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SUBJECT: Design and Analysis of a Mechanical Transition
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FROM: R. D. Cheverton and T. H. Mauney

ABSTRACT

This report discusses the design and analysis of a mechanical transition joint between materials having different thermal coefficients of expansion. The analytical treatment is based primarily on the analysis of short cylinders.

The specific design for the Zircaloy-2-stainless steel core vessel joint in the HRT is discussed in detail. Brief summaries of preliminary designs and test results are included along with results obtained from the tests performed on the final design for the HRT.

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DESIGN AND ANALYSIS OF A MECHANICAL TRANSITION JOINT BETWEEN DISSIMILAR MATERIALS - APPLICATION TO HRT

Introduction

In the design of nuclear reactor systems the choice of materials may depend upon many factors not generally considered in non-nuclear systems. Specifically, the HRT required a core vessel material having a small absorption cross section for thermal neutrons. Of course, other properties such as corrosion resistance and strength had to be considered. The material that appeared to be best suited to the particular application was an alloy of zirconium, Zircaloy-2, while the majority of the rest of the circulating system was to be constructed with an austenitic stainless steel, type 347. A simplified schematic of the system is shown in Fig. 1.

The HRT was designed to be a two-region reactor with the core vessel separating the fuel solution in the core from the D_2O reflector surrounding the core and contained within the outer pressure vessel. Thus, it was necessary to have a junction between the zirconium core vessel and the austenitic stainless steel portion of the system. Leakage across the junction was not considered to be serious from a safety standpoint; however, excessive leakage could encumber operation of the reactor. Thus, the allowable leak rate was specified as 1 cm³ of liquid/day. This was interpreted to mean that at no time during steady state or transient operating conditions should that rate be exceeded.

The large difference in coefficient of thermal expansion for Zircaloy-2 and type 347 stainless steel imposed an important requirement on the joint: stresses of a static or fatigue nature sufficient to cause excessive leakage should not develop as a result of thermal cycling between room temperature and 600°F. The minimum number of cycles for testing purposes was specified to be 500 at a minimum heating and cooling rate of 100°F/hr.

Fig. 1. HRT CORE AND PRESSURE VESSEL

There are, of course, two types of joints that can be considered for joining the two materials: a metallic bond such as welding and brazing, and a mechanical flange-type joint. At the time the problem was being considered for the HRT, there were no metallic bond methods considered satisfactory for the particular application. A mechanical joint was the only alternative.

Joint Design for HRT

A mechanical joint design had to incorporate means for compensating for the radial and longitudinal differences in coefficients of thermal expansion. Many basically different designs* were considered, several of which were tested, before a satisfactory design was conceived.

Results from the above tests indicated that a leak-tight joint could not be sustained during thermal cycling, if the mating surfaces of the gasket and one flange were allowed to move relative to each other. The movement, resulting from the differences in radial thermal expansion, was scarifying the sealing surfaces in the direction the leakage would occur. It appeared that the solution to the problem was to prevent the relative movement at the gasket-flange interface. However, this required that either the flanges, gasket, or both had to absorb the radial movement.

With the proper combinations of type 347 stainless steel and Zircaloy-2 a slip-on-type flange can be designed that will permit the difference in radial expansion to be taken as radial deflection in the cylindrical pipe wall section of the core vessel. A design considered for the HRT is shown in Fig. 2. There was one main objection in that design that was responsible for its rejection:

* Several of the preliminary designs and tests are discussed briefly in Appendix I.

Fig. 2. First version of transition joint between core vessel and pressure vessel.

the maximum and somewhat excessive stresses occurred in the outlet pipe of the vessel where a complete circumferential failure would permit one end of the vessel to be unrestrained. An alternate design allowed essentially all the radial expansion and therefore the maximum stresses to be taken in the gasket instead of the vessel.

The final design, therefore, consisted of two heavy flanges, one type 347 stainless steel and one Zircaloy-2, with a comparatively radially flexible gasket. The pertinent characteristics of the design, shown in Fig. 3, are as follows: (1) the ends of the gasket are secured in the flanges so that there is essentially no relative motion between the mating surfaces; (2) essentially all of the differences in radial expansion are taken as radial deflection in the gasket; (3) thin gold foils are used at the sealing surfaces to reduce the necessary seating stress required to prevent leakage.

Design Considerations

Only the design of the cylindrical gasket will be considered in detail in this report. However, some other important considerations will be mentioned.

Referring to Fig. 3, it is observed that as the flange unit is heated uniformly the flange bolts will have to bend in a meridional plane. The diameter of the bolt and the lengths of bolt and ferrule must be such as to prevent excessive bolt stresses. Also, the differences in longitudinal expansion must be considered. One might note that the design in Fig. 2 does not require the former design consideration.

The calculation of stresses in the flanges follows conventional methods and, therefore, will not be discussed further.

Three stress conditions in the gasket must be considered: (1) the direct and bending stresses from the standpoint of static and fatigue conditions; (2) stability against longitudinal buckling; and (3) stability against buckling under external pressure.

Fig. 3. Transition joint used in HRT between Zircaloy-2 core vessel and stainless steel clad pressure vessel.

Literature¹ concerning buckling of cylinders under axial compression indicates that for cylinders with ratios of radius to thickness no greater than considered for this particular application the critical stress when buckling begins is about equal to the yield strength of the material. That is, when the compressive membrane stress exceeds the elastic limit, the cylinder will begin to buckle. In the actual application discussed in this report the cylinder is subjected to longitudinal bending stresses as well as direct compressive membrane stresses. In designing the gasket for the HRT it was assumed that the bending stresses had little effect on the buckling stress, primarily because the bending stresses were localized near the ends. Test results from thermal cycling tests of the complete flange assembly showed no indications of buckling even though the calculated maximum total stress was greater than the yield stress for the gasket material. However, by no means should these results be interpreted to mean that the above assumption was correct because the actual magnitude of the stresses in such a test are too uncertain, probably being less than calculated.

As will be explained later it was found desirable to make the cylindrical gasket of several interference fitted laminations. To determine what effect this design had on buckling, two axial compression buckling tests were performed with aluminum cylinders. One test used a single cylinder, while the other used two cylinders, essentially the same size as the other, press fitted together. The experimental proportional limit stresses for the two gaskets were approximately the same and approximately equal to the proportional limit for the material. Therefore, it was concluded that the laminations acted independently insofar as buckling was concerned.

Buckling of the gasket will also be affected to some extent by the axially symmetrical offset loading as shown in Fig. 4. The offset is caused by the radial

deflection of the gasket when the flange temperatures are other than room temperature. No attempt was made to calculate this effect or to design tests specifically

Fig. 4

for investigating it. The influence of bending stresses and off-set loading could not be separated completely in a test of the type that was used to check the overall characteristics of the flange assembly for the HRT.

The stresses in the gasket result from axial loading (gasket load), radial shear loads, and pressure, either internal or external. The off-set effect shown in Fig. 4 results in an additional bending stress which cannot be superimposed on the others because it is dependent on the amount of radial deflection. Since the radial deflections being considered here are small compared to the wall thickness the latter effect is neglected.

One of the important characteristics of the cylindrical gasket is that the ends are not free to slide or rotate relative to the mating flanges. In order to prevent radial movement of the ends a shearing force is required, and to prevent rotation end moments are necessary. Therefore, the gasket is loaded and held in equilibrium by the forces and moments shown in Fig. 5. Of course it is, for all practical purposes, impossible to completely prevent rotation of the ends, the

Fig. 5

degree of success depending a great deal upon how large the end moments must be. With this thought and the desire to hold stresses to a minimum in mind a design was sought that would provide such a result.

