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Hermetic Bellows Seal for Braun Linear Engine — Design, Test, and Demonstration

Final Report

Submitted by

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under
Subcontract 86X-61613C

for

OAK RIDGE NATIONAL LABORATORY
operated by
MARTIN MARIETTA ENERGY SYSTEMS, INC.
for the

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ABSTRACT

The work performed on this project consisted of defining seal requirements, analyzing and testing seal performance to develop a design data base, and applying that design criteria to evolve a seal solution suitable for application to a light-commercial-sized heat pump.

At the point where the compressor piston rod joins the drive mechanism of the BR-105 linear engine, a linear acting seal is required to ensure the hermetic enclosure of the refrigerant (R-22) system. The estimated maximum pressure difference across the seal, under all conditions including start-up, will be 80 psi, and the estimated temperature range will be -30° to 100°F. Maximum life goal is 15 years with operation of at least 4000 hours per year. The seal will be exposed to an oil/refrigerant mixture on one side and atmosphere contaminated with combustion products on the other side. The rod diameter is 1.0 inch, the maximum piston stroke is 6 inches and the speed range is from 800 to 1200 cycles per minute.

After consideration of the requirements, the evolved seal design consists of two or more bellows elements placed in series and joined to the underface of the piston at one end and to the crankcase at the other. The intermediary ends of the bellows are joined together so that all the joints are made leak tight. The compressor piston rod is enclosed within the bellows. Bellows elements are driven at their point of junction by a mechanism synchronized to the piston motion but so constrained as to limit the stress in each element, thereby insuring an effective, long-life hermetic seal.

EXECUTIVE SUMMARY

This is a final report on a project for Test and Demonstration of a Hermetic Compressor Seal (Phase I), Union Carbide Contract No. 86X-61613C. The period of performance was from July 1, 1980 through January 31, 1981. The first draft report was submitted March 6, 1981.

The objective of the project was to perform the analysis, design and tests leading to an early demonstration of a hermetic seal for a linear-engine-driven compressor for application in a heat-actuated heat pump system.

The technology base of this seal development program was based on previous work by Tectonics. A metal bellows seal had been used successfully on a lever-type engine/refrigerant compressor (Model BR-300) where the dynamic motion of the seal was oscillatory as opposed to the linear motion of the BR-105 engine/compressor. Some early work also had been done on the concept of a linear seal solution wherein the necessary drive mechanism had previously been put to practice.

The effort on this program concentrated on the development and advancement of a linear motion metal bellows hermetic seal. Thus, wherever "seal" is referenced in this report, it will refer to the linear metal bellows seal.

A major effort on the program was the adaptation of analytical methods necessary to predict the dynamic behavior and failure modes of the seals as a function of their design, material of construction and environmental parameters. Once refined, the engineering analysis procedures were used to investigate the suitability of available off-the-shelf seals to the expected heat pump application.

Actual seals, specified from engineering analysis and available standard shelf stock, were procured from two seal manufacturers and were tested under various conditions and in various combinations. Test rigs were adapted for the seal tests so that both sinusoidal motion and actual engine dynamics of the heat pump drive could be investigated. The sinusoidal drive permitted precise variation of speed and displacement for analytical correlation. Correlating these test results with predicted performance validated the accuracy and suitability of the analytical methods.

By integrating the test and analytical results, a suitable seal solution to the BR-105 application was defined and developed. This solution, in turn, has been analyzed, tested and demonstrated. The solution is shown to be

compatible with the requirements for a linear-motion hermetic refrigerant seal. Further, based on the materials used and seal manufacturer's experience, the seal solution is shown to exhibit stresses low enough to be considered to have "infinite life."

TECHNICAL DISCUSSION

Seal Procurement

The seal development effort was predicated on the analysis and testing of existing seals selected from manufacturers' stock. This approach served two important functions. First, program analyses would benefit from the manufacturers' extensive experience, which would provide data for accurate seal stress and life expectancy calculations. Second, a seal solution based on available off-the-shelf materials would provide a timely and reliable input to the heat pump development program to follow.

Two independent seal manufacturers were selected as suppliers for this project, a primary supplier and a backup supplier. Six metal bellows seals were ordered in the early part of the program from the primary supplier. Two more were ordered later from the backup supplier. Table 1 gives a description of these purchased seals.

