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APPLICATION OF LOW PROFILE FLANGE DESIGN FOR SPACE VEHICLES

By

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ABSTRACT

There is evidence that the "low profile" flange configuration may be used advantageously in bolted flange connections.

Test data substantiate the concepts of the "low profile" design.

The application of this flange for short tubes, manhole, and manhole covers will increase the reliability, and reduce weight and space for the installation.

The low profile flange equalled or exceeded the performance of the Taylor forge-type flange. The tested specimens were up to 37 percent lighter and also less bulky.

No data are available yet on this flange design used on long tube connections. Additional testing is underway. Results will be published soon.

SUMMARY

There is evidence that a new approach to the design of bolted separable connectors may be used advantageously in missile and space vehicles as well as in other applications.

Test data substantiates concepts of the "low profile" design.

The application of this flange design for short tubes and manholes will increase the reliability of flange connections by reducing weight.

The "low profile" flange equalled or exceeded the performance of the Taylor Forge type flange. The tested low profile flanges were lighter (up to 37 percent) and were less bulky. Operating stresses in the "low profile" flanges are much more uniform and are lower than in conventional flanges.

No experience data are available yet on the behavior of this flange design as used on longer tube connections, but additional testing is underway. Test results will be published as time and accomplishment warrant.

BOLTED FLANGE JOINTS have been used extensively throughout the whole aerospace industry in missile and spacecraft design. These flange connections are exposed to all conceivable environments, such as high and low temperature, high and low pressure, exiting vibration, shock, pressure peaks, and differences of expansion and contraction of gasket and flange material as well as of the bolts themselves. These conditions made it more and more difficult for the designer to fulfill the requirements of leak tight connectors. We are all aware of the great risk to mission and of missile loss due to flange leakage which can cause an explosion, a pressure loss, or propellant or gas depletion, etc.

In man-rated vehicles, safety requirements do not permit leakage potentials which would jeopardize the astronaut's life. There has been a tremendous effort throughout the whole country to improve separable connectors. The results obtained are promising. We are not too far from reaching our final goal - absolutely no leaks.

This report introduces a concept of a flange design that is not yet very well known but test results are encouraging for the application of this configuration in the missile and spacecraft design as well as for other applications.

The basic principles of this design concept were developed and presented by Professor lr. H. H. Boon and Mr. lr. H. H. Lok in their article Investigation on Flanges and Gaskets (Untersuchungen an Flanschen and Dichtungen) published December 1958 in the Terman technical periodical VDI #100-34. This theory has been used as a basis for the development of equations for the design of "low profile" flanges. for practical applications.

BASIC THEORY OF THE "LOW PROFILE" FLANGE FROM THE REPORT INVESTIGATION ON FLANGES AND GASKETS BY PROF. E. F. BOON AND H. H. LOK DELFT, NETHERLANDS

From a comparison of the results of tests conducted on models and full size flanges, simple non-dimensional formulas were derived for the design of the low profile flange.

Because of the mathematical interrelation of loads and flange rolling (radial rotation), and since flange distortion can be measured quite reliabily, this interrelationship is used for determining the highest possible flange load. Flange Loads also have a great influence on the sealing characteristics of a joint. Model tests made it possible to determine the behavior of a flange under pressure and temperature fluctuation.

For the gaskets, it is known that the sealing characteristics depend on many factors which are often hard to define and standardize such as: the surface finish of the gasket and the flange, the influence of temperature, the applied pressure as a function of time, etc. It is also known that the method of leakage detection and the determination of the magnitude of a leak present still more problems, certainly at extremely high pressure and high and low temperature ranges.

In the last few years, major steps have been made in gasket design and in the development of new materials for gaskets.

The characteristics of the "low profile" flange configuration are as follows: (see Fig. 1 and 1a)

a. The distance "a_i" between the force acting along the tube and the one acting on the bolt is kept a minimum.

b. The axial tube pressure load F_T and gasket load F_D fall almost in the same line. This is done so that when F_T caused by pressurizing the tube is added to the resultant of the couple F_F and F_D, the relative movement of their lines of action is very slight and the resultant of tube load F_T and gasket load F_D do not shift relative positions. Thus, the flange will not deflect (rotate) additionally. For this assumption, the gasket is considered to be nonflexible (thin metal, for example). By placing the gasket close to the inner edge of the flange, it will oppose a flange rotation. In addition, the influence of the radial action force must be accounted for as it affects the flange distortion by tending to rotate it additionally, because the internal pressure, acting radially, stretches the tube wall more than the flange.

c. The neck is not thickened. This reduces the moment arm which simplifies the calculations since the position of the plastic joint is fixed at the transition from the tube to the flange.

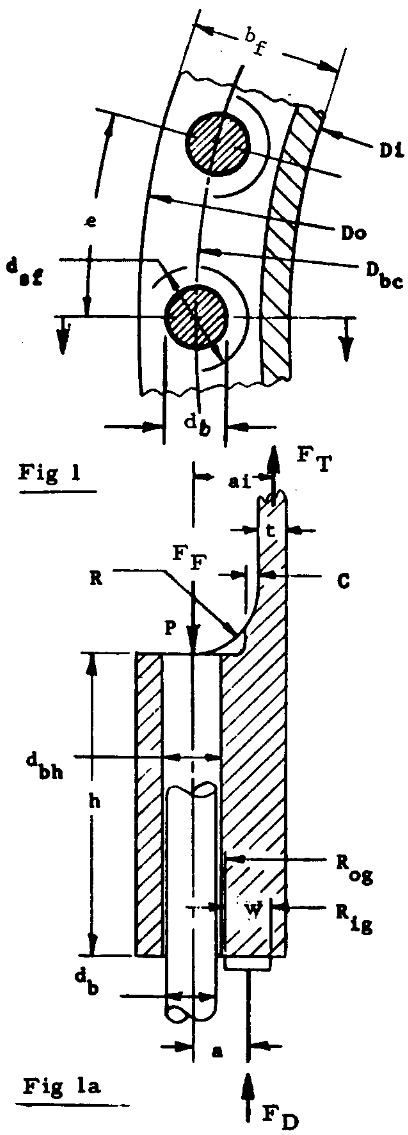


Fig. 1 - Cross-section of the low profile configuration

d. Size and type of gasket are chosen to suit the flange.

e. The bolts are under stress because of internal pressure which in turn acts on the area of the bolt circle diameter.

f. Creep of the material for bolts and flanges used can normally be neglected but if it is expected then insert the value for the pressure that will cause an elongation of 10^{-3}% after testing for 1,000 hrs.

