORMAT(16H EQUV APPL MOMENT (IN-LB/IN) NNX=9E16.8/
221 FORMAT(16H AXIAL DEFLACTION (IN) NMY=9E16.8/
222 FORMAT(16H RADIAL DEFLACTION (IN) NNZ=9E16.8/
223 FORMAT(16H FLANGE ROTATION (Radian) NM=9E16.8/
224 FORMAT(16H BOLT STRESS (PSI) WNK=9E16.8/
225 FORMAT(16H GASKET STRESS (PSI) OOK=9E16.8/
226 FORMAT(16H STRESS IN FLANGE TOP (PSI) IX=9E16.8/
227 FORMAT(16H STRESS IN FLANGE BOTTOM (PSI) IY=9E16.8/
228 FORMAT(16H STRESSES AT NECK (PSI) MAX TAU=9E16.8/

WRITE(6,101)
WRITE(6,102) (HEAD(I), I=1,12)
WRITE(6,103)
WRITE(6,104)
WRITE(6,201) ((A(I,J), J=1,4), I=1,9)
WRITE(6,202) ((SR(I,J), J=1,4), I=1,5)
WRITE(6,203) ((S(I,J), J=1,4), I=1,5)

RETURN
END
Section 5

NUMERICAL EXAMPLES

In this section one example is presented that has been computed by hand. Corresponding computer results of this and of additional examples are also given.

5.1 EXAMPLE: STEEL FLANGE WITH STEEL GASKET

Given:
- Nominal pressure \( p = 1500 \) psi
- Nominal diameter \( d_i = 8.00 \) inch
- Tube thickness \( t = .438 \) inch

Safety Factors:
- Proof 1.5
- Burst 2.0
- General 1.5
- Gasket 2.0

Material:
- Tube and Flange: 347 SS steel
- Bolt Material: A286 Steel
- Gasket Material: CRES 321-A

The material data are given in Tables 2-10 and 2-11. Following the outline in Appendix A, the following results are obtained:

(a) Tube thickness given as \( t = 0.4375 \) inch

The thickness based on Eq. (2.23) would be

\[
 t = \frac{1.5 \times 1.5 \times 10^3 \times 4.000}{35 \times 10^3} = \frac{9.0 \times 10^3}{35 \times 10^3} = 0.257 \text{ inch}
\]

For a more accurate thickness computation Eq. (2.18) should be used, giving
\[
\begin{align*}
  t &= \frac{1.5 \times 10^3 \times 4.000}{35 \times 10^3 / 1.5 - 1.5 \times 10^3 / 2.0} = \frac{6.0 \times 10^3}{23.3 \times 10^3 - 0.75 \times 10^3} \\
  &= \frac{6.0 \times 10^3}{22.55} = 0.266 \text{ inch}
\end{align*}
\]

For higher internal pressures this difference is more distinct.

(b) Initial guess of bolt size

\[d_B = 0.4375 \text{ inch, size number 4 (Table 2-3)}\]

Machined spot faces

\[e_1 = 1.03 \times 0.4375 = 0.4506 \text{ inch}\]
\[e_2 = 0.91 \times 0.4375 = 0.3981 \text{ inch}\]

Hole diameter

\[d_{\text{hole}} = 0.437 + 0.005 = 0.442 \text{ inch}\]

Spot face diameter

\[d_{\text{spot}} = 2 \times 0.4506 = 0.9012 \text{ inch}\]

Fillet radius for spot face

\[r_{\text{spot}} = 0.062 \text{ inch}\]

(c) Bolt circle radius

\[r_B = 4.00 + 0.4375 + 0.062 + 0.4506 = 4.9501\]

On the plot appears the diameter of the bolt circle

\[(\text{diam})_B = 2 \times 4.9501 = 9.900 \text{ inch}\]

(d) Flange width

\[b = 4.950 + 0.3981 - 4.000 = 1.348 \text{ inch}\]

On the plot appears the outer diameter of the flange as

\[(\text{diam})_{\Phi F} = 8.000 + 2.696 = 10.696 \text{ inch}\]
(e) Gasket width and gasket radius

Estimate for gasket radius

\[ r_G = \frac{1}{2} (4.950 - 0.221 + 4.000) \]
\[ = 1.2 (8.729) = 4.3645 \text{ inch} \]

Gasket width, calculated on the assumption that the gasket is initially stressed to the yield strength \( K_G \), but under proof pressure it is allowed to the pressure dependent seating stress \( \sigma_G = k_p p \) (see A.5(b))

\[ b_G = \frac{1.5 \times 1500 \times 4.3645}{2 \left[ 1.0 \times 40.0 \times 10^3 - 1.0 \times 1.5 \times 5.50 \times 1.5 \times 10^3 \times 2.0 \right]} \]
\[ = \frac{9.82 \times 10^3}{80.0 \times 10^3 - 49.5 \times 10^3} = \frac{9.82 \times 10^3}{30.5 \times 10^3} = 0.322 \text{ inch} \]

The gasket will be located close to the bolts, allowing a tolerance of \( c_2 = 0.05 \text{ inch} \)

\[ r_G = 4.950 - 0.221 - 0.161 - 0.050 = 4.518 \text{ inch} \]

The inner radius of the gasket is

\[ r_{G_i} = 4.518 - 0.161 = 4.357 \text{ inch} \]

and the corresponding diameter appearing on the plot

\[ (\text{diam})_{G_i} = 2 \times 4.357 = 8.714 \text{ inch} \]

The outer radius of the gasket is

\[ r_{G\phi} = 4.518 + 0.161 = 4.679 \text{ inch} \]

and the corresponding diameter

\[ (\text{diam})_{G\phi} = 2 \times 4.679 = 9.358 \text{ inch} \]
(f) Required bolt force

\[ P^{(1)}_B = 2\pi \times 4.518 \times 0.322 \times 1.0 \times 40.0 \times 10^3 \]
\[ = 6.28 \times 1.455 \times 40.0 \times 10^3 \]
\[ = 9.1374 \times 40.0 \times 10^3 = 365.5 \times 10^3 \text{ lb} \]

\[ P^{(2)}_B = \pi \times (4.518)^2 \times 1.5 \times 10^3 \times 1.5 \]
\[ + 2\pi \times 4.518 \times 0.322 \times 1.0 \times 1.5 \times 1.5 \times 10^3 \times 5.50 \times 2.0 \]
\[ = 3.14 \times 20.41 \times 2.25 \times 10^3 + 6.28 \times 1.455 \times 4.5 \times 10^3 \times 5.50 \]
\[ = 63.24 \times 2.25 \times 10^3 + 9.075 \times 24.75 \times 10^3 \]
\[ = (142.3 + 224.6) \times 10^3 = 366.9 \times 10^3 \text{ lb} \]

(g) Number of bolts

\[ n_{B1} = \frac{366.9 \times 10^3}{131 \times 10^3 \times 0.10631} \]
\[ = \frac{366.9 \times 10^3}{13.93 \times 10^3} = 26.3 \approx 26 \text{ bolts} \]

\[ n_{B2} = \frac{(2.0/1.5) \times 366.9 \times 10^3}{200.0 \times 0.10631} \approx \frac{488 \times 10^3}{21.3 \times 10^3} \]
\[ = 22.9 \approx 23 \text{ bolts} \]

\[ n_B = 26 \text{ bolts are required.} \]

(h) Bolt spacing

\[ s = \frac{2\pi \times 4.518}{26} = \frac{6.28 \times 4.518}{26} = \frac{28.37}{26} = 1.09 \text{ inch} \]
Minimum allowable spacing is

\[ s_{\text{min}} = 1.81 \times 0.4375 = 0.792 \text{ inch} < 1.09 \text{ inch} \]

Maximum allowable bolt spacing

\[ s_{\text{max}} = 8 \times 0.4375 = 3.500 \text{ inch} > 1.09 \text{ inch} \]

(i) Flange height

Internal lever arm

\[ e = 4.950 - 4.518 = 0.432 \text{ inch} \]

Radius of the shell middle surface

\[ r_o = 4.000 + 0.219 = 4.219 \text{ inch} \]

Thickness required to carry axial force

\[ t_N = \frac{0.438}{2} = 0.219 \text{ inch} \]

Ultimate moment to be carried

\[ m_{\text{Fu}} = \frac{1.5 \times 366.9 \times 10^3 \times 0.432}{2\pi \times 4.219} \]

\[ = \frac{550.4 \times 10^3 \times 0.432}{6.28 \times 4.219} \]

\[ = \frac{237.7 \times 10^3}{26.5} = 8.97 \times 10^3 \text{ in-lb/in.} \]

Effective flange width

\[ b = 1.348 - 0.442 \sqrt{0.442/1.09} \]

\[ = 1.348 - 0.442 \sqrt{0.406} \]

\[ = 1.348 - 0.442 \times 0.637 \]

\[ = 1.348 - 0.282 = 1.066 \text{ inch} \]
Assume
\[ \xi_1 = 0.8 \]
\[ \xi_2 = 0.18 \]

The coefficients of the quadratic equation for \( h \) are

\[ A = 35 \times 10^3 \times \frac{1.066}{6 \times 4.219} = \frac{37.31}{25.31} \times 10^3 = 1.474 \times 10^3 \]
\[ B = 35 \times 10^3 \times 0.18 \times \frac{0.438 - 0.219}{2} = 35 \times 0.0197 \times 10^3 = 0.6895 \times 10^3 \]
\[ C = 35 \times 10^3 \times 0.8 \times \left( \frac{(0.438)^2 - (0.219)^2}{4} - 8.97 \times 10^3 \right) = 35 \times 10^3 \times 0.2 \times (0.192 - 0.048) - 8.97 \times 10^3 = (1.008 - 8.97) \times 10^3 = -7.962 \times 10^3 \]
\[ R^2 = \left[ (0.6895)^2 + 4 \times 1.474 \times 7.962 \right] \times 10^6 = \left[ 0.4754 + 5.896 \times 7.962 \right] \times 10^6 = \left[ 0.4754 + 46.9440 \right] \times 10^6 = 47.42 \times 10^6 \]
\[ R = 6.886 \times 10^3 \]
\[ h = \frac{6.886 - 0.690}{2 \times 1.474} = \frac{6.196}{2.948} = 2.102 \text{ inch} \]

If the contribution of the plastic hinge is neglected, i.e., if \( \xi_1 = \xi_2 = 0 \) is assumed, then

\[ A = 1.474 \times 10^3 \]
\[ B = 0 \]
\[ C = -8.97 \times 10^3 \]
\[ R^2 = 4 \times 1.474 \times 8.97 \times 10^3 \]
\[ = 5.896 \times 8.97 \times 10^3 \]
\[ = 52.89 \times 10^3 \]

\[ R = 7.273 \]

\[ h = \frac{7.273}{2.948} = 2.467 \text{ inch} \]

This is 0.3 inch more than the previous result.

