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HUNTSVILLE, ALABAMA

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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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APPROVED: George D. Perry

for J. S. Farrior
Resident Director

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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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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 Prasthofer (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 χ 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

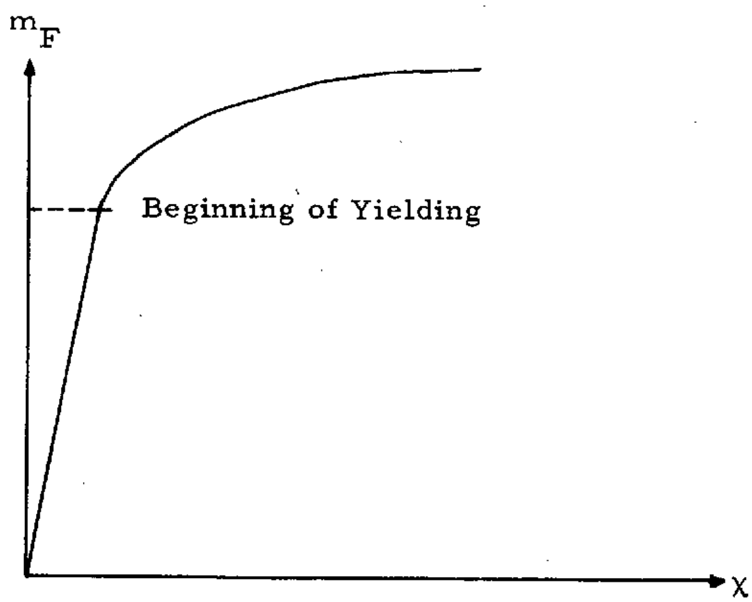
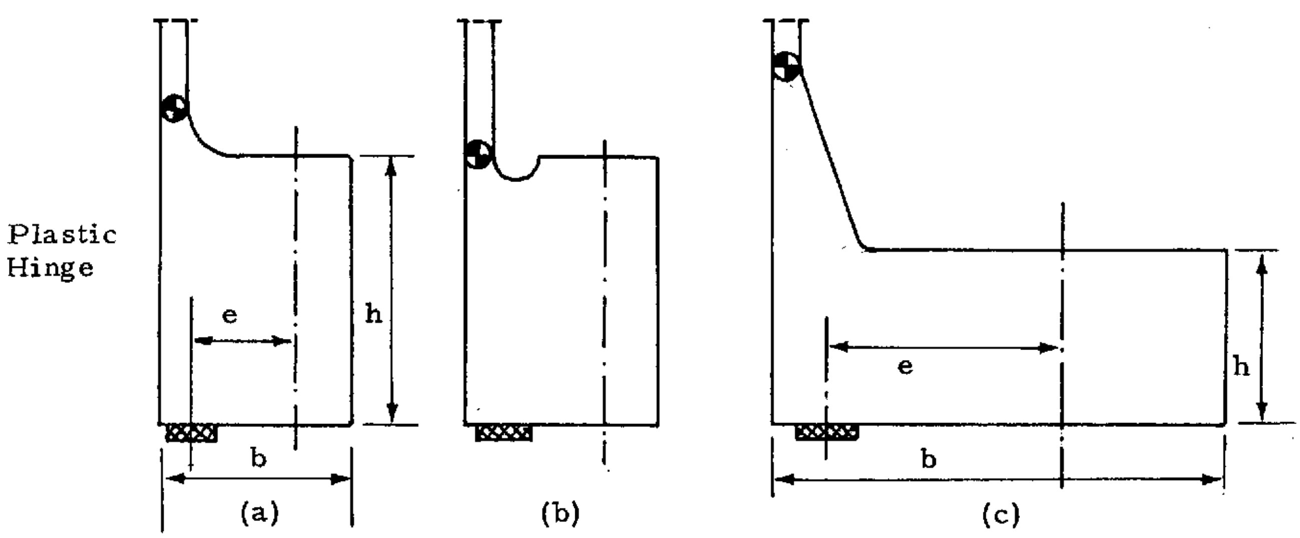


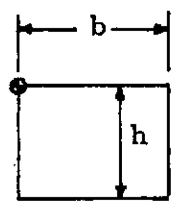
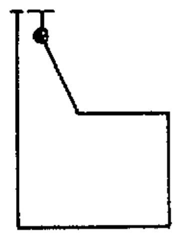
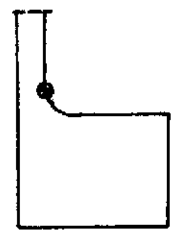
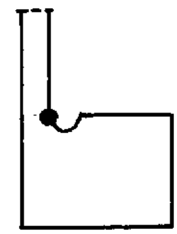
Fig. 1-1 - Applied Moment vs Roll Angle of the Flange



Low Profile Flanges

Conventional Flange

Fig. 1-2 - Location of the Plastic Hinge

	Configuration (a)	(b)	(c)	(d)
				
$^{*}m_{F_1}$	.474	1.0	.587	.332
$^{**}m_{F_2}$	.387	1.0	.635	.403

*0.2% permanent set at ⊕

**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 apprcximately 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

$$\sigma_{e_{max}} = \frac{K}{(F.S.)} \tag{1.1}$$

where $\sigma_{e_{max}}$ 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, $\sigma_{e_{max}}$, 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

$$\sigma_e = \frac{1}{\sqrt{2}} \sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2} \tag{1.2}$$

where σ_1, σ_2, σ_3 are the principal stresses. For components subjected to high tensile stresses, i.e., if $\sigma_1 > 0$, the equivalent stress is

$$\sigma_e = \sigma_1 \tag{1.3}$$

This failure mode is fracture.

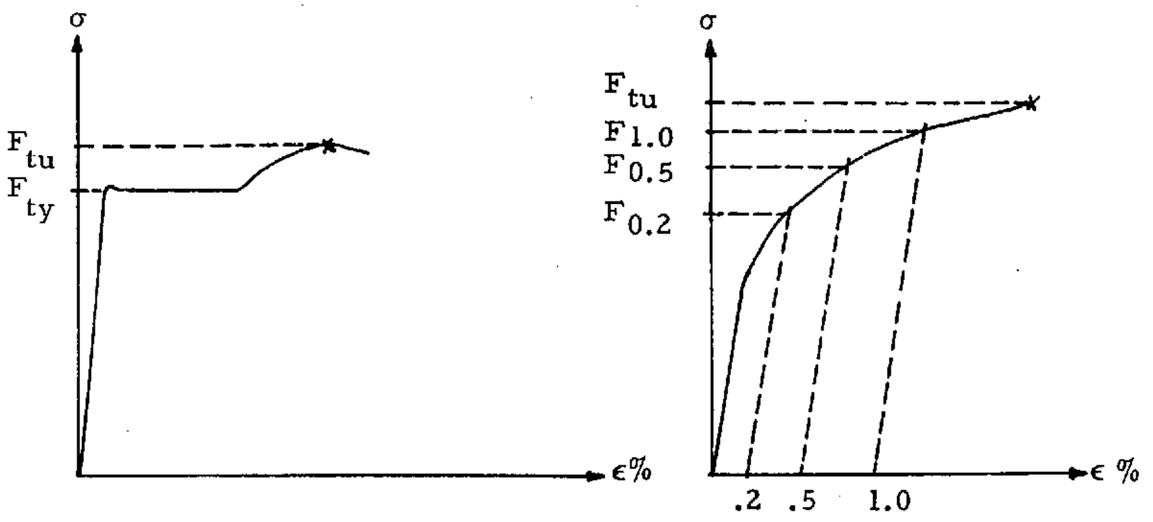
A third equivalent stress occasionally considered is the one defined by Tresca (Ref. 10) and is used for shear failures,

$$\sigma_e = \sigma_{max} - \sigma_{min} = 2\tau_{max} \quad . \tag{1.4}$$

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 $\sigma_{e_{max}}$ 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_{ty}. The ultimate tensile strength is F_{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 $F_{ty} = F_{.2}$, where $F_{.2}$ is the 0.2% stress at permanent set. If F_{tu} is much larger than $F_{.2}$ 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, ϵ_{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.



(a) Steel (b) Aluminum

Fig. 1-4 - Typical Stress-Strain Diagrams

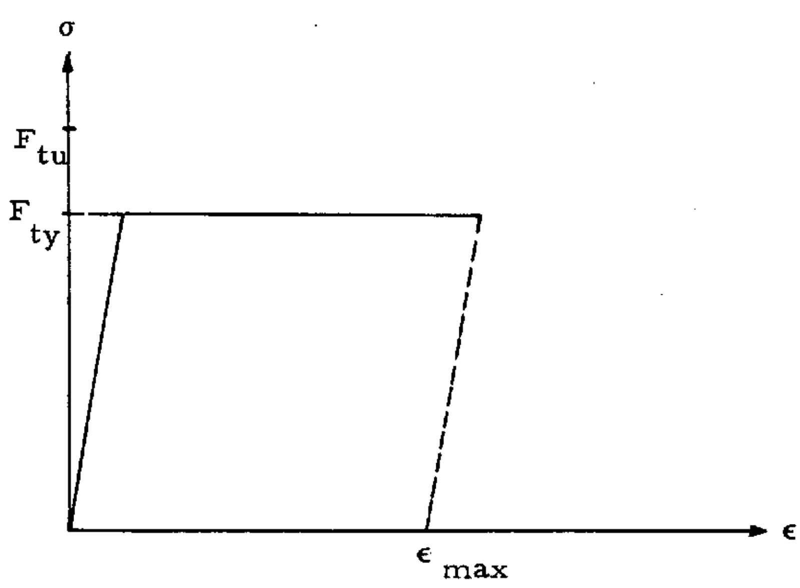


Fig. 1-5 - Idealized Stress-Strain Diagram

(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

$$m_{Fu} = (F.S.)\, m_F \tag{1.5}$$

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

$$m_{Fu} = Z_F\, F_{ty} \tag{1.6}$$

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

$$(F.S.) = (F.S.)_1 \cdot (F.S.)_2 \cdot (F.S.)_3 \cdot\cdot\cdot (F.S.)_n. \tag{1.7}$$

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

$$\sigma_r = -\frac{(b/r)^2 + 1}{(b/a)^2 - 1}\, p \ , \tag{2.1}$$

$$\sigma_x = \frac{1}{(b/a)^2 - 1}\, p \ , \tag{2.2}$$

$$\sigma_\varphi = \frac{(b/r)^2 + 1}{(b/a)^2 - 1}\, p \ . \tag{2.3}$$

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

$$\sigma_e = \frac{1}{\sqrt{2}} \sqrt{(\sigma_r - \sigma_\varphi)^2 + (\sigma_\varphi - \sigma_x)^2 + (\sigma_x - \sigma_r)^2} = \frac{\sqrt{3}\,(b/r)^2}{(b/a)^2 - 1}\, p \ , \tag{2.4}$$

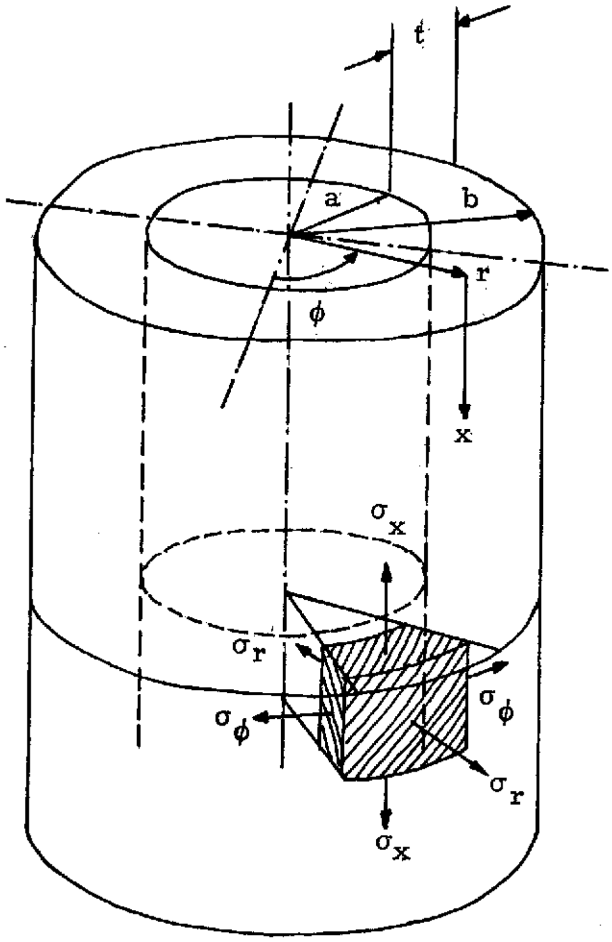


Fig. 2-1 - Coordinate System for a Thick-Walled Tube with a maximum at r = a,

$$\sigma_{e_{max}} = \frac{\sqrt{3}\,(b/a)^2}{(b/a)^2 - 1}\,p\,. \tag{2.5}$$

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

$$\sigma_e = \frac{\sqrt{3\,(b/r)^4 + (1 - \sigma_x'/\sigma_x)^2}}{(b/a)^2 - 1}\,p\,. \tag{2.6}$$

The average equivalent stress is generally

$$\sigma_{e_{av}} = \frac{(b/a)^2 + 1}{(b/a)^2 - 1}\,\frac{p}{2}\,. \tag{2.7}$$

When the equivalent stress σ_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

$$\bar{\sigma}_r = -\frac{2}{\sqrt{3}}\,F_{ty}\,\ln\frac{b}{r} \tag{2.8}$$

$$\bar{\sigma}_x = \frac{2}{\sqrt{3}}\,F_{ty}\left(\frac{1}{2} - \ln\frac{b}{r}\right) \tag{2.9}$$

$$\bar{\sigma}_\varphi = \frac{2}{\sqrt{3}}\,F_{ty}\left(1 - \ln\frac{b}{r}\right) \tag{2.10}$$

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

$$\bar{\sigma}_e = \frac{2}{\sqrt{3}}\,\ln\left(\frac{b}{r}\right) \tag{2.11}$$

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

$$\frac{b}{a} = 1.2 \tag{2.12}$$

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

$$\frac{p}{\sigma_{e_{max}}} \approx \frac{p}{\sigma_{e_{av}}} \approx \frac{p}{\bar{\sigma}_e} \quad \text{if} \quad \frac{b}{a} \leq 1.2 \tag{2.13}$$

where

$$\frac{p}{\sigma_{e_{max}}} = \frac{\left(\frac{b}{r}\right)^2 - 1}{\sqrt{3}\left(\frac{b}{a}\right)^2} , \tag{2.14}$$

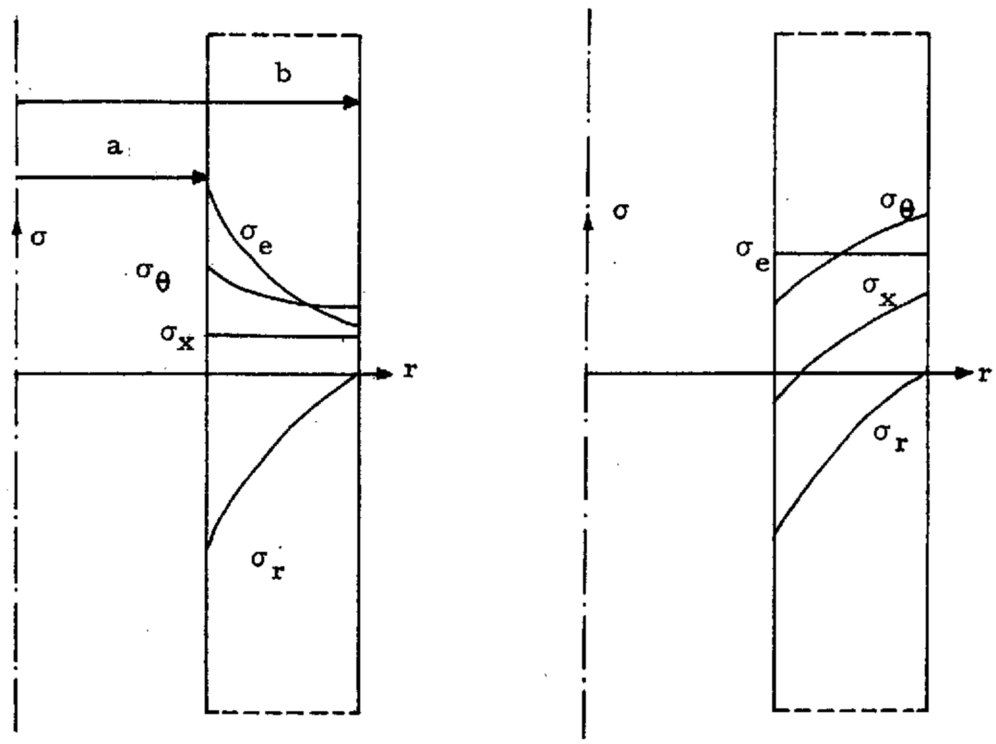
$$\frac{p}{\bar{\sigma}_e} = \frac{2}{\sqrt{3}} \ln\left(\frac{b}{r}\right) , \tag{2.15}$$

and

$$\frac{p}{\sigma_{e_{av}}} = \frac{2t}{2a + t} \tag{2.16}$$

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

$$p = \frac{2t}{2a + t} \frac{K}{(F.S.)} \tag{2.17}$$



(a) Elastic (b) Plastic

Fig. 2-2 - States of Stress in a Thick-Walled Cylindrical Shell Under Internal Pressure which can be solved for the wall thickness, giving

$$t = \frac{pa}{\frac{K}{(F.S.)} - \frac{p}{2}} . \tag{2.18}$$

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 Δt and a factor ψ for weakening by welds the wall thickness t in Eq. (2.12) is replaced by

$$\overline{t} = (t - \Delta t)\,\psi \tag{2.19}$$

which gives the formula for the thickness as

$$t = \frac{pa}{\left(\frac{K}{(F.S.)} - \frac{p}{2}\right)\psi} + \Delta t \tag{2.20}$$

This is the formula used in Ref. 22.

In Ref. 21 this equation has been modified by taking

$$t = \frac{1.1\,p\,a}{\frac{K}{(F.S.)} - .4\,p} \tag{2.21}$$

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

$$t = \frac{p\,a}{\left[\frac{K}{(F.S.)}\right]\psi} + 2\,\Delta t \tag{2.22}$$

The simplest formula, based only on circumferential stress, is

$$t = \frac{pa}{\frac{K}{(F.S.)}} \qquad (2.23)$$

The weakening factor ψ has been used in the order of

$$0.70 \leq \psi \leq 1.00 \qquad (2.24)$$

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

$$\sigma_n = \sigma_x \sin^2\gamma + \sigma_\varphi \cos^2\gamma \qquad (2.25)$$

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

$$\psi = \frac{2\psi'}{1 + \cos^2\gamma} \qquad (2.26)$$

The factor ψ' 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.

<u>Creep</u>: 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

$$\dot{\epsilon}_e = B\,\sigma_e^n \qquad (2.27)$$

where ϵ_e is the equivalent strain defined by

$$\epsilon_e^2 = \frac{\sqrt{2}}{3}\sqrt{(\epsilon_r - \epsilon_\varphi)^2 + (\epsilon_\varphi - \sigma_x)^2 + (\epsilon_x - \epsilon_r)^2} \tag{2.28}$$

and the creep constant B is

$$B = \beta e^{\alpha T} \tag{2.29}$$

where α and β 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

$$\sigma_r = -\frac{\left(\frac{b}{r}\right)^{2/n} - 1}{\left(\frac{b}{a}\right)^{2/n} - 1}\, p \tag{2.30}$$

$$\sigma_x = \frac{\frac{(1-n)}{n}\left(\frac{b}{r}\right)^{2/n} + 1}{\left(\frac{b}{a}\right)^{2/n} - 1}\, p \tag{2.31}$$

$$\sigma_\varphi = \frac{\frac{(2-n)}{n}\left(\frac{b}{r}\right)^{2/n} + 1}{\left(\frac{b}{a}\right)^{2/n} - 1}\, p \tag{2.32}$$

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),

$$\sigma_{e_{max}} = \frac{\frac{\sqrt{3}}{n}\left(\frac{b}{a}\right)^{2/n}}{\left(\frac{b}{a}\right)^{2/n} - 1} \; p \tag{2.33}$$

To design a tube for a given lifetime until creep rupture occurs, the ultimate equivalent strength is computed by

$$\sigma_{u_{max}} = (F.S.)\, \sigma_{e_{max}} \tag{2.34}$$

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

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

$$P = c_1 T \left[\log (t_{rupt}) + c_2\right] \tag{2.35}$$

which can be solved for t_{rupt} giving

$$t_{rupt} = \text{antilog}\left(\frac{P}{c_1 T} - c_2\right). \tag{2.36}$$

This excursion into creep analysis methods may suffice.

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

$$t = \frac{(\Delta p) a}{\frac{K_F}{(F.S.)} - \frac{\Delta p}{2}} + c \tag{2.37}$$

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

$$\Delta p = p_{max} - p_{min} \tag{2.38}$$

The maximum pressure p_{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

$$\sigma_x = \frac{M_1}{\pi R^2 t} \cos\varphi \tag{2.39}$$

and

$$\tau_{x\varphi} = \frac{S_1}{\pi R t} \sin\varphi \tag{2.40}$$

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

$$n_x = \frac{M_1}{\pi R^2} , \tag{2.41}$$

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

$$p = \frac{t}{a}\frac{K_T}{(F.S.)_T} \tag{2.42}$$

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

$$P_B = \pi a^2 p \tag{2.43}$$

has to be equal to

$$P_B = \frac{2\pi r_B}{s}\frac{\pi d_B^2}{4}\frac{K_B}{(F.S.)_B} \tag{2.44}$$

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),

$$P_B = \pi a^2 \left[\frac{t}{a}\frac{K_T}{(F.S.)_T}\right] \tag{2.45}$$

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

$$\pi a^2 \left[\frac{t}{a}\frac{K_T}{(F.S.)_T}\right] = \frac{2\pi r_B}{s}\frac{\pi d_B^2}{4}\frac{K_B}{(F.S.)_B}\ , \tag{2.46}$$

or, after rearranging,

$$\frac{d_B}{t} = \frac{2}{\pi}\frac{s}{d_B}\frac{a}{r_B}\frac{\dfrac{K_T}{(F.S.)_T}}{\dfrac{K_B}{(F.S.)_B}}\ . \tag{2.47}$$

An estimate for the ratio d_B/t is arrived at by assuming the following ratios

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

$$\frac{r_B}{a} \approx 1.1 \, , \tag{2.49}$$

$$\frac{s}{d_B} \approx 2.5 \, , \tag{2.50}$$

so that

$$\frac{d_B}{t} \approx \frac{2}{\pi} (2.5) \frac{1}{(1.1)} \frac{1}{1.5} \approx 1.0 \, . \tag{2.51}$$

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

$$\eta_0 = s/d_B \tag{2.52}$$

$$\eta_1 = e_1/d_B \tag{2.53}$$

and

$$\eta_2 = e_2/d_B \tag{2.54}$$

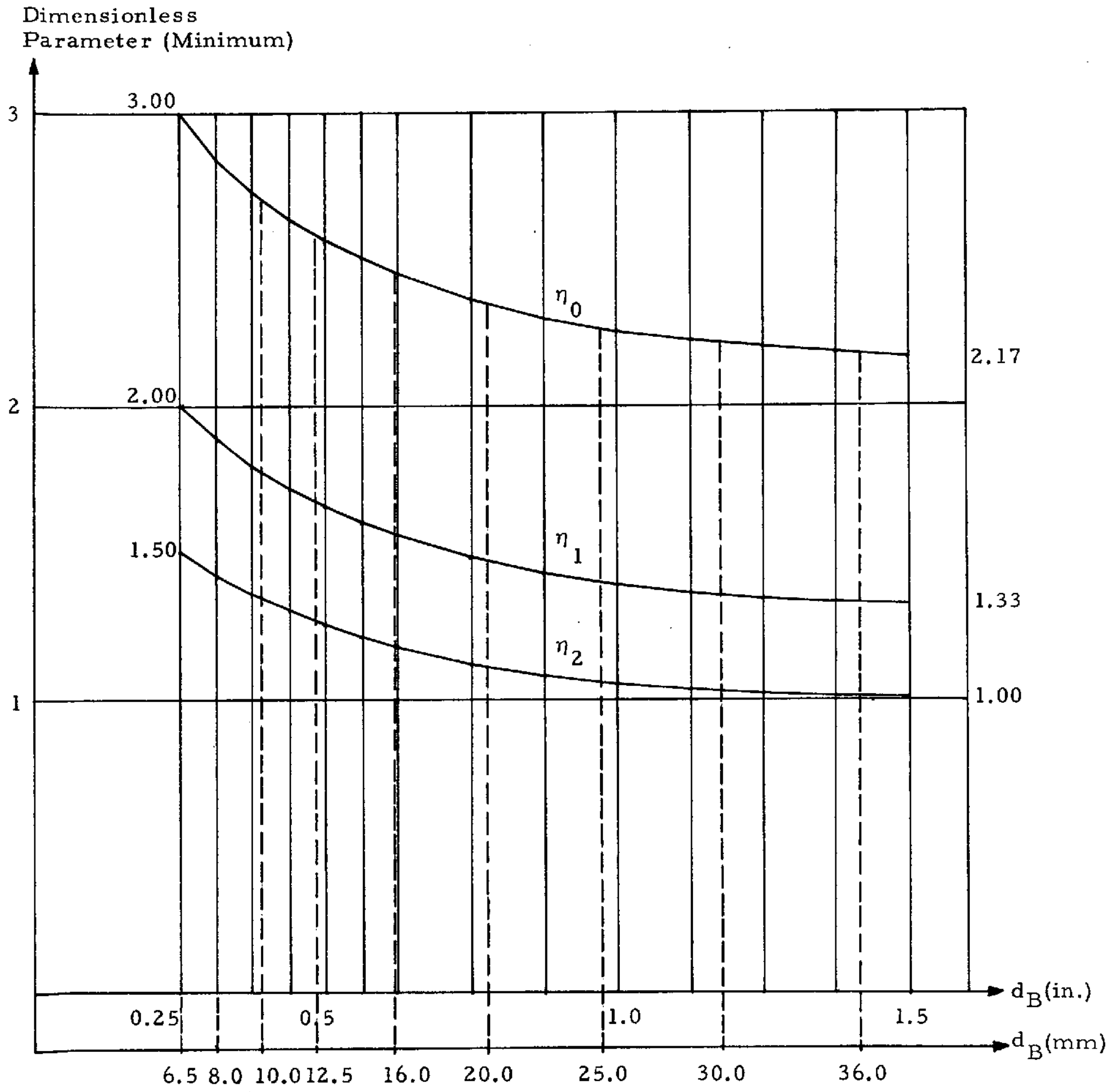


Figure 2-3 - Design Parameters for Open-End Wrenching

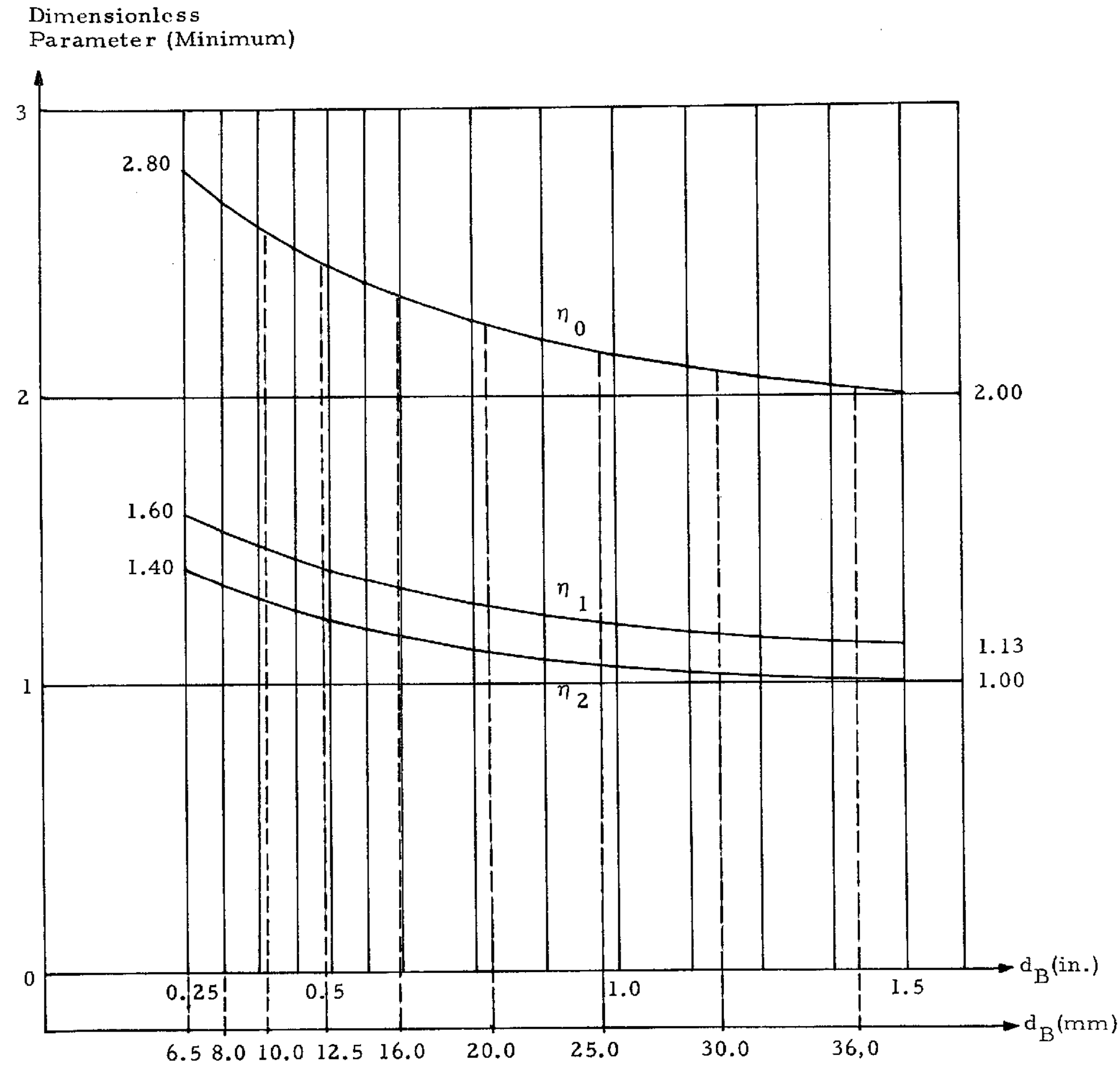


Fig. 2-4 - Design Parameter for Socket Wrenching

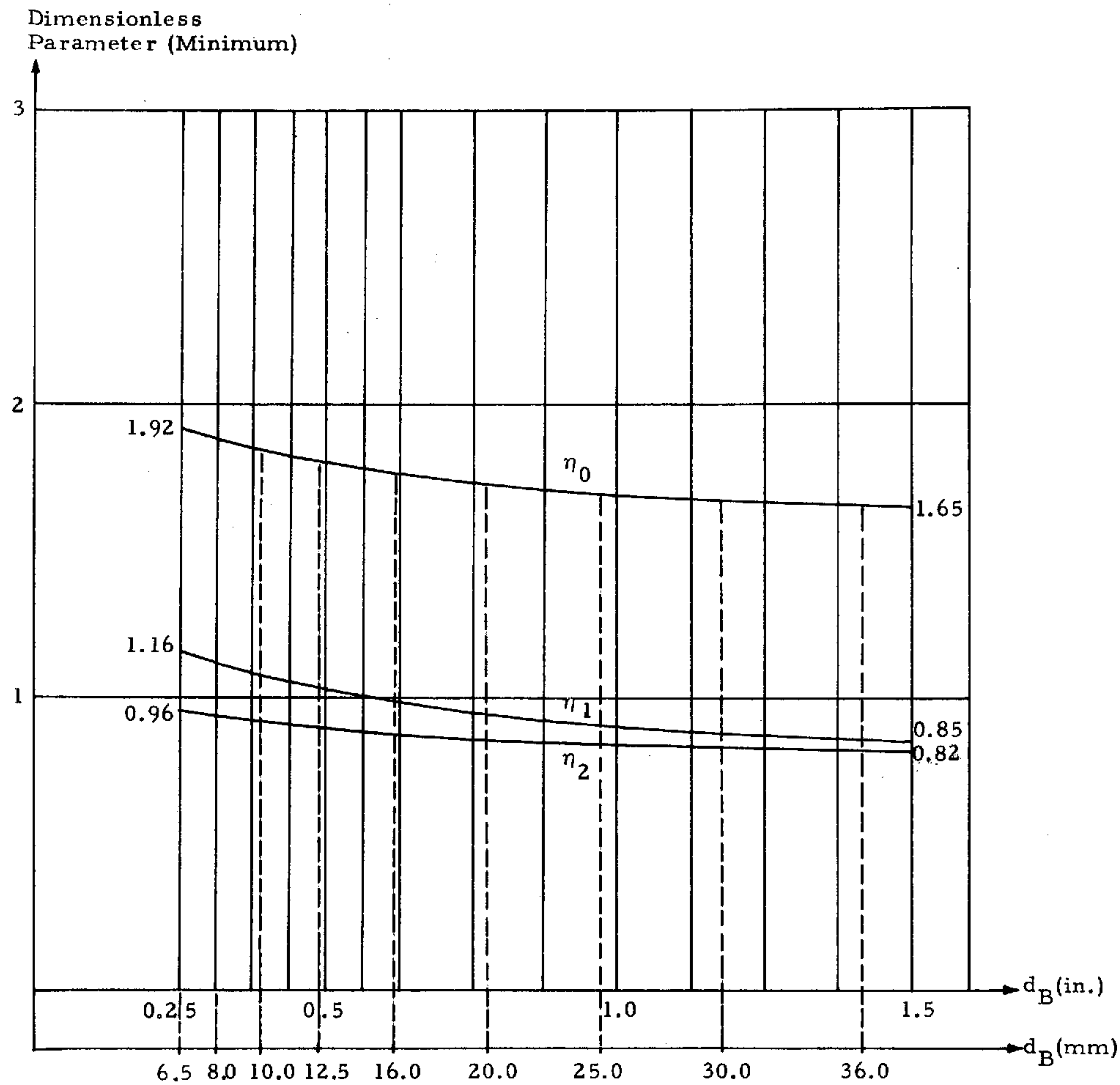


Fig. 2-5 - Design Parameters for Internal Wrenching

Table 2-1

BOLT TABLE FOR OPEN-END WRENCHING (REF. 18)

Size	d_B(in.)	η_o	η_1	η_2	A_{oB} (in^2)
1	0.2500	3.00	2.00	1.50	0.03182
2	0.3125	2.60	1.80	1.40	0.05243
3	0.3750	2.67	1.67	1.33	0.07749
4	0.4375	2.57	1.57	1.29	0.10631
5	0.5000	2.50	1.62	1.24	0.14190
6	0.5625	2.45	1.56	1.22	0.18194
7	0.6250	2.40	1.50	1.20	0.22600
8	0.7500	2.33	1.49	1.08	0.33446
9	0.8750	2.35	1.43	1.07	0.46173
10	1.0000	2.25	1.37	1.06	0.60574
11	1.1250	2.22	1.33	1.00	0.76327
12	1.2500	2.25	1.40	1.00	0.92905
13	1.3750	2.23	1.36	1.00	1.15488
14	1.5000	2.17	1.33	1.00	1.40525

Legend:

Size = size number of the bolt

d_B = nominal diameter of the bolt

$$\left.\begin{aligned} \eta_0 &= \frac{s}{d_B} \\ \eta_1 &= \frac{e_1}{d_B} \\ \eta_2 &= \frac{e_2}{d_B} \end{aligned}\right\} \text{spacing parameter (dimensionless)}$$

A_{oB} = stress area of one bolt

d_{hole} = d_B + 0.005 in.

Table 2-2

BOLT TABLE FOR SOCKET WRENCHING (REF. 18)

Size	d_B(in.)	η_0	η_1	η_2
1	0.2500	2.76	1.60	1.40
2	0.3125	2.53	1.50	1.28
3	0.3750	2.37	1.33	1.20
4	0.4375	2.26	1.25	1.14
5	0.5000	2.18	1.20	1.10
6	0.5625	2.20	1.22	1.11
7	0.6250	2.22	1.25	1.12
8	0.7500	2.12	1.17	1.07
9	0.8750	2.28	1.31	1.14
10	1.0000	2.19	1.25	1.10
11	1.1250	2.14	1.22	1.07
12	1.2500	2.09	1.18	1.04
13	1.3750	2.00	1.16	1.02
14	1.5000	2.02	1.13	1.00

Legend:

Size = size number of the bolt

d_B = nominal diameter of the bolt

$$\left.\begin{aligned} \eta_0 &= \frac{S}{d_B} \\ \eta_1 &= \frac{e_1}{d_B} \\ \eta_2 &= \frac{e_2}{d_B} \end{aligned}\right\} \text{spacing parameter (dimensionless)}$$

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)

Size	d_B(in.)	η_0	η_1	η_2
1	0.2500	1.92	1.16	0.96
2	0.3125	1.86	1.09	0.93
3	0.3750	1.79	1.04	0.91
4	0.4375	1.80	1.03	0.91
5	0.5000	1.78	1.00	0.90
6	0.5625	1.76	0.98	0.89
7	0.6250	1.75	0.96	0.88
8	0.7500	1.68	0.91	0.84
9	0.8750	1.69	0.90	0.85
10	1.0000	1.67	0.89	0.84
11	1.1250	1.86	0.96	0.92
12	1.2500	1.67	0.87	0.83
13	1.3750	1.80	0.93	0.89
14	1.5000	1.65	0.85	0.82

Legend:

Size = size number of the bolt

d_B = nominal diameter of the bolt

$$\left.\begin{aligned} \eta_0 &= \frac{S}{d_B} \\ \eta_1 &= \frac{e_1}{d_B} \\ \eta_2 &= \frac{e_2}{d_B} \end{aligned}\right\} \text{spacing parameter (dimensionless)}$$

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

d_{hole} = d_B + 0.005 in.

are given. The distances e_1 and e_2 refer to a flange with machined spot-faces 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 $d_{hole} = d_B + 0.005$ 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 $d_{spot} = 2e_1$, where e_1 is given in the bolt tables. A fillet radius of $r_{spot} = 0.062$ 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

Table 2-4

METRIC BOLT TABLE FOR OPEN-END WRENCHING (REF. 18)

i_{size}	d_B (mm)	d_B (in.)	η_0	η_1	η_2	A_{oB} (mm^2)	A_{oB} ($in.^2$)
1	6.300	0.2480	3.00	2.00	1.50	22.276	0.035
2	8.000	0.3150	2.85	1.90	1.43	36.126	0.055
3	10.000	0.3937	2.70	1.80	1.85	57.261	0.089
4	12.500	0.4921	2.59	1.68	1.28	91.524	0.142
5	16.000	0.6299	2.45	1.57	1.18	155.070	0.240
6	20.000	0.7874	2.35	1.48	1.12	242.297	0.375
7	25.000	0.9843	2.28	1.40	1.07	382.801	0.593
8	30.000	1.1811	2.23	1.35	1.02	555.296	0.861
9	36.000	1.4173	2.19	1.33	1.01	809.423	1.255

Table 2-5

METRIC BOLT TABLE FOR SOCKET WRENCHING (REF. 18)

i_{size}	d_B (mm)	η_0	η_1	η_2
1	6.300	2.80	1.60	1.40
2	8.00	2.69	1.53	1.36
3	10.000	2.57	1.48	1.29
4	12.500	2.46	1.41	1.23
5	16.00	2.34	1.33	1.16
6	20.000	2.25	1.27	1.11
7	25.000	2.15	1.22	1.07
8	30.000	2.08	1.17	1.03
9	36.000	2.02	1.14	1.01

Table 2-6
METRIC BOLT TABLE FOR INTERNAL WRENCHING (REF. 18)

i_{size}	d_B (mm)	η_0	η_1	η_2
1	6.300	1.92	1.16	0.96
2	8.00	1.88	1.12	0.94
3	10.000	1.85	1.08	0.92
4	12.500	1.81	1.03	0.90
5	16.000	1.77	0.99	0.87
6	20.000	1.73	0.94	0.85
7	25.000	1.70	0.91	0.84
8	30.000	1.68	0.88	0.83
9	36.000	1.66	0.86	0.82

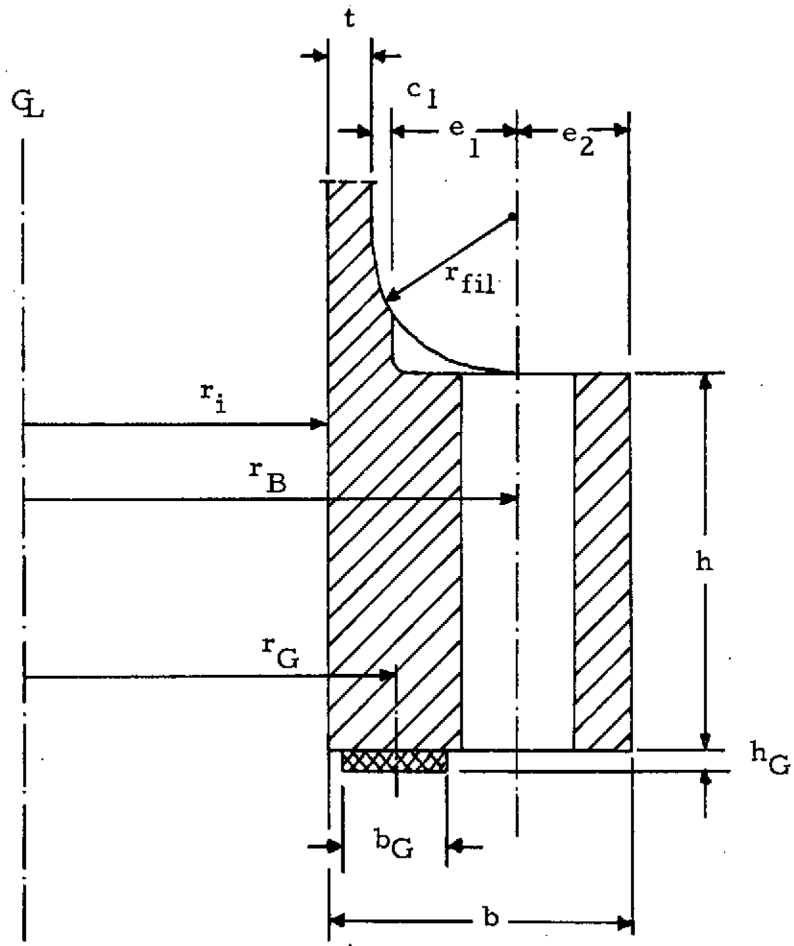


Fig. 2-6 - Low Profile Flange with Machined Spot Faces

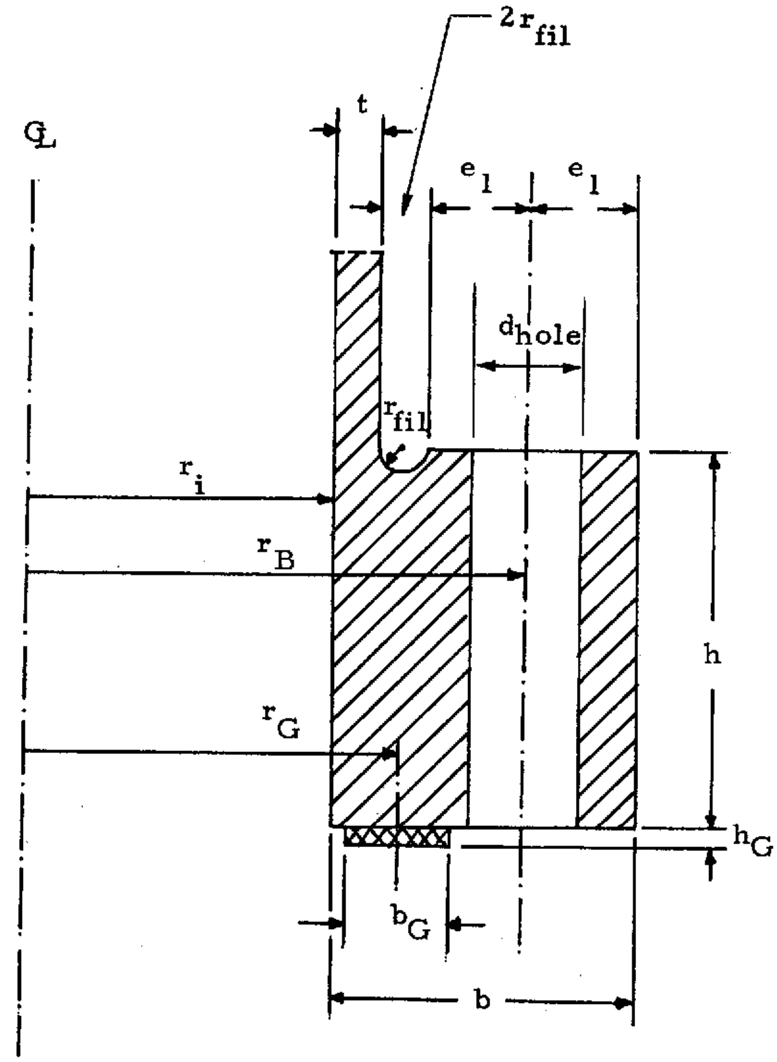


Fig. 2-7 - Low Profile Flange with Machined Groove introduced by the flange, is

$$\frac{A_1}{A_2} = e^{2\pi} = e^{\rho L/r_o} , \tag{2.55}$$

where

$$\rho = \sqrt[4]{3(1-\nu^2)\, r_o^2/t^2} . \tag{2.56}$$

The radius of the cylinder middle surface, r_o, is

$$r_o = a + t/2 . \tag{2.57}$$

From the logarithmic decrement

$$\rho L/r_o = 2\pi \tag{2.58}$$

the wavelength is

$$L = 2\pi\, r_o/\rho \tag{2.59}$$

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,

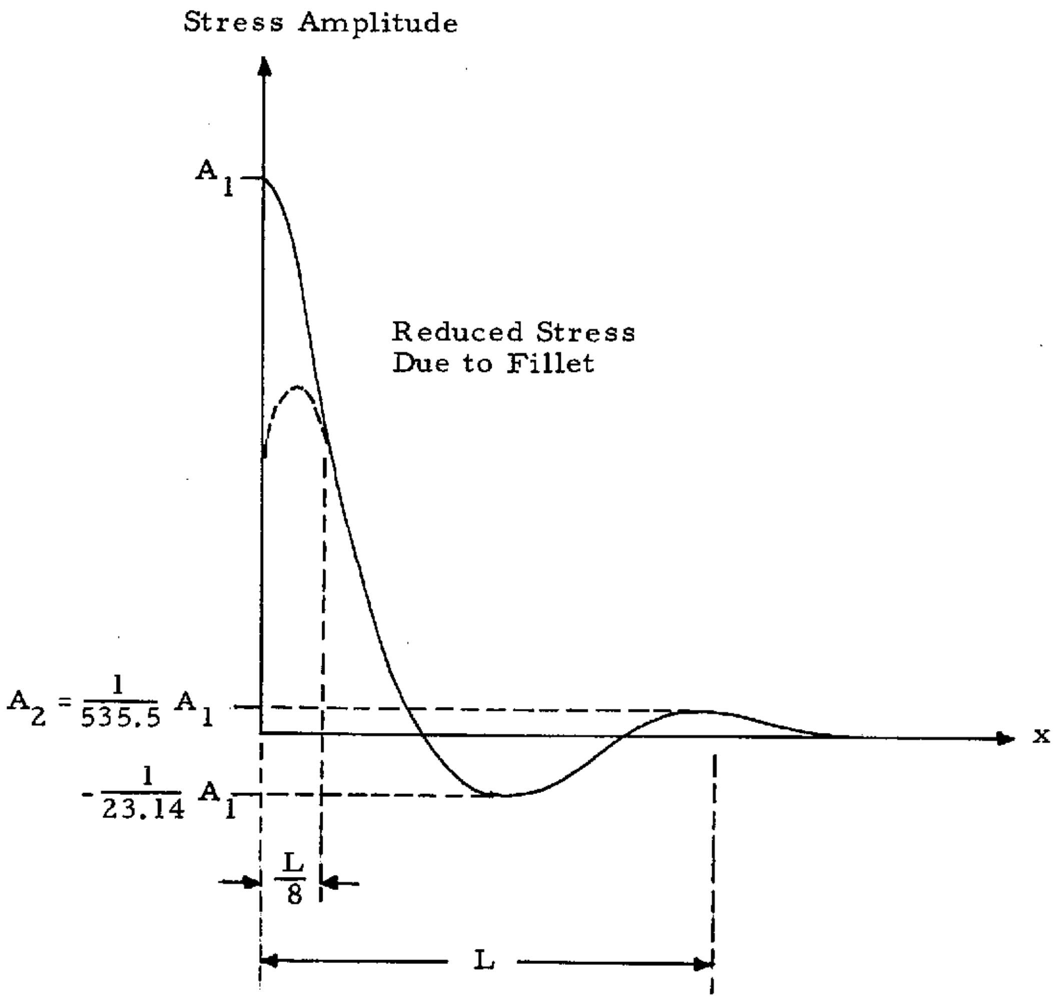


Fig. 2-8 - Relation of Fillet Radius to Shell Stresses

Table 2-7

FILLET RADIUS FOR FLANGE WITH MACHINED SPOT FACES

t (in.)	t (mm)	r_{fil} (in.)	r_{fil} (mm)
≥ 0.2	5	0.3750	10
< 0.2	5	0.3125	8
< 0.15	3	0.2500	6
< 0.10	2.5	0.1875	4
< 0.05	1	0.1250	3

Legend:

t = tube thickness

r_{fil} = fillet radius

Table 2-8

GROOVE RADIUS FOR FLANGE WITH MACHINED GROOVE

t (in.)	t (mm)	r_{fil} (in.)	r_{fil} (mm)
≥ 0.2	5	0.1250	3
< 0.2	5	0.1000	2.5
< 0.15	3	0.0750	2.
< 0.10	2.5	0.0500	1.
< 0.05	1	0.0250	0.5

Legend:

t = tube thickness

r_{fil} = groove radius

Fig. 2-6, as

$$r_B = r_i + t + c_1 + e_1 \tag{2.60}$$

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

$$r_B = r_i + t + 2 r_{fil} + e_1 \, . \tag{2.61}$$

The flange width is then

$$b = r_B + e_2 - r_i \, . \tag{2.62}$$

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

$$P_G^{(1)} = 2\pi r_G S_G^{(1)} \tag{2.63}$$

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

$$P_G^{(2)} = 2\pi r_G S_G^{(2)} \tag{2.64}$$

The total required flange force P_F is the sum of the force caused by internal pressure, P_p,

$$P_p = \pi r_G^2 p \qquad (2.65)$$

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

$$P_F^{(2)} = \pi r_G^2 p + 2\pi r_G S_G^{(2)} \quad . \qquad (2.66)$$

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

$$P_G^{(0)} = \alpha P_G^{(1)} + (1 - \alpha) \sqrt{P_F^{(2)} P_G^{(1)}} \qquad (2.67)$$

The coefficient α 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.

