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Experimental Stresses for a Contact Load on Simulated SPF
Reactor Fuel Elements

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ABSTRACT

As part of the SNAP reactor program a structural test was conducted on simulated fuel elements to determine experimentally the stresses on the inner surface of the cylinder due to local contact loads. These local loads were applied by simulated fuel elements with helical fins 0.072 inch to 0.160 inch wide on a 5-1/2 inch pitch, acting on 3/4 inch diameter thin walled cylinders, 21 inches long which were internally strain gaged. The results were correlated with analytical efforts utilizing a Bijlaard analysis. Stresses were also determined for the case where plain tube was loaded against a bowed plain tube.

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REV	SUMMARY OF CHANGE	APPROVALS AND DATE
	<p><u>CHANGE # 1</u> Appendix F, "Original Test Data" is added to this T.R. This section includes the strain gage readings as they were obtained from the Strain Gage Data Acquisition System. Appendix F is not to be duplicated because of its length (pages 121 to 341). This section is being stored at Engineering Data for personnel who wish to review it.</p>	<p><i>RK</i> <i>RC</i> <i>E Donaway</i> <i>4/3/72</i></p> <p>REL. DATE <i>4.4.72*</i></p>

I. INTRODUCTION

A series of tests were conducted to determine the local load contact stresses that develop in the SPF reactor fuel element cladding as a result of the contact that occurs between adjacent elements within the core during reactor operation.

This was accomplished in support of the Space Power Facility (SPF) Reactor program at Atomics International. The SPF reactor is a 300 kwz Zirconium Hydride Reactor with outlet coolant temperatures of 1200°F. The heat produced in the fuel elements within the core will be transferred to the primary NaK-78 coolant and then to the heat exchanger of a Brayton cycle power conversion system. There are 199 reactor fuel elements--positioned between the grid plates (Figure 1) on a pitch of 0.737 inches. The basic fuel element (Figure 2) is a Hastelloy X tubular member with end closures. The inner surface of the tube is coated with a ceramic coating which acts as a hydrogen barrier to prevent permeation through the Hastelloy X during reactor operation. In order to promote better hydraulic mixing in the core the external surface of 138 of the fuel elements have integrally hobbled fins helixed on a 5.5 inch pitch. Half of these are right hand helix and the other half a left hand helix. The remaining 61 elements have no fins at all. The fuel elements are arranged for maximum hydraulic advantage.

During reactor operation there are thermal gradients dictated by the power and coolant flow distribution. These gradients cause each element to bow as a beam supported at the grid plates until it contacts its neighbors. The contact loads in the core are then redistributed and balanced. Because of the fin geometry the helical fin contacts the adjacent element on the cylindrical section of the element. The magnitude of stress on the inner surface induced by these predetermined loads is important because of the inability of the ceramic hydrogen permeation barrier to withstand high stresses.

REACTOR VERTICAL SECTION

FIGURE 1

FIGURE 2

Before the subject test data were available the local stress magnitudes on the inner surface of the fuel elements were estimated using the analysis method developed by Professor Bijlaard⁽¹⁾. This analysis predicts the stresses adjacent to a load pad acting on a cylinder as a function of the geometry of the cylinder and the load pad dimensions. The test data generated is compared to the results of this analysis. Assuming the analysis method is correct, the effective pad dimensions for various fin or load pads can be evaluated.

Testing was conducted on two tubes of different thicknesses (0.032 inch and 0.020 inch). The tube length was 21 inches and approximately 0.75 inches in diameter. Ten strain gage (0.015 inch gage length) rosettes were inserted inside the tube at various locations to determine the detail strains. The strain gaged tube was then positioned in the test fixture (Figure 3) and loaded with a simulated fin section. The fin geometry was a major parameter in the test and several configurations were tested on each of the two tubes. Also the loads and contact positions were varied producing a great deal of strain data. This data was then correlated to the Bijlaard analysis for interpolation to the specific geometry of the SPF reactor fuel element.

Testing was conducted at the LMEC Strain Gage Laboratory (Building 6) in a temperature controlled environment. Special gratitude is extended to Gerald Czechanski and Gordan Twa of LMEC for their help in the wiring of the gages, use of the equipment and associated knowledge.

II. TEST SYSTEM DESCRIPTION

The test items consisted of two tubes approximately 21 inches long, 3/4 inches in diameter and nominally 0.032 or 0.020 inches thick. The tubes were 304 stainless steel with yield strengths of approximately 50,000 psi. One tube was purchased as 0.035 wall thickness and centerless ground to a nominal 0.032 inch wall thickness while the other was purchased as 0.020 inch nominal wall. The detailed wall thickness and outside diameter measurements were obtained at the 9 positions indicated in Figure 4. A water bath vidigage was used for the wall thickness measurements and verification made with standards and a ball micrometer. The measurements are presented in Appendix A along with the material certifications. The gages used were Micro Measurements micro-miniature, three element 60° delta rosettes. The gage type was EA-09-015 YD-120 and they were installed by Premtronics Incorporated.

The gages were applied on the inner surface of the tube by first placing them in their exact location on a developed view of the inner grid. This was accomplished by drawing the developed view of the inner tube circumference with the exact gage location spotted. Then cellophane tape was laid on the developed view, sticky side up and the prewired gages positioned to the tape (Figure 5). The tape was then wrapped around an inflatable mandrel. This mandrel was then inflated under pressure to set the gages against the inner surface of the tube. The mandrel was removed and the cement previously placed on the gages was cured. After this process was complete the gage locations were checked by x-ray, Figure 6. The gages all appeared to be within a degree of requested circumferential position and within the longitudinal tolerances for the 0.032 inch tube (Figure 7). For the 0.020 inch tube the circumferential position of the gages was shifted 30° from the desired position as shown in Figure 8, the longitudinal positions were within tolerances. No problems were encountered due to the shift. Sequential testing again verified the positioning.

The rosette wiring in Figure 7 for the 0.032 nominal wall tube is a three wire setup with a common wire between gages B and C. The common wire eliminated the advantage of the temperature com-

INSTRUMENTED TUBE

FIGURE 4

FIGURE 5: Strain gage layout, mandrel and tube before gage installation

FIGURE 6: Tube x-ray after gage installation

GAGE WIRING DIAGRAM FOR THE 32 MIL TUBE, FIG 7

DEVELOPED VIEW OF ROSETTE
LOCATIONS FOR THE 32 MIL TUBE

GAGE WIRING DIAGRAM FOR 20 MIL TUBE

FIG 8

pensation inherent in the three wire system and lead to feedback problems when subject to test conditions. To eliminate this problem an individual three wire arrangement was used for the 0.020 inch wall tube as shown in Figure 8. The position of the rosettes remained the same with the exception of gage No. 4 which was moved to the 180° position. Rosette positions inside the tube were verified by x-rays similar to Figure 6.

Photographs of the complete tubes before installation in the test fixture are shown in Figures 9 and 10.