The same result would have been obtained by taking the old formula from Ref. 1,

\[ h = \sqrt{\frac{6 \times 4.219 \times 8.97 \times 10^3}{35.0 \times 10^3 \times 1.066}} \]
\[ = \sqrt{\frac{25.31 \times 8.97}{37.31}} = \sqrt{\frac{227.0}{37.31}} \]
\[ = \sqrt{6.084} = 2.467 \text{ inch}. \]

The weight savings accomplished by considering the plastic hinge is therefore approximately 10%.

(j) Flange weight

Weight area

\[ A_w = (1.348 - 0.438) \times 2.102 \]
\[ = 0.910 \times 2.102 = 1.913 \text{ in}^2 \]

Centroidal radius

\[ r_w = 4.000 + \frac{0.438 + 1.348}{2} \]
\[ = 4.000 + 0.893 = 4.893 \text{ inch} \]

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Volume
\[ \text{Vol} = 2\pi \times 4.893 \times 1.913 = 58.78 \text{ in}^3 \]

Weight
\[ \Delta w = 0.288 \times 58.78 = 16.9 \text{ lb} \]

On Fig. 5-1 the design geometry is summarized. Figures 5-2 and 5-3 show the stresses and radial displacement at initial torqueing and at proof pressure, respectively. The axial stresses \( \sigma_x \) are indicated by the curve labeled "X," the circumferential stresses \( \sigma_\phi \) by "Y" and the transverse shear stresses, \( \tau_{xz} \), by "T." The radial displacement is shown as "W." At the bottom of the plot the total bolt-force and the applied moment in (in-lb/in), as well as the axial displacement and the rotation of the flange are given. A sample printout is given for verification.

5.2 EXAMPLES FOR WEIGHT COMPARISON WITH CONVENTIONAL FLANGES

Before some weight comparisons with conventional flanges are made it is instructive to discuss a series of designs computed by the program. This series points out the need for judgement in the selection of the design parameters and materials.

Figures 5-4 through 5-6 show a flange which was designed to meet the algorithms for minimum tube wall thickness and minimum gasket width. Possibly the tube wall thickness is less than the minimum gage requirements for handling and accidental impact loads. The gasket width should be selected to fill out the available space between the inside of the tube and the inside of the bolts, including some dimensional tolerance. Possibly a thinner gasket should be designed. An algorithm is available in the program to automatically compute the gasket width to make use of the available space. Figure 5-4 shows a design with 6 bolts, which is the minimum. Figures 5-7 through 5-9 show a design similar to the previous one with double the pressure. In both this and the previous designs the stresses are well below the allowable ones. This is due to increased flange height based on bolt spacing allowing \( h \) not to be less than \( s/3 \). This requirement is based on experience since it is difficult to assess it analytically. The intent is to avoid waviness of the flange between the bolts.
Fig. 5-1 - Design Example
Fig. 5-2 - Stresses at Initial Torquing

Legend:  

- $X = \sigma_X$  axial stress (ksi)  
- $Y = \sigma_\varphi$  circumferential stress (ksi)  
- $T = \tau_{XZ}$  transverse shear stress (ksi)  
- $W =$ radial displacement (10-fold magnified)
Fig. 5-3 - Stresses at 1500 psi (proof pressure)
FLANGE PARAMETRIC CASE 1 DIA = 4 IN PRESS = 100 PSI

PRESSURE 100 PSIG
WEIGHT OF FLANGE 0.694 LB

4.0000IA
0.010

4.7250IA

5.2050IA

4.3010IA

4.1700IA

FLANGE MATERIAL: AL6061T6 AT FTY = 25.000 ksi, FTY = 42.000 ksi
BOLT MATERIAL: AL2024T3 AT FTY = 50.000 ksi, FTY = 62.000 ksi
GASKET MATERIAL: ASBESTO 1/16 SEATING STRESS = 3.700 ksi
YIELD STRENGTH = 10.000 ksi

Fig. 5-4 - Flange 1, Design

5-12
Fig. 5-5 - Flange 1, Stresses at Initial Torquing
Fig. 5-6 - Flange 1, Stresses at Proof Pressure
FLANGE PARAMETRIC CASE 2  DIA= 4IN  PRESS = 200 PSI

PRESSURE 200 PSIG
WEIGHT OF FLANGE 0.700 LB

4.000DIA
0.021

4.747DIA

5.227DIA

4.239DIA

4.192DIA

0.125R
0.255DIA, 6 HOLES
0.580DIA SPOTFACE
0.062R FILLET

1.578
0.828
0.062

FLANGE MATERIAL  AL6061T6 RT FTY= 35,000KSI, FTYI= 42,000KSI
BOLT MATERIAL  AL2024T3 RT FTY= 50,000KSI, FTYI= 62,000KSI
GASKET MATERIAL  ASBEST/16 SEATING STRESS= 3,700KSI
YIELD STRENGTH= 10,000KSI

Fig. 5-7 - Flange 2, Design
FLANGE PARAMETRIC CASE 2  DIA= 4IN  PRESS = 200 PSI

SUMMARY OF ANALYSIS

<table>
<thead>
<tr>
<th>PHASE</th>
<th>BOLT FORCE (LB)</th>
<th>APPLIED MOMENT (IN-LB)</th>
<th>AXIAL DISPLACEMENT (IN)</th>
<th>ROTATION (RAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>BOLT-UP</td>
<td>$9.34600 \times 10^0$</td>
<td>$1.62969 \times 10^2$</td>
<td>$0.00000 \times 10^0$</td>
<td>$2.42379 \times 10^{-3}$</td>
</tr>
</tbody>
</table>

Fig. 5-8 - Flange 2. Stresses at Initial Torquing
**FLANGE PARAMETRIC CASE 2**
**DIA = 4IN**
**PRESS = 200 PSI**

**SUMMARY OF ANALYSIS**

<table>
<thead>
<tr>
<th>Phase</th>
<th>Bolt Force (lb)</th>
<th>Applied Moment (in-lb)</th>
<th>Axial Displacement (in)</th>
<th>Rotation (rad)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operation</td>
<td>$1.15301 \times 10^{-04}$</td>
<td>$2.62247 \times 10^{-02}$</td>
<td>$9.46790 \times 10^{-04}$</td>
<td>$3.90032 \times 10^{-03}$</td>
</tr>
</tbody>
</table>

**Fig. 5-9** - Flange 2, Stresses at Proof Pressure
Figures 5-10 through 5-12 show a design in which the required gasket width is controlling the width of the flange. A decrease in the minimum seating stress at proof pressure would reduce the required gasket width. A different gasket should be used in this case. Since the flange height was selected based on the strength requirements the peak stresses as shown on Figs. 5-11 and 5-12 are close to the allowable ones.

Figures 5-13 through 5-15 again show a flange design controlled by the bolt-spacing-to-height ratio of 1/3. Consequently the stresses are low. Figures 5-16 through 5-18 show a flange with more balanced proportions. It has the same inner diameter as the previous one but the pressure is doubled. The third flange with this diameter is again controlled by the gasket width requirements (see Figs. 5-19 through 5-21). For this flange a different gasket should be selected.

The flange shown on Figs. 5-22 through 5-24 is well proportioned and the stresses are well under the allowable stresses, although here as before bolt spacing is the controlling factor. Figs. 5-25 through 5-26 show a flange designed for the same inner diameter but twice the pressure. This is a strength-controlled design.

Finally, Figs. 5-28 through 5-30 show three typical low profile designs. The last two are again partially controlled by the width of the gasket although only slightly.

Table 5-1 presents a comparison of flanges designed with conventional and low profile contours. The weight savings are impressive even using the unfavorable configuration with the gasket located toward the inside of the tube. Figures 5-31 through 5-45 present plots of the low profile flanges with the conventional contour indicated by a dashed line and shading. The saving in space requirements is obvious when the outer diameters of these flanges are compared.
FLANGE PARAMETRIC CASE 3

DIA = 4 IN
PRESS = 500 PSI

PRESSURE 500 PSIG
WEIGHT OF FLANGE 0.603 LB

4.000DIA

0.052

0.187R

0.295DIA, 12 HOLES
0.580DIA SPOTFACE
0.062R FILLET

5.001DIA

5.481DIA

4.100DIA

4.446DIA

FLANGE MATERIAL AL6061T6 RT FTY = 35.000ksi, FTU = 42.000ksi
BOLT MATERIAL AL2024T3 RT FTY = 50.000ksi, FTU = 62.000ksi
GASKET MATERIAL ASBESTO/16 SEATING STRESS = 3.700ksi
VYELD STRENGTH = 10.000ksi

Fig. 5-10 - Flange 3, Design

5-19
FLANGE PARAMETRIC CASE 3 DIA = 4 IN PRESS = 500 PSI

**SUMMARY OF ANALYSIS**

<table>
<thead>
<tr>
<th>PHASE</th>
<th>BOLT FORCE (LB)</th>
<th>APPLIED MOMENT (IN-LB)</th>
<th>AXIAL DISPLACEMENT (IN)</th>
<th>ROTATION (RAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>BOLT-UP</td>
<td>$1.90075 \times 10^{4}$</td>
<td>$4.68911 \times 10^{4}$</td>
<td>$0.00000 \times 10^{-1}$</td>
<td>$1.16499 \times 10^{-2}$</td>
</tr>
</tbody>
</table>

Fig. 5-11 - Flange 3, Stresses at Initial Torquing
FLANGE PARAMETRIC CASE 3 DIA= 4IN PRESS = 500 PSI

SUMMARY OF ANALYSIS

<table>
<thead>
<tr>
<th>PHASE</th>
<th>BOLT FORCE (LB)</th>
<th>APPLIED MOMENT (IN-LB)</th>
<th>AXIAL DISPLACEMENT (IN)</th>
<th>ROTATION (RAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>OPERATION</td>
<td>$2.05749 	imes 10^{-4}$</td>
<td>$6.88662 	imes 10^{-6}$</td>
<td>$7.51977 	imes 10^{-6}$</td>
<td>$1.71770 	imes 10^{-2}$</td>
</tr>
</tbody>
</table>