<u>Preliminary estimation of flange dimensions.</u>
The wall thickness "t" of the tube has a value of

$$\frac{t}{R_i} = \frac{p_i \cdot N_f}{F_{yf}} \qquad (1)$$

where N_f assumes a value of 1.5.
The number of bolts "n" is expressed by

$$n = \frac{2\pi \cdot R_{bc}}{e}$$

according to assumption (e)

$$\pi \cdot R_{bc}^2 \cdot p_i = \frac{2\pi R_{bc}}{e} \cdot \frac{\pi \cdot d^2}{4} \cdot \frac{1}{N} \cdot F_{yb} \qquad (2)$$

Then from equation (1) and (2) we obtain

$$\frac{d}{t} = \frac{2}{\pi} \frac{R_{bc}}{R_i} \cdot \frac{F_{yf}}{F_{yb}} \cdot \frac{e}{d} \qquad (3)$$

The ratio R_{bc}/R_i is dependent on the internal pressure and the smallest possible bolt diameter and should fall between 1.05 and 1.5.

It is recommended that the bolts be as close as possible to the tube wall; therefore, many small diameter bolts should be used instead of fewer large diameter bolts.

Select bolt material having yield stress considerably higher than that of the flange or tube material.

According to this reasoning, one derives from equation (3)

$$\frac{R_{bc}}{R_i} = 1.2 \quad \text{and} \quad \frac{F_{yf}}{F_{yb}} = \frac{1}{3}$$

as well as $\frac{e}{d} = 4$

$t \approx d$

A flat cover base of the height "h" is regarded as a plate freely supported at the bolt circle. Then the following is considered valid through the elastic range.

$$\left(\frac{h}{R_{bc}}\right)^2 = 1.25 \frac{p_i N_f}{F_{yf}} \qquad (4)$$

For a rigid plastic material the plate begins to deform at:

$$\left(\frac{h}{R_{bc}}\right)^2 = \frac{2}{3} \frac{p_i}{F_{yf}} \qquad (5)$$

and it is advisable to choose:

$$\left(\frac{h}{R_{bc}}\right)^2 = 1.25 \frac{p_i}{F_{yf}} \qquad (6)$$

If in a compact flange with a cylindrical neck, the influence of the flange neck is neglected, then the internal tangential moment in the flange M_t resists the bolt moment in a flange sector of arc length 1 (one), i.e., $M_s/2\pi$. The forces acting on this flange sector are shown in Fig. 2 and 2a. If after deducting the

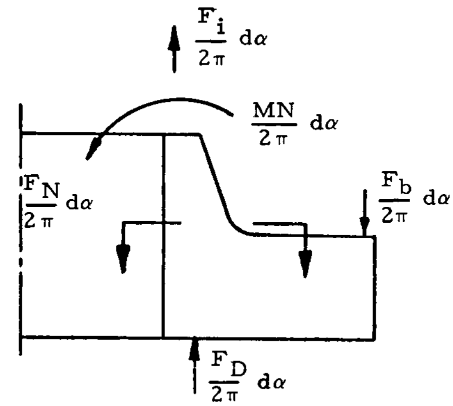


Fig. 2

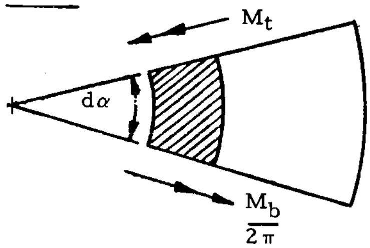


Fig. 2a

Axial loads: Axial internal pressure load F_{ii} gasket load F_D; external. bolt load F_b.

Radial loads: Sheer load in flange neck F_N; radial internal pressure load; radial portion of the tangential pressure

Moments in the radial plans:
Radial portion of the internal tangential moment M_t; internal moment in flange neck M_{Ni} moment of the sheer load in the flange neck; external boltload M_b. $d\alpha$; sector angle

Fig. 2 - Forces acting on flange sector

bolt hole the total stressed flange width is
$2t + 1.3d$
and the moment arm of the bolt is

$$a = 1.1d + 1.5t$$

then the value of $M_t = M_s/2\pi$

or with $d \approx t$

$$\left(\frac{h}{R_{bc}}\right)^2 \approx 1.6 \frac{p_i}{F_{yf}}$$

Using a safety factor factor of 1.5 we obtain

$$\left(\frac{h}{R_{bc}}\right)^2 \approx 2.4 \frac{p_i}{F_{yf}} \qquad (7)$$

The greatly simplified formulas recorded here show the relationship between the principal dimensions of a flange for a pressure vessel as follows:
Proportionality relationship of wall thickness "t" and tube radius "R_i" per equation (1).
Proportionality relationship of wall thickness "t" and bolt diameter "d" equation (3).
Proportionality relationship of flange height "h" and bolt circle radius R_{bc} per equation (7).
In these simplified equations of course the influence of the forces in the neck are not considered.
For the present, one can put in flanges with cylindrical necks

$$\left(\frac{h}{R_s}\right)^2 \approx 1.5 \frac{p_i}{F_{yf}} \qquad (8)$$

whereby the factor 1.5 is to be considered as a function of h/R_{bc} to be verified by the final test results.
Figures 3, 4 and 5 show how the shape of the flange ND 25NW1000 according to DIN (German Industrial Standards) changes when the preceeding equations or formulas are used.
The gasket dimensions are derived from the equilibrium of forces for the installed and operating condition. For the installed (unpressurized) condition the following equation applies:

$$\pi R_{bc} p_i = 2\pi . R_i . W_i . p'_G$$

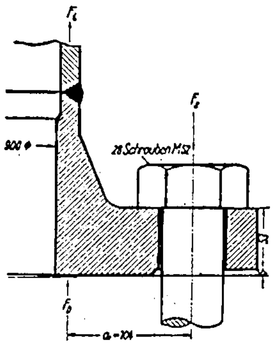


FIGURE 3

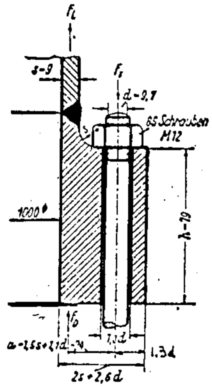


FIGURE 4

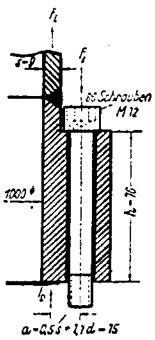


FIGURE 5

Fig. 3 - 4 - 5 - Changes in flange configuration resulting from application of different formulas

For the operating (pressurized) condition the equation is:

$$\pi R_i^2 \, p_i + 2\pi . R_i . W_i . p_G = \pi R_{bc}^2 \, p_i \quad \pi . R_{bc}^2 . p_i$$

By again putting $R_{bc} = 1.2 R_i$ one obtains

$$\frac{2w}{1.2R_{bc} - R_i} \approx \frac{p_i}{p'_G}$$

and with $P_G > p_i$ we get the condition

$$\frac{2w}{1.2\,R_{bc} - R_i} < 1$$ which is always satisfied.