FIGURE 9

0.032" Tube - Complete

FIGURE 10
0.020" Tube - Complete

III. HOOKUP AND AUTOMATIC READOUT

The instrumentation used to acquire the data was a B&F Strain Gage Data Acquisition System Model SY-156-16 shown in Figure 11. The system had thirty channels designed to accept resistance strain inputs, condition them, change them to digital data, and then print them on tape. All thirty channels and the time were printed in 8 seconds. A sample run is shown in Figure 12. Time is indicated as the first 000 channel and the following 000 to 029 channels indicate the strains. Apparent strain values are one half the actual strain values because the bridge excitation voltage was half the normal 1 volt value. The voltage was lowered to prevent overheating of the micro-miniature gages. Calibration data for the gages is given in Appendix B and the method of data analysis is given in Appendix C

FIGURE 11
DATA ACQUISITION
SYSTEM

FIGURE 12

Sample Output from Strain Gage Data Acquisition System

029 -00000 0	29 +00000 0	29 +00000 0	29 +00003 0	29 -00003 0
028 +00000 0	28 +00002 0	28 +00003 0	28 +00006 0	28 +00000 0
027 -00002 0	27 -00039 0	27 -00065 0	27 -00119 0	27 -00001 0
026 -00003 0	26 -00000 0	26 +00000 0	26 +00003 0	26 -00001 0
025 -00001 0	25 +00005 0	25 +00011 0	25 +00024 0	25 +00000 0
024 -00003 0	24 -00041 0	24 -00067 0	24 -00119 0	24 -00002 0
023 -00004 0	23 -00000 0	23 +00003 0	23 +00009 0	23 -00002 0
022 -00002 0	22 +00008 0	22 +00018 0	22 +00036 0	22 +00001 0
021 -00003 0	21 -00041 0	21 -00070 0	21 -00125 0	21 -00003 0
020 -00003 0	20 +00007 0	20 +00015 0	20 +00031 0	20 -00005 0
019 +00000 0	19 +00016 0	19 +00027 0	19 +00050 0	19 -00002 0
018 +00010 0	18 -00032 0	18 -00065 0	18 -00142 0	18 -00006 0
017 -00002 0	17 +00015 0	17 +00028 0	17 +00053 0	17 -00001 0
016 +00003 0	16 +00035 0	16 +00058 0	16 +00102 0	16 +00007 0
015 -00007 0	15 -00054 0	15 -00088 0	15 -00152 0	15 -00002 0
014 +00027 0	14 +00293 0	14 +00488 0	14 +00828 0	14 +00039 0
013 -00012 0	13 +00020 0	13 +00066 0	13 +00091 0	13 -00015 0
012 -00001 0	12 +00029 0	12 +00020 0	12 +00107 0	12 -00002 0
011 +00000 0	11 +00012 0	11 +00020 0	11 +00036 0	11 +00000 0
010 +00001 0	10 +00014 0	10 +00023 0	10 +00039 0	10 +00002 0
009 -00007 0	09 +00034 0	09 +00063 0	09 +00120 0	09 -00006 0
008 -00003 0	08 -00045 0	08 -00058 0	08 -00105 0	08 -00000 0
007 -00004 0	07 -00038 0	07 -00062 0	07 -00108 0	07 -00000 0
006 -00003 0	06 -00039 0	06 -00064 0	06 -00116 0	06 +00000 0
005 -00006 0	05 -00025 0	05 -00035 0	05 -00060 0	05 +00003 0
004 -00000 0	04 -00016 0	04 -00027 0	04 -00048 0	04 +00000 0
003 -00002 0	03 -00000 0	03 -00000 0	03 -00001 0	03 -00002 0
002 -00000 0	02 +00011 0	02 +00019 0	02 +00034 0	02 -00003 0
001 -00002 0	01 +00009 0	01 +00020 0	01 +00038 0	01 -00000 0
000 -00010 0	00 +00036 0	00 +00068 0	00 +00129 0	00 -00007 0
999 999999 9	99 999999 9	99 999999 9	99 999999 9	9 999999 9
000 110815 0	100 111300 0	00 111746 0	00 112003 0	0 112346 0

0#

5#

10#

15#

20#

Run 186 - 0.072 Std fin not parallel to tube

IV. TEST FIXTURE

The test fixture was designed to accommodate the S8DR fuel element strain gage test which was a test where two bare tubes were initially bowed and then brought together under controlled loading (Reference 3). There was little modification of the basic fixture necessary for the SPf reactor fuel element testing. The strain gaged cylinder was installed in the rigid end fitting which are in turn attached to pivot points twenty-one inches apart in the test fixture. The pivot point in the end fittings for this strain gaged upper tube is supported in the fixture frame such that the upper tube cannot move vertically, but can move a small amount laterally to compensate for the change in apparent tube length due to the tube bow. The lower tube end fittings are mounted on a slide which allows the tube to move vertically in the frame as well as allowing for a small amount of the lateral motion required. The lower tube in this fixture is a 0.540 diameter solid steel rod in which a 2-inch length section of tube with the desired fin geometry to be tested was placed. This steel rod was clamped in the rigid end fittings. Attached to the end fittings was a bell crank. A tension tie was placed between the two bell cranks and when tensioned, the tie produced an end moment on the steel rod. This caused curvature of the rod so that during the test intimate contact would be controlled and occur at the fin on the finned segment. An attempt was made to minimize the friction loads in the test setup. All the pulleys were ball bearing mounted and the rigging was nylon cord. Loads are applied to the lower tube by putting a known weight attached to rod suspended below the fixture in Figure 13. The rod is attached to nylon cord which is run over the pulley to a member which is rigidly tied to the end fitting of the lower tube. The load applied to this will cause the lower tube to rise and contact the strain gaged tube.

V: CONTACT FIN DETAIL GEOMETRY

The details of the finned element segment are important in evaluating the data. The fins are shown in Figure 14. From the foot print of the fin, which is reproduced in Figure 15, the longitudinal contact length was measured for each fin configuration assuming very small widths. These lengths were then assumed to be the contact pressure pad lengths. The unknown parameter in correlation to the Bijlaard analysis reduced then to the magnitude of the contact width. These known lengths are listed below for the various fin configurations.

	<u>Length</u>
0.072" Standard Fin	0.20"
0.072" Fin Ground Flat	0.25"
0.125" Fin	0.30"
0.125" Fin Ground Flat	0.31"
0.160" Fin	0.40"
0.125" Fin With 0.4 inch Radius	0.32"
Reverse Curvature	
0.125" Fin With 0.5 inch Radius	0.32"
Reverse Curvature	

In addition to the measurements of the fins themselves foot prints of the fins on a similar tube were measured and the two seemed to be in agreement. These are shown in Figure 16. The fin geometries tested were all profilometered to determine their exact geometry. These profilometer traces are presented in Figures 17 through 28.

FIGURE 14
FIN GEOMETRY

0.125" Fin with 0.5"
Radius Reverse Curvature

.072 STANDARD FIN

$\frac{1}{8}$ " STANDARD FIN

.072" GROUND FLAT FIN

$\frac{1}{8}$ " GROUND FLAT FIN

.160 STANDARD FIN

$\frac{1}{8}$ " STANDARD FIN WITH .9" R UNDERCUT

$\frac{1}{8}$ " STANDARD FIN WITH .5" R UNDERCUT

FIN PROFILES

FIGURE 15

FIN FOOTPRINTS
FIGURE 16

0.72" Standard Fin
Trace Perpendicular to the Fin

FIGURE 17

0.72" Standard Fin
Trace Perpendicular to the Tube

FIGURE 18

0.72" Fin with Flat Spot
Trace Perpendicular to the Fin
FIGURE 19

0.72" Fin with Flat Spot
Trace Perpendicular to Tube
FIGURE 20

.125" Standard Fin
Trace Perpendicular to the Fin

FIGURE 21

.125" Standard Fin
Trace Perpendicular to the Tube

FIGURE 22

.125" Fin with Flat Spot
Trace Perpendicular to Fin

FIGURE 23

.125" Fin with Flat Spot
Trace Perpendicular to the Tube

FIGURE 24

.160 Standard Fin
Trace Perpendicular to Fin

FIGURE 25

.160 Standard Fin
Trace Perpendicular to Tube

FIGURE 26

.125" Scalloped Fin
Radius = .5 Inches
Trace Perpendicular to Tube

FIGURE 27

.125" Scalloped Fin
Radius = .4 inches
Trace Perpendicular to Tube

FIGURE 28

VI. TESTING AND RESULTS

The testing was conducted on the two tubes utilizing the previously described fixture and peripheral equipment. All together seven fin configurations were utilized in applying loads to the strain gaged tube. These fin configurations were all helixes on a 5-1/2 inch pitch. The first series of tests had loads applied directly over the gages in order to find the maximum circumferential stress in the tube. Later other load locations were used to find a distribution of stress away from the load.