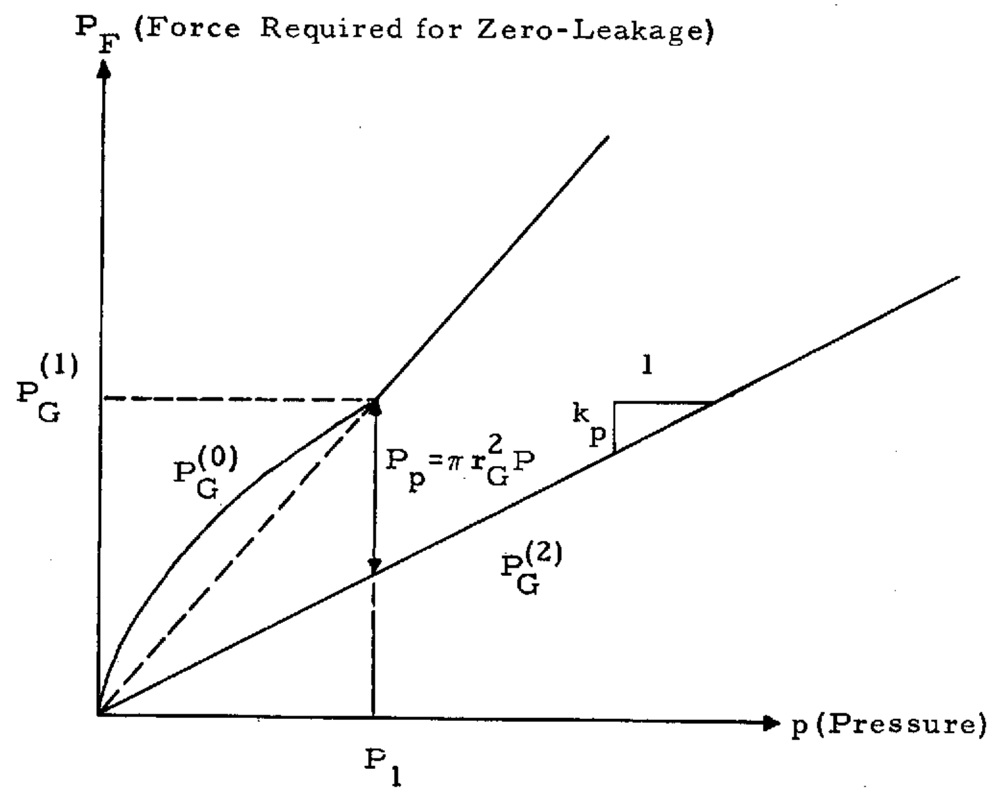


Fig. 2-9 - Required Gasket Forces

For flat gaskets the line loads are

$$S_G^{(1)} = b_{eff}^{(1)} \sigma_G \tag{2.68}$$

and

$$S_G^{(2)} = b_{eff}^{(2)} k_p \, p \tag{2.69}$$

For other than flat gaskets the line load $S_G^{(1)}$ is given directly while

$$S_G^{(2)} = K_p \, p \tag{2.70}$$

The quantity σ_G has the units of stress and k_p is dimensionless. The quantity K_1 has the dimensions of a length. The effective widths $b_{eff}^{(1)}$ and $b_{eff}^{(2)}$ 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

$$h^3 = \frac{K_e}{\sigma^m} \tag{2.71}$$

where h^3 is the "conductance parameter", K_e is a constant and σ 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

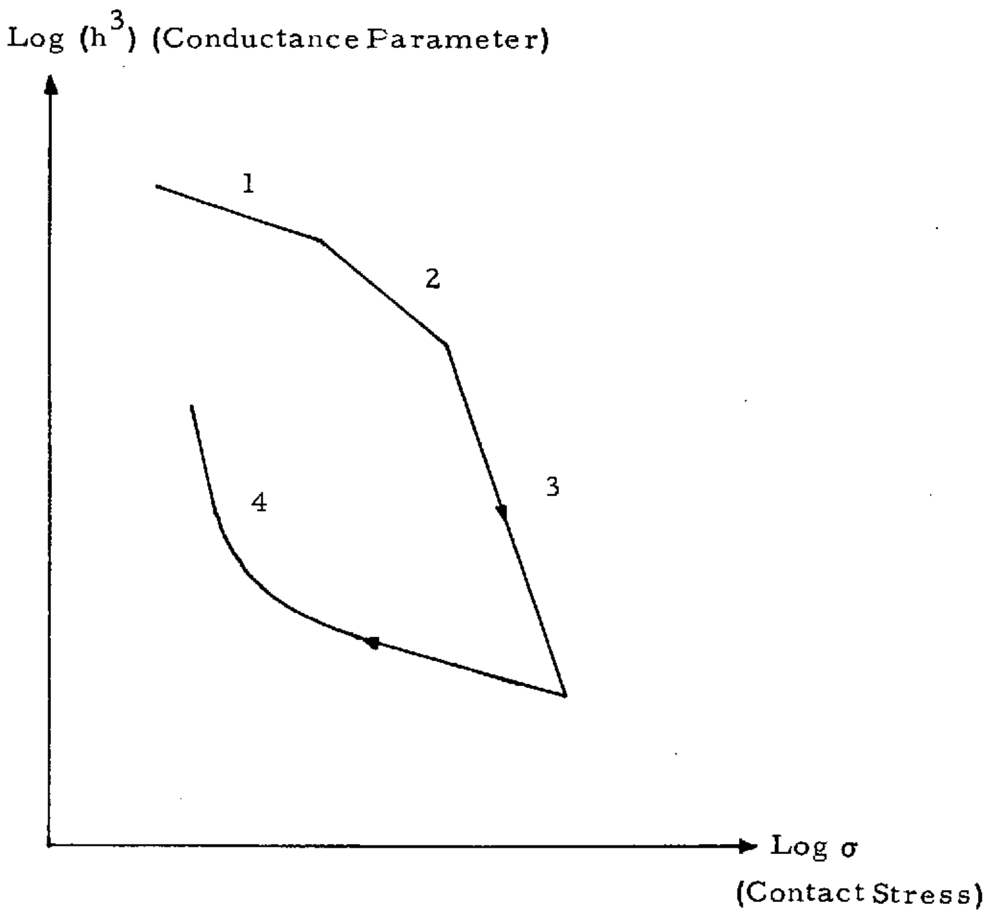


Fig. 2-10 - Relation Between Conductance Parameter and Interface Contact Stress (Ref. 31)

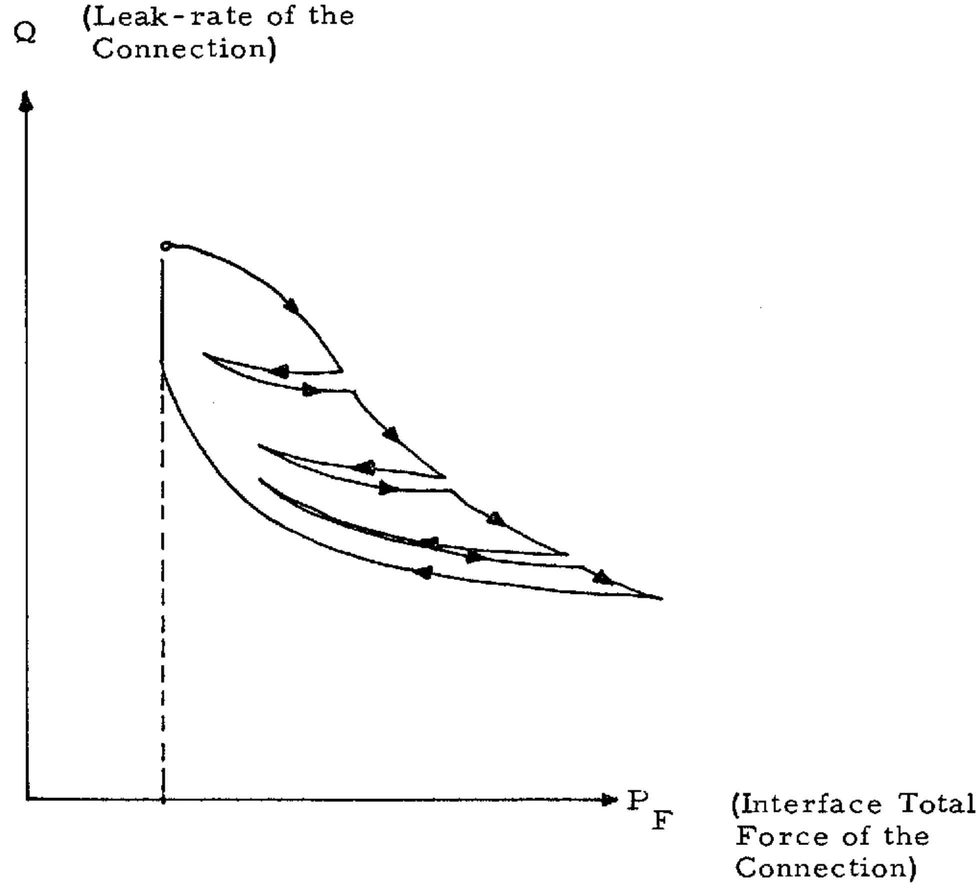


Fig.2-11 - Hysteresis Effect (Ref.31)

(weight) leak rate, is the basis for computing the required conductance parameter. For laminar, isothermal, compressible (gas) flow the volume leak rate is

$$Q = \frac{w\,(p_2^2 - p_1^2)}{24\mu\, p_o\, L}\, h^3 \qquad (2.72)$$

where w is the width and L is the length of the leak path, μ 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

$$W = \frac{pm}{RT}\, Q \qquad (2.73)$$

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

$$W = \frac{\rho\, w\,(p_2 - p_1)}{12\mu\, L}\, h^3 \qquad (2.74)$$

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

$$\left.\begin{aligned} w &\approx 2\pi r_G \\ L &\approx b_G \,. \end{aligned}\right\} \qquad (2.75)$$

and

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 coversion 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

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

This leads to

$$2\pi \; b_{eff}^{(1)} r_G \, \sigma_G = \pi \; r_G^2 \, P + 2\pi \; b_{eff}^{(2)} r_G \, k_p \, P \qquad (2.77)$$

which can be solved for b_G after substituting

$$b_{eff}^{(1)} = \gamma_1 \, b_G \, , \qquad (2.78)$$

$$b_{eff}^{(2)} = \gamma_2 \, b_G \, , \qquad (2.79)$$

resulting in

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

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 σ_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.

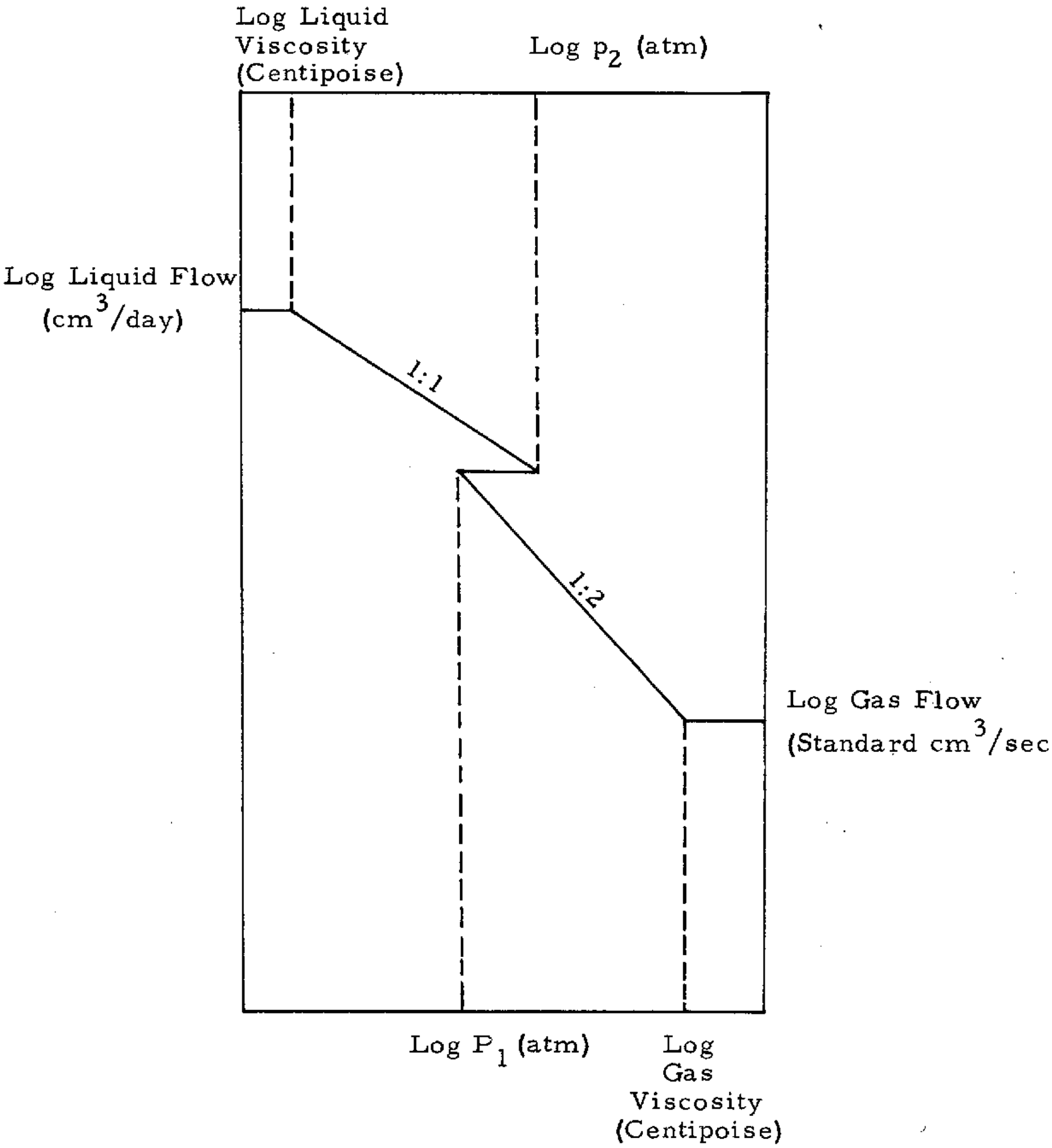


Fig. 2-12 - Fluid Flow Conversion Graph (Ref. 32)

Table 2-9
COMPARISON OF GASKET WIDTHS

Pressure p (psi)	Gasket Thickness, h_G		
	1/8	1/16	1/32
100	$0.0102\ r_G$	$0.0103\ r_G$	$0.0104\ r_G$
1000	$0.125\ r_G$	$0.138\ r_G$	$0.154\ r_G$

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

$$b_{G_{avail}} = r_B - \frac{d_{hole}}{2} - r_i - 2\Delta r \tag{2.81}$$

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

$$P_F^{(2)} = \pi r_G^2 p + 2\pi b_{G_{avail}} \gamma_2 r_G k_p p \text{ (F.S.)} \tag{2.82}$$

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

$$\sigma_G = \frac{P_F^{(2)}}{2\pi b_{G_{avail}} \gamma_1 r_G} \tag{2.83}$$

If σ_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 σ_G as computed with Eq. (2.83) is less than 3.7 ksi, the initial flange load should be $P_F^{(2)} = 2\pi b_{G_{avail}} \gamma_1 r_G$ (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

$$P_B = (F.S.)\, P_p \tag{2.84}$$

where

$$P_p = \pi\, r_s^2\, p \tag{2.85}$$

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

$$P_B = (F.S.)\, P_p \left(1 + \frac{e}{e_2}\right) . \tag{2.86}$$

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

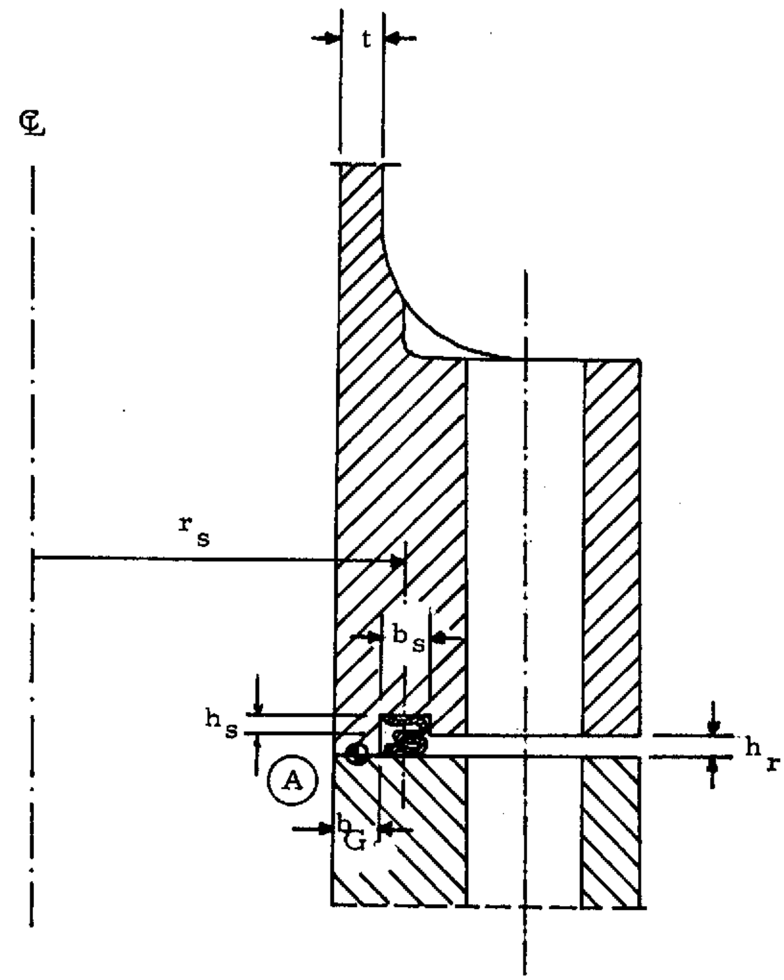


Fig.2-13 - Cantilever Flange with Pressure Energized Seal

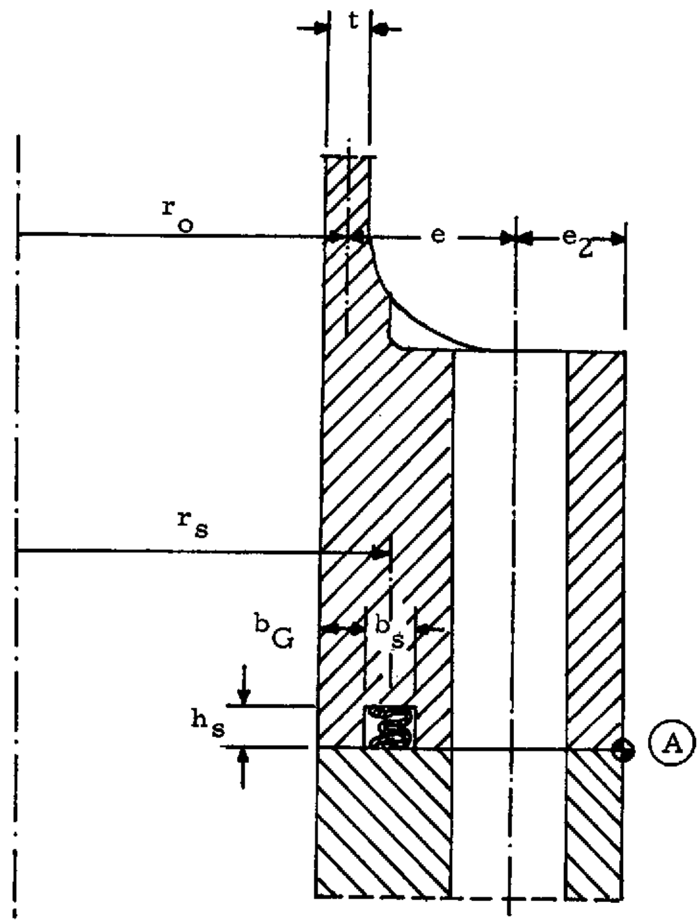


Fig. 2-14 - Metal-to-Metal Flange with Pressure Energized Seal determined from the roll angle and corresponding differential axial displacement at the outer edge of the flange cross section. If the recess is not high enough, this will result in the same situation as for the metal-to-metal flange and 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

$$n_{B1} = \frac{P_B}{F_{ty}^{(B)} A_{oB}} , \tag{2.87}$$

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

$$n_{B2} = \frac{P_{B_{(burst)}}}{F_{tu}^{(B)} \Delta_{oB}} \tag{2.88}$$

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

$$s = \frac{2\pi\ r_B}{n_B} \tag{2.89}$$

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 $s = 8\ d_B$, 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 η_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

$$m_{Fu} = \frac{(F.S.)\ P_B\ e}{2\pi\ r_o} \tag{2.90}$$

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,

$$e = r_B - r_G \tag{2.91}$$

The radius r_o is the one of the middle surface of the tube wall,

$$r_o = r_i + t/2 \tag{2.92}$$

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,

$$\overline{b} = b - d_{hole}\sqrt{\frac{d_{hole}}{s}} \tag{2.93}$$

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

$$Z_T = \zeta_1 \frac{t^2 - t_N^2}{4} + \zeta_2 (t - t_N) \frac{h}{2} \tag{2.94}$$

where ζ_1 and ζ_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

$$\zeta_1 = .8, \quad \zeta_2 = .18 \tag{2.95}$$

In Section 3 the computation of ζ_1 and ζ_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

$$S_F = \frac{\bar{b}\, h^2}{6\, r_o} \tag{2.96}$$

The design formula for the flange is now derived by requiring

$$m_{Fu} = F_{ty}^{(F)} \left[S_F + \frac{F_{ty}^{(T)}}{F_{ty}^{(F)}} Z_T \right] \tag{2.97}$$

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

$$Ah^2 + Bh + C = 0 , \tag{2.98}$$

where

$$A = F_{ty}^{(F)}\, \bar{b}/6\, r_o , \tag{2.99}$$

$$B = F_{ty}^{(F)}\, \zeta_2 (t - t_N)/2, \tag{2.100}$$

$$C = F_{ty}^{(F)}\, \zeta_1 (t^2 - t_N^2)/4 - m_{Fu}. \tag{2.101}$$

The solution for h is

$$h = \frac{\sqrt{B^2 - 4AC} - B}{2A} \tag{2.102}$$

If the contribution of the plastic hinge in the neck is neglected in the design of the flange, i.e., when ζ_1 and ζ_2 are assumed to be zero, then

$$h = \sqrt{6\, r_o\, m_{Fu}/F_{ty}^{(F)}\, \bar{b}} \quad . \tag{2.103}$$

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,

$$\text{if} \qquad s/h \geq 3 \rightarrow h = s/3 \tag{2.104}$$

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

$$A_w = (b - t)\, h \tag{2.105}$$

and the centroidal radius

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

so that

$$\text{vol} = 2\pi\, r_w\, A_w \tag{2.107}$$

the actual weight is

$$\Delta W = \rho_F\ \text{vol} \tag{2.108}$$

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

No.	Material	E (psi)	ν (-)	ρ (lb/in.^{3})	α (in./in./^{o}F)	F_{ty} (psi)	F_{tu} (psi)
1	Aℓ 6061-T6 @ RT	9.9×10^6	.33	.098	12.5×10^{-6}	35.0×10^3	42.2×10^3
2	Aℓ 6061-T6 @ 200^{o}F	9.9×10^6	.33	.098	12.5×10^{-6}	32.2×10^3	38.1×10^3
3	Aℓ 2024-T3 @ RT	9.9×10^6	.33	.098	12.5×10^{-6}	50.0×10^3	62.0×10^3
4	Aℓ 2024-T3 @ 200^{o}F	9.9×10^6	.33	.098	12.5×10^{-6}	47.0×10^3	59.0×10^3
5	347 SS @ RT	28.0×10^6	.30	.288	9.5×10^{-6}	35.0×10^3	90.0×10^3
6	347 SS @ 200^{o}F	28.0×10^6	.30	.288	9.5×10^{-6}	30.0×10^3	76.0×10^3
7	347 SS @ 600^{o}F	28.0×10^6	.30	.288	9.5×10^{-6}	25.0×10^3	68.0×10^3
8	A286 @ RT	28.0×10^6	.30	.288	9.5×10^{-6}	131.0×10^3	200.0×10^3
9	A286 @ 200^{o}F	28.0×10^6	.30	.288	9.5×10^{-6}	128.0×10^3	196.0×10^3
10	A286 @ 600^{o}F	28.0×10^6	.30	.288	9.5×10^{-6}	120.0×10^3	180.0×10^3

Legend: E = elastic modulus
ν = Poisson's ratio
ρ = weight density
α = linear thermal expansion coefficient
F_{ty} = tensile yield strength
F_{tu} = ultimate tensile strength

Table 2-11

PROPERTIES OF GASKET MATERIALS

No.	Material	E (psi)	K_G (psi)	σ_G (psi)	α (in/in/^oF)	μ (-)	h_G (in.)	k_p
1	Asbestos 1/32 in.	44.0×10^3	10.0×10^3	6.5×10^3	1.3×10^{-3}	.5	.03125	3.50
2	Asbestos 1/16 in.	44.0×10^3	10.0×10^3	3.7×10^3	1.3×10^{-3}	.5	.06250	2.75
3	Asbestos 1/8 in.	44.0×10^3	10.0×10^3	1.6×10^3	1.3×10^{-3}	.5	.12500	2.00
4	KEL-F81	180.0×10^3	8.0×10^3	4.0×10^3	3.8×10^{-5}	.12	.06250	3.00
5	CRES 321-A	28.0×10^6	40.0×10^3	18.9×10^3	9.5×10^{-6}	.30	.02500	5.50

Legend:

E = elastic modulus

K_G = yield (crushing) strength

σ_G = minimum seating stress

α = linear thermal expansion coefficient

μ = friction coefficient

h_G = thickness of the gasket

k_p = ratio of required seating stress to given pressure (see Fig. 2-9).

Assumed:

$\gamma_1 = \gamma_2 = 0.5$ for No's 1, 2 and 3

$\gamma_1 = \gamma_2 = 1.0$ for No's 4 and 5

Table 2-12

METRIC PROPERTIES OF METALLIC MATERIALS FOR TUBES, FLANGES AND BOLTS

No.	Material		E (N/mm^2)	ν (-)	ρ (g/mm^3)	α mm/mm/°C	F_{ty} (N/mm^2)	F_{tu} (N/mm^2)
1	Aℓ 6061-T6	RT	68×10^3	.33	$.271 \times 10^{-2}$	22.5×10^{-6}	242	290
2	Aℓ 6061-T6	93C	68×10^3	.33	$.271 \times 10^{-2}$	22.5×10^{-6}	222	263
3	Aℓ 2024-T3	RT	68×10^3	.33	$.271 \times 10^{-2}$	22.5×10^{-6}	345	428
4	Aℓ 2024-T3	93C	68×10^3	.33	$.271 \times 10^{-2}$	22.5×10^{-6}	324	407
5	347 SS	RT	193×10^3	.30	$.798 \times 10^{-2}$	17.1×10^{-6}	242	621
6	347 SS	93C	193×10^3	.30	$.798 \times 10^{-2}$	17.1×10^{-6}	207	528
7	347 SS	315C	193×10^3	.30	$.798 \times 10^{-2}$	17.1×10^{-6}	173	469
8	A286	RT	193×10^3	.30	$.798 \times 10^{-2}$	17.1×10^{-6}	904	1380
9	A286	93C	193×10^3	.30	$.798 \times 10^{-2}$	17.1×10^{-6}	883	1352
10	A286	315C	193×10^3	.30	$.798 \times 10^{-2}$	17.1×10^{-6}	828	1242

Table 2-13

METRIC PROPERTIES OF GASKET MATERIALS

No.	Material	E (N/mm^2)	k_G (N/mm^2)	σ_G (N/mm^2)	α ($mm/mm\,^\circ C$)	μ (-)	h_G (mm)	k_p
1	Asbestos, 0.8 mm	304.0	69.0	45.0	2.3×10^{-3}	.5	.8	3.5
2	Asbestos, 1.6 mm	304.0	69.0	26.0	2.3×10^{-3}	.5	1.6	2.75
3	Asbestos, 3.2 mm	304.0	69.0	11.0	2.3×10^{-3}	.5	3.2	2.00
4	KEL-F81 1.6 mm	1242.0	55.0	28.0	7.0×10^{-5}	.12	1.6	3.00
5	CRES 321-A, 6 mm	193×10^3	276.0	130.0	17.1×10^{-6}	.30	.6	5.50

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 ΔT is characterized by the stress resultants.

$$n_x = \frac{p\, r_o}{2} \tag{3.1}$$

and

$$n_\varphi = p\, r_o \tag{3.2}$$

where n_x, n_φ 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

$$W = \frac{p\, r_o^2}{Et} \left(1 - \frac{\nu}{2}\right) + r_o\, \alpha\, \Delta T \tag{3.3}$$

where α is the linear thermal expansion coefficient and E and ν 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

$$\frac{d^4w}{dx^4} + k^4 w = 0 \tag{3.4}$$

where

$$k^4 = \frac{12\,(1-\nu^2)}{r_o^2 t^2} \tag{3.5}$$

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

$$w = e^{-kx}\,(C_1 \cos kx + C_2 \sin kx) \tag{3.6}$$

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

$$m_x\Big|_{x=0} = -B\,\frac{d^2w}{dx^2}\Big|_{x=0} = m_o \tag{3.7}$$

$$q_x\Big|_{x=0} = -B\,\frac{d^3w}{dx^3}\Big|_{x=0} = -q_o \tag{3.8}$$

where

$$B = \frac{Et^3}{12(1-\nu^2)} \tag{3.9}$$

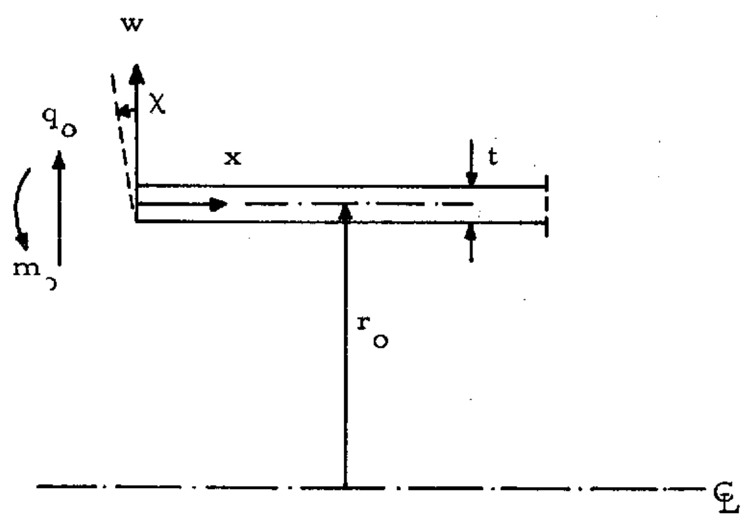


Fig. 3-1 - Edge-Loaded Cylindrical Shell

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

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

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

$$m_x = e^{-kx}\left[m_o\,(\cos kx + \sin kx) - \frac{q_o}{k}\sin kx\right] \tag{3.17}$$

and the meridional shear is

$$q_x = e^{-kx}\left[\,q_o\,(\sin kx - \cos kx) + 2k\,m_o\,\sin kx\right] \tag{3.18}$$

The circumferential bending moment is

$$m\varphi = \nu\, m_x \tag{3.19}$$

and the circumferential stress resultant is

$$n\varphi = \frac{Et}{r_o}\,w \tag{3.20}$$

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

$$\sigma_x = \frac{n_x}{t} + \frac{m_x}{\frac{t^3}{12}}\,z \tag{3.21}$$

the circumferential stress

$$\sigma_\varphi = \frac{n_\varphi}{t} + \frac{m_\varphi}{\frac{t^3}{12}}\,z, \tag{3.22}$$

and the shear stress

$$\tau_{xz} = \frac{q_x}{\left(\frac{t^3}{6}\right)}\left(\frac{t^2}{4} - z^2\right) \tag{3.23}$$

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,

$$t_n = \frac{n_x}{Y_o} , \tag{3.24}$$

is required to carry the axial force, where Yo is the uniaxial tensile yield strength of the material. This leaves for the plastic moment, m_x^p,

$$\sigma_x^p = \frac{m_x^p}{(t^2 - t_n^2)/4} \tag{3.25}$$

and the plastic shear force, q_x^p,

$$\tau_{xz}^p = \frac{q_x^p}{(t - t_n)} \tag{3.26}$$

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

$$\bar{\sigma} = \frac{1}{\sqrt{2}}\sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2} \leq Y_o \tag{3.27}$$

where σ_1, σ_2, σ_3 are the principal stresses.

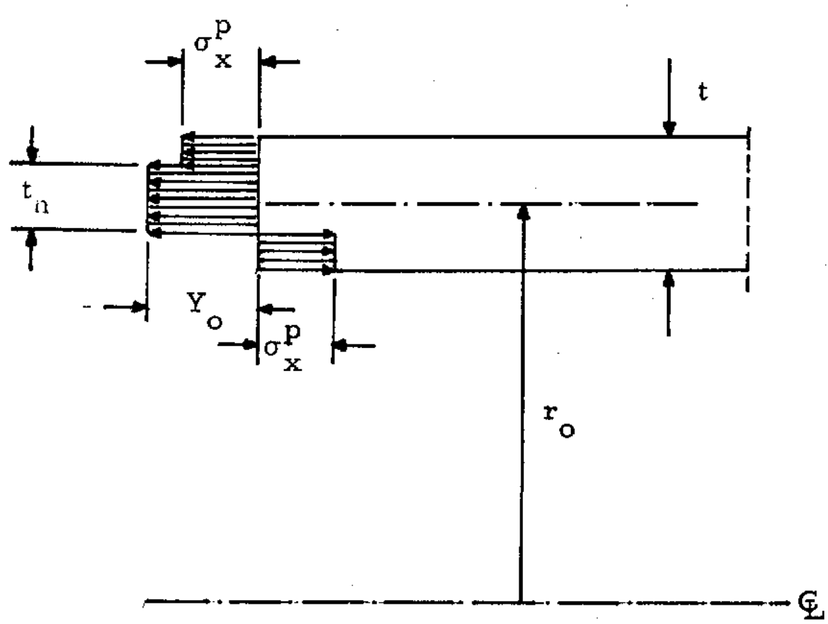


Fig. 3-2 - Assumed Stress Distribution in the Plastic Hinge

The principal stresses for the problem at hand are

$$\sigma_1 = \frac{\sigma_x^p}{2} + \sqrt{\left(\frac{\sigma_x^p}{2}\right)^2 + \left(\tau_{xz}^p\right)^2} \qquad (3.28)$$

$$\sigma_2 = \frac{\sigma_x^p}{2} - \sqrt{\left(\frac{\sigma_x^p}{2}\right)^2 + \left(\tau_{xz}^p\right)^2} \qquad (3.29)$$

$$\sigma_3 = \sigma_\varphi^p \qquad (3.30)$$

The expressions for the principal stresses are simplified by introducing σ_x^p as a reference stress, where

$$\sigma_\varphi^p = \alpha_1 \ \sigma_x^p \qquad (3.31)$$

and

$$\tau_{xz}^p = \alpha_2 \ \sigma_x^p \qquad (3.32)$$

then

$$\sigma_{1,\,2} = \sigma_x^p \left(\tfrac{1}{2} \pm \sqrt{\tfrac{1}{4} + \alpha_2^2}\ \right) \qquad (3.33)$$

$$\sigma_3 = \alpha_1 \ \alpha_x^p \qquad (3.34)$$

and the equivalent stress $\bar{\sigma}$ of Eq. (3.27) is

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

the computation of α_1, α_2 and σ_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 χ of the interface point (A) (Fig. 3-3) caused by an applied moment m_F per unit length,

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

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

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

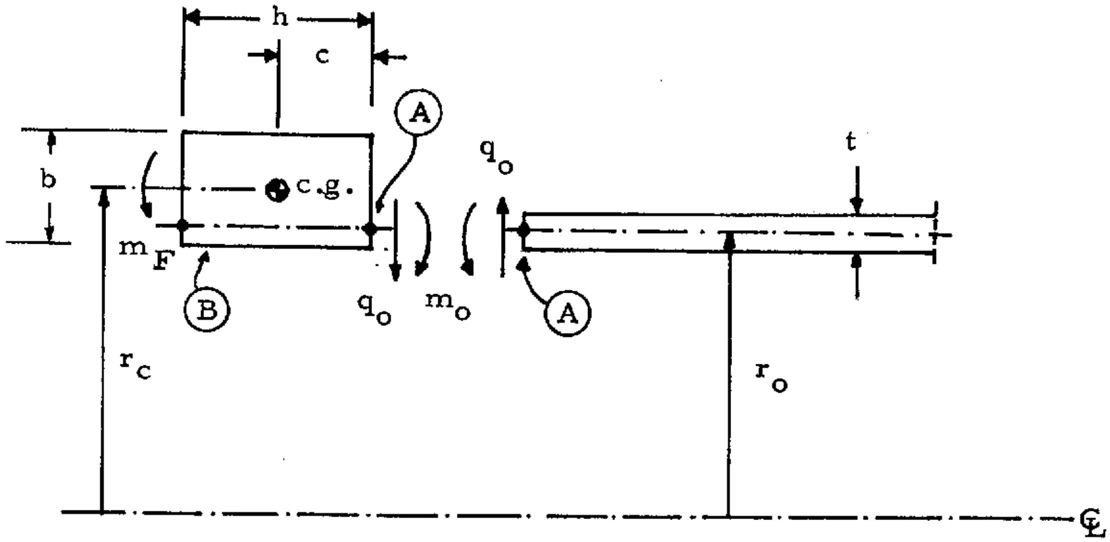


Fig. 3-3 - Ring-Shell Interface

which can be combined as

$$\begin{bmatrix} \frac{1}{2k^2} + \beta\,(c^2 + \frac{I}{A}) & -\frac{1}{2k} + c\beta \\ -\frac{1}{2k} + c\beta & 1 + \beta \end{bmatrix} \begin{bmatrix} q_o \\ m_o \end{bmatrix} = \beta \begin{bmatrix} c \\ 1 \end{bmatrix} m_F \tag{3.38}$$

where

$$\beta = \frac{B\,k\,r_o r_c}{E\,I} \; . \tag{3.39}$$

The determinant of this equation is

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

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

$$q_o = \frac{\beta}{D}\,(c + \frac{1}{2k})\,m_F \;, \tag{3.41}$$

and

$$m_o = \frac{\beta}{D}(\frac{1}{2k^2} + \beta\,\frac{I}{A} + \frac{c}{2k})\,m_F \tag{3.42}$$

The rotation of the cross section is then

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

$$= \frac{\beta}{2k^3 BD}(2k^2 \beta \frac{I}{A} + \frac{1}{2}) m_F \tag{3.43}$$

The rotational spring constant is obtained from Eq. (3.43) by dividing m_F by the rotation χ,

$$c_F = \frac{m_F}{\chi} = \frac{BD}{\beta\left(\frac{\beta}{K}\frac{I}{A} + \frac{1}{4k^3}\right)} \tag{3.44}$$

The second loading condition to be considered in this paragraph is a differential radial displacement Δw between the ring and the shell where

$$\Delta w = w_{shell} - w_{ring} \tag{3.45}$$

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

$$-\frac{\beta EI}{r_o r_c}\begin{bmatrix} 1 \\ o \end{bmatrix} \Delta w \quad . \tag{3.46}$$

The solution is given by

$$q_o = \frac{-\beta EI}{r_o r_c D}(1 + \beta) \quad \Delta w \quad , \tag{3.47}$$