Seal design and selection followed preliminary in-house performance analysis and comprehensive discussions and visits with the manufacturers. The seals were selected to allow:

- Maximum flexibility of arrangement and combination of conceivable alternatives in seal arrangement,

- Configuration compatibility with the engine/compressor,
- Reasonable assurance of desired strength and life expectations.

The present seals are constructed of readily available steel and shaped to form a compressible/expandable bellows.

Table 1. Linear Seals Procured for the Project

SEAL NO.	DESCRIPTION	QUANTITY
1	BELLOWS ASSEMBLY, 0.003-INCH PLATE THICKNESS, 1.00-INCH I.D., 1.75-INCH O.D., AM-350, CONDITION H, 30 CONVOLUTIONS	1
2	AS ABOVE WITH 39 CONVOLUTIONS	1
3	AS ABOVE WITH 78 CONVOLUTIONS	1
4	AS ABOVE WITH 30 CONVOLUTIONS	1
5	AS ABOVE WITH 60 CONVOLUTIONS	1
6	AS ABOVE WITH 78 CONVOLUTIONS	1
7	BELLOWS ASSEMBLY, 0.007-INCH PLATE THICKNESS, 0.500-INCH I.D., 1.00-INCH O.D., 60 CONVOLUTIONS	2

Throughout the program, discussions were held with the manufacturers to determine the practicality of seal design options suggested by analysis and test results.

Analysis Procedures

Earlier work with seals at Tectonics revealed that for a successful development of a reliable and practical seal solution a thorough understanding of seal behavior under dynamic conditions was necessary. This required a strong analytical base so that the seal behavior could be analyzed under all the possible conditions that a seal may experience in the Braun linear engine-refrigerant compressor system.

A literature review was conducted to gather available information on the subject matter. It was discovered that, although there exists a wealth of information on the dynamics of springs, and that such analysis is relatively straightforward, a considerable amount of effort and original thinking would be required to utilize the standard methods for the present application.

A lumped-parameter method was followed in the analysis, where the primary outputs become the amplitude of oscillation of individual seal convolutions and the fundamental and resonant frequencies. The amplitude information was used to calculate stress levels that were then plotted in the form of Goodman diagrams[1] to determine endurance strength and seal life. Infinite life is considered to occur for every point within the bounds of the endurance limit envelope. The fundamental and resonant frequencies output provided information on the forbidden range for seal operation.

Manufacturer's data consists of formulas, graphs, and charts for accurate seal stress calculations and requires input from dynamic analysis or measurements of actual deflections. Existing in-house analysis procedures were modified to produce the necessary input data for the manufacturer's stress nomographs.

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[1]Shigley, Joseph Edward, Mechanical Engineering Design, 3rd Ed., (New York: McGraw-Hill, 1977).

Comprehensive parametric studies of seal dynamics were performed in order to understand seal behavior, explore and predict failure modes, and optimize design of a practical seal. The parameters varied included material thickness, spring rate, number of convolutions, pressure differential across seal wall, acceleration (drive speed), and driving force function.

The initial studies included simple cases of seal configurations that, combined with actual seal testing, provided a means of checking the validity of and establishing confidence in the analytical method for the present application.

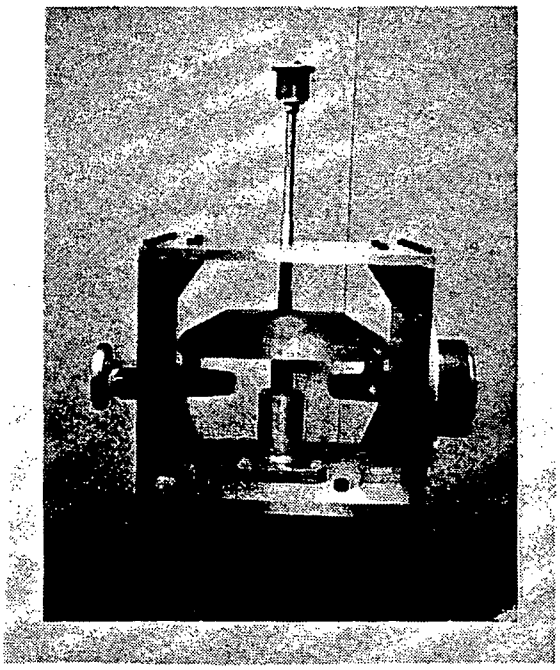
Seal Test Description

Two test drive mechanisms were available for this program: a sinusoidally driven BR-420 machine and a BR-105 engine. The BR-420 fixed-stroke machine provided a simple and manageable tool where different speeds were obtained by changing drive pulley size. The BR-105 in turn provided the actual engine dynamic environment.

