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ANALYSIS OF STRESSES IN BELLOWS

Part I. Design Criteria and Test Results

By
W. F. Anderson

October 15, 1964

Division of North American Aviation, Inc.
Atomics International
Canoga Park, California

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PART I

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ABSTRACT

Design charts and systematic design forms are presented for simplified calculations to check the number of convolutions and thickness required to limit the deflection and pressure stress range in three types of bellows: (1) convoluted bellows, (2) convoluted bellows with reinforcing rings and (3) toroidal bellows. The design charts are based on the equations for stresses derived from an asymptotic solution for the equations of toroidal shells found in previously published literature.

Proposed stress limitations to be used with the calculated stresses are based on those of the ASA Code for Pressure Piping. Data from 108 fatigue tests at 70°F and 18 fatigue tests at 1200°F for all three types of bellows are presented. These data are evaluated statistically and justify use of the proposed allowable stresses. The conclusion is drawn that expansion joint bellows can be designed to the same stress levels as other components of a piping system with equal confidence in the reliability of the design.

The mathematical derivation and equations programmed and solved by a digital computer are to be presented in Part II of this report. Limited experimental data with references are also to be presented in Part II to confirm the accuracy of the analysis for stresses.

I. INTRODUCTION

The problem of thermal stresses is especially important in sodium cooled reactor power plant systems because, while pressure stresses are low, large temperature changes are associated with the high operating temperatures.

Bellows expansion joints are frequently considered as elements in a piping system for purposes of relieving stresses induced by thermal expansion of the system. Methods of analysis for expansion stresses in a piping system without bellows have been accepted by industry and, while cumbersome, are relatively convenient. Maximum allowable stresses for expansion and pressure stresses in conventional components have been established and accepted in code form.[1] However, no such methods of analysis or system of allowable stresses have been established or accepted for the bellows used in sodium cooled reactor power plant systems, or for general use in other systems.

This study of bellows stress analysis and application was undertaken as part of the Advanced Sodium Graphite Reactor (ASGR) Project, under a subaccount established for thermal stress fatigue studies. The object of this study was to establish a method of stress analysis and a system of allowable stress for reliable application of bellows in sodium cooled reactor systems.

The final goal in developing this system of analysis and stresses is a specification for the purchase of reliable bellows. A copy of such a specification is attached as Appendix C. Calculation forms (described in the text) used with the specification primarily established a chronology for calculations based on the formulae in the specification. These formulae, based on simple strip beam relations with correction factors, serve to check proposed designs rather than to direct design.

A. DESIGN APPLICATION OF BELLOWS

Preliminary investigations disclosed that bellows capabilities for reducing piping expansion stresses are often grossly over-estimated. In one case, bellows were being suggested for installation to reduce piping expansion stresses from 40,000 psi to 26,000 psi (the ASA code allowable, in this case), whereas the bellows, as proposed, would have been subjected to calculated stresses of 800,000 psi. Such design practice may be rational where bellows are treated as an expendable item to be easily replaced with little economic penalty. However, except where it is established that this is a reasonable design practice, such inconsistency in allowable stresses should be avoided, or the implications made obvious. In nuclear power plant installations, where containment of radioactive fluids is a paramount consideration, good design practice would require safety factors in a system of allowable stresses in the bellows, or experimentally determined safety factors, consistent with those allowed in the remainder of the system. If design is based on calculated stresses, methods of analysis for expansion and pressure stresses in bellows should be as accurate as those used in the analysis of stresses in the remainder of the system.

The design procedure, including the method of calculating stresses, must be applicable with sufficient ease and directness as to permit inclusion in standard design practice manuals and codes.

As presented herein, the method of calculating stresses in bellows uses the same mathematical formulation and solution for stresses in toroidal shells as was used to calculate the stress concentration and flexibility factors for piping elbows. Both bellows and piping elbows are sectors of toroidal shells. Flexibility and stress factors were calculated, relating the bellows to simple strip beams, and arranged in design charts as shown in Figures 1 through 8.

Three types of bellows commonly used in piping systems were studied: (1) convoluted bellows, (2) convoluted bellows with reinforcing rings, and (3) toroidal bellows. Standard design forms were prepared to facilitate calculation of the stresses induced by internal pressure and deflections, for comparison with allowed stresses in a bellows. The design forms are presented in this report with directions for use of the design charts which were developed for each type of bellows. These have been used as integral parts of specifications for the purchase of bellows.

It is recognized that ultimate confirmation of the suitability of this or any other analytical technique, as with the analysis of stresses in piping elbows,[2,3] should depend upon evaluation of the method with a statistically significant amount of data from fatigue tests and data on the performance of bellows under operating conditions.

Experimental data were evaluated using the analytical technique discussed herein, and the design charts. Safety factors were determined, comparable to those used in the piping code. Accordingly, the principles and basic values of the system of allowable stresses established by the ASA Code of Pressure Piping[1] are proposed for application to bellows when incorporated in a piping system.

Fatigue data were gathered from whatever sources were available; Reference 19 was the only published source. Bellows manufacturers submitted most of the data on toroidal and reinforced bellows. Results from the bellows test program of the Rocketdyne Division of North American Aviation provided the major source of data on convoluted bellows. Unless otherwise specified, the tests discussed in this report were performed in air at room temperature.

In addition to assuring satisfactory cyclic life, bellows design problems other than stresses and spring rate may be handled with increased confidence, and a more simplified approach may be used when the stresses are limited. Angular movement and lateral translation of a bellows must be related to equivalent axial movement for determination of stresses and spring rate. With stresses limited, a general strength of materials approach is used to equate these various types of motion. Formulae developed for these relations in Part II* of this report are included in the design forms.

Bellows as a structure may become unstable or "Squirm" under internal pressure. An approach to this problem is outlined in Reference 17. When subjected to angular deflection, the tendency toward instability tends to further increase the deflection stresses at the center of the bellows. Equations based on elastic behavior[17] are used to study this problem (Appendix A). Theoretical limitations have been placed on internal pressure to limit increase in stresses or avoid instability. Formulae outlining these limitations are included in the design forms. Limited experimental confirmation for these formulae and pressure limitations are available in Tables 1-4 included in Section II of this report.

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*To be published

B. MATHEMATICAL ANALYSIS OF STRESSES IN BELLOWS

The mathematical analysis of stresses and flexibility of bellows has tended to follow the same chronological pattern as mathematical analysis of stresses and flexibility of piping elbows.

The first analyses of piping elbows expressed the distortion of the elbow by terms of a Fourier Series and solved for coefficients of these terms[4, 5, 6, 7]. As in all series approximations, the accuracy of these analyses varied with the number of terms; the usefulness of the results was limited to a range where the form factor Rt/r^2 was not small.

Clark[8] attacked the shell theory equations and obtained a solution asymptotic to the exact solution at very small values of Rt/r^2. The overlapping applicability of solutions based on these two approaches provides a full range of analytical design capability for piping elbows.

Similarly, stress problems in bellows have been studied by the same two mathematical techniques of the series solution and the asymptotic solution. The series solutions are most accurate at large values of Rt/r^2 and the asymptotic solutions being most accurate at small values of Rt/r^2.

Salzman[9] studied convoluted bellows using a three term series approximation for the shape of the toroidal shell sections. The solutions were applicable only to deflection stresses for convoluted bellows with an included flat plate section ranging from zero to full convolution depth. Design curves were presented but test data were not included.

Turner and Ford[10] used a five term series for the shape of the toroidal shell sections. The solutions presented covered deflection stresses in convoluted bellows with no flat sections and included reversed curve sections with 270° of arc. The study included test data from static tests but did not present design curves.

Dahl[11] used a four term series to express the bending deformation in studying toroidal bellows. Results of solutions were compared with the asymptotic solutions by Clark.[12] Solutions were applicable to deflection stresses in toroidal bellows only. Design curves were presented in addition to data from one test.

Clark[12] extended his asymptotic solution for curved tubes to both toroidal and convoluted bellows. The solution for convoluted bellows did not include flat plate sections. Deflection stresses were studied for the convoluted type of bellows. Design formulae are presented for each case, applicable with small values of the form factor.

Laupa and Weil[13] used a five term series to express the bending deformation in analyzing toroidal sections of convoluted bellows, applicable with large values of the form factor. The method of forming the matrix of equations to obtain solutions is outlined in the reference. Convoluted bellows, with and without flat sections, subjected to both deflection and internal pressure are included. A sample problem is solved but neither design charts nor formulae are presented.

Welded disc bellows were studied by Feeley and Goryl.[14] The geometry and range of commercial sizes of such bellows allowed a beam concept to be used in a simplified analysis. Stresses due to axial and angular deflection forces and pressure stresses were studied. Design formulae and limited test results were presented.

A study of welded curved disc bellows was presented by Hetenyi and Timms,[15] based on the work by Clark.[12] Design formulae for deflection, deflection stress and pressure stress were presented for the range where Rt/r^2 is very small. Haringx[16] presented analytical and experimental results on similar types of bellows.

The need for a comprehensive study of piping expansion bellows and readily applicable design charts becomes apparent from a review of the previous reference studies. Results of the study presented herein attempt to satisfy that need. The mathematical solution utilized is based on the asymptotic solution by Clark[12] and is presented in Part II with some of the modifications introduced by Hetenyi and Timms.[15] Review of the mathematical approximations made by Clark, to obtain a solution valid only at small values of Rt/r^2, indicated that compensating corrections could be introduced after a solution was obtained, which would result in reasonable accuracy over the full range of geometric parameters.

Geometric parameters considered were: (1) the form factor taken as $a^2/0.778\ Rt$; (2) the ratio (a/h) of the crest radius "a" to the sum "h" of one-half the flat plus the crest radius; and (3) the three types of bellows. To limit the range of geometric parameters to be covered by the calculations for the design charts, it was decided to exclude the effect of the following:

1) variations in thickness,

2) variations in crest radius between ID and OD,

3) the ratio of OD to ID, and

4) sloping or shell curvature of the annular disc between crests.

Item 3) of these is compensated for in the calculation form. Wahl and Lobo[18] showed that an approximation considering a radial strip of a circular plate as a beam, yields satisfactory results. This approximation is valid for values ranging from 1.0 to 1.5 for the ratio between OD and ID. Further examination of Reference 18 shows that similar approximations can be made for peak bending moments. These approximations are compensated for in the design forms to reduce the error introduced by excluding item 3). The effect of item 4) has been evaluated separately. Reference 23 contains design charts for annular shells which show that shells with slight curvature can be treated as flat strips with minor corrections.

The mathematical analysis of stresses is presented in Part II of this report. Matrices of four equations were formed, expressing the boundary conditions of only the toroidal shell sections for each of the various types of bellows. All equations were nondimensionalized to relate results to the behavior of simple geometric forms, namely a flat strip of uniform width with guided support at each end.

A digital computer program was devised for the following:

1) To set up the coefficients for, and solve, the boundary condition equations.

2) Calculate the distribution of moments, forces and deflections for the toroidal sections, plus deflections for the flat annular section.

3) Correct the calculated moments to compensate for the mathematical approximations.

4) Print out the results and also plot graphically the bending moment distributions in the toroidal sections.

Results provided by the computer program were again cross plotted to provide design charts for spring rate, deflection stresses, and pressure stresses for the three types of bellows. Results of this method were compared with those of References 9, 10, 11 and 12, to confirm the existence of reasonable accuracy.

The computer program which was devised can be used to solve equations for bellows which include the effects of the four parameters not included in this study. This can be done by forming of the proper boundary condition equations and values of parameters describing the geometry. The validity of the simplified approach was checked with sample calculations for certain cases of these additional parameters.

Results of static loading tests, to study stress distributions, are included in Part II of this report for comparison of analytical and experimental results.

II. DESIGN OF BELLOWS

A. CONVOLUTED BELLOWS

Convoluted bellows with undisturbed semi-toroidal sections are more amenable to stress analysis, both mathematical and experimental, than either of the other types of bellows discussed in this report. Data on stresses for this type of bellows has been made available from both static and fatigue tests.

Examination of a few sample problems will indicate that convoluted bellows appear best suited for low pressure applications. Exceeding the yield stress at the root of the convolution by applying internal pressure can result in plastic instability (squirm) due to local yielding. Reinforcing this area with root rings prevents local yielding and raises the pressure at which instability occurs. Conservative design requires that pressure bending stresses without these reinforcing rings be limited to the elastic or low-creep-rate range.

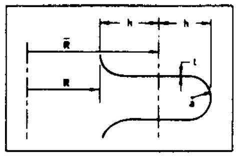
In the following design analysis, using design sheet Form 1 pressure bending stresses determine the minimum thickness required for a given bellows convolution. The number of convolutions required are based on the design thickness, required deflection and remaining allowable stress range. Spring rate and stability factors are calculated after the number of convolutions is established. The coefficients C_p, C_f, and C_d from Figures 1, 2, 3 used in these calculations relate the behavior of a convolution segment to a simple strip beam.

In this report one convolution is considered as the length from one inner crest to the next inner crest. Each convolution is separated, for calculation purposes into four segments, each of which has a projected length, "h", and a developed length $[a\pi/2 + (h - a)]$. Flat strips of length "h" simulating each of these segments provide the basis for the design formulae. The factors C_p, C_f, and C_d serve to adjust the easily predicted behavior of the flat strips to the calculated behavior of a bellows convolution segment.

Special consideration in analysis for stresses must be given to the effect of "squirm". Limiting the ratio of operating pressure and critical pressure to a value less than 0.1 also limits the value of C_{ps} (stress increase factor) to less than 1.18. This factor could result in a similar reduction of the safety factor for bellows used as hinges when bending stresses are increased by this amount, due to squirm. Therefore, the value of equivalent deflection due to angular rotation is multiplied by 1.20 to maintain an equal safety factor for all loadings. Limiting the ratio of operating to critical pressure also provides a margin for the effect of yield strength on squirm pressure.

It must be pointed out that, theoretically, the peak bending stresses induced by pressure need not coincide geometrically with the peak bending stresses induced by deflection. For most bellows they will coincide, but if more refined analysis is desired which would more accurately determine the combined peak stress, the graphs of bending moment distributions in Part II may be consulted.

DESIGN SHEET FORM 1
CONVOLUTED BELLOWS
(REFERENCE FIGURES 1, 2 & 3)



BELLOWS IDENTIFICATION __________

DESIGN CONDITIONS

1 INTERNAL PRESSURE p = ______ psig	2 TEMPERATURE ______ °F	3 NUMBER OF CYCLES ______
4 AXIAL DEFLECTION Δ_a = ______ in.	5 LATERAL TRANSLATION δ ______ in.	6 ANGULAR MOVEMENT θ = ______ radian
7 TORSIONAL ANGULAR ROTATION ALLOWED BY CLEARANCES IN HARDWARE OF RESTRAINTS Φ = ______ radian	8 TORSIONAL MOMENT OBTAINED BY PIPING STRESS ANALYSIS IF HARDWARE DOES NOT OFFER PROTECTION AGAINST TORSIONAL ROTATION M_t = ______ in. lb	

MATERIAL PROPERTIES

9 ALLOWABLE STRESS AT ROOM TEMPERATURE S_c = ______ psi	10 ALLOWABLE STRESS AT DESIGN TEMPERATURE S_h = ______ psi	11 ALLOWABLE STRESS RANGE FOR COMBINED STRESSES $S_{AC} = 1.25f(S_c + S_h)$ = ______ psi
12 YOUNG'S MODULUS AT ROOM TEMPERATURE E_c = ______ psi	13 YOUNG'S MODULUS AT OPERATING TEMPERATURE E_h = ______ psi	f STRESS RANGE REDUCTION FACTOR

BELLOWS DIMENSIONS AND PARAMETERS (SEE FIGURE ABOVE)

14 R = ______ in.	15 h = ______ in.	16 a = ______ in.
17 $\bar{R} = R + h$ = ______ in.	18 $\left(\frac{\pi}{2} - 1\right)a + h$ = ______ in.	19 a/h = ______
20 t = ______ in.	21 NUMBER OF PLIES N_p = ______	22 NUMBER OF CONVOLUTIONS N_c = ______
23 OVERALL ACTIVE LENGTH OF BELLOWS L = ______ in.		USE $\boxed{19}$ AND $\boxed{25}$ TO DETERMINE FACTORS C_p, C_d, C_f
24 $x = 0.778\sqrt{\boxed{17} \times \boxed{20}}$ = ______ in.	25 $a/x = \boxed{16} / \boxed{24}$ = ______	
26 C_p = ______ FROM FIG. 1	27 C_d = ______ FROM FIG. 2	28 C_f = ______ FROM FIG. 3

STRESS CALCULATION

29 MEMBRANE STRESS DUE TO INTERNAL PRESSURE

$$S_{mp} = \frac{\boxed{1}}{\boxed{21} \times \boxed{20}} \left(\boxed{15} + \frac{\boxed{17} \times \boxed{16}}{\boxed{18}} \right) = \_\_\_\_\_\_ \text{ psi}$$

MUST NOT EXCEED $\boxed{10}$

30 BENDING STRESS DUE TO INTERNAL PRESSURE

$$S_{bp} = \frac{2 \times \boxed{26} \times \boxed{1} \times \boxed{15} \times \boxed{15} \times \boxed{17}}{\boxed{20} \times \boxed{20} \times \boxed{21} \times \boxed{14}} = \_\_\_\_\_\_ \text{ psi}$$

31 COMBINED PRESSURE STRESSES

$$S_p = \boxed{29} + \boxed{30} = \_\_\_\_\_\_ \text{ psi}$$

MUST NOT EXCEED 1.25 X $\boxed{10}$

32 EQUIVALENT TOTAL AXIAL DEFLECTION

$$\Delta = \boxed{4} + \frac{6 \times \boxed{5} \times \boxed{17}}{\boxed{23}} + 1.2 \times \boxed{17} \times \boxed{6} = \_\_\_\_\_\_ \text{ in.}$$

33 BENDING STRESS DUE TO DEFLECTION

$$S_{bd} = \frac{0.412 \times \boxed{12} \times \boxed{20} \times \boxed{32} \times \boxed{17}}{\boxed{15} \times \boxed{15} \times \boxed{22} \times \boxed{27} \times \boxed{14}} = \_\_\_\_\_\_ \text{ psi}$$

34 COMBINED MEMBRANE AND BENDING STRESSES

$$S_d = \boxed{29} + \boxed{30} + 0.5 \times \boxed{33} = \_\_\_\_\_\_ \text{ psi}$$

35 TORSIONAL STRESSES

a) $$S_t = \frac{\boxed{7} \times \boxed{12} \times \boxed{17}}{10.4 \times \boxed{22} \times \boxed{18}} = \_\_\_\_\_\_ \text{ psi}$$

OR

b) $$S_t = \frac{\boxed{8}}{2\pi \times \boxed{14} \times \boxed{14} \times \boxed{20} \times \boxed{21}} = \_\_\_\_\_\_ \text{ psi}$$

ITEM b) MUST NOT EXCEED 0.5 X $\boxed{10}$

36 TOTAL COMBINED STRESSES

$$S = \sqrt{(\boxed{34} \times \boxed{34}) + (4 \times \boxed{35} \times \boxed{35})} = \_\_\_\_\_\_ \text{ psi}$$

MUST NOT EXCEED $\boxed{11}$

SPRING RATES

37 AXIAL SPRING RATE

$$K_a = \frac{0.431 \times \boxed{17} \times \boxed{13} \times \boxed{20} \times \boxed{20} \times \boxed{20} \times \boxed{21}}{\boxed{22} \times \boxed{15} \times \boxed{15} \times \boxed{15} \times \boxed{28}} = \_\_\_\_\_\_ \text{ lb/in.}$$

38 ROTATION SPRING RATE

$$K_r = 0.5 \times \boxed{17} \times \boxed{17} \times \boxed{37} = \_\_\_\_\_\_ \text{ in. lb/radian}$$

STABILITY

39 FACTOR OF STABILITY

$$F_s = \frac{2\pi \times \boxed{37}}{\boxed{1} \times \boxed{23}} = \_\_\_\_\_\_$$