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

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

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

The same rotation can be produced by an applied moment of

$$m_F = c_F \chi = \frac{B\left(c + \frac{1}{2k}\right)}{\left(\frac{\beta}{K}\frac{I}{A} + \frac{1}{4k^2}\right)}\Delta w \;. \tag{3.50}$$

The stresses in the ring at points A and B are

$$\sigma_\varphi^A = \frac{E}{r_o} w \tag{3.51}$$

and

$$\sigma_\varphi^B = \frac{E}{r_o}(w - h\chi) \tag{3.52}$$

where w and χ 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

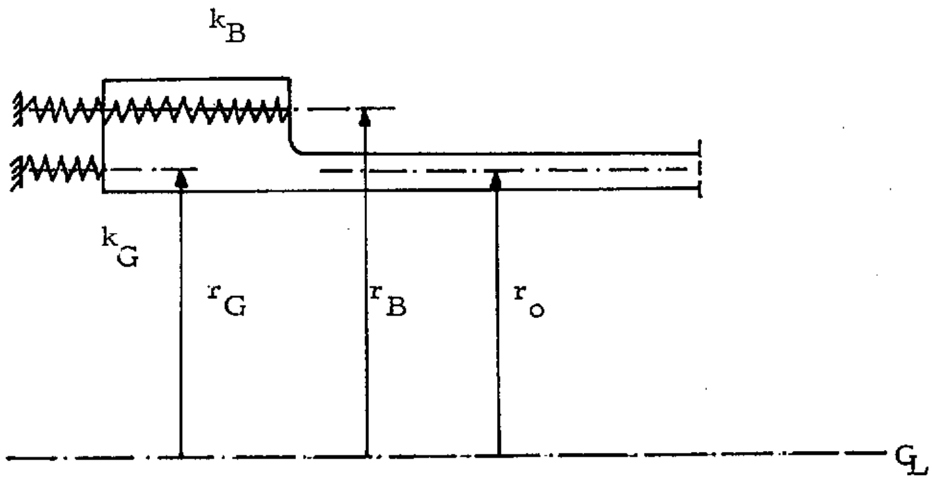


Fig. 3-4 - Gasket and Bolts Modeled as Springs

$$k_G = \frac{A_G E_G}{2\pi r_o t_G} \tag{3.53}$$

where the gasket area, A_G, is

$$A_G = 2\pi r_G b_G \tag{3.54}$$

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

$$k_B = \frac{A_B E_B}{2\pi r_o \ell_B} \tag{3.55}$$

where the total bolt area, A_B, for n_B bolts is

$$A_B = n_B A_{oB} \tag{3.56}$$

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

$$a = r_B - r_G \tag{3.57}$$

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

$$c_E = \frac{k_G k_B}{k_G + k_B} a^2 \tag{3.58}$$

The centroid of both springs is given by the radius

$$r_a = r_G + \frac{k_B}{k_B + k_G} a = r_B - \frac{k_G}{k_B + k_G} a \tag{3.59}$$

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

$$\delta_G = u + \chi \frac{K_B}{K_G + K_G} a \tag{3.60}$$

and

$$\delta_B = u - \chi \frac{K_G}{K_B + K_G} a \tag{3.61}$$

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

$$\Delta\sigma_G = \frac{E_G \delta_G}{t_G}, \tag{3.62}$$

and

$$\Delta\sigma_B = \frac{E_B \delta_B}{\ell_B}. \tag{3.63}$$

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,

$$f_B^{(o)} = \frac{n_B \sigma_B^{(o)} A_{oB}}{2\pi r_o} \tag{3.64}$$

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

$$m_F^{(o)} = a\ f_B^{(o)} \tag{3.65}$$

It is evident that from Eq. (3.43) the rotation χ 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

$$f_T = \frac{p\, r_o}{2} \tag{3.66}$$

and a force

$$f_F = \frac{r_G^2 - r_i^2}{2 r_o}\, p \tag{3.67}$$

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

$$r_F = \frac{2}{3}\ \frac{r_G^2 + r_i r_G + r_i^2}{r_G + r_i} \tag{3.68}$$

The corresponding applied flange moment is

$$m_F^{(1)} = f_T(r_a - r_o) + f_F(r_a - r_F) \tag{3.69}$$

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 ΔT 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 ΔT and therefore this term does not produce a Δw. The radial expansion of the flange ring due to internal pressure is

$$w_{ring} = \left(\frac{r_o r_i}{EA}\right) \left(\frac{ph\, r_i}{r_o}\right) \tag{3.70}$$

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

$$\chi^{(o)} = \frac{m_p^{(o)}}{c_F} \tag{3.71}$$

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

$$u^{(o)} = 0. \tag{3.72}$$

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

$$\chi^{(p)} = \frac{m_F^{(1)} + m_F^{(2)}}{c_E + c_F} \tag{3.73}$$

and the axial displacement is

$$u^{(p)} = \frac{f_T + f_F}{K_G + K_B} \tag{3.74}$$

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

$$\chi^{(T)} = \chi^{(o)} + \chi^{(p)} \tag{3.75}$$

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 χ^p and $w_o^{(T)}$ increases to w^p. The stresses are then

$$\sigma_x^p = \frac{B\left(\frac{1}{2k^2} + \beta\frac{I}{A} + \frac{c}{2k}\right)}{\frac{t^2 - t_n^2}{4}\left(\frac{\beta}{k}\frac{I}{A} + \frac{1}{4k^3}\right)}\chi^p \tag{3.76}$$

$$\tau_{xz}^p = \frac{B\left(c + \frac{1}{2k}\right)}{(t - t_n)\left(\frac{\beta}{k}\frac{I}{A} + \frac{1}{4k^3}\right)}\chi^p \tag{3.77}$$

$$\sigma_\varphi^p = \frac{E}{r_o} w^p + \nu \sigma_x^p \tag{3.78}$$

so that

$$\alpha_1 = \frac{t^2 - t_n^2}{4} \frac{E}{r_o} \frac{\left(\frac{\beta}{K} \frac{I}{A} + \frac{1}{4k^3}\right)}{B\left(\frac{1}{2k^2} + \beta\frac{I}{A} + \frac{c}{2k}\right)} w^p + \nu \tag{3.79}$$

and

$$\alpha_2 = \frac{t + t_n}{4} \quad \frac{c + \frac{1}{2k}}{\frac{1}{2k^2} + \beta\frac{I}{A} + \frac{c}{2k}} \tag{3.80}$$

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

$$m_{Fu} = Y_o \left[Z_F + Z_T\right] \tag{3.81}$$

where Y_o as 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 $\bar{b}$ the two section moduli are

$$Z_F = \frac{\bar{b}h^2}{4r_o} \tag{3.82}$$

and

$$S_F = \frac{\bar{b}h^2}{6r_o} \tag{3.83}$$

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

$$\bar{\sigma} = Y. \tag{3.84}$$

then

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

or simply

$$Z_T = \zeta_1 \frac{t^2 - t_n^2}{4} + \zeta_2 (t - t_n) \frac{\ell}{2} \tag{3.86}$$

The two dimensionless parameters are

$$\zeta_1 = \frac{1}{\sqrt{1 + \alpha_1 + \alpha_1^2 + 3\alpha_2^2}} \tag{3.87}$$

and

$$\zeta_2 = \frac{\alpha_2}{\sqrt{1 + \alpha_1 + \alpha_1^2 + 3\alpha_2^2}} \tag{3.88}$$

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 grove. 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.

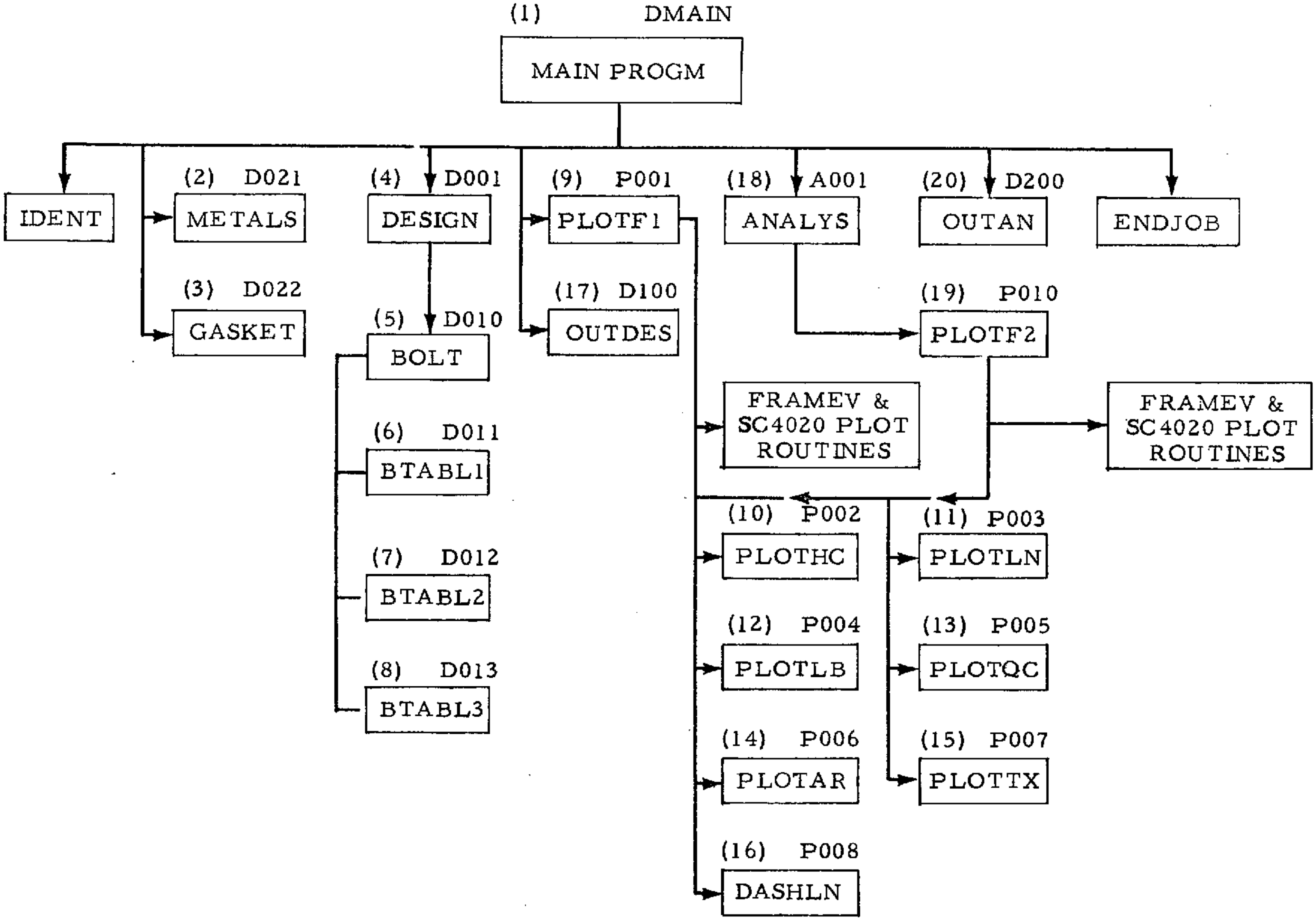


Chart 4-1 - Organization of the Flange Design and Analysis Program

Table 4-1

PROGRAM DESCRIPTION

No.	Symbol	Name	Description
1	DMAIN		Design program for low profile flanges
2	D021	METALS	Table of metallic materials design properties for tubes, flanges and bolts
3	D022	GASKET	Table of gasket materials design properties
4	D001	DESIGN	Design routine for low profile flanges
5	D010	BOLT	Bolt data handling
6	D011	BTABL1	Bolt table for open wrenching
7	D012	BTABL2	Bolt table for socket wrenching
8	D013	BTABL3	Bolt table for internal wrenching
9	P001	PLOTF1	Plot routine for low profile flanges with flat gasket and machined spotfaces for the holes
10	P002	PLOTHC	Plot a half circle from IA to IB
11	P003	PLOTLN	Plot a line
12	P004	PLOTLB	Plot label
13	P005	PLOTQC	Plot a quarter circle from IA to IB
14	P006	PLOTAR	Plot an arrow head for different orientations
15	P007	PLOTTX	Plot text
16	P008	DASHLN	Dashed-dotted line
17	D100	OUTDES	Output of the design routine
18	A001	ANALYS	Analysis routine
19	P010	PLOTF2	Plot of the analysis results
20	D200	OUTAN	Output of the analysis results

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 $7\frac{1}{2}$ by $7\frac{1}{2}$ inch.

Sample computer output, printed and plotted, is presented in Section 5.

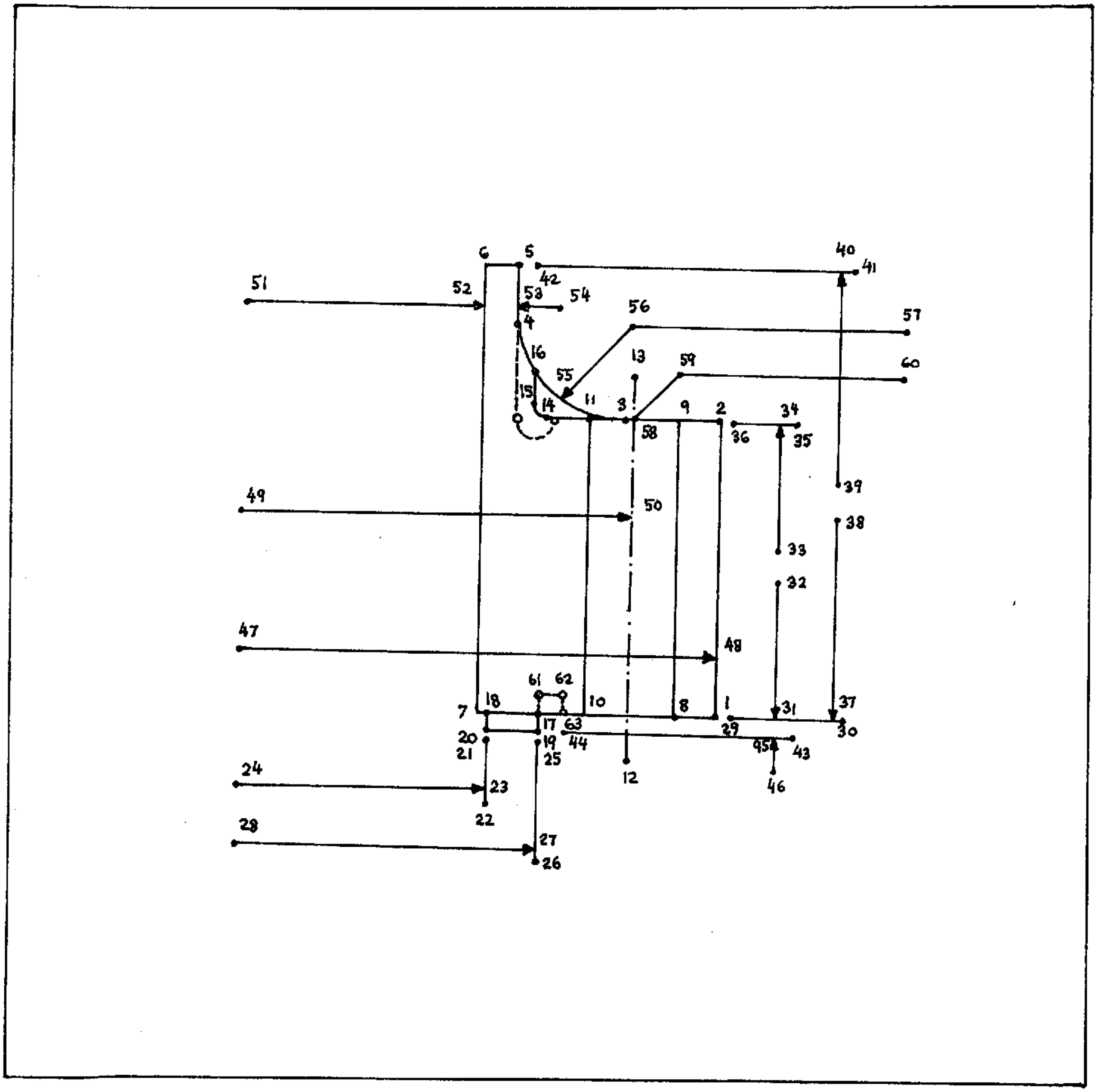


Fig. 4-1 - Layout of Design Summary SC 4020 Plot

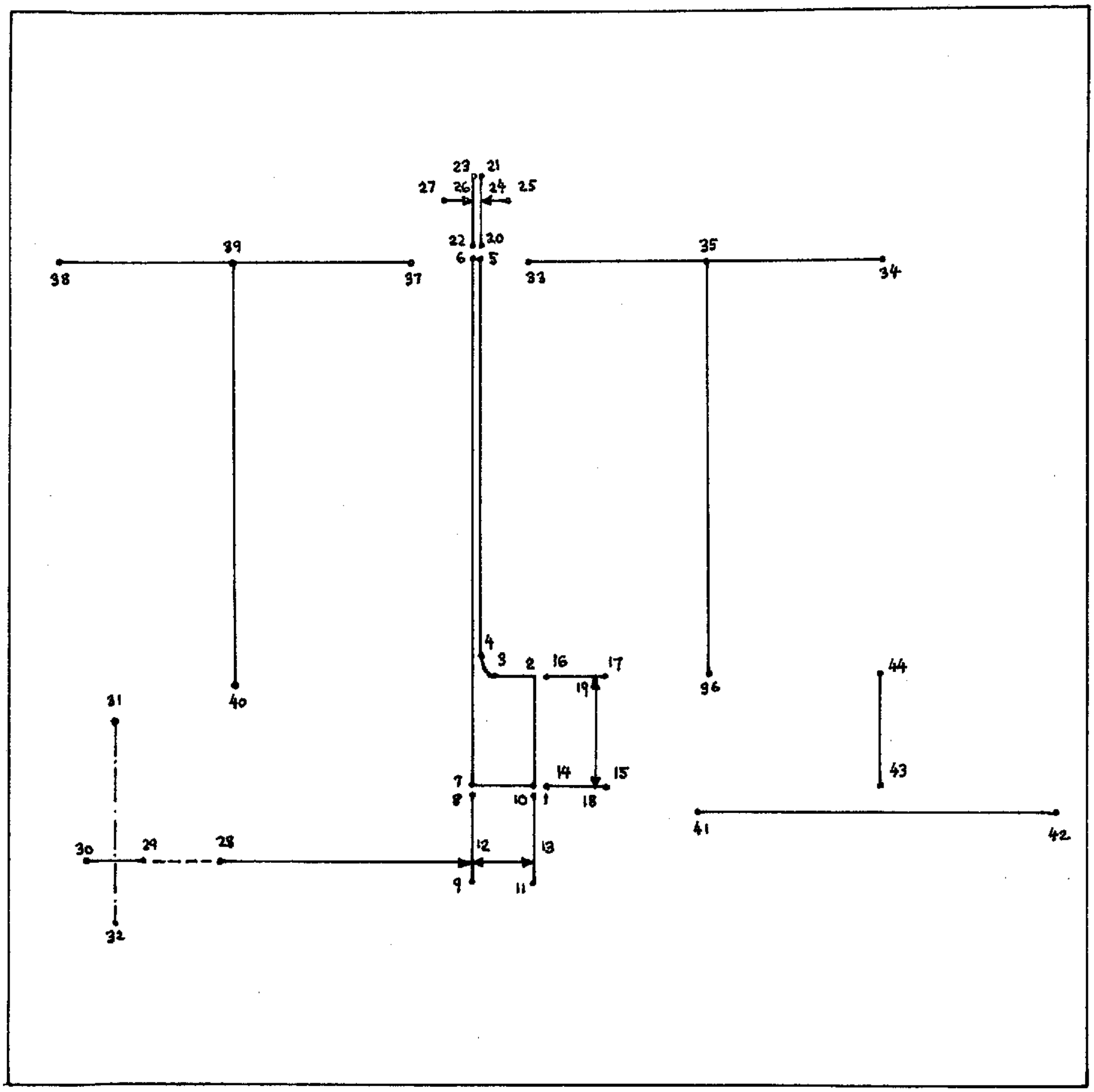


Fig. 4-2 - Layout of Analysis Summary SC 4020 Plot

PROGRAM LISTING

LIST OF ROUTINES IN FLANGE DESIGN AND ANALYSIS PROGRAM

```
434600*TPF$.LIST1
        @HDG LIST OF ROUTINES IN FLANGE DESIGN AND ANALYSIS PROGRAM
        @PRT,C LIST1
        @HDG  DMAIN      (1) (MAIN PROGM)
        @PRT,C  DMAIN
        @HDG  D021       (2) (METALS)
        @PRT,C  D021
        @HDG  D022       (3) (GASKET)
        @PRT,C  D022
        @HDG  D001       (4) (DESIGN)
        @PRT,C  D001
        @HDG  D010       (5) (BOLT)
        @PRT,C  D010
        @HDG  D011       (6) (BTABL1)
        @PRT,C  D011
        @HDG  D012       (7)  (BTABL2)
        @PRT,C  D012
        @HDG  D013       (8) (BTABL3)
        @PRT,C  D013
        @HDG  P001       (9) (PLOTF1)
        @PRT,C  P001
        @HDG  P002      (10) (PLOTHC)
        @PRT,C  P002
        @HDG  P003      (11) (PLOTLN)
        @PRT,C  P003
        @HDG  P004      (12) (PLOTLB)
        @PRT,C  P004
        @HDG  P005      (13) (PLOTWC)
        @PRT,C  P005
        @HDG  P006      (14) (PLOTAR)
        @PRT,C  P006
        @HDG  P007      (15) (PLOTTX)
        @PRT,C  P007
        @HDG  P008      (16) (DASHLN)
        @PRT,C  P008
        @HDG  D100      (17) (OUTDES)
        @PRT,C  D100
        @HDG  A001      (18) (ANALYS)
        @PRT,C  A001
        @HDG  P010      (19) (PLOTF2)
        @PRT,C  P010
        @HDG  D200      (20) (OUTAN)
        @PRT,C  D200


@HDG   DMAIN      (1) (MAIN PROGM)


@PRT,C   DMAIN
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```

```
434600*TPFS.DMAIN
      C
      C       DESIGN PROGRAM FOR LOW PROFILE FLANGES
      C       K.R.LEIMBACH, LOCKHEED-HUNTSVILLE, EXT.353
      C       21 NOVEMBER 1972
      C
            1 FORMAT(12A6)
            2 FORMAT(8E10.4)
            3 FORMAT(16I5)
      C
              DIMENSION HEAD(12),AD(22),KOPT(10)
              DIMENSION A(9,4),SRES(5,4),STR(5,4),AP(8)
              DATA(AD(I),I=1,22)/22*6H      /
              READ(5,1) (AD(I),I=1,12)
              CALL IDENT(9,AD)
              READ(5,3) NCASES
              ICASE=1
           10 READ(5,1) (HEAD(I),I=1,12)
              READ(5,2) P,DI,T,DELT,HT
              READ(5,2) PF,BF,FS,GF
              READ(5,3) IT,IF,IB,IG
              IF(IT.EQ.0) READ(5,2) ET,ANUT,RHOT,ALFAT,FTYT,FTUT
              IF(IF.EQ.0) READ(5,2) EF,ANUF,RHOF,ALFAF,FTYF,FTUF
              IF(IB.EQ.0) READ(5,2) EB,ANUB,RHOB,ALFAB,FTYB,FTUB
              IF(IG.EQ.0) READ(5,2) EG,AKG,SG,ALFAG,AMUG,GAMU,GAMS,HG,SP
              IF(IG.LT.0) READ(5,2) HS,BS,HR
      C
              IF(IT.GT.0) CALL METALS(IT,ET,ANUT,RHOT,ALFAT,FTYT,FTUT)
              IF(IF.GT.0) CALL METALS(IF,EF,ANUF,RHOF,ALFAF,FTYF,FTUF)
              IF(IB.GT.0) CALL METALS(IB,EB,ANUB,RHOB,ALFAB,FTYB,FTUB)
              IF(IG.GT.0) CALL GASKET(IG,EG,AKG,SG,ALFAG,AMUG,GAMU,GAMS,HG,SP)
      C
              READ(5,3) (KOPT(I),I=1,10)
              DIMENSION TUBMTL(2),FLAMTL(2),BOLMTL(2),GASMTL(2)
              READ(5,1) (TUBMTL(I),I=1,2),(FLAMTL(I),I=1,2)
             *              ,(BOLMTL(I),I=1,2),(GASMTL(I),I=1,2)
              READ(5,3) NPHASE
              READ(5,2) DELTAT
          101 FORMAT(1H1)
          102 FORMAT(' NOMINAL PRESSURE  P=',F16.3,' PSI'/
             *         ' NOMINAL DIAMETER DI=',F16.3,' INCH'/
             *         ' TUBE THICKNESS    T=',F16.3,' INCH'/
             *         ' TUBE THICKN TOLR DT=',F16.3,' INCH'/
             *         ' HEIGHT TO WELD   HT=',F16.3,' INCH'///)
          103 FORMAT(' PROOF FACTOR      PF=',F10.3/
             *         ' BURST FACTOR      BF=',F10.3/
             *         ' SAFETY FACTOR     FS=',F10.3/
             *         ' GASKET FACTOR     GF=',F10.3///)
          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=',I5/
             *         ' ELASTIC MODULUS          E=',E16.8,' PSI'/
             *         ' POISSON-S RATIO         NU=',F10.3/
             *         ' DENSITY                RHO=',F10.4,' LB/CUBIC-INCH'/
```

```
     *           ' THERM EXP COEFF      ALFA=',E16.8,' INCH/INCH/F'/
     *           ' TENSILE YIELD STR     FTY=',E16.8,' PSI'/
     *           ' ULTIMATE TENS STR     FTU=',E16.8,' PSI'////)
  109 FORMAT('  MATERIAL TABLE NO.        I=',I5/
     *           ' ELASTIC MODULUS          E=',E16.8,' PSI'/
     *           ' YIELD STRENGTH          KG=',E16.8,' PSI'/
     *           ' SEATING STRESS          SG=',E16.8,' PSI'/
     *           ' THERM EXP COEFF       ALFA=',E16.8,' INCH/INCH/F'/
     *           ' COEFF OF FRICTION       MU=',F10.3/
     *           ' WIDTH COEFFICIENT     GAMU=',F10.3/
     *           ' WIDTH COEFFICIENT     GAMS=',F10.3/
     *           ' GASKET THICKNESS        HG=',F10.4,' INCH'/
     *           ' SEALING STRESS RATE      SP=',F10.4///)
  110 FORMAT('  OPTIONS'/)
  111 FORMAT(10I5)
  119 FORMAT('  PRESSURE ACTIVATED SEAL'/
     *           ' DEPTH OF THE SEAL GLAND    HS=',F10.3,' INCH'/
     *           ' WIDTH OF THE SEAL GLAND    BS=',F10.3,' INCH'/
     *           ' DEPTH OF THE RECESS        HR=',F10.3,' INCH'///)
  120 FORMAT('  NUMBER OF PHASES TO BE CONSIDERED IN THE ANALYSIS =',I3/
     *           ' TEMPERATURE DIFFERENTIAL =',F10.2,' DEG F'/)
  121 FORMAT('  COMPUTED THICKNESS   T=',F10.4,' INCH'/)
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,ALFAT,FTYT,FTUT
      WRITE(6,105)
      WRITE(6,108) IF,EF,ANUF,RHOF,ALFAF,FTYF,FTUF
      WRITE(6,106)
      WRITE(6,108) IB,EB,ANUB,RHOB,ALFAB,FTYB,FTUB
      IF(IG.LT.0) GO TO 20
      WRITE(6,107)
      WRITE(6,109) IG,EG,AKG,SG,ALFAG,AMUG,GAMU,GAMS,HG,SP
      GO TO 25
   20 WRITE(6,119) HS,BS,HR
      HG=HR
   25 CONTINUE
      WRITE(6,110)
      WRITE(6,111) (KOPT(I),I=1,10)
      WRITE(6,120) NPHASE,DELTAT
C
      CALL DESIGN(P,DI,T,DELT,PF,BF,FS,GF
     *            ,ET,ANUT,RHOT,ALFAT,FTYT,FTUT
     *            ,EF,ANUF,RHOF,ALFAF,FTYF,FTUF
     *            ,EB,ANUB,RHOB,ALFAB,FTYB,FTUB
     *            ,EG,AKG,SG,ALFAG,AMUG,GAMU,GAMS,HG,SP,HS,BS
     *            ,KOPT,ADB,WEIGHT,PB
     *            ,B,H,RI,RG,RB,RFIL,RSPOT,DHOLE,DSPOT,N,BG,HT)
      WRITE(6,121) T
C
      CALL PLOTF1(B,H,T,RI,RG,RB,RFIL,RSPOT,DHOLE,DSPOT,N,BG,HG,BS,HS,HT
     *            ,FTYF,FTUF,FTYB,FTUB,SG,AKG,P,WEIGHT
     *            ,FLAMTL,BOLMTL,GASMTL,HEAD,KOPT)
C
```

DMAIN (1) (MAIN PROGM)

```
C
      CALL OUTDES(HEAD,AOB,WEIGHT,KOPT,T
     *            ,B,H,RI,RG,RB,RFIL,RSPOT,DHOLE,DSPOT,N ,DG,HT)
C
      CALL ANALYS(P,DI,T,DELT,PF,BF,FS,GF
     *           ,ET,ANUT,RHOT,ALFAT,FTYT,FTUT
     *            ,EF,ANUF,RHOF,ALFAF,FTYF,FTUF
     *            ,EB,ANUB,RHOB,ALFAB,FTYB,FTUB
     *            ,EG,AKG,SG,ALFAG,AMUG,GAMUG,GAMS,HG,HS,BS
     *            ,KOPT,AOB,NPHASE,DELTAT,PB
     *            ,B,H,RI,RG,RB,RFIL,RSPOT,DHOLE,DSPOT,N ,DG,HT
     *            ,A,SRES,STR,AP,HEAD)
C
      CALL OUTAN(HEAD,A,SRES,STR)
C
      ICASE=ICASE+1
      IF(ICASE.LE.NCASES) GO TO 10
C
      CALL ENDJOB
      STOP
      END
```

@HDG D021 (2) (METALS)

@PRT,C D021
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```
D021        (2) (METALS)

434600*TPFS.D021
              SUBROUTINE METALS(N,E,ANU,RHO,ALFA,FTY,FTU)
        C
        C     TABLE OF METALLIC MATERIALS DESIGN PROPERTIES FOR
        C     TUBES, FLANGES AND BOLTS
        C     K.R.LEIMBACH, 28 NOVEMBER 1972
        C
              COMMON/PROMTX/P(10,6)
        C
              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, 38.1 E+3,
             *  9.9 E+6,  0.33  ,  0.098  , 12.5 E-6, 50.0 E+3, 62.0 E+3,
             *  9.9 E+6,  0.33  ,  0.098  , 12.5 E-6, 47.0 E+3, 59.0 E+3,
             * 28.0 E+6,  0.30  ,  0.288  ,  9.5 E-6, 35.0 E+3, 90.0 E+3,
             * 28.0 E+6,  0.30  ,  0.288  ,  9.5 E-6, 30.0 E+3, 76.5 E+3,
             * 28.0 E+6,  0.30  ,  0.288  ,  9.5 E-6, 25.0 E+3, 66.0 E+3,
             * 28.0 E+6,  0.30  ,  0.288  ,  9.5 E-6,131.0 E+3,200.0 E+3,
             * 28.0 E+6,  0.30  ,  0.288  ,  9.5 E-6,128.0 E+3, 196.0E +3,
             * 28.0 E+6,  0.30  ,  0.288  ,  9.5 E-6,120.0 E+3,180.0 E+3/
        C
        C     LEGEND- AL 6061-T6  (1) AT RT, (2) AT 200F
        C             AL 2024-T3  (3) AT RT, (4) AT 200F
        C             347 SS       (5) AT RT, (6) AT 200F, (7) AT 600F
        C             A286         (8) AT RT, (9) AT 200F, (10)AT 600F
        C
              E=P(N,1)
              ANU=P(N,2)
              RHO=P(N,3)
              ALFA=P(N,4)
              FTY=P(N,5)
              FTU=P(N,6)
        C
              RETURN
              END


@HDG  D022        (3) (GASKET)


@PRT,C  D022
FURPUR 24H1-03/10-14:53
```

```
D022        (3) (GASKET)

434600*TPFS.D022
              SUBROUTINE GASKET(N,E,AKG,SG,ALFA,AMU,GAMU,GAMS,HG,SP)
        C
        C     TABLE OF GASKET MATERIALS DESIGN PROPERTIES
        C     K.R.LEIMBACH, 28 NOVEMBER 1972
        C
              COMMON/PROGSK/P(5,9)
        C
              DATA((P(I,J),J=1,9),I=1,5)/
             * 44.0 E+3, 10.0 E+3,  6.5 E+3,  1.3 E-3,  0.5   ,
             *  0.5    ,  0.5    ,  0.03125,  3.50   ,
             * 44.0 E+3, 10.0 E+3,  3.7 E+3,  1.3 E-3,  0.5   ,
             *  0.5    ,  0.5    ,  0.0625 ,  2.75   ,
             * 44.0 E+3, 10.0 E+3,  1.6 E+3,  1.3 E-3,  0.5   ,
             *  0.5    ,  0.5    ,  0.125  ,  2.00   ,
             *180.0 E+3,  8.0 E+3,  4.0 E+3,  3.86E-5,  0.12  ,
             *  1.0    ,  1.0    ,  0.0625 ,  3.00   ,
             * 28.0 E+6, 40.0 E+3, 18.9 E+3,  9.50E-6,  0.30  ,
             *  1.0    ,  1.0    ,  0.025  ,  5.50   /
        C
        C     LEGEND- ASBESTOS (1) 1/32, (2) 1/16, (3) 1/8
        C                      (4) KEL-F81 1/16
        C                      (5) CRES 321-A 1/16
        C
              E=P(N,1)
              AKG=P(N,2)
              SG=P(N,3)
              ALFA=P(N,4)
              AMU=P(N,5)
              GAMU=P(N,6)
              GAMS=P(N,7)
              HG=P(N,8)
              SP=P(N,9)
        C
              RETURN
              END


@HDG  D001        (4) (DESIGN)


@PRT,C  D001
FURPUR 24H1-03/10-14:53
```

```
434600*TPF$.0001
      SUBROUTINE DESIGN(P,DI,T,DELT,PF,BF,FS,GF
     *                  ,ET,ANUT,RHOT,ALFAT,FTYT,FTUT
     *                  ,EF,ANUF,RHOF,ALFAF,FTYF,FTUF
     *                  ,EB,ANUB,RHOB,ALFAB,FTYB,FTUB
     *                  ,EG,AKG,SG,ALFAG,AMUG,GAMU,GAMS,HG,SP,HS,BS
     *                  ,KOPT,AOB,WEIGHT,PB
     *                  ,B,H,RI,RG,RB,RFIL,RSPOT,DHOLE,DSPOT,NB,BG,HT)
C
C     DESIGN ROUTINE FOR LOW PROFILE FLANGES
C     K.R.LEIMBACH, 28 NOVEMBER 1972
C
      DIMENSION KOPT(10)
C
C     TUBE THICKNESS
      JBOLT=KOPT(4)
      RI=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*RI/FTYT
      GO TO 40
   20 ALAMB=.75
      T1=BF*P*RI/(FTUT*ALAMB)+2.*DELT
      T2=PF*P*RI/(FTYT*ALAMB)+2. *DELT
      T=AMAX1(T1,T2)
      GO TO 40
   30 T1=1.1*PF*P*RI/(FTYT-.4*PF*P)
      T2=1.1*BF*P*RI/(FTUT-.4*BF*P)
      T=AMAX1(T1,T2)
   40 CONTINUE
C
C     BOLT SIZE
      DB=T
      ID=0
   45 CALL BOLT(DB,ETA0,ETA1,ETA2,AOB,DHOLE,DSPOT,RSPOT,ISIZE,JBOLT,ID)
      IF(KOPT(2).EQ.1) GO TO 50
      IF(KOPT(2).EQ.2) GO TO 60
   50 E1=ETA1*DB
      E2=ETA2*DB
      IF(T.GE.0.20) RFIL=0.375
      IF(T.LT.0.20) RFIL=0.3125
      IF(T.LT.0.15) RFIL=0.250
      IF(T.LT.0.10) RFIL=0.1875
      IF(T.LT.0.05) RFIL=0.125
      GO TO 70
   60 E1=ETA2*DB
      E2=E1
      IF(T.GE.0.20) RFIL=0.125
      IF(T.LT.0.20) RFIL=0.100
      IF(T.LT.0.15) RFIL=0.075
      IF(T.LT.0.10) RFIL=0.050
      IF(T.LT.0.05) RFIL=0.025
   70 CONTINUE
C
```

```
C       BOLT CIRCLE RADIUS
        C1=0.0625
        C2=0.05
        IF(KOPT(2).EQ.1) RB=RI+T+C1+E1
        IF(KOPT(2).EQ.2) RB=RI+T+2.*RFIL+E1
C
C       FLANGE WIDTH
        B=RB+E2-RI
C
C       GASKET WIDTH
        IF(KOPT(3).GT.0) GO TO 80
        RG=.5*(RB-.5*DB+RI)
        IF(KOPT(5).EQ.0) BG=PF*P*RG/(2.*(GAMU*AKG-GAMS*SG*GF))
        IF(KOPT(5).EQ.1) BG=PF*P*RG/(2.*(GAMU*AKG-GAMS*SP*P*PF*GF))
        IF(KOPT(5).EQ.2) BG=RB-.5*DHOLE-RI-2.*C2
        IF(KOPT(6).EQ.3) GO TO 76
        RG=RB-.5*DHOLE-.5*BG-C2
        R1=RG-.5*BG-C2
        R2=RI
        IF(R1.GT.R2) GO TO 75
        RG=RI+.5*BG+C2
     74 RB=RG+.5*BG+.5*DHOLE+C2
     75 B=RB+E2-RI
        GO TO 90
     76 RG=RI+.5*BG+C2
        R1=RG+.5*BG+C2
        R2=RB-.5*DHOLE-C2
        IF(R1.LT.R2) GO TO 75
        GO TO 74
     80 BG=T
        RG=RI+.5*BG
        R1=RI+BG+2.*BS
        R2=RB-.5*DHOLE
        IF(R1.GT.R2) RB=RG+.5*BG+2.*BS+.5*DHOLE
        B=RB+E2-RI
     90 CONTINUE
C
C       BOLT FORCES
        PI=3.141593
     95 IF(KOPT(3).GT.0) GO TO 100
        IF(KOPT(5).EQ.2) GO TO 98
        PB1=2.*PI*RG*BG*GAMU*AKG
        IF(KOPT(5).EQ.0) PB2=2.*PI*RG*BG*GAMS*SG*GF+PI*RG**2*PF*P
        IF(KOPT(5).EQ.1) PB2=2.*PI*RG*BG*GAMS*SP*P*PF*GF+PI*RG**2*PF*P
        PB=AMAX1(PB1,PB2)
        GO TO 110
     98 PB2=2.*PI*RG*BG*GAMS*SP*P*PF*GF+PI*RG**2*PF*P
        PB1=PB2
        SG2=PB2/(2.*PI*BG*GAMU*RG)
        IF(SG2.LT.SG) PB1=2.*PI*RG*BG*GAMU*SG
        PB=AMAX1(PB1,PB2)
        GO TO 110
    100 PB=PI*RG**2*PF*P
    110 CONTINUE
C
C       NUMBER OF BOLTS
```

```
DOOI          (4) (DESIGN)

                ENB1=PB/(FTYB*AOB)
                ENB2=(BF/PF)*PB/(FTUB*AOB)
                ENB=AMAX1(ENB1,ENB2)
                NB=ENB
                IF(NB.GE.6) GO TO 115
                NB=6
                PB=6.*FTYB*AOB
            115 CONTINUE
          C
          C     BOLT SPACING
                ENB=NB
                S=2.*PI*RB/ENB
                ID=1
                SOVD=S/DB
                IF(SOVD.GT.8.0) GO TO 120
                IF(SOVD.LT.ETA0) GO TO 130
                GO TO 140
            120 ISIZE=ISIZE-1
                IF(ISIZE.LT.1) GO TO 140
                GO TO 45
            130 ISIZE=ISIZE+1
                IF(ISIZE.GT.14) GO TO 140
                GO TO 45
            140 CONTINUE
          C
          C
          C     FLANGE HEIGHT
                E=RB-RG
                RO=RI+.5*T
                TN=T/2.
                EMFU=FS*PB*E/(2.*PI*RO)
                BBAR=B-DHOLE*SQRT(DHOLE/S)
                ZETA1=0.80
                ZETA2=0.18
                CAPA=FTYF*BBAR/(6.*RO)
                CAPB=FTYF*ZETA2*(T-TN)/2.
                CAPC=FTYF*ZETA1*(T**2-TN**2)/4.-EMFU
                RTSQ=CAPB**2-4.*CAPA*CAPC
                RT=SQRT(RTSQ)
                H=(RT-CAPB)/(2.*CAPA)
          C
          C     CHECK FLANGE HEIGHT
                SOVH=S/H
                IF(SOVH.GT.3) H=S/3.
          C
          C     WEIGHT COMPUTATION
                RWT=.5*(2.*RI+T+B)
                AWT=(B-T)*H
                VOL=2.*PI*RWT*AWT
                WEIGHT=RHOF*VOL
          C
                RETURN
                END
```

```
0001        (4) (DESIGN)
)HDG   0010        (5) (BOLT)


PRT,C   0010
URPUR 24H1-03/10-14:53
```

```
D010       (5) (BOLT)

34600*TPFS.D010
                SUBROUTINE BOLT(D,ETA0,ETA1,ETA2,A0B,DHOLE,DSPOT,RSPOT,ISIZE
               *                    ,JBOLT,ID)
          C
                COMMON/BOLTDT/DX(14,5)
          C
                IF(JBOLT.EQ.1) CALL BTABL1
                IF(JBOLT.EQ.2) CALL BTABL2
                IF(JBOLT.EQ.3) CALL BTABL3
                IF(JBOLT.LE.0.OR.JBOLT.GE.4) STOP
          C
                IF(ID.GT.0) GO TO 16
                I=1
             15 DI=DX(I,1)
                DD=D-DI
                IF(DD.LE.0) GO TO 20
                I=I+1
                GO TO 15
             16 I=ISIZE
          C
             20 D=DX(I,1)
                ETA0=DX(I,2)
                ETA1=DX(I,3)
                ETA2=DX(I,4)
                A0B=DX(I,5)
                DHOLE=D+.005
                DSPOT=2.*ETA1*D
                RSPOT=.062
                IF(ID.EQ.0) ISIZE=I
          C
                RETURN
                END


HDG  D011        (6) (BTABL1)


PRT,C  D011
URPUR 24H1-03/10-14:53
```

```
D011       (6) (BTABL1)

434600*TPF$.D011
                  SUBROUTINE BTABL1
           C
           C      BOLT TABLE FOR OPEN WRENCHING
           C      14 SIZES FROM .25 IN TO 1.5 IN NOMINAL DIAMETER
           C
                  COMMON/BOLTD1/DX(14,5)
                  COMMON/BOLTDT/D(14,5)
           C
                  DATA((DX(I,J),J=1,5),I=1,14)/
                *    .2500,   3.00  ,  2.00  ,  1.50  ,   .03182,
                *    .3125,   2.60  ,  1.80  ,  1.40  ,   .05243,
                *    .3750,   2.67  ,  1.67  ,  1.33  ,  0.07749,
                *    .4375,   2.57  ,  1.57  ,  1.29  ,   .10631,
                *    .5000,   2.50  ,  1.62  ,  1.24  ,   .14190,
                *    .5625,   2.45  ,  1.56  ,  1.22  ,   .18194,
                *    .6250,   2.40  ,  1.50  ,  1.20  ,   .22600,
                *    .7500,   2.33  ,  1.49  ,  1.08  ,   .33446,
                *    .8750,   2.35  ,  1.43  ,  1.07  ,   .46173,
                *   1.0000,   2.25  ,  1.37  ,  1.06  ,   .60574,
                *   1.1250,   2.22  ,  1.33  ,  1.00  ,   .76327,
                *   1.2500,   2.25  ,  1.40  ,  1.00  ,   .92905,
                *   1.3750,   2.23  ,  1.36  ,  1.00  ,  1.15486,
                *   1.5000,   2.17  ,  1.33  ,  1.00  ,  1.40525/
           C
                  DO 10 I=1,14
                  DO 10 J=1,5
               10 D(I,J)=DX(I,J)
                  RETURN
                  END


)HDG  D012      (7)  (BTABL2)


@PRT,C  D012
FURPUR 24H1-03/10-14:53
```

```
D012        (7)   (BTABL2)

434600*TPFS.D012
      SUBROUTINE BTABL2
C
C     BOLT TABLE FOR SOCKET WRENCHING
C     14 SIZES FROM .25 IN TO 1.5 IN NOMINAL DIAMETER
C
      COMMON/BOLTD2/DX(14,5)
      COMMON/BOLTDT/D(14,5)
C
      DATA((DX(I,J),J=1,5),I=1,14)/
     *   .2500,   2.76  ,   1.60  ,   1.40  ,    .03182,
     *   .3125,   2.53  ,   1.50  ,   1.28  ,    .05243,
     *   .3750,   2.37  ,   1.33  ,   1.20  ,    .07749,
     *   .4375,   2.26  ,   1.25  ,   1.14  ,    .10631,
     *   .5000,   2.18  ,   1.20  ,   1.10  ,    .14190,
     *   .5625,   2.20  ,   1.22  ,   1.11  ,    .18194,
     *   .6250,   2.22  ,   1.25  ,   1.12  ,    .22600,
     *   .7500,   2.12  ,   1.17  ,   1.07  ,    .33446,
     *   .8750,   2.28  ,   1.31  ,   1.14  ,    .46173,
     *  1.0000,   2.19  ,   1.25  ,   1.10  ,    .60574,
     *  1.1250,   2.14  ,   1.22  ,   1.07  ,    .76327,
     *  1.2500,   2.09  ,   1.18  ,   1.04  ,    .92905,
     *  1.3750,   2.00  ,   1.16  ,   1.02  ,   1.15468,
     *  1.5000,   2.02  ,   1.13  ,   1.00  ,   1.40525/
C
      DO 10 I=1,14
      DO 10 J=1,5
   10 D(I,J)=DX(I,J)
      RETURN
      END


@HDG  D013        (8) (BTABL3)


@PRT,C  D013
FURPUR 24H1-03/10-14:53
```

```
D013        (8) (BTABL3)

434600*TPFS.D013
              SUBROUTINE BTABL3
      C
      C       BOLT TABLE FOR INTERNAL WRENCHING
      C       14 SIZES FROM .25 IN TO 1.5 IN NOMINAL DIAMETER
      C
              COMMON/BOLTD3/DX(14,5)
              COMMON/BOLTDT/D(14,5)
      C
              DATA((DX(I,J),J=1,5),I=1,14)/
             *   .2500,   1.92 ,   1.16 ,   0.96 ,   .03182,
             *   .3125,   1.86 ,   1.09 ,   0.93 ,   .05243,
             *   .3750,   1.79 ,   1.04 ,   0.91 ,   .07749,
             *   .4375,   1.80 ,   1.03 ,   0.91 ,   .10631,
             *   .5000,   1.78 ,   1.00 ,   0.90 ,   .14190,
             *   .5625,   1.76 ,   0.98 ,   0.89 ,   .18194,
             *   .6250,   1.75 ,   0.96 ,   0.88 ,   .22600,
             *   .7500,   1.68 ,   0.91 ,   0.84 ,   .33446,
             *   .8750,   1.69 ,   0.90 ,   0.85 ,   .46173,
             *  1.0000,   1.67 ,   0.89 ,   0.84 ,   .60574,
             *  1.1250,   1.86 ,   0.96 ,   0.92 ,   .76327,
             *  1.2500,   1.67 ,   0.87 ,   0.83 ,   .92905,
             *  1.3750,   1.80 ,   0.93 ,   0.89 ,  1.15468,
             *  1.5000,   1.65 ,   0.85 ,   0.82 ,  1.40525/
      C
              DO 10 I=1,14
              DO 10 J=1,5
           10 D(I,J)=DX(I,J)
              RETURN
              END


HDG   P001        (9) (PLOTF1)


PRT,C   P001
URPUR 24H1-03/10-14:53
```

```
'001        (9) (PLOTF1)

J4600*TPFS.P001
              SUBROUTINE PLOTF1(B,H,T,RI,RG,RB,RFIL,RSPOT,DHOLE,DSPOT,N,BG,HG
     *                         ,BS,HS,HT
     *                         ,FTYF,FTUF,FTYB,FTUB,SG,FYG,P,WEIGHT
     *                         ,FLAMTL,BOLMTL,GASMTL,HEAD,KOP1)
      C
      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)
           *         ,DATA1(1),DATA2(1),DATA3(3),DATA4(3),DATA5(3)
           *         ,DATA6(4),DATA7(11),DATA8(11),DATA9(11),DATA10(11)
           *      ,DATA11(5),DATA12(5)
           *         ,X(100),Y(100),IX(100),IY(100)
           *         ,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
        210 E1=RI+B-RB
            X(3)=X(2)-2.*E1
            Y(3)=Y(2)
            X(4)=X(3)-2.*RFIL
            Y(4)=Y(2)
        215 CONTINUE
            X(5)=X(4)
            Y(5)=Y(2)+HT
            X(6)=X(5)-T
            Y(6)=Y(5)
            X(7)=X(6)
            Y(7)=Y(1)
            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)
```

```
      X(12)=X(8)-D/2.
      Y(12)=Y(1)-.25
      X(13)=X(12)
      Y(13)=Y(2)+.25
      Y(14)=Y(2)
      X(14)=X(13)-DSPOT/2.+RSPOT
      X(15)=X(14)-RSPOT
      Y(15)=Y(2)+RSPOT
      X(16)=X(15)
      IF(X(16).LT.X(3)) GO TO 95
      X(14)=X(2)
      X(14)=X(3)
      X(15)=X(3)
      X(16)=X(3)
      Y(14)=Y(3)
      Y(15)=Y(3)
      Y(16)=Y(3)
      I16=1
      GO TO 102
   95 CONTINUE
C
      DX=X(3)-X(16)
      IF(DX.GE.RFIL) GO TO 101
      DY=SQRT(RFIL**2-DX**2)
      DYBAR=RFIL-DY
      Y(16)=Y(2)+DYBAR
      GO TO 102
  101 X(16)=X(4)
      Y(16)=Y(4)
  102 CONTINUE
C
      X(17)=X(7)+RG-RI+BG/2.
      Y(17)=Y(1)
      X(18)=X(17)-BG
      Y(18)=Y(1)
      X(19)=X(17)
      Y(19)=Y(1)-HG
      X(20)=X(18)
      Y(20)=Y(19)
C
      X(21)=X(18)
      Y(21)=Y(20)-.125
      X(22)=X(18)
      Y(22)=Y(21)-.375
      X(23)=X(18)
      Y(23)=Y(22)+.125
      X(24)=X(6)-2.0
      Y(24)=Y(23)
      X(25)=X(17)
      Y(25)=Y(21)
      X(26)=X(17)
      Y(26)=Y(25)-.750
      X(27)=X(17)
      Y(27)=Y(26)+.125
      X(28)=X(24)
      Y(28)=Y(27)
```

```
      X(29)=X(1)+.125
      Y(29)=Y(1)
      X(30)=X(29)+.750
      Y(30)=Y(1)
      X(31)=X(29)+.375
      Y(31)=Y(1)
      X(32)=X(31)
      Y(32)=Y(1)+H/2-.125
      X(33)=X(31)
      Y(33)=Y(32)+.25
      X(34)=X(31)
      Y(34)=Y(2)
      X(35)=X(31)+.125
      Y(35)=Y(2)
      X(36)=X(2)+.125
      Y(36)=Y(2)
      X(37)=X(30)-.125
      Y(37)=Y(1)
      X(38)=X(37)
      Y(38)=Y(1)+(H+HT)/2.-.125
      X(39)=X(37)
      Y(39)=Y(38)+.25
      X(40)=X(37)
      Y(40)=Y(5)
      X(41)=X(30)
      Y(41)=Y(5)
      X(42)=X(5)+.125
      Y(42)=Y(5)
      X(43)=X(31)+.125
      Y(43)=Y(19)
      X(44)=X(19)+.125
      Y(44)=Y(19)
      X(45)=X(31)
      Y(45)=Y(19)
      X(46)=X(31)
      Y(46)=Y(19)-.375
      X(47)=X(24)
      Y(47)=Y(1)+.125
      X(48)=X(1)
      Y(48)=Y(47)
      X(49)=X(24)
      Y(49)=Y(2)-.125
      X(50)=X(12)
      Y(50)=Y(49)
      X(51)=X(24)
      Y(51)=Y(5)-.125
      X(52)=X(6)
      Y(52)=Y(51)
      X(53)=X(5)
      Y(53)=Y(51)
      X(54)=X(5)+.375
      Y(54)=Y(51)
C
      DR=RFIL*(1.-SQRT(2.)/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(40)+.1
      Y(57)=Y(56)
      X(58)=X(12)
      Y(58)=Y(2)
      X(59)=X(12)+.10
      Y(59)=Y(2)+.10
      X(60)=X(40)+.1
      Y(60)=Y(59)
      IF(KOPT(3).EQ.0) GO TO 310
      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 XSCLVI(X(I),IX(I),IERR)
      CALL YSCLVI(Y(I),IY(I),IERR)
   10 CONTINUE
C
      CALL PLOTLN(1,2,IX,IY)
      CALL PLOTLN(2,3,IX,IY)
      IF(KOPT(2).EQ.1) CALL PLOTQC(3,4,IX,IY)
      IF(KOPT(2).EQ.2) CALL PLOTHC(3,4,IX,IY)
      CALL PLOTLN(4,5,IX,IY)
      CALL PLOTLN(5,6,IX,IY)
      CALL PLOTLN(6,7,IX,IY)
      IF(KOPT(3).EQ.1) GO TO 320
      CALL PLOTLN(7,1,IX,IY)
      GO TO 325
  320 CALL PLOTLN(18,20,IX,IY)
      CALL PLOTLN(20,19,IX,IY)
      CALL PLOTLN(19,17,IX,IY)
      CALL PLOTLN(17,61,IX,IY)
      CALL PLOTLN(61,62,IX,IY)
      CALL PLOTLN(62,63,IX,IY)
      CALL PLOTLN(63,1, IX,IY)
  325 CONTINUE
      CALL PLOTLN(8,9,IX,IY)
      CALL PLOTLN(10,11,IX,IY)
      IF(KOPT(2).EQ.2) GO TO 220
      CALL PLOTLN(3,14,IX,IY)
      IF(I16.EQ.1) GO TO 96
      CALL PLOTQC(14,15,IX,IY)
   96 CONTINUE
      CALL PLOTLN(15,16,IX,IY)
  220 CONTINUE
C
      CALL DASHLN(12,13,IX,IY)
```

```
      C
            IF(KOPT(3).EQ.1) GO TO 330
            CALL PLOTLN(17,19,IX,IY)
            CALL PLOTLN(18,20,IX,IY)
            CALL PLOTLN(19,20,IX,IY)
        330 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)
      C
            CALL PLOTAR(1,23,IX,IY)
            CALL PLOTAR(1,27,IX,IY)
            CALL PLOTAR(1,48,IX,IY)
            CALL PLOTAR(1,50,IX,IY)
            CALL PLOTAR(1,52,IX,IY)
            CALL PLOTAR(2,53,IX,IY)
            CALL PLOTAR(5,55,IX,IY)
            CALL PLOTAR(4,31,IX,IY)
            CALL PLOTAR(3,34,IX,IY)
            CALL PLOTAR(4,37,IX,IY)
            CALL PLOTAR(3,40,IX,IY)
            CALL PLOTAR(3,45,IX,IY)
      C
            DATA(DATA1(I),I=1,1)/6H   DIA/
            DATA(DATA2(I),I=1,1)/6H   R  /
            DATA(DATA3(I),I=1,3)/18H   DIA,      HOLES/
            DATA(DATA4(I),I=1,3)/18H   DIA SPOTFACE    /
            DATA(DATA5(I),I=1,3)/18H   R FILLET        /
            DATA(DATA6(I),I=1,4)/24H PRESSURE        PSIG  /
            DATA(DATA7(I),I=1,11)/66H FLANGE MATERIAL                  FTY=
           *  KSI, FTU=       KSI      /
            DATA(DATA8(I),I=1,11)/66H BOLT MATERIAL                    FTY=
           *  KSI, FTU=       KSI      /
            DATA(DATA9(I),I=1,11)/66H GASKET MATERIAL                      SEA
           *ING STRESS=       KSI      /
            DATA(DATA10(I),I=1,11)/66H                                      YI
           *LD STRENGTH=       KSI      /
            DATA(DATA11(I),I=1,5)/30H WEIGHT OF FLANGE           LB/
```

```
      DATA(DATA12(I),I=1,5)/30H PRESSURE ENERGIZED SEAL        /
C
      DGO=2.*RG+BG
      DGI=DGO-2.*BG
      DI=2.*RI
      DB=2.*RB
      DFO=DI+2.*B
      HWELD=H+HT
C
      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,RFIL)
      CALL PLOTTX(57,52,5,IX,IY,DATA2,6)
      CALL PLOTLB(60,24,5,IX,IY,DHOLE)
      CALL PLOTTX(60,56,5,IX,IY,DATA3,18)
      IPL=IX(60)+120
      JPL=IY(60)+5
      DN=N
      IF(KOPT(2).EQ.2) GO TO 340
      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)
  340 CONTINUE
      CALL PLOTLB(32,-24,10,IX,IY,H)
      CALL PLOTLB(38,-24,10,IX,IY,HWELD)
      CALL PLOTLB(43,24,-20,IX,IY,HG)
      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,-108,IX,IY,DATA10,66)
      GO TO 51
   50 CALL PLOTTX(26,-300,-108,IX,IY,DATA12,30)
   51 CONTINUE
      CALL PLOTTX(26,-156,-50,IX,IY,FLAMTL,12)
      CALL PLOTTX(26,-156,-70,IX,IY,BOLMTL,12)
      IF(KOPT(3).GT.0) GO TO 60
      CALL PLOTTX(26,-156,-90,IX,IY,GASMTL,12)
   60 CONTINUE
C
      FTYF=FTYF/1000.
      FTUF=FTUF/1000.
      FTYB=FTYB/1000.
      FTUB=FTUB/1000.
      SG=SG/1000.
      FYG=FYG/1000.
```

```
P001        (9) (PLOTF1)

                  CALL PLOTLB(26,-12,-50,IX,IY,FTYF)
                  CALL PLOTLB(26,116,-50,IX,IY,FTUF)
                  CALL PLOTLB(26,-12,-70,IX,IY,FTYB)
                  CALL PLOTLB(26,116,-70,IX,IY,FTUB)
                  IF(KOPT(3).GT.0) GO TO 70
                  CALL PLOTLB(26,116,-90,IX,IY,SG)
                  CALL PLOTLB(26,116,-108,IX,IY,FYG)
               70 CONTINUE
            C
                  CALL PLOTTX(6,-200,50,IX,IY,DATA6,24)
                  IXP=IX(6)-120
                  IYP=IY(6)+50
                  CALL LABLV(P,IXP,IYP,4,1,4)
                  CALL LABLV(DN,IPL,JPL,3,1,3)
            C
                  CALL PLOTTX(6,-200,30,IX,IY,DATA11,30)
                  IXW=IX(6)-56
                  IYW=IY(6)+30
                  CALL LABLV(WEIGHT,IXW,IYW,6,1,2)
            C
                  RETURN
                  END


@HDG  P002      (10) (PLOTHC)


@PRT,C  P002
FURPUR 24H1-03/10-14:53
```

