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FINAL REPORT

STRESS ANALYSIS OF MFTF-B GETTER SYSTEM BELLOWS

by

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prepared for

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INTRODUCTION

Background

The Lawrence Livermore National Laboratory (LLNL) is constructing a Tandem Mirror Fusion Test Facility (MFTF-B) at LLNL scheduled for completion in 1984. The MFTF-B is an upgrade and extension at the Mirror Fusion Test Facility (MFTF), a single mirror facility, to a tandem mirror configuration.

The MFTF-B design includes a retractable getter system. Eight getter assemblies are planned (4 in each end plug). The getter system will be the sublimated-titanium type now in use on Beta II and TMX, and planned for the MFTF. Electrically heated Ti wires are mounted on a telescoping insertion mechanism and, between machine shots (pulses), are extended into the chamber in the vicinity of inward-facing water-cooled magnet liners. Heating current is applied, and a Ti coating is sublimated onto the liners, thus providing a clean surface that will trap deuterium. During the shots, the sublimators must be withdrawn because they will intrude into plasma and diagnostic space.

Each of the getter assemblies will be mounted on the exterior of the vacuum vessel. Bellows are used to keep essentially all of the mechanism isolated from the vessel vacuum. As the system is used prior to every machine pulse, it requires high reliability (100,000 operational cycles over a 10-year machine life).

The bellows are being designed and constructed by Metal-Fab Corporation, Ormond Beach, Florida. The bellows come in two sizes (8.25" O.D. and 14" O.D.). The smaller of the two bellows has been qualified by testing up to 94,000 cycles by empirically adjusting details of the bellow design (geometry and thickness). The process required 12 different test samples and took over a one-year period to accomplish. The bellows consistently failed in the inside diameter weld heat-affected zone. This was contrary to the predicted failure location by Metal-Fab's stress analysis (Reference 1) performed using Battelle

Memorial Laboratory's bellow's design code (Reference 2). This computer analysis showed an area of the I.D. Toroidal to be the highest stressed. A cycle life was expected of greater than 100,000 cycles. Metal-Fab's stress analysis (Reference 3) of the large bellows does suggest the weld area at the inside diameter to be the highest stressed. And the analysis results again suggests a cycle life of greater than 100,000 cycles.

Tooling for the qualification of the large bellows is nearly complete with production of the first test samples imminent. SMA was requested* to perform a stress analysis of both bellow designs to assist the qualification of the large bellow design and assure that the overall design schedule of the MFTF-B experiment is met. This report documents our analysis.

Study Objectives

The objectives of the analysis are two-fold: First, to attempt to explain why the smaller bellow design consistently failed during the qualification testing in the inside diameter weld heat-treated zone rather than in other locations as suggested by previous analyses. And second, to use the comparison of the stress analysis results from the two designs to form the basis for guiding possible alternate configurations (shape and thickness) in conjunction with testing to develop a qualified large bellow design.

Approach

Stress analysis was performed using the finite element computer program MODSAP (Reference 4). The analysis was conducted on the LLNL Q-time-sharing computer system. The analysis involved the following steps:

(1) The development of a detailed finite element model of the small bellow design (several convolutions).

(2) A detailed stress analysis of the smaller bellow design. Loading consisted of an externally applied atmospheric pressure and a superimposed prescribed stroking displacement.

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* L. Valby of LLNL MFTF-B Mechanical Systems was technical contact for this study.

(3) The examination of stress analysis results of the smaller bellow design to reconcile calculated stresses with the observed behavior during the qualification tests. Focus was on highest calculated stresses in the observed failure region with particular interest on the magnitude of the mean stress and range of alternating stresses (i.e., those from fully extended to fully collapsed positions). Degradation affects of material strength and fatigue life characteristics due to welding and heat treatment were also considered. However, potential stress concentrations in the weld region were not modeled.

(4) The repeat of steps (1) and (2) for the larger bellow design.

(5) A comparison of calculated stress levels of the two bellow designs. This comparison suggested a need for several additional analyses of the larger bellow design using alternate (larger) material thickness to reduce the highest stresses to optimize the large bellow design configuration.

(6) The development of conclusions.

STRESS ANALYSIS

Finite Element Models

The exact geometry (included in the Appendix) of each bellow design was modeled. Table 1 presents some of the most significant basic design information. Both designs are of the nested ripple type. After press forming, the male and female leafs are fusion welded at their inside and outside diameters. Characteristics of such bellows are very low axial spring rates, a long stroke capability and having poor resistance to differential pressure.

Six complete convolutions (i.e., a male and female pair) of each bellow design were modeled to assure the elimination of boundary condition effects on calculated stresses. (See Figures 1 and 2). Each convolution was modeled with 52 shell type elements giving a total of 312 elements and 313 nodes for each design. Each bellow convolution consists of (flat) conical shell portions at both the inside and outside diameter regions and 3 large toroidal shell sections between these conical shells. Either 4 or 5 elements were used to model each of the larger toroidal sections and conical sections near the outside diameter. Both the male and female conical sections near the inside diameter (i.e., where failure occurred during testing of the 8.25" O.D. bellows) were modeled with 9 elements each.

The free pitch of the bellows is approximately the same (0.1582" and 0.1628" for the small and large designs, respectively). At the inside diameter the conical portions are tilted with respect to the bellow axis at approximately 57° and having an included angle of about 0.8°. At the outside diameter the conical portions are approximately perpendicular to the bellow axis with a 2° included angle between the leaves.

Bellow material for both the small and large design is A347 stainless steel with a basic stock thickness of 0.009". However, the thickness was varied in the analysis to evaluate its sensitivity on the calculated stresses. (Both 0.009" and 0.007" were considered for the small bellows and 0.009", 0.0105", and 0.012" for the large bellows).

Analysis Results

Sixteen different analyses were performed. These are listed in Table 2 with terms defined in Figure 1. All cases included an external pressure of 14.7 psi. The pitch of the bellows was varied from a fully collapsed position of 0.0296" to a fully extended position of 0.1704". Nominal free (or manufactured) pitch positions of 0.1582" and 0.1628" were used for the small and large bellows, respectively. Case S4 is similar to S1, but with an assumed free pitch of 0.1440" instead of the 0.1582" nominal value.

Cases S1-S4 and L1-L3 reflect the actual thickness (i.e., 0.009 inches) used for the small bellows and the thickness planned for the larger bellows. Cases S5-S7 and L4-L9 assumed different thickness values.

The most significant analysis results are summarized in Figures 3 through 7 and Tables 3 and 4. The figures present plots of the meridional moment in the bellows versus radial distance. Note, a positive moment tends to spread the bellows apart at its inside diameter and compress the bellows at its outside diameter. Tables 3 and 4 summarizes the calculated maximum stresses at both the inside and outside diameters of the bellows.

DISCUSSION OF RESULTS

Fatigue failure is one of the most common types of failures in bellows. In normal service, the life of a bellows is determined by the cumulative effect of pressure and deflection (i.e., stroking) stresses (strains) to which it is subjected. Fatigue life can be improved by reducing stress concentrations at points of geometrical discontinuity, and reducing material variations, residual stresses and weld heat-affected zone material.

In contrast to formed bellows, the maximum pressure and deflection stresses in welded bellows of the more standard* designs always occur near the root and crown welds. This is extremely undesirable since it means that the maximum stresses occur in a heat-affected zone and also in a section which represents a possible source of stress concentration.

To reduce this undesirable condition nested ripple welded bellows which have tilted bellow flats with respect to the axis of the bellows, such as that used on the MFTF-B getter system, are often used. By reducing the stresses near welds, so that the maximum stresses occur away from the weld areas and in an area where the metal has properties of the original sheet material, the fatigue life of welded bellows is significantly improved.

Unfortunately, there is no <u>completely</u> satisfactory correlation between fatigue life and the theoretically predicted maximum strain for welded bellows made of Type A347 stainless steel (Reference 1). Strains predicted appear to be satisfactorily accurate. However, there is considerable variation in both fatigue life and failure location for bellows subjected to the same maximum strain range. Thus, fatigue life variation often appears to be the result of manufacturing variations associated with the welding process. The variability of fatigue life resulting from manufacturing variations must be experimentally determined for each manufacturer's process. To obtain the longest fatigue life it is desirable that the bellows be in compression as much as possible. Properly designed bellows with tilted edges are expected to experience fatigue failure in the parent material rather than in the weld areas.

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* <u>Standard</u> bellow designs normally have bellow flats at their root and crown which are perpendicular to the axis of the bellows.

In spite of the above we did perform a stress analysis and compared calculated stresses and performed a fatigue evaluation of the several design options in an attempt to provide guidance in the design of the larger bellows.

Analysis effort focussed on the calculation of meridional stresses near weld areas (crown and root of bellow convolutions . Eleven of the twelve small bellow specimens tested to failure todate . ied through the weld heat affected zone (References 9-11). Figure 8 shows the typical observed failure, at inside diameter, of the small bellows during qualification tests. Membrane stresses, in general, were quite small. Thus, meridional bending stresses were used as "yardstick" for evaluations. Note, although calculated stresses are larger at the crown of the bellows than at the root of bellows, the stress state is always one of compression which is believed to explain why failures did not occur there but at the root of the bellows. The calculated stresses in the region of observed failure shows that both the small and large bellows will operate under conditions of tensile mean-stress (S_M)*. Such a condition is known to reduce fatigue-life (Reference 6).

Figures 3 through 7 show the variation of the meridional bending moment across both the male and female leaves of a typical bellows convolution. (The meridional bending stress can be calculated by simply taking six times the moment and dividing by the thickness squared). Tables 3 and 4 present the calculated meridional bending stresses at the root and crown of the bellows. Shown are maximum stresses in the fully collapsed (stroke = 0.0296") and the fully extended (stroke = 0.1704") positions, the mean stress (S_M), and alternating stress amplitude (S_A)**. We used the parameters S_M and S_A combined with fatigue life-cycle characteristics of the bellows material below to predict the fatigue performance.

