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SUBJECT: Design and Development of a 1/2-inch Titanium to Stainless Flange

TO: I. Spiewak

FROM: B. D. Draper and H. C. Roller

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SUMMARY

A 1/2-in. titanium to stainless ring joint flange with a leak detection facility has been designed, fabricated, and given preliminary thermal cycling tests. The flange consists of a titanium stub end and carbon steel lap joint flange; a 17-4 PH stainless steel octagonal ring; and a carbon steel, welding neck, ring joint flange clad with stainless steel.

Preliminary thermal cycling tests indicate that the transition flange will function properly in the intended application; namely, the installation of a titanium letdown heat exchanger in the HRT mockup.

INTRODUCTION

A Design and Development Program was initiated to provide a flanged stainless to titanium joint to be used in conjunction with the proposed titanium letdown heat exchanger for the HRT mockup.

Requirements to be incorporated in the design of the flange were as follows:

1. Operating pressure: 2000 psi
2. Design pressure: 2500 psi
3. Operating temperature: 280°C (subject to temperature cycling)
4. Fluid medium handled: UO_2SO_4
5. Leak detection: Flange to be provided with means for leak detection.
6. Maintenance: Flange to be easily assembled and disassembled, even by remote means if necessary.
7. Size: 1/2-in. Schedule 80 nominal pipe size.

PROCEDURE

a. Design

The need for a stainless-to-titanium transition joint became apparent when it was decided to fabricate a titanium letdown heat exchanger and install it in the existing stainless HRT mockup system. The problem of a joint made from these two materials is complicated due to the great dissimilarity between the thermal expansion coefficient of the two metals.

The task of joining dissimilar materials is certainly not a new one and was faced early in the HRT design program when it was necessary to design a Zircaloy-to-stainless steel joint(1). The problem was solved in this instance by the development of a laminated titanium expansion sleeve and gold gasket combination. Since this type of joint had successfully passed thermal cycling tests and some information was available on its design, a preliminary stainless-to-titanium joint design based upon this principal was made (Figure 1). It was decided,

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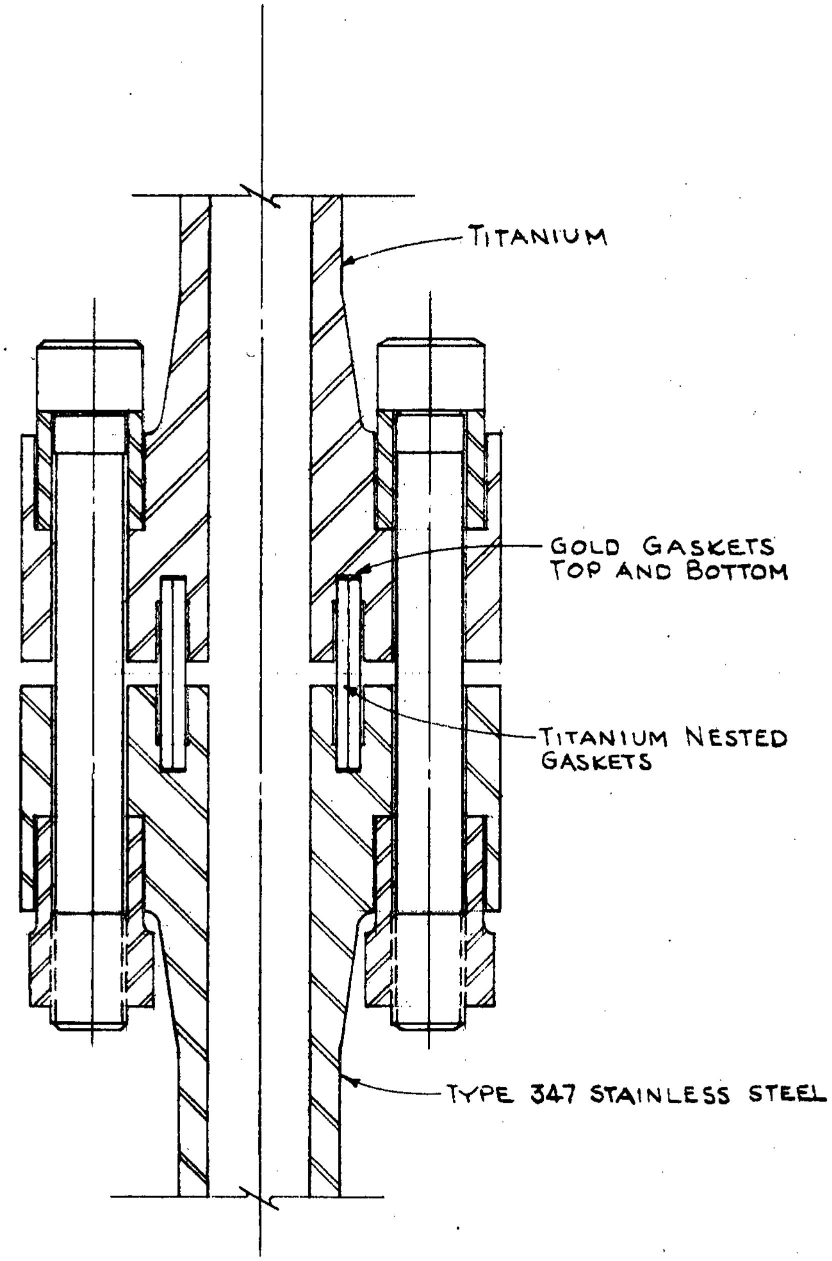