Fig. 5-12 - Flange 3, Stresses at Proof Pressure
FLANGE PARAMETRIC CASE 7

DIA = 8 IN
PRESS = 100 PSI

PRESSURE 100 PSIG
WEIGHT OF FLANGE 2.404. LB

8.000DIA

0.020

0.125R

0.259DIA, 6 HOLES

0.580DIA SPOTFACE

0.062R FILLET

8.746DIA

2.276

1.526

9.226DIA

0.062

8.257DIA

8.391DIA

FLANGE MATERIAL AL6061T6 RT FTY = 35.000KSI, FTU = 42.000KSI
BOLT MATERIAL AL2024T3 RT FTY = 50.000KSI, FTU = 62.000KSI
GASKET MATERIAL ASBEST1/16 SEATING STRESS = 3.700KSI
YIELD STRENGTH = 10.000KSI

Fig. 5-13 - Flange 7, Design
### Summary of Analysis

<table>
<thead>
<tr>
<th>Phase</th>
<th>Bolt Force (LB)</th>
<th>Applied Moment (IN-LB)</th>
<th>Axial Displacement (IN)</th>
<th>Rotation (RAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bolt-Up</td>
<td>$8.9140 \times 10^{-03}$</td>
<td>$7.45269 \times 10^{-01}$</td>
<td>$0.00000 \times 10^{-01}$</td>
<td>$6.76743 \times 10^{-04}$</td>
</tr>
</tbody>
</table>

**Fig. 5-14** - Flange 7, Stresses at Initial Torquing
SUMMARY OF ANALYSIS

STRESSES ON INSIDE

-40 -30 -20 -10 0 10 20 30 40

STRESSES ON OUTSIDE

-40 -30 -20 -10 0 10 20 30 40

<table>
<thead>
<tr>
<th>PHASE</th>
<th>BOLT FORCE (LB)</th>
<th>APPLIED MOMENT (IN-LB)</th>
<th>AXIAL DISPLACEMENT (IN)</th>
<th>ROTATION (RAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>OPERATION</td>
<td>$1.19645 \times 10^{-04}$</td>
<td>$1.70368 \times 10^{-02}$</td>
<td>$2.17031 \times 10^{-03}$</td>
<td>$1.54703 \times 10^{-03}$</td>
</tr>
</tbody>
</table>

Fig. 5-15 - Flange 7, Stresses at Proof Pressure

5-24
Fig. 5-16 - Flange 8, Design
**FLANGE PARAMETRIC CASE 8**  
**DIA: 8IN PRESS: 200 PSI**

<table>
<thead>
<tr>
<th>PHASE</th>
<th>BOLT FORCE (LB)</th>
<th>APPLIED MOMENT (IN-LB)</th>
<th>AXIAL FLEXURAL (IN)</th>
<th>ROTATION (Radian)</th>
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<tr>
<td>BOLT-UP</td>
<td>$1.93656 \times 10^{-4}$</td>
<td>$1.92988 \times 10^{-2}$</td>
<td>$0.00000 \times 10^{-1}$</td>
<td>$1.22911 \times 10^{-2}$</td>
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</table>

**Fig. 5-17 - Flange 8, Stresses at Initial Torquing**

5-26
FLANGE PARAMETRIC CASE 8 DIA = 8IN PRESS = 200 PSI

SUMMARY OF ANALYSIS

STRESSES ON INSIDE
-90 -30 -20 -10 0 10 20 30 40

STRESSES ON OUTSIDE
-90 -30 -20 -10 0 10 20 30 40

OPERATION

<table>
<thead>
<tr>
<th>PHASE</th>
<th>BOLT FORCE (LB)</th>
<th>APPLIED MOMENT (IN-LB)</th>
<th>AXIAL DISPLACEMENT (IN)</th>
<th>ROTATION (RAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>OPERATION</td>
<td>$2.23584 \times 10^{-4}$</td>
<td>$3.32845 \times 10^{-2}$</td>
<td>$1.23510 \times 10^{-3}$</td>
<td>$2.11988 \times 10^{-2}$</td>
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Fig. 5-18 - Flange 8, Stresses at Proof Pressure
FLANGE PARAMETRIC CASE 9  DIA= 8IN  PRESS = 500 PSI

PRESSURE 500 PSIG
WEIGHT OF FLANGE 2.952 LB

8.000DIA
8.100DIA
9.143DIA
9.998DIA
9.518DIA

0.25OR
0.580DIA SPOTFACE
0.062R FILLET
0.255DIA, 49 HOLES
1.178
1.920
0.062

FLANGE MATERIAL AL6061T6 AT FTV= 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL AL2024T3 AT FTV= 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL ASBEST/16 SEATING STRESS= 3.700KSI
YIELD STRENGTH= 10.000KSI

Fig. 5-19 - Flange 9, Design

5-28
Fig. 5-20 - Flange 9, Stresses at Initial Torquing
FLANGE PARAMETRIC CASE 9 DIA = 8 IN PRESS = 500 PSI

**SUMMARY OF ANALYSIS**

<table>
<thead>
<tr>
<th>Phase</th>
<th>Bolt Force (LB)</th>
<th>Applied Moment (IN-LB)</th>
<th>Axial Displacement (IN)</th>
<th>Rotation (RAD)</th>
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</thead>
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</table>

Fig. 5-21 - Flange 9, Stresses at Proof Pressure
**FLANGE Parametric Case 13**

**DIA** = 12 IN

**PRESS** = 100 PSI

**PRESSURE 100 PSIG**

**WEIGHT OF FLANGE** = 2.374 LB

---

**FLANGE MATERIAL**

AL6061 T6 RT

**FTY =** 35,000 KSI, **FTU =** 42,000 KSI

**BOLT MATERIAL**

AL2024 T3 RT

**FTY =** 50,000 KSI, **FTU =** 62,000 KSI

**GASKET MATERIAL**

ASBEST/16

**SEATING STRESS =** 3,700 KSI

**YIELD STRENGTH =** 10,000 KSI

---

**Fig. 5-22 - Flange 13, Design**

5-31
FLANGE PARAMETRIC CASE 13 DIA=12IN PRESS = 100 PSI

SUMMARY OF ANALYSIS

STRESSES ON INSIDE

<table>
<thead>
<tr>
<th>PHASE</th>
<th>BOLT FORCE (LB)</th>
<th>APPLIED MOMENT (IN-LB)</th>
<th>AXIAL DISPLACEMENT (IN)</th>
<th>ROTATION (RAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>BOLT-UP</td>
<td>1.9459x10^4</td>
<td>1.1717x10^2</td>
<td>0.0000x10^-1</td>
<td>6.4910x10^-3</td>
</tr>
</tbody>
</table>

-40, -30, -20, -10, 0, 10, 20, 30, 40

STRESSES ON OUTSIDE

1.028

BENDING STRESSES IN FLG

12,000DIA

0.623

Fig. 5-23 - Flange 13, Stresses at Initial Torquing

5-32
Fig. 5-24 - Flange 13, Stresses at Proof Pressure
FLANGE PARANETRIC CASE 14  DIA=121N  PRESS = 200 PSI

PRESUSE 200 PSIG
HEIGHT OF FLANGE 2.030 LB

12.000DIA
12.896DIA
13.276DIA
12.100DIA
12.541DIA

0.187R
0.255DIA, 28 MILES
0.500DIA SPOTFACE
0.062R FILLET
0.827
1.577

FLANGE MATERIAL  AL6061T6 RT FY= 35.000KSI, FYU= 42.000KSI
BOLT MATERIAL  AL2024T3 RT FY= 50.000KSI, FYU= 62.000KSI
GASKET MATERIAL  ASBESTO1/16 SEATING STRESS= 3.700KSI
YIELD STRENGTH= 10.000KSI

Fig. 5-25 - Flange 14, Design
FLANGE PARAMETRIC CASE 14 DIA=12IN PRESS = 200 PSI

SUMMARY OF ANALYSIS

<table>
<thead>
<tr>
<th>PHASE</th>
<th>BOLT FORCE (LB)</th>
<th>APPLIED MOMENT (IN-LB)</th>
<th>AXIAL DISPLACEMENT (IN)</th>
<th>ROTATION (RAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>BOLT-UP</td>
<td>(4.28180 \times 10^{-01})</td>
<td>(3.75975 \times 10^{02})</td>
<td>(0.00000 \times 10^{-01})</td>
<td>(2.03468 \times 10^{-02})</td>
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Fig. 5-26 - Flange 14, Stresses at Initial Torquing
FLANGE PARAMETRIC CASE 14  DIA=12IN  PRESS = 200 PSI

SUMMARY OF ANALYSIS

Fig. 5-27 - Flange 14, Stresses at Proof Pressure
FLANGE PARAMETRIC CASE 19  DIA=22IN  PRESS = 100 PSI

PRESSURE 100 PSIG
WEIGHT OF FLANGE 3.961 LB

22.00DIA

22.02DIA

22.10DIA

22.465DIA

22.30DIA

0.255DIA, 43 HOLES
0.500DIA SPOTFACE
0.062R FILLET

0.187R

0.062

0.956

1.706

FLANGE MATERIAL  AL6061T6 RT  FTY= 35.000KSI, FTY= 42.000KSI
BOLT MATERIAL  AL2024T3 RT  FTY= 50.000KSI, FTY= 62.000KSI
GASKET MATERIAL  ASBEST1/16  SEALING STRESS= 3.700KSI
YIELD STRENGTH= 10.000KSI

Fig. 5-28 - Flange 19, Design
Fig. 5-29 - Flange 20, Design
FLANGE PARAMETRIC CASE 25  DIA=451IN  PRESS = 100 PSI

PRESSURE 100 PSIG
WEIGHT OF FLANGE 18.688 LB

0.250R
0.25501A, 179 HOLES
0.580DIA SPOTFACE
0.062R FILLET

2.584
1.834

0.062

FLANGE MATERIAL  AL6061T6 RT  FTY: 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL  AL2024T3 RT  FTY: 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL  ASBEST/16  SEATING STRESS: 3.700KSI
  YIELD STRENGTH: 10.000KSI