MUST BE GREATER THAN 10 FOR COMBINED MOVEMENT
MUST BE GREATER THAN 5 IN CASE OF AXIAL DEFLECTION ONLY

Design Form 1 - Convoluted Bellows

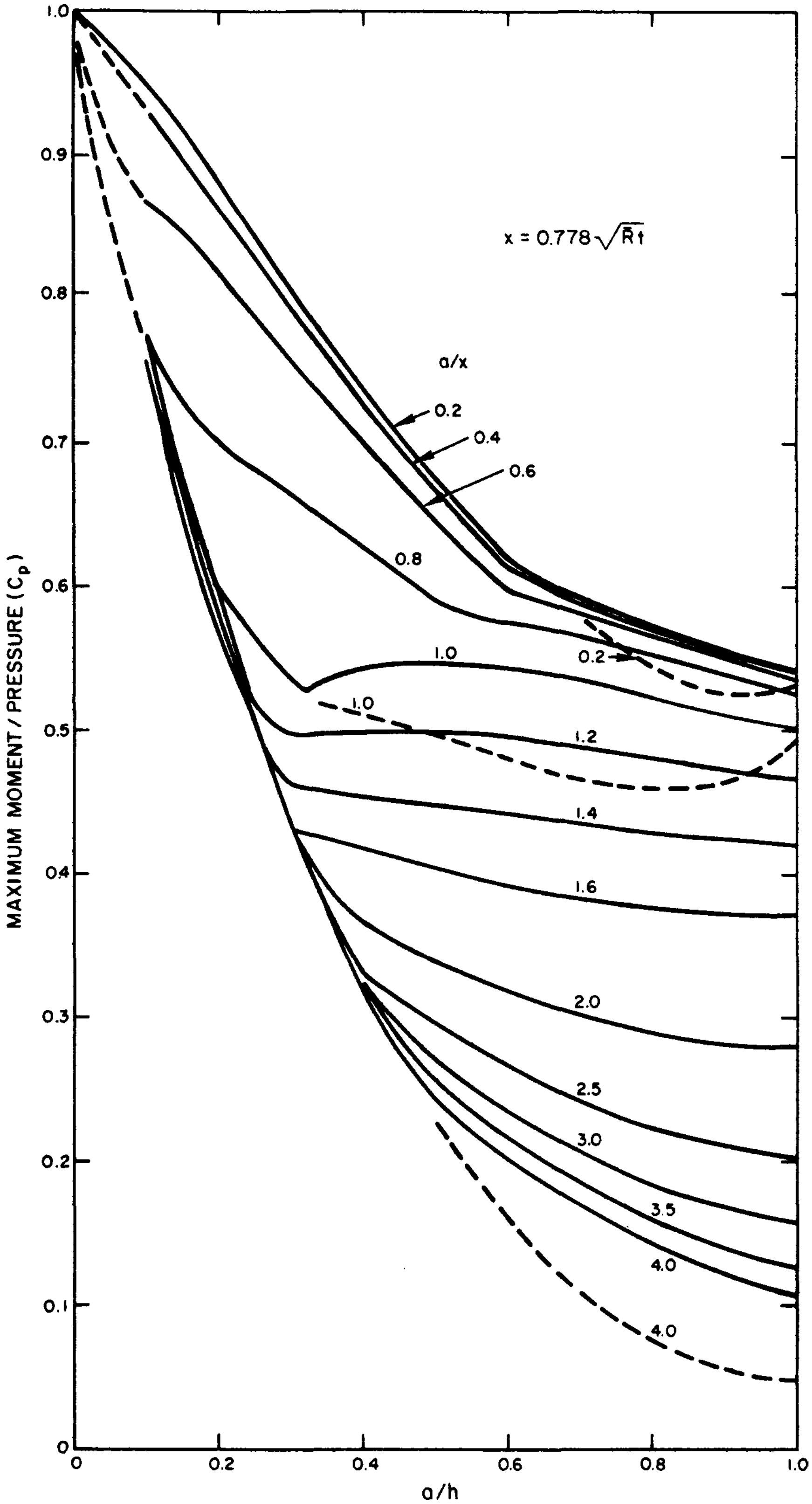


Figure 1. Maximum Moment/Pressure "C_p" for Convoluted Bellows

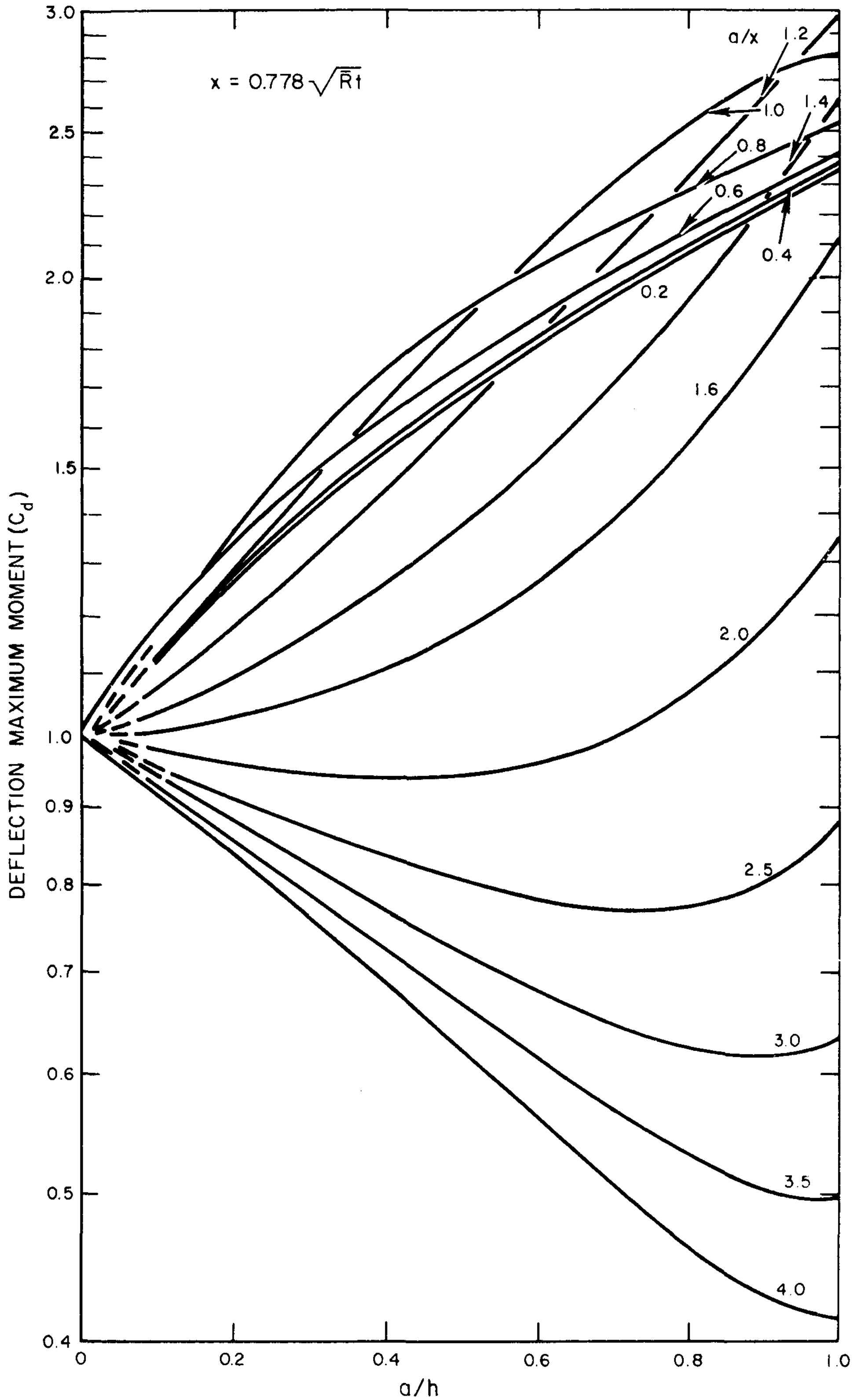


Figure 2. Deflection/Maximum Moment "C_d" for Convoluted Bellows

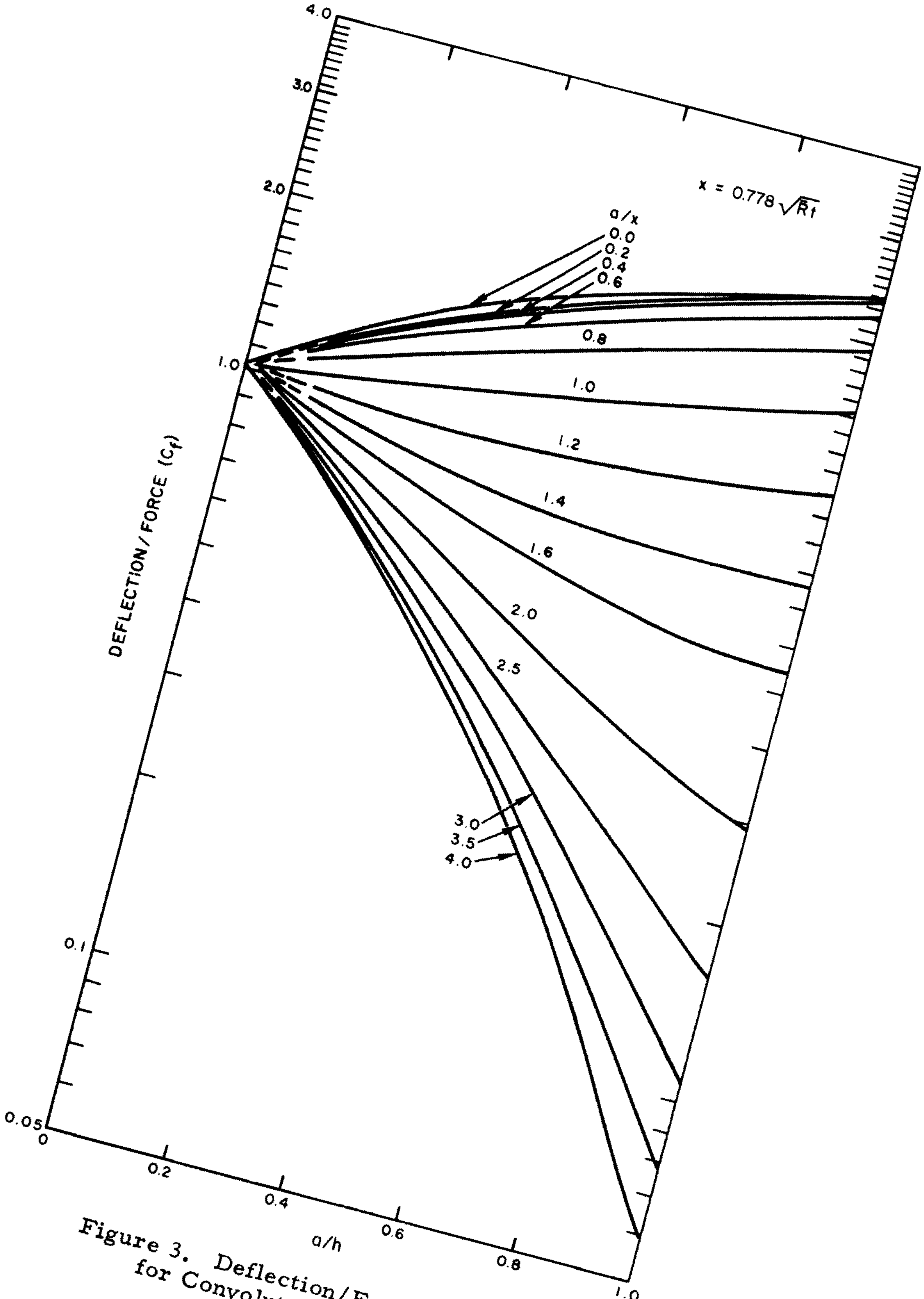


Figure 3. Deflection/Force "C_f" for Convoluted Bellows

B. RING REINFORCED BELLOWS

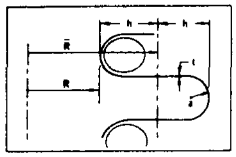
The wide use of ring reinforced bellows, especially in the intermediate pressure range, requires their inclusion in this report even though inherent problems in the analysis of stresses have not been resolved. As pointed out in the discussion of convoluted bellows, the presence of reinforcing rings adds to the stability of bellows when subjected to internal pressure. With bellows under test, one can observe that reinforcing rings eliminate any tendency for the bellows to collapse at the root due to internal pressure. One can also observe that deformations due to internal pressure tend to produce a modified shape of convolution more capable of resisting internal pressure forces by membrane stresses. Ring reinforced bellows generally find application in the intermediate pressure range of 30 psi to 500 psi.

A major question in the analysis for stress in ring reinforced bellows concerns the effect of contact of the convoluted sheet material with the reinforcing ring. The location of the line of initial contact and the effect of the movement of this line of contact on flexibility stress and fatigue life are parts of the problem. For this analysis, one primary assumption considers the radius is identical at both crests of the convolution. Further assumptions made for deflection flexibility and stresses were that (1) the peak bending stress at the initial point of contact of the ring and the convolution is equal to maximum at the exterior crest of the convolution, and (2) the slope at the initial point of contact is that which provides geometric compatibility for changes of slope around the convolution. For pressure stresses, similar assumptions were made, matching deflections and moments near the center of the flat span and allowing whatever moment and slope is necessary at the initial point of contact with the reinforcing ring. Assumption (1) is the most optimistic which can be made concerning peak stresses for a given deflection. In evaluating results from fatigue tests this would result in the lowest calculated peak stress for a given cyclic life and would result in the lowest allowable stresses being derived empirically. This assumption, while it may be in error, has a logical conservative basis since any improvement in analytical method will probably predict higher stresses and when used with the proposed allowables, will give a more conservative design.

As with the convoluted bellows, the peak pressure bending stresses are taken to coincide in location with the peak deflection bending stresses. The actual combined peak based on the assumption mentioned, can be derived from bending moment distribution curves of Part II of the report.

It is assumed, and should be specified, that forces in the reinforcing rings shall compensate for the circumferential membrane stress in the convolutions. In Design Sheet Form 2 for reinforced bellows, thickness requirements are based primarily on pressure membrane stresses. The combined pressure stresses are subtracted (after reduction of the pressure bending stresses by an empirically deduced factor of 0.25) from the total allowable stress range. The required number of convolutions is based on the remaining stress range. Spring rates and stability factors are determined as in the calculation forms for convoluted bellows. The coefficients C_p, C_f, and C_d from Figures 4, 5, and 6 relate the behavior of a convolution segment to the behavior of a simple strip beam.

DESIGN SHEET FORM 2
RING REINFORCED BELLOWS
(REFERENCE FIGURES 4, 5 & 6)



BELLOWS IDENTIFICATION ____

DESIGN CONDITIONS

1 INTERNAL PRESSURE p = ____ psig	2 TEMPERATURE ____ °F	3 NUMBER OF CYCLES ____
4 AXIAL DEFLECTION Δ_a = ____ in.	5 LATERAL TRANSLATION δ ____ in.	6 ANGULAR MOVEMENT Θ = ____ radian
7 TORSIONAL ANGULAR ROTATION ALLOWED BY CLEARANCES IN HARDWARE OF RESTRAINTS Φ = ____ radian	8 TORSIONAL MOMENT OBTAINED BY PIPING STRESS ANALYSIS IF HARDWARE DOES NOT OFFER PROTECTION AGAINST TORSIONAL ROTATION M_t = ____ in. lb	

MATERIAL PROPERTIES

9 ALLOWABLE STRESS AT ROOM TEMPERATURE S_c = ____ psi	10 ALLOWABLE STRESS AT DESIGN TEMPERATURE S_h = ____ psi	11 ALLOWABLE STRESS RANGE FOR COMBINED STRESSES $S_{AC} = 1.25f(S_c + S_h)$ = ____ psi; f STRESS RANGE REDUCTION FACTOR
12 YOUNG'S MODULUS AT ROOM TEMPERATURE E_c = ____ psi	13 YOUNG'S MODULUS AT OPERATING TEMPERATURE E_h = ____ psi	

BELLOWS DIMENSIONS AND PARAMETERS (SEE FIGURE ABOVE)

14 R = ____ in.	15 h = ____ in.	16 a = ____ in.
17 $\bar{R} = R + h$ = ____ in.	18 $\left(\frac{\pi}{2} - 1\right)a + h$ = ____ in.	19 a/h = ____
20 t = ____ in.	21 NUMBER OF PLIES N_p = ____	22 NUMBER OF CONVOLUTIONS N_c = ____
23 OVERALL ACTIVE LENGTH OF BELLOWS L = ____ in.		USE $\boxed{19}$ AND $\boxed{25}$ TO DETERMINE FACTORS C_p, C_d, C_f
24 $X = 0.778\sqrt{\boxed{17} \times \boxed{20}}$ = ____ in.	25 $a/x = \boxed{16} / \boxed{24}$ = ____	
26 C_p = ____ FROM FIG. 4	27 C_d = ____ FROM FIG. 5	28 C_f = ____ FROM FIG. 6

STRESS CALCULATION

29 MEMBRANE STRESS DUE TO INTERNAL PRESSURE

$$S_{mp} = \frac{\boxed{1} \times \boxed{15}}{\boxed{21} \times \boxed{20}} = \underline{\qquad\qquad} \text{ psi}$$

MUST NOT EXCEED $\boxed{10}$

30 BENDING STRESS DUE TO INTERNAL PRESSURE

$$S_{bp} = \frac{2 \times \boxed{26} \times \boxed{1} \times \boxed{15} \times \boxed{15} \times \boxed{17}}{\boxed{20} \times \boxed{20} \times \boxed{21} \times \boxed{14}} = \underline{\qquad\qquad} \text{ psi}$$

31 COMBINED PRESSURE STRESSES

$$S_p = \boxed{29} + \boxed{30} = \underline{\qquad\qquad} \text{ psi}$$

32 EQUIVALENT TOTAL AXIAL DEFLECTION

$$\Delta = \boxed{4} + \frac{6 \times \boxed{5} \times \boxed{17}}{\boxed{23}} + 1.2 \times \boxed{17} \times \boxed{6} = \underline{\qquad\qquad} \text{ in.}$$

33 BENDING STRESS DUE TO DEFLECTION

$$S_{bd} = \frac{0.412 \times \boxed{12} \times \boxed{20} \times \boxed{32} \times \boxed{17}}{\boxed{15} \times \boxed{15} \times \boxed{22} \times \boxed{27} \times \boxed{14}} = \underline{\qquad\qquad} \text{ psi}$$

34 COMBINED MEMBRANE AND BENDING STRESSES

$$S_d = \boxed{29} + 0.25 \times \boxed{30} + 0.5 \times \boxed{33} = \underline{\qquad\qquad} \text{ psi}$$

35 TORSIONAL STRESSES

$$\text{a) } S_t = \frac{\boxed{7} \times \boxed{12} \times \boxed{17}}{10.4 \times \boxed{22} \times \boxed{18}} = \underline{\qquad\qquad} \text{ psi} \quad \text{OR} \quad \text{b) } S_t = \frac{\boxed{8}}{2\pi \times \boxed{14} \times \boxed{14} \times \boxed{20} \times \boxed{21}} = \underline{\qquad\qquad} \text{ psi}$$

ITEM b) MUST NOT EXCEED 0.5 X $\boxed{10}$

36 TOTAL COMBINED STRESSES

$$S = \sqrt{(\boxed{34} \times \boxed{34}) + (4 \times \boxed{35} \times \boxed{35})} = \underline{\qquad\qquad} \text{ psi}$$

MUST NOT EXCEED $\boxed{11}$

SPRING RATES

37 AXIAL SPRING RATE

$$K_a = \frac{0.431 \times \boxed{17} \times \boxed{13} \times \boxed{20} \times \boxed{20} \times \boxed{20} \times \boxed{21}}{\boxed{22} \times \boxed{15} \times \boxed{15} \times \boxed{15} \times \boxed{28}} = \underline{\qquad\qquad} \text{ lb/in.}$$

38 ROTATION SPRING RATE

$$K_r = 0.5 \times \boxed{17} \times \boxed{17} \times \boxed{37} = \underline{\qquad\qquad} \text{ in. lb/radian}$$

STABILITY

39 FACTOR OF STABILITY

$$F_s = \frac{2\pi \times \boxed{37}}{\boxed{1} \times \boxed{23}} = \underline{\qquad\qquad}$$

MUST BE GREATER THAN 10 FOR COMBINED MOVEMENT
MUST BE GREATER THAN 5 IN CASE OF AXIAL DEFLECTION ONLY

Design Form 2. Ring Reinforced Bellows

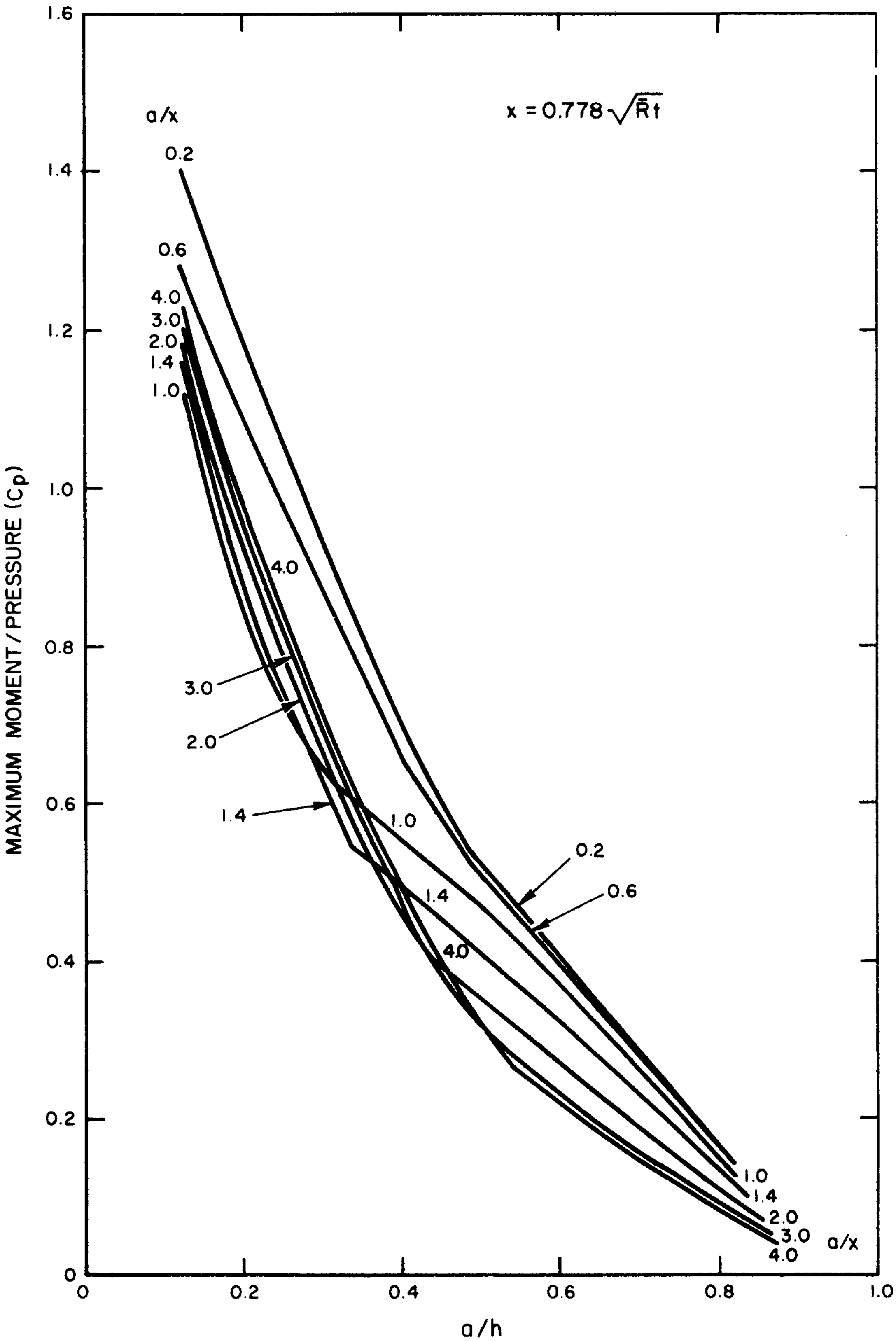


Figure 4. Maximum Moment/Pressure "C_p" for Ring Reinforced Bellows

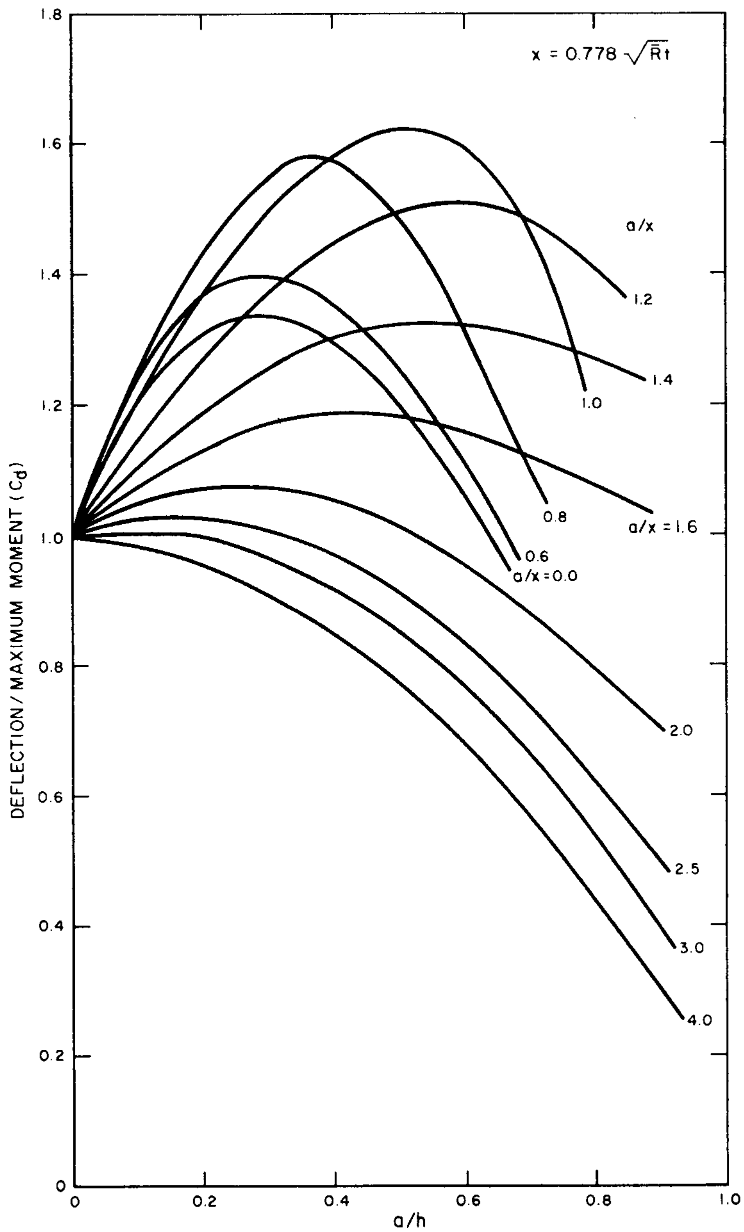


Figure 5. Deflection/Maximum Moment "C_d" for Ring Reinforced Bellows

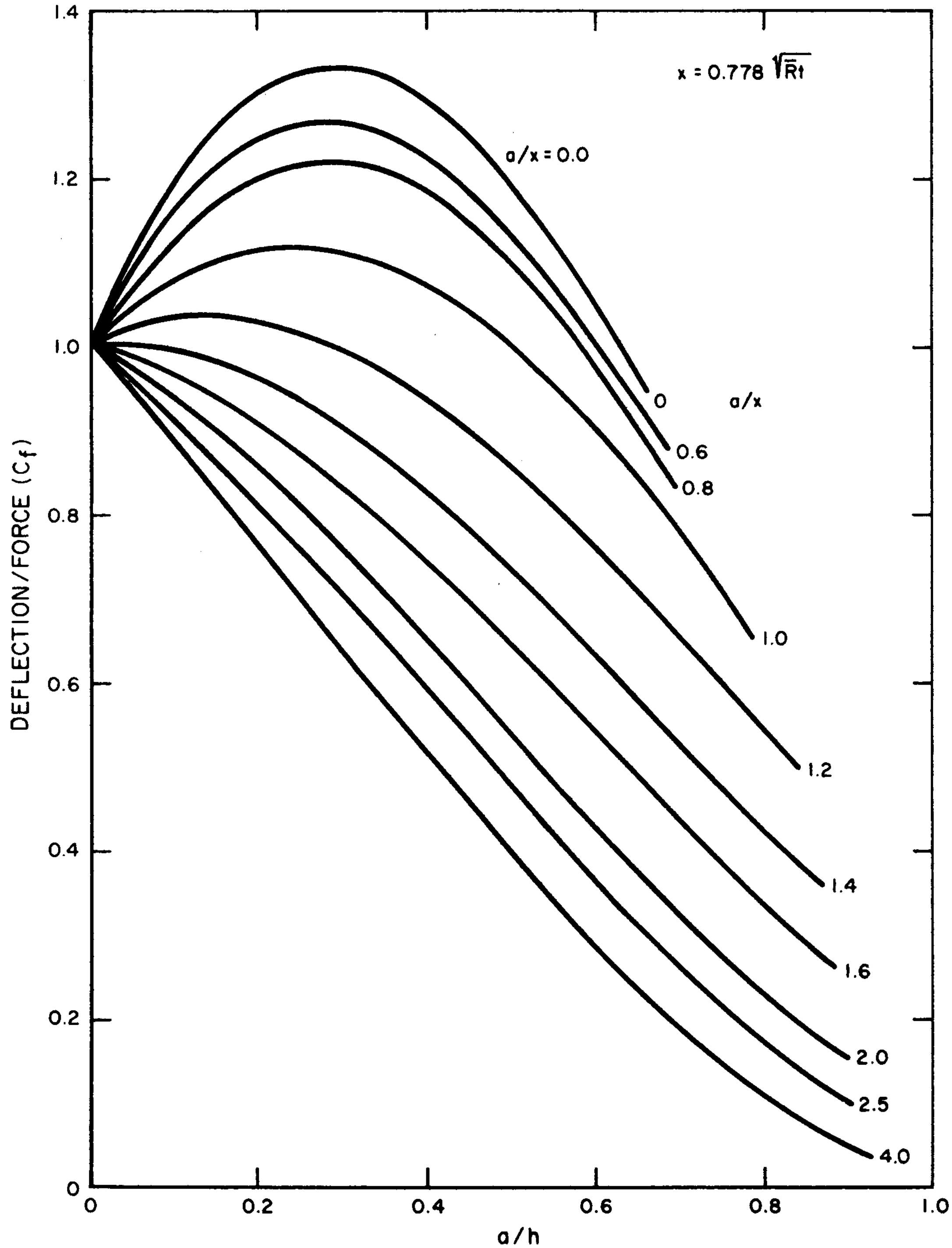


Figure 6. Deflection/Force "C_f" for Ring Reinforced Bellows

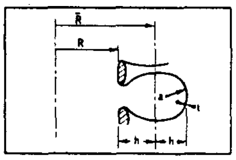
C. TOROIDAL BELLOWS

Toroidal types of bellows considered herein include those of modified toroidal shape where the cross section is not perfectly circular. Toroidal type bellows appear to be applied primarily to situations involving very high operating pressure. Mathematical methods for analysis of stresses have been made available for pure toroidal shapes. These have been confirmed by experiment and are readily applicable. In spite of this, probably for economic reasons, toroidal bellows are seldom used in industry. With the stress limits placed on the application of other types of bellows, toroidal bellows may receive more widespread application.