FIGURE 1.

however, that an adequate method of leak detection could not be easily incorporated in this design. Furthermore, the assembly of a joint of this type, with its close fit between sleeve and seat, requires precise positioning and poses an additional burden when performed remotely and under water. For the above reasons, it was deemed feasible to concentrate on adapting a flange of standard design to fulfill the requirements of the transition joint.

As noted previously, the wide divergence of the thermal expansion of titanium and stainless steel precluded their direct connection. The coefficients of thermal expansion for titanium and type 347 stainless steel are 5.3×10^{-6} in/in-°F and 9.8×10^{-6} in/in-°F, respectively. Therefore, an effort was made to incorporate into the design a metal more nearly matching the thermal expansion characteristics of titanium. Carbon steel, with a thermal coefficient of expansion of about 7×10^{-6} in/in-°F, was chosen as a base metal for the opposing flange half. Corrosion resistance to the UO_2SO_4 solution dictated that the carbon steel be clad with stainless steel. (See Figure 2 for final flange design).

For the titanium portion of the flange, a ring joint stub end was used in preference to a weld neck design in keeping with the desire to use metals approaching equal thermal expansion characteristics. This effectively reduced the volume of titanium in the flange assembly and increased the volume of carbon steel in the assembly to a maximum, thereby bringing the overall thermal expansion coefficient of the flange assembly close to that of carbon steel.

Cladding of the carbon steel, welding neck, ring joint flange presented several problems. First, a method of cladding the internal diameter of the flange had to be developed. Second, the cladding material, especially on the flange face, should be sufficiently thin so that when bonded to the carbon steel base it would follow the expansion characteristics of the base metal during temperature cycling.