If the cylinder is made long enough so that the stresses at one end are essentially independent of the forces and moments at the other, the bending stresses will be a minimum, and the problem will fit that described in reference (2) on pages 395-7. According to that solution the bending stresses, which are a maximum at the point of load application, are essentially damped out in a distance π/β . Therefore, if the cylinder has a length

$$l = 2\pi/\beta \quad (1)$$

the ends will be essentially independent of each other and minimum moments and bending stresses will be obtained for a given value of β . Actually there are two cases to be considered. Suppose that a finite cylinder is loaded on only one end. Then the effectively infinite length would be just π/β . If the cylinder is loaded on both ends, the effectively infinite length would be $2\pi/\beta$. However, since the moments drop off sharply along the cylinder from one end to the other, it might be suspected that the moments at the ends of a cylinder loaded at both ends and of length π/β would not be significantly

different in magnitude from the maximum moment in a cylinder of the same length loaded at only one end. This was the approach used for the HRT transition joint design.

As seen in equation (2) the value of

$$\beta = \left[3(1 - \nu^2) / a^2 h^2 \right]^{1/4} \quad (2)$$

β depends upon the radius and wall thickness of the cylinder. Increasing β , and therefore decreasing a and h decreases the required length of the cylinder. The end moment and shearing force are determined from equations (3) and (4).

$$M_o = \frac{Eh^2 \delta_o}{2a \left[3(1 - \nu^2) \right]^{1/2}} + \frac{p}{2\beta^2} \quad (3)$$

$$Q_o = -Eh \delta_o / a^2 \beta - p / \beta \quad (4)$$

where

M_o = Longitudinal bending moment per unit circumference at end of cylinder

Q_o = Shearing force per unit circumference at end of cylinder

a = Radius of cylinder

h = Wall thickness of cylinder

E = Young's modulus

p = Pressure across cylinder wall

δ_o = Radial deflection at end of cylinder

ν = Poisson's ratio

Assuming the radius to be a fixed value, equations (3) and (4) show that increasing β by decreasing h reduces M_o and Q_o .

There is some minimum value of h required to support the fluid pressure and another less well defined h value required to provide adequate seating characteristics and stability against axial buckling. Suppose that an h value tentatively selected on the bases of the last three considerations gives a gasket length and moments greater than desired. The problem can be eased by using a composite cylinder made up of several concentric cylinders or laminations. The h values in equations (2) and (3) then apply to the individual laminations, and p in equation (3) is the total pressure divided by the number of laminations, provided the laminations fit snugly together. Neglecting the pressure term in equation (3), it is seen that the moment at the end of the laminated gasket is proportional to nh^2 instead of $(h_1 + h_2 + \dots + h_n)^2$, where n is the number of laminations and $h = h_1 = h_2 = \dots = h_n$. Therefore, for the same total thickness the laminated gasket has a smaller end moment.

The above simple analysis is not really exact for laminated cylinders because each lamination, when fitted snugly to its neighbors, interferes with the deflection curves of its neighbors. However, the analysis is considered satisfactory for this particular type of problem.

The total maximum stresses in the gasket are found from equations (5), (6) and (7).

$$\sigma_L = \sigma_g + \frac{6 M_o}{h^2} \quad (5)$$

$$\sigma_c = \frac{ap}{h} - \frac{E \delta_o}{a} - \frac{6\nu M_o}{h^2} \quad (6)$$

$$\tau_o = Q_o/h \quad (7)$$

where σ_L = Total longitudinal stress at end of cylinder
 σ_C = Total circumferential stress at end of cylinder
 τ_o = Shearing stress at end of cylinder
 σ_g = Gasket seating stress

These maximum stresses occur at the ends of the gaskets. It should be remembered, when using the above equations, that the bending stresses (those terms containing M_o) are positive on one surface and negative on the other so that the maximum stresses are obtained by using the sign that gives the largest absolute values. It is interesting to note that the bending stresses are not dependent on h . Since p in equation (3) is the fluid pressure divided by the number of laminations, n , and since $n = (h_1 + h_2 + \dots + h_n)/h$, the pressure term in the bending moment portion of the stress equations is inversely proportional to the total thickness of the gasket.

Final design:

The dimensions selected for the experimental and final HRT gasket on the basis of the preceding analysis were as follows:

Length	= 1.75 in.
Number of laminations	= 4
Wall thickness of each lamination	= 0.0625 in.
Total wall thickness	= 0.250 in.
Outside diameter	= 7 in.
Inside diameter	= 6.5 in.

The 1.75 in. length includes 0.25 in. at each end for securing the ends of the gasket in the flanges.

Titanium was selected for the gasket, bolts and ferrules from the few materials having adequate corrosion resistance because: (1) its coefficient of thermal expansion is between that of type 347 stainless steel and Zircaloy-2, and (2) its modulus of elasticity is low.

Test results:

Several performance tests, which are discussed in greater detail in Appendix III, were conducted on two full scale models of the HRT transition joints. The results showed no excessive leakage during thermal cycling and no indication of buckling of the gasket. Detailed calculations based on the preceding method of analysis have been included for the HRT design in Appendix II.

General Analysis of Short Cylinders

The number of laminations that can be used in order to reduce the length of the effectively infinitely long cylinder is limited by practical considerations in connection with manufacturing, and the longer the cylinder the more complicated the design of the joint becomes. Therefore, it may be desirable to use a cylinder that is shorter than π/β . Of course, a paper study will not necessarily be able to determine whether the joint will be satisfactorily leak tight since only thorough testing will do this. However, as tests are completed for specific designs better analytical estimations can be made, assuming leak tightness to be affected by the magnitude of the end moments and shearing forces.

The moments, forces, and stresses in a short cylinder can be calculated, if the interdependence of stresses and strains on the forces and moments at both ends is considered.

The general differential equation representing the radial deflections of a cylinder such as in Fig. 5 is taken from reference (2) and reproduced here as equation (8).

$$\frac{d^4 w}{dx^4} + 4\beta^4 w = \frac{-p}{D}, \quad (8)$$

where w = Radial deflection: positive toward the axis

x = Distance along the cylinder from one end

p = Pressure: positive internal

D = Flexural rigidity = $\frac{Eh^3}{12(1 - \nu^2)}$

$$\beta^4 = \frac{3(1 - \nu^2)}{a^2 h^2}$$

E = Young's modulus

ν = Poisson's ratio

a = Radius of cylinder

h = Wall thickness of cylinder.

The general solution of the equation can be written as

$$w = \frac{-Pa^2}{Eh} + C_1 \sin \beta x \sinh \beta x + C_2 \sin \beta x \cosh \beta x + C_3 \cos \beta x \sinh \beta x + C_4 \cos \beta x \cosh \beta x, \quad (9)$$

where C_1, C_2, C_3 , and C_4 are arbitrary constants to be determined from the end conditions. Again from reference (2) the moments and shearing forces are

$$M_x = -D \frac{d^2 w}{dx^2} \quad (10)$$

$$M_\theta = \nu M_x \quad (11)$$

$$Q_x = \frac{dM_x}{dx} = -D \left(\frac{d^3 w}{dx^3} \right), \quad (12)$$

where

M_x = Longitudinal moment per unit circumference

M_ϕ = Circumferential moment per unit circumference

Q_x = Shearing force per unit circumference.