A typical test was conducted by first lubricating the fixture and adjusting it to minimize friction. Then the lower assembly of the fixture was counter balanced so that the subsequent applied weight would be the force of the fin against the strain gaged tube. For the first series of tests the finned section was positioned under a particular strain gage rosette and loaded to some moderate force. The gaged tube was then loosened from its end fixture and rotated until the Y legs of the rosette were equalized ensuring the rotational positioning of the gaged tubes was satisfied. Then the end fixtures were tightened to ensure the simply supported beam concept twenty-one inches between pivot points. The fin section on the lower rod, which had the fin geometry to be tested, was then rotated which caused a variation of axial contact location. The axial leg of the rosette was monitored until the reading was maximized. At the same time the Y legs of the rosette were monitored until they were maximized to ensure positioning of the applied load, through fin contact, was directly over the gage in question. After confidence was established that the fin was contacting the gaged tube directly over a gaged rosette, the loads were relieved, the system balanced and then loads applied in five pound increments until the stress response of the tube was significant.

After data was gathered for this particular location and fin geometry, the fin was moved to another strain gage location and the process repeated. The data was plotted as a function of load and repeated over enough gages to demonstrate repeatability of data. The philosophy used in interpreting the data is that the envelope of maximums is used assuming that the data below this envelope is data obtained where

the fin position was slightly off the strain gage and the response was not maximized.

It was found that on the first series of tests the fin contact plane was not parallel with the strain gaged tube to the extent that the sharp corner of the fin was doing the loading. These values are unrealistic for reactor loading and were not utilized in the correlation of the Bijlaard analysis. After the problem was discovered, parallelism was checked and verified prior to data acquisition. Looking at the detail data (Appendix D, page 66) it was observed that the stress-load relationship was linear. The results are tabulized below for the various fin and tube configurations.

TABLE I

Resulting Stresses (psi/lb load)					
Fin Configuration	.032 Tube		Max.	.020 Tube	
	Hoop	Axial		Hoop	Axial
.072 fin - non-parallel rods	1250	760	3640	3190	2240
.027 fin	1030	812	3170	3000	1840
.072 fin ground flat	855	710	2660	2370	1600
.125 fin	970	750	2640	2450	1330
.125 fin ground flat	855	600	2250	2130	1440
.160 fin	950	875	2590	2550	1070
.125 fin with a 4" radius reverse curvature	720	580	2000	1620	900
.125 fin with .5" radius reverse curvature	700	650	2250	2050	1350

The results of the 0.032 tube indicated that the principal directions were near the circumferential and axial directions. When testing and data reduction commenced on the 0.020 tube, it was discovered that the principal direction had shifted in the thinner tube. The principal stress is computed as shown in the above table of results. There is interest in determining the distribution of stress around the load application. To accomplish this, some of the fin configurations were used to study the distribution. The load was positioned over the gage and then the gage rotated so that the circumferential distribution was determined. The results indicate that the circumferential stress

significantly drops off with angular position. For instance in the case of the 0.072 inch wide rounded fin under a 30 pound load, the stress drops from a maximum of 34000 psi at 0° to 11000 psi at a position 10° from center. Similarly in the longitudinal direction in the zero degree position, the fin was rotated on this fixture and because of the helix effect this produced a longitudinal position shift on the gaged tube. The circumferential stress here dropped from the maximum of 34000 psi to 9000 psi only .2 inches from the maximum stress load position.

All of the data discussed here is presented in Appendix B.

Correlation to the Bijlaard analysis was necessary in order to interpolate to different geometries of fuel elements. This analysis determines the local stresses in a cylinder due to a pressure load over a given length and width.

The Bijlaard analysis computes the moments and forces at the center 0 of an equally distributed load as shown in Figure 29A,

Fig. 29A

Fig. 29B

where for the case of the fin contacting the fuel element tube, the load is distributed differently possibly peaking at the edges of the fin contact. Also for this special case the helix angle of the fin of 21° causes a variance of load distribution along the length of the fin.

Professor Bijlaard has a developed design curves from his equations which are functions of the tube radius, thickness, length and the contact pad geometry. These curves for the case of an axial load on the cylinder for a length/radius ratio of 8 are presented in Figures 29 through 32. The correction to this data for length/radius ratios of the subject test (approximately 56) are negligible. The parameters of these curves are a function of the pad geometry β . For a rectangular pad:

Fig. 1C—Bending moment M/P due to an external radial load P on a circular cylinder (transverse axis)

Fig. 2C—Bending moment M_x/P due to an external radial load P on a circular cylinder (transverse axis)

Fig. 3C—Membrane force $N_\phi/(P/R_m)$ due to an external radial load P on a circular cylinder

$$\beta_1 = \frac{C_1}{R_m} \quad \beta_2 = \frac{C_2}{R_m}$$

Where C_1 is half the circumferential pad width
 C_2 is half the longitudinal pad width
 R_m is the mean radius of the cylinder

For $\beta_1 / \beta_2 > 1$

$$\beta = \left[1 - 4/3 \left(1 - \frac{\beta_1}{\beta_2} \right) (1 - K_2) \right] \sqrt{\beta_1 \beta_2}$$

Where the K values are

- 1. 48 for determination of N_ϕ
- 1. 20 for determination of N_x
- .88 for determination of M_ϕ
- 1. 25 for determination of M_x

With this information and the design curves, the data was parameterized for the two tubes tested and the test data superimposed on the curves.

As described previously, the pad contact configuration is the assumed unknown. The fin profiles are shown in Figure 15 for each fin and the length along the axis determined. Once the length was measured, it is desired to determine the effective width of pad for the various configurations. From Figures 33 and 34 it can be interpolated from the curves.

The circumferential stresses presented more consistent data because the circumferential loading over the gage was a line load with more or less constant stress over the length of contact. The axial stress varies with the location of the fin over the gage because of the fin restraint. Also during the test the gages which were sensitive to the hoop strains were more closely monitored while the axial gages which were of secondary concern were not monitored as closely and the readings as a result had a larger degree of scatter.

The reference fin design for the SPF reactor is a 1/8 inch wide fin which has been ground flat in the zone of contact. The contact width as determined from the Bijlaard analysis for this fin is approximately .038 inches on the .020 wall (.0190 over the strain gage) thickness tube and .025 for the thicker .032 wall (.0327 over the strain gage)

thickness tube. This sounds logical because of the higher flexibility of the thinner tube. The deflection in a cylinder is inversely proportional to thickness to the 9/4 power (Reference 4) so that the contact width is approximated by the equation

$$w = A + B t^{-9/4} R^{3/4}$$

and solving for A & B

$$w = .01957 + 2.47 \times 10^{-6} t^{-9/4} (R/.375)^{3/4}$$

For the reference fuel element of the SPF reactor which is a .027 wall thickness cylinder, the contact width is thus interpolated to be .0268.

The SPF reactor core is divided into two zones with slightly different element radii. The wall thickness of .027 inch and other parameters for the two zones are the same. For design purposes then the following parametric stresses are determined.

Inner Zone	Outer Zone
R = .340	R = .3475
t = .027	t = .027
$\gamma = R/t = 12.6$	$\gamma = R/t = 12.85$
$C_1 = .0134$	$C_1 = .0134$
$\beta_1 = .0134/.340 = .0394$	$\beta_1 = .0134/.3475 = .0306$
$C_2 = .16$	$C_2 = .16$
$\beta_2 = .16/.340 = .47$	$\beta_2 = .16/.34 = .47$

β is computed from the equations on page 37 for each of the individual load components, and the stress magnitudes calculated from the M_ϕ/P , M_x/P , $N_\phi/P/R$, and $N_x/P/R$ values read from the design charts which utilize the above calculated β .