DERIVATION OF FORMULAS FOR THE DESIGN

Summarizing, it is established that the dimensions of the wall, flange and bolts, can first be approximated according to the following formulas:

1. $t = d$

2. $$\frac{t}{R_i} = e_1 \frac{p_i}{F_{yf}}$$

   where $e_1 = 3/2$ for cylindrical tubes

   $e_1 = 3/4$ for flanges on spheres

3. $$\frac{W}{R_i} = e_2 \frac{p_i}{p_G'}$$

   where e_2 is between .55 and 1.15

4. $$\left(\frac{h}{R_{bc}}\right)^2 = e_3 \frac{p_i}{F_{yf}}$$

   where $e_3 = 1.25$ for circular plate

   $e_3 = 1.5$ for a flange with cylindrical neck and

   $e_3 = 2.4$ for a free flange

These formulas are dimensionless. They furnish the relations between the dimensions of the tube, the bolts, and the gaskets as a function of the proportion between internal pressure and stress; they are merely to enable a first estimate of the flange dimensions.

These formulas are not valid for conduit flanges unless external forces and moments are taken into consideration. The influence of corrosion and of impact forces in the system have likewise not been considered.

Using the above mentioned theory and combining it with existing theories on Taylor forge flanges, the following procedure to calculate dimensions of a low profile flange has been derived.

Flange design procedure

There are many variables and unknown parameters to consider in designing a flanged joint. The variables can be controlled by setting limits on design criteria, choice of materials, etc.; but for the unknowns, such as effects of temperature and other conditions, reasonable assumptions based on experience and judgment must be made until more data have been obtained.

Step one (Data Compilation)

Gather all known data (operating pressure, operating temperature, operating media, flange and gasket materials, size, external forces, available space, type of fastener, etc.). Establish criteria and factors for:

a. Performance and reliability required.
b. Weight restrictions.
c. Other items that could affect the design.

Step Two (Calculation of Wall Thickness)

Calculate the wall thickness (t) of the tube or vessel. Ordinarily, this will have been done already, but if not, the calculation can be performed with the usual formulas for pressure vessels:

$$t = \frac{p_i D_i N_f}{2F_{yf}} \quad \text{(cylinder)} \qquad \text{I}$$

Where:

P_i = Design internal pressure

D_i = Inside diameter of tube

N_f = Safety factor applied to flange material

F_{yf} = Yield stress of flange material

Step Three (Preliminary Determination of Bolt Size)

The bolt size will determine the flange width. It has been shown previously that a relatively high, narrow flange will be much more stable than a low, wide flange with approximately the same cross-sectional area. Therefore, the smaller the diameter of the bolt that is used, the narrower the flange may be. An assumption can be made at this stage--based on experience and judgment--for the smallest practical bolt diameter (d_b).

Step Four (Calculation of Bolt Circle Diameter)

With the bolt size tentatively established, the bolt circle diameter (D_{bc}) can then be determined by the following equation:

$$D_{bc} = D_i + 2t + 2C + d_{sf} \qquad \text{II}$$

Where:

d_{sf} = Spotface diameter

C = Minimum practical clearance between spotface and tube wall

The spotface is selected on the basis of wrench or washer requirements, whichever is applicable. The spotface diameter should be held to a minimum.

Step Five (Calculation of Flange Width)

Since the flange will be relatively high, there is no need for excessive edge-distance from the bolt circle to the outside diameter of the flange. Unless conditions dictate otherwise, an edge-distance of one nominal bolt diameter from the bolt circle to the outside diameter should be used. Now, we can find the outside diameter (D_o) and flange width (b_f) using the following equations:

$$D_o = D_{bc} + 2d_b \qquad \text{III}$$

$$b_f = \frac{D_o - D_i}{2} \qquad \text{IV}$$

Step Six (Calculation of Gasket Width)

For relatively low pressure (up to about 1,000 psi) a soft, flat gasket can be used to advantage. For higher pressures, a pressure-energized gasket of some type should be used. The problem of the gasket has been amply covered in other works; but, a useful formula for determining the width (w) of a soft, flat gasket is as follows:

$$w = \frac{p_i\,(R_{bc} - d_b + R_i)\,N_g}{2(F_{ug} - 2p_s)} \qquad \text{V}$$

Where:

F_{ug} = Ultimate or crushing strength of gasket material

N_g = Factor of safety for gasket

With the gasket width established, the inside and outside diameters may be determined. Whether the gasket will lie near the inside diameter of the flange, or near the bolts, will depend on the prevailing conditions and the judgment of the designer. However, the following items should be considered.

a. Placing the gasket close to the inside diameter of the flange will be advantageous because it minimizes the axial pressure load affecting the joint; also, any additional flange-rolling caused by pressurizing will be held to a minimum, because the moment arms between the tube wall and bolt and the gasket and bolt are more nearly the same. A disadvantage of this arrangement is that there is no locating ring or guide and the gasket could slip out of place causing trouble.

b. If the gasket is placed near the bolts, the bolts act as a guide to keep the gasket in position. This arrangement has the disadvantages of increasing the rolling phenomena when pressurizing, and of permitting the maximum axial pressure load to exist.

Step Seven (Calculation of Gasket Load)

After the gasket dimensions have been determined, calculate the total initial gasket load (P_g) with the following equation:

$$P_g = \pi p_i \left(\frac{R_{og} + R_{ig}}{2}\right)^2 + 2\,p_s\,w\left(\frac{R_{og} + R_{ig}}{2}\right) \qquad \text{VI}$$

Where:

R_{og} = Outside radius of gasket

R_{ig} = Inside radius of gasket

Step Eight (Calculation of Number of Bolts)

Calculate the number of bolts (n) required with the following equation:

$$n = \frac{P_g N_b}{P_{yb}} \qquad \text{VII}$$

Where:

N_b = Safety factor for bolt

P_{yb} = Yield load for bolt

Because bolt torques are subject to many variants, a high safety factor should be incorporated to ensure that the yield stress of the bolt is not exceeded. Other, more accurate methods allowing better employment of the bolt strength, have been used for determining bolt pre-load, but none of these methods have been found satisfactory for general use. A safety factor utilizing approximately 1/2 to 2/3 of the yield strength of the bolt is satisfactory, except in cases in which no lubricant is used (lubricant is mandatory in critical applications), and the torque/tension ratio varies considerably.