Fig. 5-30 - Flange 25, Design

5-39
Table 5-1 - Comparison of Flanges

<table>
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<th></th>
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<tr>
<td>1</td>
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<td>140</td>
<td>23.000</td>
<td>0.140</td>
<td>26.250</td>
<td>1.000</td>
<td>2.125</td>
<td>23.565</td>
<td>1.143</td>
<td>0.7825</td>
<td>1.91</td>
<td>15.59</td>
<td>66.2</td>
<td></td>
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<tr>
<td>2</td>
<td>L1X</td>
<td>140</td>
<td>13.000</td>
<td>0.140</td>
<td>15.250</td>
<td>0.750</td>
<td>1.500</td>
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<td>13.929</td>
<td>1.239</td>
<td>0.9645</td>
<td>6.31</td>
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<tr>
<td>3</td>
<td>L0X</td>
<td>140</td>
<td>8.000</td>
<td>0.140</td>
<td>10.750</td>
<td>0.850</td>
<td>1.375</td>
<td>4.91</td>
<td>9.565</td>
<td>0.789</td>
<td>0.7826</td>
<td>2.10</td>
<td>2.92</td>
<td>57.3</td>
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<tr>
<td>4</td>
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<td>7.780</td>
<td>0.140</td>
<td>10.280</td>
<td>0.720</td>
<td>1.300</td>
<td>2.78</td>
<td>9.345</td>
<td>0.770</td>
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<td>1.06</td>
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<tr>
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<td>7.820</td>
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<td>10.500</td>
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<td>1.52</td>
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<td>7.250</td>
<td>0.630</td>
<td>1.125</td>
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<td>7.565</td>
<td>0.490</td>
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<td>1.01</td>
<td>1.21</td>
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<td>6.500</td>
<td>0.140</td>
<td>9.400</td>
<td>0.750</td>
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<td>8.425</td>
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<td>1.0425</td>
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<td>8</td>
<td>G0X</td>
<td>300</td>
<td>6.000</td>
<td>0.140</td>
<td>6.200</td>
<td>1.000</td>
<td>1.150</td>
<td>2.56</td>
<td>5.565</td>
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<td>0.7825</td>
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<td>G0X</td>
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<td>22.000</td>
<td>0.140</td>
<td>24.250</td>
<td>1.000</td>
<td>2.115</td>
<td>27.55</td>
<td>23.505</td>
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<td>6.250</td>
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<td>1.250</td>
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<td>8.565</td>
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<td>1.27</td>
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<td>6.920</td>
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<td>1.425</td>
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<td>1.22</td>
<td>1.92</td>
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<td>1.97</td>
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<td>0.71</td>
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* Based on Bolt Table 2-3
Fig. 5-31 - Flange Comparison 1

*This number refers to the list on Table 5-1.

Legend: The contour of the conventional flange is indicated by dashed lines and shading.

5-41
LIQUID OXYGEN 190 PSI, 12 IN DIA

PRESURE 190 PSI

12.000 Dia

0.000 Dia

12.000 Dia

13.000 Dia

13.920 Dia

13.000 Dia

0.312R

0.340 Dia, 56 HOLES

0.680 Dia SPOTFACE

0.062 R FILLET

1.909

1.239

0.062

FLANGE MATERIAL AL
BOLT MATERIAL AL
GASKET MATERIAL ALLPAX

FTY= 35.000 KSI, FTU= 42.000 KSI
FTY= 50.000 KSI, FTU= 62.000 KSI
SEATING STRESS= 3.700 KSI
YIELD STRENGTH= 10.000 KSI

Fig. 5-32 - Flange Comparison 2
LIQUID OXYGEN 140 PSI, 8 IN DIA

PRESURE 140 PSIG

0.000DIA

0.190

0.312R

0.275DIA, 91 HOLES
0.500DIA SPOTFACE
0.062R FILLET

0.062

0.000DIA

0.675DIA

FLANGE MATERIAL AL
BOLT MATERIAL AL
GASKET MATERIAL ALLPAK

FTY= 35.000KSI, FTU= 42.000KSI
FTY= 50.000KSI, FTU= 62.000KSI
SEATING STRESS= 3.700KSI
YIELD STRENGTH= 10.000KSI

Fig. 5-33 - Flange Comparison 3
LIQUID OXYGEN 140 PSI, 7.78 IN DIA

Fig. 5-34 - Flange Comparison 4
LIQUID OXYGEN 140 PSI, 7.82 IN DIA

PRESSURE 140 PSIG

FLANGE MATERIAL: AL
BOLT MATERIAL: AL
GASKET MATERIAL: ALLPAX

FTY = 35.000KSI, FTU = 42.000KSI

0.312R
0.275DIA, 40 HOLES
0.980DIA SPOTFACE
0.062R FILLET

0.190
1.525
0.775
0.062

Fig. 5-35 - Flange Comparison 5
LIQUID OXYGEN 90 PSI, 6 IN DIA

PRESSURE 90 PSI

0.312R
0.275DIA, 15 HOLES
0.580DIA SPOTFACE
0.062R FILLET

FLANGE MATERIAL AL
BOLT MATERIAL AL
BASKET MATERIAL ALLPAK

FTY= 35.000KSI, Ftu= 42.000KSI
FTY= 50.000KSI, Ftu= 62.000KSI
SEATING STRESS= 3.700KSI
YIELD STRENGTH= 10.000KSI

Fig. 5-36 - Flange Comparison 6
Fig. 5-37 - Flange Comparison 7
Fig. 5-38 - Flange Comparison 8
Fig. 5-39 - Flange Comparison 9
GASEOUS OXYGEN 100 PSI, 4 IN DIA

PRESSURE 100 PSIG

0.000DIA

0.160

0.312R

0.275DIA, 7 HOLES

0.500DIA SPOTFACE

0.062R FILLET

5.025DIA

5.505DIA

4.000DIA

4.250DIA

FLANGE MATERIAL AL

BOLT MATERIAL AL

GASKET MATERIAL ALLPAK

FTY= 39.000KSI, FTU= 42.000KSI

FTY= 50.000KSI, FTU= 62.000KSI

SEATING STRESS= 3.700KSI

YIELD STRENGTH= 10.000KSI

Fig. 5-40 - Flange Comparison 10
Fig. 5-41 - Flange Comparison 11
Fig. 5-42 - Flange Comparison 12
GASEOUS OXYGEN 80 PSIG, 9 IN DIA

PRESSURE 80 PSIG

5.000DIA 0.140
0.312R

5.850DIA

0.275DIA, 9 HOLES
0.900DIA SPOTFACE
0.062R FILLET

0.696

5.450DIA

0.062

5.247DIA

FLANGE MATERIAL AL
BOLT MATERIAL AL
BASKET MATERIAL ALLPAI

FTY = 35.000KSI, FTU = 42.000KSI
FTY = 50.000KSI, FTU = 62.000KSI
SEATING STRESS = 3.700KSI
YIELD STRENGTH = 10.000KSI

Fig. 5-43 - Flange Comparison 13
Baseous Oxygen 80 PSI, 9.75 in dia

Pressure 80 PSI

0.750 Dia
0.140
0.312R
0.275 Dia, 8 holes
0.500 Dia Spotface
0.062R Fillet
1.500
0.750
0.062

Flange Material: Al
FTY = 35,000 KSI, FTY = 42,000 KSI
Bolt Material: Al
FTY = 60,000 KSI, FTY = 62,000 KSI
Basket Material: All PA1
Seating Stress = 3,700 KSI
Yield Strength = 10,000 KSI

Fig. 5-44 - Flange Comparison 14
GASEOUS OXYGEN 80 PSI, 4 IN DIA

PRESSURE 80 PSI

0.000 Dia

0.000 Dia

0.000 Dia

0.000 Dia

0.000 Dia

0.000 Dia

0.327 Dia, 6 HOLES
0.585 Dia SPOTFACE
0.962R FİLLET

0.870

1.620

0.140

0.312R

FLANGE MATERIAL: AL

BOLT MATERIAL: AL

GASKET MATERIAL: ALPAX

FTY = 35,000 KSI, FTY = 92,000 KSI

FTY = 90,000 KSI, FTY = 62,000 KSI

SEATING STRESS = 3.700 KSI

YIELD STRENGTH = 10,000 KSI

Fig. 5-45 - Flange Comparison 15
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<td>000020</td>
<td>G00</td>
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</table>
**Nominal Pressure** $P = 100000.0$ PSI

**Nominal Diameter** $D = 8.000$ INCH

**Tube Thickness** $T = 0.438$ INCH

**Tube Thickness Tolerance** $DT = 0.000$ INCH

**Height to Weld Height** $HT = 1.750$ INCH

**Proof Factor** $PF = 1.500$

**Burst Factor** $BF = 2.000$

**Safety Factor** $FS = 1.500$

**Gasket Factor** $GF = 2.000$

**Properties of Tube Material**

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<tr>
<th>Material Table No.</th>
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<td>$E = 28000000 + 08$ PSI</td>
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<tr>
<td>Poisson's Ratio</td>
<td>$N_{u} = 0.300$</td>
</tr>
<tr>
<td>Density</td>
<td>$R_\theta = 2880$ LB/CUBIC-1INCH</td>
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<tr>
<td>Thermal Exp Coeff</td>
<td>$\alpha = 95000000 - 05$ INCH/INCH/F</td>
</tr>
<tr>
<td>Tensile Yield Str</td>
<td>$FTY = 35000000 + 05$ PSI</td>
</tr>
<tr>
<td>Ultimate Tens Str</td>
<td>$FTU = 90000000 + 05$ PSI</td>
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**Properties of Flange Material**

<table>
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<tr>
<th>Material Table No.</th>
<th>1 = 5</th>
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<tbody>
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<td>$E = 28000000 + 08$ PSI</td>
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<tr>
<td>Poisson's Ratio</td>
<td>$N_{u} = 0.300$</td>
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<tr>
<td>Density</td>
<td>$R_\theta = 2880$ LB/CUBIC-1INCH</td>
</tr>
<tr>
<td>Thermal Exp Coeff</td>
<td>$\alpha = 95000000 - 05$ INCH/INCH/F</td>
</tr>
<tr>
<td>Tensile Yield Str</td>
<td>$FTY = 35000000 + 05$ PSI</td>
</tr>
<tr>
<td>Ultimate Tens Str</td>
<td>$FTU = 90000000 + 05$ PSI</td>
</tr>
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**Properties of Bolt Material**

<table>
<thead>
<tr>
<th>Material Table No.</th>
<th>1 = 8</th>
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</thead>
<tbody>
<tr>
<td>Elastic Modulus</td>
<td>$E = 28000000 + 08$ PSI</td>
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<tr>
<td>Poisson's Ratio</td>
<td>$N_{u} = 0.300$</td>
</tr>
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<td>Density</td>
<td>$R_\theta = 2880$ LB/CUBIC-1INCH</td>
</tr>
<tr>
<td>Thermal Exp Coeff</td>
<td>$\alpha = 95000000 - 05$ INCH/INCH/F</td>
</tr>
<tr>
<td>Tensile Yield Str</td>
<td>$FTY = 13100000 + 06$ PSI</td>
</tr>
<tr>
<td>Ultimate Tens Str</td>
<td>$FTU = 20000000 + 06$ PSI</td>
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</table>