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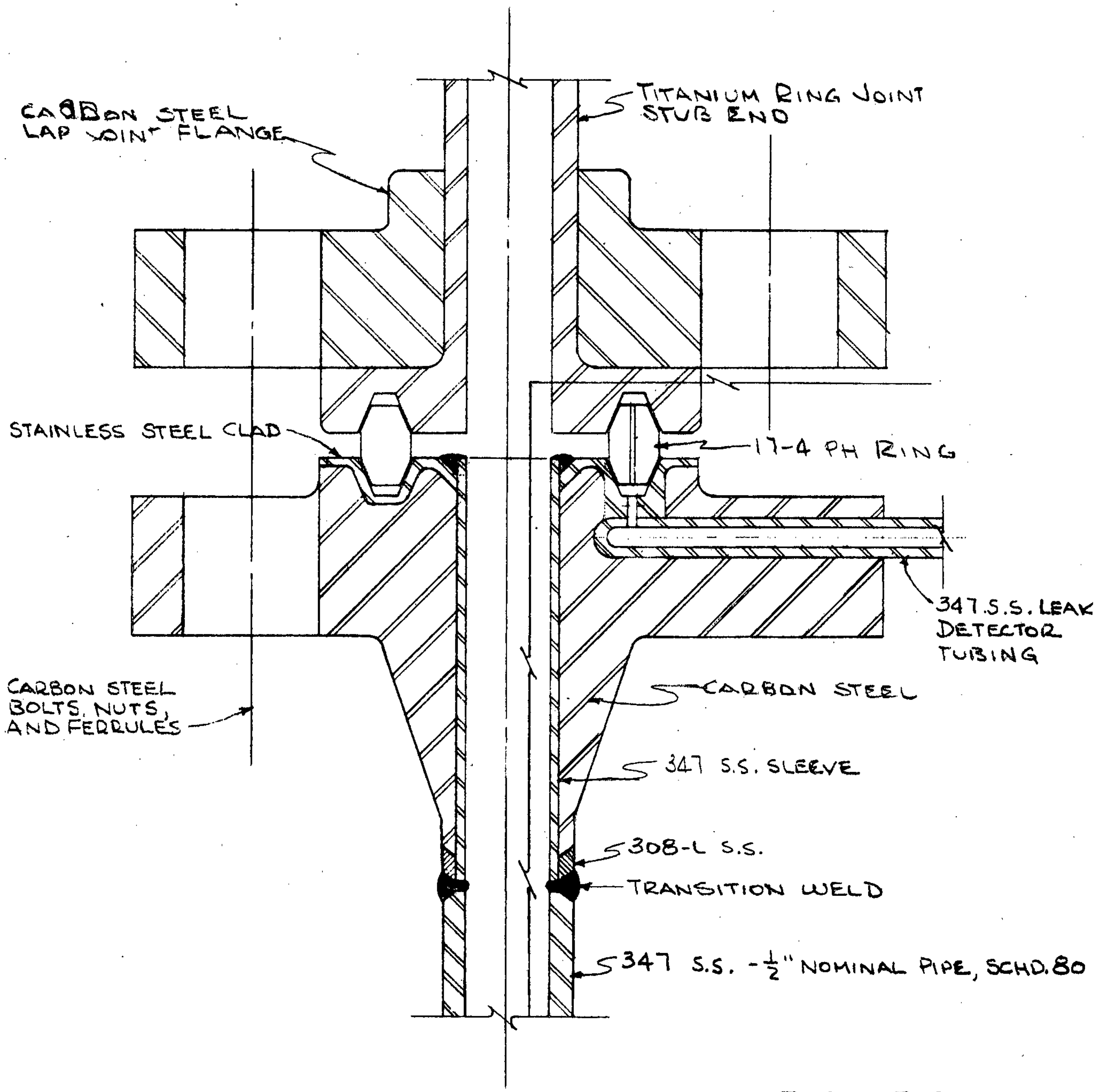

FIGURE 2

TITANIUM - STAINLESS JOINT

The cladding of the internal surface of the flange was accomplished by drilling the flange diameter oversize and inserting a stainless steel sleeve therein. The thickness of the sleeve was determined in the following manner. It was decided that the stainless steel sleeve would carry no load. Hence, the critical dimension is the thickness of the carbon steel at the welding end of the flange. Or in other words, the thicker the stainless steel sleeve, the less metal remaining on the carbon steel flange to carry the load. Therefore, knowing that the deformation of the sleeve and the flange will be the same under load, since they are in intimate contact and the sleeve carries no load, Timoshenko's equation(2) for deformation of a cylinder subject to internal and external pressure was used to determine the pressure between the sleeve and the flange. Various thicknesses were selected for the stainless sleeve thus giving a corresponding thickness for the carbon steel end. Writing the equation for the deformation of the sleeve and another for the deformation of the flange and then equating them enabled the pressure existing between the sleeve and the flange to be determined for each case. Knowing the pressure, the stress on the stainless steel sleeve and carbon steel flange could then be calculated(3). The thickness of the sleeve was therefore adjusted to give a maximum stress of 12,000 psi in the carbon steel and 14,000 psi in the stainless steel. Thus, the 60 Mil sleeve thickness was obtained. (See Appendix A for sample calculation). In addition, the standard corrosion allowance for pipes one inch and smaller in diameter is 50 Mils, hence the 60 Mil sleeve thickness appeared adequate from this standpoint. To keep the cladding thickness uniform, a 1/16-in. thick cover was placed on the face of the flange.

A slight departure from standard practice was made in the design of the ring gasket. Normally, the ring is made from a softer material than the flanges so that any distortion that occurs will be in the ring, which may be replaced, and not in the flanges. However, in this instance, it was thought best not to place a soft titanium gasket in contact with the titanium flange, since titanium is subject to galling and seizure. Therefore, an octagonal ring gasket was machined from 17-4 PH stainless steel and used in the flange. The coefficient of thermal expansion of 17-4 PH is low, being approximately 6.30×10^{-6} inch/inch-°F, which lends itself quite readily to the flange assembly.

b. Fabrication

The titanium, ring joint, stub end was machined from MST Grade III bar stock to standard tolerances. A commerical 1500 pound, carbon steel, lap joint flange was used with the titanium ring joint stub end.