Step Nine (First Design Check)

Bolt spacing (e) may be determined by dividing the circumference of the bolt circle (D_{bc}) by the number of bolts (n)

$$e = \frac{\pi D_{bc}}{n} \qquad \text{VIII}$$

At this point, the question of practicality should be introduced. If the bolt spacing is less than three times the nominal bolt diameter, a larger bolt should be used (or possibly a different bolt material). If it is decided that a larger bolt should be used, backtrack to Step Three and redesign the flange based on the larger bolt size. Step Five and Step Six may be used as originally calculated, because no significant change will result in these steps.

If the bolt spacing is greater than eight times the nominal bolt diameter, a smaller bolt may be used; however, this depends on the judgment of the designer or engineer. This does not preclude the use of bolts that are larger than necessary, but caution should be exercised, when using excessive bolt capacity, to specify a torque value that will not overstress the flanges or gasket.

Step Ten (Calculation of Flange Height)

The height of the flange depends primarily on the stresses introduced by the bolt-to-gasket and bolt-to-tube wall moments. Although these moments acting radially tend to roll the flange, the resisting moment is tangential (refer to section on Low Profile Flange Theory). The longitudinal dimension can be computed by the following equation:

$$h^2 = \frac{3(P_i a_i + P_s a_s)N_f}{\pi F_{yf}(b_f - d_{bh})} \qquad \text{IX}$$

(For rectangular flange section only)

Equation IX was derived by substitution into the basic equation for beam bending

$$F_b = \frac{M_a R_{cg} c}{I}$$

Where:

M_a (applied uniform radial moment per unit of circumference) is equal to the total applied moment (P . a) divided by the circumference of the locus of the cg of the radial section, or $\frac{P \,.\, a}{2\ R}$ $c = \frac{h}{2}$, and $I = \frac{b_f h^3}{12}$

Then substituting:

$$f_b = \frac{\left(\frac{P \,.\, a}{2\pi R_{cg}}\right)\left(R_{cg}\right)\left(\frac{h}{2}\right)}{\frac{b_f h^3}{12}}$$

$$= \frac{3(P \,.\, a)}{\pi b_f h^2}$$

Transposing for (h):

$$h^2 = \frac{3(P \,.\, a)}{f_b b_f}$$

Now, applying a safety factor (N_f), substituting the effective width ($b_f - d_{fh}$) for total width (b_f) and the yield stress (F_{yf}) for the actual stress (f_b)

$$h^2 = \frac{3(P \,.\, a)\, N_f}{\pi F_{yf}\,(b_f - d_{bh})}$$

Because there are two distinct moments involved--one from internal pressure load (P_i) with its moment arm (a_i) between the tube wall and bolt axis and one from the gasket seating load (P_s) with its moment arm (a_s) between the gasket center of pressure and bolt--that comprise the total moment (P . a), substituting these terms for (P . a) will result in equation IX for the longitudinal dimension.

If it can be seen that the internal pressure load moment will contribute a large portion of the total moment, then the following equation may be conservatively used to derive the height of the flange

$$h^2 = \frac{3(P_g a_i)(N_f)}{\pi F_{yf}\,(b_f - d_{bh})} \qquad \text{(X)}$$

Step Eleven (Second Design Check)

At this point, the flange height (h) should be checked against the bolt spacing (e) using the following formula:

$$\frac{3h}{e} > 1 \qquad \text{(XI)}$$

If this condition is not satisfied, more bolts should be used, or possibly smaller diameter bolts. If it is decided to use smaller-diameter bolts, return to Step Three and redesign the flange based on the smaller bolt size. Again, Step Five and Step Six may be used as originally calculated. If it is decided to use larger bolts, caution should be exercised in specifying torque values to avoid overstressing the flange or gasket.

The eleven preceding steps, if followed, will produce a balanced flange design with relatively uniform stresses throughout. This, of course, is basic to any good design and will result in the "best" or "most efficient" use of the total material and a maximum strength-weight ratio. The flange face can be adapted to almost any seal configuration desired.

The effect of the tube wall on the flange has not been considered in this design procedure; because at initial tightening of the bolts it was conservative to ignore it,and when pressure is applied to the system, the influence of the tube wall diminishes. This happens because the tube wall resists the initial rolling of the flange, but when pressure is applied, the tube wall, being thinner than the flange, stretches more than the flange and relieves part--and possibly all--of the initial resistance to the rolling of the flange.

Actually, the most difficult part in the design of a flange is the selection of the gasket and the determination of how best to calculate its loads, allowables, etc. The equations in Step Five are for soft, flat gaskets only; equations for any other type of gasket or configuration must be formulated according to the peculiar requirements.

The preceding eleven steps are summarized below. There has been no attempt to present a detailed theoretical analysis of flange design,because it is felt that practical application of the simplest possible equations will produce a flange as close to the ideal as could be accomplished with a highly complicated theoretical approach. These equations are actually based on the theoretical form but have been modified to a semi-empirical form for practical applications.

SUMMARY OF DESIGN PROCEDURE

Step One: Collect all known data and factors.

Step Two: Calculate wall thickness (t)

$$t = \frac{p_i D_i N_f}{2F_{yf}}$$

Step Three: Assume a bolt size (d_b)

Step Four: Determine a bolt circle diameter (D_{bc})

$$D_{bc} = D_i + 2t + 2C + d_{sf}$$

Step Five: Using edge distance = d_b, determine outside diameter (D_o)

$$D_o = D_{bc} + d_b$$

and width of flange

$$b_f = \frac{D_o - D_i}{2}$$

Step Six: Establish the gasket dimensions. (For some gaskets this step may not be necessary; dimensions may have been established already for manufactured seals such as Naflex, Raco, etc.) For a soft, flat gasket, minimum width

$$w = \frac{p_i (R_{bc} - d_b + R_i) N_g}{2(F_{ug} - 2p_s)}$$

and choose outside diameter (D_{og}) and inside diameter (D_{ig}) to suit.

Step Seven: Calculate total initial gasket load. Again, for a soft, flat gasket, and no expected external loads

$$P_g = \pi p_i \left(\frac{R_{og} + R_{ig}}{2}\right)^2 + 2\pi p_s w \left(\frac{R_{og} + R_{ig}}{2}\right)$$

Step Eight: Calculate number of bolts required. If practical, round up to a number divisible by four.

$$n = \frac{P_g N_b}{P_{yb}}$$

and bolt spacing:

$$e = \frac{\pi D_{bc}}{n}$$

Step Nine: (First Design Check) At this point, the design should be reviewed from a practical standpoint. Optimum bolt spacing should fall within the range of

$$3 < \frac{e}{d_b} < 8$$

If this condition is not satisfied, start again at Step Three with another bolt size, bolt material, flanges, material, or both.