The stresses are then equal to

$$\sigma_{\text{inside}} = \frac{6M}{t^2} - P$$

$$\sigma_{\text{outside}} = \frac{-6M}{t^2} - \frac{P}{t}$$

for M_ϕ calculation

$$\beta = \left[1 - \frac{4}{3} \left(1 - \frac{\beta_1}{\beta_2} \right) (1 - .88) \right] \sqrt{\beta_1 \beta_2}$$

$$\begin{aligned}
 &= .1168 \text{ (inner zone)} \\
 &= .1156 \text{ (outer zone)} \\
 M_{\phi}/P &= .16 \text{ (inner zone)} \\
 M_{\phi}/P &= .16 \text{ (outer zone)} \\
 \sigma_{M\phi} &= 1320 \text{ psi/pound Inner Zone} \\
 \sigma_{M\phi} &= 1320 \text{ psi/pound Outer Zone}
 \end{aligned}$$

For the M_x calculation

$$\begin{aligned}
 \beta &= \left[1 - 4/3 \left(1 - \frac{\beta_1}{\beta_2} \right) (1 - 1.25) \right] \sqrt{\beta_1 \beta_2} \\
 &= .1762 \text{ (Inner Zone)} \\
 &= .1745 \text{ (Outer Zone)} \\
 M_{x/P} &= .082 \text{ (Inner Zone)} \\
 &= .084 \text{ (Outer Zone)} \\
 \sigma_{M_x} &= 675 \text{ (Inner Zone)} \\
 &= 690 \text{ (Outer Zone)}
 \end{aligned}$$

For the N_{ϕ} calculation

$$\begin{aligned}
 \beta &= \left[1 - 4/3 \left(1 - \frac{\beta_1}{\beta_2} \right) (1 - 1.48) \right] \sqrt{\beta_1 \beta_2} \\
 &= .2131 \text{ (Inner Zone)} \\
 &= .2111 \text{ (Outer Zone)} \\
 N_{\phi}/P/R &= 1.76 \text{ (Inner Zone)} \\
 &= 1.76 \text{ (Outer Zone)} \\
 \sigma_{N\phi/} &= 191 \text{ (Inner Zone)} \\
 &= 187 \text{ (Outer Zone)}
 \end{aligned}$$

For the N_x calculation

$$\begin{aligned}
 \beta &= \left[1 - 4/3 \left(1 - \frac{\beta_1}{\beta_2} \right) (1 - 1.20) \right] \sqrt{\beta_1 \beta_2} \\
 &= .1682 \text{ (Inner Zone)} \\
 &= .1665 \text{ (Outer Zone)} \\
 N_{x/P/R} &= 2.03 \text{ (Inner Zone)} \\
 &= 2.03 \text{ (Outer Zone)} \\
 \sigma_x &= 220 \\
 \sigma_x &= 216
 \end{aligned}$$

Combining the values

<u>Zone</u>	<u>Inner Zone</u>	<u>Outer Zone</u>
$\sigma_{M\phi}$	1320	1320
σ_{Mx}	675	690
$\sigma_{N\phi}$	191	187
σ_{Nx}	220	216
σ_{ϕ} Inner Surface	1129	1133
σ_x Inner Surface	455	474
σ_{ϕ} Outer Surface	-1511	-1507
σ_x Outer Surface	-895	-906

These are the values of stress per unit load due to the local effects of the finned element contacting a unfinned element. They do not include bending or any other effect.

VII. TUBE-ON-TUBE TEST

In addition to the fin on tube tests already described, smooth tube-on-tube configurations were investigated using the same fixture and the .032 inch and .020 inch strain gaged tubes. Some previous work done using .010 inch wall tubes is described in references 3 and 4. Efforts were made to match the bow configurations used in those earlier tests.

A smooth tube with wall thickness of .032 inch and an outside diameter of .75 inch was the loading mandrel. Mandrels were inserted in the ends of the tube to facilitate bending and are shown in Figure 35. Turnbuckles were used on both tubes to produce the initial radii, no adjustments were made while the loads were applied. Relationships used to determine the amount of tube displacement necessary for the desired radii are the following:

$$\begin{aligned}\delta &= R(1 - \cos \theta) \\ \theta &= 8''/R \\ \cos \theta &= 1 - \theta^2/2 \\ \delta &= R(\theta^2/2) \\ &= 32''/R\end{aligned}$$

For a 740 inch radius the displacement is .043 inch and for a 450 inch radius the displacement is .071 inch. Figure 36 shows the nomenclature used to describe the test conditions.

Figure 36

FIGURE 35
PLAIN TUBE MANDREL

Results are given in Appendix E, page 114.. Contact lengths were observed for the various loadings and normalized to a load intensity value (load/contact length). This allows comparison of the stresses calculated for the .032 inch and .020 inch wall tubes. The stresses were primarily in the circumferential direction as shown by Figures D1 and D2, both the maximum and circumferential stress values have not been corrected for the bending. The contact lengths did not vary significantly between the runs even though R_1 for the instrumented .032 inch wall tube was initially 740 inches and for the .020 inch wall tube was infinity.

VIII. REFERENCES

1. P. P. Bijlaard, "Stresses from Radial Loads on Cylindrical Pressure Vessels", Welding Research Supplement, 1954.
2. K. R. Wichman, A. G. Hopper and J. L. Mershon, "Local Stresses in Spherical and Cylindrical Shells due to External Loadings," Welding Research Bulletin 107, December 1968.
3. TR-759-240-001, "S8DR Fuel Element Contact Stress Test Report", J. Brammer, 1-15-72.
4. TI-568-200-002, "S8DR Cladding Contact Intensity and Stress", W. H. Dauterman, 1-27-71.
6. Perry & Lessner, The Strain Gage Primer, McGraw Hill Book Co., Second Edition.

APPENDIX A

DIMENSIONAL AND MATERIAL DATA

Atomics International
North American Rockwell

CERTIFICATE OF TESTS	O.D. 3/4"	WALL .035"	MFG. PACIFIC TUBE				MT. NO. M 3805					
MECH PROPERTIES	TENSILE PSI 89,040		YIELD PSI 48,645		ELONG. IN 2" 58.0		GRAIN 6+		HARDNESS HARD'ILITY			
CHEM ANALYSIS	C .074	MN 1.59	P .016	S .016	SI .48	NI 8.93	CR 19.16	MO .24	TI	CB/TA	CU .13	
TESTS PERFORMED	DECARB	MAG	CORR X	EMB	HYDRO X	FLARE X	FLAT X	CRUSH	FLANG	R. FLAT	SURFACE X	BEND X

DESCRIPTION & REMARKS:

SMLS 304 A/C TBG CD MIL-T-8504/A EXCEPT PARA 3.7.2.