Since essentially all of the tested specimens failed through the weld heat affected zone (HAZ), for purposes of our fatigue evaluations we are using

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* $S_M = \frac{S_{max} + S_{min}}{2} = \frac{S_{Fully\ Extended} + S_{Fully\ Collapsed}}{2}$

** $S_A = \frac{S_{max} - S_{min}}{2} = \frac{S_{Fully\ Extended} - S_{Fully\ Collapsed}}{2}$

material design data representative of the HAZ or fully-annealed Type 347 stainless steel. Available information (References 12 and 13) on fully reversed (i.e., $S_M = 0$) strain-controlled fatigue behavior of this alloy is presented in Figure 9. This figure shows the number of cycles to failure as a function of both stress amplitude (S_A) and total strain-amplitude ($\Delta\varepsilon_t$). Data for three different temperature conditions (21°C, 200°C, and 350°C) as well as a design-curve based on A.S.M.E. Boiler and Pressure Vessel Code procedures are shown. Note, the design-curve reflects a factor of safety of approximately 20 on mean cycles to failure or a factor of about 2.0 on S_A. The ASME design-curve attempts to account for stress concentrations, surface flaws, etc. when assessing fatigue performance. The raw test data shown is developed under ideal conditions and with specimens free of such affects.

Figure 9 can be used to forecast fatigue life for stress states of zero mean stress (i.e., $S_M = 0$). Our stress analysis results indicate a significant non-zero mean (tensile) stress in the weld HAZ of both small and large bellow designs. Figure 10 along with Figure 9 can be used to determine the combined effect of mean stress (S_M) and alternating stress (S_A) on fatigue performance.

Figure 10, which is called a Goodman diagram, is a plot of S_A as the ordinate and S_M as the abscissa. Attention should be drawn to the line AB that connects a value on the ordinate and abscissa called the "limit of elastic behavior" which is the highest stress-amplitude the material can support without yielding even after many cycles of loading. Hence, the "limit of elastic behavior" is the dynamic yield-strength after the material shakes down to stable behavior (i.e., has undergone whatever strain-hardening or strain-softening that occurs for the particular material, initial conditions of material, temperature, and the rate of cyclic loading). Since annealed austenitic stainless steels cycled near room temperature at rates below ultrasonic frequencies will strain-harden (Reference 9) in the absence of detailed dynamic yield-strength data for this material, we assume the value to be the higher of the tensile yield-strength at room-temperature, approximately 30 KSI*, or the endurance limit, which from Figure 9 is about 35 KSI at 10^7 cycles. Regardless of the initial conditions under which any test is started, the "true" conditions after shake-down must fall inside the triangle OAB or on the ordinate above point A. Any test that is begun at conditions outside this region (OAB) would have a maximum stress greater than the yield strength (30 KSI) causing yielding to reduce the mean stress to a value that either lies on AB or on the ordinate at point A. Any stress pair, S_M and S_A, falling within the triangle OAB should never exhibit fatigue failure. Any stress pair falling outside OAB will exhibit a finite fatigue life.

Plotted on Figure 10 is also the stress conditions for each of the cases considered. All cases indicated a finite fatigue life.

In the absence of any actual fatigue data for a specific design the ASME curve is a reasonable choice for predicting fatigue performance. However, for our case we do have at least one data point - the small bellows failed at 9.2×10^4 cycles under the following predicted stress state in the observed failure location: $S_M = 20.5$ KSI and $S_A = 17.4$ KSI. We have chosen to use this information to normalize the raw test data in Figure 9, so that the normalized curve will better reflect the actual observed performance of the small bellows for prediction purposes. To do so, first use Figure 10 to estimate the alternating stress (S_A) corresponding to a zero mean stress (S_M) for the small bellows. The result is ~ 23 KSI. Hence, one data point to be used in Figure 9 is $S_A = 23$ KSI, cycles to failure $\approx 9.2 \times 10^4$. Plotting this point on Figure 9 and using a constant factor to shift the raw data fatigue curve leads to the dashed curve. For evaluation purposes,

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* Normally the yield-strength for annealed Type 347 stainless steel at room temperature is considered to be about 42 KSI. For this design we chose a relatively-low value of 30 KSI. The weld HAZ is a narrow, soft region trapped between two relatively-wide, harder regions - the weld-metal and the cold-worked sheet. Hence, yielding will occur first in the softer HAZ, and yielding will remain there as long as the maximum tensile stress in the fatigue-cycle does not exceed the smaller of the yield-strength of the weld (about 50 KSI) or the cold-worked sheet (about 65-85 KSI). Therefore, the development of failure is expected in the HAZ much before it might be expected in the sheet or weld-metal.

it is assumed that this design curve applies to both size bellows. To estimate the fatigue life of the large bellows, first obtain the alternating stress corresponding to a zero mean stress. From Figure 10, S_A = 35 KSI. Using this value of S_A and the design fatigue curve of Figure 9, an estimated fatigue life of the large bellows is $\sim$ 30,000 cycles.

Conclusions

As stated earlier in this section it is difficult to obtain a completely satisfactory correlation between fatigue life and theoretically predicted stress (strain) values in welded bellows. Therefore, one must be cautious about drawing precise quantitative conclusions from the analysis results. However, we do feel that the following conclusions can be made.

- The stress analysis indicates similar stress distributions across the leaves of both the small and large bellows. The fatigue life characteristics of the large bellows should not be radically different from that observed of the smaller bellows.
- The stress analysis predicted high stresses at the inside and outside diameters of the bellows -- the inside diameter was subjected to tensile stresses and the outside diameter, comprehensive stresses.
- High stresses of the type leading to fatigue failure were predicted at the point of observed failure.
- For the large bellows, the mean and alternating stresses are predicted to be greater than those in the small bellows.
- It is predicted that the large bellows will fail at a reduced number of cycles ($\sim$ 30,000).

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3. Stress Analysis LLL P/N 79-113586-0A, Metal-Fab Corporation dated April 17, 1980.

4. J.J. Johnson, MODSAP: A Modified Version of the Structural Analysis Program SAP IV for the Static and Dynamic Response of Linear and Localized Nonlinear Structures, General Atomic Company Report GA-A-14006 (Rev.) UC-77, June 1978.

5. L. Valby, Large Bellows (SK-5162), LLNL Engineering Note, dated April 1980.

6. L. Valby, Small Bellows (SK-5161), LLNL Engineering Note, dated April 1980.

7. SK-5161 Bellows Male/Female 2 Convolutions Titled ID Only [Geometry; Male 6 Part Model and Female 6 Part Model Dwg. No. SK-5174; Form Die, Nesting (Male and Female; Metal to Metal) Dwg. No. D01562], Metal-Fab Corporation, dated April 18, 1980.

8. SK-5162 Bellows Male/Female 2 Convolutions 14.000 x 12.000 Male 6 Part Model and Female 6 Part Model Dwg. No. SK-5174; Form Die, Nesting (Male and Female; Metal to Metal) Dwg. No. D-1562], Metal-Fab Corporation, dated May 25, 1981.

9. E.N.C. Dalder to L. Valby, "Interim Report No. 1 on MFTF-B Failed Small Diameter Getter Bellows Fatigue Test Articles," (Metal-Fab. Dwg. No. SK-5162), ENCD-81-312, June 11, 1981.

10. E. Dalder to L. Valby, "Interim Report No. 2 on MFTF-B Failed Small Diameter Getter Bellows Fatigue Test Articles (Metal-Fab. Dwg. No. SK-5162)," ENCD-81-355, July 17, 1981.

11. E. Dalder, Verbal Communication to L. Valby, October 11, 1981.

12. E.E. Baldwin, G.J. Sokol, and L.R. Coffin, Jr., "Cycle Strain Fatigue Studies on AISI Type 347 Stainless Steel," Proceedings ASTM, 57, 1957, pp. 567-86.

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15. R.W. Landgraf, "Cycle Stress-Strain Responses in Commercial Alloys" in "Work Hardening in Tension and Fatigue," A.W. Thompson (Editor), Metallurgical Society of A.I.M.E., 1975, pp. 240-259.

	Large Bellows	Small Bellows
O.D.	14.00"	8.25"
I.D.	12.00"	6.25"
Convolutions	710	710
Extended length	121"	121"
Compressed length	22"	22"
Stroke	99"	99"
Spring Rate	2.22 lbs/"/REV	2.04 lbs/"/REV
Stock Thickness	0.009"	0.009"
Material	A347 Stainless Steel	A347 Stainless Steel

Table 1. Bellows - Basic Design Information

Small Bellows (8.25" O.D.)

Case	External Pressure	Reaction	Pitch	Thickness
S1	14.7 psi	25.61 lbs.	0.1582"	0.009"
S2	14.7	26.74	0.1704	0.009
S3	14.7	13.78	0.0296	0.009
S4	14.7	24.31	0.1440	0.009
S5	14.7	26.03	0.1582	0.007
S6	14.7	26.70	0.1704	0.007
S7	14.7	19.00	0.0296	0.007

Large Bellows (14.00" O.D.)