For the mating flange half, a 1500 pound carbon steel, weld neck, ring joint flange was used. The face of the flange, including the sides and bottom of the groove, was machined off a depth of 1/16 inch and all corners rounded. The flange face at the inside diameter was beveled on a 45 degree angle. The inside diameter of the flange was then bored to receive the 0.666 inch O. D. stainless steel sleeve snugly. A hole was drilled from the outer periphery for the insertion of a 1/4-in. O. D. leak detector tube. Next, a 3/8-in. hole was drilled at the bottom of the groove to connect to the leak detector hole.

The end of a piece of leak detector tubing was fused and inserted into the drilled hole. Using 308-L stainless steel welding rod, the hole at the bottom of the groove was filled thus securing the leak detector tubing and a cover pass was placed on the face of the flange including the groove. Type 347 stainless steel weld rod was used to fill up the groove and make an additional pass across the face of the flange. The welding end of the flange was built up with 308-L stainless weld rod.

The face of the flange was remachined to the original dimension, i.e., the excess weld metal was removed leaving the 1/16-in. cladding. The weld overflow was removed from the inside diameter at the face and end of the flange. The flange face at the inside diameter was re-beveled to a 45° angle and the end of the flange was beveled to a standard HRP weld dimension of 50 degrees. The sleeve was then inserted in the flange and seal welded at the flange face. A standard groove was then machined in the flange face.

The sleeve was cut off and seal welded at the end of the flange to prevent dye penetrant from entering the crack. The flange was then welded to a 1/2-in. Schedule 80 stainless steel pipe.

Dye penetrant was used to check the cover passes for any flaws that might exist and the weld connecting the flange to the pipe was examined with dye penetrant and radiography.

The grooves in the titanium stub end and the stainless clad carbon steel flange were machined to fit as close as possible, without the use of special gauging devices, to the machined surfaces of the 17-4 PH ring gasket.

c. Testing - General

The flange was assembled and installed on flange test loop number 2 which is a multi-station flange testing facility, having a vacuum leak detection system. The loop is designed for thermal cycling and testing of from 1 to 8 flanges under steam pressure.

The normal thermal cycling of flanges in this loop consists of a four hour heating period, from 100°C at atmospheric pressure to a maximum of 336°C at 2000 psi; a two hour hold period at this temperature and pressure; a four hour cooling period, cooling to 100°C at atmospheric pressure; and a two hour hold period at 100°C at atmospheric pressure.

The date of temperature increase and decrease during the heating and cooling periods is approximately 55°C per hour. The system pressure corresponds to the saturated steam pressure for the given temperature.

The flange was made up with bolts having pin extensometers installed and were equipped with ferrules. All bolts were of ASTM A193, Grade B7 material. The nuts were ASTM A194, Class 2H and the ferrules were made of AISI 4140 material.

The flange was insulated with nine layers of crinkled aluminum foil .0015 in. thick wrapped around it, giving a total thickness of 3/4 to 1 inch.

Thermocouples were mounted on the flange hubs and on two bolts in the flange.

The initial bolting up of the flange was done by use of a torque wrench. The bolts were lubricated with a molybdenum disulfide lubricant (Bemol number 3 grease). The elongation of the bolts was measured by means of the pin extensometer (a steel pin through a hole the length of the bolt and welded at the point) and a dial indicator, measuring the difference between the bolt head and the free end of the steel pin, both of which have machined surfaces.

d. Testing - Detail

The pitch diameter of the ring and grooves were measured before assembly. The bolts were tightened to an initial loading of approximately 45,000 psi as determined by extensometer readings. A twenty-four hour period elapsed before thermal cycling tests began and during this period the leak detector system was pumped down to a required high vacuum. The flange was thermal cycled five times as previously described and then disassembled for inspection. Additional inspections were made following runs of five, eleven and twelve cycles.

During thermal cycling, data was obtained on bolt elongation; system, flange and bolt temperature; and flange leakage rates.

Following the above series of tests, the flange was assembled without the ferrules on the bolts. To date, fifteen thermal cycles have been completed and the flange inspected. Further runs are to be made on this assembly.