Proceeding with the solution,

$$\begin{aligned} \frac{dw}{dx} = \beta \left[C_1 (\sin \beta x \cosh \beta x + \cos \beta x \sinh \beta x) \right. \\ + C_2 (\sin \beta x \sinh \beta x + \cos \beta x \cosh \beta x) \\ + C_3 (\cos \beta x \cosh \beta x - \sin \beta x \sinh \beta x) \\ \left. + C_4 (\cos \beta x \sinh \beta x - \sin \beta x \cosh \beta x) \right] \end{aligned} \quad (13)$$

$$\begin{aligned} \frac{d^2 w}{dx^2} = 2\beta^2 \left[C_1 \cos \beta x \cosh \beta x + C_2 \cos \beta x \sinh \beta x - C_3 \sin \beta x \cosh \beta x \right. \\ \left. - C_4 \sin \beta x \sinh \beta x \right] \end{aligned} \quad (14)$$

$$\begin{aligned} \frac{d^3 w}{dx^3} = 2\beta^3 \left[C_1 (\cos \beta x \sinh \beta x - \sin \beta x \cosh \beta x) \right. \\ + C_2 (\cos \beta x \cosh \beta x - \sin \beta x \sinh \beta x) \\ - C_3 (\sin \beta x \sinh \beta x + \cos \beta x \cosh \beta x) \\ \left. - C_4 (\sin \beta x \cosh \beta x + \cos \beta x \sinh \beta x) \right] \end{aligned} \quad (15)$$

Boundary conditions:

$$x = 0, w = \delta_0, \frac{dw}{dx} = 0$$

$$x = l, w = \delta_l, \frac{dw}{dx} = 0$$

Substituting these into the proper equations yields the four constants:

$$C_1 = \frac{\delta_\ell \left[2SS' \right] - \delta_o \left[(KS')^2 + (SK')^2 \right] + (Pa^2/Eh) \left[2SS' - (KS')^2 - (SK')^2 \right]}{2(SS')^2 - (SK')^2 + (KS')^2} \quad (16)$$

$$C_2 = -C_3 = \frac{1}{2(SS')^2 - (SK')^2 + (KS')^2} \left\{ -\delta_\ell \left[SK' + KS' \right] + \delta_o \left[KK'(SK' + KS') - SS'(KS' - SK') \right] + \frac{Pa^2}{Eh} \left[(KK' - 1)(SK' + KS') - SS'(KS' - SK') \right] \right\} \quad (17)$$

$$C_4 = \delta_o + \frac{Pa^2}{Eh} \quad (18)$$

where $S = \sin \beta l$

$S' = \sinh \beta l$

$K = \cos \beta l$

$K' = \cosh \beta l$

The special case of $l = \infty$ is of interest.

$$\lim_{l \rightarrow \infty} C_1 = -\delta_o - \frac{Pa^2}{Eh}$$

$$\lim_{l \rightarrow \infty} C_2 = \delta_o + \frac{Pa^2}{Eh}$$

At $x = 0$

$$\frac{d^2 w}{dx^2} = 2\beta^2 C_1$$

$$\therefore \lim_{l \rightarrow \infty} \left(\frac{d^2 w}{dx^2} \right)_{x=0} = -2\beta^2 \left(\delta_o + \frac{Pa^2}{Eh} \right),$$

Therefore, for l equal infinity, M_x at $x = 0$ is

$$M_0 = 2D\beta^2 \left(\delta_0 + \frac{Pa^2}{Eh} \right)$$

$$= \frac{Eh^3 \sqrt{3(1-\nu^2)}}{6(1-\nu^2) ah} \left(\delta_0 + \frac{Pa^2}{Eh} \right),$$

but $\frac{Pa}{h} = \sigma_{cp}$, the membrane circumferential pressure stress, which will have approximately the same value for any size gasket. δ_0 , the radial deflection at $x = 0$, is

$$\delta_0 = \Delta T \Delta \alpha a,$$

where ΔT = temperature change

$\Delta \alpha$ = difference in coefficients of thermal expansion between the flange and gasket at $x = 0$.

Making these substitutions,

$$M_0 = \frac{h^2 \left(E \Delta T \Delta \alpha + \sigma_{cp} \right)}{2 \sqrt{3(1-\nu^2)}}, \quad (19)$$

which is the same as equation (3). It is interesting to note that equation (19) is independent of the diameter of the cylinder, and the bending stresses are independent of both the diameter and thickness. These statements are true, of course, only if σ_{cp} is considered to be independent of the individual values of a and h . For given values of p , a , and σ_{cp} , h must be selected accordingly.

When the length of the cylinder is not effectively infinitely long, the above statements are not correct, and the magnitudes of the bending moments on the ends will rise sharply with decreasing length.

Characteristic curves obtained from the short cylinder equations, which have been programed on the IBM 704 computer, are shown in Figs. 6 through 15. Only a sufficient number of curves for indicating the trends have been included in this report. As shown in Fig. 6 the maximum moment is at one end of the cylinder. It has not been shown in this study by analytical means that this will or will not always be the case. Therefore, an investigation of the moments along the cylinder may be necessary. This was done for all the cases presented in the curves of this report, and the maximum values were always found to occur at the end having the greatest radial deflection. All of the included curves are for that end only.

The curves are essentially self-explanatory. However, the use of such curves for design purposes may be somewhat misleading because, as mentioned previously, the magnitude of the moments, forces, and stresses do not necessarily foretell the degree of leak tightness that will be retained after a number of thermal cycles. Tests so far have been for effectively infinitely long, laminated cylinders subjected to only one type of cycle. If other gaskets are designed accordingly, the same degree of leak tightness will probably be achieved. If the product $\Delta T \alpha$ is increased, the stress and moment values attained with the infinite cylinder will be increased and very likely so will the leakage.

There are two important observations in connection with Figs. 7 through 15: (1) very little is to be gained in reduction of length once the longitudinal stress is increased above the minimum value, if a reasonable stress level is to be maintained; and (2) for a given diameter and total thickness the length can be increased to the point where the minimum stress and moment can be attained with only one lamination.

The following constants were used in the calculation of the curves in Figures 6 through 15:

$$\delta_o = 0.00314 \text{ in.}$$

$$\delta_L = - 0.00831 \text{ in.}$$

$$E = 14 \times 10^6 \text{ psi}$$

$$P = 200 \text{ psi}$$

$$\sigma = 2800 \text{ psi}$$

$$\sigma_g = 0$$

Refer to Appendix IV for discription of nomenclature.

Fig. 6a. Longitudinal moment along length of cylinder.

Fig. 6b. Radial deflection along length of cylinder.

Fig. 7. Maximum longitudinal bending stress vs. length of cylinders with $a = 3.5$ in. and $nh = 0.250$ in.

Fig. 8. Maximum longitudinal bending stress vs. length of cylinders with $a = 5.0$ in. and $nh = 0.3571$ in.

Fig. 9. Maximum longitudinal bending stress vs. length of cylinders with $a = 7.50$ in. and $nh = 0.5357$ in.

Fig. 10. Maximum longitudinal bending stress vs. length of cylinders with $a = 10.0$ in. and $nh = 0.7143$ in.

Fig. 11. Maximum longitudinal bending stress vs. length of various diameter cylinders for $n = 1$.

Fig. 12. Maximum longitudinal bending stress vs. length of various diameter cylinders for $n = 4$.

Fig. 13. Maximum longitudinal bending stress vs. length of various diameter cylinders for $n = 10$.

Fig. 14. Maximum longitudinal bending stress vs. cylinder wall thickness for $a = 3.5$ in. and for several lengths.

Fig. 15. Length vs. cylinder radius for a maximum longitudinal bending stress = 55,000 psi. (constructed using Figs. 7 through 10).

In the design of the gasket for the HRT it was assumed that a cylinder of length π/β and loaded at both ends would have maximum moments not significantly different from those in a similarly loaded cylinder of length $2\pi/\beta$. Using Fig. 7 with $n = 4$ and $\frac{\pi}{\beta} = 1.14$, it is observed that the assumption was satisfactory.

Conclusions

Based on the analysis of infinite and short cylinders and the testing of the HRT type transition joint it is tentatively concluded that a satisfactory design for a laminated, cylindrical gasket as described herein can be attained using a minimum length of π/β , provided of course that the maximum stresses for an infinite length are not excessive. In other words, equations (3) and (4) can be used to estimate the moments and shearing forces that are necessary for calculating the stresses. If a more accurate analysis is desired, the short cylinder treatment can be used. For this the Transition Joint Stress Analysis code can be used. Details of the code are included in Appendix IV.