Case	External Pressure	Reaction	Pitch	Thickness
L1	14.7 psi	46.04 lbs.	0.1628"	0.009"
L2	14.7	46.85	0.1704	0.009
L3	14.7	31.98	0.0296	0.009
L4	14.7	45.78	0.1628	0.0105
L5	14.7	46.95	0.1704	0.0105
L6	14.7	25.31	0.0296	0.0105
L7	14.7	45.59	0.1628	0.012
L8	14.7	47.23	0.1704	0.012
L9	14.7	16.96	0.0296	0.012

Table 2. Analysis Cases

Bellow Size	Fully Collapsed	Fully Extended	S_M Mean	S_A Alternating Amplitude
<u>Small</u>				
t = 0.007"	+ 21,000	+ 41,000	+ 31,000	10,000
t = 0.009"	+ 2,900	+ 37,600	+ 20,500	17,350
<u>Large</u>				
t = 0.009"	+ 12,600	+ 54,000	+ 33,300	20,700
t = 0.0105"	- 8,700	+ 45,700	+ 18,500	27,200
t = 0.012"	- 25,100	+ 39,200	+ 7,100	32,200

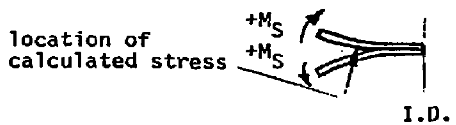


Table 3. Calculated Maximum Meridional Bending Stress (I.D.) - PSI

Bellow Size	Fully Collapsed	Fully Extended	Mean	S_A Alternating Amplitude
<u>Small</u>				
t = 0.007"	- 170,000	- 108,000	- 139,000	31,000
t = 0.009"	- 138,000	- 66,000	- 102,000	36,000
<u>Large</u>				
t = 0.009"	- 142,000	- 80,000	- 111,000	31,000
t = 0.0105"	- 129,000	- 59,000	- 94,000	35,000
t = 0.012"	- 123,000	- 45,000	- 84,000	39,000

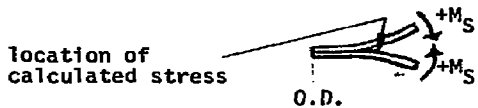
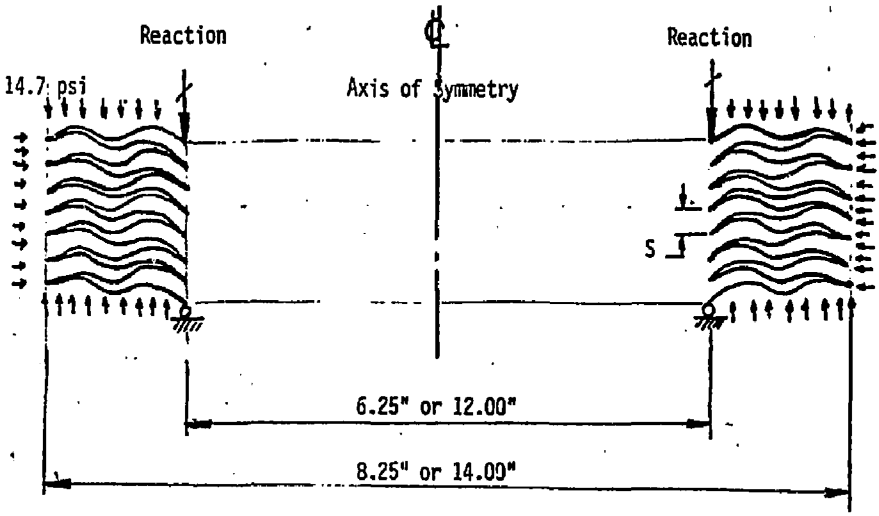


Table 4. Calculated Maximum Meridional Bending Stress (O.D.) - PSI



Pitch of Bellows(s)

- Fully Collapsed 0.02960"
- Free 0.15820" (Small Bellows)
  0.16280" (Large Bellows)
- Fully Extended 0.17040"

Figure 1. Loading and Stroke Information

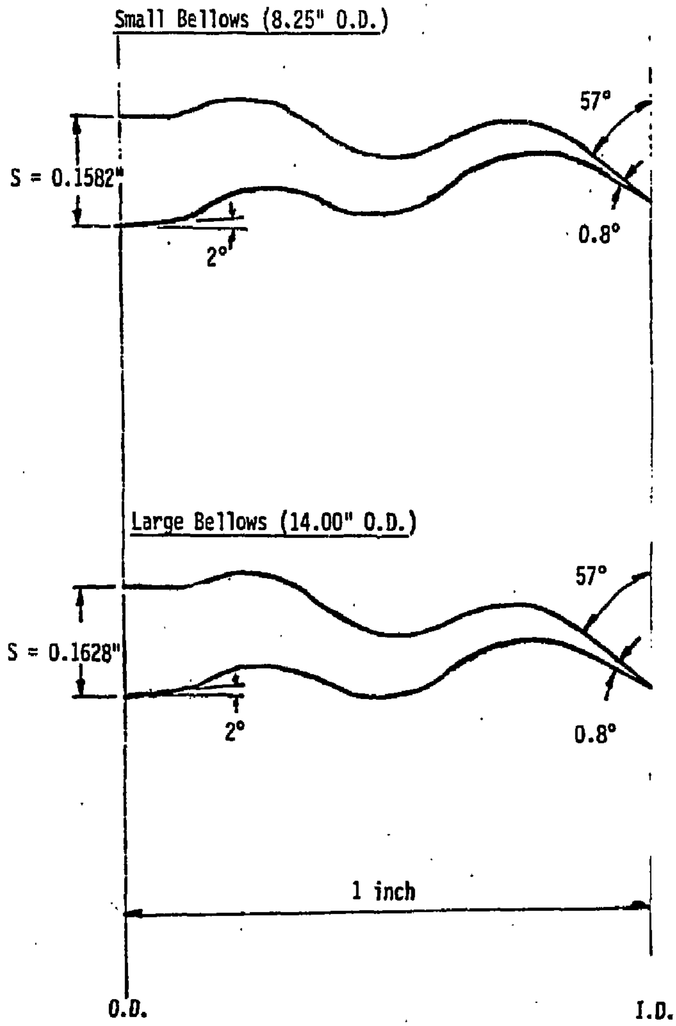


Figure 2. Bellows Geometry Modelled
(Single Convolution)

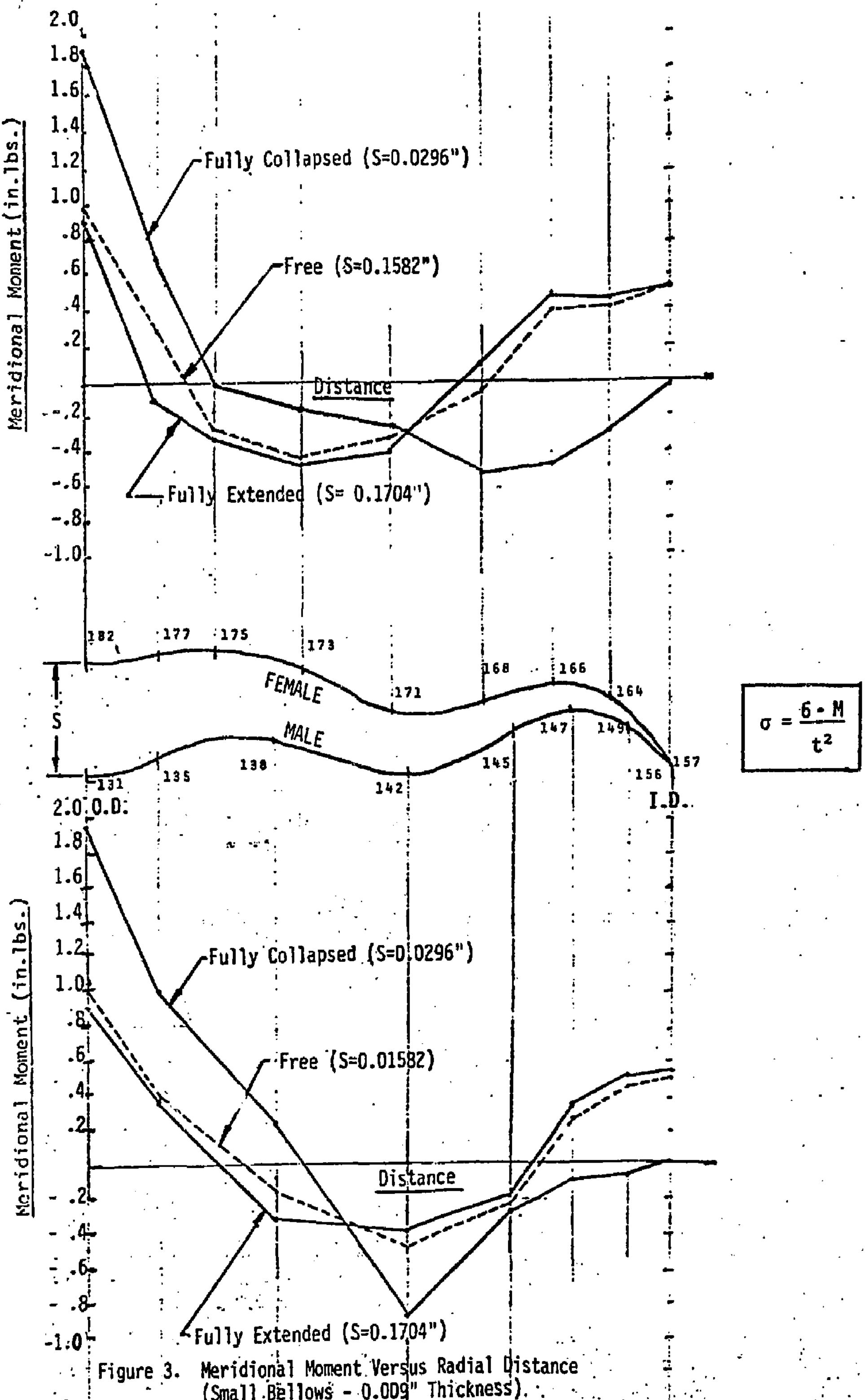


Figure 3. Meridional Moment Versus Radial Distance (Small Bellows - 0.009" Thickness).