RESULTS

In Figure 3, two flanges are compared in regard to bolt stress as a function of thermal cycling. One flange is a standard 1/2-in., stainless-to-stainless, weld-neck, ring joint flange and the other is a 1/2-in. titanium-to-stainless flange as described in this report. Examination of the way in which the bolt stress decreases more rapidly in the transition joint at room temperature following thermal cycling indicates that more plastic deformation occurs in this closure. However, the residual stress at room temperature of the bolts, pre-stressed to 45,000 psi, decreases with thermal cycling and asymptotically approaches a value of approximately 28,000 psi. This compares with a value of approximately 40,000 psi as observed in the all stainless flange.

Another point to be noted from Figure 3 is that the bolt stress in the transition flange is less at operating temperature than at room temperature, in contrast to the increased bolt stress observed in the all stainless flange. This phenonmeon has been observed previously in joints of dissimilar materials[4]. This could be attributed to the fact that the thermal expansion coefficients of titanium and 17-4 PH stainless are lower than that of the carbon steel which comprises the bulk of the flange.

In Table I are listed maximum and average leak rates observed during a series of thermal cycles of the all stainless flange and the titanium to stainless flange. The cycles chosen as representative of the all stainless flange were those which followed the installation of a new ring and thus were similar to the new titanium to stainless assembly.

It can be seen that at room temperature the leak rates are fairly constant for both flanges over a number of runs and disassemblies. However, at operating temperature and pressure, the largest leak rate occurred after the first assembly and during the first run and gradually diminished as additional runs and disassemblies were made. From this it is concluded that the increased

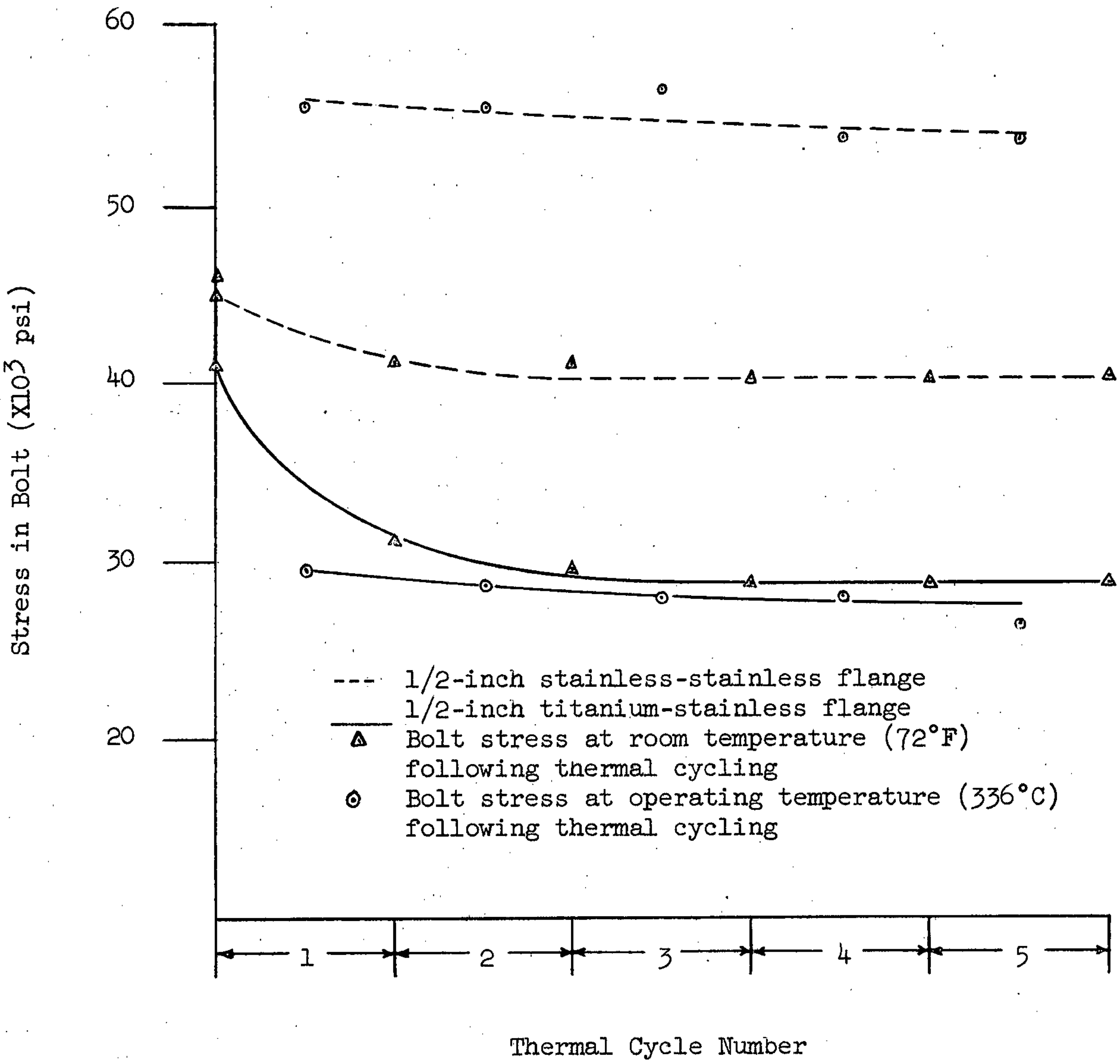
60
50
40
30
20
Stress in Bolt (X10^3 psi)
1
2
3
4
5
1/2-inch stainless-stainless flange
1/2-inch titanium-stainless flange
Bolt stress at room temperature (72°F) following thermal cycling
Bolt stress at operating temperature (336°C) following thermal cycling
Thermal Cycle Number