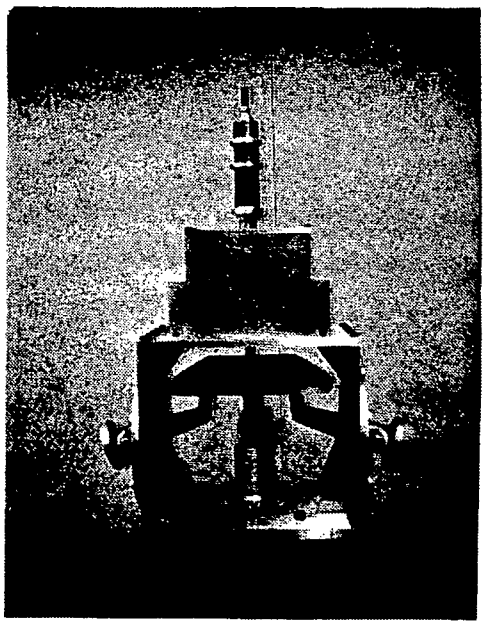
Test rigs for the two test beds were designed and fabricated to allow testing of single or multiple seals, incorporating a proprietary drive mechanism. Figures 1 and 2 are photographs of the two test rigs with seals and drive mechanisms assembled.

The seals were tested for their dynamic behavior by observing and measuring the motion of the individual convolutions as well as following the seal motion for beats or surges. The method of observation and measurement is described below.

Still photography of seal motion was used to study and record seal behavior and convolution displacements. A General Radio 1531-A stroboscope in combination with a 1531-P2 flash delay unit and a 1537 photoelectric



a. Single-Seal Arrangement



b. Multiple-Seal Arrangement

Figure 1. Fixed-Stroke Test Rig (BR-420) With Linear Drive Mechanism

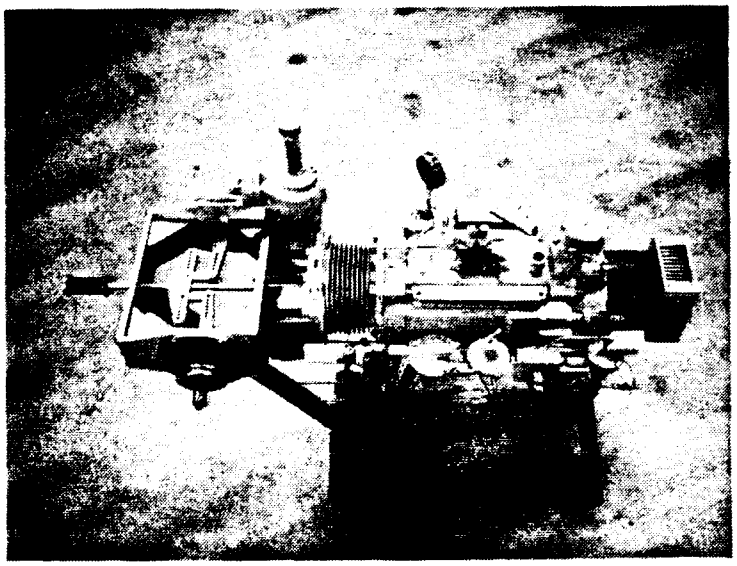


Figure 2. BR-105 Test Rig With Linear Seal (at far left) and Drive Mechanism

pick-off was used for freezing the seal motion at any desired position. The equipment provided proper synchronization of the strobe light flashes with the subject seal motion. Figure 3 shows the photographic setup used for the tests.

Photographs of the seals were made at different points of stroke. (Figure 7, to be discussed later, is a typical example). The space between two adjacent convolutions can be measured from these photographs. With the camera magnification factor known, the actual distance between the convolutions can be calculated.