P002 (10) (PLOTHC)

```
434600*TPFS.P002
      SUBROUTINE PLOTHC(IA,IB,IX,IY)
C
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
      I1=IX(IA)
      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*(1.-CPJ)
      DY=R*SPJ
      IDX=DX
      IDY=DY
      I2=IX(IA)-IDX
      J2=IY(IA)-IDY
      CALL LINEV(I1,J1,I2,J2)
      I1=I2
      J1=J2
   10 CONTINUE
C
      RETURN
      END
```

@HDG P003 (11) (PLOTLN)

@PRT,C P003

FURPUR 24R1-03/10-14:53

P003 (11) (PLOTLN)

```
434600*TPFS.P003
      SUBROUTINE PLOTLN(IA,IB,IX,IY)
C
C**** PLOT A LINE
C**** K.R.LEIMBACH, 8 NOVEMBER 1972
C
      DIMENSION IX(1),IY(1)
      CALL LINEV(IX(IA),IY(IA),IX(IB),IY(IB))
C
      RETURN
      END
```

@HDG P004 (12) (PLOTLB)

@PRT,C P004
FURPUR 24H1-03/10-14:53

```
P004      (12) (PLOTLB)

434600*TPFS.P004
              SUBROUTINE PLOTLB(I,NX,NY,IX,IY,Z)
        C
        C**** PLOT LABEL
        C**** K.R.LEIMBACH, 8 NOVEMBER 1972
        C
              DIMENSION IX(1),IY(1)
              IXP=IX(I)+NX
              IYP=IY(I)+NY
              CALL LABLV(Z,IXP,IYP,7,1,3)
        C
              RETURN
              END


@HDG  P005      (13) (PLOTQC)


@PRT,C  P005
FURPUR 24H1-03/10-14:53
```

P005 (13) (PLOTQC)

```
434600*TPF$.P005
      SUBROUTINE PLOTQC(IA,IB,IX,IY)
C
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)
      I1=IX(IA)
      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)
      IDX=DX
      IDY=DY
      I2=IX(IA)-IDX
      J2=IY(IA)+IDY
      CALL LINEV(I1,J1,I2,J2)
      I1=I2
      J1=J2
   10 CONTINUE
C
      RETURN
      END
```

@HDG P006 (14) (PLOTAR)

@PRT,C P006
FURPUR 24H1-03/10-14:53

P006 (14) (PLOTAR)

```
434600*TPFS.P006
      SUBROUTINE PLOTAR(IORNT,I,IX,IY)
C
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
C
      DIMENSION IX(1),IY(1)
      I1=IX(I)
      J1=IY(I)
      IF(IORNT.NE.1) GO TO 10
      I2=I1-10
      J2=J1+5
      I3=I2
      J3=J1-5
      GO TO 100
   10 IF(IORNT.NE.2) GO TO 20
      I2=I1+10
      J2=J1+5
      I3=I2
      J3=J1-5
      GO TO 100
   20 IF(IORNT.NE.3) GO TO 30
      I2=I1+5
      J2=J1-10
      I3=I1-5
      J3=J2
      GO TO 100
   30 IF(IORNT.NE.4) GO TO 40
      I2=I1+5
      J2=J1+10
      I3=I1-5
      J3=J2
      GO TO 100
   40 IF(IORNT.NE.5) GO TO 50
      I2=I1+11
      J2=J1+3
      I3=I1+3
      J3=J1+11
      GO TO 100
   50 RETURN
  100 CALL LINEV(I1,J1,I2,J2)
      CALL LINEV(I1,J1,I3,J3)
      CALL LINEV(I2,J2,I3,J3)
C
      RETURN
      END
```

```
 P006       (14) (PLOTAR)

@HDG  P007      (15) (PLOTTX)


@PRT,C  P007
FURPUR 24H1-03/10-14:53
```

```
P007      (15) (PLOTTX)

434600*TPF$.P007
                SUBROUTINE PLOTTX(I,NX,NY,IX,IY,AR,NTEXT)
          C
          C**** PLOT TEXT
          C**** K.R.LEIMBACH, 8 NOVEMBER 1972
          C
                DIMENSION IX(1),IY(1),AR(1)
          C
                IPLT=IX(I)+NX
                JPLT=IY(I)+NY
                CALL PRINTV(NTEXT,AR,IPLT,JPLT)
          C
                RETURN
                END


@HDG  P008      (16) (DASHLN)


@PRT,C  P008
FURPUR 24R1-03/10-14:53
```

```
P008      (16) (DASHLN)

134600*TPF$.P008
                SUBROUTINE DASHLN(IA,IB,IX,IY)
          C
          C     K.R.L. - 11/30/72
          C
                DIMENSION IX(1),IY(1)
          C
                I1=IX(IA)
                J1=IY(IA)
                J2=J1+25
             10 CALL LINEV(I1,J1,I1,J2)
                J1=J2+25
                J2=J1+25
                IF(J2.LT.IY(IB)) GO TO 10
                IF(J1.LT.IY(IB)) GO TO 15
                GO TO 20
             15 J2=IY(IB)
                GO TO 10
             20 CONTINUE
          C
                J1=IY(IA)+35
                J2=J1+5
             30 CALL LINEV(I1,J1,I1,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
             35 J2=IY(IB)
                GO TO 30
             40 CONTINUE
          C
                RETURN
                END


@HDG  D100      (17) (OUTDES)


@PRT,C  D100
FURPUR 24H1-03/10-14:53
```

D100   (17) (OUTDES)

```
34600*TPF$.D100
      SUBROUTINE OUTDES(HEAD,AOB,WEIGHT,KOPT,T
     *                  ,B,H,RI,RG,RB,RFIL,RSPOT,DHOLE,DSPOT,NB,BG,HT)
C
C**** OUTPUT OF DESIGN PARAMETERS
C**** K R L  1/8/73
C
      DIMENSION HEAD(12),KOPT(10)
  101 FORMAT(12A6)
  102 FORMAT('    AOB=',F10.4,' SQ-IN'/
     *      ' WEIGHT=',F10.4,' LB'/
     *      '      B=',F10.4,' IN'/
     *      '      H=',F10.4,' IN'/
     *      '     RI=',F10.4,' IN','     DI=',F10.4,' IN'/
     *      '     RG=',F10.4,' IN','    DGI=',F10.4,' IN'/
     *      '                         DGO=',F10.4,' IN'/
     *      '     RB=',F10.4,' IN','     DB=',F10.4,' IN'/
     *      '   RFIL=',F10.4,' IN'/
     *      '  RSPOT=',F10.4,' IN'/
     *      '  DHOLE=',F10.4,' IN'/
     *      '  DSPOT=',F10.4,' IN'/
     *      '     NB=',I10/
     *      '     BG=',F10.4,' IN'/
     *      '     HT=',F10.4,' IN'/)
  103 FORMAT(8F10.4)
C
      DI=2.*RI
      DGI=2.*RG-BG
      DGO=2.*RG+BG
      DB=2.*RB
C
      WRITE(6,101) (HEAD(I),I=1,12)
      IF(KOPT(3).GT.0) GO TO 50
      WRITE(6,102) AOB,WEIGHT,B,H,RI,DI,RG,DGI,DGO,RB,DB
     *            ,RFIL,RSPOT,DHOLE,DSPOT,NB,BG,HT
C
      RETURN
   50 WRITE(6,103) AOB,WEIGHT,T,B,H,RI,DI,RB,DB,RFIL,RSPOT,DHOLE,DSPO
     *      ,NB,HT
      RETURN
      END
```

@HDG A001   (18) (ANALYS)

@PRT,C A001
FURPUR 24R1-03/10-14:53

A001 (18) (ANALYS)

434600*TPFS.A001

```
      SUBROUTINE ANALYS(P,DI,T,DELT,PF,BF,FS,GF
     *                  ,ET,ANUT,RHOT,ALFAT,FTYT,FTUT
     *                  ,EF,ANUF,RHOF,ALFAF,FTYF,FTUF
     *                  ,EB,ANUB,RHOB,ALFAB,FTYB,FTUB
     *                  ,EG,ZKG,SG,ALFAG,AMUG,GAMU,GAMS,HG,HS,BS
     *                  ,KOPT,AOB,NPHASE,DELTAT,PB
     *                  ,B,H,RI,RG,RB,RFIL,RSPOT,DHOLE,DSPOT,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),SRES(5,4),STR(5,4),AP(8),HEAD(12),KOPT(10)
      PI = 3.14159
      RO = RI + T/2.
      FX=P*RO/2.
      FR=P*H*RI/RO
      RS=RG-BG/2.
      FP=P*(RS**2-RI**2)/(2.*RO)
      RP=(RS**2+RI*RS+RI**2)/(1.5*(RS+RI))
      AG=2.*PI*RG*BG
      IF(KOPT(3).GT.0) GO TO 50
      EKG=AG*EG/(2.*PI*RO*HG)
      GO TO 51
   50 EKG=AG*EF/(2.*PI*RO*(HG+HS))
   51 CONTINUE
      ELB=H
      ENB=NB
      AB=ENB*AOB
      EKB=AB*EB/(2.*PI*RO*ELB)
      E=RB-RG
      RA=RG+EKB*E/(EKB+EKG)
      CE=EKB*EKG*E**2/(EKB+EKG)
      AF=B*H
      AIF=AF*H**2/12.
      C=H/2.
      RC=RI+B/2.
      BEND=ET*T**3/(12.*(1.-ANUT**2))
      AK4=12.*(1.-ANUT**2)/(RO**2*T**2)
      AK2=SQRT(AK4)
      AK=SQRT(AK2)
      BETA=BEND*AK*RO*RC/(EF*AIF)
      DX=(1.+BETA)*(.5/AK2+BETA*(C**2+AIF/AF))-(C*BETA-.5/AK)**2
      CF=BEND*DX/(BETA*(BETA*AIF/(AK*AF)+.25/(AK*AK2)))
      CWF=.5/(AK2*AK*BEND)
      BETADX=BETA/DX
      CMF= BETADX *(.5/AK2+BETA*AIF/AF+.5*C/AK)
      CQF= BETADX *(C+.5/AK)
C
  101 J=1
      BFC=PB
      EAM=E*BFC/(2.*PI*RO)
      ADFL=0.
      CHI0=EAM/CF
      FROT=CHI0
```

AU01 (18) (ANALYS)

```
      BSTRO=BFC/AB
      GSTRO=BFC/AG
      BSTRS=BSTRO
      GSTRS=GSTRO
      ENX=0.
      WP=0.
      EMX=CMF*EAM
      EQX=CQF*EAM
      RDFL=+CWF*(EQX-AK*EMX)
      ENY= ET*T*RDFL/RO
      EMY= ANUT*EMX
      GO TO 201
C
  102 J=2
      EAM1=FP*(RA-RP)+FX*(RA-RO)
      DW=P*RO**2*(1.-ANUT/2.)/(ET*T)+RO*ALFAT*DELTAT-RO*RC*FR/(EF*AF)
      EAM2=CF*BETA*(C+.5/AK)*DW/DX
      ADFL=(FP+FX)/(EKG+EKB)
      CHIP=(EAM1+EAM2)/(CE+CF)
      FROT=CHIO+CHIP
      DG=ADFL+CHIP*EKB*E/(EKB+EKG)
      DB=ADFL-CHIP*EKG*E/(EKB+EKG)
      BSTRS=BSTRO+EB*DB/ELB
      GSTRS=GSTRO-EG*DG/HG
      BFC=BSTRS*AB
      ENX=FX
      WP=P*RO**2*(1.-ANUT/2.)/(ET*T )+RO*ALFAT*DELTAT
  112 EAM=CF*FROT
      EMX=CMF*EAM
      EQX=CQF*EAM
      RDFL=+CWF*(EQX-AK*EMX)+WP
      ENY= ET*T*RDFL/RO
      EMY= ANUT*EMX
      GO TO 201
C
  103 J=3
      DW=DW-RO*ALFAT*DELTAT
      EAM2=CF*BETA*(C+.5/AK)*DW/DX
      CHIP=(EAM1+EAM2)/(CE+CF)
      FROT=CHIO+CHIP
  113 DG=ADFL+CHIP*EKB*E/(EKB+EKG)
      DB=ADFL-CHIP*EKG*E/(EKB+EKG)
      BSTRS=BSTRO+EB*DB/ELB
      GSTRS=GSTRO-EG*DG/HG
      BFC=BSTRS*AB
      IF(J.EQ.3) WP=WP-RO*ALFAT*DELTAT
      IF(J.EQ.4) WP=PF*P*RO**2*(1.-ANUT/2.)/(ET/T)
      GO TO 112
C
  104 J=4
      CHIP=PF*CHIP
      FROT=CHIO+CHIP
      ADFL=PF*ADFL
      ENX=PF*FX
      GO TO 113
C
```

AGO1 (18) (ANALYS)

```
      201 WTOP=RDFL
          WBOT=RDFL-H*FROT
          SFTOP=EF*WTOP/RO
          SFBOT=EF*WBOT/RO
          IF(KOPT(2).EQ.1) GO TO 202
          SXI=-6.*EMX/T**2+ENX/T
          SXO=+6.*EMX/T**2+ENX/T
          SYI=-6.*EMY/T**2+ENY/T
          SYO=+6.*EMY/T**2+ENY/T
          TXZ=1.5*EQX/T
          GO TO 203
      202 TX=T+.5*RFIL
          SXI=-6.*EMX/TX**2+ENX/TX
          SXO= 6.*EMX/TX**2+ENX/TX
          SYI=-6.*EMY/TX**2+ENY/TX
          SYO= 6.*EMY/TX**2+ENY/TX
          TXZ=1.5*EQX/TX
      203 CONTINUE
C
          A(1,J)=BFC
          A(2,J)=EAM
          A(3,J)=ADFL
          A(4,J)=RDFL
          A(5,J)=FROT
          A(6,J)=BSTRS
          A(7,J)=GSTRS
          A(8,J)=SFTOP
          A(9,J)=SFBOT
C
          SRES(1,J)=ENX
          SRES(2,J)=ENY
          SRES(3,J)=EMX
          SRES(4,J)=EMY
          SRES(5,J)=EQX
C
          STR(1,J)=SXI
          STR(2,J)=SYI
          STR(3,J)=SXO
          STR(4,J)=SYO
          STR(5,J)=TXZ
C
          AP(1)=BFC
          AP(2)=EAM
          AP(3)=ADFL
          AP(4)=WP
          AP(5)=FROT
          AP(6)=ENX
          AP(7)=EMX
          AP(8)=EQX
          IPHASE=J
          CALL PLOTF2(DI,B,H,RB,RFIL,T,HT,ALFAT,DELTAT
     *                ,IPHASE,AP,ET,ANUT,AK,EF
     *          ,HEAD,KOPT,P,PF)
          J=J+1
          IF(J.GT.NPHASE) GO TO 300
          IF(J.EQ.2) GO TO 102
```

A001 (18) (ANALYS)

```
          IF(J.EQ.3) GO TO 103
          IF(J.EQ.4) GO TO 104
C
C         END OF STRESS AND DEFORMATION ANALYSIS
C
C          ULTIMATE MOMENT CAPACITY
      300 TN = 0.5*T
          ALFA1 = (T**2 -TN**2)/4.
          ALFA1 = ALFA1 * (BETA*AIF/(AK*AF) + 0.25/(AK2*AK))*EF*RDFL/RO
          ALFA1 = ALFA1/(BEND*(0.25/AK2 + BETA*AIF/AF +0.5*C/AK))
          ALFA1 = ALFA1 + ANU1
          ALFA2 = (0.25*(T+TN)*(C+0.5/AK))/(0.5/AK2 + BETA * AIF/AF +0.5
         1                *C/AK)
          ALBAR = 1.0 + ALFA1 + ALFA1**2 + 3.0*ALFA2**2
          ZETA1 = 1.0 / SQRT(ALBAR)
          ZETA2 = ALFA2*ZETA1
          SXX = B*H**2/4.0
          EMFU = FTYT*(SXX/RO + 0.25*ZETA1*(T**2 - TN**2) + ZETA2*(T-TN)*C
     1000 FORMAT('   MFU= ',E16.8,'   IN-LB/IN ' / ' ZETA1= ',E16.8/
         1               ' ZETA2= ',E16.8/)
          WRITE(6,1000) EMFU,ZETA1,ZETA2
          RETURN
          END
```

@HDG P010 (19) (PLOTF2)

@PRT,C P010
FURPUR 24H1-03/10-14:53

```
P010     (19) (PLOTF2)

434600*TPF$.P010
               SUBROUTINE PLOTF2(DI,B,H,RB,RFIL,T,HT,ALFAT,DELIAT
     *                        ,IPHASE,AP,ET,ANUT,AK,EF
     *         ,HEAD,KOPT,P,PF)
C
C**** PLOT ROUTINE FOR SUMMARY OF STRESS AND DEFORMATION ANALYSIS
C**** FOR LOW PROFILE FLANGES
C**** K.R.LEIMBACH, 19 DECEMBER 1972
C
      DIMENSION X(100),Y(100),IX(100),IY(100)
     *          ,HEAD(12),KOPT(10)
      DIMENSION AP(8)
C
      CALL FRAMEV(0)
      CALL SCRECT(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)
C
      SCALE=2.0
      BS=B/SCALE
      HS=H/SCALE
      RFILS=RFIL/SCALE
      TS=T/SCALE
      HTS = 2.0
      RI=DI/2.
C
      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)+RFILS
      GO TO 215
  210 CONTINUE
      EI=RI+B-RB
      EIS=EI/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
```

```
      X(9)=X(7)
      Y(9)=Y(7)-.6
      X(10)=X(1)
      Y(10)=Y(8)
      X(11)=X(1)
      Y(11)=Y(9)
      X(12)=X(7)
      Y(12)=Y(7)-.5
      X(13)=X(1)
      Y(13)=Y(12)
      X(14)=X(1)+.1
      Y(14)=Y(1)
      X(15)=X(1)+.6
      Y(15)=Y(1)
      X(16)=X(14)
      Y(16)=Y(2)
      X(17)=X(15)
      Y(17)=Y(2)
      X(18)=X(1)+.5
      Y(18)=Y(1)
      X(19)=X(18)
      Y(19)=Y(2)
      X(20)=X(5)
      Y(20)=Y(5)+.1
      X(21)=X(5)
      Y(21)=Y(5)+.6
      X(22)=X(6)
      Y(22)=Y(20)
      X(23)=X(6)
      Y(23)=Y(21)
      X(24)=X(5)
      Y(24) = Y(5) + 0.6
      X(25)=X(5)+.2
      Y(25)=Y(24)
      X(26)=X(6)
      Y(26)=Y(24)
      X(27)=X(6)-.2
      Y(27)=Y(24)
      X(28)=X(7)-1.5
      Y(28)=Y(12)
      X(29)=X(28)-.5
      Y(29)=Y(12)
      X(30)=X(29)-.5
      Y(30)=Y(12)
      X(31)=X(29)-.25
      Y(31) = Y(12) + 1.0
      X(32)=X(31)
      Y(32) = Y(12) - 0.5
      X(33)=X(2)+.1
      Y(33)=Y(5)+.1
      X(34)=X(33)+2.
      Y(34)=Y(33)
      X(35) = X(33) +1.0
      Y(35)=Y(33)
      X(36)=X(35)
      Y(36)=Y(2)
```

P010 (19) (PLOTF2)

```
      X(37) = X(6) -0.1
      Y(37) = Y(5) +0.1
      X(38)=X(37)-2.
      Y(38) = Y(37)
      X(39)=X(37)-1.
      Y(39) = Y(37)
      X(40)=X(39)
      Y(40)=Y(2)
      X(41)=X(36)
      Y(41)=Y(1)-.1
      X(42) = X(41) +2.0
      Y(42)=Y(41)
      X(43) = X(41) + 1.0
      Y(43)=Y(1)
      X(44)=X(43)
      Y(44)=Y(2)
C
      NP=44
C
      DO 10 I=1,NP
      CALL XSCLVI(X(I),IX(I),IERR)
      CALL YSCLVI(Y(I),IY(I),IERR)
   10 CONTINUE
C
      CALL PLOTLN(1,2,IX,IY)
      CALL PLOTLN(2,3,IX,IY)
      IF(KOPT(2).EQ.1) CALL PLOTQC(3,4,IX,IY)
      IF(KOPT(2).EQ.2) CALL PLOTHC(3,4,IX,IY)
      CALL PLOTLN(4,5,IX,IY)
      CALL PLOTLN(5,6,IX,IY)
      CALL PLOTLN(6,7,IX,IY)
      CALL PLOTLN(7,1,IX,IY)
      CALL PLOTLN(8,9,IX,IY)
      CALL PLOTLN(10,11,IX,IY)
      CALL PLOTLN(12,13,IX,IY)
      CALL PLOTAR(2,12,IX,IY)
      CALL PLOTAR(1,13,IX,IY)
      CALL PLOTLN(14,15,IX,IY)
      CALL PLOTLN(16,17,IX,IY)
      CALL PLOTLN(18,19,IX,IY)
      CALL PLOTAR(4,18,IX,IY)
      CALL PLOTAR(3,19,IX,IY)
      CALL PLOTLN(20,21,IX,IY)
      CALL PLOTLN(22,23,IX,IY)
      CALL PLOTLN(24,25,IX,IY)
      CALL PLOTLN(26,27,IX,IY)
      CALL PLOTAR(2,24,IX,IY)
      CALL PLOTAR(1,26,IX,IY)
      CALL PLOTLN(12,28,IX,IY)
      CALL PLOTAR(1,12,IX,IY)
      CALL DSHLNV(IX(28),IY(28),IX(29),IY(29),8,8)
      CALL PLOTLN(29,30,IX,IY)
      CALL DASHLN(31,32,IX,IY)
      CALL PLOTLN(33,34,IX,IY)
      CALL PLOTLN(35,36,IX,IY)
      CALL PLOTLN(37,38,IX,IY)
```

P010   (19) (PLOTF2)

```
      CALL PLOTLN(39,40,IX,IY)
      CALL PLOTLN(41,42,IX,IY)
      CALL PLOTLN(43,44,IX,IY)
C
      ISCALE=1
      X1=X(34)
      Y1=Y(34)
      Y2=Y1-.05
      IDY=5
   20 CALL YSCLVI(Y1,IY1,IERR)
      CALL YSCLVI(Y2,IY2,IERR)
      STRESS=40.
      IYS=IY1+IDY
      DO 21 I=1,9
      X2=X1
      CALL XSCLVI(X1,IX1,IERR)
      CALL XSCLVI(X2,IX2,IERR)
      CALL LINEV(IX1,IY1,IX2,IY2)
      IXS=IX1-8
      CALL LABLV(STRESS,IXS,IYS,3,1,2)
      X1=X1-.25
      STRESS=STRESS-10.
   21 CONTINUE
      ISCALE=ISCALE+1
      IF(ISCALE.EQ.2) GO TO 22
      IF(ISCALE.EQ.3) GO TO 23
      IF(ISCALE.EQ.4) GO TO 24
   22 X1=X(37)
      Y1=Y(37)
      Y2=Y1-.05
      IDY=5
      GO TO 20
   23 X1=X(42)
      Y1=Y(42)
      Y2=Y1+.05
      IDY=-21
      GO TO 20
   24 CONTINUE
C
      DIMENSION D1(3),D2(3),D3(3),D4(3),D5(3),D6(3)
     *          ,D7(3),D8(3),D9(3),D10(3),D11(3)
     *          ,D12(4),D13(4),D14(4),D15(4)
      DIMENSION D101(3) , D102(3) . D103(3) , D104(3)
C
      DATA( D1(I),I=1,3)/18H PHASE            /
      DATA( D2(I),I=1,3)/18H BOLT FORCE       /
      DATA( D3(I),I=1,3)/18H   (LB)           /
      DATA( D4(I),I=1,3)/18H APPLIED MOMENT   /
      DATA( D5(I),I=1,3)/18H   (IN-LB)        /
      DATA( D6(I),I=1,3)/18H AXIAL DISPLACEMT /
      DATA( D7(I),I=1,3)/18H   (IN)           /
      DATA( D8(I),I=1,3)/18H ROTATION         /
      DATA( D9(I),I=1,3)/18H   (RAD)          /
      DATA(D10(I),I=1,3)/18H                  /
      DATA(D11(I),I=1,3)/18H                  /
      DATA(D12(I),I=1,4)/24H STRESSES ON INSIDE     /
```

P010   (19) (PLOTF2)

```
      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    /
C
      CALL PRINTV(18,D1,190,151)
      CALL PRINTV(18,D2,350,151)
      CALL PRINTV(18,D3,350,131)
      CALL PRINTV(18,D4,510,151)
      CALL PRINTV(18,D5,510,131)
      CALL PRINTV(18,D6,670,151)
      CALL PRINTV(18,D7,670,131)
      CALL PRINTV(18,D8,830,151)
      CALL PRINTV(18,D9,830,131)
      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)
C
      IXP=IX(38)+20
      IYP=IY(38)+30
      CALL PRINTV(24,D12,IXP,IYP)
      IXP=IX(33)+20
      IYP=IY(33)+30
      CALL PRINTV(24,D13,IXP,IYP)
      IXP=IX(41)+20
      IYP=IY(41)-40
      CALL PRINTV(24,D14,IXP,IYP)
      CALL PRINTV(24,D15,400,961)
      CALL LINEV(400,951,568,951)
      CALL LINEV(400,947,568,947)
C
      DIMENSION DATA1(1)
      DATA(DATA1(I),I=1,1)/6H   DIA/
C
      CALL PLOTLB(28,0,5,IX,IY,D1)
      CALL PLOTTX(28,32,5,IX,IY,DATA1,6)
      CALL PLOTLB(18,10,25,IX,IY,H)
      CALL PLOTLB(9,10,-20,IX,IY,B)
      CALL PLOTLB(21,10,10,IX,IY,T)
C
      BFC=AP(1)
      EAM=AP(2)
      ADFL=AP(3)
      WP0=AP(4)
      CHI0=AP(5)
      EN0=AP(6)
      EM0=AP(7)
      EQ0=AP(8)
      SSCALE=.000025
      WSCALE=10.
      R0=RI+T/2.
      AL = HTS *2.0
      DL=0.05
      NL=AL/DL
      I=IPHASE
```

```
      AK2=AK*AK
      BEND=ET*T**3/(12.*(1.-ANUT**2))
      CWF=.5/(AK2*AK*BEND)
C
      DO 100 K=1,NL
      EK=K
      XK=(EK-1.)*DL
      SX=SIN(AK*XK)
      CX=COS(AK*XK)
      EX=EXP(-AK*XK)
      WX=CWF*EX*(EQ0*CX-AK*EM0*(CX-SX))+WP0
      EMX= EX*(EM0*(CX+SX)-EQ0*SX/AK)
      EQX= EX*(EQ0*(SX-CX)+2.*AK*EM0*SX)
      IF(K.EQ.1) WTOP=WX
      ENX=END
      ENY=ET*T*WX/R0
      IF(IPHASE.EQ.2) ENY=ENY-ET*T*ALFAT*DELTAT+P*R0*ANUT/2.
      IF(IPHASE.EQ.3) ENY=ENY+P*R0*ANUT/2.
      IF(IPHASE.EQ.4) ENY=ENY+PF*P*R0*ANUT/2.
      EMY= ANUT*EMX
C
      IF(KOPT(2).EQ.1) GO TO 202
      SXI=-6.*EMX/T**2+ENX/T
      SX0=+6.*EMX/T**2+ENX/T
      SYI=-6.*EMY/T**2+ENY/T
      SY0=+6.*EMY/T**2+ENY/T
      TMAX=1.5*EQX/T
      GO TO 205
  202 IF(XK.LE.RFIL) GO TO 203
      TX=T
      GO TO 204
  203 XP=RFIL-XK
      YP=SQRT(RFIL**2-XP**2)
      TX=T+RFIL-YP
  204 CONTINUE
      SXI=-6.*EMX/TX**2+ENX/TX
      SX0= 6.*EMX/TX**2+ENX/TX
      SYI=-6.*EMY/TX**2+ENY/TX
      SY0= 6.*EMY/TX**2+ENY/TX
      TMAX=1.5*EQX/TX
  205 CONTINUE
C
      YS=Y(2)+XK/SCALE
      XSXI=X(39)+SSCALE*SXI
      XSYI=X(39)+SSCALE*SYI
      XSX0=X(35)+SSCALE*SX0
      XSY0=X(35)+SSCALE*SY0
      XTMAX=X(6)-SSCALE*TMAX
      XWX=X(6)+WSCALE*WX
C
      CALL YSCLV1(YS,IYS,IERR)
      CALL XSCLV1(XSXI,IXSXI,IERR)
      CALL XSCLV1(XSYI,IXSYI,IERR)
      CALL XSCLV1(XSX0,IXSX0,IERR)
      CALL XSCLV1(XSY0,IXSY0,IERR)
      CALL XSCLV1(XTMAX,IXTMAX,IERR)
```

```
      CALL XSCLVI(XWX,IXWX,IERR)
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
   98 CALL PLOTV(IXSXI,IYS,29,0)
      CALL PLOTV(IXSYI,IYS,30,0)
      CALL PLOTV(IXSXO,IYS,29,0)
      CALL PLOTV(IXSYO,IYS,30,0)
      CALL PLOTV(IXTMAX,IYS,25,0)
      CALL PLOTV(IXWX,IYS,28,0)
   99 CALL PLOTV(IXSXI,IYS,35,0)
      CALL PLOTV(IXSYI,IYS,35,0)
      CALL PLOTV(IXSXO,IYS,35,0)
      CALL PLOTV(IXSYO,IYS,35,0)
      CALL PLOTV(IXTMAX,IYS,35,0)
      CALL PLOTV(IXWX,IYS,35,0)
  100 CONTINUE
C
      WBOT=WTOP-H*CHIO
      XWBOT=X(7)+WBOT
      XWTOP=X(7)+WTOP
      SIGTOP=EF*WTOP/RO
      SIGBOT=EF*WBOT/RO
      YSTOP=Y(2)
      YSBOT=Y(1)
      XSTOP=X(43)+SSCALE*SIGTOP
      XSBOT=X(43)+SSCALE*SIGBOT
C
      CALL XSCLVI(XSTOP,IXSTOP,IERR)
      CALL YSCLVI(YSTOP,IYSTOP,IERR)
      CALL XSCLVI(XSBOT,IXSBOT,IERR)
      CALL YSCLVI(YSBOT,IYSBOT,IERR)
      CALL XSCLVI(XWBOT,IXWBOT,IERR)
      CALL XSCLVI(XWTOP,IXWTOP,IERR)
C
      CALL PLOTV(IXSTOP,IYSTOP,30,0)
      CALL PLOTV(IXSBOT,IYSBOT,30,0)
      CALL LINEV(IX(44),IY(44),IXSTOP,IYSTOP)
      CALL LINEV(IX(43),IY(1),IXSBOT,IYSBOT)
      CALL LINEV(IXSTOP,IYSTOP,IXSBOT,IYSBOT)
      CALL PLOTV(IXWBOT,IY(1),28,0)
      CALL DSHLNV(IXWBOT,IY(1),IXWTOP,IY(7),5,15)
      DATA(D101(I),I=1,3)/18H BOLT-UP          /
      DATA(D102(I),I=1,3)/18H START-UP         /
      DATA(D103(I),I=1,3)/18H OPERATION        /
      DATA(D104(I),I=1,3)/18H SHUT-DOWN        /
      IF(IPHASE.EQ.1) CALL PRINTV(18,D101,190,76)
      IF(IPHASE.EQ.2) CALL PRINTV(18,D102,190,76)
      IF(IPHASE.EQ.3) CALL PRINTV(18,D103,190,76)
      IF(IPHASE.EQ.4) CALL PRINTV(18,D104,190,76)
C
      CALL LABLV(BFC,380,76,-6,1,1)
      CALL LABLV(EAM,540,76,-6,1,1)
      CALL LABLV(ADFL,700,76,-6,1,1)
```

```
P010      (19) (PLOTF2)

               CALL LABLV(CHI0.860.76,-6.1,1)
          C
               RETURN
               END


@HDG  D200      (20) (OUTAN)


@PRT,C  D200
FURPUR 24H1-03/10-14:53
```

200 (20) (OUTAN)

434600*TPFS.D200

```
      SUBROUTINE OUTAN(HEAD,A,SR,S)
C
C**** PRINT-OUT ANALYSIS RESULTS
C     K.R.LEIMBACH, 4 JANUARY 1973
C
      DIMENSION A(9,4),SR(5,4),S(5,4),HEAD(12)
C
  101 FORMAT(1H1)
  102 FORMAT(12A6)
  103 FORMAT(/' OUTPUT OF THE STRESS AND DEFORMATION ANALYSIS RESULTS'/
  104 FORMAT(/' VARIABLE ',30X,' BOLT-UP         '
     *                          ,' START-UP        '
     *                          ,' OPERATION       '
     *                          ,' SHUT-DOWN       '//)
  201 FORMAT(16H BOLT FORCE (LB),24X,4E16.8/
     *          29H EQUIV APPL MOMENT (IN-LB/IN),11X,4E16.8/
     *          22H AXIAL DEFLECTION (IN),18X,4E16.8/
     *          23H RADIAL DEFLECTION (IN),17X,4E16.8/
     *          25H FLANGE ROTATION (RADIAN),15X,4E16.8/
     *          18H BOLT STRESS (PSI),22X,4E16.8/
     *          20H GASKET STRESS (PSI),20X,4E16.8/
     *          27H STRESS IN FLANGE TOP (PSI),13X,4E16.8/
     *          30H STRESS IN FLANGE BOTTOM (PSI),10X,4E16.8/)
  202 FORMAT(40H STRESS RESULTANTS          (LB/IN)      NX=,4E16.8/
     *          40H                           (LB/IN)      NY=,4E16.8/
     *          40H                        (IN-LB/IN)      MX=,4E16.8/
     *          40H                        (IN-LB/IN)      MY=,4E16.8/
     *          40H                           (LB/IN)      WX=,4E16.8/)
  203 FORMAT(40H STRESSES AT NECK         (PSI) INNER SIGX=,4E16.8/
     *          40H                         (PSI) INNER SIGY=,4E16.8/
     *          40H                         (PSI) OUTER SIGX=,4E16.8/
     *          40H                         (PSI) OUTER SIGY=,4E16.8/
     *          40H                            (PSI) MAX TAU=,4E16.8/)
C
      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)
C
      RETURN
      END
```

@FIN

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
Tube and Flange Material:	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

$$t = \frac{1.5 \times 10^3 \times 4.000}{35 \times 10^3/1.5 - 1.5 \times 10^3/2.0} = \frac{6.0 \times 10^3}{23.3 \times 10^3 - 0.75 \times 10^3}$$

$$= \frac{6.0 \times 10^3}{22.55} = 0.266 \text{ inch}$$

For higher internal pressures this difference is more distinct.