Figure 3

TABLE 1

Stainless-to-Stainless Joint

Cycle No.	Cycles Per Run	Maximum Leak Rate Per Run at Room Temp. and 0-psig, g H_2O/day	Average Leak Rate Per Run at Room Temp. and 0-psig, g H_2O/day	Maximum Leak Rate Per Run at 336°C and 2000 psig, g H_2O/day	Average Leak Rate Per Run at 336°C and 2000 psig, g H_2O/day
44-53	10	1.09×10^{-4}	0.94×10^{-4}	16.30×10^{-4}	10.20×10^{-4}
53-61	9	1.12	1.04	7.14	6.28
62-66	5	1.08	0.89	5.39	4.59
67-76	10	1.63	1.48	3.69	2.87
Titanium-to-Stainless Joint					
1-5	5	1.18×10^{-4}	0.82×10^{-4}	6.62×10^{-4}	4.79×10^{-4}
6-10	5	0.65	0.48	2.77	2.41
11-21	11	0.97	0.73	2.67	1.59
22-33	12	0.85	0.80	0.80	0.77

deformation that occurs to the ring and grooves during a number of assembly operations is apparently beneficial from the standpoint of decreased leak rate.

In the all stainless flange some deformation occurs to both the ring and the grooves and, in some instances, the ring binds in one flange half requiring external force for removal. In the titanium to stainless flange there was no visible deformation of the 17-4 PH stainless steel ring, a very slight deformation of the stainless steel clad flange, and about a 6 Mil maximum indentation of one side of the groove of the titanium stub end. During all disassemblies, the 17-4 PH ring was easily removed from the flange with no evidence of binding or seizure.

CONCLUSIONS

The results obtained from the thermal cycle tests indicate that it is feasible to fabricate a titanium-to-stainless flange as described in this report. The leak rates obtained on this flange compare favorably with those of a comparable all stainless flange. Additional flanges are being fabricated in order to install a titanium letdown heat exchanger on the HRT mockup. This will enable the flange to be tested with UO_2SO_4 as the process fluid, however, it is anticipated that no harmful defects will arise from this application.

APPENDIX A

Figure 4 represents the end of the carbon steel flange with the stainless steel liner of unknown thickness inside.

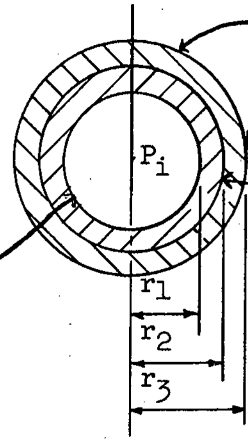

Figure 4

Writing Timoshenko's equation for the deformation of any point in concentric cylinders due to internal and external pressure and adding a term for temperature effect, the deformation at r_2 for the outside cylinder equals,

$$u_o = \frac{(1-V)}{E_o}\frac{(r^2{}_2\ P_2)\ r^2}{r_3{}^2-r_2{}^2} + \frac{(1+V)}{E_o}\frac{(r^2{}_2 r_3{}^2 P_2)}{(r_3{}^2-r^2{}_2)r_2} + a_oT_or_2 \qquad (1)$$

and the deformation at r_2 for inside cylinder equals,

$$u_i = -\frac{(1-V)}{E_i}\frac{(r_2{}^2P_2)r_2}{(r_2{}^2-r_1{}^2)} - \frac{(1+V)}{E_i}\frac{(r_1{}^2r_2{}^2P_2)}{(r_2{}^2-r_1{}^2)r_2} + a_iT_ir_2 \qquad (2).$$