Step Ten: Determine flange height (h)

$$h^2 = \frac{3\ (P\ .\ a)\ N_f}{\pi F_{yf}\ (b_f - d_{bh})}$$

Step Eleven: (Second Design Check) Height to bolt spacing:

$$\frac{3h}{e} > 1$$

If this condition is not satisfied, use more bolts--or possibly, smaller diameter bolts.

Testing of the "low profile" flanges

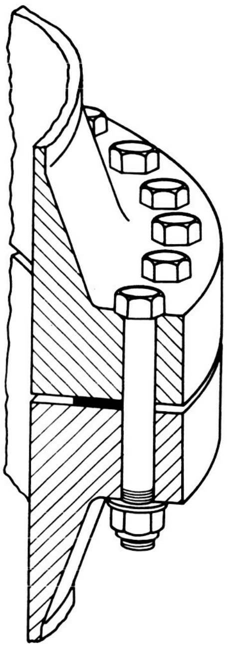
These tests were performed to verify the integrity of the "low profile" flange design concept by direct comparison with the generally accepted standard flange design.

Test Fixtures

Two different test fixtures were used for the performance of this study.

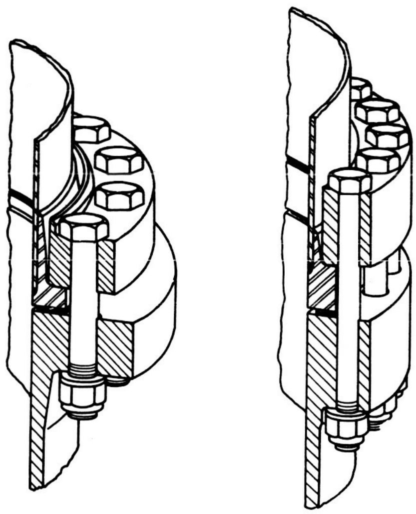
Test fixture #1, which is the Saturn I center LOX tank manhole and manifold test setup, was modified by replacing three out of six conventional flanges with "low profile" flanges (Fig. 6, 7, and 8).

Test Fixture #2 (Fig. 9) was designed and manufactured to get comparison data on the Saturn I, 8-inch LOX suction line configuration.



(a) SA-5 Configuration (b) Low Profile Configuration

Fig. 6 - LOX center tank aft manhole to cover joint



(a) SA-5 Configuration (b) Low Profile Configuration

Fig. 7 - LOX center tank aft manifold to interconnect joint

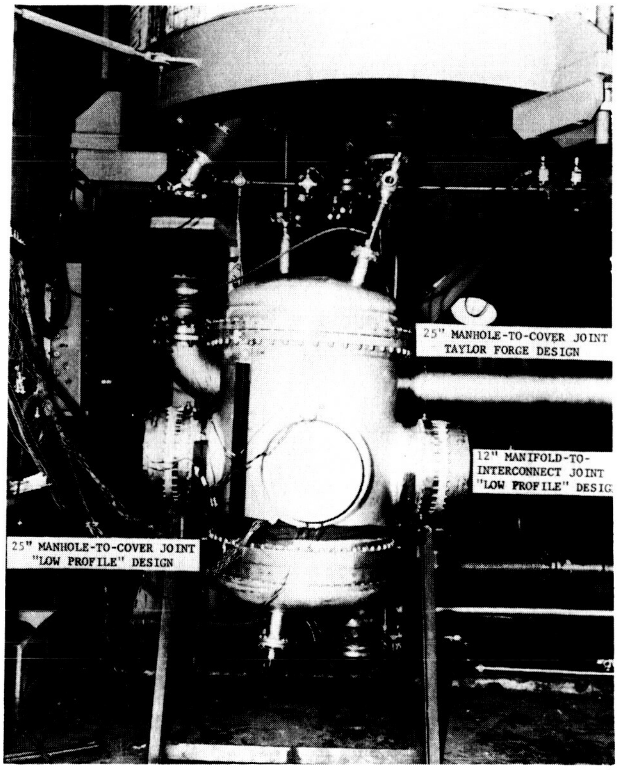


Fig. 8 - Test fixture used in comparison testing of flanged joints

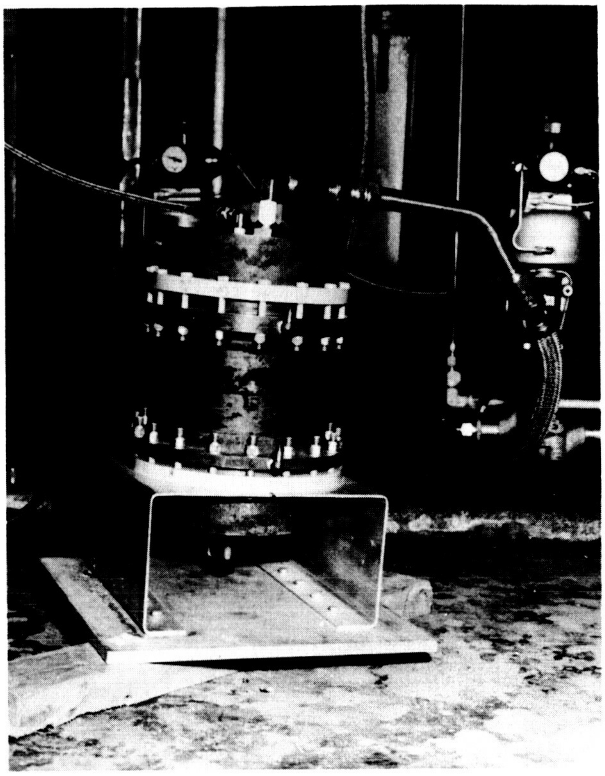


Fig. 9 - Test fixture No. 2

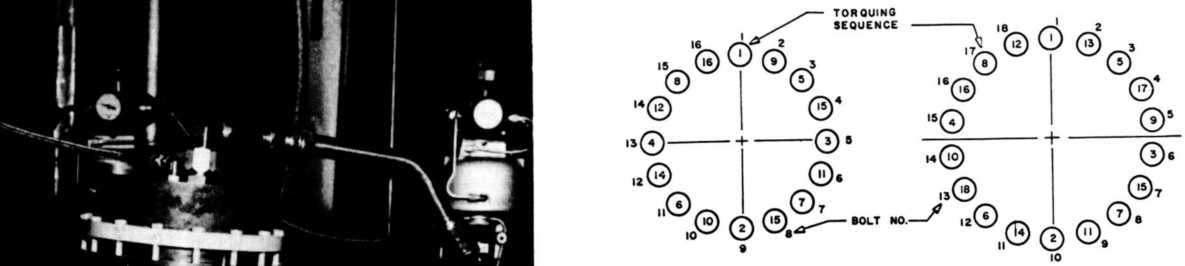
Gasket

Allpax 500 gasket material, Fluorolube treated per specification MS-750. 0B has been used throughout the testing except on the "low profile" interconnect joints on the last two tests where a new gasket material, Teflon FEP - a glass cloth laminated gasket developed by Narmco under the supervision of NASA - MSFC, was used.