(b) Initial guess of bolt size

$d_B = 0.4375$ 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

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

Spot face diameter

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

Fillet radius for spot face

$$r_{spot} = 0.062 \text{ inch}$$

(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

$$(\text{diam})_{F\phi} = 8.000 + 2.696 = 10.696 \text{ inch}$$

(e) Gasket width and gasket radius

Estimate for gasket radius

$$r_G = 1/2\ (4.950 - 0.221 + 4.000)$$
$$= 1.2\ (8.729) = 4.3645 \text{ inch}$$

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 $c_2 = 0.05$ 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

$$P_B^{(1)} = 2\pi \times 4.518 \times 0.322 \times 1.0 \times 40.0 \times 10^3$$
$$= 6.28 \times 1.455 \times 40.0 \times 10^3$$
$$= 9.1374 \times 40.0 \times 10^3 = 365.5 \times 10^3 \text{ lb}$$

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

(g) Number of bolts

$$n_{B1} = \frac{366.9 \times 10^3}{131 \times 10^3 \times 0.10631}$$

$$= \frac{366.9 \times 10^3}{13.93 \times 10^3} = 26.3 \approx 26 \text{ bolts}$$

$$n_{B2} = \frac{(2.0/1.5)\ 366.9 \times 10^3}{200.0 \times 0.10631} \approx \frac{488 \times 10^3}{21.3 \times 10^3}$$

$$= 22.9 \approx 23 \text{ bolts}$$

n_B = 26 bolts are required.

(h) Bolt spacing

$$s = \frac{2\pi\ 4.518}{26} = \frac{6.28 \times 4.518}{26} = \frac{28.37}{26} = 1.09 \text{ inch}$$

Minimum allowable spacing is

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

Maximum allowable bolt spacing

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

(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

$$t_N = 0.438/2 = 0.219 \text{ inch}$$

Untimate moment to be carried

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

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

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

Effective flange width

$$\bar{b} = 1.348 - 0.442\sqrt{0.442/1.09}$$

$$= 1.348 - 0.442\sqrt{0.406}$$

$$= 1.348 - 0.442 \times 0.637$$

$$= 1.348 - 0.282 = 1.066 \text{ inch}$$

Assume

$$\zeta_1 = 0.8$$
$$\zeta_2 = 0.18$$

The coefficients of the quadratic equation for h are

$$A = 35 \times 10^3 \frac{1.066}{6 \times 4.219} = \frac{37.31}{25.31} \times 10^3 = 1.474 \times 10^3$$

$$B = 35 \times 10^3 \times 0.18 \times \frac{0.438 - 0.219}{2} = 35 \times 0.0197 \times 10^3$$

$$= 0.6895 \times 10^3$$

$$C = 35 \times 10^3 \times 0.8 \times \frac{(0.438)^2 - (0.219)^2}{4} - 8.97 \times 10^3$$

$$= 35 \times 10^3 \times 0.2 \times (0.192 - 0.048) - 8.97 \times 10^3$$

$$= (1.008 - 8.97) \times 10^3 = -7.962 \times 10^3$$

$$R^2 = \left[(0.6895)^2 + 4 \times 1.474 \times 7.962\right] \times 10^6$$
$$= \left[0.4754 + 5.896 \times 7.962\right] \times 10^6$$
$$= \left[0.4754 + 46.9440\right] \times 10^6 = 47.42 \times 10^6$$

$$R = 6.886 \times 10^3$$

$$h = \frac{6.886 - 0.690}{2 \times 1.474} = \frac{6.196}{2.948} = 2.102 \text{ inch}$$

If the contribution of the plastic hinge is neglected, i.e, if $\zeta_1 = \zeta_2 = 0$ 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

$$r_w = 4.000 + \frac{0.438 + 1.348}{2}$$

$$= 4.000 + 0.893 = 4.893 \text{ inch}$$

Volume

$$Vol = 2\pi \times 4.893 \times 1.913 = 58.78 \text{ in.}^3$$

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 σ_x are indicated by the curve labeled "X," the circumferential stresses σ_φ by "Y" and the transverse shear stresses, τ_{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.

TEST FLANGE 1500 PSI, 8 IN DIA

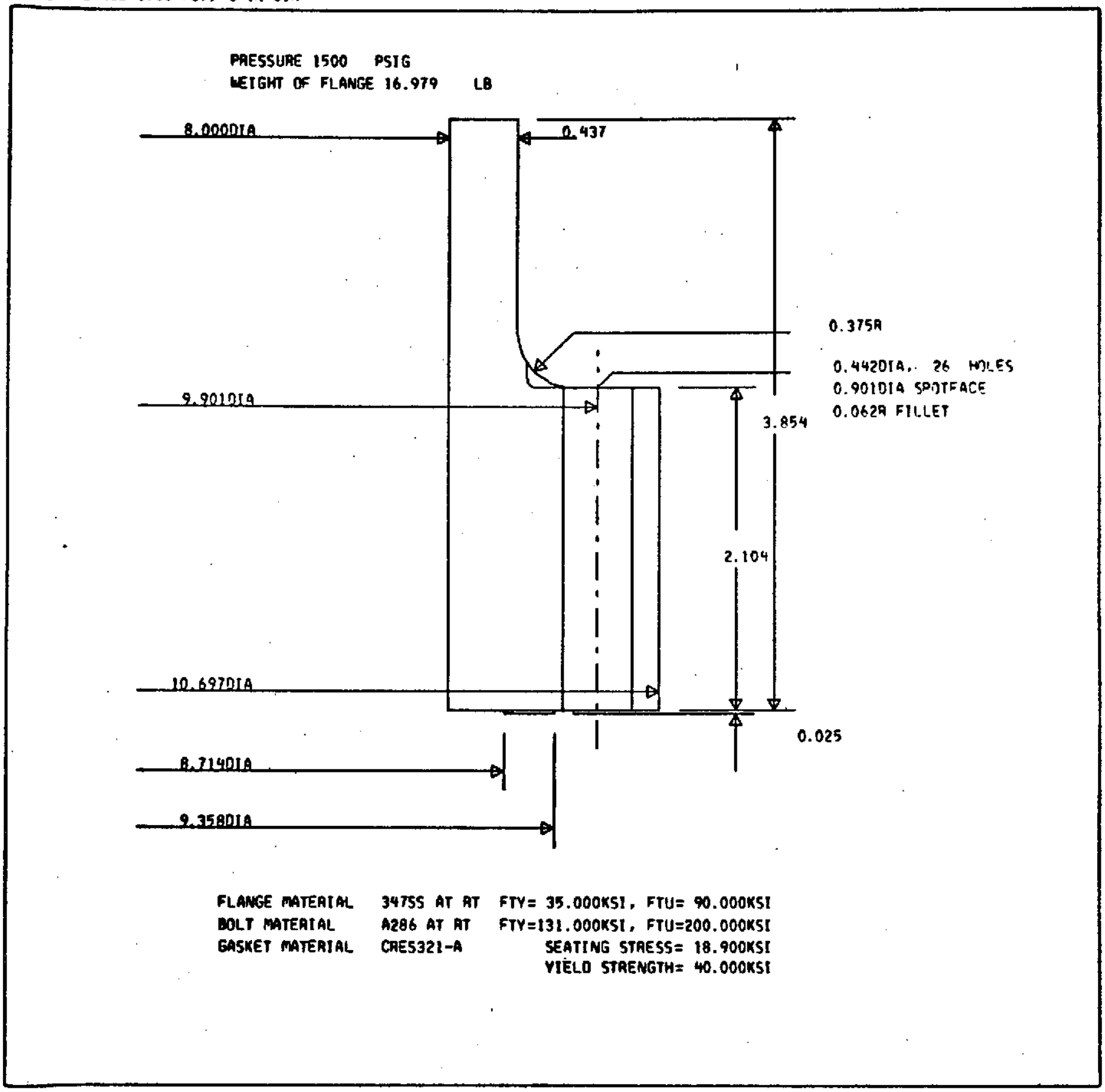
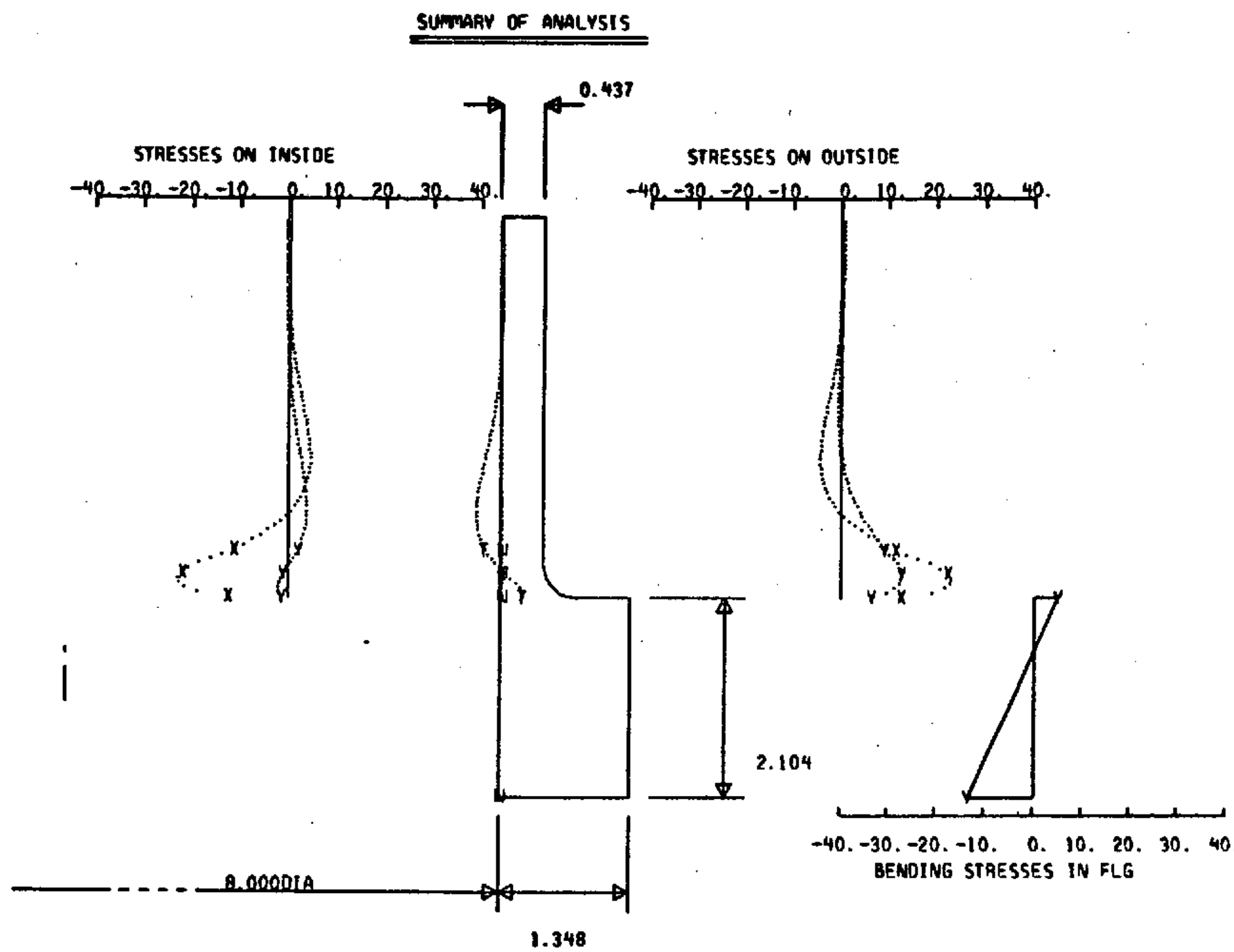


Fig. 5-1 - Design Example

TEST FLANGE 1500 PSI, 8 IN DIA



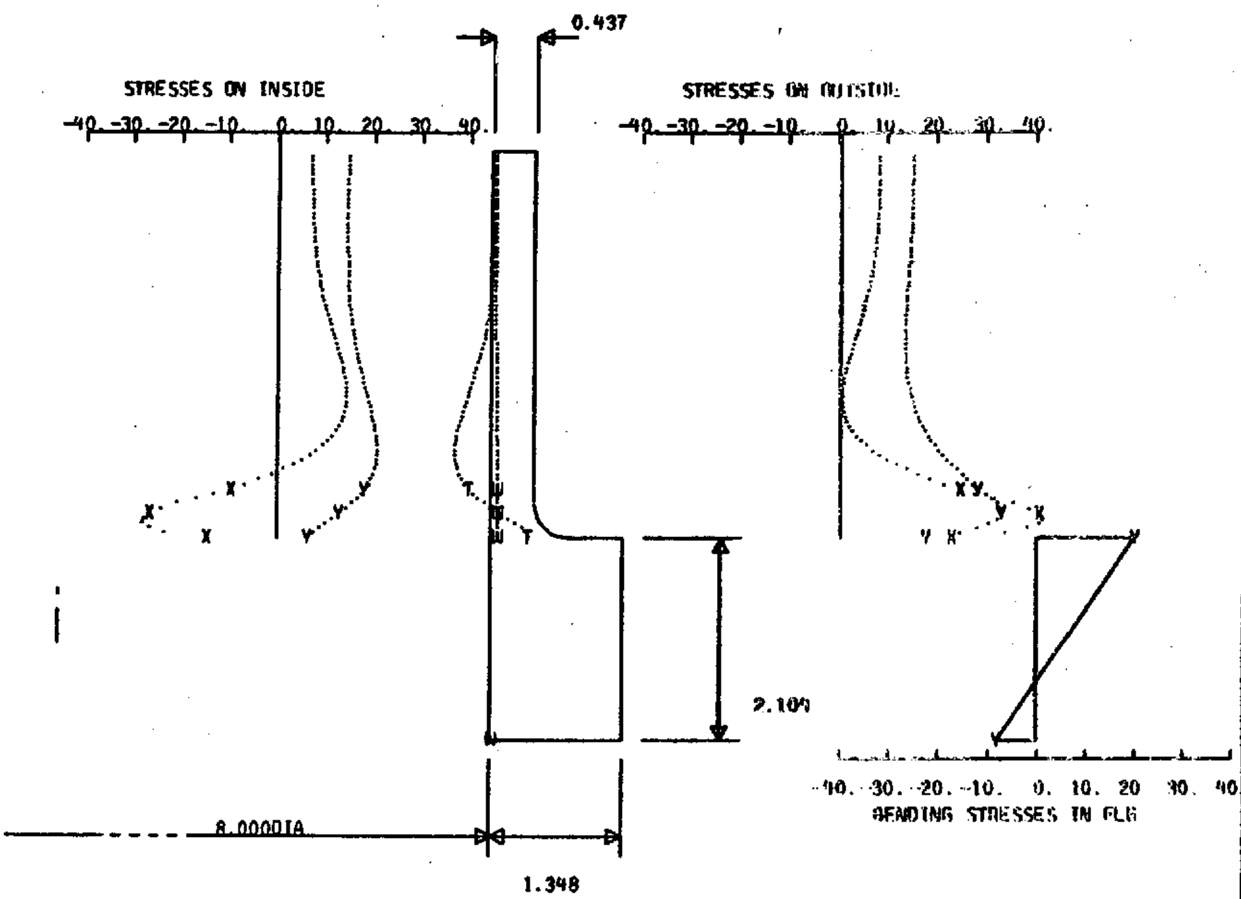
PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMENT (IN)	ROTATION (RAD)
BOLT-UP	$3.70613 \times 10^{+05}$	$6.04409 \times 10^{+03}$	0.00000×10^{-01}	1.33011×10^{-03}

Fig. 5-2 - Stresses at Initial Torquing

Legend: X = σ_X axial stress (ksi)
Y = σ_φ circumferential stress (ksi)
T = τ_{XZ} transverse shear stress (ksi)
W = radial displacement (10-fold magnified)

TEST FLANGE 1500 PSI, 8 IN DIA

SUMMARY OF ANALYSIS



PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMENT (IN)	ROTATION (RAD)
OPERATION	$3.60117X10^{+05}$	$9.15526X10^{+03}$	$9.52960X10^{-06}$	$2.01478X10^{-03}$

Fig. 5-3 - Stresses at 1500 psi (proof pressure)

FLANGE PARAMETRIC CASE 1 DIA= 4IN PRESS = 100 PSI

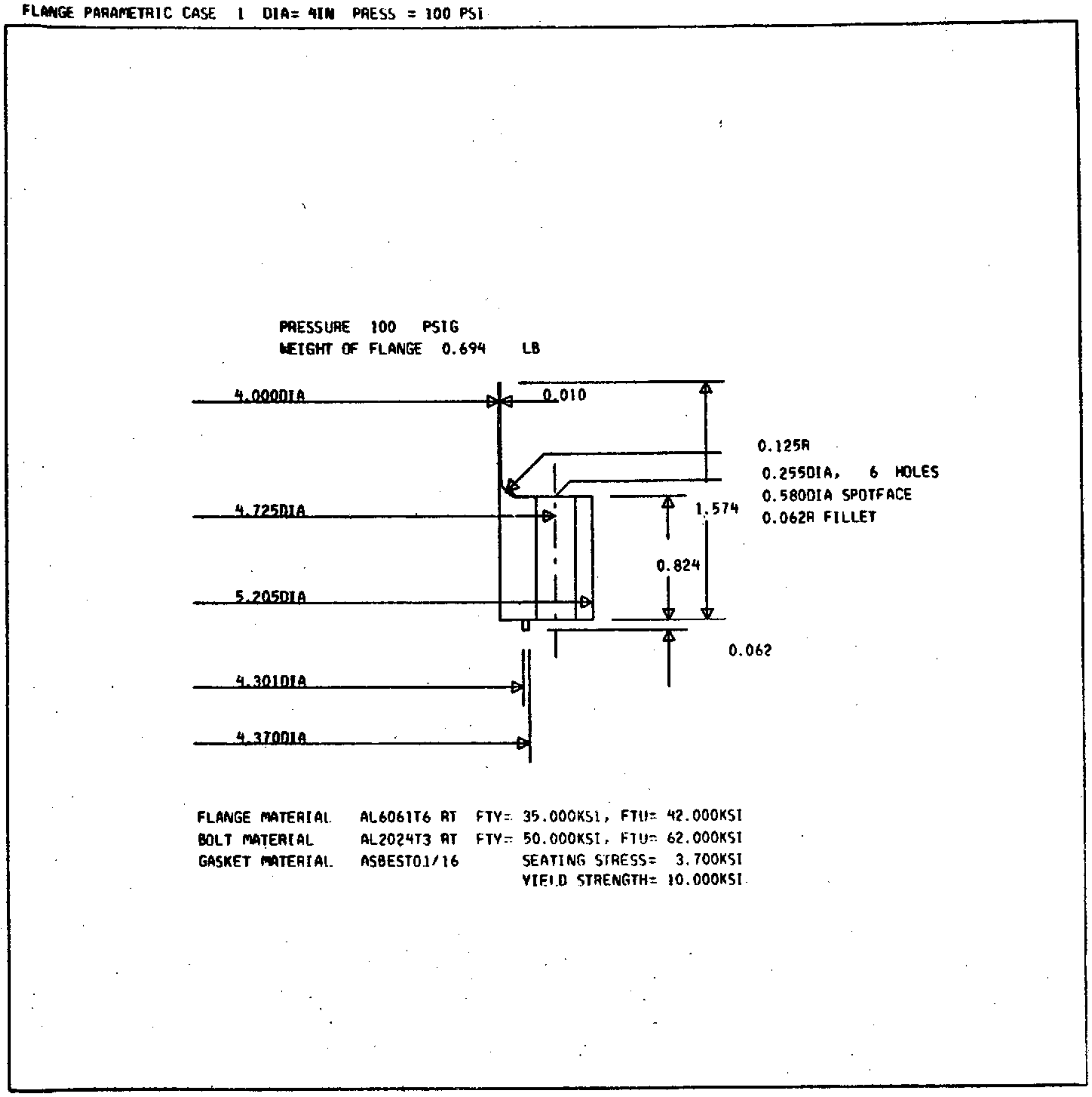
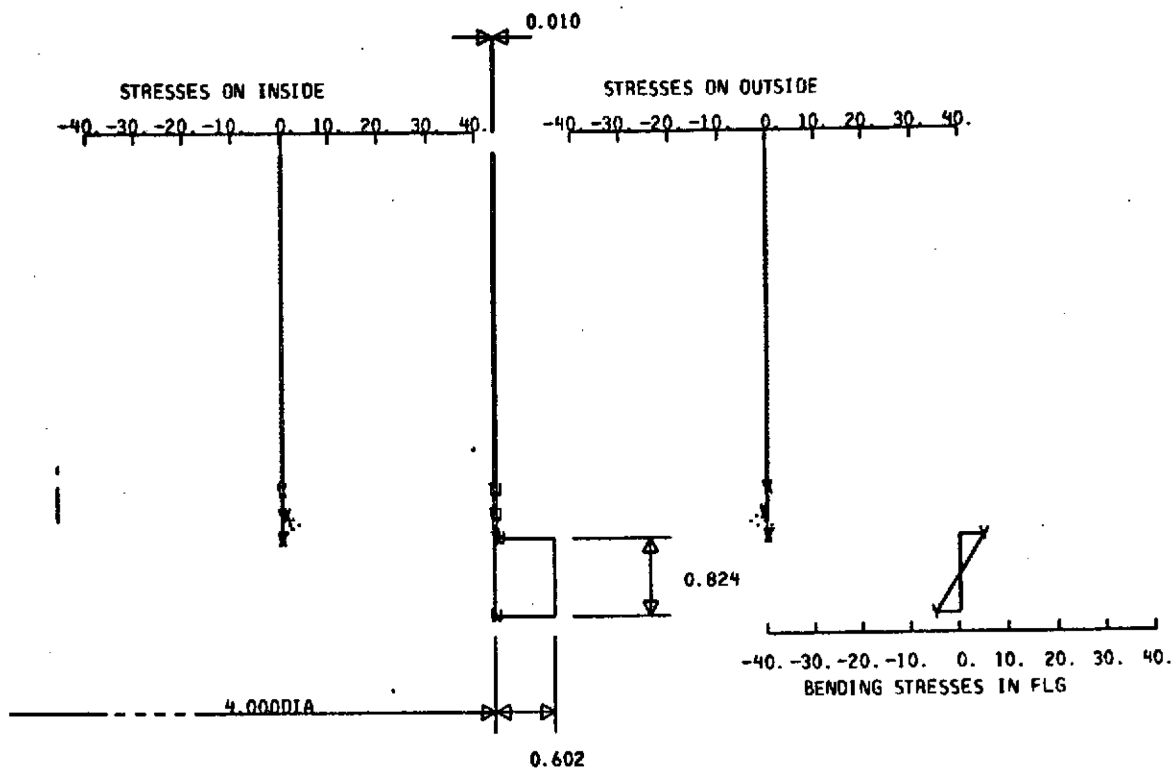


Fig. 5-4 - Flange 1, Design

FLANGE PARAMETRIC CASE 1 DIA= 4IN PRESS = 100 PSI

SUMMARY OF ANALYSIS

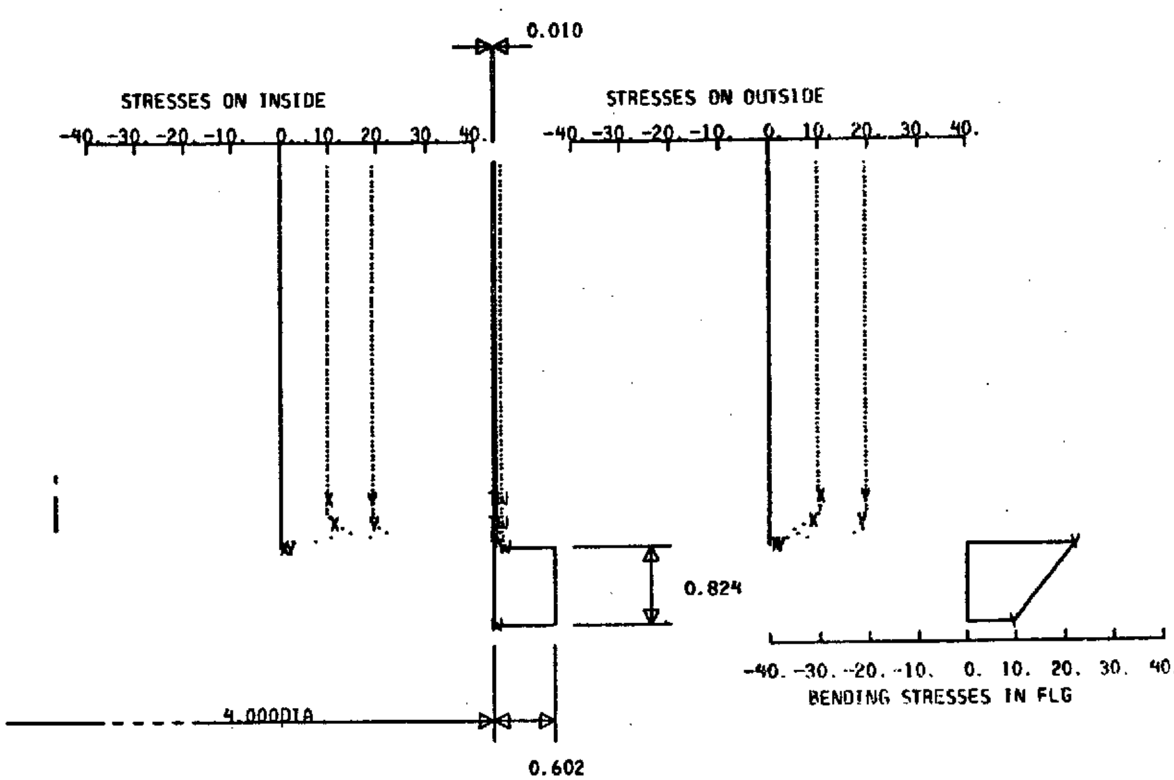


PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
BOLT-UP	$9.54600 \times 10^{+03}$	$1.47608 \times 10^{+02}$	0.00000×10^{-01}	2.37347×10^{-03}

Fig. 5-5 - Flange 1, Stresses at Initial Torquing

FLANGE PARAMETRIC CASE 1 DIA= 4IN PRESS = 100 PSI

SUMMARY OF ANALYSIS

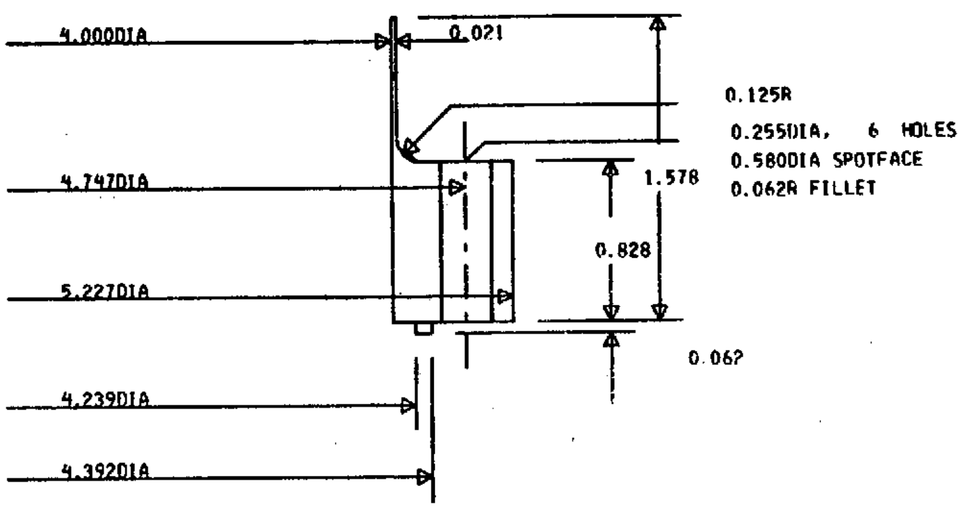


PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMNT (IN)	ROTATION (RAD)
OPERATION	$1.07769 \times 10^{+04}$	$1.96157 \times 10^{+02}$	4.56443×10^{-04}	3.15410×10^{-03}

Fig. 5-6 - Flange 1, Stresses at Proof Pressure

FLANGE PARAMETRIC CASE 2 DIA= 4IN PRESS = 200 PSI

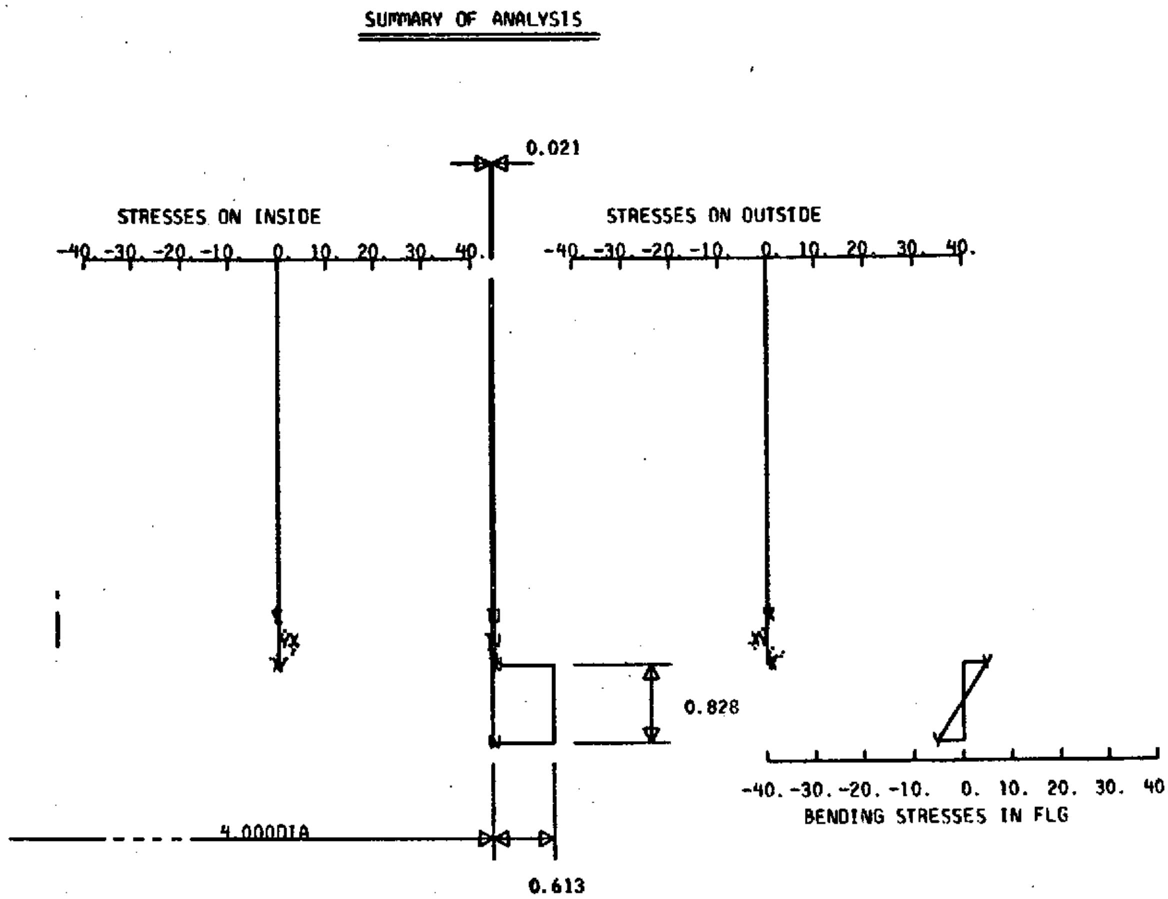
PRESSURE 200 PSIG
WEIGHT OF FLANGE 0.700 LB



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. 5-7 - Flange 2, Design

FLANGE PARAMETRIC CASE 2 DIA= 4IN PRESS = 200 PSI

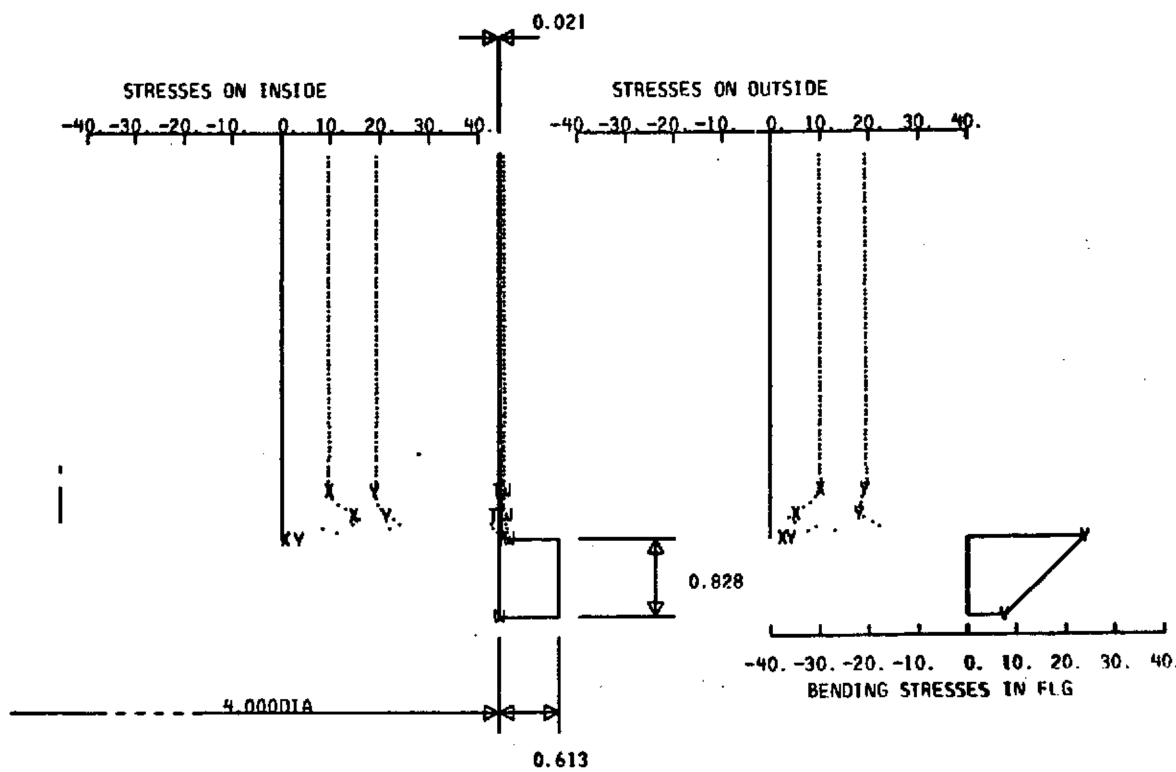


PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
BOLT-UP	$9.54600 \times 10^{+03}$	$1.62969 \times 10^{+02}$	0.00000×10^{-01}	2.42379×10^{-03}

Fig. 5-8 - Flange 2, Stresses at Initial Torquing

FLANGE PARAMETRIC CASE 2 DIA= 4IN PRESS = 200 PSI

SUMMARY OF ANALYSIS



PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
OPERATION	$1.15301 \times 10^{+04}$	$2.62247 \times 10^{+02}$	9.46790×10^{-04}	3.90032×10^{-03}

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.