**Properties of Gasket Material**

<table>
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<tr>
<td>Elastic Modulus</td>
<td>$E = 28000000 + 08$ PSI</td>
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5-57
<table>
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<tr>
<th>Property</th>
<th>Value</th>
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<tbody>
<tr>
<td>Yield Strength KG</td>
<td>400000000 + 5 PSI</td>
</tr>
<tr>
<td>Seating Stress SG</td>
<td>189000000 + 5 PSI</td>
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<tr>
<td>Therm Exp Coeff ALFA</td>
<td>950000000 + 5 INCH/INCH/F</td>
</tr>
<tr>
<td>Coeff of Friction MU</td>
<td>300</td>
</tr>
<tr>
<td>Width Coefficient GAMU</td>
<td>1.000</td>
</tr>
<tr>
<td>Width Coefficient GAMS</td>
<td>1.000</td>
</tr>
<tr>
<td>Gasket Thickness HG</td>
<td>0.0250 INCH</td>
</tr>
<tr>
<td>Sealing Stress Rate SP</td>
<td>5.5000</td>
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</table>

Options

0 1 0 3 1 0 0 0 0 0 0 0

Number of phases to be considered in the analysis = 4

Temperature differential = -500.00 DEG F

Computed thickness T = 0.4375 INCH

Test Flange 1500 PSI, 8 IN DIA

AQB = 0.1063 SQ-IN

Weight = 16.9797 LB

B = 1.3487 IN

H = 2.1049 IN

RI = 4.0000 IN

RG = 4.5183 IN

DI = 8.0000 IN

DG1 = 8.7146 IN

DGO = 9.3587 IN

RB = 4.9506 IN

RFIL = 0.3750 IN

RSPOT = 0.0620 IN

DMOLE = 0.4425 IN

DSPOT = 0.9012 IN

NB = 26

BG = 0.3221 IN

HT = 1.7500 IN

MFU = 11176485 + 5 IN-LB/IN

ZETA1 = 76976176 + 00

ZETA2 = 24211833 + 00
## TEST FLANGE 1500 PSI, 8 IN DIA

**OUTPUT OF THE STRESS AND DEFORMATION ANALYSIS RESULTS**

<table>
<thead>
<tr>
<th>VARIABLE</th>
<th>BOLT-UP</th>
<th>START-UP</th>
<th>OPERATION</th>
<th>SHUT-DOWN</th>
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<tbody>
<tr>
<td>BOLT FORCE (LB)</td>
<td>+37065256+06</td>
<td>+48851169+06</td>
<td>+36011698+06</td>
<td>+35486920+06</td>
</tr>
<tr>
<td>EQUIV APPL MOMENT (IN-LB/IN)</td>
<td>+49960856+09</td>
<td>+27671795+05</td>
<td>+91552638+04</td>
<td>+10710853+05</td>
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<tr>
<td>AXIAL DEFLECTION (IN)</td>
<td>+00000000</td>
<td>+85295984+05</td>
<td>+95295984+05</td>
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<tr>
<td>RADIAL DEFORMATION (IN)</td>
<td>+75853273+03</td>
<td>+21659449+01</td>
<td>+7014149+02</td>
<td>+1876419+02</td>
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<tr>
<td>FLANGE ROTATION (RADIUS)</td>
<td>+13301115+02</td>
<td>+60896844+02</td>
<td>+20147832+02</td>
<td>+23571190+02</td>
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<tr>
<td>BOLT STRESS (PSI)</td>
<td>+13463676+06</td>
<td>+17673339+06</td>
<td>+13028552+06</td>
<td>+12838694+06</td>
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<tr>
<td>GASKET STRESS (PSI)</td>
<td>+40532359+05</td>
<td>+4271927+05</td>
<td>+8677223+05</td>
<td>+2743354+05</td>
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<tr>
<td>STRESS IN FLANGE TOP (PSI)</td>
<td>+5034101+04</td>
<td>+14375453+04</td>
<td>+19920502+05</td>
<td>+12451492+05</td>
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<tr>
<td>STRESS IN FLANGE BOTTOM (PSI)</td>
<td>-13543458+05</td>
<td>-58698999+05</td>
<td>-22027804+04</td>
<td>-20470744+05</td>
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<td>STRESS RESULTANTS</td>
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<td></td>
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<tr>
<td>(LB/IN)</td>
<td>MX= +00000000</td>
<td>+31640625+04</td>
<td>+31640625+04</td>
<td>+4746938+04</td>
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<td>(IN-LB/IN)</td>
<td>NY= +2202554+04</td>
<td>+6289266+09</td>
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<td>(IN-LB/IN)</td>
<td>MZ= +13975052+04</td>
<td>+61693181+04</td>
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<td>(IN-LB/IN)</td>
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<td>+123394+03</td>
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<tr>
<td>(IN-LB/IN)</td>
<td>N= +99834905+04</td>
<td>+11827670+05</td>
<td>+39134061+04</td>
<td>+45781067+04</td>
</tr>
<tr>
<td>STRESSES AT NECK (PSI)</td>
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</tr>
<tr>
<td>(PSI) INNER SIGX= -20697079+05</td>
<td>= 99823225+05</td>
<td>+26289259+05</td>
<td>+29085048+05</td>
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<tr>
<td>(PSI) INNER SIGY= -26852167+04</td>
<td>= 7219995+05</td>
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<tr>
<td>(PSI) OUTER SIGX= +20697079+05</td>
<td>= 89499225+05</td>
<td>+36414258+05</td>
<td>+14277548+05</td>
<td></td>
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<tr>
<td>(PSI) OUTER SIGY= +97333967+04</td>
<td>= 12709639+06</td>
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<tr>
<td>(PSI) MAX TAU= +62001715+04</td>
<td>= 28386406+05</td>
<td>+93916945+04</td>
<td>+10987456+05</td>
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Section 6
CONCLUSION

The foundations have been laid for a simple but comprehensive design procedure for low profile flanges with a subsequent stress and deformation analysis. The algorithms have been programmed and the format for the basic output, i.e., a summary of the flange geometry and a summary of the analysis results have been established.

The computer program is set up for relatively few options of flange configurations within the class of low profile flanges. The amount of programming was limited by the number of man hours available for this contract.

From the accompanying stress analysis it becomes quite obvious whether a design is sound or whether some basic design parameters need to be changes, such as the type of the gasket. The program is not automatic in the sense that it makes selective design decisions, which normally originate in the designer's mind based on his experience. Such a design procedure falls under the category of design optimization from the operations research standpoint (Ref. 37). The method described in Ref. 37, however, could be automated and combined with the current design/analysis program. The few material data and bolt geometry data currently incorporated in the design/analysis program would then have to be expanded to large varieties. This can be done with the current program without any modifications to the existing logic. The current lists would just be longer having more entries.

Further work is needed in verification testing. A test procedure to verify the moment carrying capacity of the flange, covering the entire range from elastic stresses to the formation of the plastic hinge in the flange neck,
is needed. These tests should be carried far beyond the initial yielding. Permanent strains in the highly stressed regions as well as permanent rotations should be measured versus applied moment. These tests can be carried out with an unpressurized test fixture since the entire loading can be expressed as an equivalent externally applied moment on the flange.

The test should be carried with highly instrumented specimens of the following diameter sequence: 4, 8, 12, 22 and 45 inches, as shown on some of the examples in Section 5. Three pressure levels should be considered which have yet to be defined. High pressure levels for the small diameters and lower pressure levels for the large diameters are recommended.

The specimens should first be tested without bolt holes, then with bolt holes, but without spotfaces, finally with spotfaces. Thus the weakening effects and the stress-raising effects of both the bolt holes and the spotfaces could be measured. Finally the fillet should be machined off and a groove as described in Section 2 be established.

The instrumentation should include strain gages on the inside and the outside of the shell wall and the flange to verify essentially the stress distributions shown on the plots labeled "Summary of Analysis." The strain gages should be mounted between bolts and in line with the bolts. Further instrumentation is needed to measure bolt force and gasket contact stress. Finally the deformation measurements, rotation and axial displacement, require some optical devices, possibly mirror systems.

Further analytical work should proceed along the lines of a three-dimensioned elasticity solutions for a typical slice of the flange (see Fig. 6-1), requiring a three-dimensional finite element network. Lockheed-Huntsville's structural network analysis programs have not been made operational to include this type of analysis, although it would take only moderate further development effort to make the appropriate program modifications.
Hopefully the computer program delivered under this contract will be useful to the designers who are meant to use it. As more user's experience is accumulated it will definitely be necessary to make changes and improvements. The accompanying documentation in this report is provided for this purpose.
Fig. 6-1 - Slice $\Delta \theta$ for Finite Element Modeling
Section 7
REFERENCES


Section 8
NOTATION

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A, B, C</td>
<td>coefficients for quadratic equation for h</td>
</tr>
<tr>
<td>A</td>
<td>cross-sectional area of the flange</td>
</tr>
<tr>
<td>A₁, A₂</td>
<td>amplitudes of the stresses</td>
</tr>
<tr>
<td>A₀B</td>
<td>stress area of the bolt</td>
</tr>
<tr>
<td>A_B</td>
<td>total bolt area</td>
</tr>
<tr>
<td>A_G</td>
<td>total gasket area</td>
</tr>
<tr>
<td>A_w</td>
<td>flange cross-sectional area used for weight computation</td>
</tr>
<tr>
<td>a, b</td>
<td>inner and outer tube radius, respectively</td>
</tr>
<tr>
<td>a</td>
<td>radial lever arm between gasket and bolts</td>
</tr>
<tr>
<td>B</td>
<td>bending rigidity of the shell wall</td>
</tr>
<tr>
<td>B</td>
<td>creep constant</td>
</tr>
<tr>
<td>b</td>
<td>width of the flange</td>
</tr>
<tr>
<td>b̅</td>
<td>effective width of the flange</td>
</tr>
<tr>
<td>b_G</td>
<td>gasket width</td>
</tr>
<tr>
<td>b_eff</td>
<td>effective gasket width</td>
</tr>
<tr>
<td>b_s</td>
<td>width of the seal gland</td>
</tr>
<tr>
<td>C₁, C₂</td>
<td>integration constants of the shell equation</td>
</tr>
<tr>
<td>c</td>
<td>axial lever arms between centroid of flange and flange neck</td>
</tr>
<tr>
<td>c₁</td>
<td>distance of spotface from shell outer surface</td>
</tr>
<tr>
<td>c₁, c₂</td>
<td>constants in creep law</td>
</tr>
<tr>
<td>c_E</td>
<td>equivalent rotational spring constant of the gasket and the bolts</td>
</tr>
<tr>
<td>c_F</td>
<td>equivalent rotational spring constant of shell and flange</td>
</tr>
</tbody>
</table>
D determinant of the coefficient matrix of the shell-flange flexibility equation