Certainly it is to be concluded that for designs with higher stresses (relative to the properties of HRT gasket material) than in the HRT tested design extensive tests should be conducted to investigate the mechanical stability characteristics as well as leak tightness.

Appendix I

Preliminary Designs for the HRT

Several different designs were considered for compensating for the differences in radial deflection without allowing excessive leakage and stresses. Threaded joints, shrink fits, dished washer type gaskets, and others got no further than the drafting board. The transition joints that were constructed and tested are discussed briefly in this report since the results of the tests very decidedly pointed out what basic design considerations should be incorporated in a satisfactory transition joint.

The first tests were conducted at the Newport News Shipbuilding and Dry Dock Company under the supervision of the authors. The apparatus used for those tests is shown in Fig. 16. Leak detection was accomplished at the beginning and end of a thermal cycle by filling the interior of the flange with water and pressurizing the exterior. At that time satisfactory leak tightness was defined as no evidence of bubbles for about thirty minutes with 50 psi pressure differential. After moving the apparatus to ORNL, several modifications were made, including the addition of a continuously indicating leak detector system.

Thermal cycling for all tests was between 100° and 300°C with a heating rate of about 300°C/hr and a somewhat slower cooling rate. Particular care was taken in the adjustment of the heaters and cooling system to minimize radial and axial thermal gradients.

Spirally wound (Flexitallic) gasket:

A spirally wound gasket similar to that made by Flexitallic is designed primarily for resilience in the axial direction. However, since it is composed of many thin layers (relative to the height and diameter of the gasket) there

Fig. 16. Original Test Apparatus Used for Thermal Cycling Tests on the HRT Transition Joint. (Later Modified)

is less resistance to longitudinal bending and therefore greater flexibility in the radial direction than for a solid gasket of the same over-all dimensions. Thus, it was estimated that the mating surfaces at the interface between the flange and gasket would not move relative to each other.

Generally, a spirally wound gasket has a filler material such as asbestos between the plys. Since asbestos could not be used in the reactor, the first test was run without any filler. The joint was tightened according to the manufacturer's specifications and thermal cycled. The joint was at no time leak tight. However, indications were that slippage was not taking place, but that the layers were digging into the flange faces. To determine what effect a filler would have on leakage an asbestos filled gasket was tested in the same manner. It leaked excessively at the end of about fifty thermal cycles.

While the asbestos filled gasket was being tested, a spirally wound gasket with a gold filler was being manufactured. Although it was certain that the gold would nullify much of the axial resilience of the gasket, the gold filler idea was considered and tested because it was obvious that a non-filled, spirally wound gasket was not satisfactory. By the end of the fiftieth thermal cycle the gold filled gasket was leaking excessively and the bolt load had relaxed to about one tenth of the initial value. A dimensional check after the test showed considerable permanent set in the height of the gasket, thus varifying the earlier presumption.

Exactly how leakage was occurring past the spirally wound gaskets was not determined. However, one common result of all tests is shown in Fig. 17. The inside edge crimped or buckled in one or more places. After checking many asbestos filled, spirally wound gaskets that had been used in steam service aboard ships, it was found that buckling occurred frequently but was not necessarily responsible for excessive leakage.

Figure 17. Spirally-wound Gasket after Thermal Cycling.

It was concluded from these tests that the spirally wound type of gasket appeared to provide the desired characteristic of non-slippage between sealing surfaces, but for reasons undetermined satisfactory sealing during thermal cycling could not be maintained for fifty cycles. In the case of the gold filled gasket the ninety percent reduction in gasket and bolt load must have contributed significantly to leakage.

Solid, flat-faced gasket:

At that time no other type of gasket was known that would provide sufficient radial flexibility for preventing relative movement of the sealing surfaces. Therefore, an attempt was made to allow radial slippage without excessive damage to the sealing surfaces. The gasket considered was a flat type with a square or rectangular cross section. The mating surfaces of the flanges and gasket were lapped to a flatness of one or two light fringes (one fringe = 11.6×10^{-6} in.). The bolts were tightened initially so that approximately the yield stress of the gasket material was developed. When this did not prevent leakage, higher loads were used. Thermal cycling and leak detection was the same as described for the spirally wound gaskets until the testing was discontinued at the Newport News Shipbuilding and Dry Dock Company and resumed at ORNL. The new testing facility provided the same cycling rate, but incorporated a continuously recording, helium leak detector system that could be calibrated at any time during a cycle down to an equivalent leak rate of one cubic centimeter of liquid per day with reasonable accuracy. During thermal cycling, the joint was pressurized with helium to 50 psi. The pressure was on the inside of the gasket.

The spirally wound gaskets were tightened by measuring the compression of the height of the gasket and, therefore, it was not necessary to know the bolt load. In the case of the solid gaskets the bolt load had to be known, and it

was also desirable to know how the load varied with thermal cycling. At this time the idea of the bolt pin extensometer was conceived and put to use. The particular design is shown in Fig. 18.

Several solid gaskets were tested and are listed below.

1. Zircaloy - $1/4"$ x $1/4"$ cross section
2. Zircaloy - $1/4"$ x $1/8"$ cross section
3. Zircaloy - gold plated $1/4"$ x $1/4"$ cross section
4. Type 347 S.S.-gold plated $1/4"$ x $1/4"$ cross section
5. Titanium - $1/4"$ x $1/8"$ cross section

The maximum bolting area for the HRT joint was fixed because fabrication of the vessels was paralleling the development of a suitable gasket for the transition joint. In order to initially seat the $1/4"$ x $1/4"$ non-plated gaskets properly bolt stresses as high as 90,000 psi were often required. During thermal cycling, however, the load dropped off as shown in Fig. 19. Even so, it was decided that the required initial stress was too high, by comparison to a bolt yield stress of about 120,000 psi (17-4 PH), to be reliable. Therefore, the $1/4"$ x $1/8"$ and the $1/4"$ x $1/4"$ gold plated gaskets were considered.

By the end of approximately fifty thermal cycles all of the gaskets leaked excessively. The non-plated gaskets showed considerable resistance to slippage at the mating surfaces, the indications being a very noticeable permanent distortion of the cross section. Some slippage did occur, however, as evidenced by scarifying of the surfaces in the radial direction. The latter condition was probably largely responsible for the excessive leakage. The distortion of the cross section of the $1/4"$ x $1/8"$ gasket was more noticeable than for the wider faced gaskets, indicating less slippage.

APPARATUS FOR DETERMINING STRAIN IN FLANGE STUDS

FIGURE 18

Fig. 19. Stud strain vs. stud temperature during thermal cycling of a type 347 stainless steel-Zircaloy-2 transition joint using a $1/4" \times 1/4"$ solid Zircaloy-2 gasket.

The gold plated gaskets required less bolt load for sealing and showed indications of more slippage without scarifying. However, in addition to small cracks appearing in the gold on the sealing surfaces, the plating was actually peeling off the non-sealing surfaces.

It was concluded from the above test results that there was little chance of a satisfactory gasket being developed in the time allowed, if slippage of the bare mating surfaces was permitted. Since the gold plated gasket showed some promise it was concluded that a solid gold gasket might be suitable.

Gold foil:

The ability of gold to flow at low stress levels reduces the magnitudes of the bolt load necessary for sealing, and also permits greater radial movement of the gasket with less shearing force at the interfaces.

Two gold foil gaskets were tested. The first was 0.0015 in. thick, and the other was 0.014 in. thick. The first gasket did not extrude but tore all to pieces as shown in Fig. 20. The second gasket extruded radially. At the end of about fifty cycles the thickness between the seating surfaces was only 0.003 in., and still the bolt load had not reached an asymptotic value. As shown in Fig. 21 circumferential tears were forming.