Preparation of Test Fixtures

1. All flange surfaces were inspected for finish and flatness by quality control personnel.
2. Flange mating surfaces were cleaned with trichloroethylene.
3. All sliding surfaces between bolts, nuts, washers, and flanges were lubricated with Molylube spray dry film lubricant in accordance with procedure MSFC-PROC-226.
4. Before gasket installation, gasket thickness was measured at several locations. The gaskets were marked for orientation to the flange.
5. Strain gages were fitted to two bolts in each of the manifold flanges to determine the stress levels at which the bolts were being utilized. Unfortunately, readings from the strain gages under low temperature were erratic and therefore discounted.

Table I. Torquing Sequence and Schedule



16 HOLE PATTERN

18 HOLE PATTERN

JOINT	TORQUE SCHEDULE						
	FIRST INCREMENT (IN. LBS)	TIME LAPSE (HRS)	SECOND INCREMENT (IN. LBS)	TIME LAPSE (HRS)	FINAL TORQUE	TIME LAPSE (HRS)	TORQUE CHECK
SAT I	40	8	70	16	100	48	100
LOW PROFILE	35	8	60	16	85	48	85

Test Procedure

The flange connections were assembled using the established bolt torquing sequence shown in Table I which was developed at MSFC. Each test fixture was hydrostatically tested once for general reliability. After hydrostatic testing, each test fixture was carefully inspected for any signs of damage. A new gasket was installed and reassembled as before.

Liquid nitrogen was used as the test medium in all tests. Gaseous nitrogen was used to pressurize the test fixtures. While testing, periodic checks were made for evidence of leakage and abnormalities.

After the fixture was held the requested time under the maximum pressure, the test fixtures were depressurized and pressurized once more before final depressurization. After residual LN_2 draining and disassembling all parts, the flanges and gaskets were subjected to a thorough inspection.

Results

Table II lists the parameters of the two flange configurations of test fixture #1. A weight saving of 37% is shown for 25-inch diameter flange and 8% for the 12 inch.

The recorded test data for the flanges on test fixture #1 are given in Tables III and IV. The recorded test results show that the low profile flange is at least as effective as the conventional flange.

Table V lists the parameters of the flange configuration of the test fixture #2; a weight saving of 15% is shown for the 8-inch diameter flange.

TABLE II. TECHNICAL DATA ON TEST FIXTURE NO. 1.

		SATURN I BLOCK II DESIGN	LOW PROFILE DESIGN
	MAX. OPERATING PRESSURE (PSIG)	94	94
	TEST PRESSURE (PSIG)	113	113
	PROOF PRESSURE (PSIG)	125	125
MANHOLE JOINTS *	GASKET AREA — SQ IN.	52.769	29.965
	GASKET THICKNESS	.125	.125
	GASKET O.D.	27.500	25.810
	GASKET I.D.	25.000	25.060
	FLANGE THICKNESS	[illegible]	2.125
	FLANGE O.D.	29.250	27.000
	FLANGE I.D.	25.000	25.000
	BOLT (ALUMINUM)	AN7DD31A	AN6DD46A
	BOLT CIRCLE DIAMETER (B.C.)	28.000	26.250
	WASHER (ALUMINUM)	AN960PD716	AN960PD616
	NUT (SELF LOCKING)	MS20365-720A	MS20365-624A
	PITCH	1.955	1.586
	NO. OF BOLTS	45	52
	TOTAL WEIGHT OF JOINT (LBS)	64.58	40.55 (37% LESS)
INTERCONNECT JOINTS	GASKET AREA — SQ IN.	14.932	9.857
	GASKET THICKNESS	.062	.062
	GASKET O.D.	13.050	12.800
	GASKET I.D.	12.300	12.300
	FLANGE THICKNESS	.812	1.300
	FLANGE O.D.	15.000	13.812
	FLANGE I.D.	12.000	12.000
	BOLT (ALUMINUM)	MS35314-68	AN5DD33A
	BOLT CIRCLE DIAMETER (B.C.)	13.562	13.188
	WASHER (ALUMINUM)	AN960PD616	AN960PD516
	NUT (SELF LOCKING)	MS20365-624A	MS20365-524A
	PITCH	1.775	1.295
	NO. OF BOLTS	24	32
	TOTAL WEIGHT OF JOINT (LBS)	13.81	12.71 (8% LESS)

*UNLESS OTHERWISE SPECIFIED ALL DIMENSIONS IN INCHES

The recorded test data for the flanges on test fixture #2 are given in Table VI.

CONCLUSION

Test data shown on tables 1, 2, and 4 prove that the low profile flange design is superior to the Taylor Forge concept in the application of the test hardware in the following ways:

a. tighter seal
b. lighter weight, up to 37 percent lighter
c. more compact, up to 12 percent reduction in diameter

APPENDIX

SAMPLE CALCULATIONS

The following calculations are for the "low profile" versions of the 25-inch diameter aft LOX manhole-to-cover joint (Fig. 6), and an 8-inch diameter LOX suction line joint, being tested together with their Saturn counterparts, designed according to Taylor Force methods.

TABLE III. TEST DATA FOR 25-INCH CENTER LOX TANK MANHOLE COVER - TEST FIXTURE NO. 1.