d_B nominal diameter of the bolt

d_hole diameter of the bolt hole

E elastic modulus

E_B, E_G elastic modulus of the bolts and of the gasket, respectively

e base of the natural logarithm

e radial lever arm

e_1, e_2 radial spacing

F_{tu}, F_{ty} ultimate tensile strength and tensile yield strength, respectively

f_B, f_T, f_F bolt force, tube force, and force on flange face, respectively per unit length of radius r_o

h height of the flange

h^3 conductance parameter

h_G gasket thickness

h_R depth of the recess

h_s depth of the seal gland

I moment of inertia of the flange cross section

K strength

K_c equivalent constant in law governing interface leakage

K_p, k_p slope of sealing-force-vs-pressure curve

k shell parameter

k_B equivalent spring constant of the bolts

k_G equivalent spring constant of the gasket

L wave length of axial variation of stresses

L length of the leak channel

l_B strained length of the bolt
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( M_1 )</td>
<td>pipe bending moment</td>
</tr>
<tr>
<td>( m )</td>
<td>exponent in interface leakage law</td>
</tr>
<tr>
<td>( m_F )</td>
<td>applied flange moment</td>
</tr>
<tr>
<td>( m_o )</td>
<td>edge moment</td>
</tr>
<tr>
<td>( m_x )</td>
<td>meridional bending moment</td>
</tr>
<tr>
<td>( m_\phi )</td>
<td>circumferential bending moment</td>
</tr>
<tr>
<td>( n )</td>
<td>creep exponent</td>
</tr>
<tr>
<td>( n_B )</td>
<td>number of bolts</td>
</tr>
<tr>
<td>( n_\lambda )</td>
<td>axial stress resultant</td>
</tr>
<tr>
<td>( n_\phi )</td>
<td>circumferential stress resultant</td>
</tr>
<tr>
<td>( P )</td>
<td>creep parameter in Lawson-Miller creep law</td>
</tr>
<tr>
<td>( P_B )</td>
<td>bolt force</td>
</tr>
<tr>
<td>( P_F )</td>
<td>total force on flange required</td>
</tr>
<tr>
<td>( P_G )</td>
<td>gasket force</td>
</tr>
<tr>
<td>( P_P )</td>
<td>force caused by internal pressure</td>
</tr>
<tr>
<td>( p )</td>
<td>pressure</td>
</tr>
<tr>
<td>( Q )</td>
<td>volume leak rate</td>
</tr>
<tr>
<td>( q_o )</td>
<td>edge shear</td>
</tr>
<tr>
<td>( q_x )</td>
<td>meridional shear stress resultant</td>
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<tr>
<td>( R )</td>
<td>stress ratio for fatigue design</td>
</tr>
<tr>
<td>( r )</td>
<td>radius of a point in the shell wall</td>
</tr>
<tr>
<td>( r_o )</td>
<td>radius of the shell wall middle surface</td>
</tr>
<tr>
<td>( r_a )</td>
<td>equivalent radius of gasket and bolts spring constant</td>
</tr>
<tr>
<td>( r_B )</td>
<td>bolt circle radius</td>
</tr>
<tr>
<td>( r_c )</td>
<td>radius of the flange centroid</td>
</tr>
<tr>
<td>( r_{fil} )</td>
<td>fillet radius on the upper surface of the flange</td>
</tr>
</tbody>
</table>
\( r_F \) radius of the application point of the force acting on the flange face
\( r_G \) gasket radius
\( r_i \) inner radius of the tube
\( r_s \) radius of the seal contact surface
\( r_{\text{spot}} \) fillet radius for the spotface
\( r_w \) radius used for weight computation
\( S_I \) pipe shear force
\( S_F \) elastic section modulus of the flange
\( S_G \) line load on the gasket
\( s \) circumferential spacing
\( T \) temperature
\( t \) wall thickness
\( t_G \) thickness of the gasket
\( t_n \) part of tube wall thickness required to carry axial force due to pressure
\( t_{\text{rupt}} \) time to rupture
\( u \) axial displacement of the flange
\( \text{vol} \) volume added to the tube by the flange
\( W \) weight leak rate
\( w \) width of the leak channel
\( w \) radial displacement
\( x \) axial coordinate
\( Y_0 \) tensile yield strength
\( Z_F, Z_T \) plastic section modulus of the tube and the flange, respectively
\( \alpha \) linear thermal expansion coefficient
\( \alpha, \beta \) creep constants
\( \alpha_1, \alpha_2 \) stresses relating \( \sigma_\varphi \) and \( \tau_{xz} \) to \( \sigma_x \) when yielding occurs
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\beta$</td>
<td>dimensionless parameters containing shell and flange stiffnesses</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>angle between cylinder axis and weld</td>
</tr>
<tr>
<td>$\gamma_1', \gamma_2'$</td>
<td>coefficients relating the physical gasket width to the effective gasket width at yielding and under operating conditions, respectively</td>
</tr>
<tr>
<td>$\Delta p$</td>
<td>internal pressure difference</td>
</tr>
<tr>
<td>$\Delta T$</td>
<td>temperature difference between flange and tube</td>
</tr>
<tr>
<td>$\Delta t$</td>
<td>tube wall thickness tolerance</td>
</tr>
<tr>
<td>$\Delta W$</td>
<td>weight added to the tube by the flange</td>
</tr>
<tr>
<td>$\Delta w$</td>
<td>difference in radial displacement between flange and shell</td>
</tr>
<tr>
<td>$\Delta \sigma_B, \Delta \sigma_G$</td>
<td>change in stress of bolts and gasket, respectively</td>
</tr>
<tr>
<td>$\delta_B$</td>
<td>deflection of the bolts</td>
</tr>
<tr>
<td>$\delta_G$</td>
<td>deflection of the gasket</td>
</tr>
<tr>
<td>$\epsilon$</td>
<td>strain</td>
</tr>
<tr>
<td>$\xi_1', \xi_2'$</td>
<td>coefficients related to the state of stress in the flange neck</td>
</tr>
<tr>
<td>$\eta_0, \eta_1, \eta_2$</td>
<td>design parameters for bolt spacing</td>
</tr>
<tr>
<td>$\mu$</td>
<td>coefficient of friction</td>
</tr>
<tr>
<td>$\nu$</td>
<td>Poisson's ratio</td>
</tr>
<tr>
<td>$\rho$</td>
<td>density of the material</td>
</tr>
<tr>
<td>$\rho$</td>
<td>shell parameter</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>stress, contact stress</td>
</tr>
<tr>
<td>$\sigma_B$</td>
<td>stress in the bolts</td>
</tr>
<tr>
<td>$\sigma_G$</td>
<td>contact stress of the gasket</td>
</tr>
<tr>
<td>$\sigma_x, \sigma_\varphi, \tau_{xz}$</td>
<td>axial circumferential and radial shear stress, respectively</td>
</tr>
<tr>
<td>$\bar{\sigma}, \sigma_e$</td>
<td>equivalent stress</td>
</tr>
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<td>$\varphi$</td>
<td>circumferential angle</td>
</tr>
<tr>
<td>$\chi$</td>
<td>roll angle</td>
</tr>
<tr>
<td>$\psi$</td>
<td>weld weakening factor</td>
</tr>
<tr>
<td>(F.S.)</td>
<td>factor of safety</td>
</tr>
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</table>
Appendix A
SUMMARY OF THE DESIGN PROCEDURE
Appendix A

A.1 TUBE THICKNESS $t$ (in.) OR (mm)

(a) Given

$$ t = \frac{(F.S.) \ p \ r_i}{F_{ty}} $$

where

$(F.S.)$ = a factor of safety

$p$ = the design pressure (usually proof pressure)

$r_i$ = radius of the inside of the tube, equal one-half of the inner diameter, $d_i$, of the tube

$F_{ty}$ = tensile yield strength of the tube

(b) $t = T_{ty}$

(c) $t_1 = \frac{(B.F.) \ p_1 \ r_i}{F_{tu} \ \psi} + 2 \Delta t$

$$ t_2 = \frac{(P.F.) \ p_2 \ r_i}{F_{ty} \ \psi} + 2 \Delta t $$

where

$(B.F.)$ = safety factor for burst condition

$(P.F.)$ = safety factor for proof condition

$\psi = 0.70$ ... $1.00$ (weld reduction)

$\Delta t$ = manufacturing tolerance for the tube wall (approximately $0.01$ in)

$p_1$ = burst pressure

$p_2$ = proof pressure

$F_{tu}$ = ultimate tensile strength of the tube
\[ F_{ty} = \text{tensile yield strength of the tube} \]
\[ t = \text{maximum of } t_1 \text{ and } t_2 \]

\[(d) \quad t_1 = \frac{1.1 \text{ (B.F.) } p_1 r_i}{F_{ty} - 0.4 \text{ (B.F.) } p_1} \]
\[(d) \quad t_2 = \frac{1.1 \text{ (P.F.) } p_2 r_i}{F_{tu} - 0.4 \text{ (P.F.) } p_2} \]

where the symbols have the same meaning as in (c)

A.2 BOLT SIZE \( d_B \) (in) OR (mm)

Initial estimate \( d_B = t \)

from bolt table for the proper wrench clearance and the given \( d_B \) find from Tables 2-1 through 2-3 or Tables 2-4 through 2-6 the following quantities

\( \eta_0, \eta_1, \eta_2, A_0B, d_{\text{hole}}, r_{\text{spot}}, r_{\text{size}} \)

(a) Machined Spot Faces

\[ e_1 = \eta_1 d_B \]
\[ e_2 = \eta_2 d_B \]

\( r_{\text{fil}} \) (fillet radius) according to Table 2-7.