CERTIFICATE OF TESTS	O.D. 3/4"	WALL .020"	MFG. PACIFIC TUBE				MT. NO. M-3925					
MECH PROPERTIES	TENSILE PSI 94,980		YIELD PSI 57,815		ELONG. IN 2" 57.0		GRAIN 6+		HARDNESS HARD'ILITY			
CHEM ANALYSIS	C .068	MN 1.70	P .018	S .015	SI .47	NI 9.40	CR 18.72	MO .20	TI	CB/TA	CU .13	
TESTS PERFORMED	DECARB	MAG	CORR X	EMB	HYDRO X	FLARE X	FLAT X	CRUSH	FLANG	R. FLAT	SURFACE X	BEND X

DESCRIPTION & REMARKS:

SMLS 304 A/C TBG CD MIL-T-8504/A AMEND.1

CERTIFICATE OF TESTS

WALL THICKNESS SCAN OF NOMINAL 20 MIL WALL TUBE

Ø	(INCHES)								
	AXIAL POSITIONS								
	1	2	3	4	5	6	7	8	9
0	.0198	.0198	.0198	.0200	.0194	.0202	.0200	.0200	.0199
30	.0196	.0199	.0199	.0202	.0192	.0199	.0202	.0199	.0194
60	.0195	.0200	.0200	.0200	.0196	.0197	.0199	.0196	.0190
90	.0193	.0198	.0200	.0198	.0185	.0194	0 —	0 —	0 —
120	.0192	.0198	.0199	.0195	0 —	.0190	.0195	.0194	0 —
150	.0190	.0197	.0198	0 —	0 —	0 —	0 —	.0192	0 —
180	.0191	.0197	.0197	.0195	.0199	.0196	0 —	0 —	.0192
210	.0192	.0196	.0196	.0194	.0196	.0194	.0193	.0192	0 —
240	.0192	.0195	.0194	.0192	.0194	.0192	.0192	0 —	.0193
270	.0193	.0194	.0193	.0190	0 —	.0190	.0190	.0190	0 —
300	.0194	.0192	.0191	.0190	.0190	.0189	.0189	0 —	0 —
330	.0195	.0190	.0190	.0191	.0189	.0189	.0189	.0192	.0193

ARROW INDICATES LOCATION OF ROSETTES 5 THROUGH 10

OUTSIDE DIAMETER OF NOMINAL 20 MIL WALL TUBE (INCHES)

Ø	1	2	3	4	5	6	7	8	9
0	.7534	.7535	.7532	.7534	.7533	.7531	.7530	.7534	.7534
30	.7527	.7533	.7532	.7531	.7531	.7530	.7533	.7534	.7535
60	.7531	.7532	.7536	.7537	.7536	.7534	.7534	.7531	.7534
90	.7532	.7536	.7535	.7534	.7536	.7532	.7530	.7533	.7533
120	.7530	.7534	.7533	.7534	.7534	.7534	.7533	.7531	.7535
150	.7533	.7537	.7536	.7532	.7533	.7532	.7535	.7533	.7534
180	.7534	.7534	.7534	.7545	.7532	.7534	.7533	.7534	.7535

WALL THICKNESS SCAN OF NOMINAL 32 MIL WALL TUBE									
(INCHES)									
Ø	AXIAL POSITIONS								
	1	2	3	4	5	6	7	8	9
0	.0325	.0324	.0325	.0327	.0327	.0328	.0325	.0326	.0326
30	.0326	.0325	.0326	.0328	.0326	.0323	.0326	.0327	.0328
60	.0327	.0326	.0327	.0326	0 —	0 —	.0326	.0326	.0328
90	.0378	.0326	.0326	.0325	.0325	.0327	.0324	.0327	0 —
120	.0327	.0325	.0324	.0324	.0324	.0326	.0324	.0324	.0326
150	.0326	.0324	.0324	.0324	.0322	.0325	.0323	0 —	.0325
180	.0325	.0323	.0323	.0326	0 —	.0324	.0322	0 —	.0324
210	.0323	.0322	.0322	.0322	.0322	.0322	.0323	.0324	.0323
240	.0322	.0323	.0323	.0323	.0323	0 —	.0322	.0325	.0324
270	.0323	.0324	.0324	.0324	.0322	.0321	.0323	.0324	.0322
300	.0324	.0324	.0325	.0325	0 —	.0320	.0324	.0325	0 —
330	.0325	.0323	.0324	.0324	.0323	0 —	.0324	.0326	0 —
360	.0324	.0322	.0322	.0323	.0324	.0320	.0326	.0326	0 —
ARROW INDICATES POSITION OF ROSETTES 5 THROUGH 10									

OUTSIDE DIAMETER OF NOMINAL 32 MIL WALL TUBE (INCHES)									
Ø	1	2	3	4	5	6	7	8	9
0	.7475	.7475	.7477	.7475	.7474	.7473	.7474	.7475	.7475
30	.7473	.7475	.7475	.7475	.7475	.7474	.7475	.7476	.7473
60	.7475	.7475	.7474	.7476	.7475	.7475	.7474	.7475	.7475
90	.7474	.7476	.7475	.7475	.7476	.7474	.7475	.7474	.7474
120	.7475	.7475	.7475	.7476	.7476	.7474	.7475	.7475	.7475
150	.7473	.7474	.7475	.7474	.7475	.7475	.7474	.7474	.7473
180	.7475	.7575	.7476	.7474	.7474	.7475	.7474	.7475	.7475

APPENDIX B

CALIBRATION

CALIBRATIONS

The strain gage calibration was accomplished differently on the two tubes. The first was the 0.032 wall thickness tube. The gages were hooked up to record strains on the same equipment with the same wire lengths that were used in the actual test. The tube was laid on knife edge pivot points twenty inches apart and a known load applied hung from an approximately 0.050 diameter wire looped over the tube (Figure B1). The gages away from the influence of the local effects of the load were read and compared to the known bending stress. In reviewing the readings on the 0.032 nominal wall thickness tube all the gages were within the required accuracy and no correction factors were applied to the data.

For the 0.020 wall thickness tube the calibration procedure was different. The calibration set up is shown in Figure B2. A hole was bored in each end of the tube prior to the application of the strain gages. For calibration a pin was set through the hole and a tension force applied. Review of this data indicated that there was a minor amount of bending within the tube which was accounted for. The data was plotted and the load deflection curve indicated that Youngs Modulus was 30.0×10^6 psi. Reviewing the data for the 0.020 tube the following correction factors on the readings were established. It was felt that all but gages 1, 8 and 10 were within the accuracy required and just these three required correction factors for use in the data reduction. These are presented in Table B-1.

Calibration Setup for the .032 Inch
Wall Tube (Bending Calibration Test)

Figure B1

Figure B2

PREPARED BY:	ATOMICS INTERNATIONAL A DIVISION OF NORTH AMERICAN ROCKWELL CORPORATION	PAGE NO. 54 OF
CHECKED BY:		TR-653-240-006
DATE:	.020" TUBE CORRECTIONS	REPORT NO.
		MODEL NO.

STRAIN GAGE CORRECTION FACTORS		
TABLE B1		
GAGE #	CORRECTION FACTOR POS2	CORRECTION FACTOR POS1
1	.93	.94
2		
3		
4	1.0	
5	.98	1.03
6	.99	.99
7	.995	.98
8	1.042	1.04
9	1.032	1.016
10	.96	.953

APPENDIX C

DATA ANALYSIS METHOD

TR

653-240-006

2 OF

DATA ANALYSIS

The strain gages were configured in a rosette with all legs 60° apart. The one leg (1) was along the axis of the tube while the other two legs (2, 3) were 60° from the axis. The reduction of the data from the individual strain readings is taken from Reference 6 and repeated here for clarity.

FIG. C1 Geometrical representation of strain gage deformation due to a strain in the x direction.

FIG. C2 Geometrical representation of strain gage deformation due to a strain in the y direction.

FIG. C3 Geometrical representation of strain gage deformation due to a shearing strain.