TEST DATA

25 INCH I. D. MANHOLE JOINTS

TEST NO. & DATE	*FLANGE CONFIG	GASKET				NOM BOLT DIA	TORQUE (IN. LBS)	PRESSURE (PSIG)	RESULTS	REMARKS
		MATL	THICK	WIDTH	NEW					
1 3/25/64	B D	ALLPAX ALLPAX	.125 .062	.625 .375	YES YES	7/16 3/8	270± 10 130± 5	113 113	NO LEAKS NO LEAKS	
2 3/26/64	B D	ALLPAX ALLPAX	.125 .062	.625 .375	NO NO	7/16 3/8	270± 10 130± 5	113 113	NO LEAKS NO LEAKS	TORQUE RECHECKED PRIOR TO TEST
3 4/8/64	B D	ALLPAX ALLPAX	.125 .062	.625 .375	YES YES	7/16 3/8	270 ±10 130 ±5	113 113	NO LEAKS NO LEAKS	
4 4/8/64	B D	ALLPAX ALLPAX	.125 .062	.625 .375	NO NO	7/16 3/8	270 ±10 130 ±5	113 113	NO LEAKS NO LEAKS	NO TORQUE RECHECK PRIOR TO TEST
5 6/8/64	B D	ALLPAX ALLPAX	.125 .125	.625 .375	YES YES	7/16 3/8	270 ±10 130 ±5	113 113	NO LEAKS NO LEAKS	
6 6/8/64	B D	ALLPAX ALLPAX	.125 .125	.625 .375	NO NO	7/16 3/8	270 ±10 130 ±5	120 120	NO LEAKS 2 LN_2 LEAKS	AT 120 PSIG (IN AREA OF SLUGS)
7 6/24/64	B D	ALLPAX ALLPAX	.125 .125	.625 .375	YES YES	7/16 3/8	270 ±10 145 ±5	113 113	2 LN_2 LEAKS NO LEAKS	ICE BETWEEN FLANGES WHERE LEAKS OCCURRED
8 6/25/64	B D	ALLPAX ALLPAX	.125 .125	.625 .375	NO NO	7/16 3/8	270 ±10 145 ±5	145 145	NO LEAKS NO LEAKS	TORQUE RECHECKED PRIOR TO FILL

* FLANGE CONFIGURATION
B= SATURN I, BLOCK II, DESIGN
D= LOW PROFILE DESIGN

TABLE IV. TEST DATA FOR 12-INCH INTERCONNECT JOINTS ON TEST FIXTURE NO. 1.

TEST DATA
12 INCH I. D. INTERCONNECT JOINTS

TEST NO. & DATE	* FLANGE CONFIG	GASKET				NOM BOLT DIA	TORQUE (IN. LBS)	PRESSURE (PSIG)	RESULTS	REMARKS
		MATL	THICK	WIDTH	NEW					
1	A	ALLPAX	.062	.375	YES	3/8	135 ± 5	113	NO LEAKS	
3/25/64	C	ALLPAX	.062	.250	YES	5/16	90 ± 5	113	NO LEAKS	
2	A	ALLPAX	.062	.375	NO	3/8	135 ± 5	113	NO LEAKS	TORQUE RECHECKED
4/26/64	C	ALLPAX	.062	.250	NO	5/16	90 ± 5	113	NO LEAKS	PRIOR TO TEST
3	A	ALLPAX	.062	.375	YES	3/8	135 ± 5	113	NO LEAKS	
4/8/64	C	ALLPAX	.062	.250	YES	5/16	90 ± 5	113	NO LEAKS	
4	A	ALLPAX	.062	.375	NO	3/8	135 ± 5	113	NO LEAKS	NO TORQUE RECHECK
4/8/64	C	ALLPAX	.062	.250	NO	5/16	90 ± 5	113	NO LEAKS	PRIOR TO TEST
5	A	ALLPAX	.062	.375	YES	3/8	135 ± 5	113	NO LEAKS	
6/8/64	C	ALLPAX	.062	.250	YES	5/16	90 ± 5	113	NO LEAKS	
6	A	ALLPAX	.062	.375	NO	3/8	135 ± 5	120	NO LEAKS	NO TORQUE RECHECK
6/8/64	C	ALLPAX	.062	.250	NO	5/16	90 ± 5	120	NO LEAKS	PRIOR TO TEST
7	A	ALLPAX	.062	.375	YES	3/8	135 ± 5	113	NO LEAKS	
6/24/64	C	NARMCO	.062	.250	YES	5/16	90 ± 5	113	NO LEAKS	
8	A	ALLPAX	.062	.375	NO	3/8	135 ± 5	145	NO LEAKS	TORQUE RECHECKED
6/25/64	C	NARMCO	.062	.250	NO	5/16	90 ± 5	145	NO LEAKS	PRIOR TO FILL

* FLANGE CONFIGURATION
A = SATURN I, BLOCK II DESIGN
C = LOW PROFILE DESIGN

TABLE V. TECHNICAL DATA ON TEST FIXTURE NO. 2.

	SATURN I BLOCK II DESIGN	LOW PROFILE DESIGN
*** FLANGES**		
1. INTEGRAL		
MATERIAL	CRES	CRES
O. D.	10.250	9.520
I. D.	7.890	7.875
THICKNESS	.500	.766
2. FERRULE		
MATERIAL	CRES	CRES
O. D.	9.042	8.520
I. D.	7.875	7.875
THICKNESS	.300	.570
3. FLOATING RING		
MATERIAL	ALUMINUM 2024	ALUMINUM 2024
O. D.	10.252	9.520
I. D.	8.250	8.220
THICKNESS	.625	.700
GASKET		
O. D.	9.000	8.520
I. D.	8.125	7.915
THICKNESS	.062	.062
AREA (SQ IN.)	10.93	7.01
FASTENERS		
BOLT CIRCLE DIAMETER (B.C.)	9.410	8.890
NO. REQD & SPACING	18 AT 1.642	16 AT 1.746
BOLT (ALUMINUM)	MS35314-67	MS35314-42
NOMINAL DIAMETER	3750	.3125
NUT (STEEL)	MS20365-624A	MS20365-524A
WASHER (ALUMINUM)	AN960PD616	AN960PD516
WEIGHT		
TOTAL WEIGHT OF JOINT (LBS)	10.864	9.237 $\boxed{\text{15\% LESS}}$

* UNLESS OTHERWISE SPECIFIED
ALL DIMENSIONS IN INCHES

MANHOLE-TO-COVER JOINT CALCULATIONS

Design Data:

Design pressure: 113 pounds per square inch gage (psig)

Design temperature: minus 297 degree Fahrenheit

Flange material: 5456-T651 Aluminum alloy F_{ty} = 30,000 psi

Gasket material: Allpax "500"

Assumed crushing stress - F_{ug} = 10,000 psi

Effective seating stress - p_s = 1,850 psi

Bolt material: 2024-T4 Aluminum alloy - F_{ty} = 40,000 psi (ANxxDDxxA)

Tube I. D. = 25 inch

d Tube wall = 0.188 inch

Design Calculations:

Tube wall (t) from design data = 0.188 inch

Bolt size: Assume approximately 1 1/2-inch spacing for bolts @ 25.00 inch diameter to estimate approximate number of bolts (n) required, then roughly estimate expected bolt load (P_b)

$$n = \frac{\pi(D_i)}{1.5} = \frac{\pi(25)}{1.5} = 52.36 \text{ Try 52 bolts}$$

TABLE VI. TEST DATA FOR 8-INCH LOX SUCTION LINE JOINT - TEST FIXTURE NO. 2.