(b) Machined Groove

\[ e_1 = \eta_2 d_B \]
\[ e_2 = e_1 \]

\( r_{\text{fil}} \) (groove radius) according to Table 2-8.
A.3 BOLT CIRCLE RADIUS $r_B$ (in) OR (mm)

(a) Machined Spot Faces

$$r_B = r_i + t + c_1 + e_1$$

where

$$c_1 = 0.0625 \text{ in or } 1.5 \text{ mm}$$

(b) Machined Groove

$$r_B = r_i + t + 2r_{fil} + e_1$$

A.4 FLANGE WIDTH $b$ (in) OR (mm)

$$b = r_B + e_2 - r_i$$

A.5 GASKET WIDTH $b_G$ AND GASKET RADIUS $r_G$ (in) OR (mm)

Estimate for gasket radius, $r_G$:

$$r_G = \frac{1}{2} \left( r_B - \frac{d_{\text{hole}}}{2} + r_i \right)$$

Gasket Width, $b_G$:

(a) $b_G = \frac{(PF) p r_G}{2[\gamma_1 K_G - \gamma_2 \sigma_G (FG) \text{ or}]}$

(b) $b_G = \frac{(PF) p r_G}{2[\gamma_1 K_G - \gamma_2 K_p (PF) P (GF)]}$

where

$\gamma_1$ = a width factor for the gasket under initial deformation

$K_G$ = the yield (crushing) strength of the gasket

$\gamma_2$ = a width factor for the gasket under operating condition
\[ \sigma_G = \text{seating stress of the gasket} \]
\[ (G.F.) = \text{gasket factor} \]
\[ k_p = \text{ratio of seating stress over pressure} \]

\[ b_G = r_B - \frac{d_{\text{hole}}}{2} - r_i - 2c_2 \] (the available space is used)

where \( c_2 \) is a tolerance. \( c_2 = 0.05 \text{ in or 1.0 mm.} \)

The force required to pre-form the gasket is

\[ P_G^{(1)} = 2\pi r_G b_G \gamma_1 K_G \]

and the force required to keep a zero-leak connection is

\[ P_G^{(2)} = P_p + 2\pi r_G b_G \gamma_2 \sigma_G \] (G.F.) or

\[ P_G^{(2)} = P_p + 2\pi r_G b_G \gamma_2 k_p \] (P.F.) p (G.F.)

where

\[ P_p = \pi r_G^2 \] (P.F.).

With the width \( b_G \) computed above the condition

\[ P_G^{(1)} = P_G^{(2)} \]

should be met.

Gasket Radius, \( r_G \):

(1) Gasket Close to Inside of Tube

\[ r_G = r_i + b_G/2 + c_2 \]
Check for space: 
\[ r_1 = r_G + \frac{b_G}{2} + c_2 \]
\[ r_2 = r_B - \frac{d_{\text{hole}}}{2} - c_2 \]
if \( r_1 > r_2 \) set 
\[ r_G = r_i + \frac{b_G}{2} + c_2 \]
\[ r_B = r_G + \frac{b_G}{2} + \frac{d_{\text{hole}}}{2} + c_2 \]

(2) Gasket Close to the Bolts 
\[ r_G = r_B - \frac{d_{\text{hole}}}{2} - \frac{b_G}{2} - c_2 \]

Check for space as under (a).

A.6 PRESSURE ENERGIZED SEAL, EQUIVALENT \( b_G \) AND \( r_G \) (in) OR (mm)

Estimate \( b_G = t \)

find 
\[ r_G = r_i + \frac{b_G}{2} \]

check if space for seal gland is sufficient:

\[ r_1 = r_i + b_G + 2 b_s \]
\[ r_2 = r_B - \frac{d_{\text{hole}}}{2} \]
if \( r_1 > r_2 \) set 
\[ r_B = r_G + \frac{b_G}{2} + 2 b_s + \frac{d_{\text{hole}}}{2} \]

where \( b_s \) is the width of the seal gland.

In both cases (A.5 and A.6) the width of the flange has to be recalculated using the new bolt circle radius,

\[ b = r_B + e_2 - r_i \]

as under A.4.

A-5
A.7 REQUIRED BOLT FORCE $P_B$ (lb) OR (N)

(a) Flat Gasket (see A.5)

\[ P_B^{(1)} = 2\pi r_G b_G \gamma_1 K_G \]
\[ P_B^{(2)} = \pi r_G^2 p \text{(P.F.)} + 2\pi r_G b_G \gamma_2 \sigma_g \text{(G.F.)} \]

or

\[ P_B^{(2)} = \pi r_G^2 p \text{(P.F.)} + 2\pi r_G b_G \gamma_2 \text{(P.F.)} k_p p \text{(G.F.)} \]

\[ P_B = \text{maximum of } P_B^{(1)} \text{ and } P_B^{(2)} \]

(b) Pressure Energized Seal

\[ P_B = \pi r_G^2 p \text{(P.F.)} \]

A.8 NUMBER OF BOLTS, $n_B$

\[ n_{B1} = \frac{P_B}{F_{ty} A_o B} \]
\[ n_{B2} = \frac{(B,F./P,F.) P_B}{F_{tu} A_o B} \]

\[ n_B = \text{maximum of } n_{B1} \text{ and } n_{B2} \]

where

\[ F_{ty} = \text{tensile yield strength of the bolt} \]
\[ F_{tu} = \text{ultimate tensile strength of the bolt} \]

A.9 BOLT SPACING $s$ (in) OR (mm)

\[ s = 2\pi r_B / n_B \]

A-6
if \( s/d_B > 8 \) decrease bolt size (if possible) and go back to A.2.

if \( s/d_B < \eta_0 \) increase bolt size (if possible) and go back to A.2.

A.10 FLANGE HEIGHT \( h \) (in) OR (mm)

\[
e = r_B - r_G
\]

\[
r_o = r_1 + t/2
\]

\[
t_N = t/2
\]

Ultimate moment to be carried

\[
m_{Fu} = \frac{(F.S.) P_B e}{2\pi r_o}
\]

Subtract effect of bolt holes from the flange width

\[
\bar{b} = b - d_{\text{hole}} \sqrt{d_{\text{hole}}/s}
\]

Assume

\[
\zeta_1 = 0.8
\]

\[
\zeta_2 = 0.18
\]

and compute

\[
A = \frac{F_{ty}}{F} \frac{\bar{b}}{6 r_o}
\]

\[
B = \frac{F_{ty}}{F} \zeta_2 (t - t_N)/2
\]

\[
C = \frac{F_{ty}}{F} \zeta_1 (t^2 - t_N^2)/4 - m_{Fu}
\]

\[
R^2 = B^2 - 4 AC, \quad R = \sqrt{R^2}
\]

\[
h = (R - B)/2A
\]
where $F_{ty}^F$ = tensile yield strength of the flange.

A.11 CHECK FLANGE HEIGHT

if $s/h > 3$  \( h = s/3 \)

to prevent waviness of the flange when too thin.

A.12 WEIGHT $\Delta W$ (lb) OR (kg)

\[ r_w = \frac{2r_1 + t + b}{2} \]

\[ A_w = (b - t) \cdot h \]

\[ V_{ol} = 2\pi \cdot r_w \cdot A_w \]

\[ \Delta W = \rho_F \cdot V_{ol} \]

where $\rho_F$ = weight density of the flange.

NOTE: Material data for some flange, bolt and gasket materials are given in Tables 2-10 through 2-13.
Appendix B

SUMMARY OF THE ANALYSIS METHOD

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Appendix B

B.1 APPLIED FORCES

\[ f_x = \frac{p r_o}{2} \] (axial force in the tube wall)
\[ f_r = \frac{p h r_i}{r_o} \] (radial force on the flange)
\[ f_p = \frac{p(r_G^2 - r_i^2)}{2 r_o} \] (pressure force on the flange face)

where
- \( p \) = applied pressure
- \( r_o \) = radius of the tube wall middle surface
- \( r_i \) = inner radius of the tube
- \( h \) = flange height
- \( r_G \) = gasket radius

B.2 SPRING CONSTANTS

(a) Gasket

\[ A_G = 2\pi r_G b_G \]  
\[ K_G = \frac{A_G E_G}{2\pi r_o h_G} \] (Linear spring constant for flat gasket)

\[ K_G = \frac{A_G E_F}{2\pi r_o (h_R + h_s)} \] (Linear spring constant for cantilever flange with pressure energized seal)

where
- \( A_G \) = gasket area
- \( h_G \) = gasket thickness
(b) Bolts

$$A_B = n_B A_{oB}$$

$$K_B = A_B E_B / 2\pi r_o l_B$$ (Linear spring constant for the bolts)

where

- $A_B$ = total bolt stress area
- $A_{oB}$ = stress area of one bolt
- $n_B$ = number of bolts
- $l_B$ = stressed length of the bolt.

(c) Equivalent Rotational Spring of Bolts and Gasket

$$e = r_B - r_G$$

$$r_a = r_G + K_B e / (K_B + K_G)$$

$$c_E = K_B K_G e^2 / (K_B + K_G)$$ (Rotational spring constant for bolts and gasket)

where

- $e$ = lever arm between bolt circle and gasket circle
- $r_a$ = radius of centroid of combined springs
(d) Equivalent Rotational Spring of the Flange

\[ A_F = b h \]
\[ I_F = A_F \frac{h^2}{12} \]
\[ c = \frac{h}{2} \]
\[ r_c = r_1 + \frac{b}{2} \]
\[ B = E t^3 / 12(1 - \nu^2) \]
\[ k = \frac{12(1 - \nu^2)}{r^2 t^2} \]
\[ \beta = Bk r_0 r_c / E_F I_F \]
\[ D = (1 + \beta) \left( \frac{1}{2k^2} + \beta \left( c^2 + \frac{I F}{A_F} \right) \right) - (c\beta - \frac{1}{2k})^2 \]

\[ c_F = \frac{BD}{\beta \left( \frac{BI F}{k A_F} + \frac{1}{4k^3} \right)} \]

(Rotational spring constant for bolts and gasket)

where

- \( A_F \) = cross-sectional area of the flange
- \( b \) = flange width
- \( I_F \) = moment of inertia of the flange cross section
- \( r_o \) = radius of the centroid of the flange
- \( B \) = bending rigidity of the tube wall
- \( k \) = shell parameter
- \( \beta \) = flange parameter
- \( D \) = determinant of coefficient matrix of equation for interface bending moment and interface shear force at the flange neck