The theoretical stress analysis for both pressure and deflection stresses has been extended herein to modified toroidal shapes which include a flat annular area between toroidal half-sections. The analysis presented in Part II of this report covers the full range of ratio of flat to toroidal section (a/h). However, for simplicity of analysis and because of the limited range of a/h used in fabricating toroidal bellows as shown by the test results available, the range of a/h covered by Design Form No. 3 is taken from a/h = 0.8 to a/h = 1.0. Within this range of geometry the peak pressure bending stress is very low. The deflection stress range is practically that of a pure toroidal section for this range of a/h. Also, the peak pressure bending stress does not occur in the same area as the peak deflection stress but may be associated with the point of contact with the reinforcing hardware and not considered in the analysis. For these reasons no pressure bending stresses are considered in design analysis of toroidal type bellows and no provision is made for this in the design forms.

In the design forms, thickness requirements are based on pressure membrane stresses. Circumferential membrane stresses in the convolution are not considered significant on the basis of shell theory, but the total circumferential forces induced by pressure must be considered in the design of the reinforcing hardware. The pressure membrane stresses, after being multiplied by the empirically deduced factor of 1.5, are subtracted from the total allowable stress range. The factor of 1.5 compensates for ignoring pressure bending stresses. The required number of convolutions is based on the remaining stress range. The spring rates and stability factors are determined in the design forms as for convoluted bellows. The coefficients used in these calculations, C_d and C_f, from Figures 7 and 8 relate the behavior of a convolution segment to the behavior of a simple strip beam.

DESIGN SHEET FORM 3
TOROIDAL BELLOWS
(REFERENCE FIGURES 7 & 8)



BELLOWS IDENTIFICATION ______

DESIGN CONDITIONS

1 INTERNAL PRESSURE p = ______ psig	2 TEMPERATURE ______ °F	3 NUMBER OF CYCLES ______
4 AXIAL DEFLECTION Δ_a = ______ in.	5 LATERAL TRANSLATION δ ______ in.	6 ANGULAR MOVEMENT θ = ______ radian
7 TORSIONAL ANGULAR ROTATION ALLOWED BY CLEARANCES IN HARDWARE OF RESTRAINTS ϕ = ______ radian	8 TORSIONAL MOMENT OBTAINED BY PIPING STRESS ANALYSIS IF HARDWARE DOES NOT OFFER PROTECTION AGAINST TORSIONAL ROTATION M_t = ______ in. lb	

MATERIAL PROPERTIES

9 ALLOWABLE STRESS AT ROOM TEMPERATURE S_c = ______ psi	10 ALLOWABLE STRESS AT DESIGN TEMPERATURE S_h = ______ psi	11 ALLOWABLE STRESS RANGE FOR COMBINED STRESSES $S_{AC} = 1.25f(S_c + S_h)$ = ______ psi f STRESS RANGE REDUCTION FACTOR
12 YOUNG'S MODULUS AT ROOM TEMPERATURE E_c = ______ psi	13 YOUNG'S MODULUS AT OPERATING TEMPERATURE E_h = ______ psi	

BELLOWS DIMENSIONS AND PARAMETERS (SEE FIGURE ABOVE)

14 R = ______ in.	15 h = ______ in.	16 a = ______ in.
17 $\bar{R} = R + h$ = ______ in.	18	19 a/h = ______ MUST BE 0.8 OR MORE
20 t = ______ in.	21 NUMBER OF PLIES N_p = ______	22 NUMBER OF CONVOLUTIONS N_c = ______
23 OVERALL ACTIVE LENGTH OF BELLOWS L = ______ in.		USE $\boxed{19}$ AND $\boxed{25}$ TO DETERMINE FACTORS , C_d, C_f
24 $x = 0.778\sqrt{\boxed{17} \times \boxed{20}}$ = ______ in.	25 $a/x = \boxed{16} / \boxed{24}$ = ______	
26	27 C_d = ______ FROM FIG. 8	28 C_f = ______ FROM FIG. 9

STRESS CALCULATION

29 MEMBRANE STRESS DUE TO INTERNAL PRESSURE

$$S_{mp} = \frac{\boxed{1} \times \boxed{15}}{\boxed{21} \times \boxed{20}} = \text{______ psi}$$

MUST NOT EXCEED $\boxed{10}$

30

31

32 EQUIVALENT TOTAL AXIAL DEFLECTION

$$\Delta = \boxed{4} + \frac{6 \times \boxed{5} \times \boxed{17}}{\boxed{23}} + 1.2 \times \boxed{17} \times \boxed{6} = \text{______ in.}$$

33 BENDING STRESS DUE TO DEFLECTION

$$S_{bd} = \frac{0.412 \times \boxed{12} \times \boxed{20} \times \boxed{32} \times \boxed{17}}{\boxed{15} \times \boxed{15} \times \boxed{22} \times \boxed{27} \times \boxed{14}} = \text{______ psi}$$

34 COMBINED MEMBRANE AND BENDING STRESSES

$$S_d = 1.5 \times \boxed{29} + 0.5 \times \boxed{33} = \text{______ psi}$$

35 TORSIONAL STRESSES

a) $S_t = \dfrac{\boxed{7} \times \boxed{12} \times \boxed{17}}{10.4 \times \boxed{22} \times \boxed{18}}$ = ______ psi OR b) $S_t = \dfrac{\boxed{8}}{2\pi \times \boxed{14} \times \boxed{14} \times \boxed{20} \times \boxed{21}}$ = ______ psi

ITEM b) MUST NOT EXCEED 0.5 X $\boxed{10}$

36 TOTAL COMBINED STRESSES

$$S = \sqrt{(\boxed{34} \times \boxed{34}) + (4 \times \boxed{35} \times \boxed{35})} = \text{______ psi}$$

MUST NOT EXCEED $\boxed{11}$

SPRING RATES

37 AXIAL SPRING RATE

$$K_a = \frac{0.431 \times \boxed{17} \times \boxed{13} \times \boxed{20} \times \boxed{20} \times \boxed{20} \times \boxed{21}}{\boxed{22} \times \boxed{15} \times \boxed{15} \times \boxed{15} \times \boxed{28}} = \text{______ lb/in.}$$

38 ROTATION SPRING RATE

$$K_r = 0.5 \times \boxed{17} \times \boxed{17} \times \boxed{37} = \text{______ in. lb/radian}$$

STABILITY

39 FACTOR OF STABILITY

$$F_s = \frac{2\pi \times \boxed{37}}{\boxed{1} \times \boxed{23}} = \text{______}$$

MUST BE GREATER THAN 10 FOR COMBINED MOVEMENT
MUST BE GREATER THAN 5 IN CASE OF AXIAL DEFLECTION ONLY

Design Form 3. Toroidal Bellows

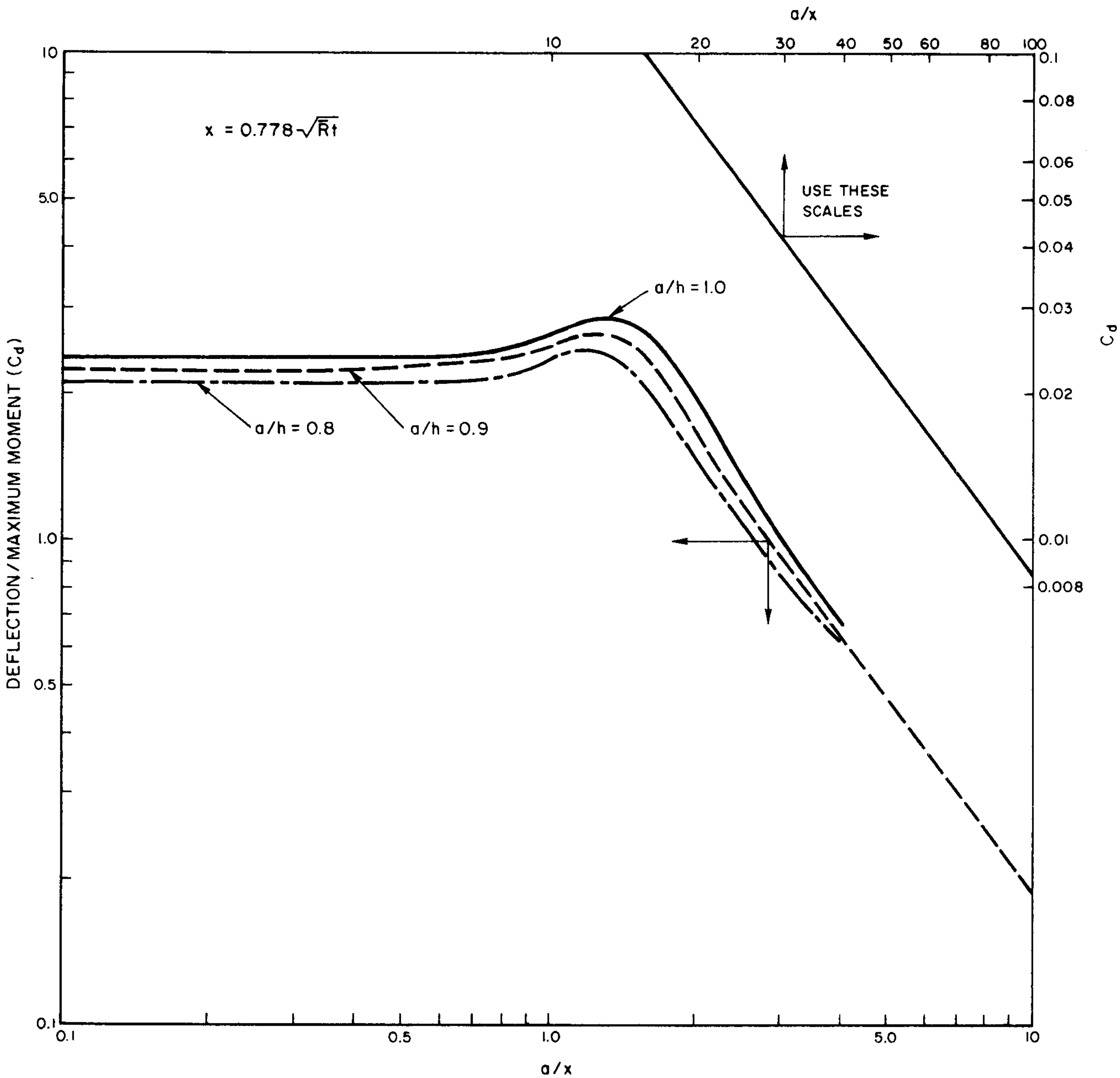


Figure 7. Deflection/Maximum Moment "C_d" for Toroidal Bellows

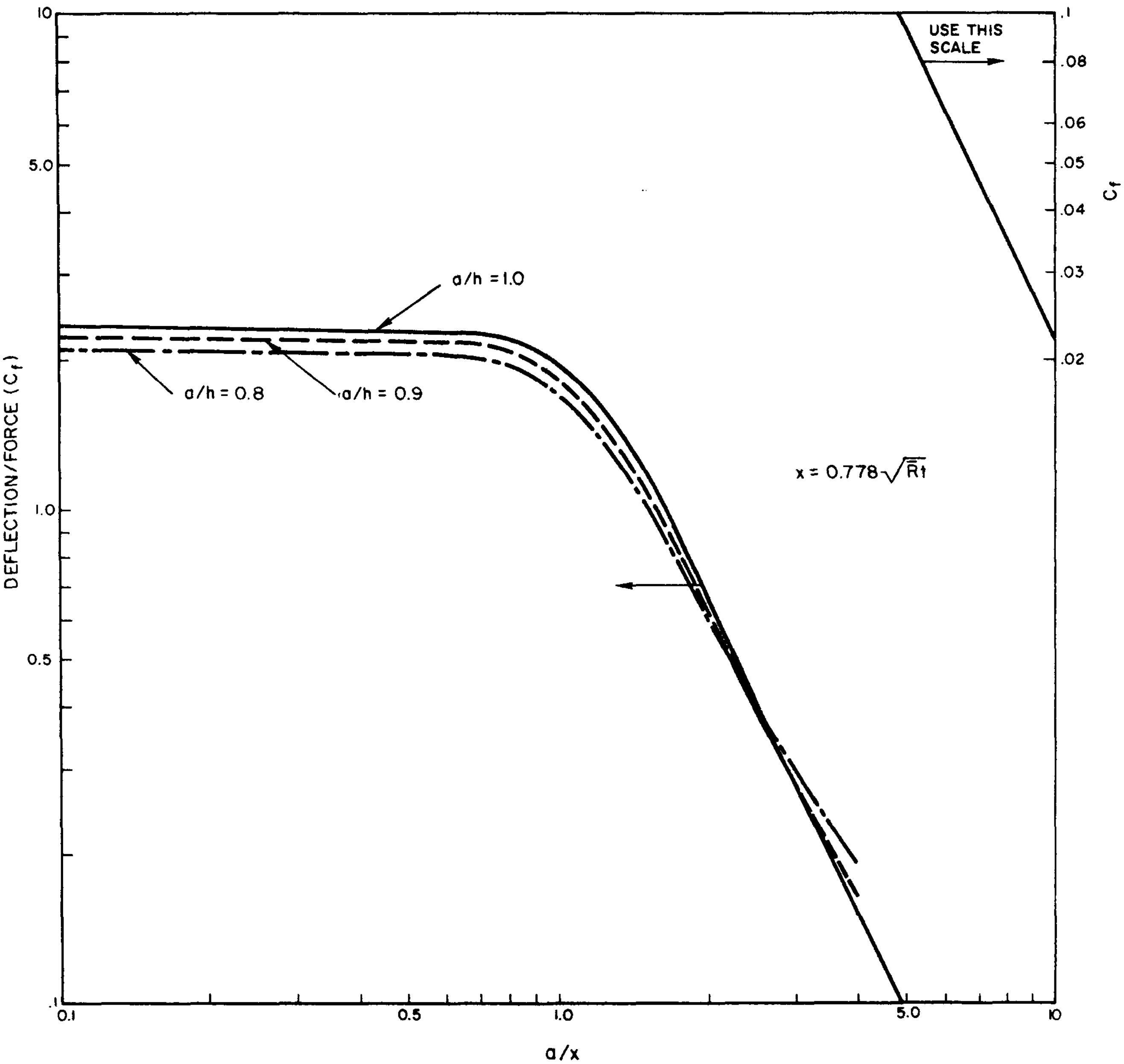


Figure 8. Deflection/Force "C_f for Toroidal Bellows

III. FATIGUE TEST RESULTS AND ALLOWABLE STRESSES

A. CONVOLUTED BELLOWS FATIGUE TEST RESULTS

Available data from fatigue tests on 45 convoluted bellows were analyzed to correlate calculated stresses with cyclic life. These data are presented in Table 1. Results of this analysis are demonstrated graphically in Figures 9 (a, b, c) and 10 (a, b). From comparison of Figures 9a and 9b, it can be observed that pressure stresses have a definite effect on cyclic life. For Figure 9b, peak pressure stresses were considered, as in the design form, to coincide with the peak deflection stress. A better correlation exists between calculated stresses and fatigue life in Figure 9b as compared to Figure 9a.

The tendency of the bellows to squirm due to internal pressure and the resulting effect upon bending stresses is considered in Appendix A. The increase in bending stresses due to this tendency is limited to bellows undergoing angular displacement. The effect of this increase is measured as a factor (C_{ps}) which is a function of the ratio of operating pressure to critical pressure. This effect is included in the calculated stresses of Figure 9c and appears to result in even closer correlation between calculated stresses and fatigue life.

It must be pointed out that these calculated stresses exceed the yield strength by an order of magnitude and therefore must be considered as theoretical elastic stresses. A correlation coefficient of 0.74 was found between the combined theoretical elastic stresses and fatigue life for the 45 bellows tested.

To provide a logical basis for considering the effect of pressure in design, a statistical evaluation was made of results from tests with and without pressure. The best method of performing such an evaluation would be to perform a non-linear multiple regression analysis of the data available. As explained in Appendix B, this would require a first estimate of the effect of pressure which would later be corrected with linearized correction forms. The following technique provided a first estimate which was deemed suitable. Since the exact value found was not used, there was little to be gained from making small corrections to the approximate value. First, the results were selected from tests of 24 bellows wherein the pressure bending stresses were zero or less than 15% of the deflection stress. Double the existing pressure membrane and pressure bending stresses, for those bellows with these lower pressures, were added to the deflection stress. These results were analyzed using techniques of statistical mathematics to fit a curve which would relate cyclic deflection bending stress to cyclic life, with least squared error. The fitted curve, a straight line log-log relation given in Figure 10a, was then taken as a basic fatigue curve. This relation could be expressed as

$$\mathrm{Log}_{10}\, S = 6.36 - 0.274\ \mathrm{Log}_{10}\, N \qquad \ldots(1)$$

where

S = stress

N = number of cycles to failure.

Remaining data from tests on 21 additional bellows, which were subjected to substantial pressure bending stresses in addition to deflection bending stresses, were then utilized. These data were fitted to the basic fatigue curve by adjusting the pressure bending stresses by an unknown factor, to obtain the factor for fit of least error. The mathematicians who were consulted deduced that the factor for a fit of least squared error could be directly derived. This was done as shown in Appendix B. Results of this effort showed that, when a factor of 1.726 is applied to the pressure stresses before adding them to the deflection stresses, the combined stress - cyclic life relation fits the basic fatigue curve with the least squared error. On this basis, the factor was taken as 2.0 in the calculation forms and specifications. With the factor of 2.0, the fitted curve of least squared error was calculated for the combined stresses and fatigue life for all 45 bellows and is illustrated in Figure 10b. This relation can also be expressed as

$$\mathrm{Log}_{10}\, S = 6.24 - 0.236\ \mathrm{Log}_{10}\, N \qquad \ldots(2)$$

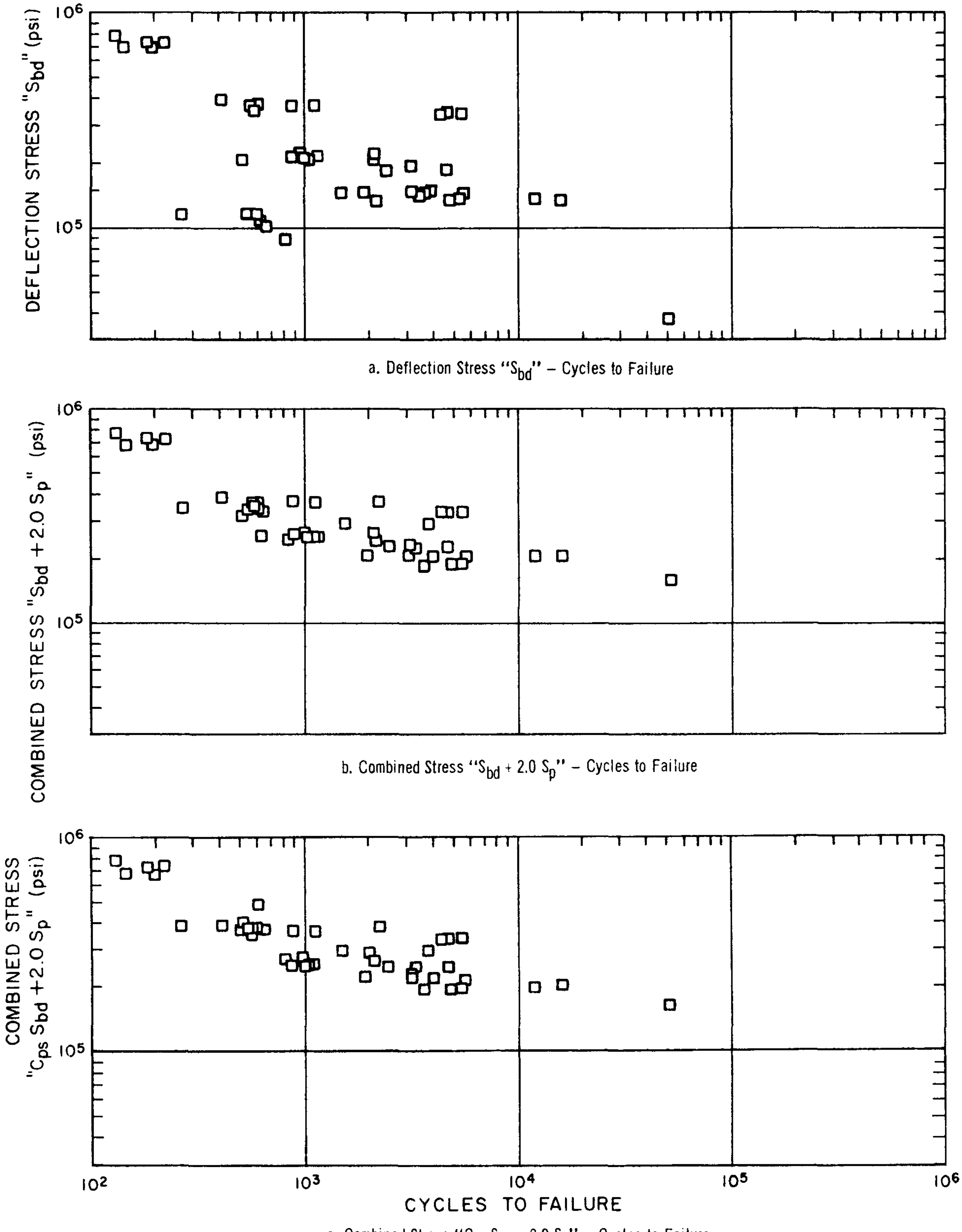
Comparison of these two relations at 10^4 cycles indicates little deviation resulting from including pressure stresses.

The results also indicate that a factor of at least four exists between ASA Allowable Stresses for 7000 cycles and the average of the calculated value of stress which induced failure. The minimum difference from calculated stress value in Figure 9c is a factor of 3.35 above the ASA Code allowable stress for that number of cycles. For this reason a proposed theoretical allowable deflection stress for convoluted bellows could be two times the experimentally established value allowed by the ASA code for piping elbows or, the actual stress concentration factor could be taken as one-half the theoretical stress concentration factor. It should be noted also, that the factor of 2.0 on pressure stresses was used in combination with the theoretical elastic deflection stresses. Dividing all the theoretical stresses by 2.0 yields an effective stress of

$$S_{mp} + S_{bp} + 0.50\, S_{bd} = S \qquad \ldots(3)$$

which should not be greater than the ASA code allowable.