Figures C1, C2, and C3 indicate the gage reading in strain for a unit strain in the X, Y or γ directions respectively. Referring to Figure C1 for the strain in the X direction

$$\epsilon_x = \frac{\delta_x}{X}$$

The strain measured by the gage is

$$\epsilon_\phi = \frac{\delta_L}{L} = \frac{\delta_x \cos^2 \phi}{X} = \epsilon_x \cos^2 \phi$$

where

$$L = \frac{X}{\cos \phi}$$

$$\delta_L = \delta_x \cos \phi$$

Referring to Figure C2 for action in the Y Direction

$$\epsilon_Y = \frac{\delta_Y}{Y}$$

and the strain measured by the gages is

$$\epsilon_\phi = \frac{\delta_L}{L} = \epsilon_Y \sin^2 \phi$$

$$L = \frac{Y}{\sin \phi}$$

$$\delta_L = \delta_Y \sin \phi$$

Referring to Figure C3 for the measurements due to a shearing strain in the X - Y plane.

$$\epsilon_\phi = \frac{\delta_L}{L} = \gamma_{XY} \sin \phi \cos \phi$$

where $L = \frac{Y}{\sin \phi}$

and

$$\delta_L = \delta_X \cos \phi = Y \tan \gamma_{XY} \cos \phi \approx Y \gamma_{XY} \cos \phi$$

Since all gages read all strains simultaneously

$$\epsilon_\phi = \epsilon_X \cos^2 \phi + \epsilon_Y \sin^2 \phi + \gamma_{XY} \sin \phi \cos \phi$$

now from Mohr's circle

$$\epsilon_1 = \frac{\epsilon_X + \epsilon_Y}{2} + \frac{\epsilon_X - \epsilon_Y}{2} \cos 2\phi_1 + \frac{\gamma_{XY}}{2} \sin 2\phi_1$$

$$\epsilon_2 = \frac{\epsilon_X + \epsilon_Y}{2} + \frac{\epsilon_X - \epsilon_Y}{2} \cos 2\phi_2 + \frac{\gamma_{XY}}{2} \sin 2\phi_2$$

$$\epsilon_3 = \frac{\epsilon_X + \epsilon_Y}{2} + \frac{\epsilon_X - \epsilon_Y}{2} \cos 2\phi_3 + \frac{\gamma_{XY}}{2} \sin 2\phi_3$$

Where ϵ_1 , ϵ_2 , and ϵ_3 are the strain gage readings and ϵ_X , ϵ_Y , and γ_{XY} are respectively the strains along the tube axis, the circumferential strains and the shearing strains.

For this test the gages were identified as shown below

Then solving for ϵ_X , ϵ_Y and γ_{XY}

$$\epsilon_X = \epsilon_1$$

$$\epsilon_Y = \frac{-\epsilon_1 + 2\epsilon_2 + 2\epsilon_3}{3}$$

$$\gamma_{XY} = \frac{2(\epsilon_2 - \epsilon_3)}{\sqrt{3}}$$

The circumferential, longitudinal and shear stresses are:

$$\sigma_X = \frac{E}{1-\mu^2} \left[\epsilon_X + \mu \epsilon_Y \right]$$

$$\sigma_Y = \frac{E}{1-\mu^2} \left[\epsilon_Y + \mu \epsilon_X \right]$$

$$\tau_{XY} = \frac{E}{2(1+\mu)} \gamma_{XY}$$

and the principal stresses and angles are:

$$\sigma_{\text{Max}} = \frac{\sigma_X + \sigma_Y}{2} + \sqrt{\left(\frac{\sigma_X - \sigma_Y}{2}\right)^2 + \tau_{XY}^2}$$

$$\sigma_{\text{Min}} = \frac{\sigma_X + \sigma_Y}{2} - \sqrt{\left(\frac{\sigma_X - \sigma_Y}{2}\right)^2 + \tau_{XY}^2}$$

$$\tau_{\text{Max}} = \frac{\sigma_{\text{Max}} - \sigma_{\text{Min}}}{2}$$

$$\tan^2 \phi_P = \frac{\tau_{XY}}{\left(\frac{\sigma_X - \sigma_Y}{2}\right)}$$

$$\phi = \frac{1}{2} \tan^{-1} \frac{2 \tau_{XY}}{\sigma_X - \sigma_Y}$$

To facilitate data reduction a computer program was written for the GE timeshare where the readings would be input and the total circumferential stress, total longitudinal stress, shear stress, and principal stresses were calculated. The listing is given below.

STR1

```

1 PRINT"THIS ANALYSIS IS FOR THE DATA REDUCTION OF THE STRAIN"
3 PRINT"GAGE, STRAIN 1, STRAIN 2, STRAIN 3. IT WILL COMPUTE AND"
4 PRINT"PRINT      MAX. STRESS      MIN STRESS      MAX SHEAR"
5 PRINT"      X STRESS      Y STRESS      X-Y SHEAR"
6 PRINT " STRAIN DATA IS INPUT IN LINE 500"
7 PRINT"INPUT THE NUMBER OF RUNS INCLUDING ZEROS - MAX IS 25"
8 DIM S(30,25),T(25),C(30,25)
9 LET K=-2.08 — 7.2 FACTOR
10 LET E=30
11 LET M=.3
13 GO TO 300
14 LET E=30


---


18 PRINT
20 PRINT E1,E2,E3
21 LET E1=E1*-K
22 LET E2=-E2*K
23 LET E3=-E3*K
30 LET S4=(E2+E1+E3)/3
40 LET S5=((E1-S4)+2+((E2-E3)/1.732)+2)
41 LET S5=SQR(S5)/(1+M)
50 LET S1=(S4/(1-M)+S5)*E
60 LET S2=(S4/(1-M)-S5)*E
70 LET S3=S5*E
72 LET A=.5*ATN((1/1.732)*(E2-E3)/(E1-(E1+E2+E3)/3))
73 LET A=57.3*A
74 LET G=G+1
75 IF G=11 THEN 140
80 PRINT G,S1,S2,S3,A
90 LET X3=1.15*(E2-E3)*E/(2*(1+M))
95 LET X1=E1
100 LET X2=(-E1+2*E2+2*E3)/3
105 LET X4=(E/(1-M+2))*(X1+M*X2)
110 LET X5=(E/(1-M+2))*(X2+M*X1)
120 PRINT G,X4,X5,X3
123 PRINT


---


125 IF L=31 THEN 440
130 GO TO 400


---


140 LET G=1
141 GO TO 80


---


300 INPUT N
312 FOR J=1 TO N
314 FOR I=1 TO 30
316 READ S(I,J)
318 NEXT I
319 PRINT S(1,J)
320 NEXT J


---


322 FOR I = 1 TO N
324 READ T(I)
325 PRINT T(I)

```

*CALCULATION OF
STRESS FOR
LINE RECEIPT*

SIGNALS END OF 10 GAGES

*LEADS & PRINTS
STRAINS FOR N
RUNS INCLUDING ZERO*

LEADS TIME FOR EACH RUN

STR1 CONTINUED

326 NEXT I

335 FOR J=2 TO (N-1)

340 FOR I=1 TO 30

350 LET C(I,J)=S(I,1)-(S(I,1)-S(I,N))*(T(J)-T(1))/(T(N)-T(1))

360 LET S(I,J)=S(I,J)-C(I,J)

370 NEXT I

380 NEXT J

*CORRECTS DATA FOR ZERO
READINGS & TIME ZERO SHIFT*

385 LET J=2

390 LET L=1

400 LET E1=S(L,J)

410 LET E2=S(L+1,J)

420 LET E3=S(L+2,J)

425 LET L=L+3

430 GO TO 18

*RENAMES STRAIN COMPONENTS
FOR STRESS CALCS**GO TO 18*

440 LET J=J+1

445 IF J=N THEN 1400

450 GO TO 390

STEPS TO NEXT RUN

1400 END

Of course this represents the total strain and stress in the member including the effects of beam bending. First it is necessary to determine the loading and then to determine the induced bending stresses. The load was applied in increments of five pounds with previously weighed bottles of shot weight. However, because of the friction in the system not all of this weight was effective in loading or unloading. Because of this the gage on the tension side of the beam, for the .032 inch nominal wall thickness tube, was used as a load cell. It is assumed that the axial stress determination was not influenced by the local stress because of its location opposite the contact point. For the second series of tests on the .020 inch nominal wall tube two gages were applied on the tension side as a check on each other and more care was taken to reduce friction. The two gages did not exactly predict the same loading and the applied load was used on the effects of friction, which had been minimized, were ignored for the 15 pound load. The average stresses for gages 1 and 4 were determined taking into account the 0.94 factor for gage 1 from the calibration data. Then for the lesser loads where friction may become important, the load was determined by reducing the average stress of gages 1 and 4 at load to the average stress of gages 1 and 4 at 15 pound load times the fifteen pounds. After the load was determined the bending stress at the gage was found.

$$\sigma = \frac{MC}{I} = \frac{PAB}{L\pi R^2 t}$$

This was also computerized to compute the bending stress from the known load and then subtract it from the previously computed total stresses. The circumferential, axial and shear are then computed for the load effects only.