As was noted previously, the inner cylinder carries no load, hence, $P_i = 0$; and since there is no external load on the carbon steel cylinder $P_o = 0$ also. Therefore, equating (1) and (2) enables one to solve for the pressure between the cylinders due to temperature increase alone.

$$\frac{(1-V)}{E_o}\frac{(r_2{}^3P_2)}{r_3{}^2-r_2{}^2} + (1+V)\frac{(r_2r_3^2P_2)}{(r_3{}^2-r_2{}^2)} + a_oT_or_2 = -\frac{(1-V)}{E_i}\frac{(r_2{}^3P_2)}{(r^2{}_2-r_1{}^2)} - \frac{(1+V)}{E_i}\frac{(r^2{}_1r_2P_2)}{(r^2{}_2-r_1{}^2)}$$

$$+ a_iT_ir_2 \qquad (3)$$

rearranging (3)

$$P_2\left[\frac{1-V}{E_o}\;\frac{(r_2^{\,3})}{(r_3^{\,2}-r_2^{\,2})}+\frac{1+V}{E_o}\;\frac{(r_2 r_3^{\,2})}{(r_3^{\,2}-r_2^{\,2})}\right]+a_oT_or_2=-P_2\left[\frac{1-V}{E_i}\;\frac{(r_2^{\,3})}{(r^2{}_2-r_1^{\,2})}+\frac{(1+V)}{E_i}\;\frac{(r_1^{\,2}r_2)}{(r_2^{\,2}-r_1^{\,2})}\right]+a_iT_ir_2 \qquad (4)$$

Transposing and solving for P_2

$$\frac{r_2(a_iT_i-a_oT_o)}{\left[\frac{(1-V)}{E_o}\;\frac{r_2^{\,3}}{(r_3^{\,2}-r_2^{\,2})}+\frac{(1+V)}{E_o}\;\frac{(r_2r_3^{\,2})}{(r^2{}_3-r_2^{\,2})}\right]+\left[\frac{(1-V)}{E_i}\;\frac{(r_2^{\,3})}{(r_2^{\,2}-r_1^{\,2})}+\frac{(1+V)}{E_i}\;\frac{(r_1^{\,2}r_2)}{(r_2^{\,2}-r_1^{\,2})}\right]} \qquad (5)$$

$$P_2=\frac{r_2(a_iT_i-a_oT_o)}{r_2\left[\frac{(1-V)}{E_o}\;\frac{r_2^{\,2}}{(r_3^{\,2}-r_2^{\,2})}+\frac{(1+V)}{E_o}\;\frac{(r_3^{\,2})}{(r_3^{\,2}-r_2^{\,2})}\right]+r_2\left[\frac{(1-V)}{E_i}\;\frac{(r_2^{\,2})}{(r_2^{\,2}-r_1^{\,2})}+\frac{(1+V)(r_1^{\,2})}{E_i(r_2^{\,2}-r_1^{\,2})}\right]} \qquad (6)$$

$$P_2=\frac{(a_iT_i-a_oT_o)}{\frac{1}{E_o(r_3^{\,2}-r_2^{\,2})}\left[(1-V)\,r_2^{\,2}+(1+V)(r_3^{\,2})\right]+\frac{1}{E_i(r_2^{\,2}-r_1^{\,2})}\left[(1-V)(r_2^{\,2})+(1+V)(r_1^{\,2})\right]} \qquad (7)$$

$$P_2=\frac{(a_iT_i-a_oT_o)}{\left[\dfrac{(1-V)\frac{r_2^{\,2}}{r_3^{\,2}}+(1+V)}{E_o\,\left(1-\frac{r_2^{\,2}}{r_3^{\,2}}\right)}\right]+\left[\dfrac{(1-V)\,\frac{r_2^{\,2}}{r_1^{\,2}}+(1+V)}{E_i\,\dfrac{(r_2^{\,2}-1)}{r_1^{\,2}}}\right]} \qquad (8)$$

This equation for P_2 is in the form of constants and ratios of the unknown cylinder thicknesses for ease in trial and error solving. The correct trial solution follows as an example.

The known values are:

r_1 = .273 inches

r_3 = .420 inches

$a_i = 9.8 \times 10^{-6}$ in/in-°F - Thermal coefficient of expansion for stainless steel.