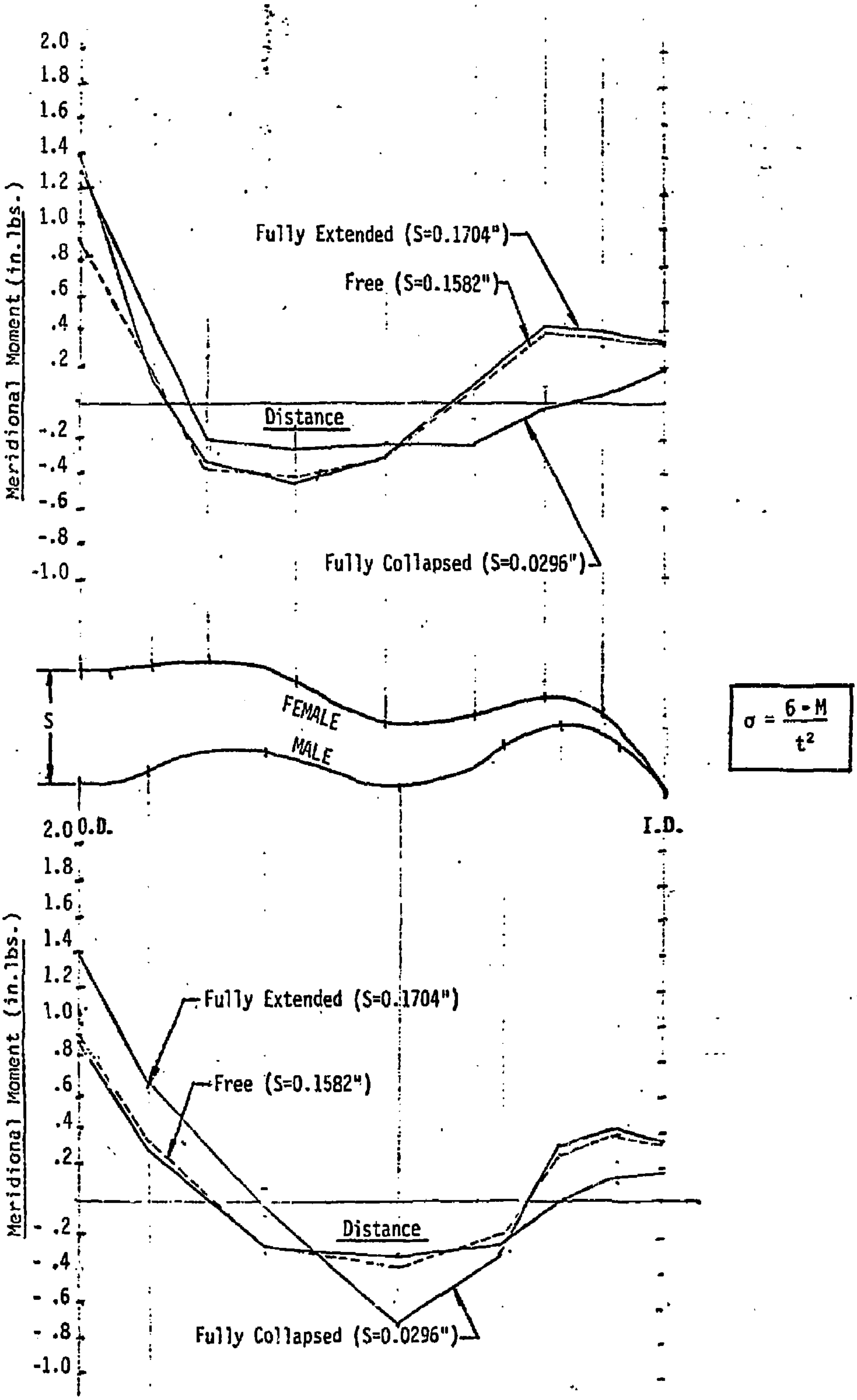


Figure 4. Meridional Moment Versus Radial Distance
(Small Bellows - 0.007" Thickness)

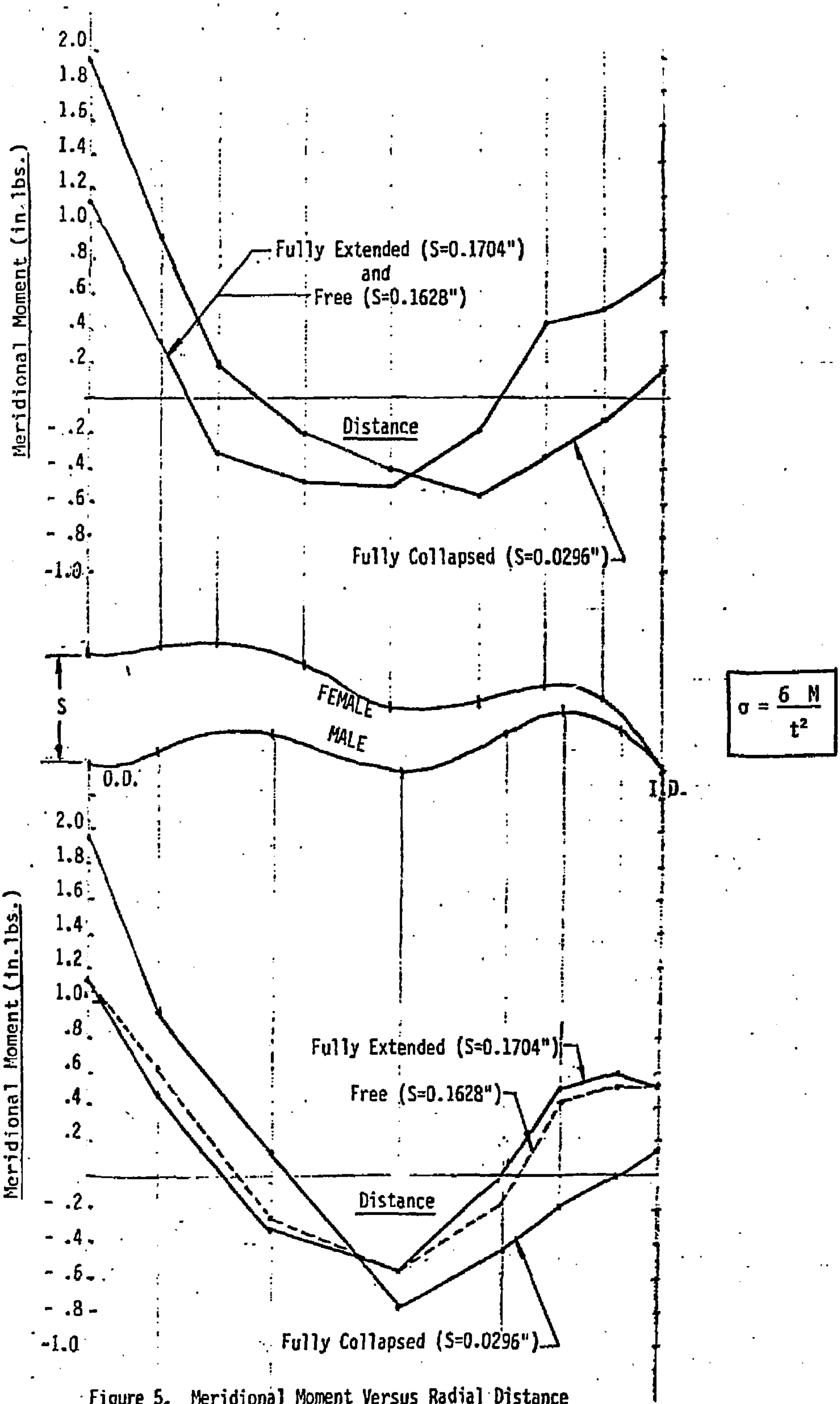
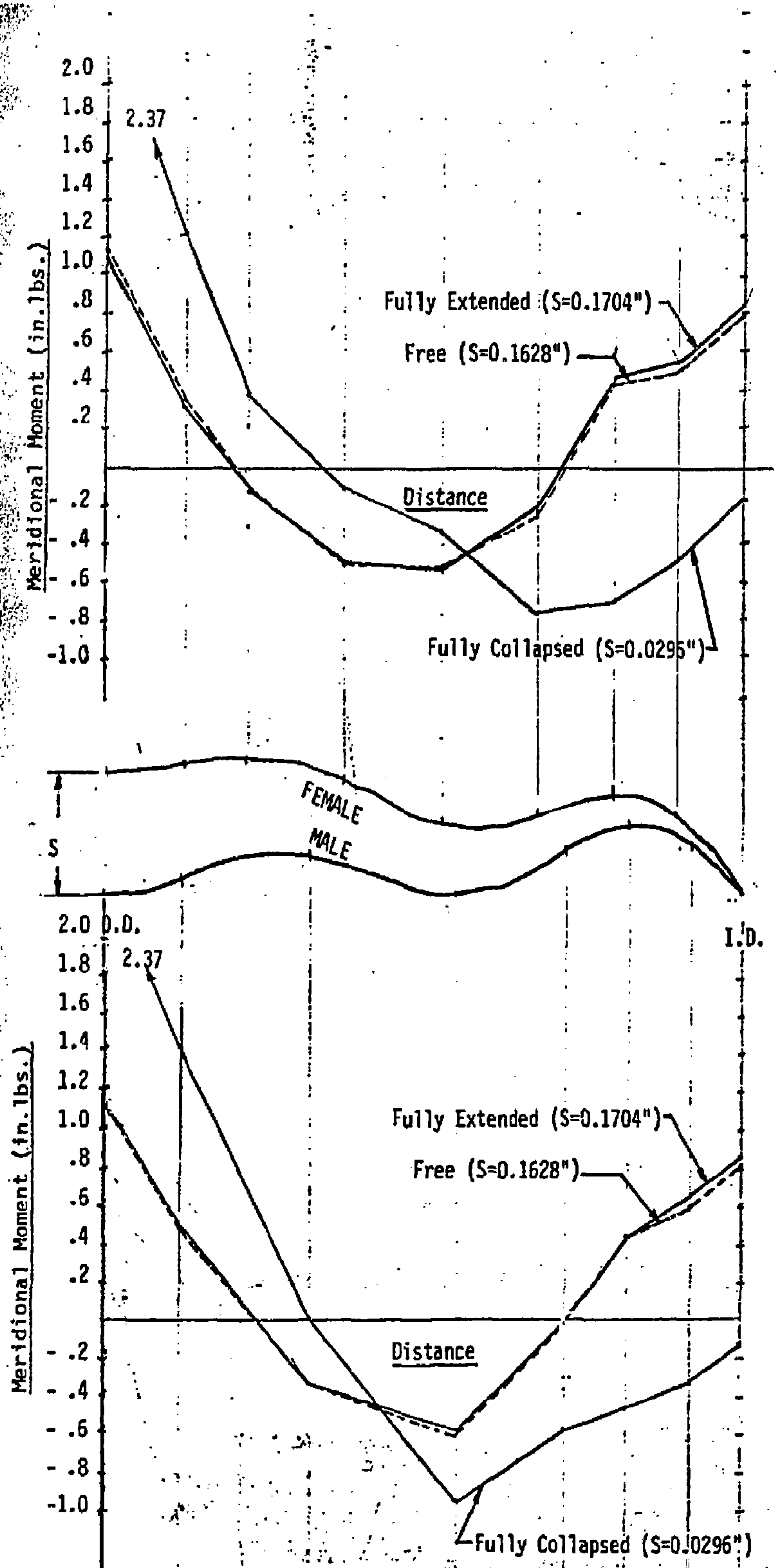