STCØRR

```
1 PRINT " THIS PRØGRAM CØRRECTS THE DATA FØR BENDING "
3 PRINT " DATA IS  GAGE NØ, LØAD, SX, SY, SXY "
5 PRINT " ØUTPUT IS  GAGE NØ, LØAD, SX, SY, SXY "
7 PRINT "          BENDING STRESS, SX LØCAL "
9 PRINT "          MAX STRESS, TAN 2 THETA, ANGLE ØF PRIN STRESS"
11 PRINT
13 READ G, P, S1, S2, S3
20 IF G = 5 THEN 28
22 IF G = 6 THEN 33
24 IF G = 7 THEN 37
26 IF G = 8 THEN 41
28 LET A = 0
31 GØ TØ 50
33 LET A = .375
35 GØ TØ 50
37 LET A = .75
39 GØ TØ 50
41 LET A = 1.125
43 GØ TØ 50
50 LET S4 = P* (110.25 - A^2)/.1723
52 LET S5 = S1 + S4
54 LET S6 = (S5+S2)/2 + SQR(((S5-S2)/2)^2 + S3^2)
56 LET Q = -2*S3/(S5 - S2)
58 LET R = ATN(Q)
60 LET U = R/2 * 57.3
62 PRINT G,P,S1,S2,S3
64 PRINT S4,S5
66 PRINT S6,Q,U
68 GØ TØ 11
100 DATA 5, 15, 23286.4, 29328.7, -13293.3
102 DATA 5, 8.4, 14047.1, 16586.9, -8190.04
103 DATA 5, 4.85, 7848.78, 9636.68, -5457.73
104 DATA 5, 15, 20878.2, 28507, -16471.7
106 DATA 5, 15, 20648.5, 27926.8, -16115.5
108 DATA 5, 4.99, 7805.67, 10467.1, -5832.9
110 DATA 5, 8.38, 12876.5, 17363.1, -9450.62
112 DATA 5, 4.85, 5919.39, 13427.6, -6331.67
114 DATA 5, 8.29, 8650.15, 24338.3, -10316.9,
116 DATA 5, 15, 19015.9, 40764.2, -18950
118 DATA 5, 15, 18958.1, 40561.4, -18874.8
120 DATA 5, 15, -3211.62, 37773.3, -4003.75
130 DATA 6, 15, 21602.6, 25823.9, -7286.4
132 DATA 6, 15, 14582.4, 44896.4, -2456.4
134 DATA 6, 15, 15310.7, 45793.8, -9522
136 DATA 6, 15, 2800.99, 46904.3, -3670.8
138 DATA 6, 15, 9624.9, 47992, -7231.2
140 DATA 7, 15, 17489.7, 41408.6, 16983.5
142 DATA 7, 15, 17113.3, 41588.1, 16973.8
144 DATA 7, 9.4, 8460.33, 28924.9, 10329
146 DATA 7, 9.5, 901.4, 29690.5, 3446.33
```


STCØRR CONTINUED

148 DATA 7, 4.7, 1427.94, 16236.6, 2271.44
149 DATA 7, 4.7, 4831.68, 15500, 4999.72
150 DATA 5, 4.85, -2015.79, 9541.94, -639.74
152 DATA 5, 8.6, -3317.87, 16885.3, -1306.88,
154 DATA 5, 15, -6686.53, 25875.3, -2085.35
155 DATA 5, 8.7, -3870.48, 15546.1, -1125.19,
156 DATA 5, 4.9, -2309.12, 8503.23, -604.06
160 DATA 5, 4.7, -2626.29, 7862.6, -269.87
161 DATA 5, 4.64, 7213.71, 7359.49, -512.133
162 DATA 5, 4.7, -2468.57, 7718.1, -561.2
163 DATA 5, 8.2, -4306.29, 11345.3, -913.87
164 DATA 5, 8.33, -4274.86, 11301.6, -880.133
165 DATA 5, 8.54, -4224, 15086.2, -578.267
166 DATA 5, 5, -2592, 7815.11, -530.533
170 DATA 6, 15, 9341.3, 37776.2, -8852.97
171 DATA 6, 15, -84.49, 34618.6, -2784.65
172 DATA 6, 15, 9104.33, 37129.2, -7578.17
173 DATA 6, 15, -3211.56, 33085.4, -1689.55
174 DATA 6, 15, 6596.16, 38504.4, -7626.25
175 DATA 6, 4.8, 2358.59, 13948.2, -3247.24
176 DATA 6, 8.3, 8372.99, 20460, -6164.56
177 DATA 6, 15, 6489.85, 39948.9, -7603.22
180 DATA 6, 15, 6690.45, 44973.5, -8603.24
182 DATA 6, 15, 4297.67, 35957.6, -4859.64
184 DATA 6, 15, 18740.7, 35013, -7196.69
190 DATA 7, 4.74, 477.6, 14483.9, -717.6
191 DATA 7, 9.46, 6319.5, 27033.4, -1959.6
192 DATA 7, 15, 3036.4, 41890.9, -3450
193 DATA 7, 15, 4532.33, 41271.7, -3698.4
194 DATA 7, 15, 10458.2, 43659.6, -5713.2
195 DATA 7, 15, 2447.74, 39536.2, -3367.2
196 DATA 7, 9.53, 753.9, 27326.6, -1959.6
197 DATA 7, 4.6, 2368.7, 14941.2, -1794
200 DATA 5, 4.7, -2478.72, 5947.41, -257.13
201 DATA 5, 8.2, -4415.6, 9521.67, -428.9
202 DATA 5, 15, -7748.06, 16565.9, -1153.99
210 DATA 5, 15, 11418.1, 33200.2, -11488.9
211 DATA 5, 15, 6025.83, 23899.7, -8657.27
212 DATA 5, 15, 7844.63, 24556.9, -12494.3
220 DATA 6, 15, 7336.31, 26170.5, -743.527
221 DATA 6, 15, -605.7, 26190.4, -2631.2
222 DATA 6, 15, 366.6, 32864.8, -2711.2
223 DATA 6, 15, -460, 30843, -2952.05
224 DATA 6, 15, -781.682, 29024.4, -96.6644
225 DATA 6, 15, 5372.12, 27528.7, -1192.79
226 DATA 6, 15, 10917.7, 29213.9, -5549.84
227 DATA 6, 8.2, 2590.15, 18374.7, -2052.49
228 DATA 6, 4.7, 3377.15, 10619.3, -1621.5
230 DATA 7, 4.66, 1684.68, 8089.58, -297.756
231 DATA 7, 4.64, -898.189, 10249.7, 156.14,

STCØRR CONTINUED

232 DATA 7, 4.63, 1116.93, 11133.8, -476.288,
233 DATA 7, 9.4, 8829.44, 17896.3, -1713.54
234 DATA 7, 9.4, -1546.6, 22193.5, 83.639
235 DATA 7, 9.5, 9005.3, 19290, 1742.97
236 DATA 7, 15, -5046.15, 29224, 310.891
237 DATA 7, 15, -1680.81, 32314.5, -16.23
238 DATA 7, 15, -1079.76, 33877.9, -1196.64
239 DATA 7, 15, -3199.77, 32415.5, 986.859
240 DATA 7, 15, 2867.27, 29951.5, -3811.43
700 END