As in the piping code, the pressure stresses (both membrane and bending) for bellows are subtracted from the total allowable stress; the remainder is the allowable bending stress due to deflection. Overall, this is akin to the technique used to calculate stresses in piping elbows where the ASA code stress concentration factor is taken as one-half of the theoretical stress concentration factor and the pressure stresses are subtracted from the total allowable stress. Modifying the allowable stress for bellows in this manner still provides a safety factor of 1.67 below the minimum data points. This is considered sufficiently conservative, since (1) bellows stresses were calculated on the basis of nominal dimensions, (2) the bellows were procured from several manufacturers in a wide range of sizes and material thickness, and (3) the bellows were tested under a wide range of pressure and deflection conditions.



a. Deflection Stress "S_{bd}" – Cycles to Failure

b. Combined Stress "$S_{bd} + 2.0\ S_p$" – Cycles to Failure

c. Combined Stress "$C_{ps}\ S_{bd} + 2.0\ S_p$" – Cycles to Failure

Figure 9. Convoluted Bellows Fatigue Test Results

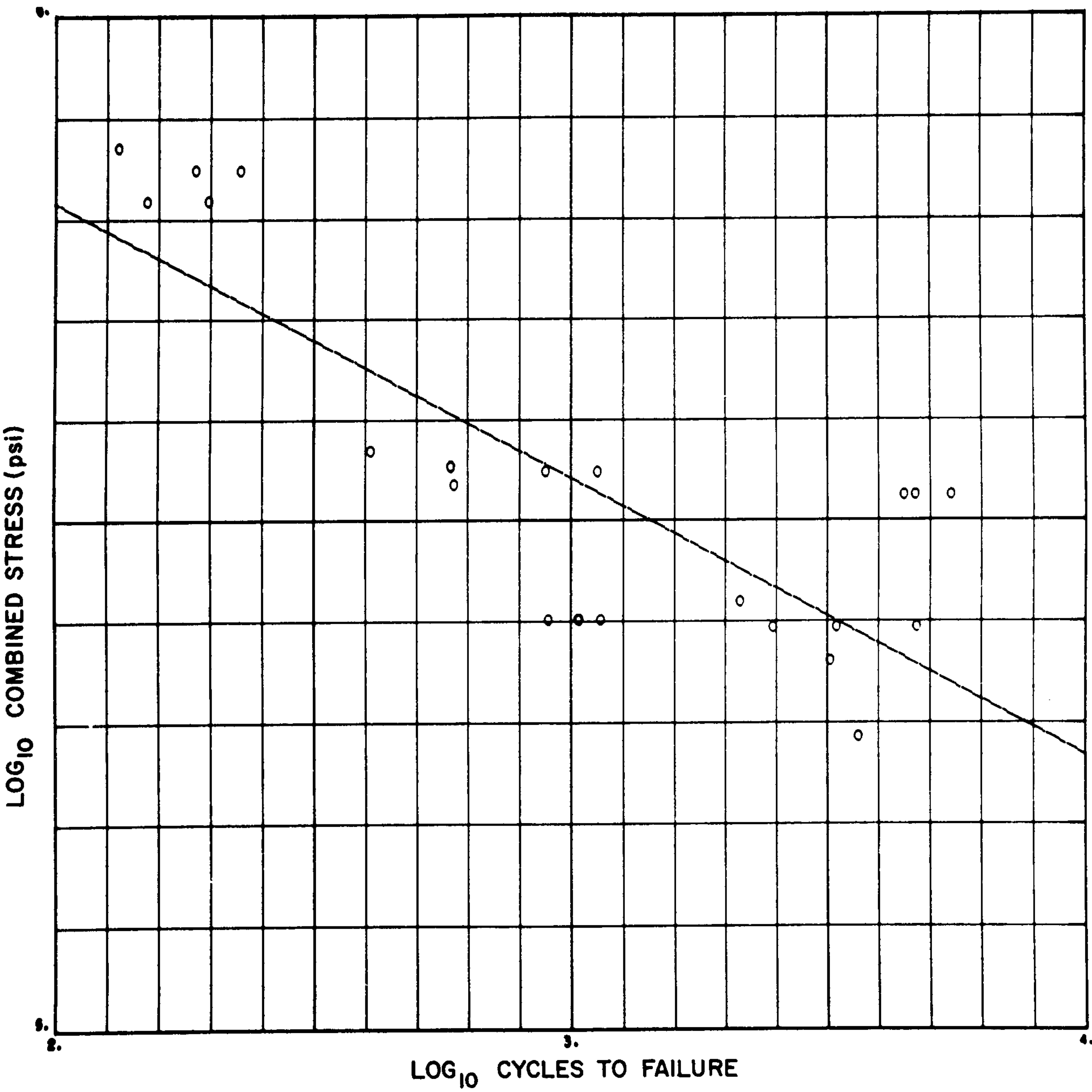


Figure 10a. Convoluted Bellows Fatigue Test Results
Best Fit Curve $S_p \approx 0$

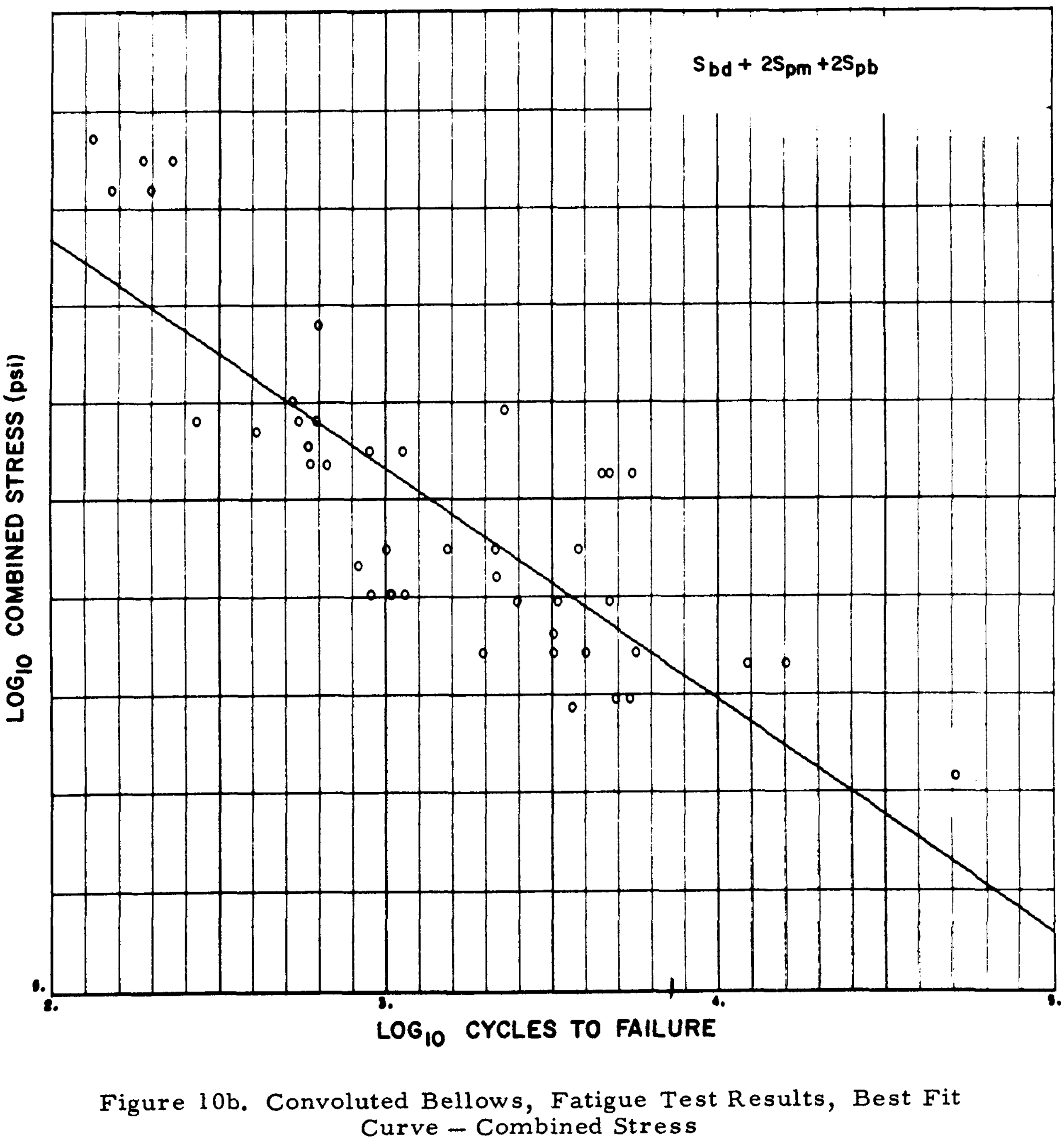


Figure 10b. Convoluted Bellows, Fatigue Test Results, Best Fit Curve — Combined Stress

All Bellows Type 321 SS Test Temperature ~70°F

1 Axial Defl. (in.)	2 Angular Defl. Degrees of Arc	3 Trans. Defl. (in.)	4 Test Press (psi)	5 $\overline{R}$ (in.)	6 a (in.)	7 h (in.)	8 t (in.)	9 L (in.)	10 N_p	11 N_c	12 K_a Meas. (lb/in.)	13 K_a Calc. (lb/in.)	14 K_r Meas. (in.lb/deg)	15 K_r Calc. (in.lb/deg)	16 Exper. Critical Pres. (psi)	17 Critical Pres. Calc. (psi) q_{cr}	18 Pres. Mem. Stress (psi) S_{mp}	19 Pres. Bending Stress (psi) S_{bp}	20 Deflec. Bending Stress (psi) S_{bd}	21 Cycles to Failure
-	0-5°	-	1795	4.79	0.082	0.255	0.020	2.62	4	8	-	-	104	1860		22,400	5,730	114,000	133,000	2,265
-	±10°	-	-	1.11	0.0425	0.140	0.020	1.87	2	11	-	-	26	52	-	-	-	-	336,000	4,698
-	±10°	-	-	1.11	0.0425	0.140	0.020	1.87	2	11	-	-	38	52	-	-	-	-	336,000	4,456
-	±10°	-	-	1.11	0.0425	0.140	0.020	1.87	2	11	-	-	24	52	-	-	-	-	336,000	5,515
0.80	-	0.70	-	6.375	0.074	0.370	0.020	8.00	1	27	221	325	-	-	-	2,560	640	19,200	212,000	1,133
0.80	-	0.70	-	6.375	0.074	0.370	0.020	8.00	1	27	218	325	-	-	-	2,560	640	19,200	212,000	896
0.80	-	0.70	-	6.375	0.074	0.370	0.020	8.00	1	27	257	325	-	-	-	2,560	640	19,200	212,000	1,032
0.80	-	0.70	-	6.375	0.074	0.370	0.020	8.00	1	27	228	325	-	-	-	2,560	640	19,200	212,000	1,026
-	±3°	-	100	8.925	0.100	0.440	0.031	4.00	1	10	-	-	-	-	-	4,440	1,420	34,900	137,000	16,065
-	±3°	-	100	8.925	0.100	0.440	0.031	4.00	1	10	-	-	-	-	-	4,440	1,420	34,900	137,000	12,250
-	±7-1/2°	-	1160	2.25	0.075	0.220	0.020	6.00	3	20	1070	2050	475	90.7	-	2,140	4,230	71,200	104,000	620
-	±7°	-	1200	2.18	0.057	0.155	0.020	4.10	2	18	7530	4060	312	168	-	6,220	4,650	54,100	201,000	520
-	±4.3°	-	550 400	2.90	0.160	0.286	0.0125	3.2	3	5	1400	1535	125	112.5	-	3,020	3,050	73,500	88,400	823
-	±9.25°	-	100	9.46	0.186	0.900	0.040	7.44	3	10	1640	2270	-	-	-	1,920	750	29,200	148,000	5,700
-	±9.25°	-	100	9.46	0.186	0.900	0.040	7.44	3	10	-	-	-	-	-	1,920	750	29,200	148,000	4,000
-	±9.25°	-	100	9.46	0.186	0.900	0.040	7.44	3	10	-	-	-	-	-	1,920	750	29,200	148,000	1,948
-	±9.25°	-	100	9.46	0.186	0.900	0.040	7.44	3	10	-	-	-	-	-	1,920	750	29,200	148,000	3,200
-	±13°	-	100	9.55	0.186	1.00	0.050	7.44	3	10	-	-	1560 2370-100	2660	-	2,820	670	23,400	220,000	2,125
-	±13°	-	100	9.55	0.186	1.00	0.050	7.44	3	10	-	-	1560 2370-100	2660	-	2,820	670	23,400	220,000	1,000
±0.125	-	-	1050	1.79	0.055	0.150	0.016	2.19	2	10	-	-	-	-	-	9,800	4,930	68,900	146,000	1,528
±0.125	-	-	1050	1.79	0.055	0.150	0.016	2.19	2	10	-	-	-	-	-	9,800	4,930	68,900	146,000	3,807
-	±9°	-	37	4.38	0.054	0.306	0.010	2.17	3	10	316	430	53.7	72	-	1,242	377	20,500	144,000	3,650
-	±0.14°	-	37	4.38	0.060	0.309	0.012	3.35	2	14	344	339	54.2	58	-	636	475	21,600	186,000	2,467
-	±0.14°	-	37	4.38	0.060	0.309	0.012	3.35	2	14	329	339	45	58	-	636	475	21,600	186,000	3,300
-	±0.14°	-	37	4.38	0.060	0.309	0.012	3.35	2	14	300	339	45.5	58	-	636	475	21,600	186,000	4,711
-	0-8.5°	-	300	2.18	0.067	0.191	0.012	1.875	1	7	625	635	-	-	-	2,124	4,780	111,300	115,200	270
-	0-8.5°	-	300	2.18	0.067	0.191	0.012	1.875	1	7	-	-	-	-	-	2,124	4,780	111,300	115,200	614
-	0-8.5°	-	300	2.18	0.067	0.191	0.012	1.875	1	7	742	635	28.3	26.4	-	2,124	4,780	111,300	115,200	543
-	0-7.5°	-	300	2.18	0.067	0.191	0.012	1.875	1	7	1483	635	61	26.4	-	2,124	4,780	111,300	101,500	660
±2.25	2°Cont.	-	37	4.38	0.068	0.305	0.010	5.48	3	20	189	207	-	-	-	218	376	19,300	195,000	3,201
-	±14°	-	37	4.38	0.064	0.302	0.0116	3.33	2	13	382	348	68	58	-	650	482	21,700	201,000	2,131
-	±10°	-	100	7.07	0.125	1.02	0.050	5.00	3	10	-	-	-	-	-	3,150	680	25,900	137,000	5,462
-	±10°	-	100	7.07	0.125	1.02	0.050	5.00	3	10	3750	2500	1930	1100	-	3,150	680	25,900	137,000	4.917
-	±3°	-	100	6.80	0.125	0.790	0.025	4.00	3	8	2200	775	-	-	-	1,216	1,050	59,300	37,600	51 462
6.00	-	-	-	6.062	0.234	0.662	0.050	7.5	1	8	1550	2760	-	-	440	2,300	-	-	708,000	226

TABLE 1

CONVOLUTED BELLOWS FATIGUE TEST DATA

	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
Axial Defl. (in.)	Angular Defl. Degrees of Arc	Trans. Defl. (in.)	Test Press (psi)	$\bar{R}$ (in.)	a (in.)	h (in.)	t (in.)	L (in.)	N_p	N_c	K_a Meas. (lb/in.)	K_a Calc. (lb/in.)	K_r Meas. (in.lb/deg)	K_r Calc. (in.lb/deg)	Exper. Critical Pres. (psi)	Critical Pres. Calc. (psi) q_{cr}	Pres. Mem. Stress (psi) S_{mp}	Pres. Bending Stress (psi) S_{bp}	Deflec. Bending Stress (psi) S_{bd}	Cycles to Failure
6.00				6.062	0.234	0.662	0.050	7.5	1	8	1710	2760			440				708,000	185
3.00				6.062	0.234	0.662	0.050	7.5	1	8	1750	2760			440				354,000	890
3.00				6.062	0.234	0.662	0.050	7.5	1	8	1600	2760			440				354,000	1,120
5.50				5.032	0.234	0.662	0.050	7.5	1	8	1570	2150			465	1,830			660,000	149
5.50				5.032	0.234	0.662	0.050	7.5	1	8	1580	2150			465	1,830			660,000	196
3.00				5.032	0.234	0.662	0.050	7.5	1	8	1300	2150			465	1,830			359,000	580
3.00				5.032	0.234	0.662	0.050	7.5	1	8	1260	2150			465	1,830			359,000	580
2.00				2.702	0.234	0.419	0.036	7.5	1	8	1210	1970			655	1,640			344,000	589
2.00				2.702	0.234	0.419	0.039	7.5	1	8	1440	2390			655	2,000			372,000	405
4.00				2.702	0.234	0.419	0.039	7.5	1	8	1430	2390			655	2,000			746,000	131

TABLE 1 (Continued)

B. RING REINFORCED BELLOWS FATIGUE TEST RESULTS

Available data from fatigue tests on 43 reinforced bellows (Table 2) were analyzed to correlate calculated stresses with cyclic life. One of these bellows was tested at 1050°F; all others were tested at room temperature. Stresses for one bellows were normalized to room temperature by multiplying by the ratio of code allowable stress at room temperature, to that at 1050°F. Results of this analysis are presented graphically in Figures 11a, b, and 12a, b. In analyzing these data, results of two tests where the bellows did not fail were treated as though the bellows had failed when the test was stopped. From comparison of Figure 11a with Figure 11b it can be observed that, as with convoluted bellows, pressure bending stresses have an effect upon the cyclic life of reinforced bellows. The combined stresses of Figure 11b include the effect of internal pressure upon the deflection stresses wherein the tendency to squirm increases the deflection stresses. Again, as with the convoluted bellows, the total combined calculated stresses far exceed the elastic limit of the material and must be considered as fictitious or theoretical elastic stresses. A correlation coefficient of 0.89 was found between the combined theoretical elastic stresses and fatigue life for the 43 bellows tested.

The same technique that was used for convoluted bellows, presented in Appendix A, was used to determine the effect of pressure stresses on cyclic life. Of the 43 ring reinforced bellows which were fatigue tested, 27 were tested with zero pressure differential between the inside and the outside of the bellows. A log-log relation with least squared error between calculated stress and cyclic life was determined for these 27 bellows as the basic fatigue curve and is shown in Figure 12a. This relation can also be expressed as

$$\text{Log}_{10} S = 6.41 - 0.284 \ \text{Log}_{10} N \quad . \qquad \ldots (4)$$

Data from tests of the remaining 16 bellows were then fitted to this curve using an unknown factor applied to pressure bending stresses. The combined stresses utilizing this factor should fit the predicted stresses from the basic fatigue curve with least squared error. The value of this unknown factor was determined directly as 0.471 and is taken in the design forms and specifications as 0.50. With the factor taken as 0.50, the fitted curve of least squared error was calculated for combined stresses and cyclic life of all 43 bellows and is shown in Figure 12b. This relation can also be expressed as

$$\mathrm{Log}_{10}S = 6.26 - 0.249\ \mathrm{Log}_{10}N \quad . \qquad \ldots (5)$$

Comparison of this with the first relation in the region of 10^4 cycles indicates little deviation resulting from the inclusion of pressure stresses with the factor of 0.50. It should be noted that this value of 0.50 is used with the theoretical elastic bending stresses.

The value of this factor, however, is strongly dependent on the results of 8 of the 16 tests which were performed with the highest pressure stresses. By omitting selected bellows from this set of 8 and using the remainder of the 16, this factor can be varied from 0.2 to greater than 1.0. This indicates that more test data are required, especially from tests at high pressure, and improved analytical techniques should be applied in order to verify the use of the factor of 0.5 in considering the effect of bending stresses induced by pressure.

Again, as with the convoluted bellows, the difference between the allowable stress range of the ASA Code for pressure piping and the theoretical elastic stress for a given value of cyclic life, is approximately 4.0. Considering the effective stress as one-half of the theoretical elastic stress, a safety factor of 2.0 on the mean would result. The effective stress would be computed as

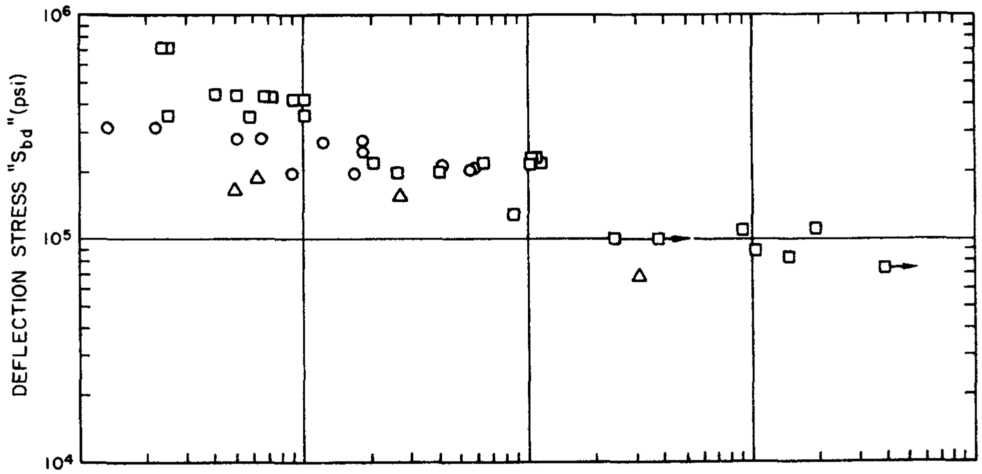
$$S = S_{mp} + 0.25\ S_{bp} + 0.5\ S_{bd} \quad , \qquad \ldots (6)$$

which should not be greater than the ASA Code allowable.

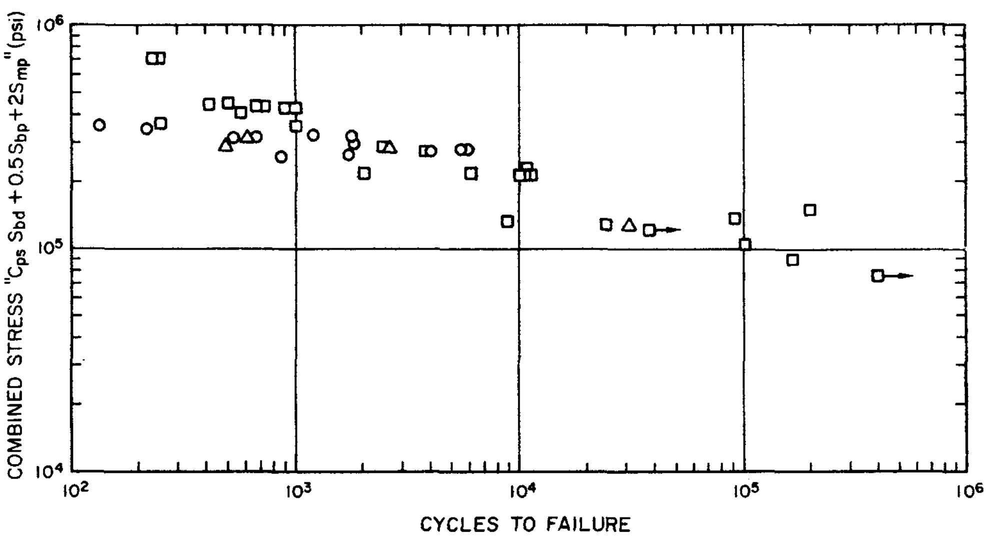
Table 2 also contains data on measured spring rates and critical pressure compared to theoretical values of each. Again, substantial differences appear between theoretical and measured values. In the case of axial spring rate, measured values are consistently lower than theoretical by a factor as great as 2.0. Differences between theoretical and measured values of axial spring rate could be explained on the basis of the analytical model used for stress analysis, wherein it is assumed that the inner crest radius is entirely in contact with the reinforcing rings and does not enter into the flexibility of the convolutions. This could and probably would be the situation under high pressures. However, all the spring rate measurements quoted were obtained at zero pressure differential so that the greater flexibility might be expected.

The measured value of critical pressure (squirm pressure) is also lower than the theoretical value by a factor as great as 6 for bellows subjected to pressure while highly strained above the yield stress with lateral offset deflection. The values of measured critical pressure range from 81 to 100% of the theoretical value for bellows with fully restrained ends. Even this small difference might be expected since the measured values are based on achieving a complete theoretical buckling load. This theoretical buckling load might be impossible to achieve on a test due to inelastic behavior of the material. Results of the tests on bellows with restrained ends indicate that the analytical model may be sufficiently correct to allow reasonably accurate prediction of spring rate and critical pressure under the proposed limited allowable operating conditions of pressure and deflection. In the meantime the factor of 10 required on stability pressure does not appear excessive.