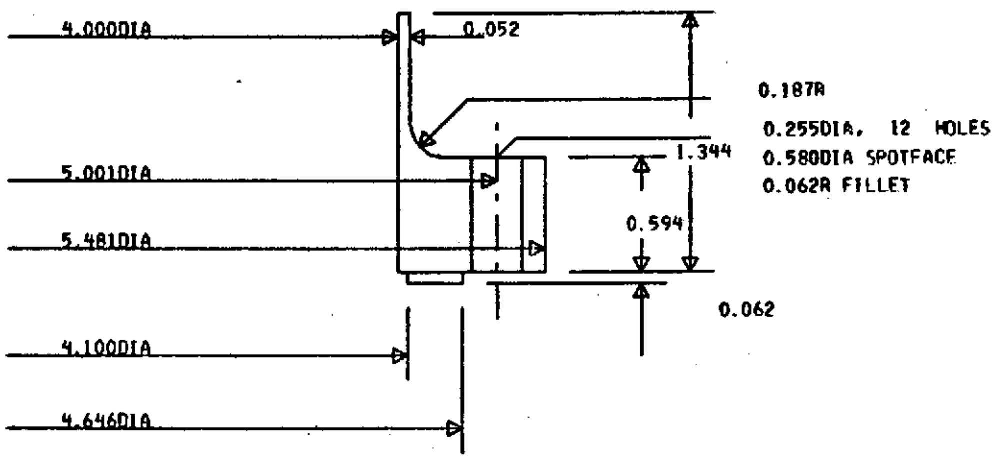
The flange shown on Figs. 5-22 through 5-24 is well proportioned and the stresses are well under the allowable stresses, although here as before bolt spacing is the controlling factor. Figs. 5-25 through 5-26 show a flange designed for the same inner diameter but twice the pressure. This is a strength-controlled 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= 4IN PRESS = 500 PSI

PRESSURE 500 PSIG
WEIGHT OF FLANGE 0.603 LB

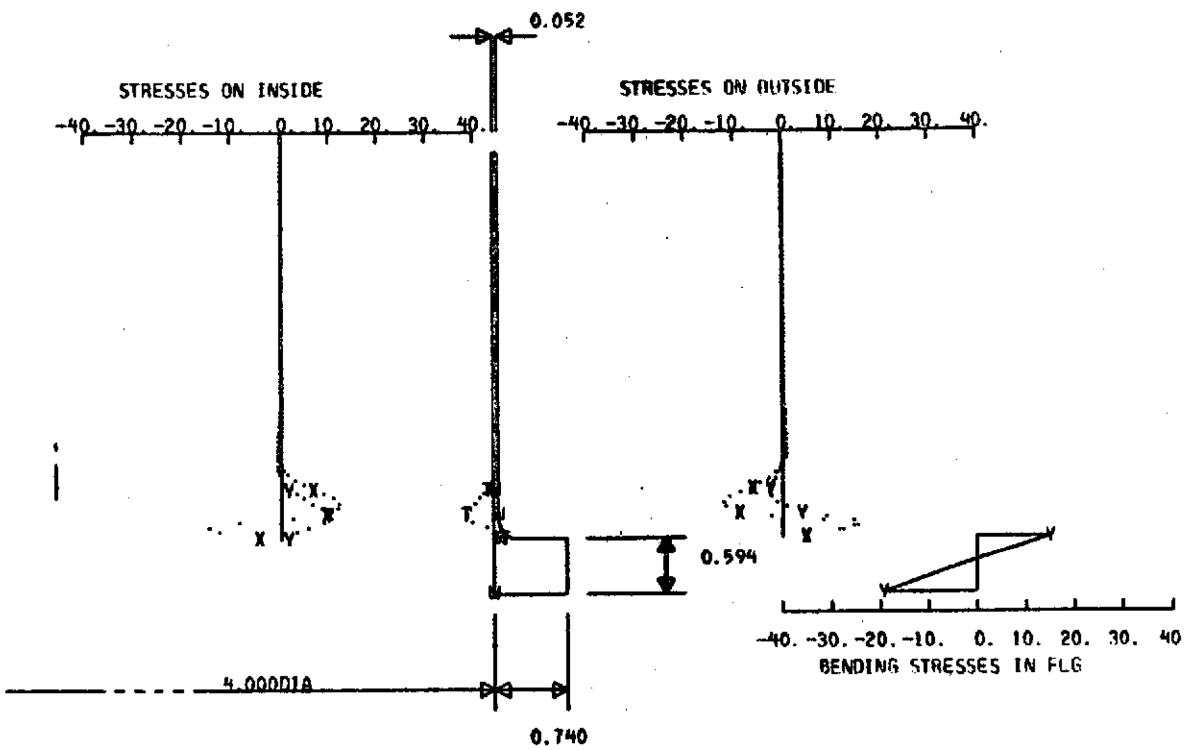


FLANGE MATERIAL	AL6061T6 RT	FTY= 35.000KSI,	FTU= 42.000KSI
BOLT MATERIAL	AL2024T3 RT	FTY= 50.000KSI,	FTU= 62.000KSI
GASKET MATERIAL	ASBESTO1/16	SEATING STRESS=	3.700KSI
		YIELD STRENGTH=	10.000KSI

Fig. 5-10 - Flange 3, Design

FLANGE PARAMETRIC CASE 3 DIA= 4IN PRESS = 500 PSI

SUMMARY OF ANALYSIS

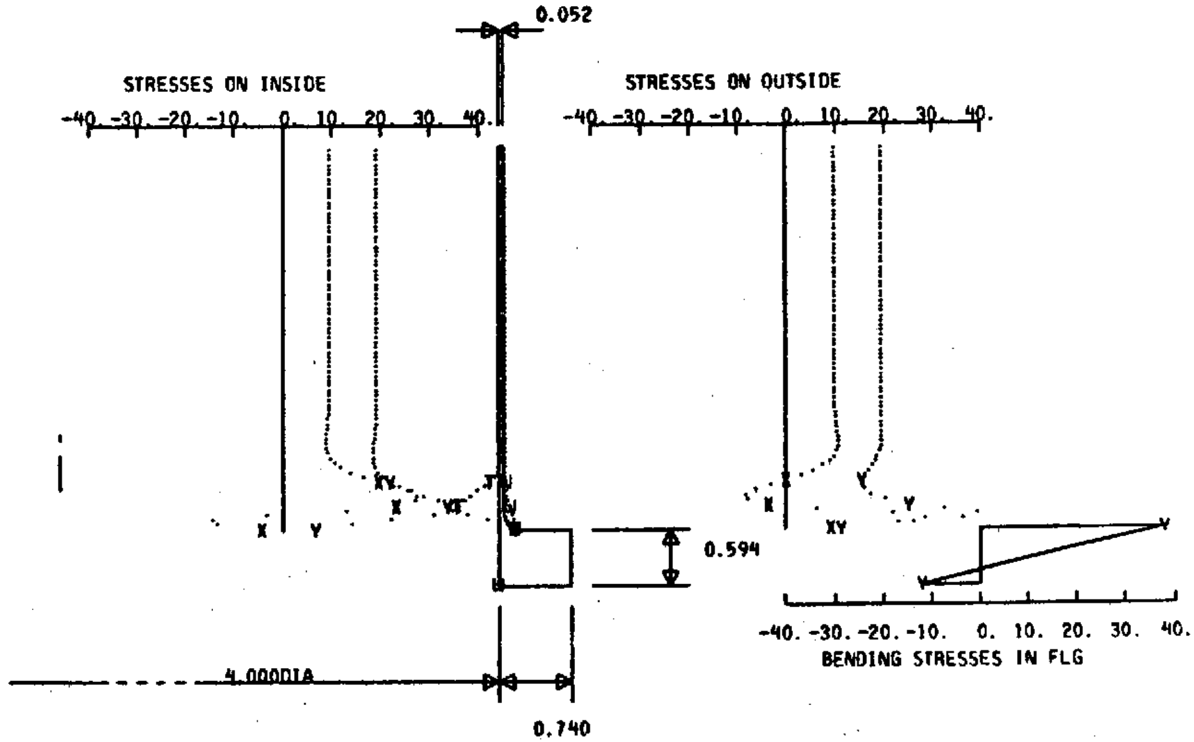


PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
BOLT-UP	$1.90075 \times 10^{+04}$	$4.68911 \times 10^{+02}$	0.00000×10^{-01}	1.16959×10^{-02}

Fig. 5-11 - Flange 3, Stresses at Initial Torquing

FLANGE PARAMETRIC CASE 3 DIA= 4IN PRESS = 500 PSI

SUMMARY OF ANALYSIS



PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
OPERATION	$2.05749 \times 10^{+04}$	$6.88662 \times 10^{+02}$	7.51977×10^{-04}	1.71770×10^{-02}

Fig. 5-12 - Flange 3, Stresses at Proof Pressure

FLANGE PARAMETRIC CASE 7 DIA= 8IN PRESS = 100 PSI

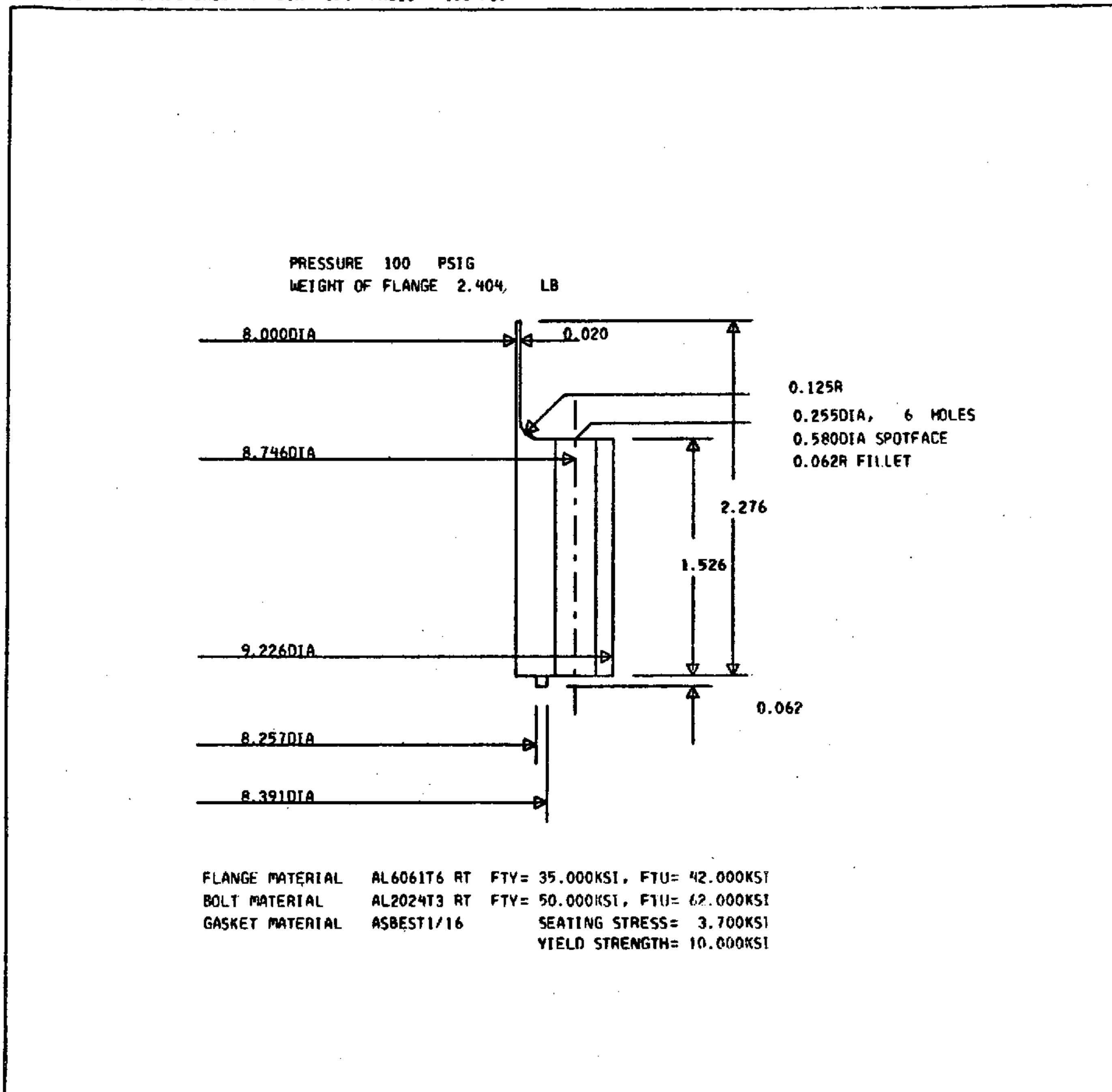
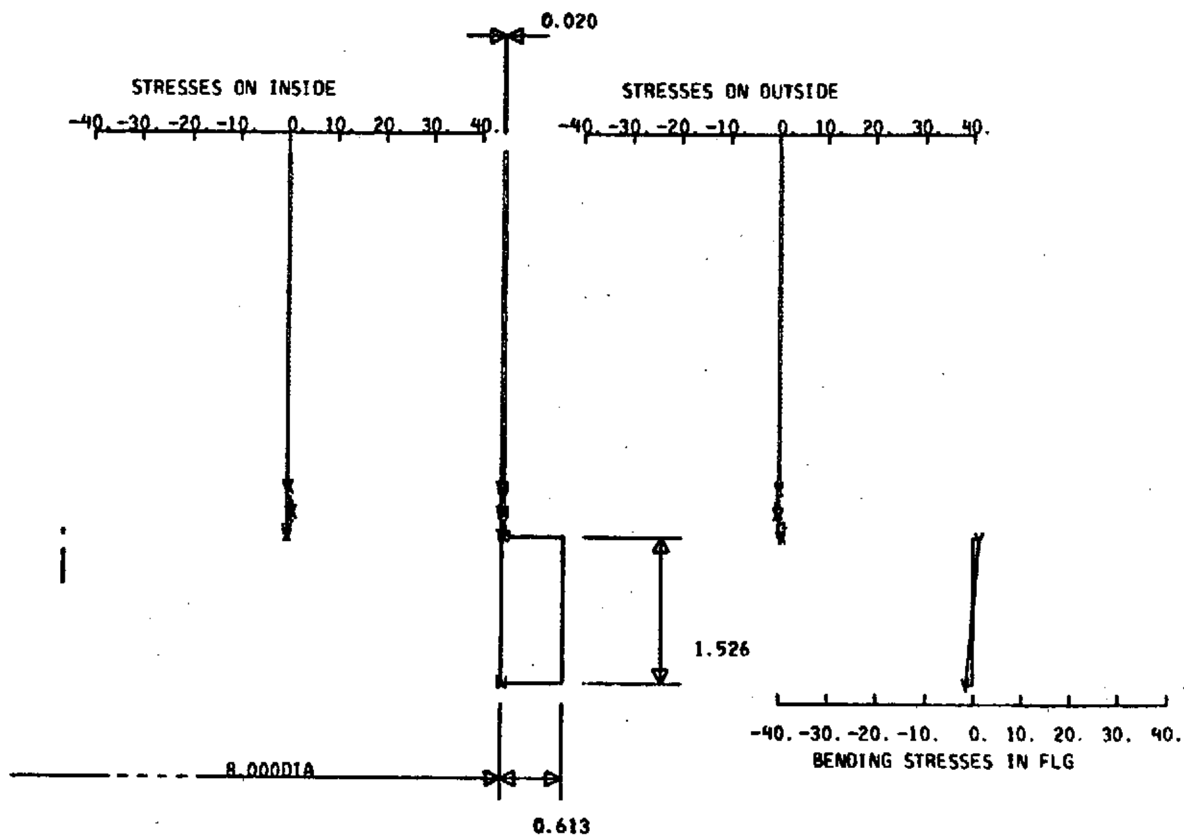


Fig. 5-13 - Flange 7, Design

FLANGE PARAMETRIC CASE 7 DIA= 8IN PRESS = 100 PSI

SUMMARY OF ANALYSIS

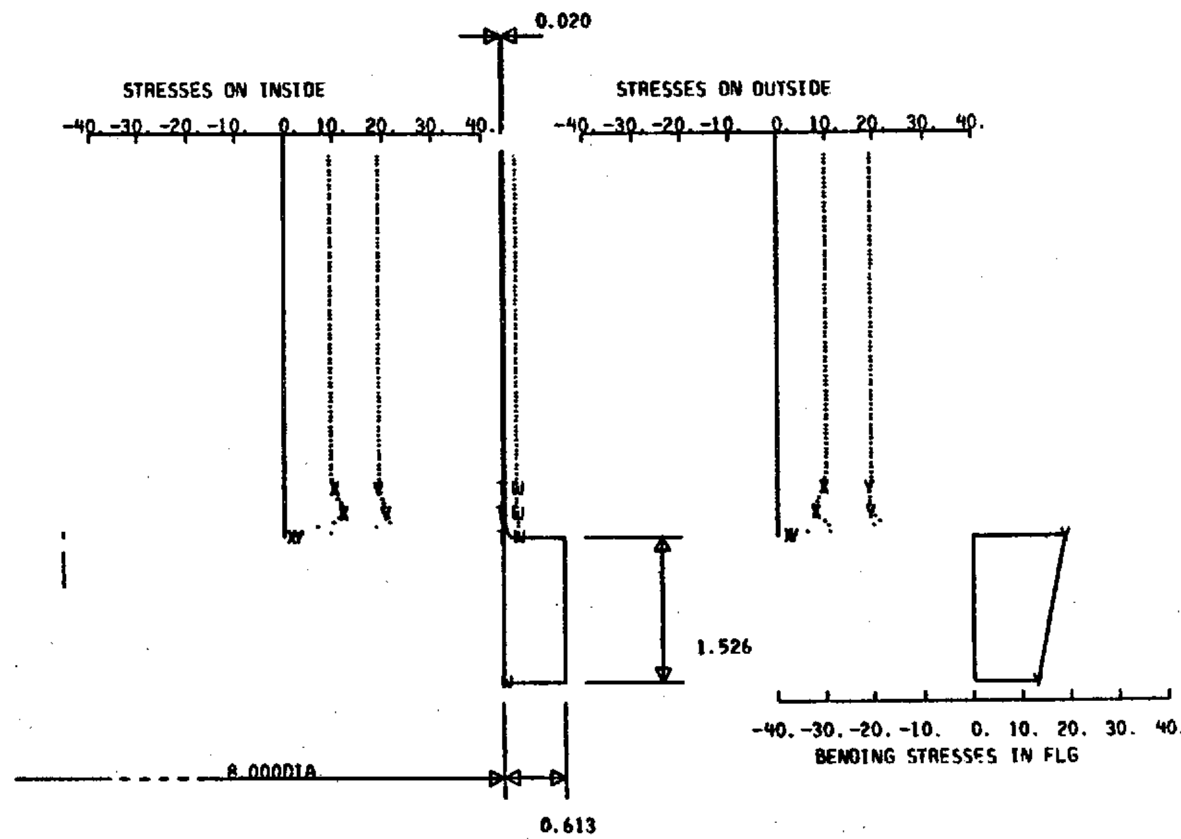


PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
BOLT-UP	$8.89140 \times 10^{+03}$	$7.45269 \times 10^{+01}$	0.00000×10^{-01}	6.76743×10^{-04}

Fig. 5-14 - Flange 7, Stresses at Initial Torquing

FLANGE PARAMETRIC CASE 7 DIA= 8IN PRESS = 100 PSI

SUMMARY OF ANALYSIS

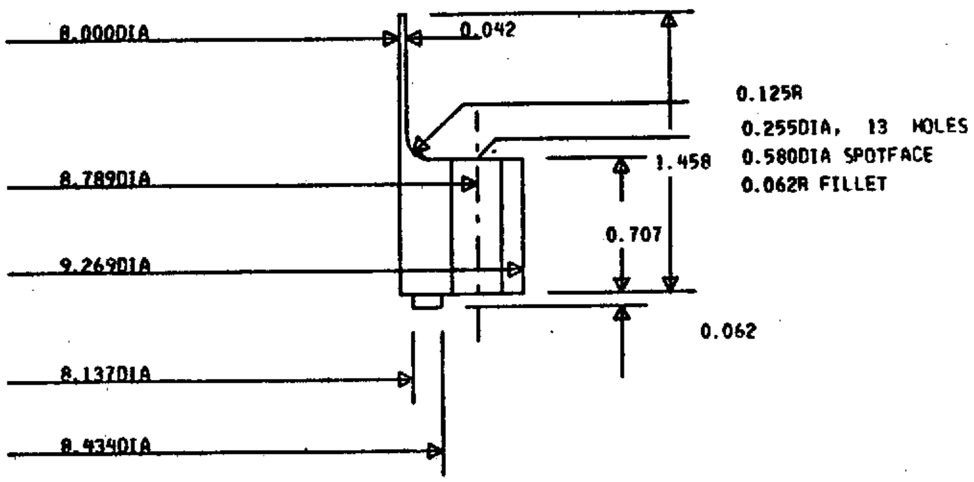


PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
OPERATION	$1.14645 \times 10^{+04}$	$1.70368 \times 10^{+02}$	2.17031×10^{-03}	1.54703×10^{-03}

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

FLANGE PARAMETRIC CASE 8 DIA= 8IN PRESS = 200 PSI

PRESSURE 200 PSIG
WEIGHT OF FLANGE 1.120 LB

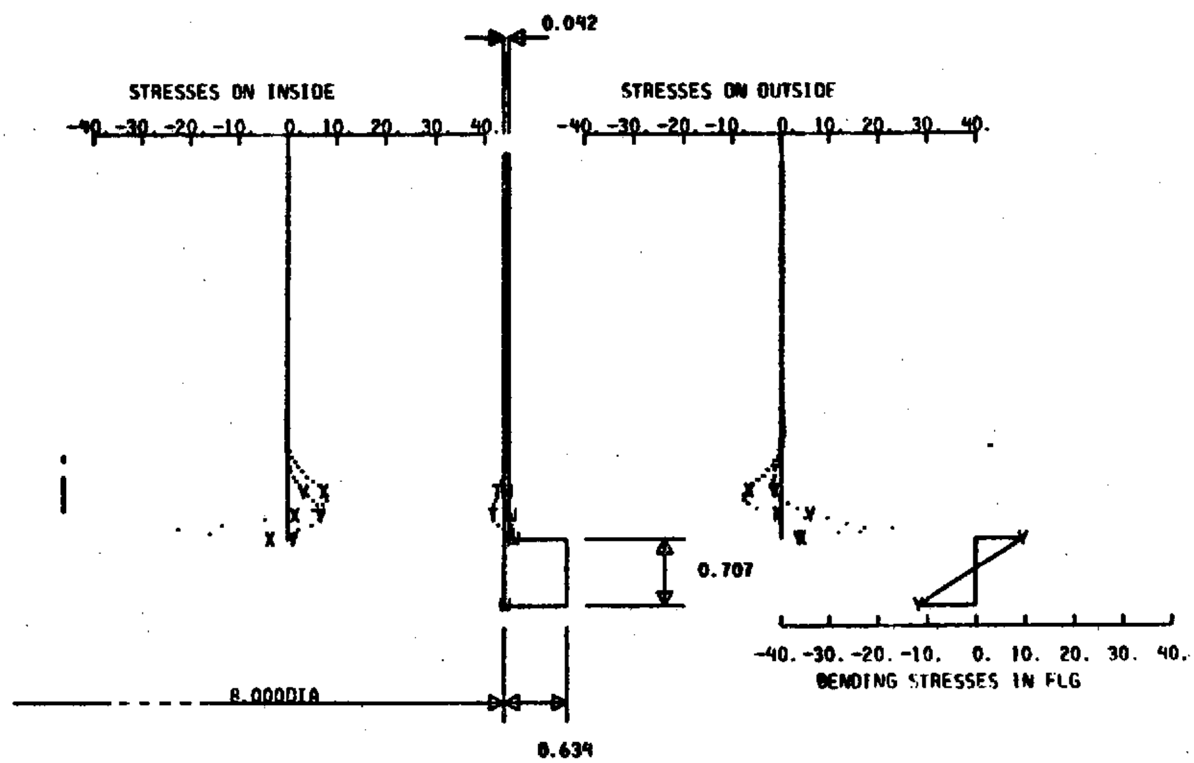


FLANGE MATERIAL	AL6061T6 RT	FTY= 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL	AL2024T3 RT	FTY= 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL	ASBESTO1/16	SEATING STRESS= 3.700KSI
		YIELD STRENGTH= 10.000KSI

Fig. 5-16 - Flange 8, Design

FLANGE PARAMETRIC CASE 8 DIA= 8IN PRESS = 200 PSI

SUMMARY OF ANALYSIS

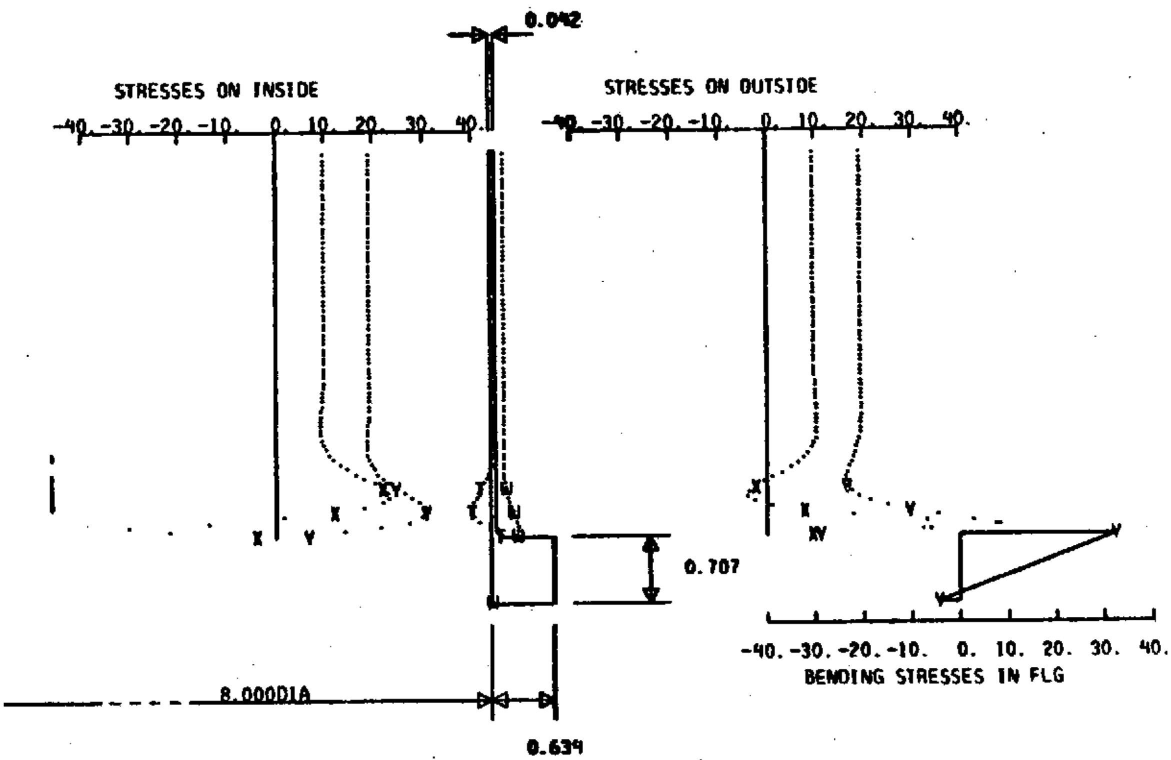


PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMENT (IN)	ROTATION (RAD)
BOLT-UP	$1.93656\times10^{+04}$	$1.92988\times10^{+02}$	0.00000×10^{-01}	1.22911×10^{-02}

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

FLANGE PARAMETRIC CASE 8 DIA= 8IN PRESS = 200 PSI

SUMMARY OF ANALYSIS



PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
OPERATION	$2.23584 \times 10^{+04}$	$3.32845 \times 10^{+02}$	1.23510×10^{-03}	2.11983×10^{-02}

Fig. 5-18 - Flange 8, Stresses at Proof Pressure

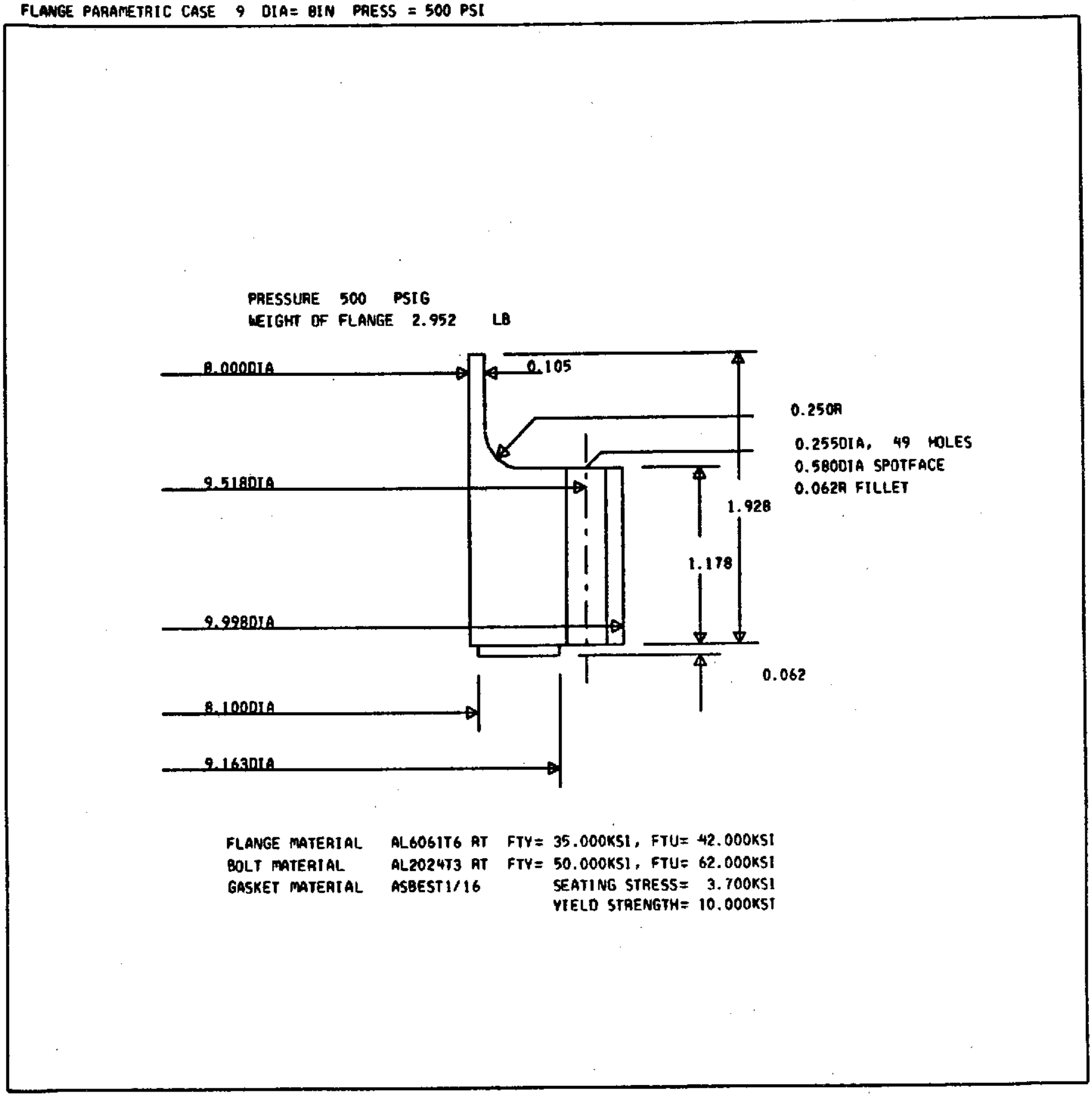
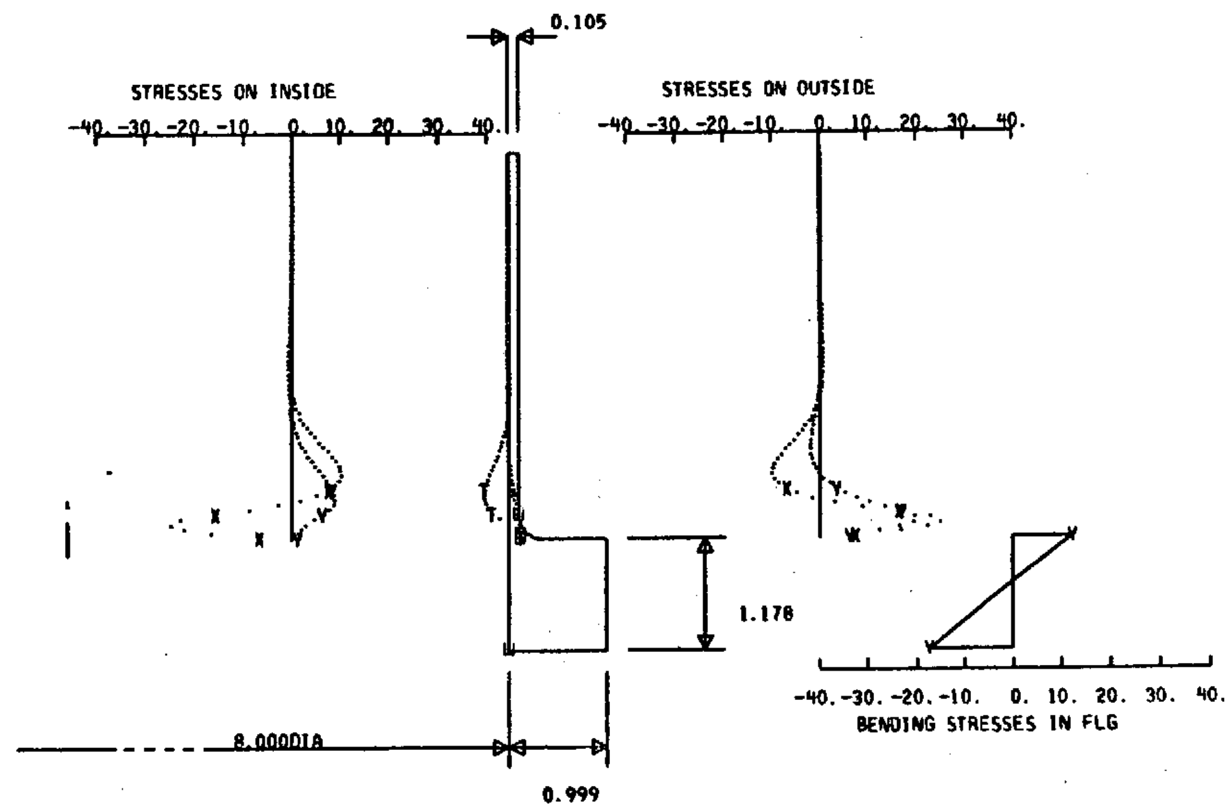


Fig. 5-19 - Flange 9, Design

FLANGE PARAMETRIC CASE 9 DIA= 8IN PRESS = 500 PSI

SUMMARY OF ANALYSIS

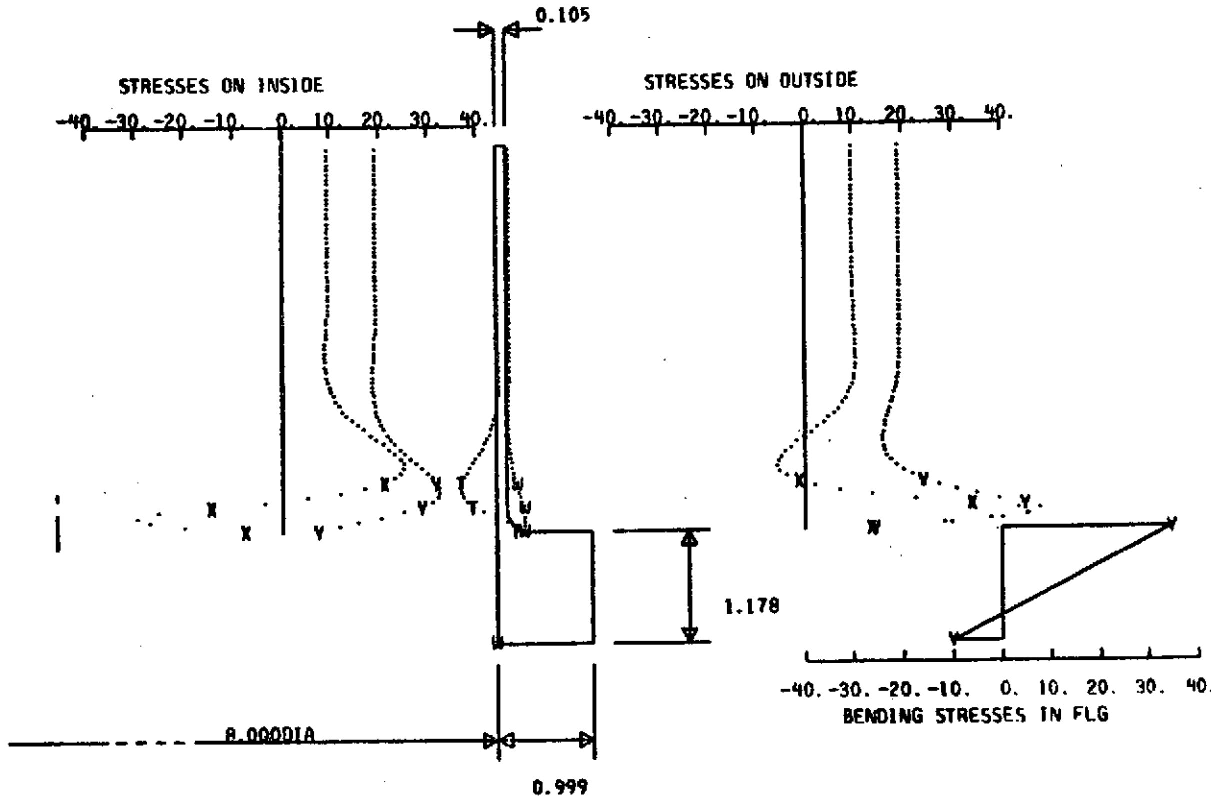


PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
BOLT-UP	$7.36401 \times 10^{+04}$	$1.28239 \times 10^{+03}$	0.00000×10^{-01}	1.02664×10^{-02}

Fig. 5-20 - Flange 9, Stresses at Initial Torquing

FLANGE PARAMETRIC CASE 9 DIA= 8IN PRESS = 500 PSI

SUMMARY OF ANALYSIS

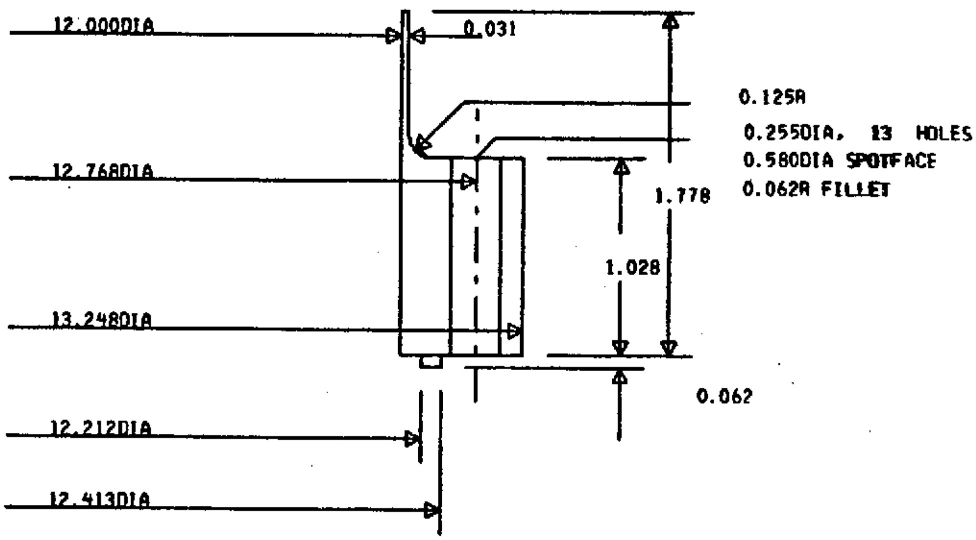


PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
OPERATION	$7.52752 \times 10^{+04}$	$1.93511 \times 10^{+03}$	1.13685×10^{-03}	1.54919×10^{-02}

Fig. 5-21 - Flange 9, Stresses at Proof Pressure

FLANGE PARAMETRIC CASE 13 DIA=12IN PRESS = 100 PSI

PRESSURE 100 PSIG
WEIGHT OF FLANGE 2.374 LB

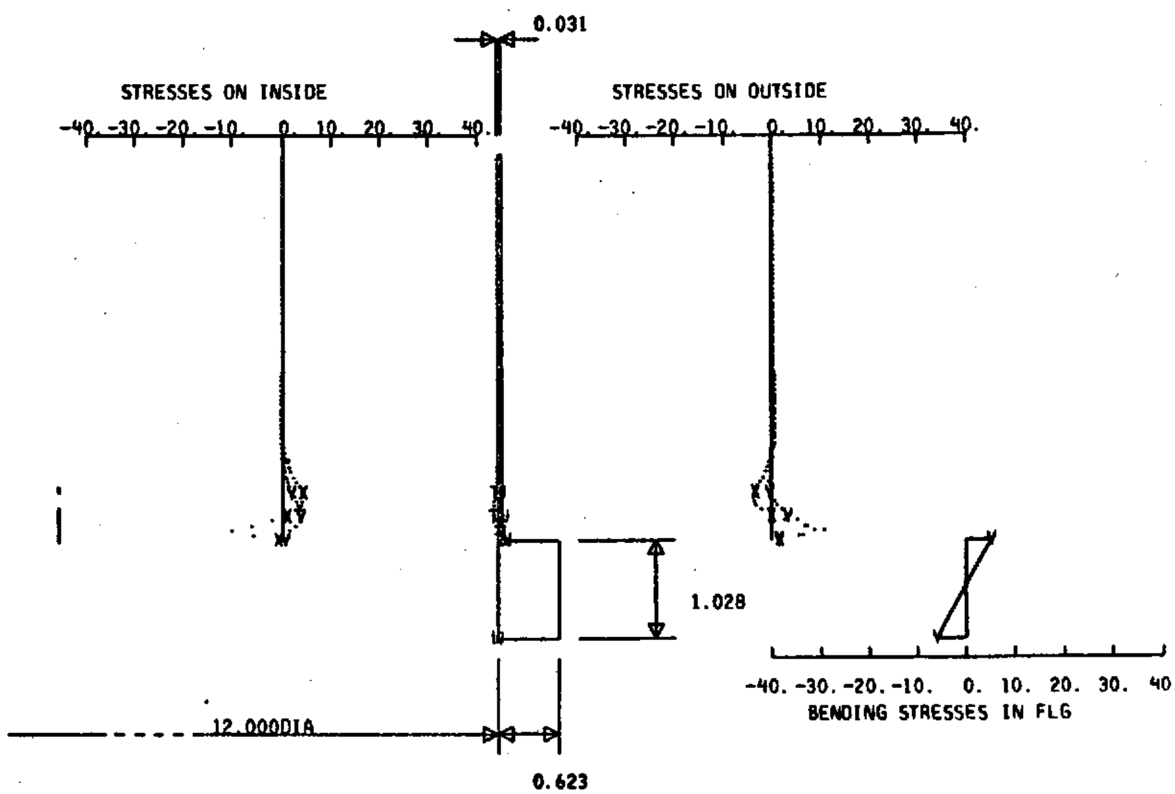


FLANGE MATERIAL AL6061T6 RT FTY= 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL AL2024T3 RT FTY= 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL ASBEST1/16 SEATING STRESS= 3.700KSI
YIELD STRENGTH= 10.000KSI

Fig. 5-22 - Flange 13, Design

FLANGE PARAMETRIC CASE 13 DIA=12IN PRESS = 100 PSI

SUMMARY OF ANALYSIS

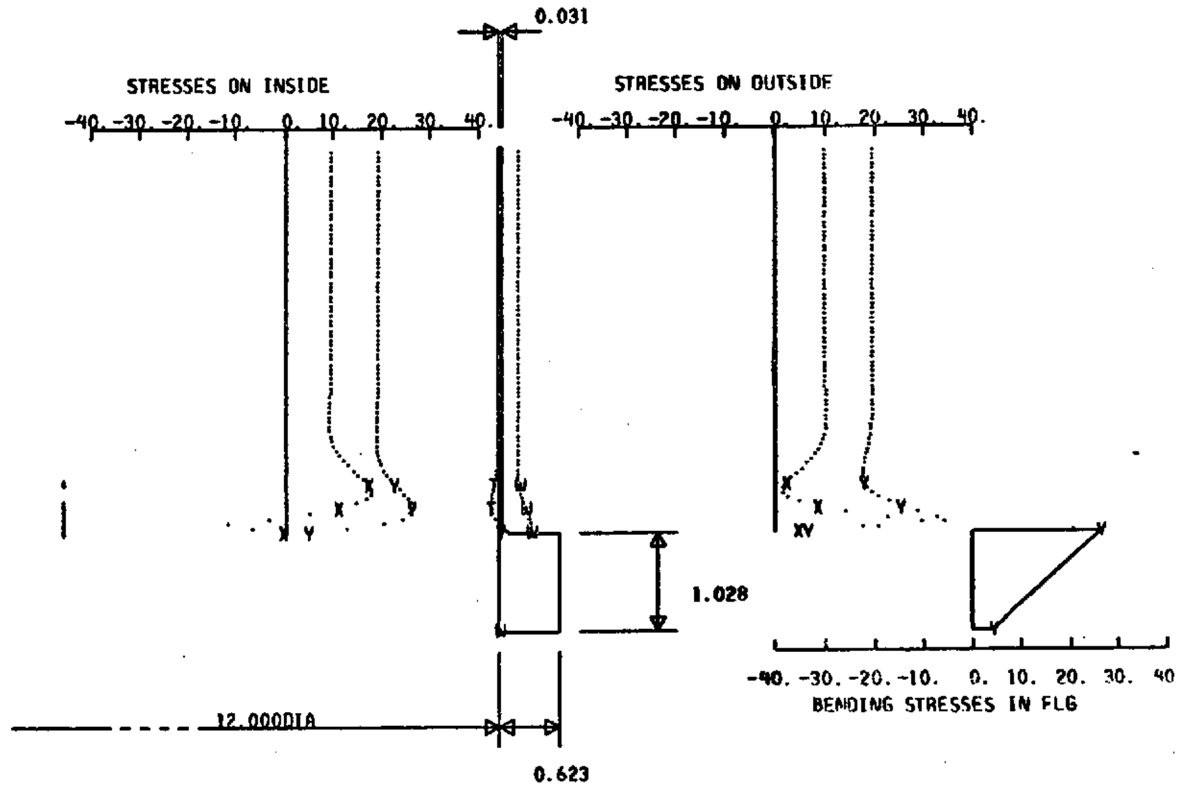


PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
BOLT-UP	$1.94594 \times 10^{+04}$	$1.17177 \times 10^{+02}$	0.00000×10^{-01}	6.49100×10^{-03}

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

FLANGE PARAMETRIC CASE 13 DIA=12IN PRESS = 100 PSI

SUMMARY OF ANALYSIS



PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
OPERATION	$2.40635 \times 10^{+04}$	$2.33837 \times 10^{+02}$	1.75444×10^{-03}	1.29533×10^{-02}

Fig. 5-24 - Flange 13, Stresses at Proof Pressure

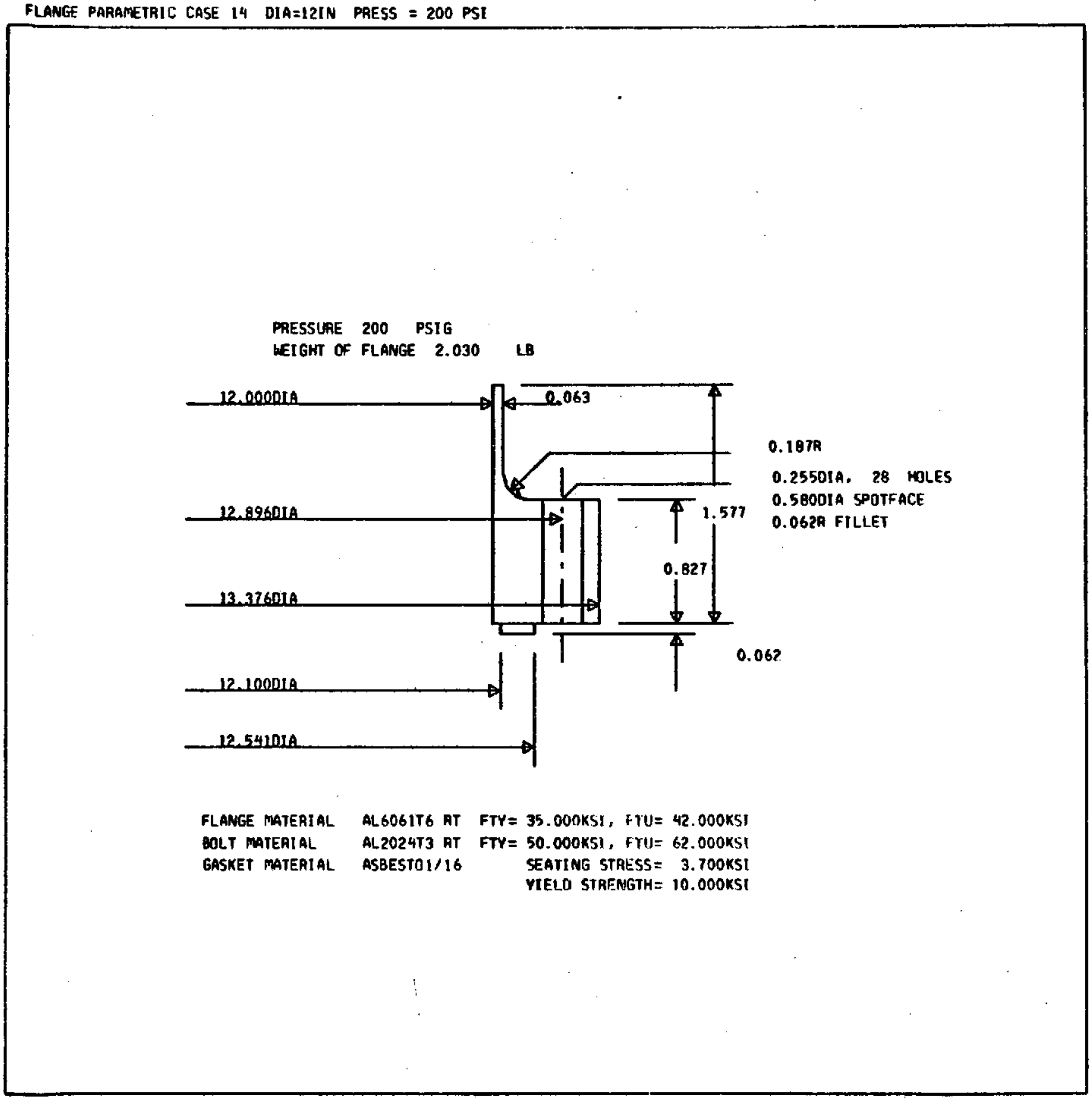
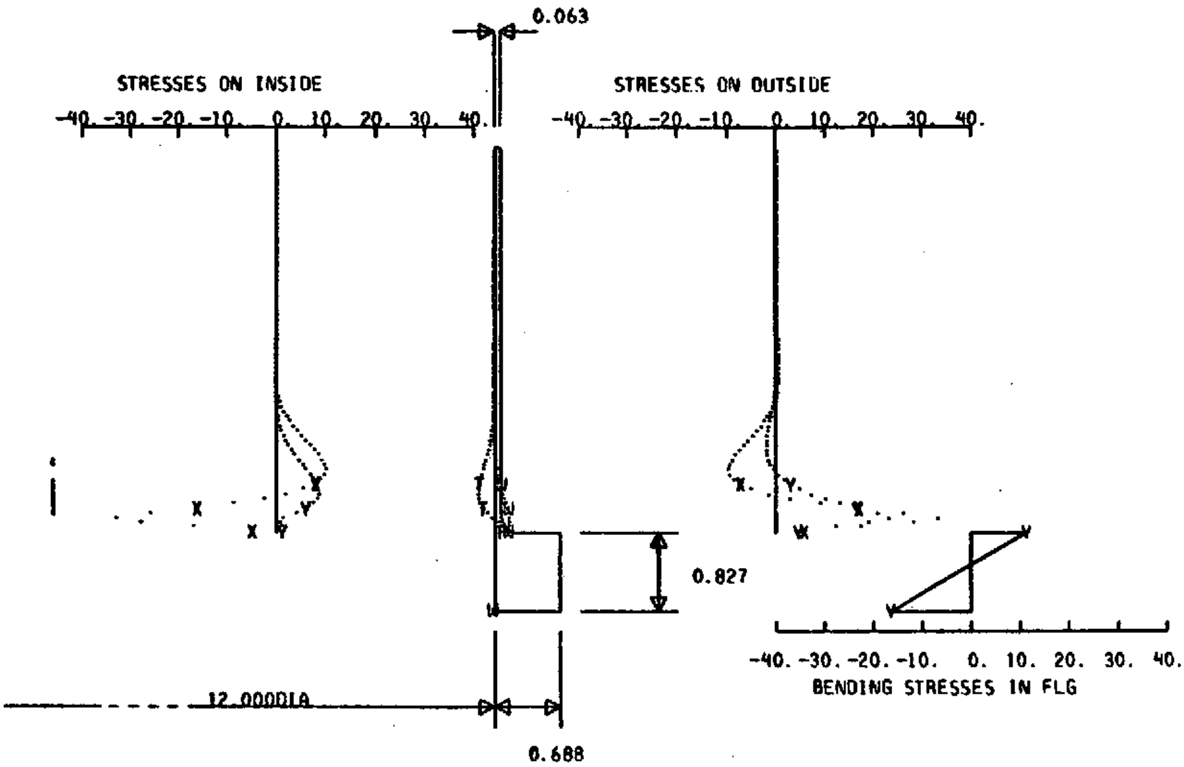