Final conclusions:

The primary purpose of the preliminary tests was to determine what characteristics were required in a gasket for the transition joint. The information obtained was adequate to show that slippage of the mating surfaces must not occur, and that if gold was to be used to lower the seating stress it would have to be confined so that extrusion could not occur.

With these conclusions in mind the laminated cylindrical gasket design was conceived.

Figure 20. Gold Foil Gasket (0.0015 in. thick) after Thermal Cycling.

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-41-

Figure 21. Gold Foil Gasket (0.014 in. thick) after Thermal Cycling.

Appendix II

Calculations for the HRT Transition Joint Gasket

- I. Loads on the joints in addition to the bolt loads that decrease or increase the gasket axial load. Refer to Fig. 22.

Fig. 22

A. Outlet flange

1. Force due to resultant pressure acting on transition piece between s.s. flange and bellows

$$= \frac{\pi (p_i - p_o) (d_g^2 - d_i^2)}{4}$$

2. Force due to differential pressure across bellows

$$= \frac{\pi (p_i - p_o) (d_o + d_i) (d_o - \sqrt{(d_o^2 + d_i^2)/2})}{4}$$

3. Force due to axial expansion or contraction of bellows

$$= \pm 1400 \text{ lbs, (measured load for 5/16 in. compression or extension of bellows)}$$

4. Total force on outlet s.s. flange

$$= F_o = \frac{\pi}{4} (p_i - p_o) (d_g^2 - d_i^2)$$

$$- \frac{\pi}{4} (p_i - p_o) (d_o + d_i) (d_o - \sqrt{(d_o^2 + d_i^2)/2})$$

$$\pm 1400 \text{ lbs}$$

B. Inlet flange

1. Forces on the Zircaloy inlet flange resulting from pressure and bellows movement = F_o

2. Force due to weight of core vessel

$$\approx -380 \text{ lbs}$$

3. Total force on inlet Zircaloy flange

$$= F_o - 380 \text{ lbs}$$

II. Minimum bolt load to prevent separation of the gasket from the flange at room temperature

$$= F_o \text{ when } F_o > 0$$

$$\text{For } d_g = 7 \text{ in.}$$

$$d_i = 5.5 \text{ in.}$$

$$d_o = 8.375 \text{ in.}$$

$$F_o = 0.67 (p_i - p_o) \text{ in.}^2 + 1400 \text{ lbs}$$

$$\text{Assume } p_i - p_o = 200 \text{ psi,}$$

$$\text{then } F_o = 1530 \text{ lbs.}$$

III. Seating area or thickness of gasket

A. Based on required initial bolt load

Let the initial bolt load = 10 x minimum bolt load to prevent separation.

Then, for $d_{go} = 7 \text{ in.}$, $\sigma_g = 20,000 \text{ psi}$, where d_{go} = outside diameter of gasket, d_{gi} = inside diameter of gasket

$$\begin{aligned} d_{gi} &= \sqrt{d_{go}^2 - \frac{40F_o}{\sigma_g}} \\ &= \sqrt{7^2 - \frac{40 \times 1530}{20,000}} \\ &= 6.778 \text{ in.} \end{aligned}$$

$$\therefore t = \frac{d_{go} - d_{gi}}{2} = 0.111 \text{ in.,}$$

where t = thickness of gasket seat.

B. Based on pressure

1. Internal pressure

$$t = \frac{(p_i - p_o) a}{\sigma_{cp}}$$

where σ_{cp} = allowable circumferential membrane pressure stress
= 5,000 psi. (This value must be assumed since it will be the difference between the allowable total stress and the circumferential bending and deflection stresses as indicated in equation (6).)

$$t = \frac{200 \times 3.5}{5000} = 0.14 \text{ in.}$$

2. External pressure

Using $t = 0.14$ in., $\ell = 1.5$ in. and a slightly modified form of Fig. UNF-28.5 of Section VIII of the ASME Pressure Vessel Code, the allowable external pressure was estimated:

$$p_o - p_i = 400 \text{ psi.}$$

The use of laminated construction tends to reduce the allowable external pressure. Assuming that one lamination will be as thin as 0.0625 in., an allowable external pressure per lamination is estimated:

$$p_o - p_i \cong 150 \text{ psi/lamination}$$

Two or three laminations would therefore be sufficient. In the actual design the axial loads and bending stresses will decrease the critical external pressure. However, since the out-of-roundness of the gasket will be essentially zero by comparison to that

incorporated in the code calculations, the code allowable external pressure can be multiplied by a factor of about three. Considering all these conditions a total gasket thickness of 0.25 in. was tentatively selected.

C. Based on axial stability

According to reference (1) only a very thin shell will buckle elastically when subjected to an axial compressive force. Assuming that the short cylinders considered for the gasket would begin to buckle when the material began to yield in compression, the following equation from reference (1) was used:

$$\sigma_{ult} = \frac{E \left(0.6 \frac{h}{a} - 10^{-7} \frac{a}{h} \right)}{1 + 0.004 \frac{E}{\sigma_{y.p.}}} \quad (20)$$

Using equation (20) and the material properties listed in Table 1, the ultimate compressive stresses at room temperature and operating temperature were determined to be

$$\sigma_{ult} (70^{\circ}\text{F}) = 74,000 \text{ psi}$$

$$\sigma_{ult} (600^{\circ}\text{F}) = 45,000 \text{ psi.}$$

Table 1

Properties of Ti used for gasket

	70°F	600°F
$\sigma_{y.p.}$	50,000 psi	25,000 psi
E	15.5×10^6 psi	12.5×10^6 psi

The characteristic shape of the load vs. axial deflection curve was obtained from axial buckling tests on two aluminum cylinders. The cylinders were 1.5 in. long and approximately 7 in. in diameter with 0.0625 in wall thickness. A typical curve is shown in Fig. 23.

Fig. 23. Load vs. deflection for a short cylinder subjected to an axial compressive load.

Since the greatest portion of the gasket load was the bolt load, Fig. 23 indicated that complete buckling would not occur during thermal cycling. The increasing amount of axial deflection, as buckling progressed, would relieve the gasket load to some value less than that required to completely collapse the gasket. However, there was some question as to whether sufficient seating stress for adequate leak tightness would be left after cycling.

IV. Length of gasket

The length of the gasket was computed, using equations (1) through (7). Equations (1) through (4) apply to an infinite cylinder loaded as shown in Fig. 24 and were assumed to apply to a finite cylinder of length ℓ as shown in Fig. 25.

Fig. 24

Fig. 25

Assuming first that $h = 0.25$ in.,

$$\beta = \left[\frac{3(1 - .3^2)}{(3.5)^2 (0.25)^2} \right]^{1/4} = 1.37 \text{ in.}^{-1}$$

$$\ell = \frac{2\pi}{\beta} = \frac{2\pi}{1.37} \text{ in.} = 4.6 \text{ in.}$$

As mentioned previously it was suspected that a cylinder length of $\ell/2$ would result in stresses essentially the same as for a length of ℓ . Therefore, the actual gasket length for a 0.25 in. wall thickness would be

$$\frac{\ell}{2} = 2.3 \text{ in.}$$

This value was considered to be too long for a practical flange design. In order to retain the same total thickness a laminated design was considered. The thinnest cylinder wall considered practical for fabrication was about 0.0625 in. Using this value for h ,

$$\frac{\ell}{2} = 2.3 \sqrt{\frac{4}{16}} = 1.15 \text{ in.}$$

This length was considered satisfactory. Therefore, the gasket was to consist of $n = 4$ laminations 0.0625 in. thick.