More data are needed from results of tests conducted under various pressure and deflection conditions. An improvement to the mathematical model is necessary before analytical design for spring rate and critical pressure under critical conditions can be performed accurately for reinforced bellows. The tendency for reinforced bellows to distort to the shape of toroidal bellows under very high pressure is obvious from photographs of high pressure tests. This is not accounted for in the theory and may partially account for those cases where measured critical pressure is substantially above that predicted herein.



a. Deflection Stress "S_{bd}" - Cycles to Failure



b. Combined Stress "C_{ps} S_{bd} + 0.5 S_{bp} + 2.0 S_{mp}" - Cycles to Failure

Figure 11. Ring Reinforced Bellows Fatigue Test Results

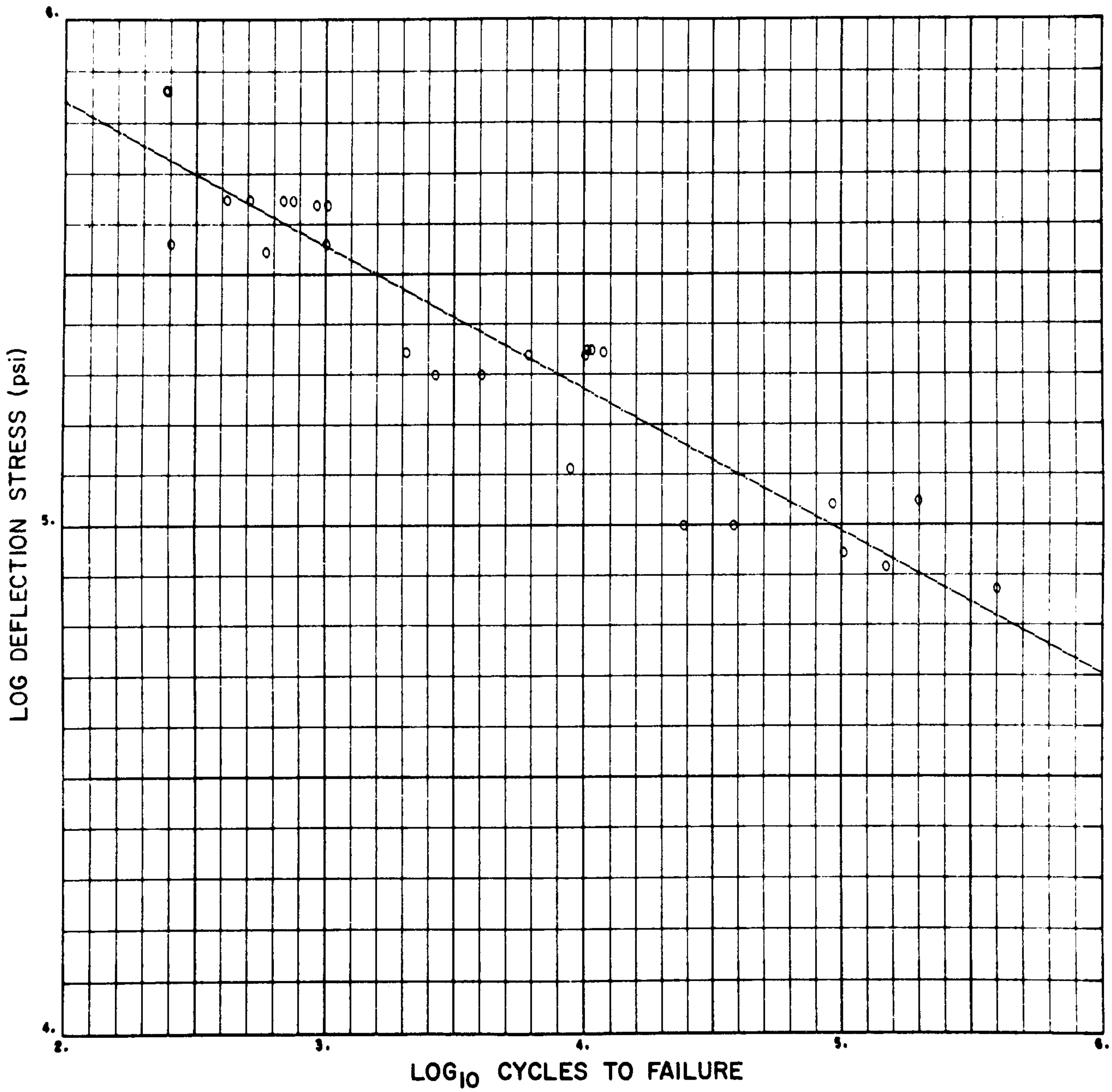


Figure 12a. Ring Reinforced Bellows Fatigue Test Results, Best Fit Curve $S_p \approx 0$

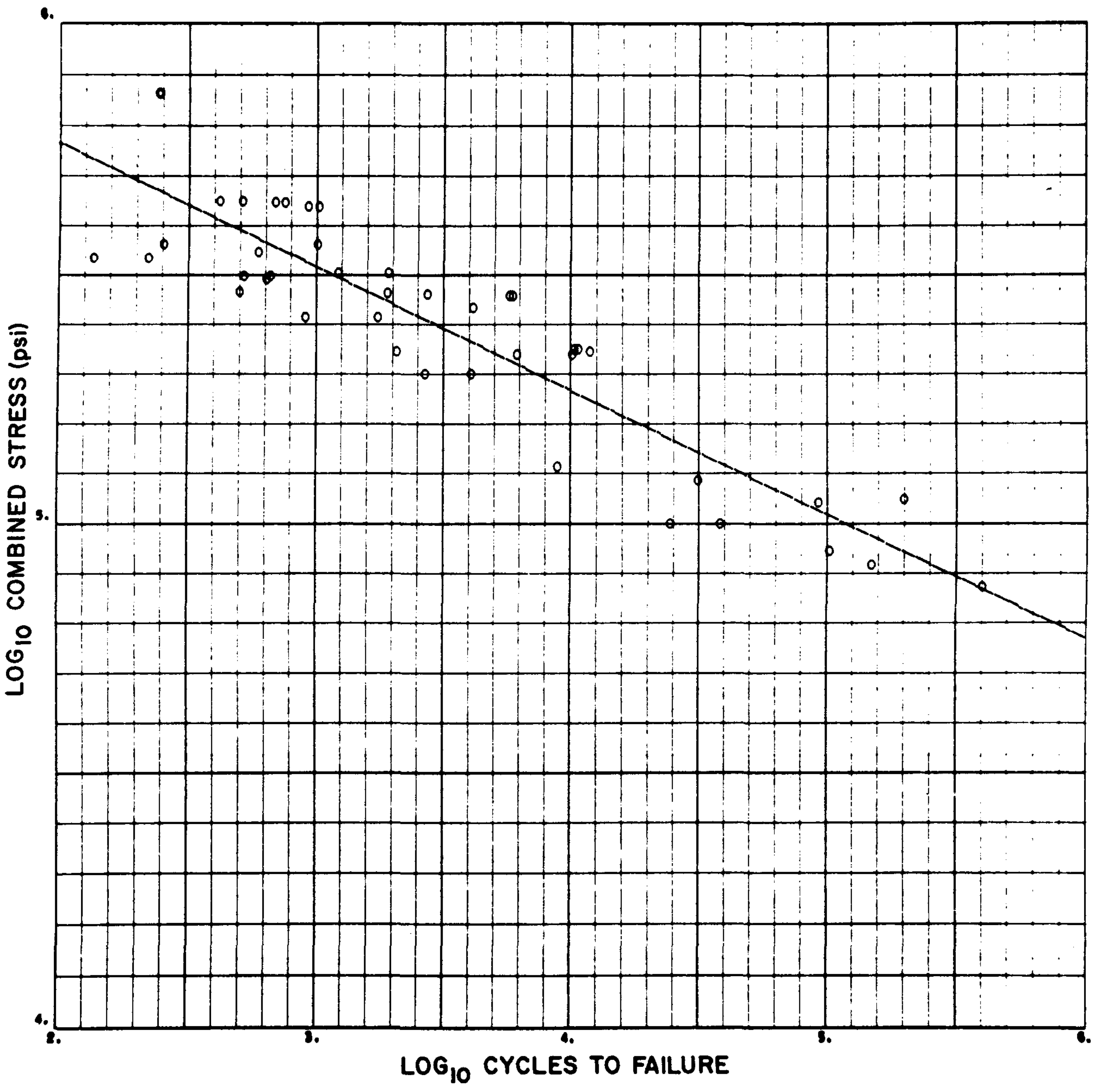


Figure 12b. Ring Reinforced Bellows Fatigue Test Results, Best Fit Curve – Combined Stress

	1 Axial Defl. (in.)	2 Angular Defl. Degrees of Arc	3 Trans. Defl. (in.)	4 Test Pres. (psi)	5 $\bar{R}$ (in.)	6 a (in.)	7 h (in.)	8 t (in.)	9 L (in.)	10 N_p	11 N_c	12 K_a Meas. (lb/in.)	13 K_a Calc. (lb/in.)	14 K_r Meas. (in.lb/Rad)	15 K_r Calc. (in.lb/Rad)	16 Exper. Critical Pres. (psi)	17 Critical Pres. Calc. (psi) q_{cr}	18 Pres. Mem. Stress (psi) S_{mp}	19 Pres. Bending Stress (psi) S_{bp}	20 Deflec. Bending Stress (psi) S_{bd}	21 Cycles to Failure
1	1.0	-	-	150	19.3	0.53	1.3	0.062	8.5	1	4	-	-	-	-	-	-	3,120	91,200	70,000*	31,511
2	.63	-	.818	520	6.12	0.25	0.625	0.050	10.0	1	10	-	-	-	-	-	-	6,500 9,400*	114,000 163,000*	109,000 156,000*	2,700
3	-	±14°	-	150	4.50	0.0835	0.25	0.012	6.68	3	20	-	-	-	-	-	-	1,040	36,100	192,000 290,000	630
4	1.5	-	-	-	3.97	0.234	0.61	0.031	-	1	3	-	-	-	-	-	-	-	-	410,000	580
5	0.375	-	-	-	3.97	0.234	0.61	0.031	-	1	3	-	-	-	-	-	-	-	-	88,000	102,192
6	0.75	-	-	-	13.0	0.40	1.0	0.050	-	1	3	-	-	-	-	-	-	-	-	138,000	91,628
7	0.75	-	-	-	19.3	0.531	1.30	0.063	6.375	1	3	-	-	-	-	-	-	-	-	82,500	148,472
8	0.75	-	-	-	7.88	0.332	0.88	0.038	-	1	3	-	-	-	-	-	-	-	-	122,000	38,213
9	1.50	-	-	-	7.88	0.332	0.88	0.038	-	1	3	-	-	-	-	-	-	-	-	200,000	2,639
10	1.50	-	-	-	7.88	0.332	0.88	0.038	-	1	3	-	-	-	-	-	-	-	-	200,000	4,008
11	0.375	-	-	-	4.03	0.234	0.672	0.031	2.81	1	3	-	-	-	-	-	-	-	-	87,500	400,000
12	-	±6°	-	350	4.03	0.234	0.672	0.031	2.81	1	3	1100	1320	-	-	2,150	2,950	7,000	154,000	165,000 202,000	493
13	0.563	-	-	-	4.03	0.234	0.672	0.031	2.81	1	3	-	-	-	-	-	-	-	-	111,500	197,463
14	0.875	-	-	-	4.03	0.234	0.672	0.031	3.75	1	4	-	-	13,900	8,100	2,150	1,670	-	-	130,000	8,777
15	1.00	-	-	-	7.88	0.332	0.88	0.038	5.31	1	4	-	-	-	-	1,700	2,130	-	-	100,000	24,383
16	3.00	-	-	0-150	7.06	0.56	1.063	0.054	12.8	1	4	-	-	-	-	-	-	2,960	49,000	312,000	134
17	3.00	-	-	0-150	7.06	0.56	1.063	0.054	12.8	1	4	-	-	-	-			2,960	49,000	312,000	220
18	3.00	-	-	0-150	7.19	0.56	1.188	0.062	14.3	1	4	-	-	-	-	-	-	2,870	53,000	282,000	656
19	3.00	-	-	0-150	7.19	0.56	1.188	0.062	14.3	1	4	-	-	-	-	-	-	2,870	53,000	282,000	515
20	3.00	-	-	0-150	7.13	0.56	1.312	0.048	12.6	1	4	-	-	-	-	-	-	4,100	110,000	197,000	893
21	3.00	-	-	0-150	7.13	0.56	1.312	0.048	12.6	1	4	-	-	-	-	-	-	4,100	110,000	197,000	1,715
22	3.00	-	-	0-150	7.19	0.42	1.188	0.060	10.7	1	4	-	-	-	-	-	-	2,960	77,600	275,000	1,207
23	3.00	-	-	0-150	7.19	0.42	1.188	0.060	10.7	1	4	-	-	-	-	-	-	2,960	77,600	275,000	1,886
24	3.00	-	-	0-150	7.25	0.56	1.25	0.049	12.4	1	4	-	-	-	-	-	-	3,820	93,600	217,000	4,095
25	3.00	-	-	0-150	7.25	0.56	1.25	0.058	12.4	1	4	-	-	-	-	-	-	3,240	68,400	249,000	1,875
26	3.00	-	-	0-150	7.06	0.45	1.063	0.034	9.45	1	4	-	-	-	-	-	-	4,700	146,500	204,000	5,846
27	3.00	-	-	0-150	7.06	0.45	1.063	0.034	9.45	1	4	-	-	-	-	-	-	4,700	146,500	204,000	5,713
28	6.00	-	-	-	6.062	0.234	0.672	0.031	7.5	1	8	500	715	-	-	Offset 238	598	-	-	443,000	740
29	6.00	-	-	-	6.062	0.234	0.672	0.031	7.5	1	8	500	715	-	-	Fixed 600	598	-	-	443,000	678
30	3.00	-	-	-	6.062	0.234	0.672	0.031	7.5	1	8	530	715	-	-		598	-	-	221,000	2,053
31	3.00	-	-	-	6.062	0.234	0.672	0.031	7.5	1	8	490	715	-	-	-	598	-	-	221,000	11,875
32	6.00	-	-	-	5.032	0.234	0.672	0.031	7.5	1	8	440	598	-	-	Offset 190	501	-	-	435,000	1,008
33	6.00	-	-	-	5.032	0.234	0.672	0.031	7.5	1	8	380	598	-	-	Fixed 435	501	-	-	435,000	913
34	3.00	-	-	-	5.032	0.234	0.672	0.031	7.5	1	8	410	598	-	-	-	501	-	-	218,000	6,098
35	3.00	-	-	-	5.032	0.234	0.672	0.031	7.5	1	8	420	598	-	-	-	501	-	-	218,000	10,008

*Corrected to compensate for 1050°F test temperature

TABLE 2

RING REINFORCED BELLOWS FATIGUE TEST DATA

	1 Axial Defl. (in.)	2 Angular Defl. Degrees of Arc	3 Trans. Defl. (in.)	4 Test Pres. (psi)	5 $\bar{R}$ (in.)	6 a (in.)	7 h (in.)	8 t (in.)	9 L (in.)	10 N_p	11 N_c	12 K_a Meas. (lb/in.)	13 K_a Calc. (lb/in.)	14 K_r Meas. (in.lb/Rad)	15 K_r Calc. (in.lb/Rad)	16 Exper. Critical Pres. (psi)	17 Critical Pres. Calc. (psi) q_{cr}	18 Pres. Mem. Stress (psi) S_{mp}	19 Pres. Bending Stress (psi) S_{bp}	20 Deflec. Bending Stress (psi) S_{bd}	21 Cycles to Failure
36	6.00				4.032	0.234	0.672	0.031	7.5	1	8	320	495			Offset 304	415			446,000	502
37	6.00				4.032	0.234	0.672	0.031	7.5	1	8	300	495			Fixed 350	415			446,000	414
38	3.00				4.032	0.234	0.672	0.031	7.5	1	8	280	495			-	415			223,000	10,620
39	3.00				4.032	0.234	0.672	0.031	7.5	1	8	280	495			-	415			223,000	10,261
40	4.00				2.702	0.234	0.422	0.031	7.5	1	8	980	1860			Offset 250	1550			730,000	238
41	4.00				2.702	0.234	0.422	0.031	7.5	1	8	920	1860			Fixed 1260	1550			730,000	241
42	2.00				2.702	0.234	0.422	0.031	7.5	1	8	940	1860			-	1550			365,000	250
43	2.00				2.702	0.234	0.422	0.031	7.5	1	8	990	1860			-	1550			365,000	1,000

TABLE 2 (Continued)

All Bellows Type 321 SS Test Temperature ~70°F

	1 Axial Defl. (in.)	2 Angular Defl. Degrees of Arc	3 Trans. Defl. (in.)	4 Test Pres. (psi)	5 $\bar{R}$ (in.)	6 a (in.)	7 h (in.)	8 t (in.)	9 L (in.)	10 N_p	11 N_c	12 K_a Meas. (lb/in.)	13 K_a Calc. (lb/in.)	14 K_r Meas.	15 K_r Calc.	16 Exper. Critical Pres. (psi)	17 Critical Pres. Calc. (psi) q_{cr}	18 Pres. Mem. Stress (psi) (S_{mp})	19 Pres. Bending Stress (psi) S_{bp}	20 Deflec. Bending Stress (psi) S_{bd}	21 Cycles to Failure
1	0.750	-	-	-	7.25	1.081	1.081	0.036	-	1	2	-	-	-	-	-	-	0	-	124,500	12,805
2	0.50	-	-	-	7.25	1.133	1.133	0.036	-	1	2	-	-	-	-	-	-	0	-	96,000	52,371
3	0.375	-	-	0-520	7.25	1.062	1.156	0.046	-	1	1	-	-	-	-	-	-	13,100	-	157,000	8,072
4	0.375	-	-	0-520	7.25	1.078	1.203	0.045	-	1	1	-	-	-	-	-	-	13,900	-	150,500	9,733
5	0.25	-	-	0-850	7.25	1.055	1.196	0.043	-	1	1	-	-	-	-	-	-	23,600	-	98,000	11,251
6	1.00	-	-	0-150	7.25	1.062	1.156	0.041	-	1	1	-	-	-	-	-	-	3,800	-	408,000	522
7	0.375	-	-	0-535	12.0	1.407	1.773	0.051	-	1	1	-	-	-	-	-	-	18,600	-	85,500	11,487
8	0.375	-	-	0-205	12.0	1.400	1.844	0.050	-	1	1	-	-	-	-	-	-	7,600	-	82,300	88,614
9	0.50	-	-	0-205	12.0	1.500	1.844	0.051	-	1	1	-	-	-	-	-	-	7,400	-	112,000	79,007
10	0.750	-	-	0.205	12.0	1.360	1.797	0.050	-	1	1	-	-	-	-	-	-	7,400	-	166,000	8,201
11	0.625	-	-	0-205	12.0	1.391	1.822	0.052	-	1	1	-	-	-	-	-	-	7,200	-	142,000	21,375
12	0.75	-	-	0-150	12.0	1.400	1.773	0.0505	-	1	1	-	-	-	-	-	-	5,300	-	169,500	16,075
13	0.50	-	-	0-205	12.0	1.400	1.789	0.049	-	1	1	-	-	-	-	-	-	7,500	-	109,000	49,757
14	1.322	-	-	0-300	12.0	1.[illegible]8	1.757	0.051	-	1	1	-	-	-	-	-	-	10,300	-	300,000	1,730
15	1.322	-	-	0-150	12.0	1.43	1.757	0.050	-	1	1	-	-	-	-	-	-	5,300	-	294,000	1,576
16	0.75	-	-	0-300	12.0	1.40	1.757	0.051	-	1	1	-	-	-	-	-	-	10,300	-	170,000	7,037
17	0.50	-	-	0-300	12.0	1.375	1.773	0.052	-	1	1	-	-	-	-	-	-	10,200	-	116,000	26,856
18	-	±3°	-	550	3.28	0.552	0.552	0.015	3.50	1	3	1,100	1053	-	-	-	1890	20,300	-	154,000 90,700	1,065
19	-	±2.1°	-	550	3.28	0.552	0.552	0.015	3.50	1	3	1,100	1053	-	-	-	1890	20,300	-	108,000 63,500	2,000
20	2.25	-	-	135	19.72	1.45	1.62	0.049	-	1	3	4,950 Tem 11,900 Comp	4750	-	-	-	-	4,500	-	132,700	41,570
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TABLE 3

TOROIDAL BELLOWS FATIGUE TEST DATA

C. TOROIDAL BELLOWS FATIGUE TEST RESULTS

Available data from fatigue tests on 20 toroidal bellows were analyzed to correlate calculated stresses with fatigue life. These data are presented in Table 3. Results of the analysis are demonstrated graphically in Figures 13 and 14. It can be observed from comparison of Figures 13a and 13b that pressure stresses have a definite effect upon fatigue life of toroidal type bellows. In Figure 13b, a better correlation exists between calculated stresses and cycles to failure than in Figure 13a. The pressure membrane stresses shown in Figure 13b have been added to the deflection bending stress at twice their calculated value; the C_{ps} factor has also been applied to increase deflection bending stresses because of the tendency to squirm under internal pressure. As with the other types of bellows, the calculated bending stresses far exceed the elastic limit and must be considered as fictitious or theoretical elastic stresses. A correlation coefficient of 0.82 was found between calculated theoretical elastic stresses and cycles to failure with the data treated as for Figure 14.

Since 18 of the 20 bellows tested were subjected to substantial internal pressure during the test, and only 2 were not subjected to internal pressure, the technique for evaluation of the effect of pressure stresses on cyclic life could not be repeated for toroidal type bellows. Instead, a least squares curve fit was made to the data in several cases, with a different factor applied to the pressure membrane stresses for each case. It was hoped that a reasonable and distinctive case with the least value for the standard deviation from the fitted curve would determine the factor which should be applied to the pressure stresses. This was in lieu of a non-linear multiple regression analysis of the data available. No distinctive case appeared, however, so a reasonably conservative case with the pressure membrane stresses multiplied by 3 and added to the theoretical bending stresses was taken as the rule. The results of this are shown directly from the digital computer output (Figure 14). The log-log relation can also be expressed as

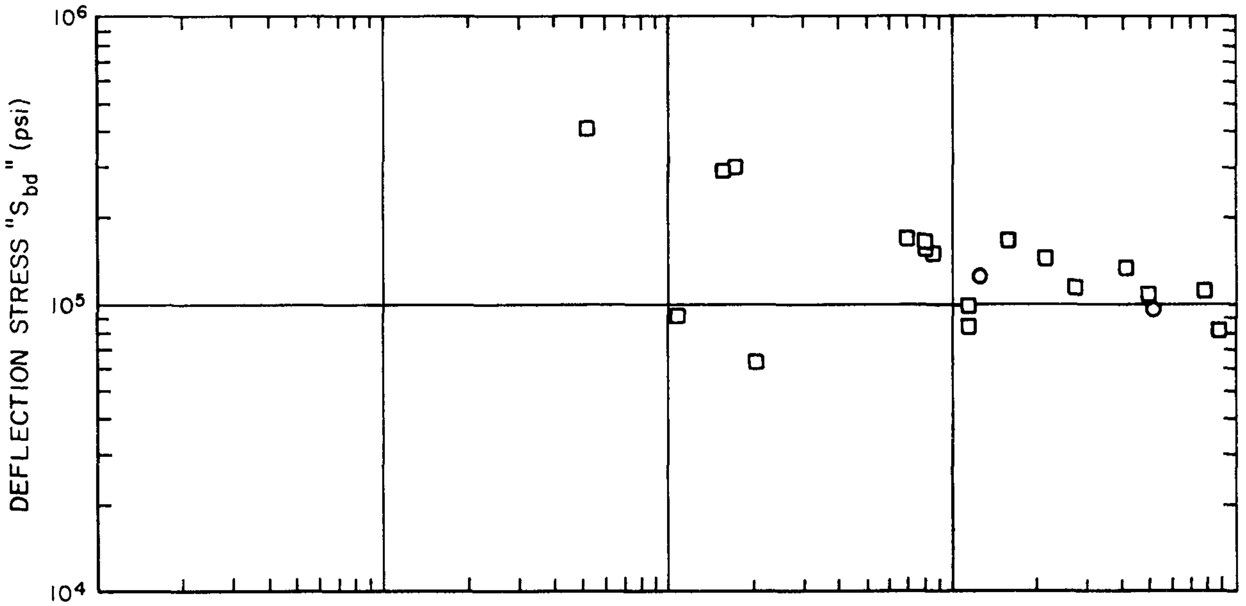
$$\mathrm{Log}_{10} S = 6.11 - 0.216\ \mathrm{Log}_{10} N \quad . \qquad \ldots(7)$$

At 10^4 cycles this results in approximately the same value of S (stress) as the expression for the other two bellows types. The factor of 3, which was applied to pressure membrane stresses, is greater than the factor of 2 which was applied to membrane stresses with the other two types of bellows and thereby makes allowance for some pressure bending stresses which are not otherwise considered. It may also be observed that analysis for the value of this factor is strongly dependent on results of only 4 of the 20 tests. On the basis of so few tests, it would be unwise to overemphasize the effect of pressure stresses, especially when the factor of 3 results in general agreement of theoretical stress-cyclic life relations with those of the other bellows.