Figure 5. Meridional Moment Versus Radial Distance
(Large Bellows - 0.009" Thickness)



$$\sigma = \frac{6 \cdot M}{t^2}$$

Figure 6. Meridional Moment Versus Radial Distance (Large Bellows - 0.0105" Thickness)

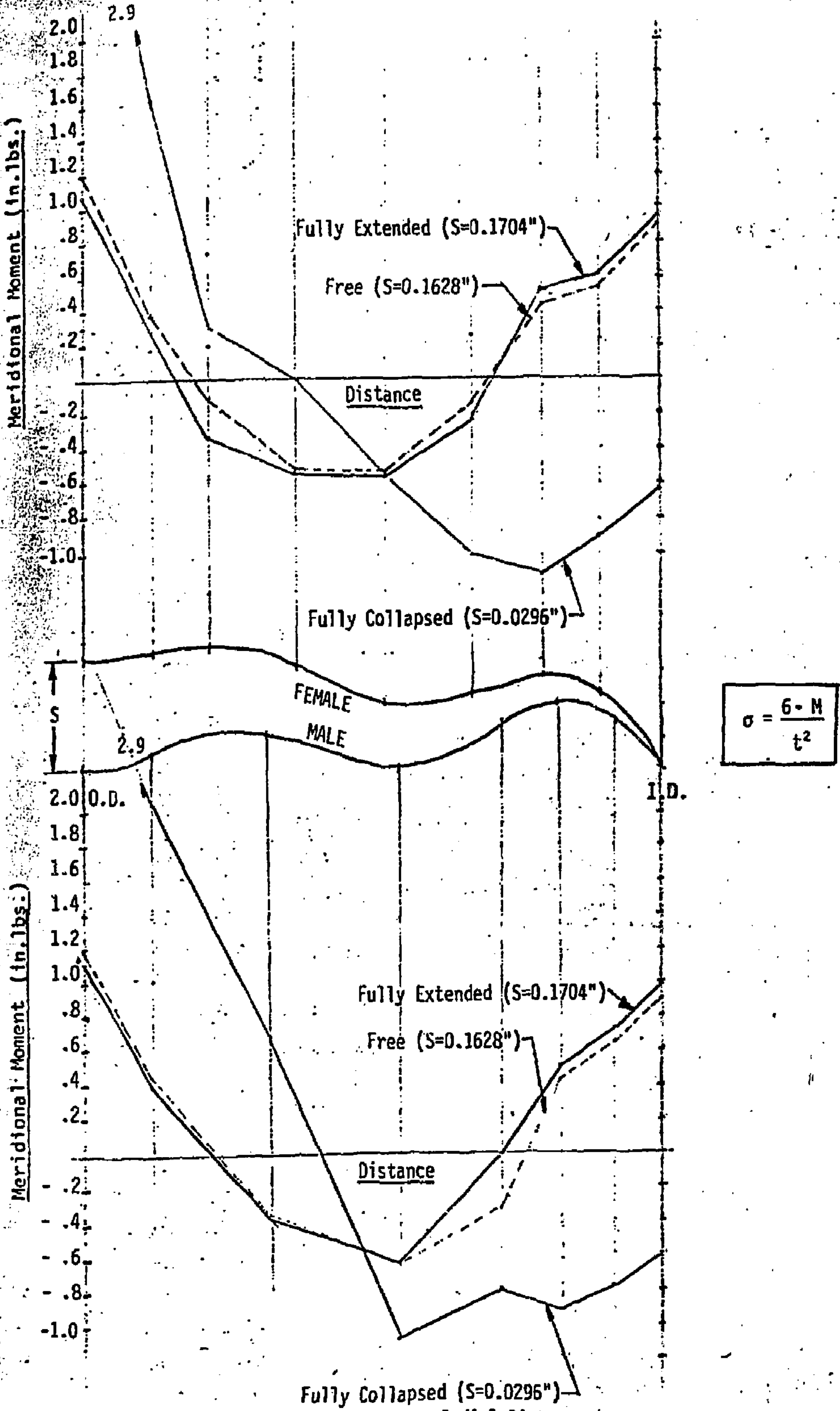


Figure 7. Meridional Moment Versus Radial Distance (Large Bellows - 0.012" Thickness)

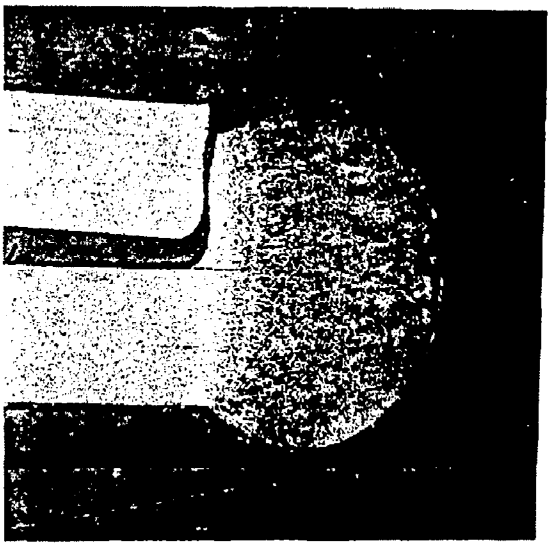


Figure 8. Typical Observed Failure (Inside Diameter) of Small Bellows During Qualification Tests.