The maximum total moment and shearing force were calculated from equations (3) and (4).

$$M_o = n \left\{ \frac{Eh^2 \delta_o}{2a [3(1-\nu^2)]^{1/2}} + \frac{p}{2\beta^2} \right\},$$

where $\delta_o = \Delta T \Delta \alpha a.$

α for 347 s.s. = $9.8 \times 10^{-6}/^{\circ}\text{F}$

α for titanium = $5.3 \times 10^{-6}/^{\circ}\text{F}$

$\Delta T = 600^{\circ}\text{F} - 72^{\circ}\text{F} = 528^{\circ}\text{F}$

$$\begin{aligned} \therefore M_o &= 4 \left\{ \frac{12.5 \times 10^6 \text{ psi } (0.0625)^2 \text{ in.}^2 \cdot 528^{\circ}\text{F} (9.8 - 5.3) \times 10^{-6}/^{\circ}\text{F} \times 3.5 \text{ in.}}{2 \times 3.5 \text{ in. } [3(1 - .3^2)]^{1/2}} \right. \\ &\quad \left. + \frac{200/4 \text{ psi } 3.5 \text{ in.} \times 0.0625 \text{ in.}}{2 \times 1.65} \right\} = 4 \times 38.5 \frac{\text{lbs in.}}{\text{in.}} \end{aligned}$$

$$\begin{aligned} Q_o &= n \left\{ \frac{Eh \delta_o}{a^2 \beta} + \frac{p}{\beta} \right\} \\ &= 4 \left\{ \frac{12.5 \times 10^6 \text{ psi } 0.0625 \text{ in. } 528^{\circ}\text{F} (9.8 - 5.3) \times 10^{-6}/^{\circ}\text{F} \cdot 3.5 \text{ in.}}{(3.5)^2 \text{ in.}^2 \times 2.74 \text{ in.}^{-1}} \right. \\ &\quad \left. + \frac{200/4 \text{ psi}}{2.74 \text{ in.}^{-1}} \right\} = 4 \times 211 \frac{\text{lbs}}{\text{in.}} \end{aligned}$$

The stresses were determined from equations (5), (6), and (7).

$$\begin{aligned}\sigma_L &= \sigma_g + \frac{6 M_o}{h^2} \\ &= -20,000 \text{ psi} - \frac{6 \times 38.5 \text{ lbs}}{(0.0625)^2 \text{ in.}^2} \\ &= -(+20,000 \text{ psi} + 59,000 \text{ psi}) \\ &= -79,000 \text{ psi}\end{aligned}$$

This value simply indicated of course that the elastic limit of the material would be exceeded. Since the maximum value occurred locally at the ends where the gasket was secured, and since the major portion of the stress was a bending stress, it was speculated that the effect on the stability and static failure of the gasket would not be serious. Also, since the joint was to be designed for only 500 thermal cycles failure by fatigue was considered not very probable. At any rate, tests were conducted as discussed in Appendix III to determine the validity of the assumptions.

The other stresses were

$$\begin{aligned}\sigma_C &= \frac{ap}{h} = \frac{E \delta_o}{a} - \frac{6\nu M_o}{h^2} \\ &= \frac{3.5 \text{ in. } 200 \text{ psi}/4}{0.0625 \text{ in.}} + \frac{12.5 \times 10^6 \text{ psi } 0.0083 \text{ in.}}{3.5 \text{ in.}} \\ &\quad + .3 \times 59,000 \text{ psi} \\ &= 1400 \text{ psi} + 29,600 \text{ psi} + 17,700 \text{ psi} \\ &= 48,700 \text{ psi} \\ \tau_o &= \frac{Q_o}{h} = \frac{211 \text{ lbs/in.}}{0.0625 \text{ in.}} = 3380 \text{ psi}\end{aligned}$$

On the basis of all the calculations and reasoning up to this point, the following gasket was specified for the first test:

Outside diameter = 7 in.

Inside diameter = 6.5 in.

Length = 1.75 in.

Number of laminations, $n = 4$

Thickness of each lamination = 0.0625 in.

The length given above provides 0.25 in. at each end for securing the gasket in grooves in the flanges. Additional circumferential reinforcement was added to the gasket as shown in Fig. 3. It was speculated that because the gasket buckling shape, as determined from the buckling test, was as shown in Fig. 26 an integral circumferential band on the outside surface midway between the ends would increase the ultimate buckling load. No tests were conducted specifically for the purpose of showing just how much, if any, the load would be affected.

Fig. 26. Short cylinder buckling shape

V. Bolt area required

$$A_B = \frac{\sigma_g A_g}{\sigma_B}$$

σ_g = gasket seating stress = 20,000 psi

A_g = gasket seating area

σ_B = allowable bolt stress = 40,000 psi

A_B = bolt area

$$A_B = \frac{20,000 \text{ psi } \pi (7^2 - 6.5^2) \text{ in.}^2}{40,000 \text{ psi } \times 4}$$

$$= 2.65 \text{ in.}^2$$

Ten 3/4-in. diameter bolts with shank areas of 0.280 in.² were selected.

VI. Length of bolt and ferrule based on axial differential expansion.

Considering only elastic deformation in the gasket, bolts and ferrules, the following equation was derived from Fig. 27:

$$L_F = \frac{1}{2} \left[\frac{E_B A_B \Delta}{A_g \times .10 \times \sigma_g} - z - L_g \frac{A_F}{A_g} \right],$$

where $\Delta = \Delta T (\Delta \alpha_1 \Delta Y_1 + \Delta \alpha_2 \Delta Y_2)$,

A_F = Ferrule area

L_F = Ferrule length

$\Delta \alpha_1 = \alpha_{s.s} - \alpha_{Ti} = 4.5 \times 10^{-6}/^{\circ}\text{F}$

$\Delta \alpha_2 = \alpha_{Ti} - \alpha_{zir.} = 1.7 \times 10^{-6}/^{\circ}\text{F}$

$\sigma_g = 20,000 \text{ psi}$

$A_B = A_F = 2.80 \text{ in.}^2$

$\Delta T = 528^{\circ}\text{F}$

Fig. 27

The $10\% \times \sigma_g$ is the desired maximum increase in the gasket stress during a thermal cycle.

$$\begin{aligned} \Delta &= 528^\circ\text{F} (4.5 \times .125 + 1.7 \times .75) \times 10^{-6} \\ &= 970 \times 10^{-6} \text{ in.} \end{aligned}$$

$$\begin{aligned} L_F &= \frac{1}{2} \left[\frac{12.5 \times 10^6 \text{ psi} \cdot 2.80 \text{ in.}^2 \cdot 970 \times 10^{-6} \text{ in.}}{5.3 \text{ in.}^2 \times 2000 \text{ psi}} - 1.125 \text{ in.} - 1.75 \text{ in.} \cdot \frac{2.80}{5.30} \right] \\ &= 1.15 \text{ in.} \end{aligned}$$

$$L_B = L_F + Z = 1.15 + 1.13 = 2.28 \text{ in.}$$

These lengths may be reduced if the decrease in E with increasing temperature is considered.

VII. Length of bolt and ferrule based on radial deflection

As the temperature of the flange is increased the bolt and ferrule must deflect radially as shown in Fig. 28. Figure 28a considers both the bolt and ferrule to be flexible while Fig. 28b considers only the bolt to be flexible.

Fig. 28

It was very likely that the ferrule would actually be in contact with the side of the counter-bore. Therefore, only the deflection of the bolt was considered.

The maximum stress in a bar whose ends are not permitted to rotate in a plane containing the axis is

$$\sigma = \frac{3E\delta d}{L^2}$$

where σ = allowable bolt stress due to bending

$$= 80,000 - 42,000 = 38,000 \text{ psi}$$

d = diameter of bar

$$\therefore L = \left(\frac{3E\delta d}{\sigma} \right)^{1/2}$$

$$= \left(\frac{3 \times 12.5 \times 10^6 \text{ psi} \times 528^\circ\text{F} (9.8 - 3.6) \times 10^{-6}/^\circ\text{F} \frac{5 \text{ in.}}{2} \times 0.625 \text{ in.}}{38,000 \text{ psi}} \right)^{1/2}$$

$$= 2.2 \text{ in.}$$

Therefore, the minimum length of the bolt, considering both radial and axial thermal expansion, is

$$L_B = 2-1/4 \text{ in.}$$

Appendix III

Performance tests conducted on the HRT transition joint.