Fig. 5-25 - Flange 14, Design

FLANGE PARAMETRIC CASE 14 DIA=12IN PRESS = 200 PSI

SUMMARY OF ANALYSIS

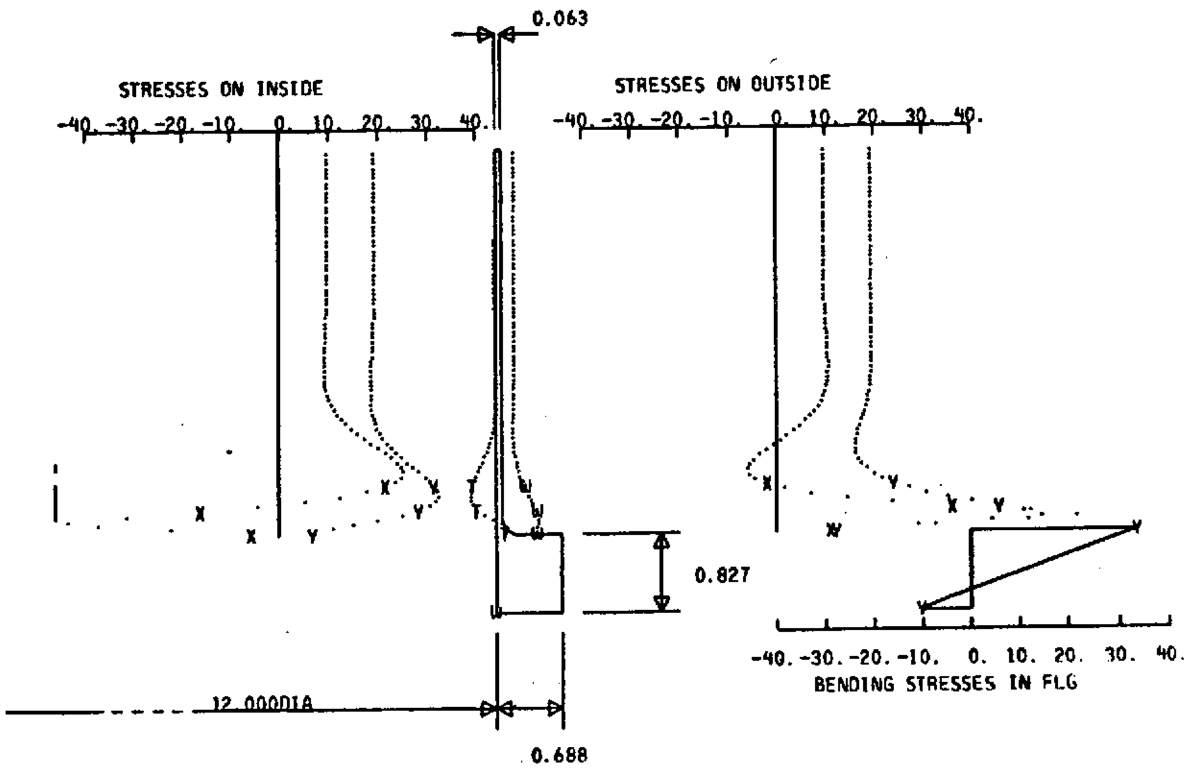


PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
BOLT-UP	$4.28180 \times 10^{+04}$	$3.25275 \times 10^{+02}$	0.00000×10^{-01}	2.03468×10^{-02}

Fig. 5-26 - Flange 14, Stresses at Initial Torquing

FLANGE PARAMETRIC CASE 14 DIA=12IN PRESS = 200 PSI

SUMMARY OF ANALYSIS



PHASE	BOLT FORCE (LB)	APPLIED MOMENT (IN-LB)	AXIAL DISPLACEMT (IN)	ROTATION (RAD)
OPERATION	$4.43483 \times 10^{+04}$	$5.17644 \times 10^{+02}$	1.39328×10^{-03}	3.23800×10^{-02}

Fig. 5-27 - Flange 14, Stresses at Proof Pressure

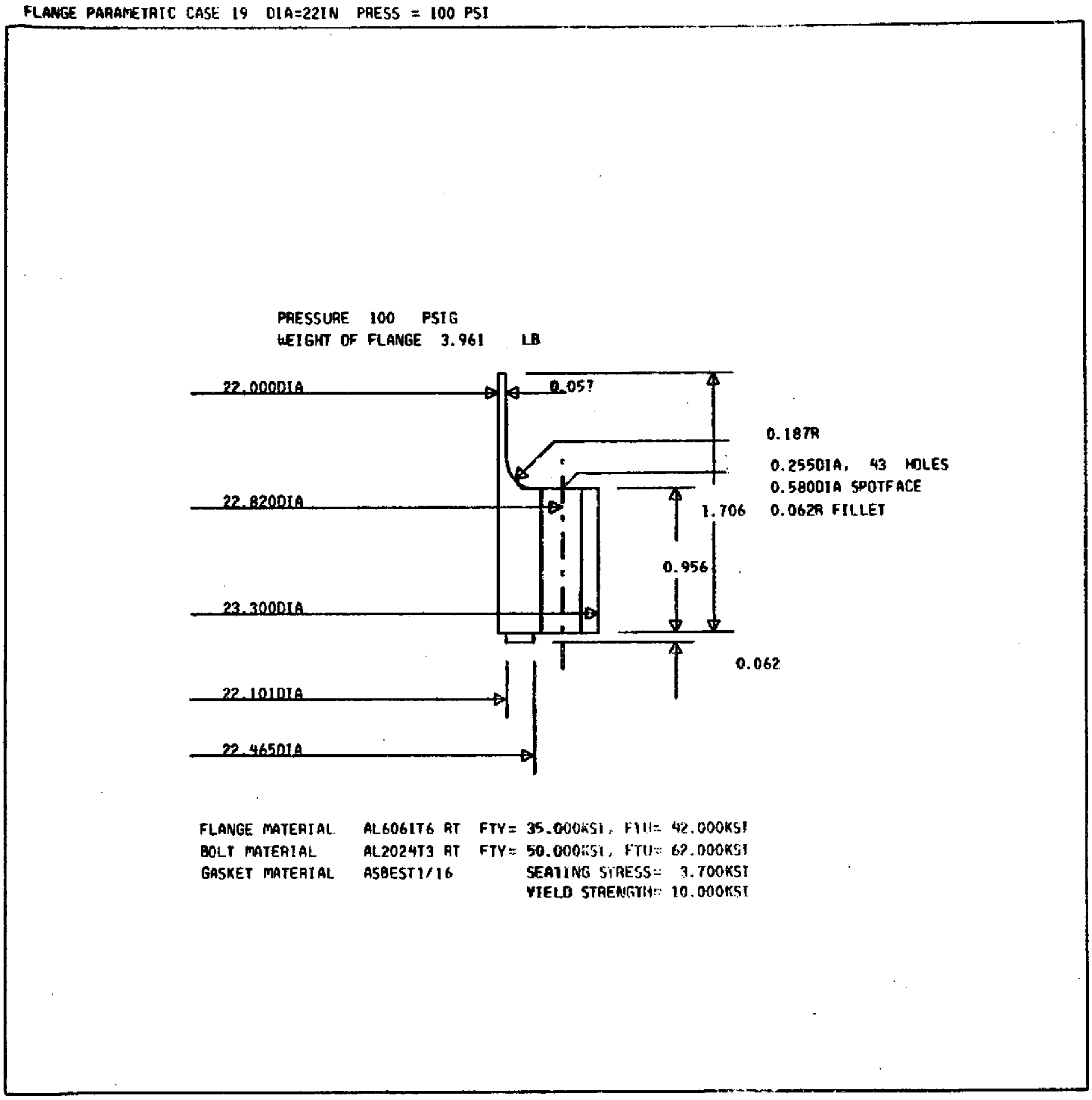


Fig. 5-28 - Flange 19, Design

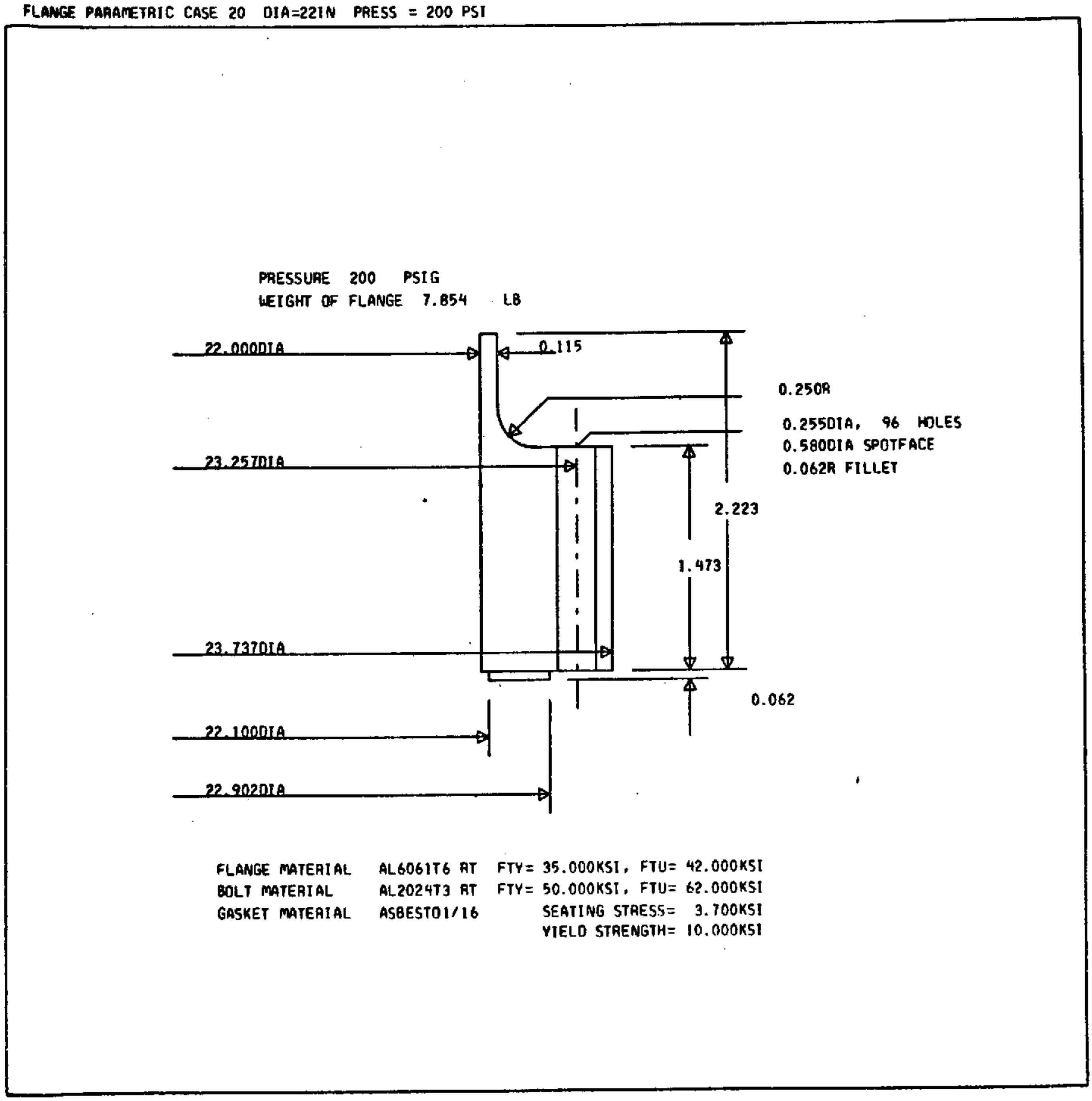


Fig. 5-29 - Flange 20, Design

FLANGE PARAMETRIC CASE 25 DIA=45IN PRESS = 100 PSI

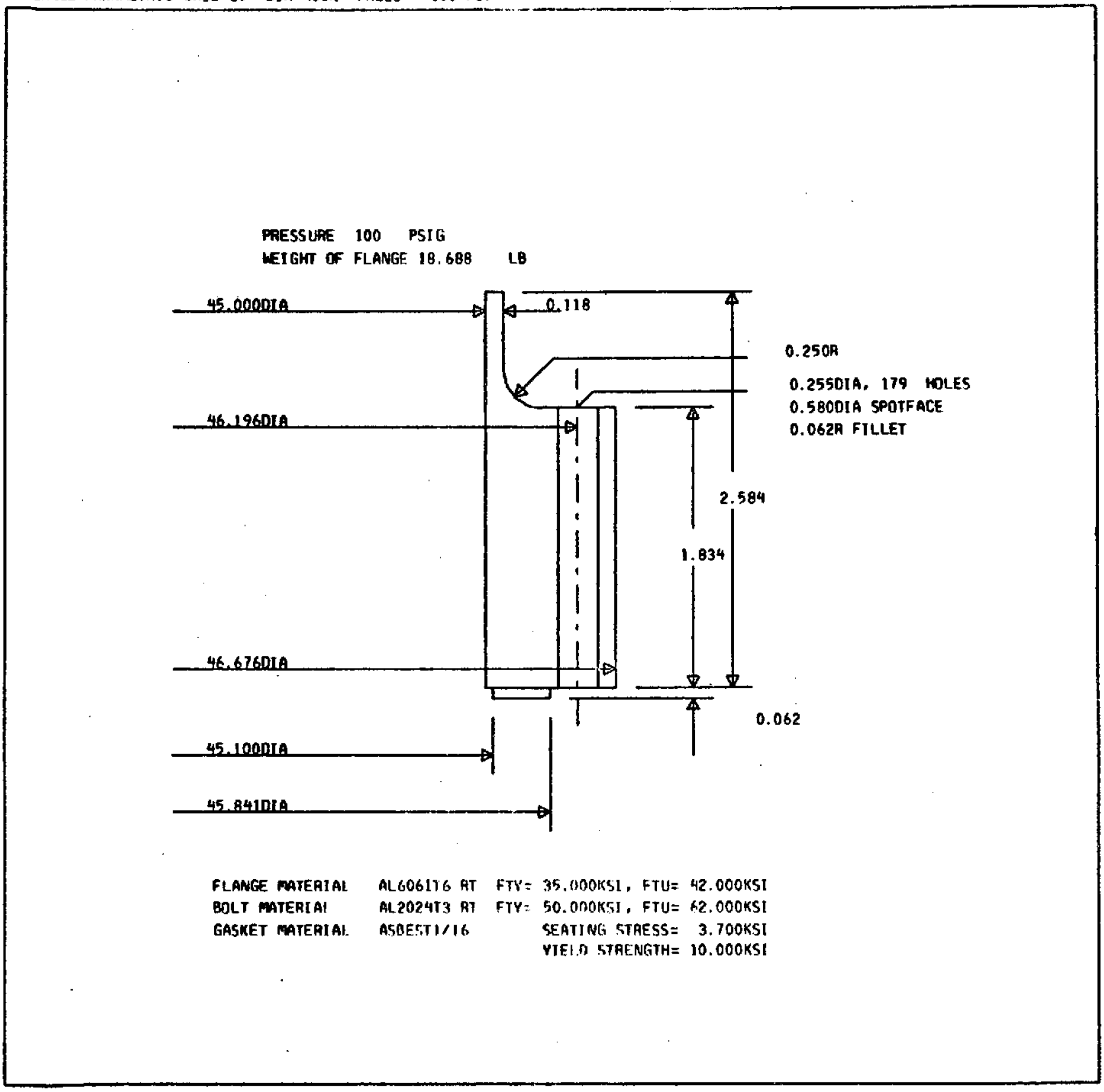


Fig. 5-30 - Flange 25, Design

Table 5-1 - Comparison of Flanges

No.	Flange for	pressure [psi]	inner diameter [in]	wall thickness [in]	Saturn II Flanges: outer diameter [in]	Saturn II Flanges: height of flange [in]	Saturn II Flanges: width of flange [in]	Saturn II Flanges: added volume [in³]	Low Profile Flanges*: outer diameter [in]	Low Profile Flanges*: height of flange [in]	Low Profile Flanges*: width of flange [in]	Low Profile Flanges*: added volume [in³]	volume difference [in³]	weight savings [%]
1	LOX	140	22.000	0.190	26.250	1.000	2.125	23.50	23.565	1.163	0.7825	7.91	15.59	66.2
2	LOX	140	12.000	0.190	15.000	0.750	1.500	6.72	13.929	1.239	0.9645	6.31	0.41	6.1
3	LOX	140	8.000	0.190	10.750	0.870	1.375	4.92	9.565	0.789	0.7825	2.10	2.82	57.3
4	LOX	140	7.780	0.190	10.280	0.720	1.300	3.78	9.345	0.770	0.7825	1.99	1.88	49.8
5	LOX	140	7.820	0.190	10.500	0.660	1.340	3.54	9.385	0.775	0.7825	2.02	1.52	43.0
6	LOX	80	6.000	0.160	8.250	0.630	1.125	2.22	7.505	0.490	0.7525	1.01	1.21	54.3
7	GOX	300	6.500	0.190	9.400	0.750	1.450	3.84	8.625	1.108	1.0625	3.74	0.10	2.6
8	GOX	300	4.000	0.190	6.300	1.000	1.150	2.56	5.565	0.632	0.7825	0.93	1.63	63.7
9	GOX	100	22.000	0.160	26.250	1.000	2.125	23.85	23.505	1.023	0.7525	6.94	16.91	71.0
10	GOX	100	4.000	0.160	6.250	0.810	1.125	2.06	5.505	0.751	0.7525	1.09	0.97	47.1
11	GOX	80	7.000	0.140	9.500	0.750	1.250	3.49	8.465	0.546	0.7325	1.27	2.22	63.5
12	GOX	80	6.900	0.140	9.750	0.810	1.425	3.43	8.365	0.538	0.7325	1.24	2.19	63.4
13	GOX	80	5.000	0.140	7.750	0.870	1.375	3.20	6.465	0.696	0.7325	1.22	1.98	61.9
14	GOX	80	4.750	0.140	7.500	0.750	1.375	2.90	6.215	0.750	0.7325	1.24	1.66	57.3
15	GOX	80	4.000	0.140	6.270	0.560	1.135	1.97	5.465	0.870	0.7325	1.26	0.71	36.1

*Based on Bolt Table 2-3

LIQUID OXYGEN 140 PSI, 22 IN DIA

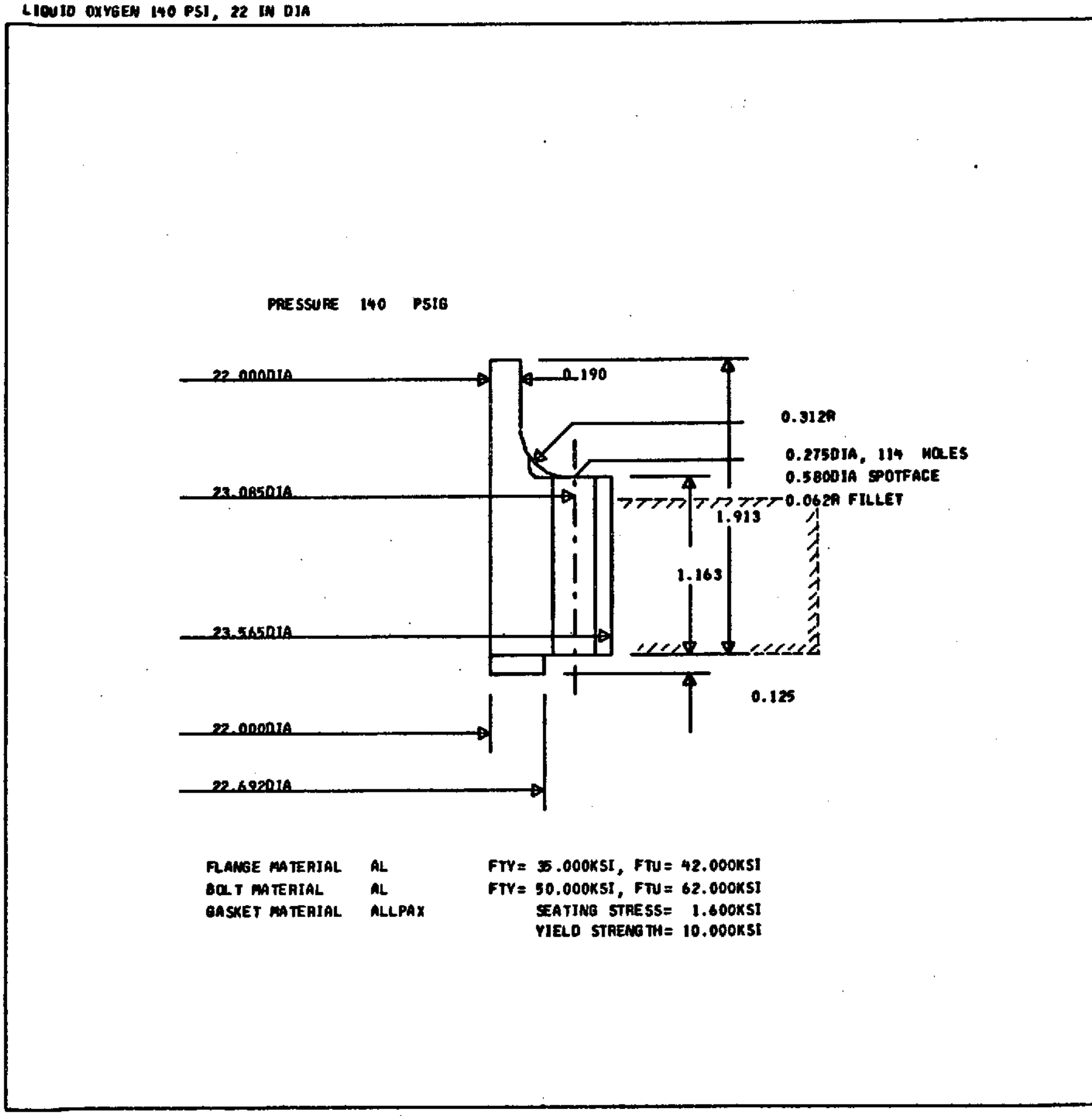


Fig. 5-31 - Flange Comparison 1*

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

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

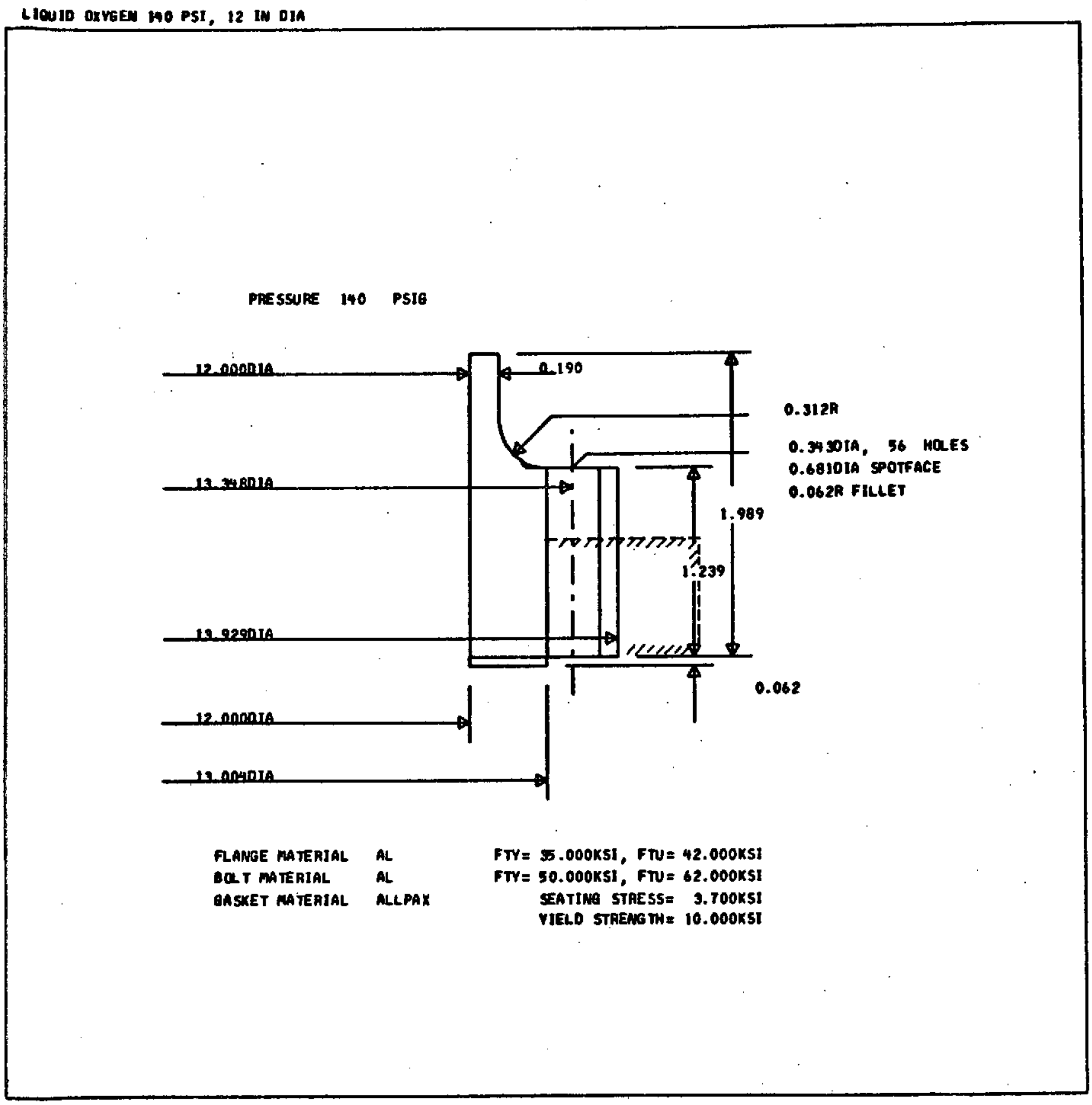


Fig. 5-32 - Flange Comparison 2

LIQUID OXYGEN 140 PSI, 8 IN DIA

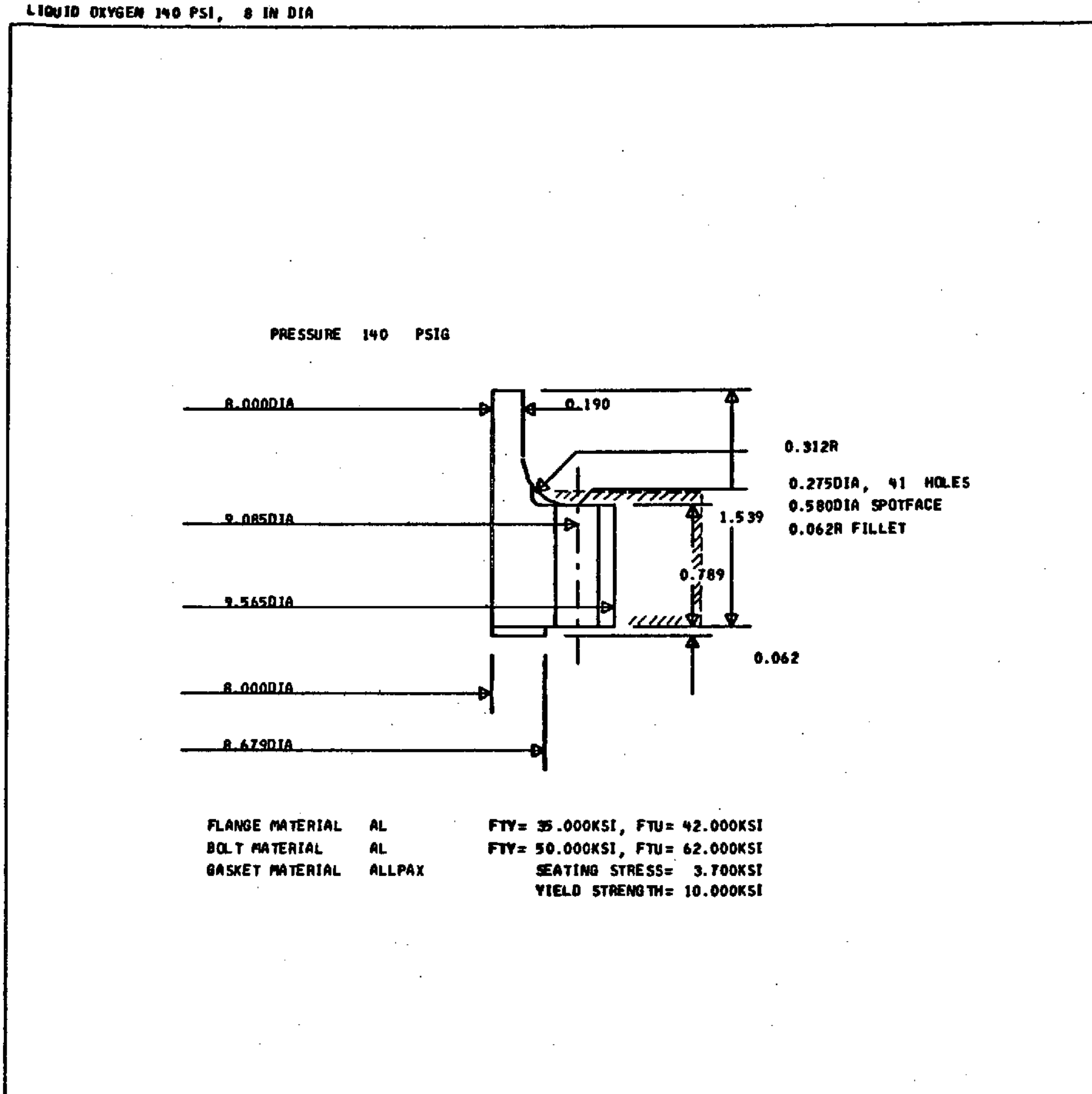
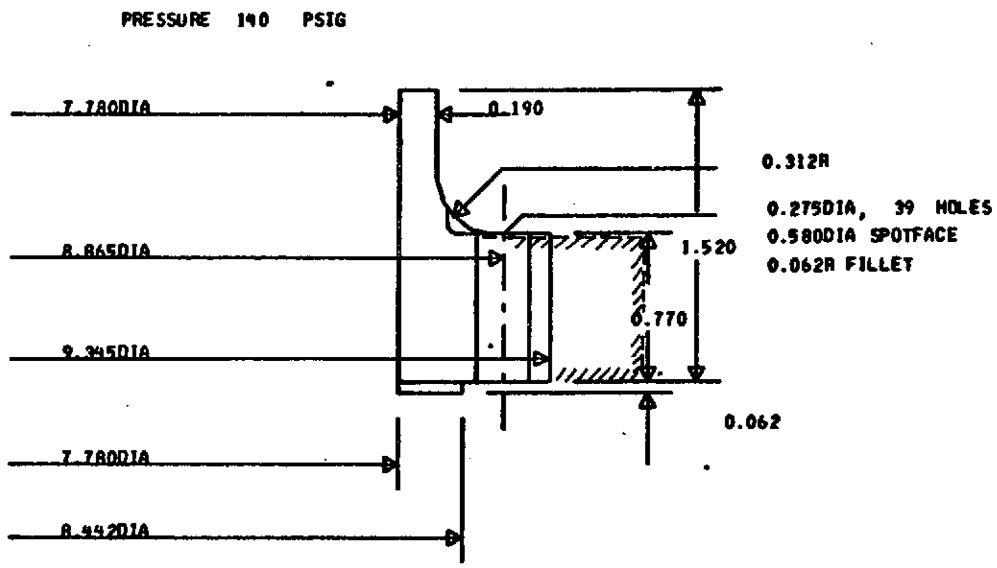


Fig. 5-33 - Flange Comparison 3

LIQUID OXYGEN 140 PSI, 7.78 IN DIA



FLANGE MATERIAL	AL	FTY= 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL	AL	FTY= 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL	ALLPAX	SEATING STRESS= 3.700KSI
		YIELD STRENGTH= 10.000KSI

Fig. 5-34 - Flange Comparison 4

LIQUID OXYGEN 140 PSI, 7.82 IN DIA

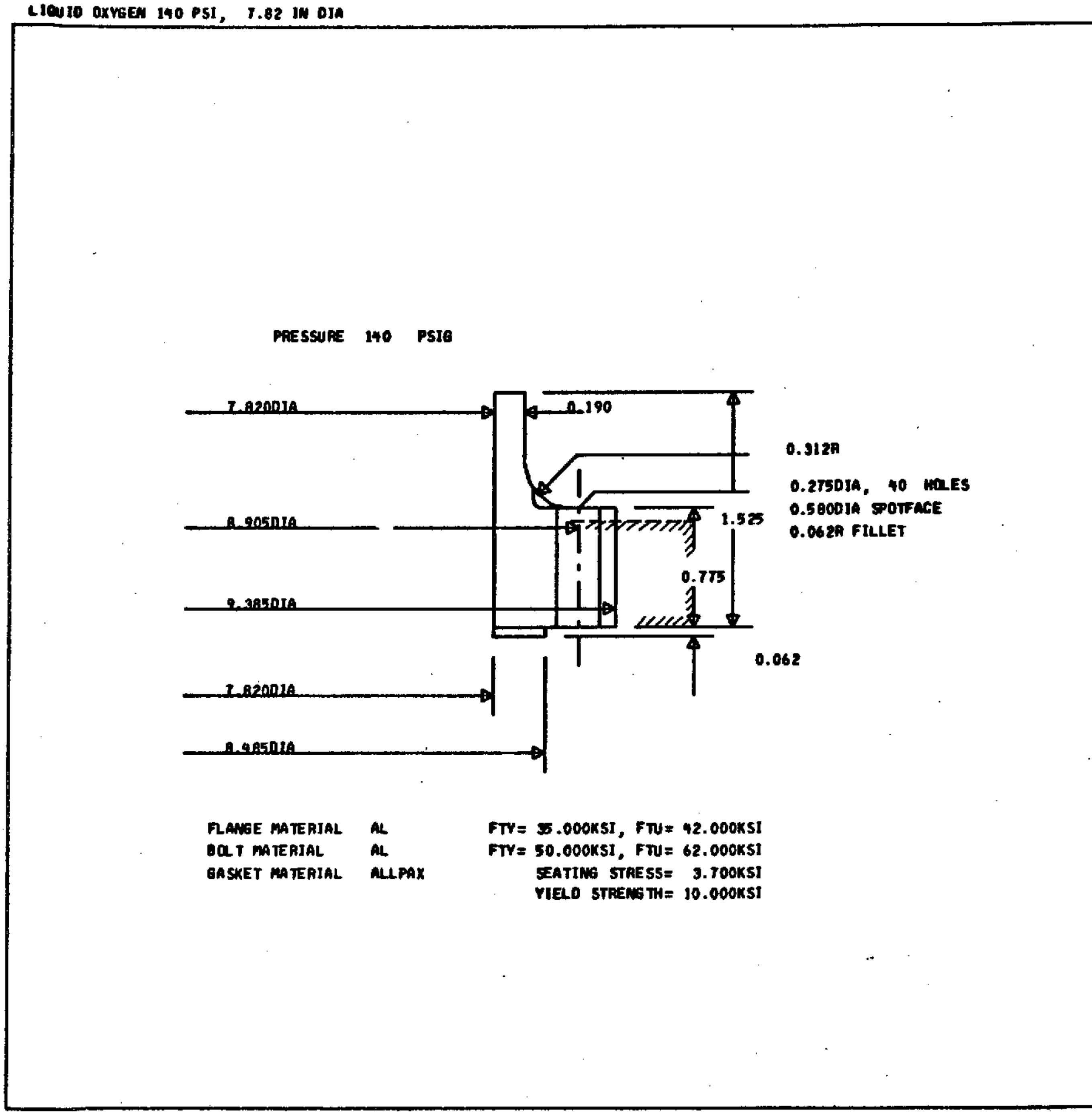
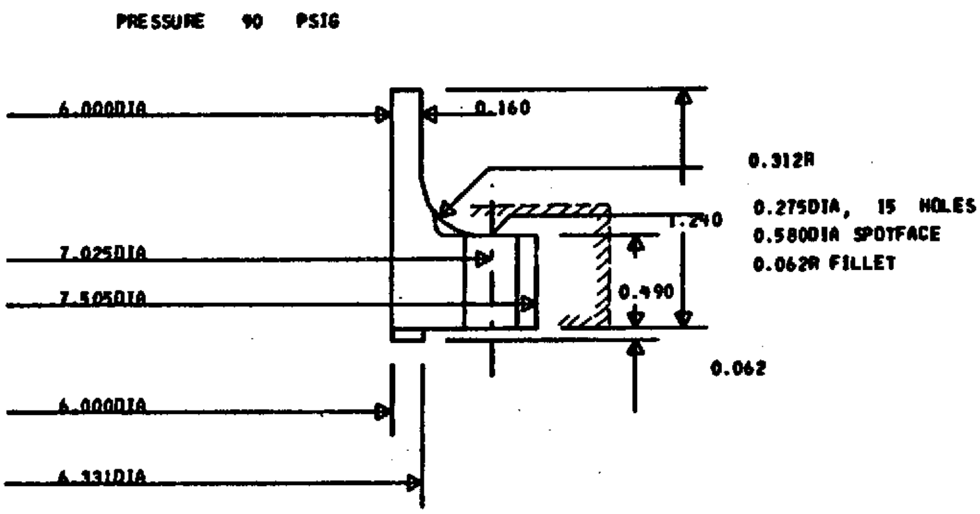


Fig. 5-35 - Flange Comparison 5

LIQUID OXYGEN 90 PSI, 6 IN DIA

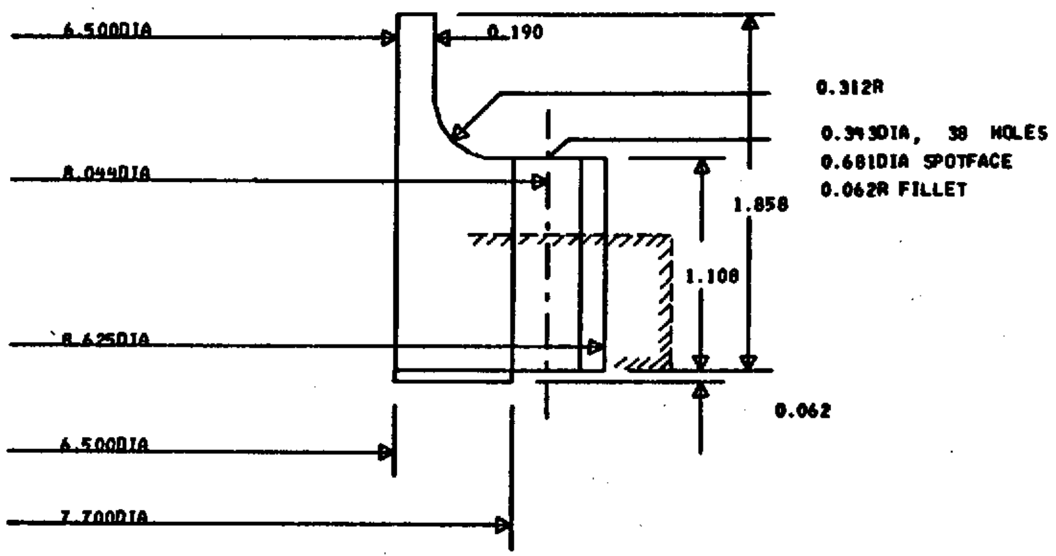


FLANGE MATERIAL	AL	FTY= 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL	AL	FTY= 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL	ALLPAX	SEATING STRESS= 3.700KSI
		YIELD STRENGTH= 10.000KSI

Fig. 5-36 - Flange Comparison 6

GASEOUS OXYGEN 300 PSI, 6.5 IN DIA

PRESSURE 300 PSIG



FLANGE MATERIAL	AL	FTY= 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL	AL	FTY= 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL	ALLPAX	SEATING STRESS= 3.700KSI
		YIELD STRENGTH= 10.000KSI

Fig. 5-37 - Flange Comparison 7

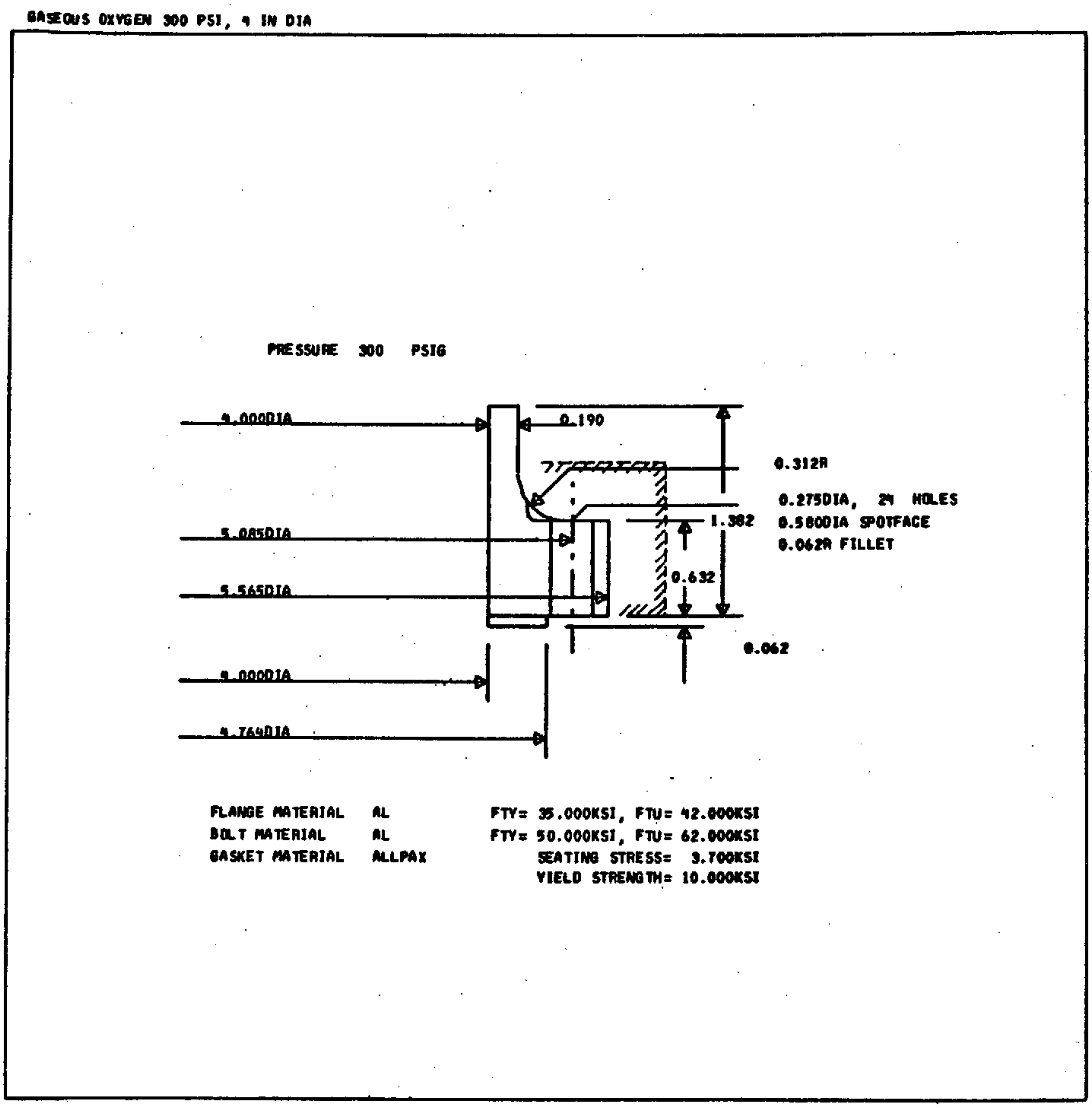
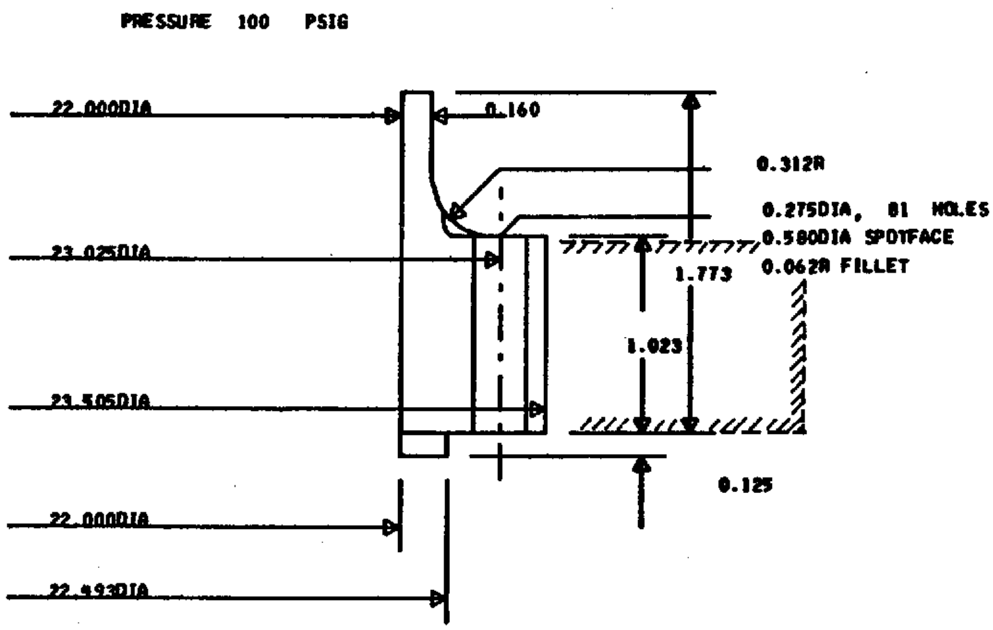