$a_o = 6.8 \times 10^{-6}$ in/in-°F - Thermal coefficient of expansion for carbon steel.

$T_i = T_o = 500°F$

V = Poisson's ratio = .3 (for elastic range)

$E_i = 12.9 \times 10^6$ psi - carbon steel

$E_o = 25 \times 10^6$ psi - stainless steel

Assume an inner cylinder thickness of 0.063 inches which makes $r_2 = 0.336$ inches.

$$P_2 = \frac{(9.8 \times 10^{-6} \times 500 - 6.8 \times 10^{-6} \times 500)}{\left[\dfrac{(0.7)\dfrac{(.336^2)}{(.420^2)} + 1.3}{25 \times 10^6(1 - \dfrac{.336^2}{.420^2})}\right] + \left[\dfrac{(0.7)\dfrac{(.336^2)}{(.273^2)} + 1.3}{12.9 \times 10^6(\dfrac{.336^2}{.273^2} - 1)}\right]} \qquad (9)$$

$$P_2 = \frac{15 \times 10^{-4}}{\dfrac{1.748}{9 \times 10^6} + \dfrac{2.362}{6.68 \times 10^6}} \qquad (10)$$

$$P_2 = 2740 \text{ psi} \qquad (11)$$

Using Timoshenko's Equation 200 noted previously gives the stress in the stainless steel shell as:

$$\text{Stress} = \frac{a^2P_i - b^2P_o}{b^2-a^2} + \frac{(P_i - P_o)a^2b^2}{r^2(b^2-a^2)} \qquad (12)$$

when $a = .273$ inches

$b = .336$ inches

$r = b$

$$\text{Stress} = \frac{(.273^2)(0)-(.336^2)(2740)}{(.336^2-.273^2)} + \frac{(0-2740)(.273^2)(.336^2)}{(.336^2)(.336^2-.273^2)} \tag{13}$$

$$\text{Stress} = \frac{-310}{0.038} - \frac{23.1}{.00429} \tag{14}$$

$$\text{Stress} = -13{,}600 \text{ psi (compression)} \tag{15}$$

Stress in the carbon steel cylinder

when:

$a = .336$ inches

$b = .420$ inches

$r = b$

Substituting in (12)

$$\text{Stress} = \frac{(.336^2)(2740)-(.420^2)(0)}{(.420^2-.336^2)} + \frac{(2740-0)(.336^2)(.420^2)}{.420^2(.420^2-.336^2)} \tag{16}$$

$$\text{Stress} = \frac{310}{.063} + \frac{54.5}{.0111} \tag{17}$$

$$\text{Stress} = 9830 \text{ psi} \tag{18}$$

Both of these stress values are below the maximum limits of 14,000 psi and 12,000 psi established previously and the 1/16-in. (0.063) thick sleeve is thus obtained.

APPENDIX B

The leak rate calculations are based on a memo by R. D. Cheverton of December 7, 1956, in which the following equation is presented:

$$\text{L.R.} = \frac{V(\Delta P)\ 31}{\Delta t(273+T)} \qquad (1)$$

where:

L.R. = Leak Rate, $\frac{\text{c.c. at standard atms conditions}}{\text{day}}$

V = Leak detector volume, c.c.

ΔP = Change of pressure in volume V, microns of Hg.

Δt = Time for pressure change, seconds

T = Average temperature of gas in volume, V, °C.

An additional term is required to convert to grams of H_2O/day at standard conditions.

$$\text{L.R.} = \frac{V(\Delta P)\ \ 31}{\Delta t(273+T)} \times \frac{18}{22,400} \qquad (2)$$

$$\text{L.R.} = \frac{V(\Delta P)\ .0249}{\Delta t(273+T)} \qquad (3)$$

The volume of the leak detector system is 49.25 c.c. and the average absolute temperature at operating conditions is 360°K and at room temperature 300°K.

$\therefore$ substituting in Equation (3).

$$\text{L.R. (Hot)} = \frac{49.25(.0249)\Delta P}{360\Delta t} = .00341\ \frac{(\Delta P)}{\Delta t}\ \frac{\text{Grams } H_2O}{\text{Day}} \qquad (4)$$

$$\text{L.R. (Cold)} = \frac{49.25(.0249)\Delta P}{300\Delta t} = .00409\ \frac{(\Delta P)}{\Delta t}\ \frac{\text{Grams } H_2O}{\text{Day}} \qquad (5).$$

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