A. Test No. 1

Equipment and procedure:

Two test facilities were used for testing two full scale models of the HRT transition joint. The first tests were conducted using the apparatus shown in Fig. 16. This apparatus as shown was originally designed for testing the spirally wound and flat gasket but was later modified for installation and testing of the laminated cylindrical gasket. Modifications also included an internal cooling coil and a helium leak detector system. Leak detection was accomplished by picking up the leaking helium in a nitrogen purge gas that flowed through an annulus between the cylindrical gasket and an outer secondary gasket. The nitrogen-helium mixture was monitored by a mass spectrometer-type helium leak detector. Calibration of the system was accomplished by bubbling helium through water into the purge gas annulus, the flow rate of helium being determined by counting the number of bubbles. This system was reasonably accurate down to the equivalent of 1 c.c. of water/day leakage past the laminated gasket, and it would record continuously during thermal cycling so that the variation of leak rate with position in the cycle could be determined. A calibration could be performed at any time during the cycle.

Since improper bolt tightening could result in buckling of the laminated gasket, the titanium bolts were equipped with pins as shown in Fig. 18. In this manner very accurate initial loading was achieved, and variations in loading could be determined as cycling proceeded.

Testing conditions were as follows:

Pressure differential across gasket = 50 psi

Temperature range = 100°C to 300°C

Heating time ~ 40 minutes

Cooling time ~ 60 minutes

Total number of cycles = 212

The two heaters and coolers were adjusted to give minimum temperature gradients in the radial and axial directions.

Results:

At no time during cycling was the leak rate greater than 1 cc of water/day. Actually, all leak rate measurements indicated the leakage to be considerably less than that. At the end of the 212th cycle the helium leak detector sampling probe was used for a final leak check with 50 psi helium pressure inside the flange. No leakage was indicated.

The flange bolts were initially tightened to produce about 25,000 psi gasket seating stress. At the end of 123 thermal cycles the bolt load had been reduced by twenty-four percent. According to other test results, most of that decrease occurred during the first three or four cycles. At the end of 212 cycles there was no further reduction in bolt load. This meant that the asymptotic gasket seating stress was about 18,000 psi, which was considered more than sufficient to assure adequate leak tightness.

Upon disassembly of the flanges at the end of the thermal cycling tests there was no evidence of buckling or cracks in any of the laminations. However, the dimensional check showed that the end of the gasket that had been in the stainless steel flange was about 0.006 in. greater in diameter than the other end. This was to be expected since calculations indicated that the stresses near the ends would be inelastic.

It was concluded from the results of these tests that the design was satisfactory for the HRT.

B. Test No. 2

Equipment and procedure:

The purpose of the second test was to subject a model of the HRT transition joint to fuel solution, during simulated reactor operation, to see what effect the corrosive solution would have on the integrity of the joint. The apparatus used is shown in Fig. 29. Details of the tests and results are given in references (3) through (7). The testing procedure is summarized in Table 2.

Table 2

Summary of thermal cycling tests run in fuel solution

Test No.	Thermal cycles (100°C - 300°C)	Time hrs
1	54	1250
2	50	1140
3	12	167
4	50	1201
5	1	951
6	144	4084
Total	311	8793

-58a-

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Figure 29

A pressure drop from outside to inside of the joint was maintained at about 40 psi during the tests. The heating and cooling cycles were at a rate of about 50°C/hr with hold periods of sixteen hours at 300°C. Test number five maintained 300°C for the duration of the test.

Two methods of leak detection were used. As shown in Fig. 29 the interior of the joint was filled with distilled water. Leakage of the fuel solution could be detected by sampling the distilled water. The other method made use of the helium leak detector sampling probe during shutdown with 50 psi helium pressure across the joint.

Results:

No leakage was detectable at any time by either of the two methods used. As yet the joint has not been disassembled so no direct results concerning the condition of the gasket can be reported at this time.

Appendix IV

Transition Joint Stress Analysis Code

The analysis of the stresses in short cylinders as used in the transition joint analysis described in this report has been programmed for the IBM 704 computer. The input data for the code are as follows:

- P = Internal (+) or external (-) pressure, psi
- E = Young's modulus, psi
- δ_L = Radial deflection of cylinder at $x = L$, in., (+) toward center
- δ_0 = Radial deflection of cylinder at $x = 0$, in., (+) toward center
- ν = Poisson's ratio
- σ = Allowable pressure hoop stress, psi

a = Radius of cylinder

Δx = Magnitude of equal distances between points where stress values are desired, in.

σ_g = Gasket seating stress (-), psi

n = Number of laminations

ℓ = Length of cylinder, in.

J = Number of n values (maximum of 10)

K = Number of ℓ values (maximum of 10)

The output of the code is as follows:

$C(1), C(2), C(4)$ = arbitrary constants, in.

$B = \beta$, in.⁻¹

$H = h$, wall thickness, in.

$S(x) = \sigma_x$, maximum summation of longitudinal stress at x , psi

$S(T) = \sigma_\phi$, maximum summation of tangential stress at x , psi

$M(x)$ = Longitudinal moment at x , lbs.

$W = w(x)$, radial deflection at x , in.

$Q = \tau(x)$, shearing stress at x , psi

An example input data sheet is shown in Fig. 30, and an output sheet for the same problem is shown in Fig. 31.

Blank first column of all fields unless -

Input Data Chg. 1543

Card 1

1	P	7	8	E	14	15	δL	21	22	$\delta 0$	28	29	v	35	36	σ	42	43	a	49	50	ΔX	56	57	σ_g	63	64	J	66	67	K	69
2000 + 3			1400 + 8			-8310 -2			3140 -2			3000 + 0			2800 + 4			3500 + 1			1000 + 0			0			01			03		

Card 2

n_1 through n_j

1	7	8	14	15	21	22	28	29	35	36	42	43	49	50	56	57	63	64	70	
4000 + 1																				

Card 3

l_1 through l_k

1	7	8	14	15	21	22	28	29	35	36	42	43	49	50	56	57	63	64	70	
5000 + 0		1000 + 1		3000 + 1																

Form of Numbers

J and K blank XX (Integer, 1-10)

All others $\left(\begin{smallmatrix} \text{blank} \\ \text{or} \end{smallmatrix} \right) \text{XXXX} \pm Y$

XXXX = decimal fraction

Y = power of 10

Fig. 30. Input data sheet.