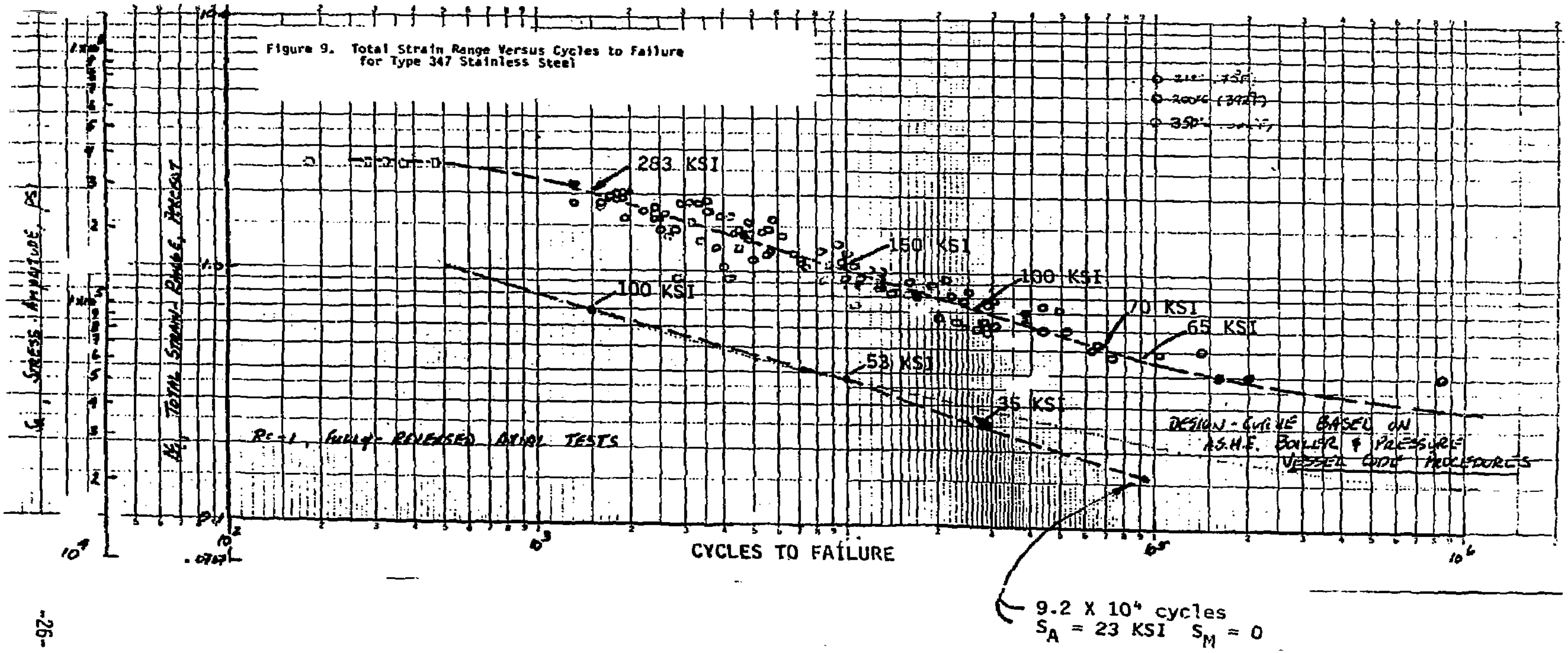
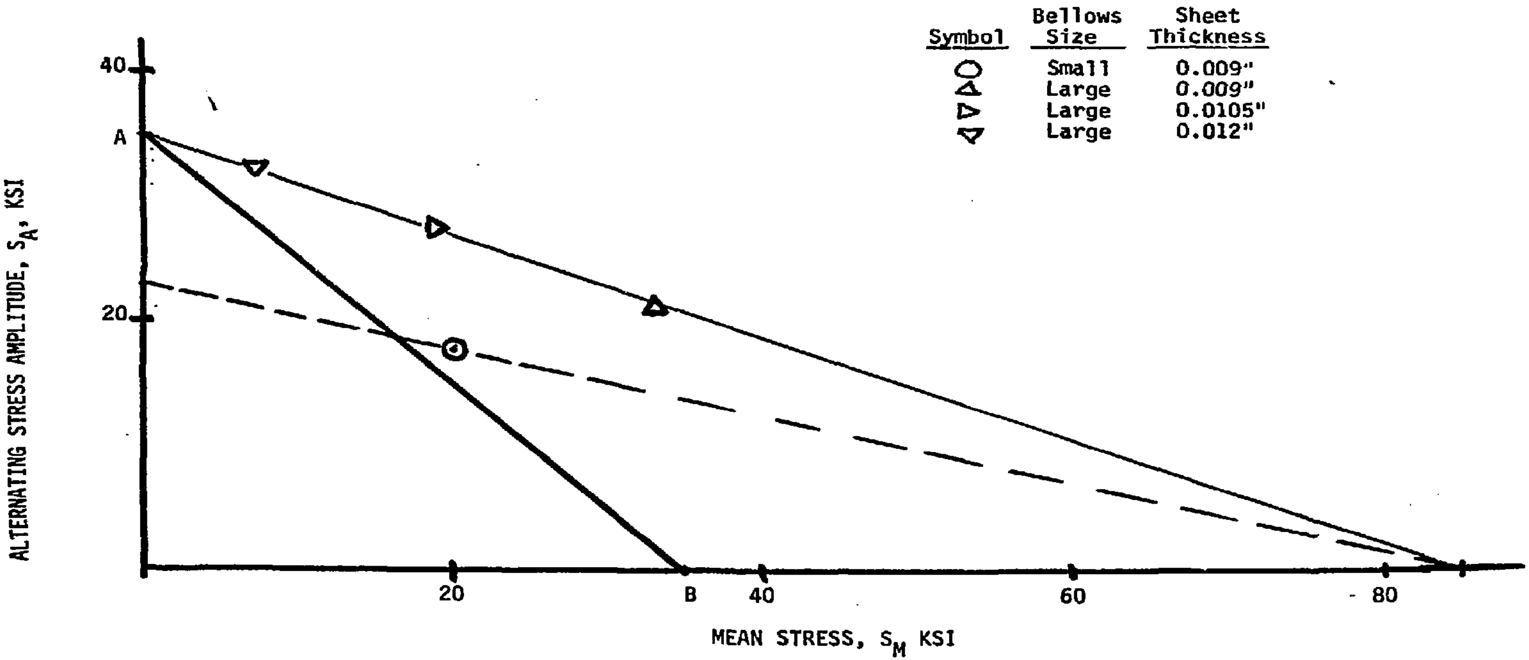


Figure 9. Total Strain Range Versus Cycles to Failure for Type 347 Stainless Steel

Figure 10. Comparison of Mean Stress and Alternating Stress Effects on Fatigue Performance of MFTF-B Getter Bellows



APPENDIX

Geometry and Forming Data

[illegible] BELLOWS FEMALE/MALE [illegible] CONVOLUTIONS [illegible] ONLY — From Metal Fab 5/22/81

METAL FAB

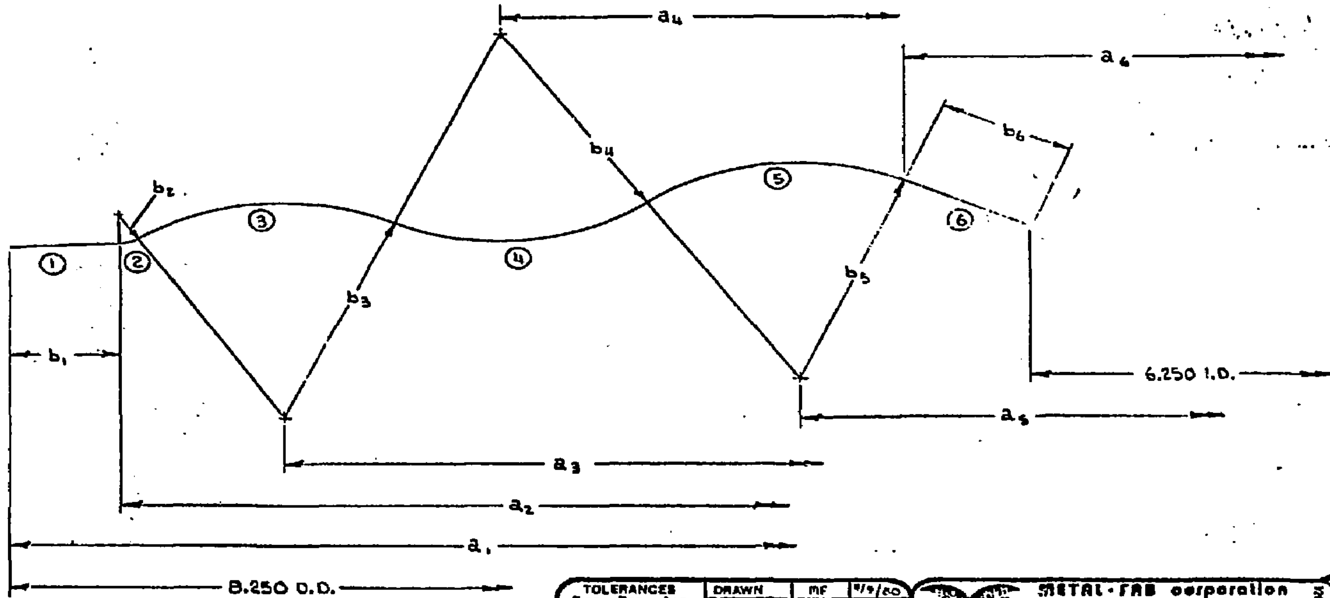
APRIL 18 1980

II

* GEOMETRY *

PART	TYPE	NO. OF SEGMENTS	COORDINATES INITIAL	COORDINATES FINAL	COORDINATES DIMENSIONS	A,INCHES	B,INCHES	C,INCHES	THICKNESS, INCHES CONSTANT	THICKNESS, INCHES VALUE
1	CONICAL	2	178.00000		DEGREES	4.12500	.10852		YES	.00900
2	TOROIDAL	2	178.00000	147.13920	DEGREES	4.01617	-.03500		YES	.00900
3	TOROIDAL	2	147.13920	204.59097	DEGREES	3.85760	.25950		YES	.00900
4	TOROIDAL	2	204.59097	144.59097	DEGREES	3.64583	-.24950		YES	.00900
5	TOROIDAL	2	144.59097	212.86150	DEGREES	3.35090	.25950		YES	.00900
6	CONICAL	2	212.86150		DEGREES	3.21610	.10130		YES	.00900
7	CONICAL	2	32.86150		DEGREES	3.12500	.15720		YES	.00900
8	TOROIDAL	2	32.86150	-24.59097	DEGREES	3.39240	-.24950		YES	.00900
9	TOROIDAL	2	-24.59097	35.40903	DEGREES	3.60430	.25950		YES	.00900
10	TOROIDAL	2	35.40903	-22.33729	DEGREES	3.89920	-.24950		YES	.00900
11	TOROIDAL	2	-22.33729	2.00000	DEGREES	4.01350	.04500		YES	.00900
12	CONICAL	2	2.00000		DEGREES	4.01350	.11150		YES	.00900
13	CONICAL	2	178.00000		DEGREES	4.12500	.10852		YES	.00900
14	TOROIDAL	2	178.00000	147.13920	DEGREES	4.01617	-.03500		YES	.00900
15	TOROIDAL	2	147.13920	204.59097	DEGREES	3.85760	.25950		YES	.00900
16	TOROIDAL	2	204.59097	144.59097	DEGREES	3.64580	-.24950		YES	.00900
17	TOROIDAL	2	144.59097	212.86150	DEGREES	3.35090	.25950		YES	.00900
18	CONICAL	2	212.86150		DEGREES	3.21010	.10130		YES	.00900
19	CONICAL	2	32.86150		DEGREES	3.12500	.15720		YES	.00900
20	TOROIDAL	2	32.86150	-24.59097	DEGREES	3.39240	-.24950		YES	.00900
21	TOROIDAL	2	-24.59097	35.40963	DEGREES	3.60430	.25950		YES	.00900
22	TOROIDAL	2	35.40903	-22.33729	DEGREES	3.89920	-.24950		YES	.00900
23	TOROIDAL	2	-22.33729	2.00000	DEGREES	4.01350	.04500		YES	.00900
24	CONICAL	2	2.00000		DEGREES	4.01350	.11150		YES	.00900