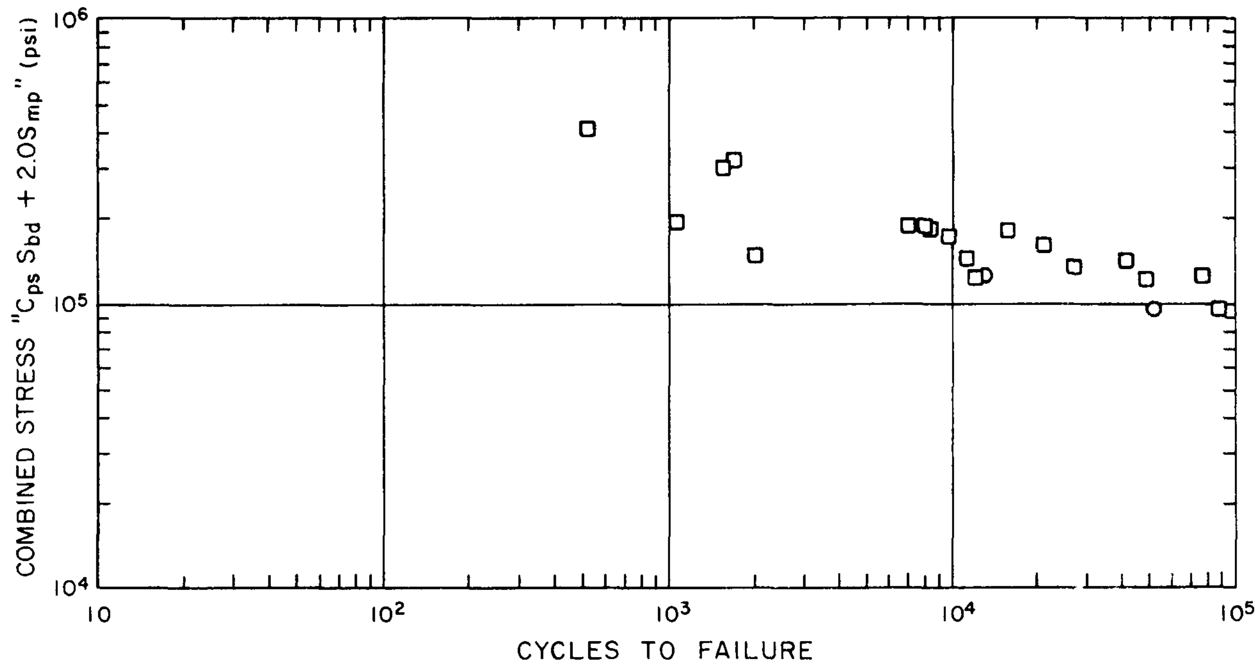
On the basis of these results, it can be concluded that for toroidal bellows, as with other bellows types, theoretical values of stresses should be divided by 2.0 to obtain the effective stress and still yield a safety factor of 2.0 on the mean above the ASA Code allowable. The effective stress would then be

$$S = 1.5\, S_{mp} + 0.5\, S_{bd} \quad . \qquad \ldots (8)$$

Table 3 also contains data on calculated and experimentally determined spring rates from 3 of the 20 bellows tested. It may be observed that calculated spring rates agree well with experimentally observed spring rates for the two smaller bellows and the tension spring rate of the larger bellows, but are in error by a factor greater than 2.0 for the compressive spring rate of the larger bellows. Prints were available for the larger bellows; the factor of 2.0 can almost be allowed for by the extremes of dimensional tolerances, again indicating that accurate prediction of spring rate will be a significant problem for bellows design.



a. Deflection Stress "S_{bd}" – Cycles to Failure



b. Combined Stress "C_{ps} S_{bd} + 2.0 S_{mp}" – Cycles to Failure

Figure 13. Toroidal Bellows Fatigue Test Results, Cycles to Failure

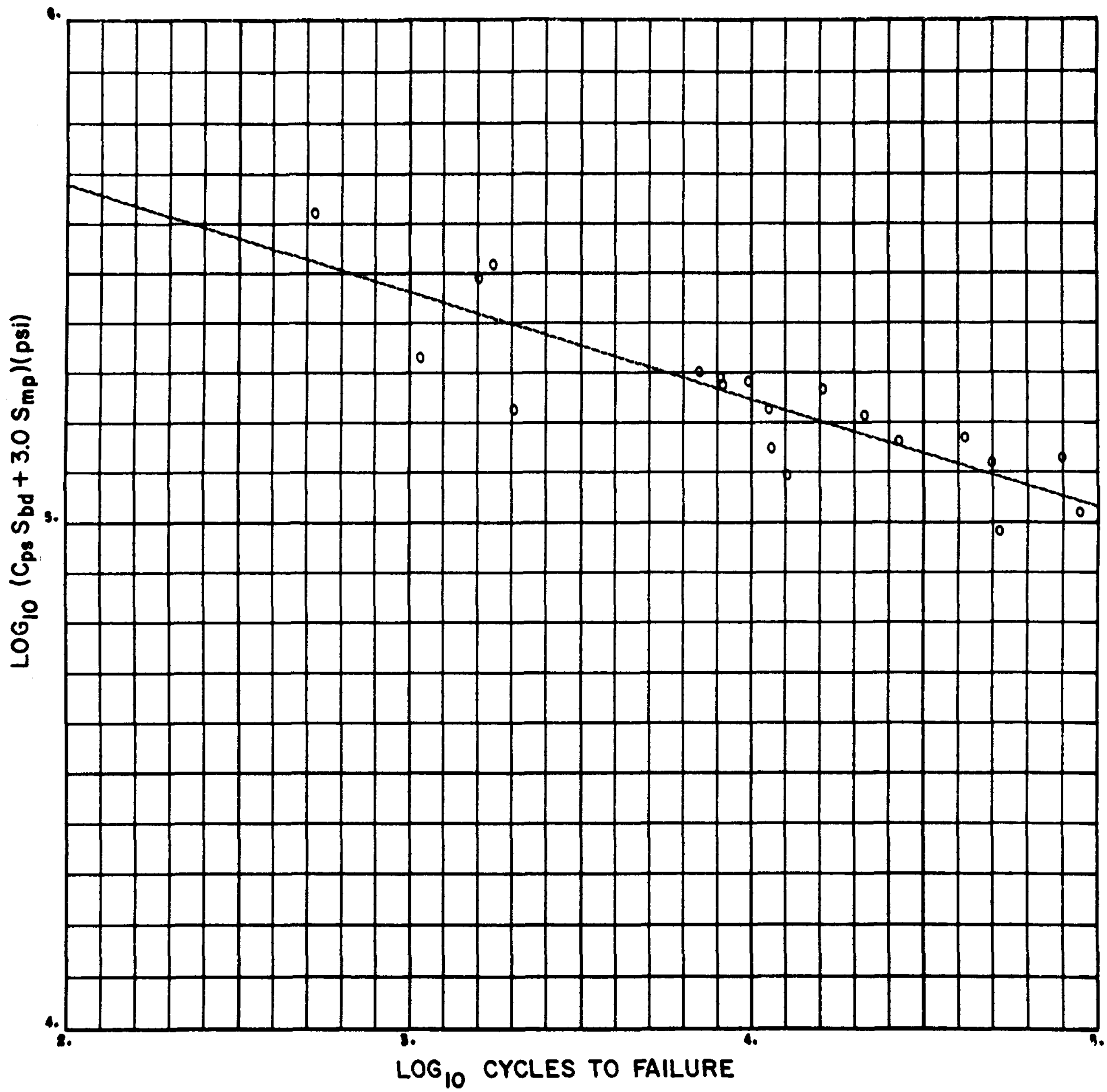


Figure 14. Toroidal Bellows Fatigue Test Results, Best Fit Curve - Combined Stress "C_{ps} S_{bd} + 3.0 S_{mp}"

D. COMBINED RESULTS OF FATIGUE TESTS AT 70°F

Results of the 108 room temperature fatigue tests on the three types of bellows were analyzed statistically as a single group. Calculated theoretical bending deflection stresses were combined with theoretical bending pressure stresses and the pressure membrane stresses in previously specified proportions to obtain total theoretical stresses equal to twice the effective stresses. A correlation coefficient of 0.87 was found between the calculated theoretical elastic stresses and cycles to failure for this combined group.

Constants of equations were determined which express the combined stress as a function of cycles to failure using a log-log relation with least squared error. The evaluation of least squared error was made using two techniques, i.e., (1) the error was used directly, and (2) the error was weighted by dividing by the value of the Log S (Method of Fractional Residuals[20]). Constants for both first order and second order equations were developed using both these methods, resulting in four evaluations of the data. Previously, the data for each of the three types of bellows had been separately evaluated with the same four procedures.

Some of the results of this analysis are presented in Table 4. Fitting the second order curve to test data yielded unreasonable results in the range of 250,000 cycles; these results are therefore not presented. The data on Figures 15, 16, 17, and 18 indicate that weighting of the residual (or errors) would be the most realistic approach since the scatter of data is obviously larger at the higher stress values; therefore, only those results using weighted error are included in Table 4. The weighting factor applied to the squared residuals was the inverse of square of the Log 10 of the stress for each point.

An equivalent RMS deviation for this weighted curve was desired. The RMS deviation with the weighted errors was obtained for the fitted curve and this was multiplied by the Log 10 of the calculated mean stress at any given number of cycles to determine an equivalent RMS deviation, not weighted, at that number of cycles. This method then predicts larger RMS deviations at higher stresses and is reasonable if not mathematically rigorous.

Table 4 also presents calculations for the factor of safety between predicted effective mean stresses and the proposed allowable stresses. At 7,000 cycles and 250,000 cycles the safety factor is not less than 1.75 in any case. However, at 250,000 cycles the safety factor is consistently lower than at 7,000 cycles.

TABLE 4

RESULTS FROM STATISTICAL EVALUATION OF FATIGUE TEST DATA

	Bellows Type			
	108 Combined	Convoluted	Reinforced	Toroidal
Curve with Min. RMS Deviation	Log S = 6.260 - 0.2479 Log N	Log S = 6.220 - 0.2307 Log N	Log S = 6.260 - 0.2493 Log N	Log S = 6.096 - 0.213 Log N
Equivalent RMS Deviation on Theoretical Stresses	0.01744 x Log S	0.01641 x Log S	0.01866 Log S	0.01507 Log S
Predicted Theoretical Stress Range at 7,000 Cycles	201,400 psi	214,300 psi	201,000 psi	189,200 psi
Predicted Effective Stress Range at 7,000 Cycles	100,700 psi	107,150 psi	100,500 psi	94,600 psi
ASA Allowable Stress Range at 7,000 Cycles	46,875 psi	46,875 psi	46,875 psi	46,875 psi
Factor of Safety	2.14	2.29	2.14	2.02
Reliability 75% Confidence Level	0.9995	0.9999	0.999	0.9993
Predicted Theoretical Stress Range at 250,000 Cycles	83,180 psi	94,200 psi	82,400	88,300
Predicted Effective Stress Range at 250,000 Cycles	41,590 psi	47,100 psi	41,200	44,150
ASA Allowable Stress at 250,000 cycles	23,438 psi	23,438 psi	23,438	23,438
Factor of Safety	1.77	2.01	1.75	1.88
Reliability 75% Confidence Level	0.995	0.9995	0.991	0.9991
Correlation Coefficient	0.87	0.74	0.89	0.82

It should be observed that in all 108 tests, no failures were obtained above 197,000 cycles and the one bellows tested to 400,000 cycles did not fail but was treated so in the analysis. Therefore, the predicted values at 250,000 cycles are an extrapolation and may be considered conservative.

For each type of bellows and for the combined test results, Table 4 also presents an estimate of the reliability in service at the allowable stress level when subjected to the full cyclic life. This estimate of reliability is at the 75% confidence level for a large population.[22] It is based on an assumed normal distribution. Since system design requirements seldom specify a number of operating cycles as high as 7,000 cycles, which is the minimum number considered in establishing ASA Code allowables, these estimates of the reliability may be very conservative in practice. The estimates do not include allowance for unusual corrosive conditions or vibration problems which may be encountered in application.

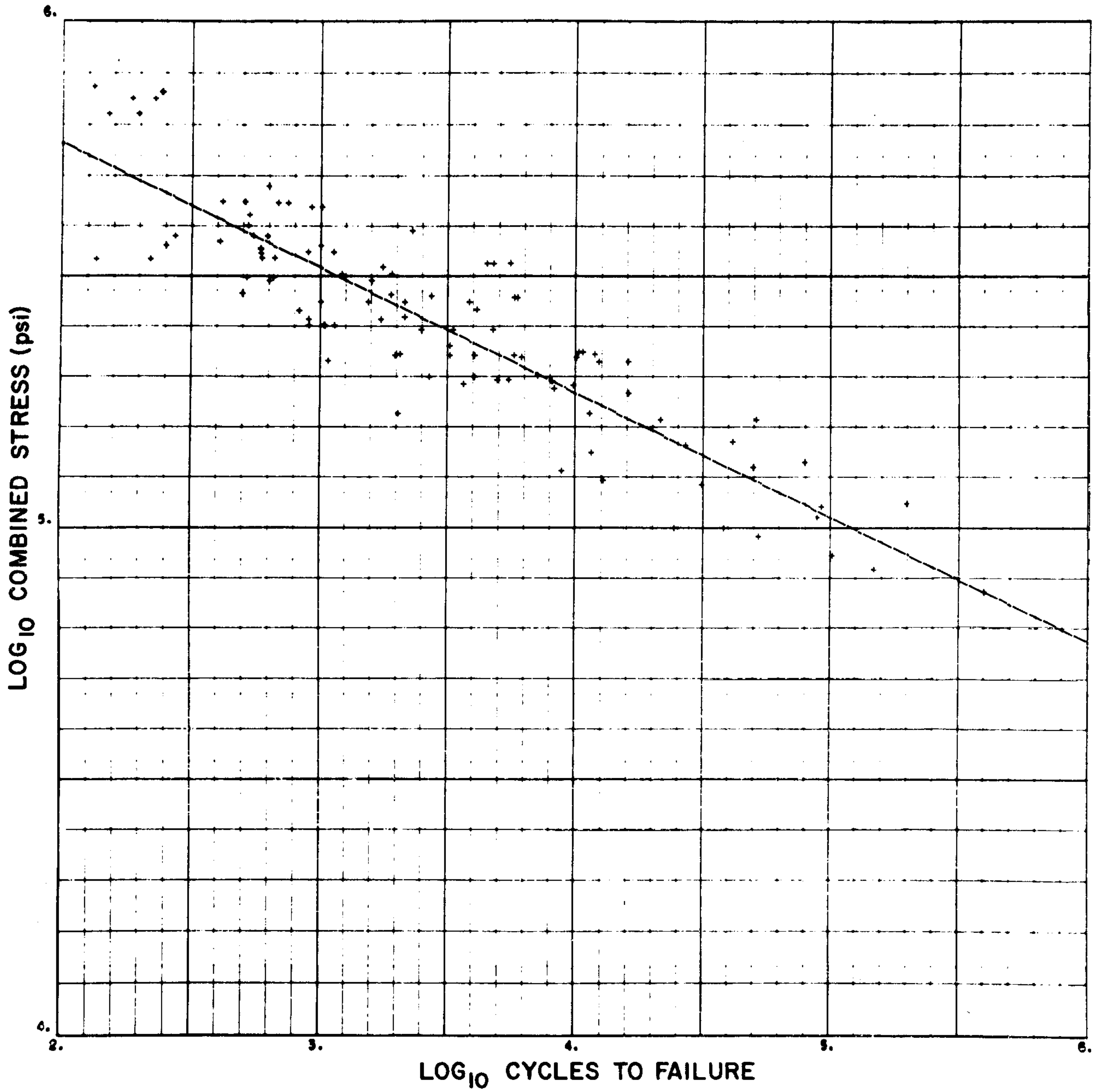


Figure 15. 108 Bellows Combined Fatigue Test Results,
Best Fit Curve-Weighted Area

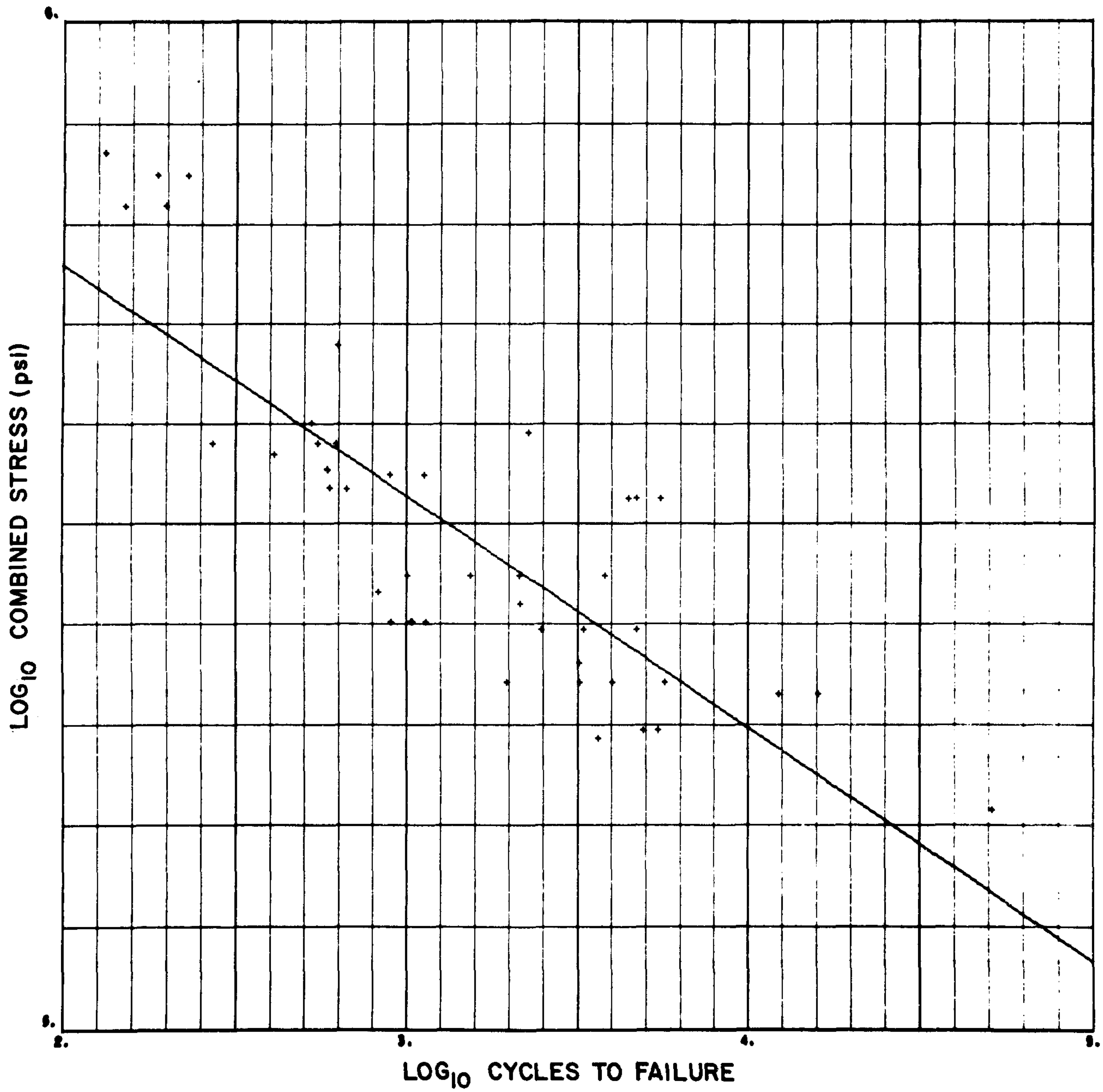


Figure 16. Convoluted Bellows Fatigue Test Results, Best Fit Curve-Weighted Error

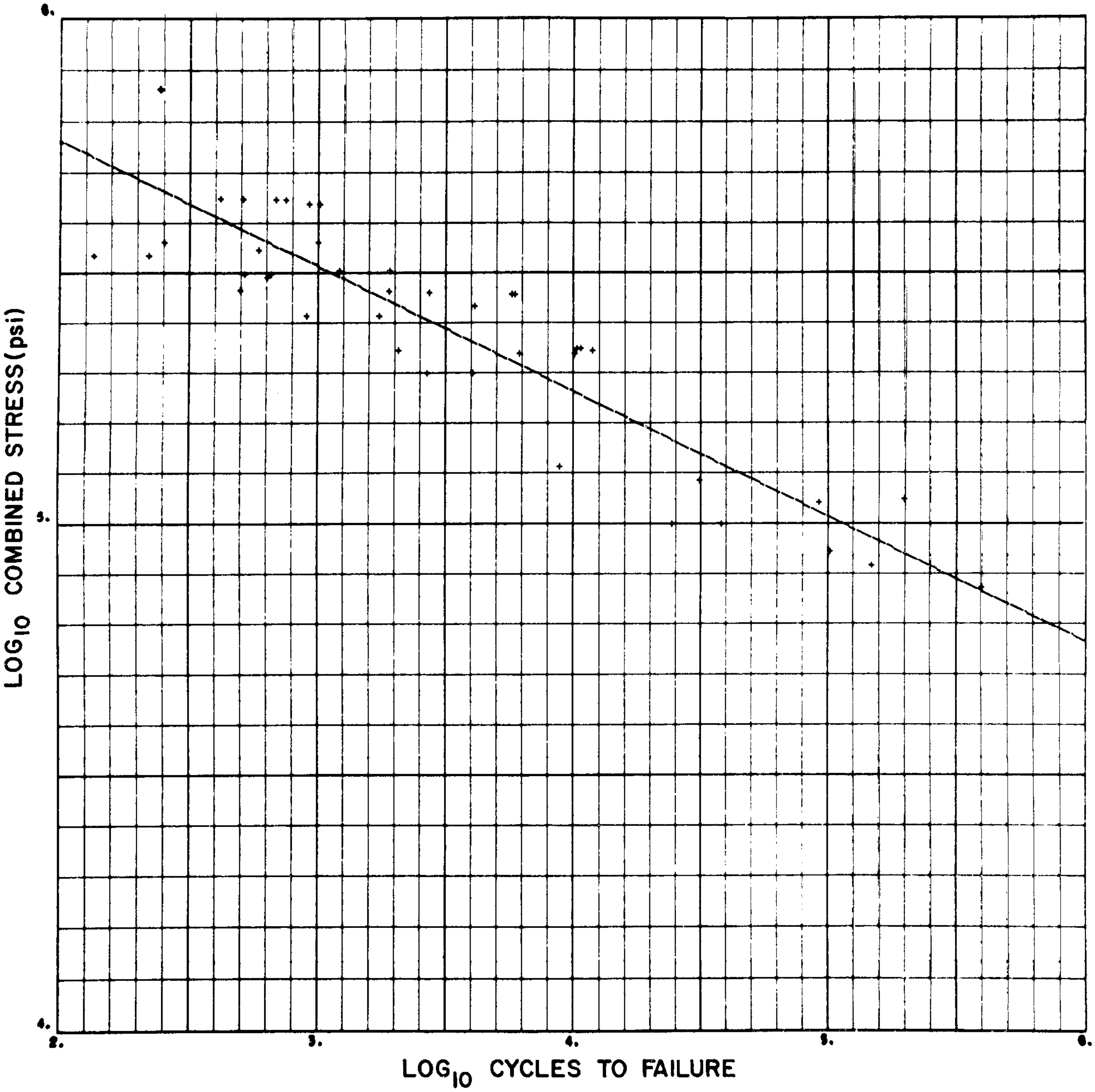


Figure 17. Ring Reinforced Bellows Fatigue Test Results,
Best Fit Curve-Weighted Error

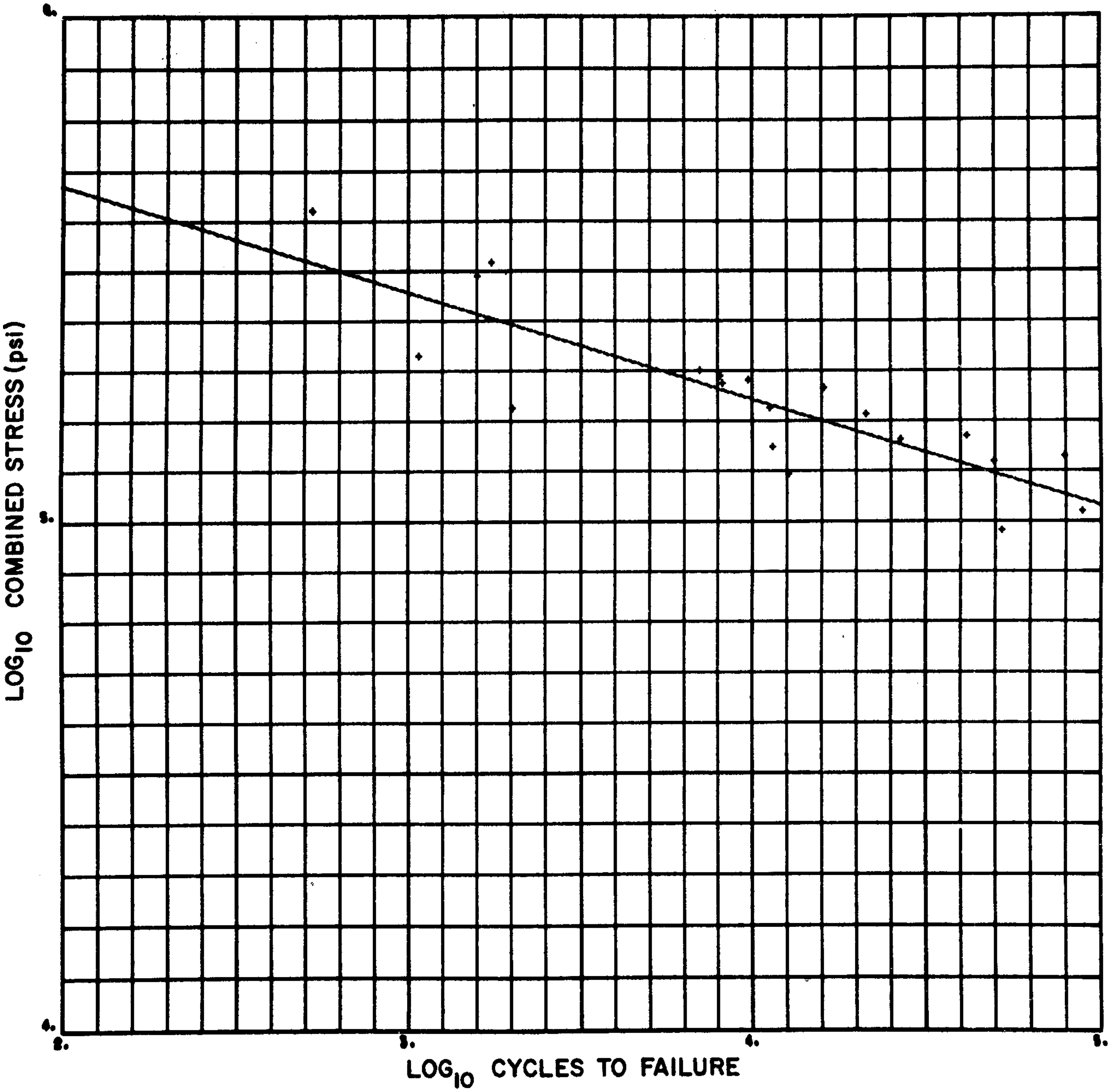


Figure 18. Toroidal Bellows Fatigue Test Results,
Best Fit Curve-Weighted Error

E. RESULTS OF HIGH TEMPERATURE FATIGUE TESTS

Results from fatigue tests performed at 1200°F on the three types of bellows are presented in Table 5. Some of these tests were performed as part of an experimental investigation of bellows performance as explained in Reference 21. Several of the larger bellows were made especially for this test program as non-standard production items, and were purchased from the manufacturer on a best-effort basis. Consequently, these bellows contained geometrical variations within and between convolutions which would result in rejection on a normal inspection basis. These variations did not, however, interfere with the basic purpose of the program and the bellows were accepted and tested for stress distribution. The fatigue test was an additional part of the program and, in spite of the observed large stress concentrations, all the bellows were subjected to fatigue tests. An estimate of the additional stress concentrations can also be found in the experimental data of Reference 21. Data from tests on bellows which would normally be rejected are therefore also included in Table 5 and may be given special consideration in light of the above.