Fig. 5-38 - Flange Comparison 8

GASEOUS OXYGEN 100 PSI, 22 IN DIA

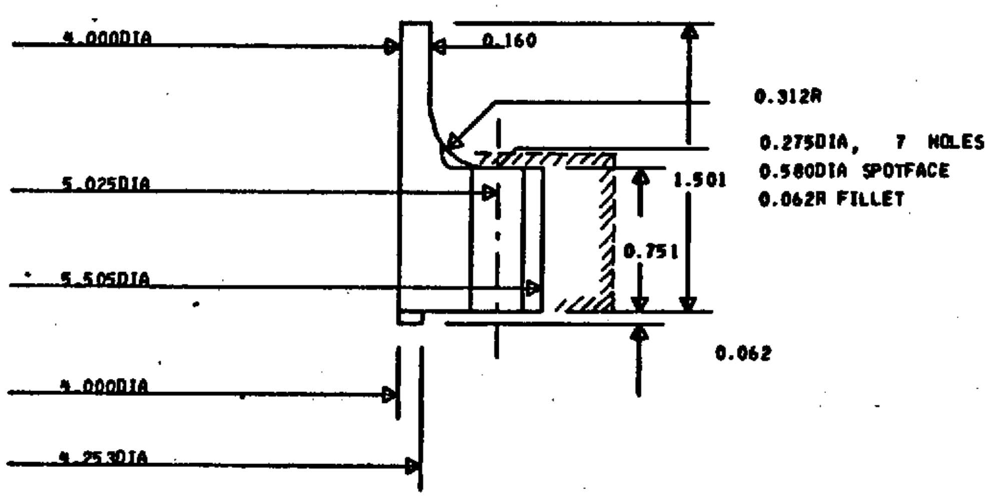


FLANGE MATERIAL	AL	FTY= 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL	AL	FTY= 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL	ALLPAX	SEATING STRESS= 1.600KSI
		YIELD STRENGTH= 10.000KSI

Fig. 5-39 - Flange Comparison 9

GASEOUS OXYGEN 100 PSI, 4 IN DIA

PRESSURE 100 PSIG

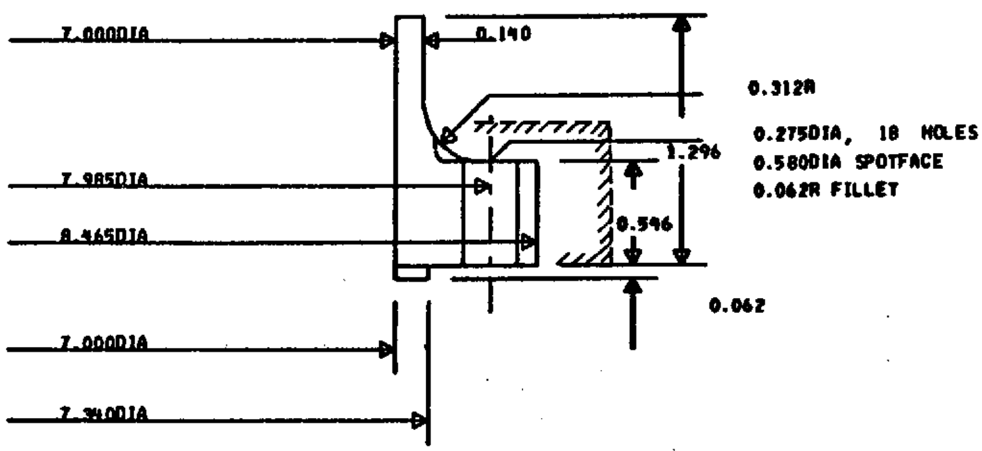


FLANGE MATERIAL	AL	FTY= 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL	AL	FTY= 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL	ALLPAX	SEATING STRESS= 3.700KSI
		YIELD STRENGTH= 10.000KSI

Fig. 5-40 - Flange Comparison 10

GASEOUS OXYGEN 80 PSI, 7 IN DIA

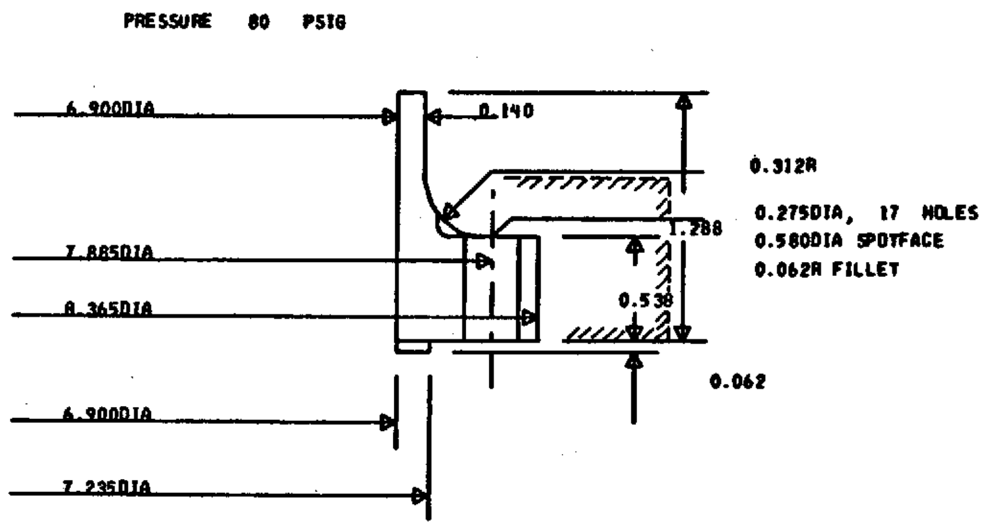
PRESSURE 80 PSIG



FLANGE MATERIAL AL FTY= 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL AL FTY= 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL ALLPAX SEATING STRESS= 3.700KSI
YIELD STRENGTH= 10.000KSI

Fig. 5-41 - Flange Comparison 11

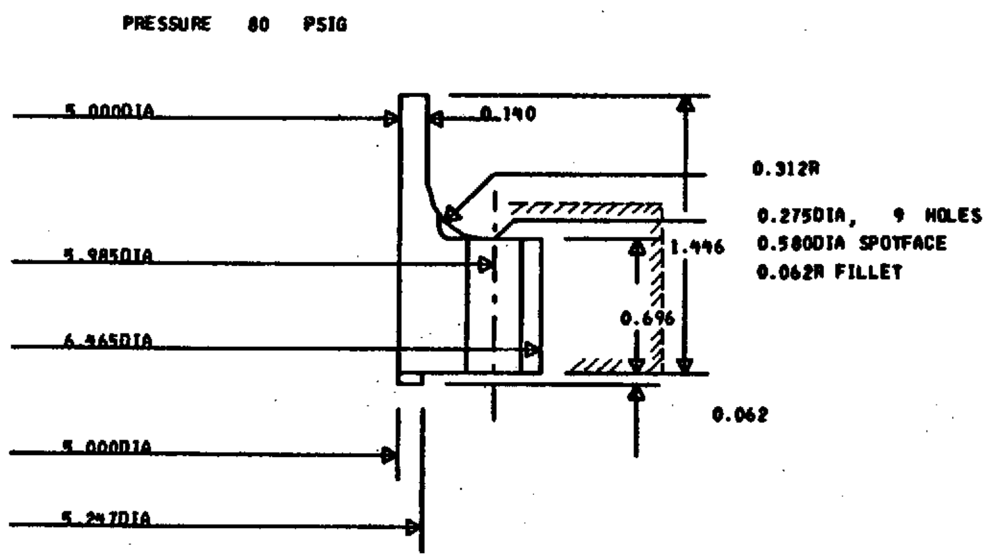
GASEOUS OXYGEN 80 PSI, 6.9 IN DIA



FLANGE MATERIAL AL FTY= 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL AL FTY= 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL ALLPAX SEATING STRESS= 3.700KSI
YIELD STRENGTH= 10.000KSI

Fig. 5-42 - Flange Comparison 12

GASEOUS OXYGEN 80 PSI, 5 IN DIA



FLANGE MATERIAL	AL	FTY= 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL	AL	FTY= 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL	ALLPAX	SEATING STRESS= 3.700KSI
		YIELD STRENGTH= 10.000KSI

Fig. 5-43 - Flange Comparison 13

GASEOUS OXYGEN 80 PSI, 4.75 IN DIA

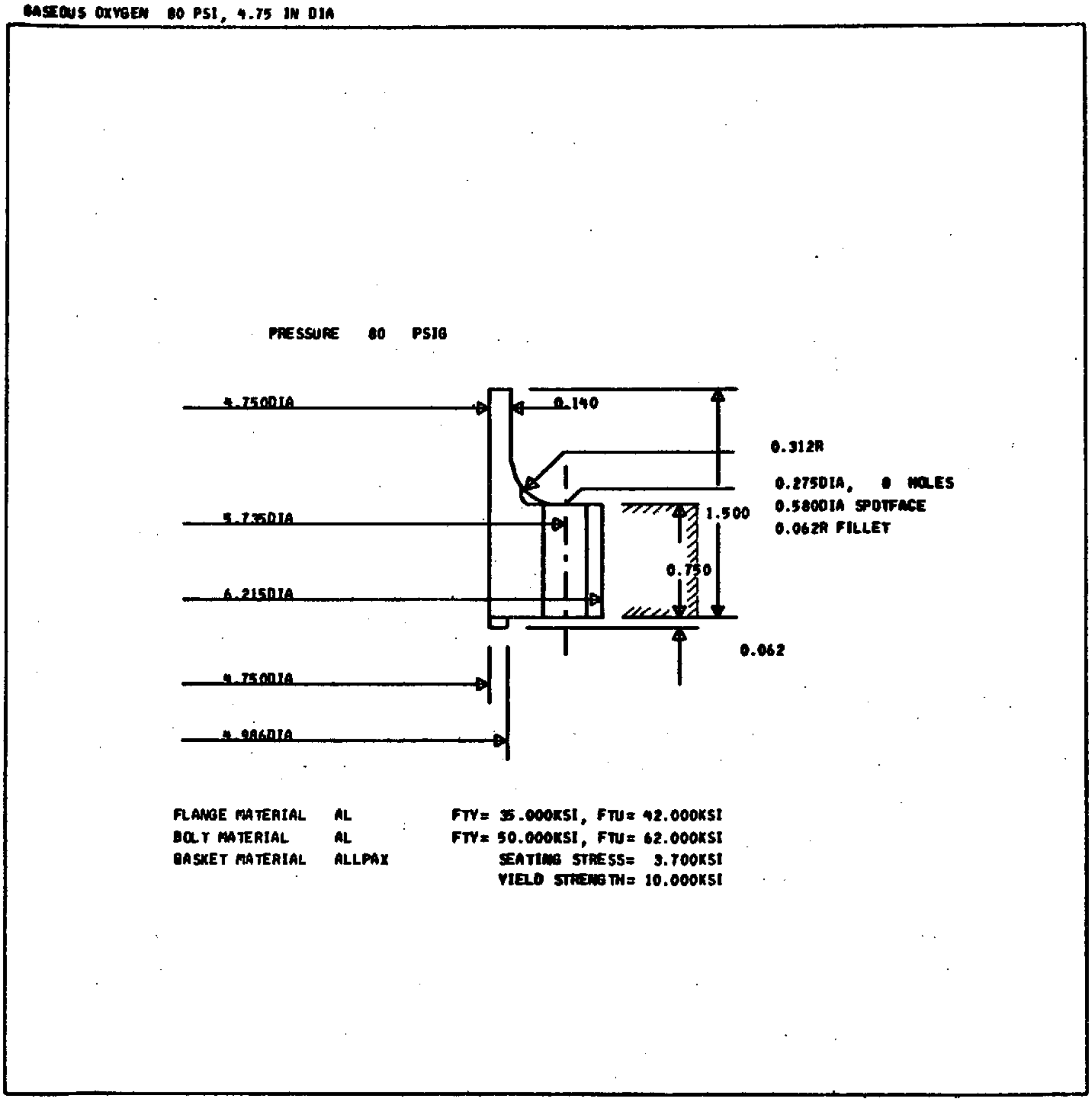
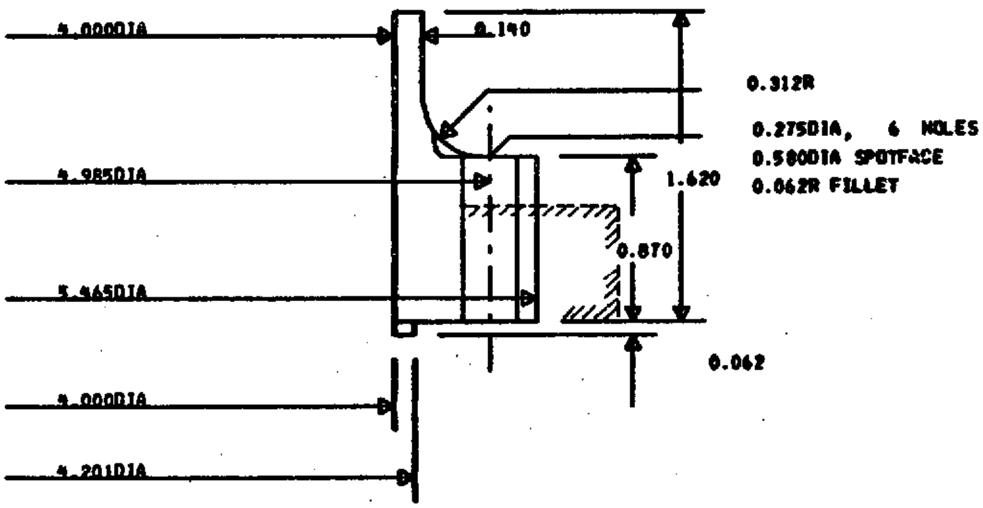


Fig. 5-44 - Flange Comparison 14

GASEOUS OXYGEN 80 PSI, 4 IN DIA

PRESSURE 80 PSIG



FLANGE MATERIAL	AL	FTY= 35.000KSI, FTU= 42.000KSI
BOLT MATERIAL	AL	FTY= 50.000KSI, FTU= 62.000KSI
GASKET MATERIAL	ALLPAX	SEATING STRESS= 3.700KSI
		YIELD STRENGTH= 10.000KSI

Fig. 5-45 - Flange Comparison 15

SAMPLE PRINTOUT

5-55-a

```
@ELT,DIL DATA3,,DATA3
ELT006-RL1867-10  03/10-14:52:51
CYCLE (00)
000001        000   @XQT      DABS
000002        000    HARDCOPY
000003        000        2
000004        000    TEST FLANGE 1500 PSI, 8 IN DIA
000005        000        1.5 +3     8.0          .4375                  1.75
000006        000       1.5         2.0          1.5         2.0
000007        000        5     5     8     5
000008        000        0     1     0     3     1
000009        000    347SS AT RT 347SS AT RT A286 AT RT  CRES321-A
000010        000        4
000011        000   -500.
000012        000    SPACE SHUTTLE MAIN ENGINE FLANGE  8000 PSI, 16 IN DIA
000013        000       8.    +3  16.0           1.                     2.
000014        000       1.5         2.0           1.5        2.0
000015        000        8     8     8    -1
000016        000        .3          .6          .05
000017        000        0     1     1     3
000018        000    A286  AT RT A286  AT RT A286 AT RT  K-SEAL
000019        000        4
000020        000   +600.
```

THESE DATA WERE USED FOR EXAMPLE 1 IN SECT. 5.1

NOMINAL PRESSURE	P=	1500.000 PSI
NOMINAL DIAMETER	DI=	8.000 INCH
TUBE THICKNESS	T=	.438 INCH
TUBE THICKN TOLR	DT=	.000 INCH
HEIGHT TO WELD	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

MATERIAL TABLE NO.	I=	5
ELASTIC MODULUS	E=	.28000000+08 PSI
POISSON-S RATIO	NU=	.300
DENSITY	RHO=	.2880 LB/CUBIC-INCH
THERM EXP COEFF	ALFA=	.95000000-05 INCH/INCH/F
TENSILE YIELD STR	FTY=	.35000000+05 PSI
ULTIMATE TENS STR	FTU=	.90000000+05 PSI

PROPERTIES OF FLANGE MATERIAL

MATERIAL TABLE NO.	I=	5
ELASTIC MODULUS	E=	.28000000+08 PSI
POISSON-S RATIO	NU=	.300
DENSITY	RHO=	.2880 LB/CUBIC-INCH
THERM EXP COEFF	ALFA=	.95000000-05 INCH/INCH/F
TENSILE YIELD STR	FTY=	.35000000+05 PSI
ULTIMATE TENS STR	FTU=	.90000000+05 PSI

PROPERTIES OF BOLT MATERIAL

MATERIAL TABLE NO.	I=	8
ELASTIC MODULUS	E=	.28000000+08 PSI
POISSON-S RATIO	NU=	.300
DENSITY	RHO=	.2880 LB/CUBIC-INCH
THERM EXP COEFF	ALFA=	.95000000-05 INCH/INCH/F
TENSILE YIELD STR	FTY=	.13100000+06 PSI
ULTIMATE TENS STR	FTU=	.20000000+06 PSI

PROPERTIES OF GASKET MATERIAL

MATERIAL TABLE NO.	I=	5
ELASTIC MODULUS	E=	.28000000+08 PSI

```
YIELD STRENGTH           KG=    .40000000+05 PSI
SEATING STRESS           SG=    .18900000+05 PSI
THERM EXP COEFF        ALFA=    .95000000-05 INCH/INCH/F
COEFF OF FRICTION        MU=      .300
WIDTH COEFFICIENT      GAMU=     1.000
WIDTH COEFFICIENT      GAMS=     1.000
GASKET THICKNESS         HG=     .0250 INCH
SEALING STRESS RATE      SP=     5.5000


OPTIONS
  0    1    0    3    1    0    0    0    0    0
NUMBER OF PHASES TO BE CONSIDERED IN THE ANALYSIS =  4
TEMPERATURE DIFFERENTIAL =   -500.00 DEG F

COMPUTED THICKNESS    T=      .4375 INCH

TEST FLANGE 1500 PSI, 8 IN DIA
  AOB=      .1063 SQ-IN
WEIGHT=   16.9797 LB
     B=    1.3487 IN
     H=    2.1044 IN
    RI=    4.0000 IN      DI=     8.0000 IN
    RG=    4.5183 IN     DGI=     8.7146 IN
                            DGO=     9.3587 IN
    RB=    4.9506 IN      DB=     9.9012 IN
  RFIL=     .3750 IN
 RSPOT=     .0620 IN
 DHOLE=     .4425 IN
 DSPOT=     .9012 IN
    NB=       26
    BG=     .3221 IN
    HT=    1.7500 IN

   MFU=     .11176485+05    IN-LB/IN
 ZETA1=     .76976176+00
 ZETA2=     .24211833+00
```

DATE 03107

TEST FLANGE 1500 PSI, 8 IN DIA

OUTPUT OF THE STRESS AND DEFORMATION ANALYSIS RESULTS

VARIABLE			BOLT-UP	START-UP	OPERATION	SHUT-DOWN
BOLT FORCE (LB)			.37061256+06	.48850169+06	.36011698+06	.35486920+06
EQUIV APPL MOMENT (IN-LB/IN)			.60440856+04	-.27671795+05	.91552638+04	.10710853+05
AXIAL DEFLECTION (IN)			.00000000	.95295984-05	.95295984-05	.14294398-04
RADIAL DEFLECTION (IN)			.75853278-03	-.21659444-01	.30014149-02	.18760619-02
FLANGE ROTATION (RADIAN)			.13301115-02	-.60896844-02	.20147832-02	.23571190-02
BOLT STRESS (PSI)			.13408267+06	.17673339+06	.13028552+06	.12838694+06
GASKET STRESS (PSI)			.40532359+05	.42713927+05	.28673023+05	.22743354+05
STRESS IN FLANGE TOP (PSI)			.50344101+04	-.14375453+06	.19920502+05	.12451492+05
STRESS IN FLANGE BOTTOM (PSI)			-.13543458+05	-.58698989+05	-.82202786+04	-.20470744+05
STRESS RESULTANTS	(LB/IN)	NX=	.00000000	.31640625+04	.31640625+04	.47460938+04
	(LB/IN)	NY=	.22025544+04	-.62892606+05	.87152195+04	.54475279+04
	(IN-LB/IN)	MX=	.13475052+04	-.61693181+04	.20411301+04	.23879426+04
	(IN-LB/IN)	MY=	.40425155+03	-.18507954+04	.61233904+03	.71638279+03
	(LB/IN)	QX=	.25834048+04	-.11827670+05	.39132061+04	.45781067+04
STRESSES AT NECK	(PSI) INNER	SIGX=	-.20697679+05	.99823225+05	-.26289259+05	-.29085048+05
	(PSI) INNER	SIGY=	-.26852167+04	-.72199953+05	.45388235+04	-.22875950+04
	(PSI) OUTER	SIGX=	.20697679+05	-.89698225+05	.36414258+05	.44272548+05
	(PSI) OUTER	SIGY=	.97333907+04	-.12905639+06	.23349879+05	.19719684+05
	(PSI) MAX	TAU=	.62001715+04	-.28386406+05	.93916945+04	.10987456+05

5-59

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.

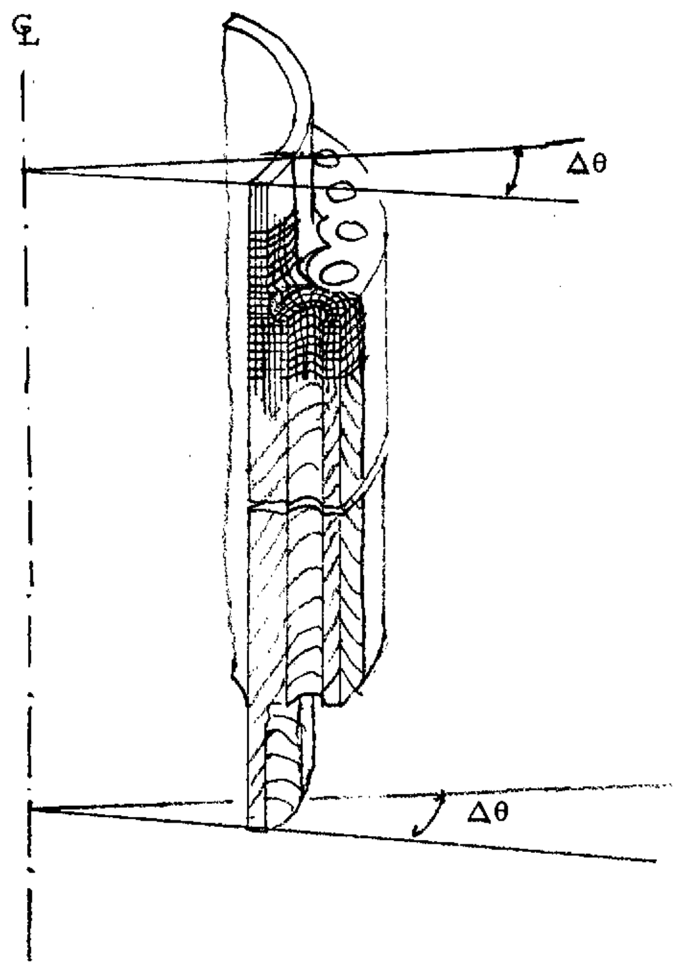


Fig. 6-1 - Slice Δθ for Finite Element Modeling

Section 7
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Section 8
NOTATION

A, B, C	coefficients for quadratic equation for h
A	cross-sectional area of the flange
A_1, A_2	amplitudes of the stresses
A_{oB}	stress area of the bolt
A_B	total bolt area
A_G	total gasket area
A_w	flange cross-sectional area used for weight computation
a, b	inner and outer tube radius, respectively
a	radial lever arm between gasket and bolts
B	bending rigidity of the shell wall
B	creep constant
b	width of the flange
$\bar{b}$	effective width of the flange
b_G	gasket width
b_{eff}	effective gasket width
b_s	width of the seal gland
C_1, C_2	integration constants of the shell equation
c	axial lever arms between centroid of flange and flange neck
c_1	distance of spotface from shell outer surface
c_1, c_2	constants in creep law
c_E	equivalent rotational spring constant of the gasket and the bolts
c_F	equivalent rotational spring constant of shell and flange

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_e	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

M_1	pipe bending moment
m	exponent in interface leakage law
m_F	applied flange moment
m_o	edge moment
m_x	meridional bending moment
m_φ	circumferential bending moment
n	creep exponent
n_B	number of bolts
n_x	axial stress resultant
n_φ	circumferential stress resultant
P	creep parameter in Lawson-Miller creep law
P_B	bolt force
P_F	total force on flange required
P_G	gasket force
P_p	force caused by internal pressure
p	pressure
Q	volume leak rate
q_o	edge shear
q_x	meridional shear stress resultant
R	stress ratio for fatigue design
r	radius of a point in the shell wall
r_o	radius of the shell wall middle surface
r_a	equivalent radius of gasket and bolts spring constant
r_B	bolt circle radius
r_c	radius of the flange centroid
r_{fil}	fillet radius on the upper surface of the flange

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_{spot}	fillet radius for the spotface
r_w	radius used for weight computation
S_1	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_{rupt}	time to rupture
u	axial displacement of the flange
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_o	tensile yield strength
Z_F, Z_T	plastic section modulus of the tube and the flange, respectively
α	linear thermal expansion coefficient
α, β	creep constants
α_1, α_2	stresses relating σ_φ and τ_{xz} to σ_x when yielding occurs

β	dimensionless parameters containing shell and flange stiffnesses
γ	angle between cylinder axis and weld
γ_1, γ_2	coefficients relating the physical gasket width to the effective gasket width at yielding and under operating conditions, respectively
Δp	internal pressure difference
ΔT	temperature difference between flange and tube
Δt	tube wall thickness tolerance
ΔW	weight added to the tube by the flange
Δw	difference in radial displacement between flange and shell
$\Delta \sigma_B, \Delta \sigma_G$	change in stress of bolts and gasket, respectively
δ_B	deflection of the bolts
δ_G	deflection of the gasket
ϵ	strain
ζ_1, ζ_2	coefficients related to the state of stress in the flange neck
η_0, η_1, η_2	design parameters for bolt spacing
μ	coefficient of friction
ν	Poisson's ratio
ρ	density of the material
ρ	shell parameter
σ	stress, contact stress
σ_B	stress in the bolts
σ_G	contact stress of the gasket
$\sigma_x, \sigma_\varphi, \tau_{xz}$	axial circumferential and radial shear stress, respectively
$\bar{\sigma}, \sigma_e$	equivalent stress
φ	circumferential angle
χ	roll angle
ψ	weld weakening factor
(F.S.)	factor of safety

Appendix A
SUMMARY OF THE DESIGN PROCEDURE

Appendix A

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

(a) Given

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

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}^{T} = tensile yield strength of the tube

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

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

where

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

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

ψ = 0.70 ... 1.00 (weld reduction)

Δt = manufacturing tolerance for the tube wall (approximately 0.01 in)

p_1 = burst pressure

p_2 = proof pressure

F_{tu}^{T} = ultimate tensile strength of the tube

F_{ty}^{T} = tensile yield strength of the tube

t = maximum of t_1 and t_2

(d) $$t_1 = \frac{1.1\,(B.F.)\,p_1\,r_i}{F_{ty}^{T} - 0.4\,(B.F.)\,p_1}$$

$$t_2 = \frac{1.1\,(P.F.)\,p_2\,r_i}{F_{tu}^{T} - 0.4\,(P.F.)\,p_2}$$

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

$$\eta_o,\ \eta_1,\ \eta_2,\ A_{oB},\ d_{hole},\ r_{spot},\ i_{size}$$

(a) Machined Spot Faces

$$e_1 = \eta_1\,d_B$$
$$e_2 = \eta_2\,d_B$$

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

(b) Machined Groove

$$e_1 = \eta_2\,d_B$$
$$e_2 = e_1$$

r_{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

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

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:

$$r_G = 1/2\,(r_B - \frac{d_{hole}}{2} + r_i)$$

Gasket Width, b_G:

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

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

where

γ_1 = a width factor for the gasket under initial deformation

K_G = the yield (crushing) strength of the gasket

γ_2 = a width factor for the gasket under operating condition

σ_G = seating stress of the gasket

(G.F.) = gasket factor

k_p = ratio of seating stress over pressure

(c) $b_G = r_B - \frac{d_{hole}}{2} - r_i - 2\,c_2$ (the available space is used)

where c_2 is a tolerance. c_2 = 0.05 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

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

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

where

$$P_p = \pi\ r_G^2\ p\ (P.F.)\ .$$

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:

$$r_1 = r_G + b_G/2 + c_2$$

$$r_2 = r_B - d_{hole}/2 - c_2$$

if $r_1 > r_2$ set

$$r_G = r_i + b_G/2 + c_2$$

$$r_B = r_G + b_G/2 + d_{hole}/2 + c_2$$

(2) Gasket Close to the Bolts

$$r_G = r_B - \frac{d_{hole}}{2} - \frac{b_G}{2} - c_2$$

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

$$r_G = r_i + b_G/2$$

check if space for seal gland is sufficient:

$$r_1 = r_i + b_G + 2\,b_s$$

$$r_2 = r_B - d_{hole}/2$$

if $r_1 > r_2$ set

$$r_B = r_G + b_G/2 + 2\,b_s + d_{hole}/2$$

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.7 REQUIRED BOLT FORCE P_B (lb) OR (N)

(a) Flat Gasket (see A.5)

$$P_B^{(1)} = 2\pi\, r_G\, b_G\, \gamma_1\, K_G$$

$$P_B^{(2)} = \pi\, r_G^2\, p\, (P.F.) + 2\pi\, r_G\, b_G\, \gamma_2\, \sigma_G\, (G.F.)$$

or

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

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

(b) Pressure Energized Seal

$$P_B = \pi\, r_G^2\, p\, (P.F.)$$

A.8 NUMBER OF BOLTS, n_B

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

$$n_{B2} = \frac{(B.F./P.F.)\, P_B}{F_{tu}^B\, A_{oB}}$$

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

where

F_{ty}^B = tensile yield strength of the bolt

F_{tu}^B = ultimate tensile strength of the bolt

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

$$s = 2\pi\, r_B / n_B$$

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

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

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

$$e = r_B - r_G$$

$$r_o = r_i + t/2$$

$$t_N = t/2$$

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

$$\overline{b} = b - d_{hole}\sqrt{d_{hole}/s}$$

Assume

$$\zeta_1 = 0.8$$

$$\zeta_2 = 0.18$$

and compute

$$A = F_{ty}^F\, \overline{b}/6\, r_o$$

$$B = F_{ty}^F\, \zeta_2\, (t - t_N)/2$$

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

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

$$h = (R - B)/2A$$

where F_{ty}^{F} = tensile yield strength of the flange.

A.11 CHECK FLANGE HEIGHT

$$\text{if } s/h > 3 \qquad h = s/3$$

to prevent waviness of the flange when too thin.

A.12 WEIGHT ΔW (lb) OR (kg)

$$r_w = \frac{2\,r_i + t + b}{2}$$

$$A_w = (b - t)\,h$$

$$Vol = 2\pi\, r_w\, A_w$$

$$\Delta W = \rho_F \cdot Vol$$

where ρ_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

Appendix B

B.1 APPLIED FORCES

$$f_x = p\,r_o/2 \quad \text{(axial force in the tube wall)}$$

$$f_r = p\,h\,r_i/r_o \quad \text{(radial force on the flange)}$$

$$f_p = p(r_G^2 - r_i^2)/2\,r_o \quad \text{(pressure force on the flange face)}$$

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

$$A_G = 2\pi\, r_G\, b_G$$

$$K_G = A_G E_G/2\pi\, r_o\, h_G$$

(Linear spring constant for flat gasket)

$$K_G = A_G\, E_F/2\pi\, r_o(h_R + h_s)$$

(Linear spring constant for cantilever flange with pressure energized seal)

where

A_G = gasket area

h_G = gasket thickness

h_R = depth of the recess

h_s = depth of the seal gland

E_G = elastic modulus of the gasket

E_F = elastic modulus of the flange

(b) Bolts

$$A_B = n_B A_{oB}$$

$$K_B = A_B E_B / 2\pi r_o \ell_B$$ (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

$$A_F = b\,h$$

$$I_F = A_F\,h^2/12$$

$$c = h/2$$

$$r_c = r_i + b/2$$

$$B = E\,t^3/12(1-\nu^2)$$

$$k = 12(1-\nu^2)/r_o^2\,t^2$$

$$\beta = Bk\,r_o\,r_c/E_F\,I_F$$

$$D = (1+\beta)\left[\frac{1}{2k^2} + \beta\,(c^2 + \frac{I_F}{A_F})\right] - (c\beta - \frac{1}{2k})^2$$

$$c_F = \frac{B\,D}{\beta\left(\frac{\beta I_F}{k\,A_F} + \frac{1}{4k^3}\right)}$$ (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

β = 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

$$c_w = 1/2k^3 B$$

$$c_m = \frac{\beta}{D\left(\frac{1}{2k^2} + \beta \frac{I_F}{A_F} + \frac{c}{2k}\right)}$$

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

B.3 INITIAL TORQUING (o)

$$m_F^{(o)} = e\, P_B/2\pi\, r_o \qquad \text{(Applied flange moment)}$$

$$\chi^{(o)} = m_F^{(o)}/c_F \qquad \text{(Flange rotation)}$$

$$\sigma_B^{(o)} = P_B/A_B \qquad \text{(Bolt stress)}$$

$$\sigma_G^{(o)} = P_B/A_G \qquad \text{(Gasket stress)}$$

Variables at the flange neck:

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

$$m_x^{(o)} = c_m\, m_F^{(o)} \qquad \text{(Meridional bending moment)}$$

$$q_x^{(o)} = c_q\, m_F^{(o)} \qquad \text{(Shear force)}$$

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

$$n_y^{(o)} = E_T\, t\, w^{(o)}/r_o \qquad \text{(Circumferential force)}$$

$$m_y^{(o)} = \nu\, m_x^{(o)} \qquad \text{(Circumferential bending moment)}$$

B.4 PRESSURIZATION (p)

$$m_F^{(1)} = f_p\,(r_a - r_p) + f_x\,(r_a - r_o)$$ (Applied flange moment from pressure)

$$\Delta w = \frac{p\,r_o^2\,(1 - \nu/2)}{E_T\,t} + r_o\,\alpha_T\,\Delta T - \frac{r_o\,r_c\,f_r}{E_F\,A_F}$$ (Difference in radial deflection between tube and flange)

$$m_F^{(2)} = \frac{\beta\,(c + 1/2k)\,\Delta w}{D}\,c_F$$ (Equivalent applied flange moment due to Δw)

$$u^{(p)} = (f_p + f_x)/(K_G + K_B)$$ (Axial displacement)

$$(p) = \frac{m_F^{(1)} + m_F^{(2)}}{c_E + c_F}$$ (Additional flange rotation due to pressurization)

$$\chi^{(T)} = \chi^{(o)} + \chi^{(p)}$$ (Total flange rotation)

$$\delta_G = u^{(p)} + \chi^{(p)}\,K_B\,e/(K_G + K_B)$$ (Deformation of the gasket)

$$\delta_B = u^{(p)} - \chi^{(p)}\,K_G\,e/(K_B + K_B)$$ (Deformation of the bolts)

$$\sigma_G^{(T)} = \sigma_G^{(o)} + E_G\,\delta_G/h_G$$ (Total stress in the gasket)

$$\sigma_B^{(T)} = \sigma_B^{(o)} + E_B\,\delta_B/\ell_B$$ (Total stress in the bolts)

Variables at the flange neck:

$$n_x^{(p)} = f_x$$ (Axial force)

$$m_F^{(T)} = c_F\,\chi^{(T)}$$ (Total equivalent applied flange moment)

$$m_x^{(T)} = c_m\,m_F^{(T)}$$ (Meriodional bending moment)

$$q_x^{(T)} = c_q\,m_F^{(T)}$$ (Shear force)

$$w^{(T)} = c_w \left(q_x^{(o)} - k\, m_x^{(o)}\right) + \frac{p\, r_o^2 (1 - \nu/2)}{E_T\, t} + r_o\, \alpha_T\, \Delta T \quad \text{(Radial deflection)}$$

$$n_y^{(T)} = \frac{E_T\, t\, w^{(T)}}{r_o} - r_o\, \alpha_T\, \Delta T + p\, r_o \frac{\nu}{2} \quad \text{(Circumferential force)}$$

$$m_y^{(T)} = \nu\, m_x^{(T)} \quad \text{(Circumferential bending moment)}$$

B.5 STRESSES IN THE FLANGE

$$w_{top} = w$$

$$w_{bottom} = w - h\chi$$

$$\sigma_{top} = E_F\, w_{top}/r_o$$

$$\sigma_{bottom} = E_F\, w_{bottom}/r_o$$

B.6 STRESSES IN THE FLANGE NECK

$$\sigma_x = \pm \frac{6\, m_x}{t^2} + \frac{n_x}{t}$$

$$\sigma_y = \pm \frac{6\, m_y}{t^2} + \frac{n_y}{t}$$

$$\tau_{xz} = 1.5\, q_x/t \quad \text{(max)}$$

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

$$n_x(x) = n_x(o)$$

$$w_x(x) = c_w e^{-k_x} \left[q_x(o) \cos kx - k\, m_x(o) (\cos kx - \sin kx) \right]$$

$$+ \frac{p\, r_o^2 \left(1 - \frac{\nu}{2}\right)}{E_T\, t} + r_o\, \alpha_T\, \Delta T$$

$$m_x(x) = e^{-k_x} \left[m_x(o) (\cos kx + \sin kx) - \frac{q_x(o)}{k} \sin kx \right]$$

$$q_x(x) = e^{-k_x} \left[q_x(o) (\sin kx - \cos kx) + 2\, k\, m_x(o) \sin kx \right]$$

$$n_y(x) = Et\, w_x(x)/r_o - r_o\, \alpha_T\, \Delta T + p\, r_o\, \nu/2$$

$$m_y(x) = \nu\, m_x(x)$$

The stresses at a point x are then computed according to paragraph B.6.

B.8 PLASTIC HINGE

$$\alpha_1 = \frac{t^2 - t_n^2}{4} \frac{E}{r_o} \frac{\frac{\beta}{k} \frac{I_F}{A_F} + \frac{1}{4k^3}}{B\left(\frac{1}{2k^2} + \beta \frac{I_F}{A_F} + \frac{c}{2k}\right)} w + \nu$$

$$\alpha_2 = \frac{t + t_n}{4} \frac{c + \frac{1}{2k}}{\left(\frac{1}{2k^2} + \beta \frac{I_F}{A_F} + \frac{c}{2k}\right)}$$

$$\bar{\alpha} = \sqrt{1 + \alpha_1 + \alpha_1^2 + 3\, \alpha_2^2}$$

$$\zeta_1 = 1/\overline{\alpha}$$

$$\zeta_2 = \alpha_2/\overline{\alpha}$$

$$S_x = b\,h^2/2$$

$$m_{F_u} = Y_o\left[\frac{S_x}{r_o} + \frac{\zeta_1}{4}\,(t^2 - t_n^2) + \zeta_2\,(t - t_n)\,\frac{h}{2}\right] \qquad \text{(Ultimate applied flange moment)}$$

Appendix C
INPUT INSTRUCTIONS FOR DESIGN AND ANALYSIS PROGRAM

Appendix C

Card	Format	Column	Description	Units
0	12A6	1-72	Instruction to plotter operator	—
1	I5	1-5	Number of cases	—
2	12A6	1-72	Title of the plots	
3	E10.4	1-10	p = pressure	psi (N/mm^2)
		11-20	d_i = inner diameter of the tube	in. (mm)
		21-30	t = thickness of the tube	in. (mm)
		31-40	Δt = thickness tolerance	in. (mm)
		41-50	h_T = height of tube frustum being part of the flange	in. (mm)
4	E10.4	1-10	P.F. = proof factor	—
	E10.4	11-20	B.F. = burst factor	—
	E10.4	21-30	F.S. = factor of safety	—
		31-40	G.F. = gasket factor	—
5	I5	1-5	i_T = tube material number (see Table 2-6)	—
		6-10	i_F = flange material number (see Table 2-6)	—
		11-15	i_B = bolt material number (see Table 2-6)	—
			i_G = gasket material number (see Table 2-7)	
6			Only if i_T = 0 Material properties of tube	
	E10.4	1-10	E_T = elastic modulus	psi (N/mm^2)
		11-20	ν_T = Poisson's ratio	—
		21-30	ρ_T = density	lb/in^3 (kg/mm^3)

Card	Format	Column	Description	Units
6	E10.4	31-40	α_T = thermal expansion coefficient	in/in/^{o}F (mm/mm/^{o}C)
		41-50	F_{ty}^T = tensile yield strength	psi (N/mm^2)
		51-60	F_{tu}^T = ultimate tensile strength	psi (N/mm^2)
7			Only if $i_F = 0$ Material properties of flange (same format and description as card 6)	
8			Only if $i_B = 0$ Material properties of bolt (same format and description as card 6)	
9			Only if $i_G = 0$ Material properties of gasket	
	E10.4	1-10	E_G = elastic modulus	psi (N/mm^2)
	E10.4	11-20	K_G = compressive strength	psi (N/mm^2)
	E10.4	21-30	σ_G = seating stress	psi (N/mm^2)
		31-40	α_G = thermal expansion coefficient	in/in/^{o}F
		41-50	μ_G = friction coefficient	— (mm/mm/^{o}C)
			Only if $i_G < 0$ Pressure energized seal	
	E10.4	1-10	h_s = depth of the seal gland	in.(mm)
	E10.4	11-20	b_s = width of the seal gland	in. (mm)
		21-30	h_R = depth of the recess	in. (mm)
10	10I5	1-50	Options:	
			Option 1 = 0: Read tube thickness from card 3:	—
			Option 1 = 1: Tube thickness computed according to Appendix A, paragraph A.1(b):	—

Card	Format	Column	Description	Units
10	10I5	1-50	Option 1 = 2: Same, but paragraph A.1(c)	–
			Option 1 = 3: Same, but paragraph A.1(d)	–
			Option 2 = 1: Machined spot faces	–
			Option 3 $\leq$ 0: Flat gasket	–
			Option 3 $\geq$ 1: Pressure activated seal	–
			Option 4 = 1: Open wrenching (see Table 2-1 or 2-4)	–
			Option 4 = 2: Socket wrenching (see Table 2-2 or 2-5)	–
			Option 4 = 3: Internal wrenching (see Table 2-3 or 2-6)	–
			Option 5 = 0: Gasket width according to paragraph A.5(a)	
			Option 5 = 1: Gasket width according to paragraph A.5(b)	
			Option 5 = 2: Gasket width according to paragraph A.5(c)	
			Option 6 $\neq$ 3: Gasket close to bolt circle	
			Option 6 = 3: Gasket close to inside of tube	
11	2A6	1-12	Name of tube material	–
	2A6	13-24	Name of flange material	–
	2A6	25-36	Name of bolt material	–
	2A6	37-48	Name of gasket material (or seal, as applicable)	–
12	I5	1-5	Number of loading phases	–
13	E10.4	1-10	Temperature differential between tube and flange	^{o}F (^{o}C)