REVISION				
LTR.	DESCRIPTION	DATE	DRAWN	APPROVED

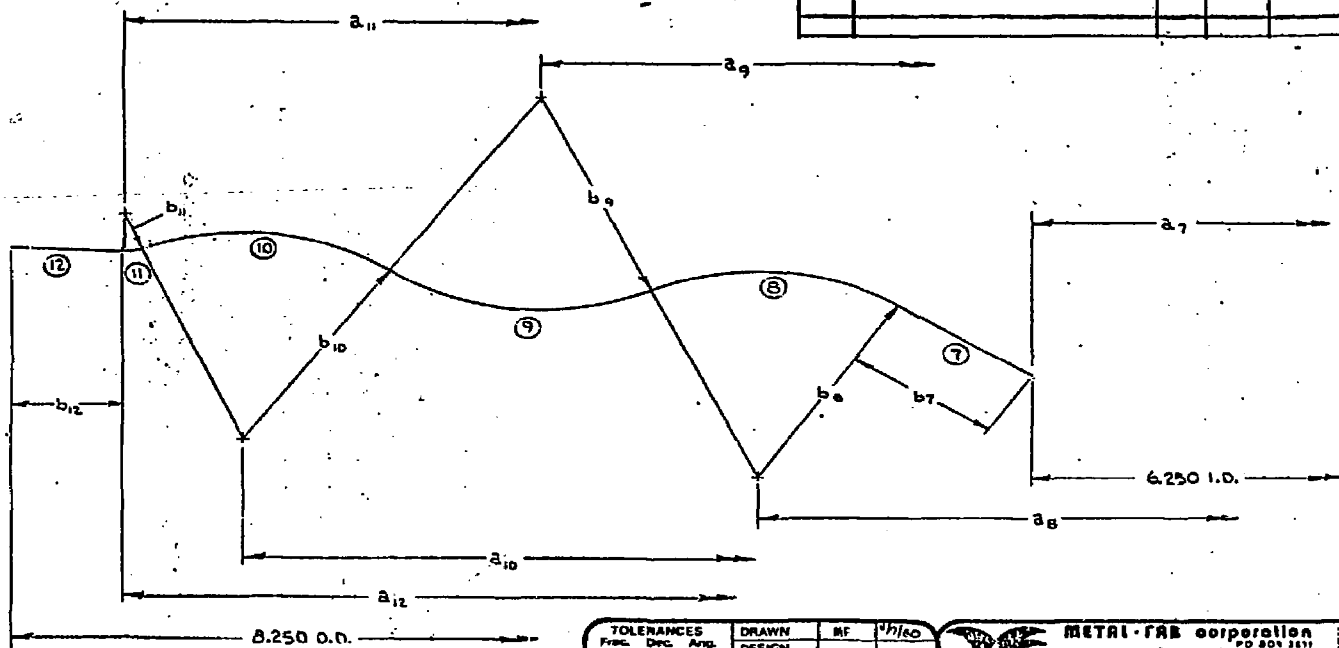


III

TOLERANCES Frac. Dec. Ang. ± ± ±	DRAWN	MF	8/9/80
	DESIGN		
	STRESS		
MATERIAL	PROCESS		
	CHECK		
	PROJECT		
	REF D-1582		

METAL-FAB corporation P.O. BOX 2011 1-904-677-2148 Ormond Beach, Florida

TITLE MALE 6 PART MODEL		
CODE IDENT NO. 2F449	DWG. NO. SK-5161	
SCALE 10 X		SHEET 1 OF 2



REVISION				
LTR.	DESCRIPTION	DATE	DRAWN	APPROVED

TOLERANCES Frac. Dec. Ang. ± ± ±	DRAWN	MF	4/7/80
	DESIGN		
	STRESS		
MATERIAL	PROCESS		
	CHECK		
	PROJECT		
	Ref D- 1582		

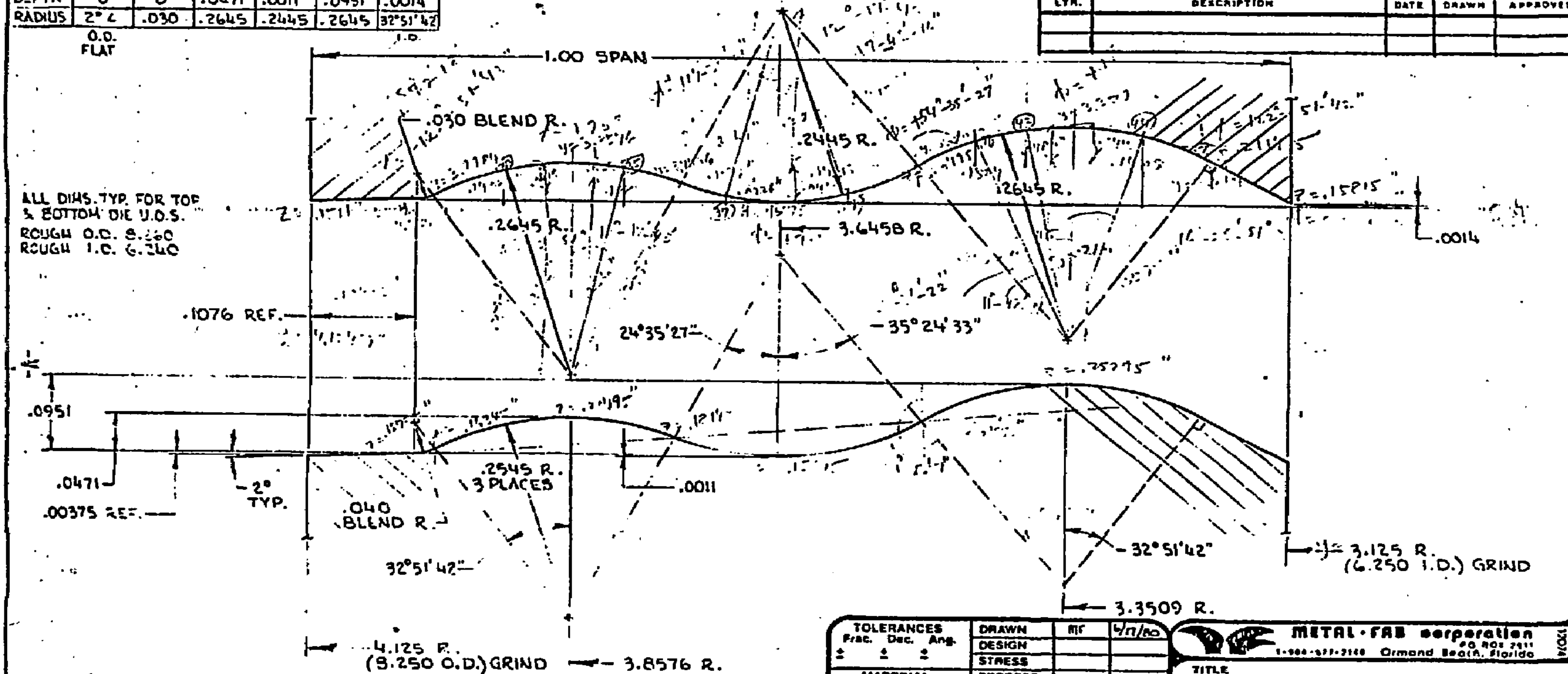
METAL-FAB corporation
P.O. BOX 3511
1-904-677-3100 Ormond Beach, Florida

TITLE
FEMALE 6 PART MODEL

CODE IDENT NO 2F449	DWG. NO. SK-5161
SCALE 10 X	SHEET 2 OF 2

DIST	.1126	.1126	.2724	.4842	.7791	1.005
DEPTH	0	0	.0471	.0011	.0951	.0014
RADIUS	2°4	.030	.2645	.2445	.2645	32°51'42"
	O.D. FLAT					I.D.

ALL DIMS. TYP. FOR TOP & BOTTOM DIE U.O.S.
ROUGH O.D. 8.260
ROUGH I.D. 6.240



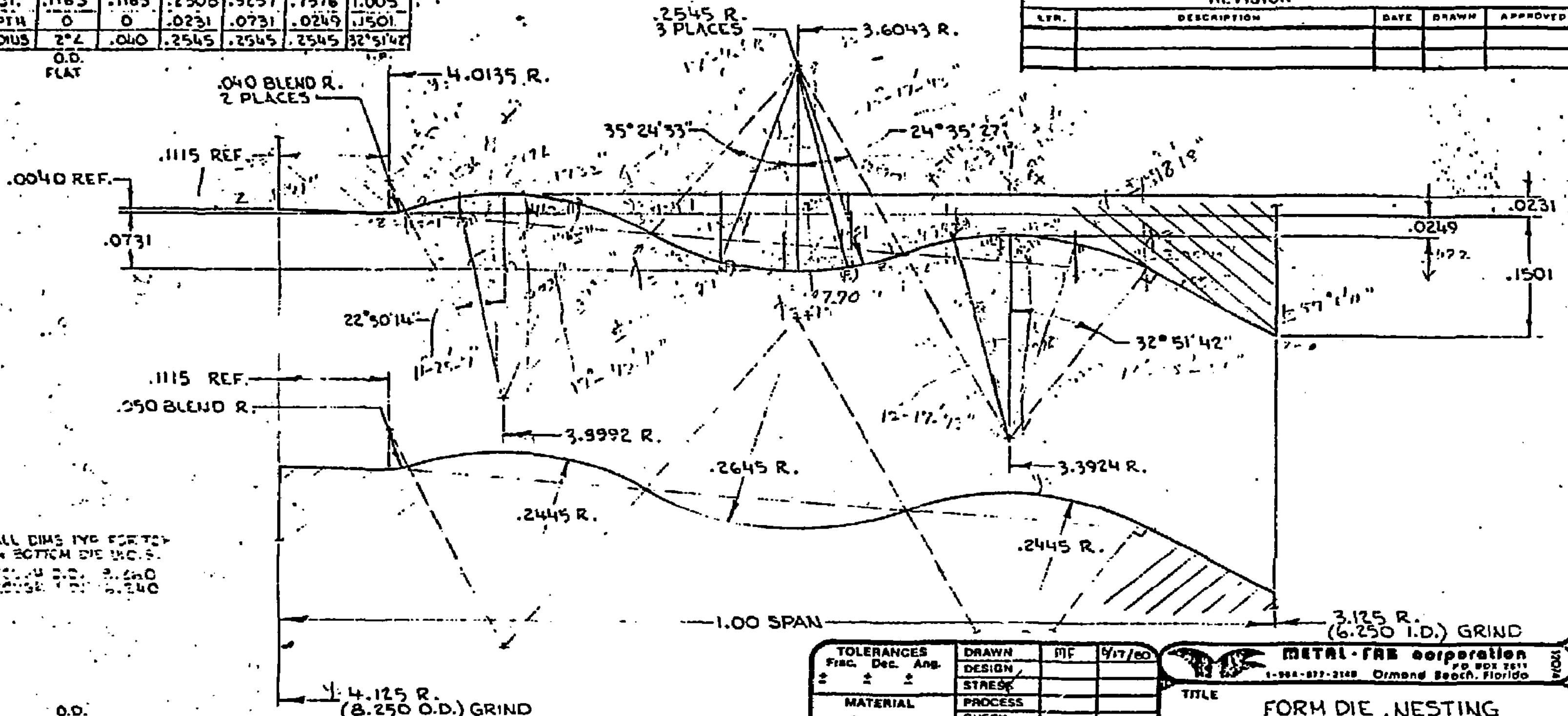
	O.D FLAT					I.D.
DIST	.1126	.1126	[illegible]	.4842	.7791	[illegible]
DEPTH	[illegible]	[illegible]	[illegible]	[illegible]	.0951	.0014
RADIUS	2°4	.040	.2545	.2545	[illegible]	32°51'42"