APPENDIX D

DETAIL DATA

AXIAL STRESS UNDER LOAD.072 FIN (ROUNDED)PARALLEL RODS1532 TUBE

FIGURE D7

CIRCUMFERENTIAL STRESSUNDER LOAD1072 FIN GROUND FLAT (.005)1032 DIA

FIGURE D8

STRESS - KSI

○ RUN 71 OVER GAGE 8

△ RUN 72 OVER GAGE 7

○ RUN 73 OVER GAGE 6

◇ RUN 74 OVER GAGE 5

▽ RUN 75 OVER GAGE 5

◇ RUN 72 OVER GAGE 4

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○ TEST 48 OVER GAGE 5	▽ TEST 51 OVER GAGE 8
△ TEST 49 OVER GAGE 6	▽ TEST 66 OVER GAGE 6
□ TEST 50 OVER GAGE 7	○ TEST 65 OVER GAGE 5
○ TEST 51 OVER GAGE 9	▽ TEST 172 OVER GAGE 5
△ RUN 111 OVER GAGE 8	○ RUN 171 OVER GAGE 6
△ RUN 109 OVER GAGE 6	○ RUN 170 OVER GAGE 5

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○ GAGE B RUN 20
△ GAGE 7 RUN 20
□ GAGE 6 RUN 20
◇ GAGE 5 RUN 20

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○ RUN 129	5	○ 135	6	○ 141	7
△ 136	5 + 1/4	○ 136	6 + 1/4	△ 142	7 + 1/4
▽ 137	5 + 1/2	▽ 137	6 + 1/2	○ 143	7 + 1/2
△ 138	5 + 3/4	○ 138	6 + 3/4	△ 144	7 + 3/4
○ 139	5 + 1/2	○ 139	6 + 1/2	▽ 145	7 + 1/2
○ 134	5 + 1/4	△ 140	6 + 1/4

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MAXIMUM STRESS

.020" WALL

.072" STANDARD FIN

PARALLEL

FIGURE D27

LEGEND

○	RW 175	GAGE #5
△	176	#5
□	199	#6
▽	201	#6
◇	209	#7

STRESS - KSI

3470 PSI/TENON

LOAD - LB

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MAXIMUM STRESS
0.20" WALL
0.72" FLAT FIN
PARALLEL

LEGEND		
	RUN	CASE
O	178	5
□	179	15
V	195	26
D	210	37

FIGURE D-30

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.072" FLAT FIN
AXIAL STRESS

FIGURE D 32

LEGEND		
	RUN	GAGE
○	178	#5
□	179	#5
▽	195	#6
◇	210	#7

STRESS (KSI)

44
42
40
38
36
34
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0

LOAD (LB)

5

10

15

400 PSI / Pound

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CIRCUMFERENTIAL STRESS
 .020" WALL TUBE
 .125 STANDARD FIN
 PARALLEL

LEGEND

RUN	GAGE
O	182 #5
□	183 #5
△	196 #6
D	205 #7

FIGURE D 34

STRESS (KSI)

LOAD (LB)

R450 R51/LB

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EUGENE DIETZGEN CO.
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125" FLAT FIN AXIAL STRESS

0.020" WALL
PARALLEL

LEGEND		
ROW	GAGE	
O	181	*5
△	197	*6
D	207	*7

FIGURE D 38

STRESS (KSI)

46
44
42
40
38
36
34
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0

MAXIMUM STRESS
0.30" WALL
160" STANDARD FRI
PARALLEL

LEGEND

RUN	GAGE
○ 187	#5
□ 188	#5
△ 198	#6
D 206	#7

FIGURE D 39

2590 PSI

CIRCUMFERENTIAL STRESS
0.90" WALL TUBE
160" STANDARD FIN
PARALLEL

LEGEND

	ROW	GAGE
O	182	#5
D	188	#5
Δ	198	#6
D	206	#7

FIGURE D-40

STRESS KSI

2550 PSI/LB

LOAD - POUNDS

.160" STANDARD FIN
AXIAL STRESS

.020" WALL
PARALLEL

LEGEND	
RUN	GAGE
○ 187	#5
□ 188	#5
△ 198	#6
◇ 206	#7

FIGURE D-41

MAXIMUM STRESS
0.20" WALL

4" RADIUS SCALLOPED .125" FAN
PARALLEL

LEGEND		
RUN	GAGE	
O	191	#5
U	200	#6
D	208	#7

EDGE IN
CONTACT

FIGURE D42

2000 PSI/POUND

CIRCUMFERENTIAL STRESS
6.00" WALL TUBE
.4" RADIUS SCALLOP ON .125" STD. FIN
0.000000

LEGEND	
SYM	GAGE
○	192 #5
□	200 #6
▷	203 #7

EDGE CONTACT

FIGURE D 43

STRESS KSI

1620 PSI / LB

LOAD - POUNDS

4" RADIUS SCALLOPED 125" FIN
AXIAL STRESS

0.020" WALL
PARALLEL

LEGEND

	RUN	GAGE
O	192	#5
D	200	#6
D	208	#7

FIGURE D-44

900 PSI / POUND

0.020" WALL
PARALLEL

40
38
36
34
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0
-2

0

5

10

15

MAXIMUM STRESS
0.020" WALL

5" RADIUS SCALLOPED .125" FIN
PARALLEL

LEGEND

	RUN	GAGE
○	187	#5
□	190	#5
▽	191	#5
△	199	#6
D	202	#7

FIGURE D45

CIRCUMFERENTIAL STRESS
.020" WALL TUBE
.5" RADIOS SCALLOPED .125" STD FIN
PARALLEL

LEGEND

	ROW	GAGE
O	189	#5
E	190	#5
F	191	#5
A	199	#6
D	209	#7

FIGURE D 46

STRESS
KSI

2050 PSI/LB

APPENDIX E

TUBE-ON-TUBE DETAIL DATA

TUBE ON TUBE

LOAD INTENSITY VS. CIRCUMFERENTIAL STRESS

FIGURE E1

STRESS (KSI) BENDING INCLUDED

9
8
7
6
5
4
3
2
1
0

CIRCUMFERENTIAL

LEGEND			
R ₁ IN	TUBE WALL	R ₁	R ₂
Δ 202	.020 IN.	∞	450 IN.
□ 78	.032 IN.	740 IN.	450 IN.
○ 79	.032 IN.	740 IN.	450 IN.

LOAD INTENSITY (POUNDS/INCH)

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TUBE ON TUBE LOAD INTENSITY VS. MAXIMUM STRESS

FIGURE E2

MAXIMUM STRESS (KSI) BENDING INCLUDED

20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
0

LOAD INTENSITY (POUNDS/INCH)

LEGEND			
RUN	TUBE WALL	R ₁	R ₂
△ 202	.020 IN.	∞	450 IN.
□ 78	.032 IN.	740 IN.	450 IN.
○ 79	.032 IN.	740 IN.	450 IN.

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STRESS (KSI)

TUBE ON TUBE

LOAD VS. AVERAGE STRESS

FIGURE 13

20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
0

5

10

15

20

25

30

LOAD (LB)

LEGEND

RUN	TUBE WALL	R ₁	R ₂
Δ 2.2	.020 IN.	∞	450 IN.
□ 1.8	.032 IN.	740 IN.	450 IN.
○ 7.7	.032 IN.	740 IN.	450 IN.

TUBE ON TUBE CONTACT LENGTH VS. APPLIED LOAD

FIGURE E4

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