TRANSITION JOINT STRESS ANALYSIS

INPUT PARAMETERS

P = 2.0000E 02 E = 1.4000E 07 DELTA L = -8.3100E-03 DELTA O = 3.1400E-03 NU = 3.0000E-01
 SIGMA = 2.8000E 03 A = 3.5000E 00 DELTA X = 10.00000E-02 SIGMA G = 0.
 NO. L = 1 NO. J = 3

C(1)= -0.1807E-01 C(2)= 0.1387E-01 C(4)= 0.3840E-02 B= 0.2748E 01 N= 0.4000E 01 H= 0.6250E-01 L= 0.5000E 00

S(X)= -0.1313E 06 S(T)= -0.4914E 05 M(X)= 0.8546E 02 W= 0.3140E-02 Q= -0.3606E 03
 S(X)= -0.7787E 05 S(T)= -0.2842E 05 M(X)= 0.5070E 02 W= 0.1963E-02 Q= -0.3361E 03
 S(X)= -0.2725E 05 S(T)= 0.1433E 05 M(X)= 0.1774E 02 W= -0.8382E-03 Q= -0.3264E 03
 S(X)= -0.2357E 05 S(T)= 0.2670E 05 M(X)= -0.1535E 02 W= -0.4206E-02 Q= -0.3394E 03
 S(X)= -0.7820E 05 S(T)= 0.5457E 05 M(X)= -0.5091E 02 W= -0.7077E-02 Q= -0.3753E 03
 S(X)= -0.1397E 06 S(T)= 0.7794E 05 M(X)= -0.9093E 02 W= -0.8310E-02 Q= -0.4266E 03

C(1)= -0.4611E-02 C(2)= 0.3331E-02 C(4)= 0.3840E-02 B= 0.2748E 01 N= 0.4000E 01 H= 0.6250E-01 L= 0.1000E 01

S(X)= -0.3349E 05 S(T)= -0.1981E 05 M(X)= 0.2180E 02 W= 0.3140E-02 Q= -0.8658E 02
 S(X)= -0.2226E 05 S(T)= -0.1522E 05 M(X)= 0.1449E 02 W= 0.2834E-02 Q= -0.5991E 02
 S(X)= -0.1488E 05 S(T)= -0.9898E 04 M(X)= 0.9685E 01 W= 0.2059E-02 Q= -0.3721E 02
 S(X)= -0.1049E 05 S(T)= -0.4220E 04 M(X)= 0.6827E 01 W= 0.9685E-03 Q= -0.2124E 02
 S(X)= -0.7909E 04 S(T)= 0.6544E 04 M(X)= 0.5149E 01 W= -0.3430E-03 Q= -0.1390E 02
 S(X)= -0.5707E 04 S(T)= 0.1179E 05 M(X)= 0.3715E 01 W= -0.1820E-02 Q= -0.1653E 02
 S(X)= -0.2266E 04 S(T)= 0.1713E 05 M(X)= 0.1475E 01 W= -0.3413E-02 Q= -0.3017E 02
 S(X)= -0.4156E 04 S(T)= 0.2424E 05 M(X)= -0.2706E 01 W= -0.5048E-02 Q= -0.5539E 02
 S(X)= -0.1534E 05 S(T)= 0.3376E 05 M(X)= -0.9987E 01 W= -0.6588E-02 Q= -0.9207E 02
 S(X)= -0.3295E 05 S(T)= 0.4388E 05 M(X)= -0.2145E 02 W= -0.7799E-02 Q= -0.1387E 03
 S(X)= -0.5829E 05 S(T)= 0.5353E 05 M(X)= -0.3795E 02 W= -0.8310E-02 Q= -0.1918E 03

Fig. 31. Output data sheet

C(1)= -0.3847E-02 C(2)= 0.3842E-02 C(4)= 0.3840E-02 B= 0.2748E 01 N= 0.4000E 01 H= 0.6250E-01 L= 0.3000E 01

S(X)= -0.2794E 05	S(T)= -0.1814E 05	M(X)= 0.1819E 02	W= 0.3140E-02	Q= -0.9986E 02
S(X)= -0.1469E 05	S(T)= -0.1320E 05	M(X)= 0.9562E 01	W= 0.2899E-02	Q= -0.7303E 02
S(X)= -0.5352E 04	S(T)= -0.8188E 04	M(X)= 0.3484E 01	W= 0.2346E-02	Q= -0.4918E 02
S(X)= -0.6520E 03	S(T)= -0.4096E 04	M(X)= -0.4245E-00	W= 0.1675E-02	Q= -0.2980E 02
S(X)= -0.4049E 04	S(T)= -0.2469E 04	M(X)= -0.2636E 01	W= 0.1014E-02	Q= -0.1523E 02
S(X)= -0.5559E 04	S(T)= 0.2735E 04	M(X)= -0.3619E 01	W= 0.4331E-03	Q= -0.5121E 01
S(X)= -0.5815E 04	S(T)= 0.4680E 04	M(X)= -0.3786E 01	W= -0.3389E-04	Q= 0.1234E 01
S(X)= -0.5329E 04	S(T)= 0.5924E 04	M(X)= -0.3469E 01	W= -0.3813E-03	Q= 0.4681E 01
S(X)= -0.4483E 04	S(T)= 0.6618E 04	M(X)= -0.2918E 01	W= -0.6184E-03	Q= 0.6050E 01
S(X)= -0.3539E 04	S(T)= 0.6911E 04	M(X)= -0.2304E 01	W= -0.7624E-03	Q= 0.6069E 01
S(X)= -0.2657E 04	S(T)= 0.6928E 04	M(X)= -0.1730E 01	W= -0.8328E-03	Q= 0.5333E 01
S(X)= -0.1915E 04	S(T)= 0.6765E 04	M(X)= -0.1247E 01	W= -0.8476E-03	Q= 0.4304E 01
S(X)= -0.1332E 04	S(T)= 0.6489E 04	M(X)= -0.8672E 00	W= -0.8223E-03	Q= 0.3320E 01
S(X)= -0.8805E 03	S(T)= 0.6141E 04	M(X)= -0.5733E 00	W= -0.7691E-03	Q= 0.2623E 01
S(X)= -0.5032E 03	S(T)= 0.5741E 04	M(X)= -0.3276E-00	W= -0.6974E-03	Q= 0.2377E 01
S(X)= -0.1220E 03	S(T)= 0.5298E 04	M(X)= -0.7942E-01	W= -0.6153E-03	Q= 0.2685E 01
S(X)= -0.3523E 03	S(T)= 0.5029E 04	M(X)= 0.2294E-00	W= -0.5308E-03	Q= 0.3593E 01
S(X)= -0.1012E 04	S(T)= 0.4919E 04	M(X)= 0.6586E 00	W= -0.4539E-03	Q= 0.5084E 01
S(X)= -0.1939E 04	S(T)= 0.4976E 04	M(X)= 0.1262E 01	W= -0.3986E-03	Q= 0.7057E 01
S(X)= -0.3193E 04	S(T)= 0.5295E 04	M(X)= 0.2079E 01	W= -0.3841E-03	Q= 0.9293E 01
S(X)= -0.4788E 04	S(T)= 0.5983E 04	M(X)= 0.3117E 01	W= -0.4367E-03	Q= 0.1141E 02
S(X)= -0.6663E 04	S(T)= 0.7156E 04	M(X)= 0.4338E 01	W= -0.5894E-03	Q= 0.1282E 02
S(X)= -0.8646E 04	S(T)= 0.8917E 04	M(X)= 0.5629E 01	W= -0.8808E-03	Q= 0.1266E 02
S(X)= -0.1041E 05	S(T)= 0.1133E 05	M(X)= 0.6780E 01	W= -0.1352E-02	Q= 0.9807E 01
S(X)= -0.1145E 05	S(T)= 0.1439E 05	M(X)= 0.7453E 01	W= -0.2038E-02	Q= 0.2836E 01
S(X)= -0.1099E 05	S(T)= 0.1794E 05	M(X)= 0.7156E 01	W= -0.2960E-02	Q= -0.9873E 01
S(X)= -0.8036E 04	S(T)= 0.2164E 05	M(X)= 0.5232E 01	W= -0.4106E-02	Q= -0.2999E 02
S(X)= -0.1328E 04	S(T)= 0.2485E 05	M(X)= 0.8645E 00	W= -0.5413E-02	Q= -0.5892E 02
S(X)= -0.1055E 05	S(T)= 0.3292E 05	M(X)= -0.6871E 01	W= -0.6739E-02	Q= -0.9737E 02
S(X)= -0.2904E 05	S(T)= 0.4284E 05	M(X)= -0.1891E 02	W= -0.7833E-02	Q= -0.1446E 03
S(X)= -0.5530E 05	S(T)= 0.5263E 05	M(X)= -0.3600E 02	W= -0.8310E-02	Q= -0.1978E 03

Fig. 31. Output data sheet - contd.

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