Single Seal Analysis and Test Validation

Eleven variations of a single seal arrangement (as listed in Table 2) were defined for the initial analysis. The arrangements were selected to produce

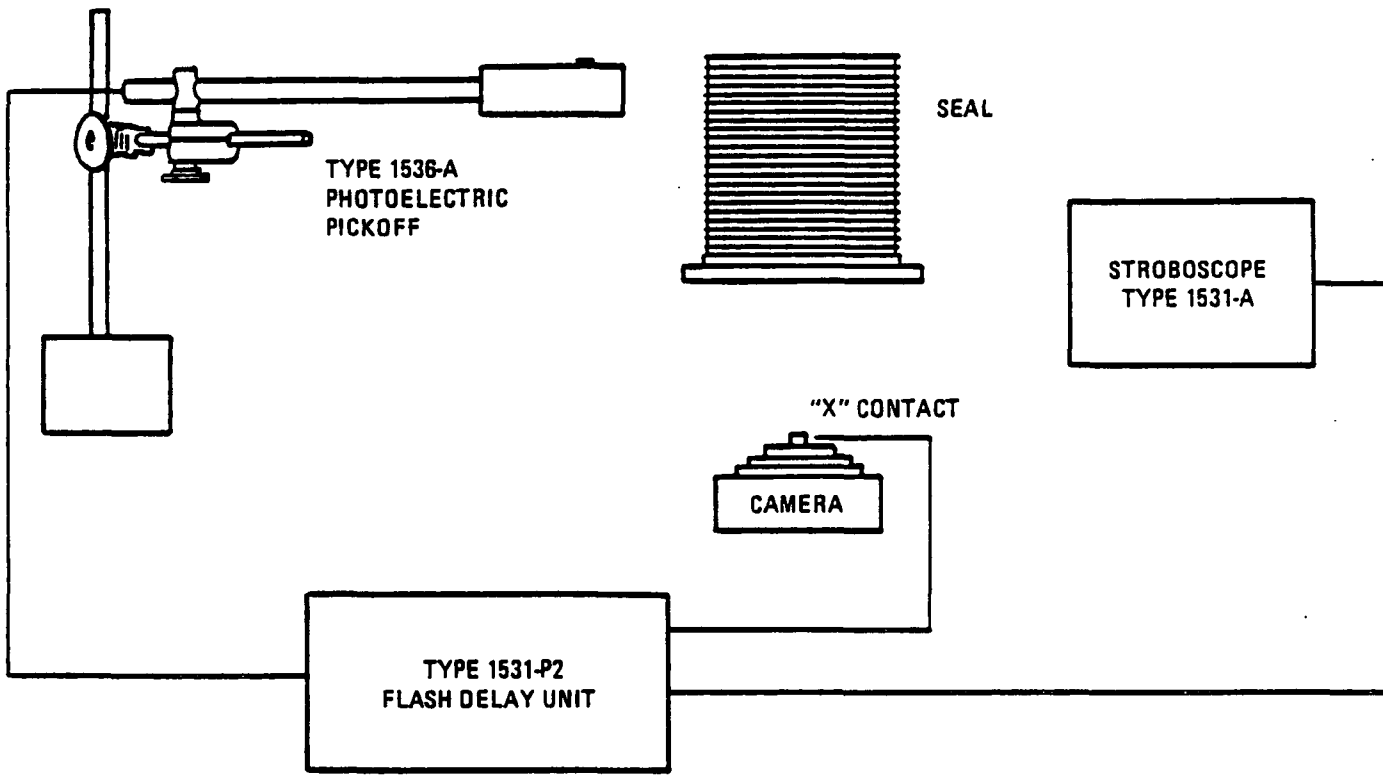


Figure 3. Setup for Photographing Seal in Motion

results that could later be correlated with actual tests. The simulated test conditions relate to conditions, shown schematically in Figure 4.

In the eleven analytical cases, the variables investigated included bellows material thickness, number of bellows convolutions, and linear speed (cycles per minute).