(e) Constants for the Determining \( w_o \), \( m_o \) and \( q_o \) at the Flange Neck

\[ c_w = 1/2k^3 B \]
c_m = \frac{\beta}{D \left( \frac{1}{2k} + \beta \frac{I_F}{A_F} + \frac{c}{2k} \right)}

c_q = \frac{\beta}{D (c + \frac{1}{2k})}

B.3 INITIAL TORQUING (o)

m_{F}^{(o)} = e \frac{P_B}{2\pi r_o} \quad \text{(Applied flange moment)}

\chi^{(o)} = \frac{m_{F}^{(o)}}{c_F} \quad \text{(Flange rotation)}

c_B^{(o)} = \frac{P_B}{A_B} \quad \text{(Bolt stress)}

c_G^{(o)} = \frac{P_B}{A_G} \quad \text{(Gasket stress)}

Variables at the flange neck:

n_x^{(o)} = 0 \quad \text{(Axial force)}

m_x^{(o)} = c_m m_{F}^{(o)} \quad \text{(Meridional bending moment)}

q_x^{(o)} = c_q m_{F}^{(o)} \quad \text{(Shear force)}

w^{(o)} = c_w (q_x^{(o)} - k m_x^{(o)}) \quad \text{(Radial deflection)}

n_y^{(o)} = E_T t \frac{w^{(o)}}{r_o} \quad \text{(Circumferential force)}

m_y^{(o)} = \nu m_x^{(o)} \quad \text{(Circumferential bending moment)}
B.4 PRESSURIZATION (p)

\[ m_F^{(1)} = f_p (r_a - r_p) + f_x (r_a - r_o) \]
\[ \Delta w = \frac{p r_o^2 (1 - \nu/2)}{E_T t} + r_o \alpha_T T - \alpha_T r \frac{r_c f_r}{E_F A_F} \]
\[ m_F^{(2)} = \frac{\beta (c + 1/2k) \Delta w}{D} c_F \]
\[ u_p = (f_p + f_x)/(K_G + K_B) \]
\[ (p) = \frac{m_F^{(1)} + m_F^{(2)}}{c_E + c_F} \]
\[ \chi^{(T)} = \chi^{(o)} + \chi^{(p)} \]
\[ \delta_G = u_p + \chi^{(p)} K_B e/(K_G + K_B) \]
\[ \delta_B = u_p - \chi^{(p)} K_G e/(K_B + K_B) \]
\[ \sigma_G^{(T)} = \sigma_G^{(o)} + E_G \delta_G/h_G \]
\[ \sigma_B^{(T)} = \sigma_B^{(o)} + E_B \delta_B/f_B \]

Variables at the flange neck:
\[ n_x = f_x \]
\[ m_F^{(T)} = c_F \chi^{(T)} \]
\[ m_x^{(T)} = c_m m_F^{(T)} \]
\[ q_x^{(T)} = c_q m_F^{(T)} \]
\[ w^{(T)} = c_w (q_x^{(o)} - k m_x^{(o)}) + \frac{p r_o^2 (1 - \nu/2)}{E_T t} + r_o \alpha_T \Delta T \] (Radial deflection)

\[ n_y^{(T)} = \frac{E_T t w^{(T)}}{r_o} - r_o \alpha_T \Delta T + p r_o \nu \] (Circumferential force)

\[ m_y^{(T)} = \nu m_x^{(T)} \] (Circumferential bending moment)

### B.5 STRESSES IN THE FLANGE

\[ w_{\text{top}} = w \]
\[ w_{\text{bottom}} = w - h X \]
\[ \sigma_{\text{top}} = E_F w_{\text{top}} / r_o \]
\[ \sigma_{\text{bottom}} = E_F w_{\text{bottom}} / r_o \]

### B.6 STRESSES IN THE FLANGE NECK

\[ \sigma_x = \pm \frac{6 m_x}{t^2} + \frac{n_x}{t} \]
\[ \sigma_y = \pm \frac{6 m_y}{t^2} + \frac{n_y}{t} \]

\[ \tau_{xz} = 1.5 q_x / t \text{ (max)} \]

where

- \( m_x \) = meridional bending moment
- \( m_y \) = circumferential bending moment
- \( n_x \) = axial force
- \( n_y \) = circumferential force
- \( q_x \) = shear force.
B.7 VARIATION OF THE SHELL VARIABLES ALONG THE TUBE

\[ n_x(x) = n_x(0) \]

\[ w_x(x) = c_w e^{-k_x} \left[ q_x(0) \cos kx - k m_x(0) (\cos kx - \sin kx) \right] \]

\[ + \frac{p r^2_o (1 - \nu^2)}{E_T t} + r_o \alpha_T \Delta T \]

\[ m_x(x) = e^{-k_x} \left[ m_x(0) (\cos kx + \sin kx) - \frac{q_x(0)}{k} \sin kx \right] \]

\[ q_x(x) = e^{-k_x} \left[ q_x(0) (\sin kx - \cos kx) + 2 k m_x(0) \sin kx \right] \]

\[ n_y(x) = E_t w_x(x)/r_o - r_o \alpha_T \Delta T + p r_o \nu/2 \]

\[ m_y(x) = \nu m_x(x) \]

The stresses at a point x are then computed according to paragraph B.6.

B.8 PLASTIC HINGE

\[ \alpha_1 = \frac{t^2}{4} \frac{t_n}{E} \frac{\beta I_F}{k A_F} + \frac{1}{4k^3} \frac{w + \nu}{B \left( \frac{1}{2k^2} + \beta \frac{I_F}{A_F} + \frac{c}{2k} \right)} \]

\[ \alpha_2 = \frac{t + t_n}{4} \frac{c + \frac{1}{2k}}{\left( \frac{1}{2k^2} + \beta \frac{I_F}{A_F} + \frac{c}{2k} \right)} \]

\[ \overline{\alpha} = \sqrt{1 + \alpha_1 + \alpha_2 + \sqrt{3 \alpha_2^2}} \]
\[ \zeta_1 = \frac{1}{\alpha} \]

\[ \zeta_2 = \frac{\alpha_2}{\alpha} \]

\[ S_x = b h^2/2 \]

\[ m_{F_u} = Y_o \left[ \frac{S}{r_o} + \frac{\zeta_1}{4} (t_2^2 - t_n^2) + \zeta_2 (t - t_n) \frac{h}{2} \right] \] (Ultimate applied flange moment)
Appendix C
INPUT INSTRUCTIONS FOR DESIGN AND ANALYSIS PROGRAM
Appendix C

<table>
<thead>
<tr>
<th>Card</th>
<th>Format</th>
<th>Column</th>
<th>Description</th>
<th>Units</th>
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<td>1-72</td>
<td>Instruction to plotter operator</td>
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</tr>
<tr>
<td>1</td>
<td>I5</td>
<td>1-5</td>
<td>Number of cases</td>
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<td>12A6</td>
<td>1-72</td>
<td>Title of the plots</td>
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<tr>
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<td>E10.4</td>
<td>1-10</td>
<td>( p = \text{pressure} )</td>
<td>psi (N/mm(^2))</td>
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<tr>
<td></td>
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<td>11-20</td>
<td>( d_i = \text{inner diameter of the tube} )</td>
<td>in.(mm)</td>
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<td>21-30</td>
<td>( t = \text{thickness of the tube} )</td>
<td>in.(mm)</td>
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<tr>
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<td>31-40</td>
<td>( \Delta t = \text{thickness tolerance} )</td>
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<td>41-50</td>
<td>( h_T = \text{height of tube frustum being part of the flange} )</td>
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<tr>
<td>4</td>
<td>E10.4</td>
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<td>P.F. = proof factor</td>
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<td>11-20</td>
<td>B.F. = burst factor</td>
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<td>21-30</td>
<td>F.S. = factor of safety</td>
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<td>G.F. = gasket factor</td>
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<td>( i_T = \text{tube material number} ) (see Table 2-6)</td>
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<td>( i_F = \text{flange material number} ) (see Table 2-6)</td>
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<td>( i_B = \text{bolt material number} ) (see Table 2-6)</td>
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<td>( i_G = \text{gasket material number} ) (see Table 2-7)</td>
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<td>Material properties of tube</td>
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<td>E10.4</td>
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<td>( E_T = \text{elastic modulus} )</td>
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<td>11-20</td>
<td>( \nu_T = \text{Poisson's ratio} )</td>
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<td>21-30</td>
<td>( \rho_T = \text{density} )</td>
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C-1
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<td>$F_{ty}$ = tensile yield strength</td>
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<td>$F_{tu}$ = ultimate tensile strength</td>
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<td>$\sigma_G$ = seating stress</td>
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<td>41-50</td>
<td>$\mu_G$ = friction coefficient</td>
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<td>Only if $i_G &lt; 0$</td>
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<td>1-10</td>
<td>$h_s$ = depth of the seal gland</td>
<td>in.(mm)</td>
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<td>E10.4</td>
<td>11-20</td>
<td>$b_s$ = width of the seal gland</td>
<td>in.(mm)</td>
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<td>$h_R$ = depth of the recess</td>
<td>in.(mm)</td>
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<td>1015</td>
<td>1-50</td>
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<td>Option 1 = 1:</td>
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<td>Tube thickness computed according to Appendix A, paragraph A.1(b):</td>
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C-2
<table>
<thead>
<tr>
<th>Card</th>
<th>Format</th>
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<th>Description</th>
<th>Units</th>
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<tr>
<td>10</td>
<td>10I5</td>
<td>1-50</td>
<td>Option 1 = 2:&lt;br&gt;Same, but paragraph A.1(c)</td>
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<td>Option 1 = 3:&lt;br&gt;Same, but paragraph A.1(d)</td>
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<td>Option 2 = 1:&lt;br&gt;Machined spot faces</td>
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<td>Option 3 ≤ 0:&lt;br&gt;Flat gasket</td>
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<tr>
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<td>Option 3 ≥ 1:&lt;br&gt;Pressure activated seal</td>
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<td>Option 4 = 1:&lt;br&gt;Open wrenching (see Table 2-1 or 2-4)</td>
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<td>Option 4 = 2:&lt;br&gt;Socket wrenching (see Table 2-2 or 2-5)</td>
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<td>Option 4 = 3:&lt;br&gt;Internal wrenching (see Table 2-3 or 2-6)</td>
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<td>Option 5 = 0:&lt;br&gt;Gasket width according to paragraph A.5(a)</td>
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<td>Option 5 = 1:&lt;br&gt;Gasket width according to paragraph A.5(b)</td>
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<td>Option 5 = 2:&lt;br&gt;Gasket width according to paragraph A.5(c)</td>
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<td>Option 6 ≠ 3:&lt;br&gt;Gasket close to bolt circle</td>
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<td>Option 6 = 3:&lt;br&gt;Gasket close to inside of tube</td>
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<td>Name of tube material</td>
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<td>Name of flange material</td>
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<td>25-36</td>
<td>Name of bolt material</td>
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<tr>
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<td>2A6</td>
<td>37-48</td>
<td>Name of gasket material (or seal, as applicable)</td>
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<tr>
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<td>I5</td>
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<td>Number of loading phases</td>
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<td>E10.4</td>
<td>1-10</td>
<td>Temperature differential between tube and flange</td>
<td>°F (°C)</td>
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</table>

C-3