The tests were performed at 1200°F with molten sodium metal on one surface, either the inside or the outside of the bellows. In the tests of the small bellows, the second atmosphere was helium gas and in tests of the large bellows, the second atmosphere was air entrapped in thermal insulation. During the tests, each of the bellows was subjected to several levels of cyclic stress. The data were transformed to one level of cyclic stress at 7000 cycles by the use of Miners Hypothesis and the fatigue relation $N^{-1/4} = KS$. This fatigue relation may be derived from results of the statistical analysis of bellows fatigue behavior presented in Table 4 with

N = number of cycles

S = cyclic stress range

K = a constant, empirically determined

The result of these calculations permitted determination of a factor of safety defined as the ratio between the effective stresses range applied for a 7000-cycle life and that which would be allowed by the ASA Code at this temperature. It should be noted that at constant high temperature cycling the code has been interpreted herein as allowing a total stress range of $1.25\ (S_c + S_h)$ where $S_c = S_h$.

The safety factor determined for these bellows averages 3.80. The low value of 1.15 is for a bellows with a large deviation from typical convolution conformation. This resulted in an experimentally observed stress concentration[21] of approximately 3.0 above the calculated and experimentally verified theoretical stress distribution. If this low value is ignored or adjusted on this basis, there would then be a minimum safety factor of 2.52. This is above the minimum value of 1.75 on the mean which is observed at 70°F by the data of Table 4. It is felt that this margin between 2.52 and 1.75 will be adequate to provide for the effect upon cyclic life resulting from the difference in rate of cycling between that of the test and that of service conditions.

1 Bellows No., Type, & Material	2 R (in.)	3 h (in.)	4 a (in.)	5 $\bar{R}$ (in.)	6 $\left(\frac{\pi}{2}-1\right)a+h$ (in.)	7 t (in.)	8 N_p	9 N_c	10 $X = 0.778\sqrt{\bar{R}t}$	11 $\frac{a}{h}$	12 $\frac{a}{x}$	13 C_p	14 C_d	15 Δ (in.)	16 S_{mp} (psi)	17 S_{bp} (psi)	18 S_{bd} (psi)	19 S (psi)	20 N_{test}	21 $N_{equiv.}$ at S_{AC}	22 $N_{equiv.}$ total	23 $S_{equiv.}$ at 9000 cycles (psi)	24 Safety Factor
#2	9.50	1.00	1.00	10.50	1.57	0.050	1	4	0.565	1	1.77	0.34	1.8	0.428	770	1500	28,200	16,370	1,000	4,500			
Convoluted	304	SS												0.534	770	1500	35,200	19,870	820	8,000	12,500	13,000	1.15
#3	10.125	0.875	0.468	11.00	1.142	0.056	1	4	0.61	0.535	0.765	0.59	1.92	0.500	480	1570	44,300	24,200	3,000	64,000			
Convoluted	304	SS												0.624	480	1570	55,300	29,700	1,000	47,500			
														0.750	480	1570	66,400	35,250	1,000	96,200			
														0.876	480	1570	77,400	40,750	1,000	173,000			
														1.000	480	1570	88,500	46,300	1,000	287,000			
														1.124	480	1570	99,600	51,850	90	40,700	708,400	35,700	3.17
#5	10.00	0.75	0.20	10.75	0.864	0.050	1	5	0.576	0.267	0.345	0.82	1.37	0.316	100	1980	37,800	20,980	18,260	220,000			
Convoluted	304	SS												0.404	100	1980	48,300	26,230	5,170	153,000			
														0.480	100	1980	57,500	30,830	1,180	66,400			
														0.552	100	1980	66,000	35,080	720	68,000	507,400	32,800	2.92
#6	10.00	0.75	0.20	10.75	0.864	0.050	1	5	0.576	0.267	0.345	0.82	1.37	0.400	100	1980	47,900	26,030	5,000	143,600			NF
Convoluted	304	SS												0.600	100	1980	71,800	37,980	5,000	650,000			
														0.800	100	1980	95,700	49,930	5,000	1,970,000			
														1.000	100	1980	119,600	61,880	140	122,500	2,886,100	50,750	4.5
#7	10.34	1.14	0.367	11.48	1.349	0.062	1	4	0.653	0.322	0.562	0.75	1.5	0.560	350	2800	42,100	24,200	1,000	21,400			
Convoluted	304	SS												0.700	350	2800	52,500	29,400	1,000	46,200			
(welded)														0.840	350	2800	63,000	34,650	1,000	90,000			
														0.980	350	2800	73,600	39,950	1,000	158,500			
														1.120	350	2800	84,100	45,200	1,050	273,000	589,100	34,100	3.03
#8 Conv.	10.10	1.15	0.438	11.16	1.400	0.030	1	4	0.45	0.381	0.973	0.55	1.65	1.946	780	8920	63,400	41,400	1,590	289,000	289,000	28,500	2.54
#9	10.10	1.15	0.438	11.16	1.400	0.050	1	4	0.58	0.381	0.755	0.55	1.7	1.184	465	3795	63,200	35,860	5,000	515,000			
Convoluted	304	SS												1.480	465	3795	79,000	43,760	5,000	1,140,000			NF
														1.776	465	3795	94,700	51,610	3,000	1,335,000	2,990,000	51,100	4.55
#10 Ring Reinf.	304 SS 10.10	1.15	0.438	11.16	-	0.030	1	4	0.45	0.381	0.973	0.58	1.57	1.946	190	9430	66,600	35,850	2,730	282,000	282,000	28,300	2.52
#11	10.625	1.375	1.325	12.00	-	0.050	1	2	0.603	0.96	2.2	-	1.55	0.526	140	-	43,400	21,910	27,850	402,000			
Toroidal	304 SS													0.788	140	-	65,000	32,710	5,130	366,000			NF
														1.050	140	-	86,500	43,460	2,570	575,000	1,343,000	42,000	3.74

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TABLE 5

BELLOWS FATIGUE TEST DATA FROM HIGH TEMPERATURE TESTS

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
	Bellows No., Type, & Material	R (in.)	h (in.)	a (in.)	$\bar{R}$ (in.)	$\left(\frac{\pi}{2} - 1\right) a+h$ (in.)	t (in.)	N_p	N_c	$X = 0.778\sqrt{\bar{R}t}$	$\frac{a}{h}$	$\frac{a}{x}$	C_p	C_d	Δ (in.)	S_{mp} (psi)	S_{bp} (psi)	S_{bd} (psi)	S (psi)	N_{test}	$N_{equiv.}$ at S_{AC}	$N_{equiv.}$ total	$S_{equiv.}$ at 7000 cycles (psi)	Safety Factor
1	#8	1.755	0.407	0.221	2.162	0.533	0.03	1	10	0.198	0.543	1.12	0.52	1.8	0.545	110	590	57,700	29,550	1,000	31,100			
2	Convoluted														0.683	110	590	72,300	36,850	1,000	75,700			
3	347 SS														0.796	110	590	84,400	42,900	1,000	138,000			
4															0.898	110	590	95,000	48,200	1,000	213,000			
5															1.035	110	590	109,600	55,500	11,000	4,268,000	4,725,800	63,700	5.1
6	#10	1.755	0.407	0.221	2.162	0.533	0.03	1	10	0.198	0.543	1.12	0.52	1.8	0.554	110	590	58,600	30,000	1,000	33,100			
7	Convoluted														0.693	110	590	73,400	37,400	1,000	80,000			
8	347 SS														0.832	110	590	88,100	44,750	1,000	164,000			
9															0.971	110	590	103,000	52,200	1,000	303,000			
10															1.110	110	590	117,700	59,550	1,000	512,000			NF
11															1.249	110	590	132,200	66,800	11,000	8,910,000	10,002,100	76,900	6.15
12	#9	1.755	0.407	0.218	2.162	0.531	0.03	1	10	0.198	0.543	1.12	0.52	1.8	0.554	110	590	58,600	30,000	1,000	33,100			
13	Convoluted														0.693	110	590	73,400	37,400	1,000	80,000			
14	347 SS														0.832	110	590	88,100	44,750	1,000	164,000			
15															0.971	110	590	103,000	52,200	1,000	303,000			
16															1.110	110	590	117,700	59,550	971	499,000	1,079,100	44,100	3.52
17	#2	1.755	0.347	0.125	2.102	0.418	0.03	1	24	0.194	0.36	0.645	0.70	1.6	0.714	80	560	47,700	24,490	1,000	14,800			
18	Convoluted														0.892	80	560	59,600	30,440	1,000	35,000			
19	347 SS														1.070	80	560	71,500	36,390	1,000	70,800			NF
20															1.248	80	560	83,400	42,340	18,000	2,360,000	2,480,600	54,300	4.34
21	#1	1.755	0.347	0.125	2.102	0.418	0.03	1	24	0.194	0.36	0.645	0.70	1.6	0.714	80	560	47,700	24,490	1,000	14,800			
22	Convoluted 347	SS													0.892	80	560	59,600	30,440	1,000	35,000			
23															1.070	80	560	71,500	36,390	1,000	70,800			NF
24															1.248	80	560	83,400	42,340	17,000	2,225,000	2,345,600	53,500	4.28
25	#5	1.765	0.197	0.08	1.962	0.2426	0.015	1	30	0.134	0.406	0.596	0.70	1.6	0.666	140	670	46,100	23,860	1,000	13,300			
26	Convoluted														0.832	140	670	57,600	29,610	1,000	31,400			
27	347 SS														0.998	140	670	69,100	35,360	1,000	63,800			
28															1.164	140	670	80,600	41,110	1,000	116,500			
29															1.330	140	670	92,000	46,810	8,576	1,718,000	1,943,000	51,100	4.09
30	#6	1.765	0.197	0.08	1.962	0.2426	0.015	1	30	0.134	0.406	0.596	0.70	1.6	0.654	140	670	45,200	23,410	1,000	12,300			
31	Convoluted														0.817	140	670	56,500	29,060	1,000	29,200			
32	347 SS														0.980	140	670	67,800	34,710	1,000	59,000			
33															1.143	140	670	79,100	40,360	1,000	108,000			
34															1.306	140	670	90,400	46,010	1,000	184,000			
35															1.469	140	670	101,700	51,660	5,326	1,550,000	1,942,500	57,100	4.09
1	#3	1.755	0.347	0.133	2.102	0.423	0.03	1	23	0.194	0.383	0.685	0.67	1.65	0.696	85	535	47,200	24,220	1,000	14,000			
2	Convoluted														0.870	85	535	58,900	30,070	1,000	33,500			
3	347 SS														1.044	85	535	70,700	35,970	1,000	66,200			
4															1.218	85	535	82,500	41,870	1,000	125,300			
5															1.392	85	535	94,400	47,820	1,000	213,000			
6															1.566	85	535	106,100	53,670	1,000	338,000			
7															1.740	85	535	119,300	60,270	1,000	540,000			
8															1.914	85	535	129,800	65,520	1,000	756,000			
9															2.064	85	535	140,000	70,620	353	360,000	2,446,000	54,100	4.33
10	#4	1.755	0.347	0.133	2.102	0.423	0.03	1	23	0.194	0.383	0.685	0.67	1.65	0.708	85	535	48,000	24,620	1,000	15,100			
11	Convoluted														0.885	85	535	60,000	30,620	1,000	36,000			
12	347 SS														1.062	85	535	72,000	36,620	1,000	74,000			
13															1.239	85	535	84,000	42,620	12,169	1,635,000			
14															1.419	85	535	96,000	48,620	1,000	230,000			
15															1.593	85	535	108,000	54,620	1,000	365,000			
16															1.770	85	535	120,000	60,620	730	403,000	2,758,000	55,700	4.46

Table 5 (Continued)

IV. CONCLUSIONS

A method of evaluating the design application of bellows on the basis of calculated allowable stresses has been presented which correlates well with test results. This calculation method allows evaluation of both deflection stresses and pressure stresses on convoluted bellows, convoluted bellows with reinforcing rings and toroidal bellows. The reliability and correlation coefficients in Table 4 are sufficiently high to advise acceptance of the calculation method regardless of its mathematical basis.

Methods of statistical mathematics have been used with data from tests to determine the effect of pressure stresses on cyclic life of the bellows. Factors for this effect have been included in the proposed calculation procedure.

The proposed stress calculation method and the proposed allowable stress limits for bellows to be used in a piping system both have the same basis as those for other components of the system. The proposed limits have been shown to have reasonable safety factors at 70°F and at 1200°F on the basis of experimental data from tests evaluated herein.

Test data have been presented and so evaluated as to provide a basis for economic optimization of allowable stresses in applications other than piping systems.

The effect of instability induced by internal pressure upon bending stresses and spring rate has been evaluated analytically and observed in experimental data. This effect has been compensated for in the proposed analytical method.

Additional test data and data from service performance would be helpful in:

a) Confirming the effect of pressure upon cyclic life, especially for toroidal and reinforced bellows.

b) Confirming the safety of the proposed allowable stresses, and the effect of pressure stresses upon cyclic life at high temperatures.

c) Evaluating other problems such as vibration-induced fatigue failure which may occur in service.

d) Evaluating the manufacturing process controls which must be exerted to ensure reliable bellows.

e) Evaluating other design features such as end connection details which help ensure reliable bellows.

Additional analytical effort might be especially helpful in studying the effect of contact with reinforcing rings on the fatigue life of reinforced bellows.

As in the opening paragraph, it has been found that stresses are predicted more accurately than deflection for a given load. Therefore, where spring rate is a controlling factor, prototype testing under design conditions is recommended, since substantial error might result from using the calculation methods presented herein. This error can be in either direction.

The availability of additional test data and funds would justify and allow the use of non-linear multiple regression analysis of all test data. This method could be applied to evaluating the effect of internal pressure and other factors on the fatigue life and performance of bellows.

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APPENDIX A
INSTABILITY OF BELLOWS WITH INTERNAL PRESSURE

The problem to be investigated is that of instability of a bellows under internal pressure when used as a hinge. The bending stresses in the bellows when near instability occurs will be investigated.

The assumption is made that the bellows can be approximated by a thin walled hollow cylinder having a bending rigidity $\overline{EI}$ which can be taken as a continuum quality. This assumption might require 10 convolutions to avoid appreciable error; its effect has not been evaluated.

Taking the bellows as a thin walled tube with internal pressure as shown in Figure 19, two sources of transverse force are found which contribute to instability, the end force "Q" and the internal pressure "P." Both of these sources require some curvature (d^2y/dx^2) for the existence of a transverse force.

It can easily be shown that the transverse, or y component of force from Q for a length "dx" equals

$$f_{Qy} dx = -Q \frac{d^2y}{dx^2} \cdot dx \quad . \qquad \ldots(1)$$

The transverse force caused by internal pressure results from increased area on one side and decreased area on the other side due to the curvature of the tube. The length of an element of the outside surface where the length of the center is "dx" may be expressed as

$$dL = dx \left[1 + R \cdot \sin\varphi \left(\frac{-d\theta}{dx} \right) \right] , \qquad \ldots(2a)$$

or

$$dL = dx \left(1 - R \sin\varphi \cdot \frac{d^2y}{dx^2} \right) \quad , \qquad \ldots(2b)$$

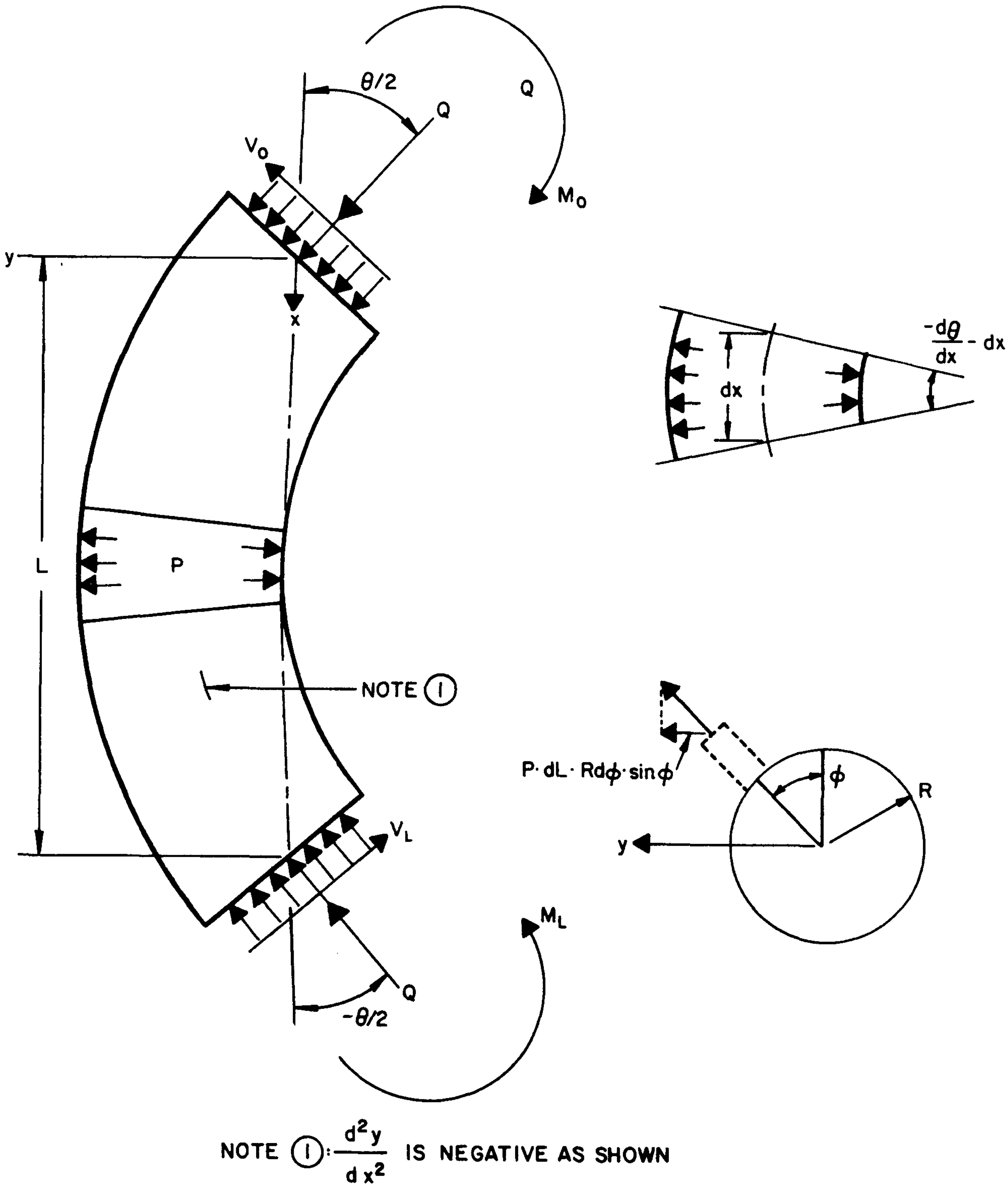

Figure 19. Dimensions and Forces on Bellows

since

$$\frac{d\theta}{dx} \cong \frac{d}{dx}\left(\frac{dy}{dx}\right) = \frac{d^2y}{dx^2} \quad . \qquad \ldots(3)$$

The y component of the internal pressure is

$$df_{py} = P \cdot dL \cdot Rd\varphi \cdot \sin\varphi \quad , \qquad \ldots(4)$$

then the net transverse force from internal pressure f_{py} for a length dx equals

$$f_{py} \cdot dx = \int_{\varphi=0}^{\varphi=2\pi} Pdx\left(1 - R\sin\varphi\frac{d^2y}{dx^2}\right) \cdot R\sin\varphi\, d\varphi \quad , \qquad \ldots(5a)$$

which can be integrated to

$$f_{py} \cdot dx = \int_{\varphi=0}^{\varphi=2\pi}\left[-P \cdot R^2\frac{d^2y}{dx^2} \cdot dx \cdot \sin^{-2}\varphi + PRdx\sin\varphi\right]d\varphi \quad , \qquad \ldots(5b)$$

$$f_{py}dx = PR^2\frac{d^2y}{dx^2} \cdot dx \cdot \pi + 0 \quad , \qquad \ldots(5c)$$

$$f_{py}dx = -P\pi R^2 \cdot \frac{d^2y}{dx^2} \cdot dx \quad . \qquad \ldots(5d)$$

One can interpret this to mean that f_{py} is equal to the pressure times the cross-sectional area of the tube times the curvature. Also, since the total transverse force under consideration is then

$$f_y = -\left(P\pi R^2 + Q\right)\frac{d^2y}{dx^2} \quad , \qquad \ldots(6)$$

a ready explanation can be made for the reason that buckling of tubes due to internal pressure is seldom observed. When end caps or elbows exist on tubes, the force, Q, is opposite in sign to that shown on and proceding page, and has a value

$$Q = -P\pi R^2 \quad .$$

The result would then be

$$f_y = 0 \quad ,$$

and no transverse (or buckling) forces exist.

The equation of equilibrium for forces in the y direction on an element of length dx, considering bending of the tube, then becomes

$$\frac{d^2}{dx^2}\left(\frac{\overline{EI}\, d^2y}{dx^2}\right) - f_y = 0 \quad . \qquad \ldots(7)$$

If $\overline{EI}$ is a constant and elastic action prevails, then by substituting for f_y,

$$\overline{EI}\,\frac{d^4y}{dx^2} + \left(P\,\pi R^2 + Q\right)\frac{d^2y}{dx^2} = 0 \quad . \qquad \ldots(8)$$

Taking as boundary conditions at x = 0,

a) $y = y_o = 0$ defined

b) $y' = \theta_o = \theta/2$ defined as $\dfrac{M_i L}{2\overline{EI}}$

c) $\overline{EI}_y'' = -M_o$

d) $\overline{EI}_y''' = V_o$

at x = L,

e) $y = y_L = 0$ defined

f) $y' = \theta_L = -\theta/2$ defined as $\dfrac{M_i L}{2\,\overline{EI}}$

g) $\overline{EI}_y'' = -M_L$

h) $\overline{EI}_y''' = V_L$

Also, since symmetry exists, then at x = L/2,

$$\theta_{L/2} = 0 \text{ and } V_{L/2} = 0 \quad .$$

Integrating Equation 8 from x = 0, to $x = x_{arbitrary}$ and preserving the meaning of the limits as boundary conditions (b) and (d) yields

$$\overline{EI}\,\frac{d^3y}{dx^3}\Bigg]_o^x + \left(P\pi R^2 + Q\right)\frac{dy}{dx}\Bigg]_o^x = 0 \quad , \qquad \ldots(9)$$

or

$$\overline{EI}\,\frac{d^3y}{dx^3} - V_o + (P\pi R^2 + Q)\,\frac{dy}{dx} - (P\pi R^2 + Q)\,\theta_o = 0 \quad . \qquad \ldots(10)$$

Integrating again in this way and substituting for boundary conditions in (a) and (c) yields

$$\overline{EI}\,\frac{d^2y}{dx^2} + (P\pi R^2 + Q)\,y = V_o x - M_o + (P\pi R^2 + Q)(\theta_o x + y_o) \quad . \qquad \ldots(11)$$

This now requires solution as a differential equation:

$$\frac{d^2y}{dx^2} + K^2 y = (V_o + F\theta_o)\,\frac{x}{\overline{EI}} + (Fy_o - M_o)\,\frac{1}{\overline{EI}} \quad , \qquad \ldots(12)$$

$$F = P\pi R^2 + Q \quad , \qquad \ldots(13)$$

$$K^2 = \frac{F}{EI} \quad . \qquad \ldots(14)$$

The solution is of a classic form

$$y = A \cos Kx + B \sin Kx + C + Dx \quad . \qquad \ldots(15)$$

Solve for the coefficients and unknown boundary conditions V_o, V_L, M_o, M_L, by substituting into the equations as follows:

Substituting Equation 15 into Equation 11 yields

$$K^2C + K^2Dx = (V_o + F\theta_o)\frac{x}{EI} + (Fy_o - M_o)\frac{1}{EI} \quad , \qquad \ldots(16)$$

$$C = \frac{Fy_o - M_o}{F} \quad , \qquad \ldots(17)$$

$$D = \frac{F\theta_o + V_o}{F} \quad . \qquad \ldots(18)$$

From boundary condition a),

$$C = \frac{-M_o}{F} \quad . \qquad \ldots(19)$$

Substituting boundary condition a) into Equation 15 yields

$$y = 0 = A \cdot 1 + B \cdot 0 + C + D \cdot 0 \quad , \qquad \ldots(20a)$$

$$A = -C = \frac{M_o}{F} \quad . \qquad \ldots(20b)$$

Substituting boundary condition b) into Equation 15 yields

$$y' = \theta_o = -KA \sin Kx + KB \cos Kx + D \quad , \qquad \ldots(21a)$$

$$B = \frac{\theta_o - D}{K} \quad , \qquad \ldots(21b)$$

then

$$y = -C \cos Kx + \frac{\theta_o - D}{K} \sin Kx + C + Dx \quad . \qquad \ldots(22)$$

Using the symmetry conditions by taking equilibrium of forces in the y direction for the length x = 0 to x = L/2,

$$F\theta_{L/2} + V_{L/2} - F\theta_o - V_o = 0 \quad , \qquad \ldots(23a)$$

yields

$$F\theta + V_o = 0, \text{ or } D = 0 \quad , \qquad \ldots(23b)$$

then

$$y = + \frac{M_o}{F} \cos Kx + \frac{\theta_o}{K} \sin Kx - \frac{M_o}{F} \quad . \qquad \ldots(24)$$

Substituting into boundary condition f) and boundary condition b),

$$y' = -\frac{\theta}{2} = -K \frac{M_o}{F} \sin KL + K \frac{\theta}{2K} \cos KL \quad , \qquad \ldots(25)$$

$$M_o = \frac{F}{K \sin KL} \cdot \frac{\theta}{2} \cdot (1 + \cos KL) \quad , \qquad \ldots(26)$$

then

$$y = \frac{1}{K} \frac{\theta}{2} \left(\frac{1 + \cos KL}{\sin KL}\right)\left(\cos Kx - 1\right) + \frac{\theta}{2K} \sin Kx \quad , \qquad \ldots(27)$$

or since $\theta/2 = \frac{M_i L}{2 \overline{EI}}$ where M_i is the bending moment when pressure P = 0,

$$y = \frac{M_i L}{2 \overline{EI}} \frac{1}{K} \left[\left(\frac{1 - \cos KL}{\sin KL}\right)\left(\cos Kx - 1\right) + \sin Kx\right] \quad , \qquad \ldots(28)$$

and

$$M = \overline{EI}\, y'' = \frac{-M_i LK}{2} \left[\left(\frac{1 + \cos KL}{\sin KL}\right) \cos Kx + \sin Kx\right] \quad . \qquad \ldots(29)$$

It can be shown that when sin KL = 0 and at KL = π, the value of M is not infinite. However, when sin KL = 0 at KL - 2π, the value of M becomes infinite, or KL = 2π determines the critical buckling load. This critical load

$$F_{cr} = \frac{4\pi^2 \overline{EI}}{L^2} \quad , \qquad \ldots(30)$$

is the same as for a regular fixed end column of length L.