TEST DATA 8 INCH I. D. LOX SUCTION LINE JOINT										
TEST NO. & DATE	*FLANGE CONFIG	GASKET				NOM BOLT DIA	TORQUE (IN. LBS)	PRESSURE (PSIG)	RESULTS	REMARKS
		MATL	THICK	WIDTH	NEW					
1 4/16/64	A B	ALLPAX ALLPAX	.062 .062	.375 .250	YES YES	3/8 5/16	100±5 85±5	200 200	NO LEAKS NO LEAKS	
2 4/16/64	A B	ALLPAX ALLPAX	.062 .062	.375 .250	NO NO	3/8 5/16	100±5 85±5	180 180	NO LEAKS NO LEAKS	
3 5/1/64	A B	ALLPAX ALLPAX	.062 .062	.375 .250	YES YES	3/8 5/16	100±5 85±5	200 200	NO LEAKS NO LEAKS	
4 5/1/64	A B	ALLPAX ALLPAX	.062 .062	.375 .250	NO NO	3/8 5/16	100±5 85±5	200 200	NO LEAKS NO LEAKS	
5 5/6/64	A B	ALLPAX ALLPAX	.062 .062	.375 .250	NO NO	3/8 5/16	100±5 85±5	200 200	NO LEAKS NO LEAKS	
6 5/22/64	A B	ALLPAX ALLPAX	.062 .062	.375 .250	YES YES	3/8 5/16	100±5 85±5	215 215	LN_2 LEAK NO LEAKS	
7 5/22/64	A B	ALLPAX ALLPAX	.062 .062	.375 .250	NO NO	3/8 5/16	100±5 85±5	260 260	LN_2 LEAK NO LEAKS	AT 180 PSIG
*FLANGE CONFIGURATION A= SATURN I, BLOCK II, DESIGN B= LOW PROFILE DESIGN										

Approximate load per bolt, using 3/8-inch wide gaskets

$$P_b = \frac{P_i \cdot \pi \left(\frac{D_i}{2}\right)^2 + p_s\,(\pi)\left(D_i + 3/8\right)\left(\frac{3}{8}\right)}{52}$$

$$= \frac{113(\pi)\,(12.5)^2 + 1,850(\pi)\,(25.375)\,(0.375)}{52}$$

$$= \frac{55,470 + 55,300}{52}$$

$$= 2,130 \text{ pounds}$$

try 3/8 - 24 Aluminum bolt, AN6DDxxA,
P_{yb} = 3,240 pounds

Aluminum bolts are necessary because the different thermal coefficient of expansion of steel bolts will cause the flanges to shrink away from the bolts and become loose at cryogenic temperatures.

Bolt circle diameter (D_{bc})

$$D_{bc} = D_i + 2t + 2C + d_{sf}$$

$$= 25 + 2(0.188) + 2(0.062) + 0.750$$

$$= 26,250 \text{ inches}$$

Flange O. D. (D_o)

Using edge distance = d_b

then:

$$D_o = D_{bc} + 2\,d_b$$

$$= 26.250 + 2(0.375)$$

$$= 27,000 \text{ inches}$$

and:

$$b_f = \frac{D_o - D_i}{2}$$

$$= \frac{27.00 - 25.00}{2}$$

$$= 1.00 \text{ inch}$$

Gasket dimensions

$$b_g = \frac{p_i\,(R_{bc} - d_b + R_i)Ng}{2\,(F_{ug} - 2p_s}$$

$$= \frac{113(13.125 - 0.375 + 12.5)(1.5)}{2\ [10,000 - 2(1,850)]}$$

$= 0.340$ inch (Use 0.375 inch)

Setting gasket just inside the bolt circle

$D_{od} = D_{bc} - d_{bh}$

$= 26.25 - 0.406$

$= 25.844$ inches (Use 25.812 inches)

then:

$D_{ig} = D_{og} - 2b_g$

$= 25.812 - 2(0.375)$

$= 25.062$ inches

Loads

$$P_i = \pi p_i \left(\frac{D_{og} + D_{ig}}{4}\right)^2$$

$$= \pi(113)\left(\frac{25.812 + 25.062}{4}\right)^2$$

$= 57,425$ pounds

$$P_s = \pi p_s \left(\frac{D_{og} + D_{ig}}{2}\right)(b_g)$$

$$= (\pi)(1,850)\left(\frac{25.812 + 25.062}{2}\right)(.375)$$

$= 55,440$ pounds

$P_g = P_i + P_s$

$= 112\ 865$ pounds

Number of bolts (n)

$$n = \frac{P_g N_b}{P_{yb}}$$

$$= \frac{112,865(1.5)}{3,240}$$

$= 52.25$ (Use 52 bolts)

Bolt spacing (e)

$$e = \frac{\pi D_{bc}}{n}$$

$$= \frac{\pi(26.250)}{52}$$

$= 1.586$ inches

First design check

$$3 < \frac{e}{d_b} < 8$$

$$\frac{e}{d_b} = \frac{1.586}{0.375}$$

$= 4.23$ OK

Flange height (h)

$$h^2 = \frac{3(P_i a_i + P_s a_s)N_f}{\pi F_{yf}(b_f - d_{bh})}$$

$$= \frac{3[(57,425)(0.531) + (55,440)(0.406)]\ 1.25}{\pi(26,000)(1 - 0.406)}$$

$h = \sqrt{4.096445}$

$= 2.024$ inches (Use 2.125 inches)

Second design check (Height to bolt spacing)

$$\frac{3h}{e} > 1$$

$$\frac{3(2.125)}{1.586} > 1$$ OK

SYMBOLS USED:

a: moment arm between bolts and gasket load

a_i: the moment arm between tube wall and bolt

b_f: Flange width

C: minimum practical clearance between spotface and tube wall

d: thread root diameter of bolts (inches)

d_b: bolt diameter

d_{bh}: bolt hole diameter

D_{bc}: bolt circle diameter

e: bolt spacing (inches)

F_{yf}: yield stress of flange material (psi)

F_{yb}: yield stress of bolt material (psi)

F_i: axial internal pressure load

F_{ug}: ultimate or crushing strength of gasket material

F_D: gasket load

F_b: external bolt load

F_N: shear load in flange neck

h: flange height (inches)

M_t: internal tangential moment in the flange

M_s: bolt moment

M_N: internal moment in flange neck

n: number of bolts

N_f: safety factor applied to flange material

N_b: safety factor for bolt material

N_g: safety factor for gasket material

P_G: pressure on the gasket (unpressurized)

P_{yb}: yield load for bolt

P_G: operating pressure on gasket

P_g: initial gasket load

P_s: effective seating stress

P_b: bolt load

P_i: internal pressure

R_{bc}: radius of the bolt circle (inches)

R_i: inside radius of the tube (inches)

R_{og}: outside radius of gasket

R_{ig}: inside radius of gasket

t: wall thickness of the tube (inches)

w: gasket width

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