Table 2. Analysis/Test Cases--Single Seal

CASE	RELATIVE DISPLACEMENT FLANGE$_2$ TO FLANGE F_1	DESCRIPTION	BELLOWS PLATE THICKNESS (INCHES)	SPEED (CYCLES PER MINUTES)	NUMBER OF CONVOLUTIONS
1	$F_2=F_1$	SHAKER ARRANGEMENT	0.003	1000	31
2	F_2=VAR	FREE END	0.003	800	31
3	$F_2=0.8\ F_1$	DIFFERENTIAL DRIVE	0.003	1250	39
4	$F_2=0.7\ F_1$	DIFFERENTIAL DRIVE	0.003	1250	39
5	$F_2=0.8\ F_1$	DIFFERENTIAL DRIVE	0.003	800	39
6	$F_2=0.7\ F_1$	DIFFERENTIAL DRIVE	0.003	800	39
7	$F_2=0$	FIXED END (SAME AS ACTUAL INSTALLATION)	0.003	1000	30
8	$F_2=0.8\ F_1$	DIFFERENTIAL DRIVE	0.003	1000	31
9	$F_2=0.8\ F_1$	DIFFERENTIAL DRIVE	0.005	1000	31
10	$F_2=0.8\ F_1$	DIFFERENTIAL DRIVE	0.003	2000	31
11	$F_2=0.8\ F_1$	DIFFERENTIAL DRIVE	0.005	2000	31

Note: Each case was analyzed on the basis of inputs from the schematic arrangement shown in Figure 4 using the BR-420 fixed-stroke drive.

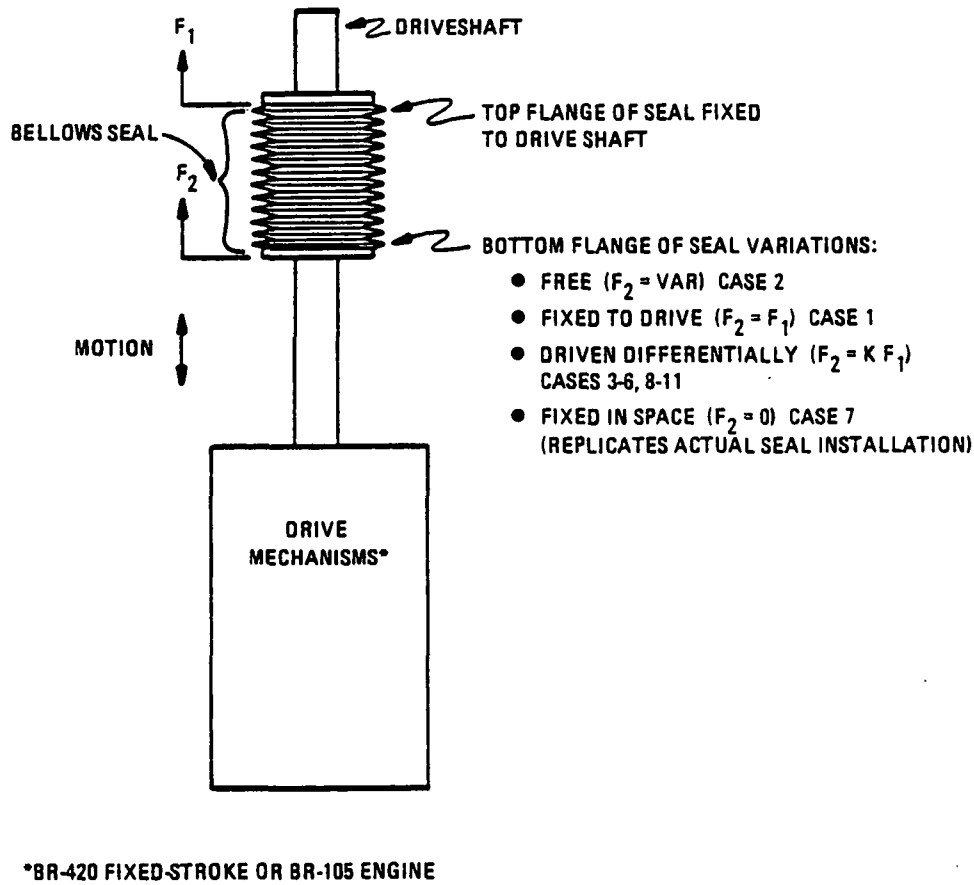


Figure 4. Analytical and Test Arrangements of Single Seal

For each of the defined test cases, the maximum displacement amplitude of each seal convolution was analytically determined as represented in Figure 5.