To find the point of maximum moment

$$\frac{dM}{dx} = 0 = V \quad .$$

Since V = 0 at x = L/2, maximum moment occurs at L/2, and since

$$\frac{1 + \cos KL}{\sin KL} = \cot \tfrac{1}{2}KL = \frac{\cos \frac{1}{2} KL}{\sin \frac{1}{2} KL} \quad , \qquad \ldots(31)$$

then

$$M_{L/2} - \frac{-M_i LK}{2} \left[\frac{\cos^2 \frac{1}{2} KL}{\sin \frac{1}{2} KL} + \sin \tfrac{1}{2} KL \right] \quad , \qquad \ldots(32)$$

$$M_{L/2} = \frac{-M_i LK}{2} \cdot \frac{1}{\sin \frac{1}{2} KL} \text{ or } \frac{M_{L/2}}{M_i}$$

$$= -\frac{LK}{2} \Big/ \sin \tfrac{1}{2} KL \quad . \qquad \ldots(33)$$

Using this formula and the moment at x = L/2 as a function of $\frac{1}{2}KL/2\pi$, (the ratio of operating pressure to critical pressure), the effect of internal pressure on maximum bending stress was evaluated. This is presented graphically as

$$C_{ps} = \frac{M_{L/2}}{M_i} \quad ,$$

as a function of P/P_{cr} in Figure 20.

Examination of Equation 29 shows also that at the ends of the bellows, $x = 0$, and $x = L$; M_o and M_L become zero when $\frac{1}{2}KL = \pi/2$ or $F/F_{cr} = 1/4$.

This suggests a strong effect of internal pressure on rotation spring rate K_r. When $F/F_{cr} > 1/4$, the spring rate becomes negative.

A similar derivation was made for bellows subjected to transverse motion and internal pressure. No increase in peak deflection bending stresses is found to be induced when internal pressure is less than $\frac{1}{2}P_{cr}$ . .

The continuum quality $\overline{EI}$ can be estimated by using the easily calculated values of axial spring rate which could be related to the properties of a simple tube so that

$$\delta = \frac{FL}{\overline{EA}} \text{ or } K_a = \frac{F}{\delta} = \frac{\overline{EA}}{L} \quad . \qquad \ldots(34)$$

From the equation

$$\bar{I} = (\bar{r})^2 \bar{A} \quad ,$$

$$\bar{r} = \text{radius of gyration}$$

$$\bar{r} = \frac{\bar{R}}{\sqrt{2}} \text{ for thin cylinders}$$

then

$$\bar{I} = \frac{(\bar{R})^2 \cdot \bar{A}}{2} \quad , \qquad \ldots(35)$$

and

$$\frac{\overline{EI}}{L} = \frac{F}{\delta} \cdot \frac{(\bar{R})^2}{2} = 0.5\,(\bar{R})^2 K_a \quad . \qquad \ldots(36)$$

To find rotational spring constant K_r,

$$\theta = \frac{ML}{\overline{EI}} \text{ or } K_r = \frac{M}{\theta} = \frac{\overline{EI}}{L} = 0.5\,R^2 K_a \quad , \qquad \ldots(37)$$

and to find the critical pressure, or internal pressure which would cause buckling when Q = 0,

$$F_{cr} = \pi R^2 P_{cr} = 4\pi^2 \frac{\overline{EI}}{L^2} = \frac{4\pi^2}{L} \cdot 0.5 R^2 K_a \quad , \qquad \ldots(38)$$

$$P_{cr} = \frac{2\pi K_a}{L} \quad . \qquad \ldots(39)$$

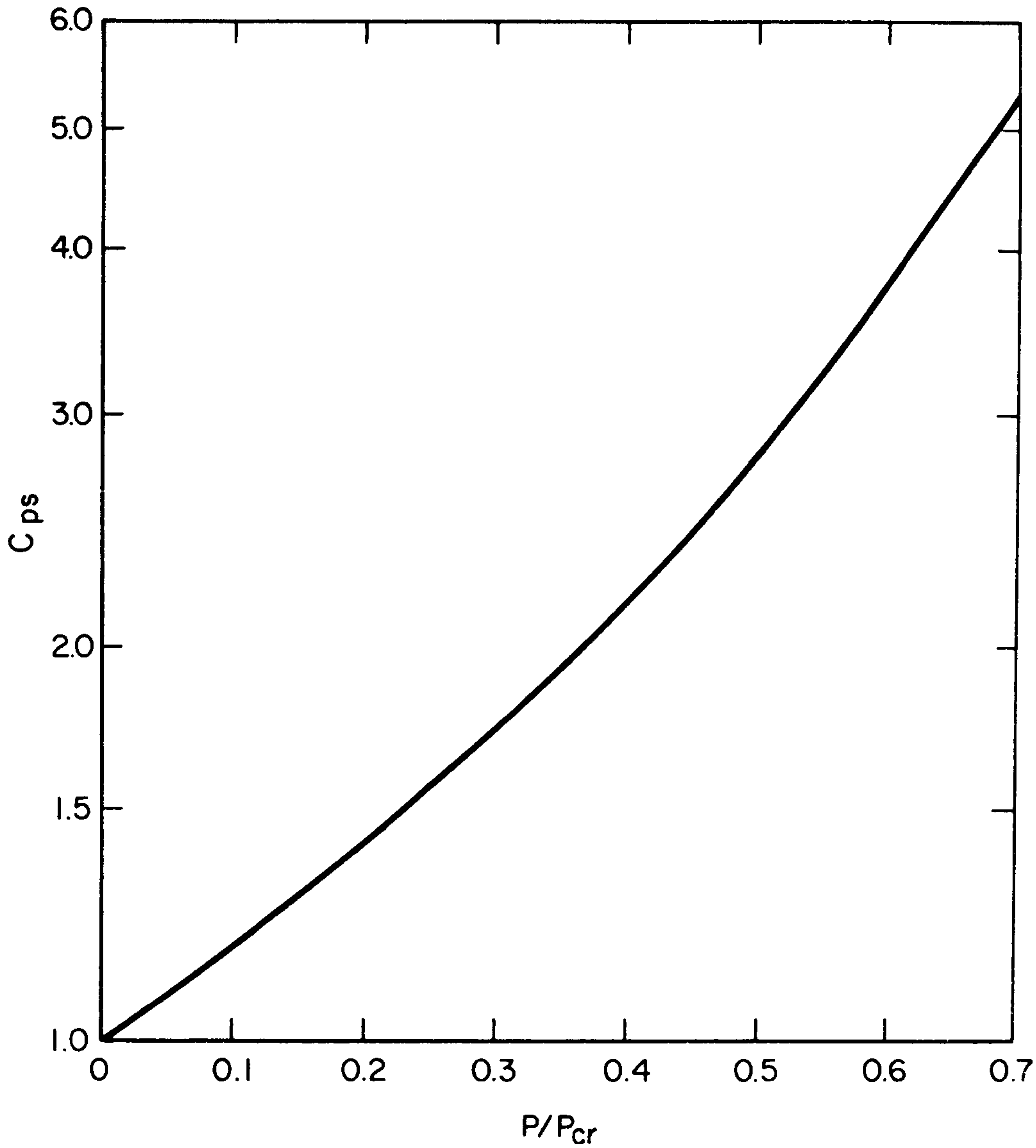

Figure 20. Factor for Increase in Angular Deflection Stress From Internal Pressure Cps - P/P_{cr}

APPENDIX B

MATHEMATICAL EVALUATION OF THE EFFECT OF PRESSURE STRESSES ON FATIGUE LIFE

From the empirically derived relation of cyclic stress to fatigue life of bellows without significant pressure stress

$$\text{Log } S'_{bd} = X - n \text{ Log } N_i \qquad \ldots(1a)$$

one can also state

$$S'_{bd} = \frac{X}{N_i^{\,n}} \qquad \ldots(1b)$$

where subscript "i" denotes individual values, prime denotes predicted value

S_{bd} = bending stress from deflection, psi

N = number of cycles to failure

X, n = empirically derived constants

C_{ps} = factor for increase of deflection stress due to internal pressure

S_{mp} = membrane stress induced by internal, pressure, psi

S_{bp} = bending stress induced by internal pressure, psi

The assumption is made that the value of "X" and "n" determined from tests on a group of bellows with no significant pressure stress determine the relation of cyclic stress to fatigue life for the entire population of bellows, including those bellows where significant pressure bending stress of undetermined effectiveness exists. Then a factor for the effectiveness of pressure bending stress on fatigue life "β" will be determined.

Given the test data from a bellows with internal pressure one can calculate a predicted stress from the given number of cycles Ni

$$S'_i = \frac{X}{(N_i)^n} \qquad \ldots(1c)$$

or to separate the stress components

$$S_i' = C_{ps_i} S_{bd_i}' + 2S_{mp_i}' + S_{bp_i}' = \frac{X}{(N_i)^n} . \qquad \ldots(1d)$$

This predicted stress would be compared to the actual calculated theoretical stress from the known conditions of pressure and deflection as

$$C_{ps} \cdot S_{bd_i} + 2S_{mp_i} + \beta S_{bp_i} = S_i \quad . \qquad \ldots(2)$$

It is further assumed that

$$C_{ps_i} S_{bd_i}' + 2S_{mp_i}' = C_{ps} S_{bd_i} + 2S_{mp_i} \qquad \ldots(3)$$

which implies that any error existing would result from the effectiveness of S_{bp} on cyclic life or

$$\epsilon_i = \beta S_{bp_i} - S_{bp_i}' \quad . \qquad \ldots(4)$$

When normalized by dividing by S' and squared to avoid the mathematical problems of using absolute values of the error then

$$\sum_{i=1}^{n} \epsilon^2_i = \sum_{i=1}^{n} \frac{\left(\beta S_{bp_i} - S_{bp_i}'\right)^2}{(S_i')^2} . \qquad \ldots(5)$$

To find that value of "β" which yields the minimum value of $\sum_{i=i}^{n} \epsilon_i^2$

one takes

$$\frac{\partial \Sigma \epsilon_i^2}{\partial \beta} = 0 = \sum \frac{2 S_{bp_i} (\beta S_{bp_i} - S'_{bp_i})}{(S_i')^2} \qquad \ldots(6)$$

which yields

$$\sum \beta \frac{\left(S_{bp_i}\right)^2}{\left(S'_i\right)^2} = \sum \frac{S'_{bp_i} S_{bp_i}}{\left(S'_i\right)^2} \qquad \ldots(7)$$

or as a constant

$$\beta = \frac{\sum_{i=1}^{n} \frac{S'_{bp_i} S_{bp_i}}{\left(S_i\right)^2}}{\sum_{i=1}^{n} \left(S_{bp_i}\right)^2 / \left(S_i\right)^2} \; . \qquad \ldots(8)$$

This yields a value of "β" which would result in the least squared error between the predicted values of total stress and the actual calculated values of stress which include pressure bending stress.

If more exact evaluation of the effect of pressure stress upon cyclic life were desired, a multiple non-linear regression would be a better way of evaluating the best fit value of "β". One general method of performing a multiple non-linear regression is to obtain an estimate of the unknown coefficients and then solve for corrections to these terms using multiple linear regression techniques.

The data may be expressed by

$$\text{Log } y' = A_o + R \text{ Log } (A_1 X_1 + A_2 X_2) \qquad \ldots(9)$$

where A_2 would be related to "β".
This could also be expressed as

$$\text{Ln } y' = n L_n (a_1 X_1 + a_2 X_2) \qquad \ldots(10)$$

where the values of n, a, and a_2 must be found for least squared error. Expressing the values found in the previous approximate analysis as n_o, a_{10}, and a_{20}

one could define the best fit values as corrections of these by

$$n = n_o + \epsilon_o \qquad \text{...(11a)}$$

$$a_1 = a_{10} + \epsilon_1 \qquad \text{...(11b)}$$

$$a_2 = a_{20} + \epsilon_2 \qquad \text{...(11c)}$$

then

$$Lny' = (n_o + \epsilon_o)\, L_n\left[(a_{10} + \epsilon_1)X_1 + (a_{20} + \epsilon_2)X_2\right] . \qquad \text{...(12)}$$

Expanding the right side as a Taylor series expression for natural logs and taking only the first term correction

$$Ln(X + \epsilon) = LnX + \frac{\epsilon}{1!} \frac{d(LnX)}{dX} = LnX + \frac{\epsilon}{X} \qquad \text{... 13)}$$

then

$$Lny' = n_o L_n(a_{10}X_1 + a_{20}X_2) + \epsilon_o L_n(a_{10}X_1 \times a_{20}X_2) \qquad \text{...(14)}$$

$$+ \frac{n_o \epsilon_1 X_1}{a_{10}X_1 + a_{20}X_2} + \frac{n_o \epsilon_2 X_2}{a_{10}X_1 + a_{20}X_2} . \qquad \text{...(14)}$$

This expression is linear in ϵ_o, ϵ_1 and ϵ_2 and so would allow the use of multiple linear regression techniques.

This method was not used, however, as: (1) no computer code to solve for these corrections was readily available; and (2) the method which was used would serve to supply the values of n_o, a_{10}, and a_{20} to be corrected. The uncorrected values proved to provide a correlation close enough for practical purposes.

APPENDIX C

SPECIFICATIONS FOR THE PURCHASE OF BELLOWS

TABLE OF CONTENTS

*Included in body of report - see contents page.

BELLOWS TYPE EXPANSION JOINTS

1. Scope This specification covers the requirements for design, fabrication, inspection, testing, and packaging of bellows type expansion joints which assure flexibility of piping systems for high temperature liquid metal service subject to thermal expansion or other movement. The bellows type expansion joint assembly consists of the convolutions or toruses, connections to piping or equipment, and hardware for support and restraint of movements defined by the design criteria. Application of this specification is limited to convoluted, ring reinforced convoluted and toroidal bellows as far as the shape of convolutions is concerned.

2. REQUIREMENTS

2.1 Design Conditions

	Item 1	Item 2	Item 3	Item 4	Item 5	Item 6
Number required						
Pipe size						
Pipe Material						
Pipe Connection type						
Type of bellows						
Type of hardware						
Design pressure, psig						
Design temperature, °F						
Operating temperature, °F						
Axial deflection, in.						
Lateral translation, in.						
Angular movement, °						
Number of lifetime cycles						
Torsional moment (obtained by piping stress analysis)						

2.2 Configuration of Bellows The convolutions and toruses shall be of commercially available configuration but limited to those specified in paragraph 1.

2.2.1 There shall be no appreciable variation of the spring constants of individual convolutions and toruses.

2.2.2 The shape of end convolutions and toruses shall be consistent with all other bellows elements.

2.2.3 The design of attachment between pipe and bellows material shall assure that no detrimental stresses will be generated which may cause the failure of the bellows material or the weldment.

2.3 Hardware

2.3.1 All bellows shall be fitted with permanent protective covers so spaced or articulated as to allow full movement of the bellows while minimizing the possibility of entry of foreign particles between convolutions or damage during the construction period.

2.3.2 Reinforcing rings, hardware, or other items in contact with the bellows material shall be smooth and free of projections which could induce failure.

2.3.3 Bellows hardware or packaging shall be fitted with stops or restraints to prevent excess deflection during the construction period.

2.4 Allowable Stresses and Stress Ranges The stresses due to internal pressure and thermal expansion shall not exceed the following allowable stresses and stress ranges where S_c, S_h and f are defined in the applicable section of the American Standard Code for Pressure Piping ASA B31.

2.4.1 Membrane stresses in the corrugations, reinforcing rings and hardware due to internal pressure (S_{mp}) shall not exceed the allowable stresses at design temperature (S_h).

2.4.2 In nonreinforced type bellows, the combination of membrane and bending stresses due to internal pressure ($S_p = S_{mp} + S_{bp}$) shall be limited to $1.25\ S_h$.

2.4.3 Shear stresses shall be limited by $0.5\ S_h$.

2.4.4 Combination of calculated membrane, bending and torsional stresses due to internal pressure and deflection caused by thermal expansion (S_{mp}, S_{bp}, S_{bd}, and S_t) shall not exceed the Code allowable stress range for combined stresses $S_{AC} = f\ (1.25 S_c + 1.25\ S_h)$.

2.5 Stability The factor of stability, that is the ratio of critical to operating pressure, must be greater than 5 if the bellows are designed for axial movement only and must be greater than 10 for bellows designed for combined movements.

2.6 Calculations Calculations to substantiate compliance with the requirements of paragraphs 2.4 and 2.5 shall be performed by the means of the attached Design Sheet Forms 1, 2 or 3, whichever is applicable and in accordance with the following definitions and equations:

2.6.1 Definitions

R - Mean radius of connecting piping, in.

h - Half wave height, in.

a - Radius of convolution, in.

t - Bellows thickness, in.

$\bar{R} = R + h$ - Mean bellows radius, in.

N_p - Number of plies

N_c - Number of convolutions

L - Overall active length of bellows, in.

A - Cross section area of the reinforcing rings

p - Internal pressure, psig

Δ_a - Axial deflection, in.

δ - Lateral translation, in.

$\Delta_b = \dfrac{6\delta\bar{R}}{L}$ - Lateral translation equivalent axial deflection, in.

Θ = Angular movement, radian

$\Delta_c = \bar{R}\Theta$ - Angular movement equivalent axial deflection, in.

$\Delta = \Delta_a + \Delta_b + 1.2\Delta_c$ Total equivalent axial deflection, in.

M_t - Torsional moment, in lb.

φ - Angular rotation due to torsional moment, radian

E_c - Young's modulus at room temperature, psi

E_h - Young's modulus at operating temperature, psi

$\nu = 0.3$ Poisson's ratio

2.6.2 Equations

(a) Membrane stresses due to internal pressure:

$$S_{mp} = \frac{ph}{N_p t} = \frac{pRa}{\left[\left(\frac{\pi}{2} - 1\right) a + h\right] N_p t}$$ for convoluted bellows

$$S_{mp} = \frac{ph}{N_p t}$$ for ring reinforced and toroidal bellows

$$S_{mp} = \frac{phL}{\Sigma A}$$ for reinforcing rings

(b) Bending stresses due to internal pressure:

$$S_{bp} = \frac{2C_p ph^2 R}{t^2 N_p R}$$

Obtain constant C_p from Figure 1 for convoluted bellows, from Figure 4 for ring reinforced convoluted bellows.

(c) Bending stresses due to deflection:

$$S_{bd} = \frac{0.412\, E_c t \Delta \bar{R}}{h^2 N_c C_d R}$$

Obtain constant C_d from Figure 2 convoluted bellows, from Figure 5 for ring reinforced convoluted bellows and from Figure 7 for toroidal bellows.

(d) Combined membrane and bending stresses due to internal pressure and deflection:

$$S_d = S_{mp} + S_{bp} + 0.5S_{bd}$$ for convoluted bellows

$$S_d = S_{mp} + 0.25S_{bp} + 0.5S_{bd}$$ for ring reinforced convoluted bellows

$$S_d = 1.5S_{mp} + 0.5S_{bd}$$ for toroidal bellows

(e) Torsional stresses:

$$S_t = \frac{M_t}{2\pi R^2 t N_p} = \frac{\varphi E_h R}{10.4 N_c (.57\, a + h)}$$

Where M_t may be obtained from the stress analysis of the piping configuration or φ from the clearances maintained in the hardware of restraints.

(f) Combined total membrane, bending and torsional stresses:

$$s = \sqrt{S_d^{\,2} + 4S_t^{\,2}}$$

(g) Axial spring rate:

$$K_a = \frac{0.431\ \overline{R}\ E_h t^3 N_p}{N_c h^3 C_f} \qquad \text{lb/in.}$$

Obtain constant C_f from Figure 3 for convoluted bellows, from Figure 6 for ring reinforced convoluted bellows and from Figure 8 for toroidal bellows.

(h) Rotation spring rate:

$$K_r = 0.5\ \overline{R}^2\ K_a \quad \text{in.lb/radian}$$

(i) Factor of stability:

$$F_s = \frac{2\pi K_a}{L_p}$$

2.7 Material

2.7.1 Material in contact with the fluid shall be austenitic stainless steel, any 300 series. Minimum thickness of bellows material shall not be less than 0.030 in. nominal.

2.7.2 Material not in contact with the fluid such as reinforcing rings and hinge or gimbal hardware may be of other than 300 series stainless for reasons of reduction of wear or strength, provided it is compatible with the other materials.

2.7.3 All materials shall be corrosion resistant and metallurgically stable under long time exposure at operating temperatures.

2.8 Fabrication All fabrication shall be performed in accordance with the provisions of the pertinent section of the American Standards Code for Pressure Piping ASA B31 unless otherwise noted in this specification.

2.8.1 Welding Welding shall be according to Atomics International Specification NA0107-002.

2.8.2 Passivation Passivation of the bellows material shall be according to the supplier's process or to commercially acceptable practice, if applicable. When applicable the process shall be submitted to Atomics International for in-information.

2.8.3 Heat Treatment Heat treatment of the bellows material shall be cording to the supplier's process or to commercially acceptable practice, if applicable. The process shall be submitted to Atomics International for information.

2.9 Cleaning Cleaning shall be done according to commercially acceptable practice to remove dirt, grit, scale, weld spatter, grease, oil, cleaning fluid, and rinsing water.

2.9.1 Halogen-ion bearing cleaning solutions shall be prohibited.

2.9.2 All openings shall be sealed immediately after cleaning to prevent entry of foreign matter and moisture during transit and storage.

2.9.3 All gas, or moisture, or both trapped between layers of multi-ply bellows shall be evacuated and the space sealed at design temperature to prevent deformation of the bellows caused by thermal expansion of any gas present.

2.10 Identification Identification of the product to be in accordance with commercially accepted practice using an electro-chemical marking on a non-critical stress area. Impression stamping or vibrating needle identification shall not be used.

2.11 Packaging The bellows shall be packaged to prevent damage and contamination during handling, shipment, and storage, and be acceptable by common carriers.

3. INSPECTION AND TESTS

3.1 Access An Atomics International representative shall have access to the Seller's plant for surveillance of work at any time during the fabrication, assembly, inspection, testing, and packaging of the equipment.

3.2 The Seller shall supply all materials and equipment for conducting, and conduct all necessary inspections as required by this specification.

3.3 The bellows material shall be determined to be free of injurious defects by definitive inspection methods prior to forming.

3.4 After forming, the bellows shall be determined to be free from injuriour defects by definitive inspection methods.

3.5 All welds of the expansion joint assembly shall be radiographed in accordance with the provisions of Atomics International Specification NA0107-002. Where radiography is not practical, liquid penetrant examination may be substituted. Application of the liquid penetrant shall be done by painting or dipping.

3.6 Visual examination and measurement of the completed expansion joint assembly shall determine that variances of fabrication such as notches, crevices, material thinning, metal buildup, upsetting or others will not serve as points of stress concentration or stress raisers.

3.7 The completed expansion joint shall be tested for leakage by the helium leak detection method. The maximum allowable leakage shall be 1×10^{-7} cc per second for a one atmosphere pressure differential at standard conditions. The Seller shall submit his detailed procedure to Atomics International and obtain Atomics International approval before start of testing.

3.8 All inspection and test reports and forms including material certifications shall be submitted to Atomics International.

4. ENGINEERING DATA The seller shall furnish the following data for Atomics International's review and approval prior to fabrication.

a. Bellows assembly drawing including the following information: configuration, weldments, controlling dimensions, tolerances, materials, finishes, applicable specifications.

b. Design sheet forms completed with calculations.

c. Fabrication procedures.

d. Inspection and test procedures.

3.4 After forming, the bellows shall be determined to be free from injuriour defects by definitive inspection methods.

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