T = .009

TOLERANCES Frac. ± Dec. ± Ang. ±	DRAWN	MF	4/17/80
MATERIAL: TOOL STEEL 52100 Rc 60	DESIGN		
	STRESS		
	PROCESS		
	CHECK		
	PROJECT		

METAL-FAB corporation, P.O. BOX 2511, Ormond Beach, Florida	
TITLE: FORM DIE, NESTING (MALE ; METAL TO METAL)	
CODE IDENT NO. 2F449	DWG. NO. D-1582
SCALE 10 X	SHEET 1 OF 2

DIST.	.1165	.1165	.2308	.5257	.7376	1.005
DEPTH	0	0	.0231	.0731	.0249	.1501
RADIUS	2° 4	.040	.2545	.2545	.2545	32°51'42"
	O.D. FLAT					I.D.



REVISION				
LTR.	DESCRIPTION	DATE	DRAWN	APPROVED

ALL DIMS TYP FOR TOP
& BOTTOM DIE U.O.S.

	O.D. FLAT					I.D.
DIST.	.1165	.1165	.2306	.5257	.7376	1.005
DEPTH	0	C	.0231	.0731	.0249	.1501
RADIUS	2° 4	.250	.2445	.2445	.2445	32°51'42

T· .009

TOLERANCES Frac. Dec. Ang. ± ± ±	DRAWN	MF	6/17/80
	DESIGN		
	STRESS		
MATERIAL TOOL STEEL 52100 Rc60	PROCESS		
	CHECK		
	PROJECT		

METAL-FAB corporation
PO BOX 7611
Ormond Beach, Florida

TITLE
FORM DIE, NESTING
(FEMALE ; METAL TO METAL)

CODE IDENT NO.	DWG. NO.
2F449	D-1582

SCALE 10 X

SHEET 2 OF 2

D-1562 BELLOWS MALE/FEMALE 2 CONVOLUTIONS 14.000 X 12.000

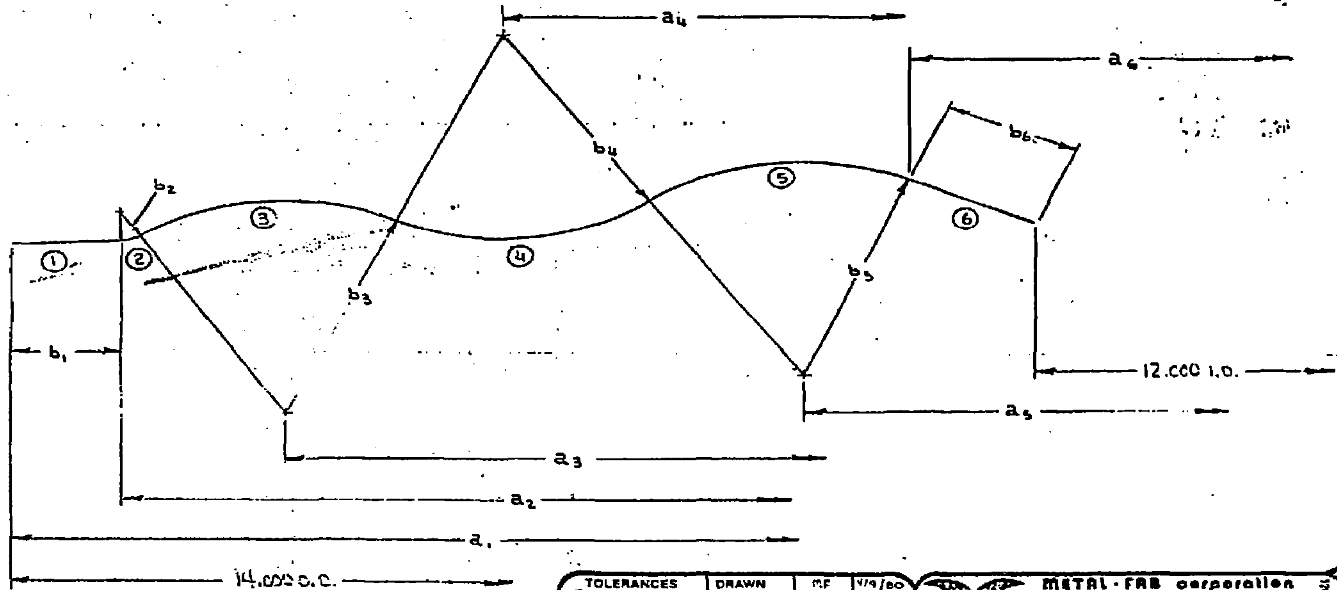
METAL FAB

MAY 25, 1981

** GEOMETRY **

PART	TYPE	NO. OF SEGMENTS	<-- CORDINATES ------> INITIAL	FINAL	DIMENSIONS	A, INCHES	B, INCHES	C, INCHES	THICKNESS, CONSTANT	INCHES VALUE
1	CONICAL	2	178.00000		DEGREES	7.00000	0.10760		YES	0.00900
2	TOROIDAL	2	178.00000	147.13820	DEGREES	6.89240	-0.04000		YES	0.00900
3	TOROIDAL	2	-147.13820	204.58920	DEGREES	6.73260	0.25450		YES	0.00900
4	TOROIDAL	2	204.58920	144.59370	DEGREES	6.52080	-0.25450		YES	0.00900
5	TOROIDAL	2	144.59370	220.00000	DEGREES	6.22590	0.25450		YES	0.00900
6	CONICAL	2	220.00000		DEGREES	6.06230	0.08130		YES	0.00900
7	CONICAL	2	40.00000		DEGREES	6.00000	0.13550		YES	0.00900
8	TOROIDAL	2	40.00000	-24.58920	DEGREES	6.26740	-0.25450		YES	0.00900
9	TOROIDAL	2	-24.58920	-35.40630	DEGREES	6.47930	0.25450		YES	0.00900
10	TOROIDAL	2	35.40630	-22.83730	DEGREES	6.77420	-0.25450		YES	0.00900
11	TOROIDAL	2	-22.83730	2.00000	DEGREES	6.83350	0.04000		YES	0.00900
12	CONICAL	2	2.00000		DEGREES	6.88850	0.11150		YES	0.00900
13	CONICAL	2	178.00000		DEGREES	7 00000	0.10760		YES	0.00900
14	TOROIDAL	2	178.00000	147.13820	DEGREES	6.89240	-0.04000		YES	0.00900
15	TOROIDAL	2	-147.13820	204.58920	DEGREES	6.79260	0.25450		YES	0.00900
16	TOROIDAL	2	204.58920	144.59370	DEGREES	6.52080	-0.25450		YES	0.00900
17	TOROIDAL	2	144.59370	220.00000	DEGREES	6.22590	0.25450		YES	0.00900
18	CONICAL	2	220.00000		DEGREES	6.06230	0.08130		YES	0.00900
19	CONICAL	2	40.00000		DEGREES	6.00000	0.13550		YES	0.00900
20	TOROIDAL	2	40.00000	-24.58920	DEGREES	6.26740	-0.25450		YES	0.00900
21	TOROIDAL	2	-24.58920	-35.40630	DEGREES	6.47930	0.25450		YES	0.00900
22	TOROIDAL	2	35.40630	-22.83730	DEGREES	6.77420	-0.25450		YES	0.00900
23	TOROIDAL	2	-22.83730	2.00000	DEGREES	6.88850	0.04000		YES	0.00900
24	CONICAL	2	2.00000		DEGREES	6.88850	0.11150		YES	0.00900

REVISION				
LTR.	DESCRIPTION	DATE	DRAWN	APPROVED



TOLERANCES Frac. Dec. Ang. ± ± ±	DRAWN	MF	9/9/80
	DESIGN		
	STRESS		
MATERIAL	PROCESS		
	CHECK		
	PROJECT		
	REF D-1562		

METAL-FAB corporation PO BOX 2811 1-904-677-2144 Ormond Beach, Florida

TITLE MALE 6 PART MODEL

CODE IDENT NO.	DWG. NO.
2F449	SK-5174
SCALE 10 X	SHEET 1 OF 2