As background for the data of Figure 5, the case 4 results are elaborated in Figure 6, which illustrates the deflections for the full-stroke cycle from top dead center (TDC) to bottom dead center (BDC). The two maxima are symmetrical, assuming no mechanical limits are encountered. The analytical deflection values plotted in Figures 5 and 6 are relative to the displacements at the midstroke position.

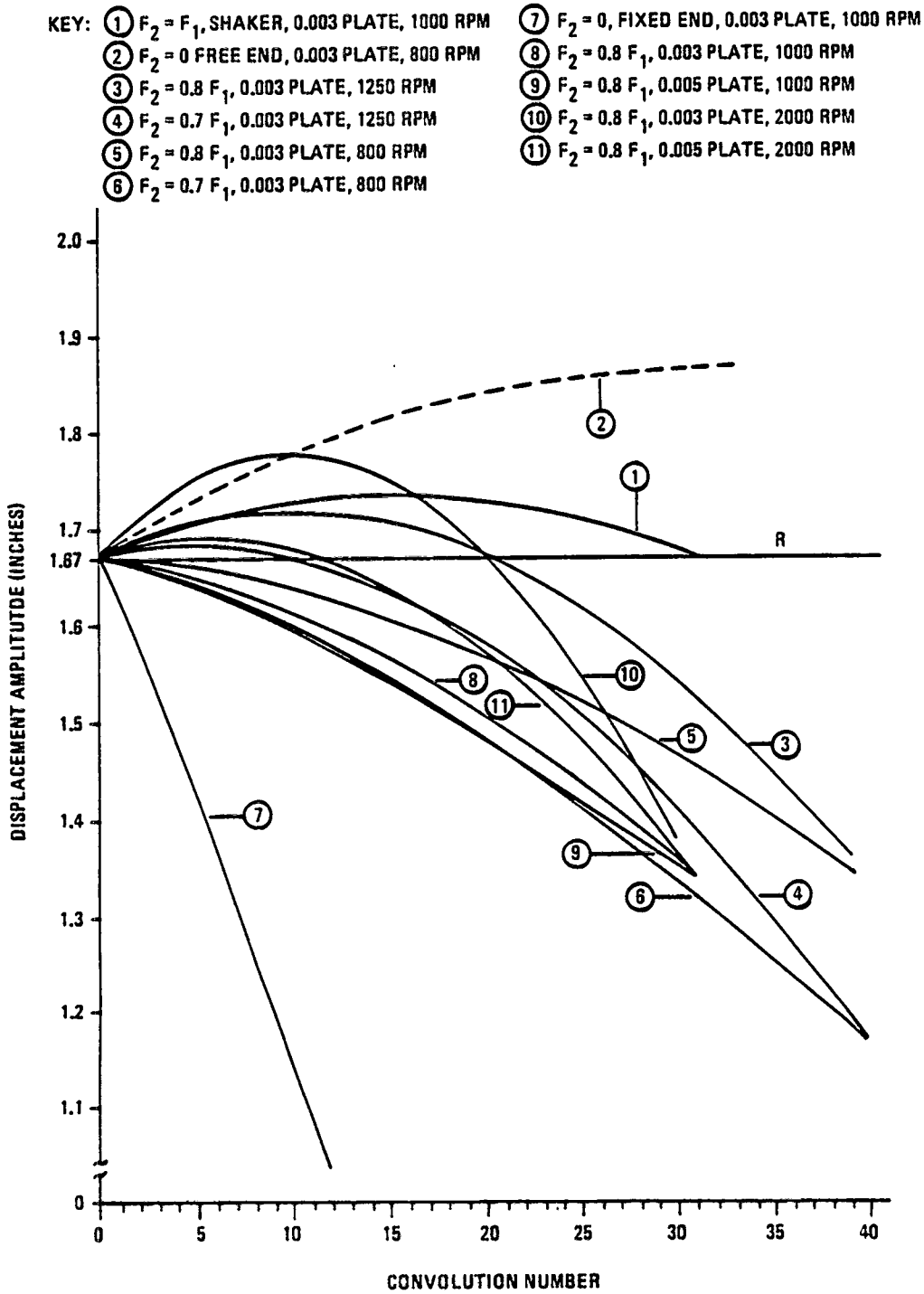
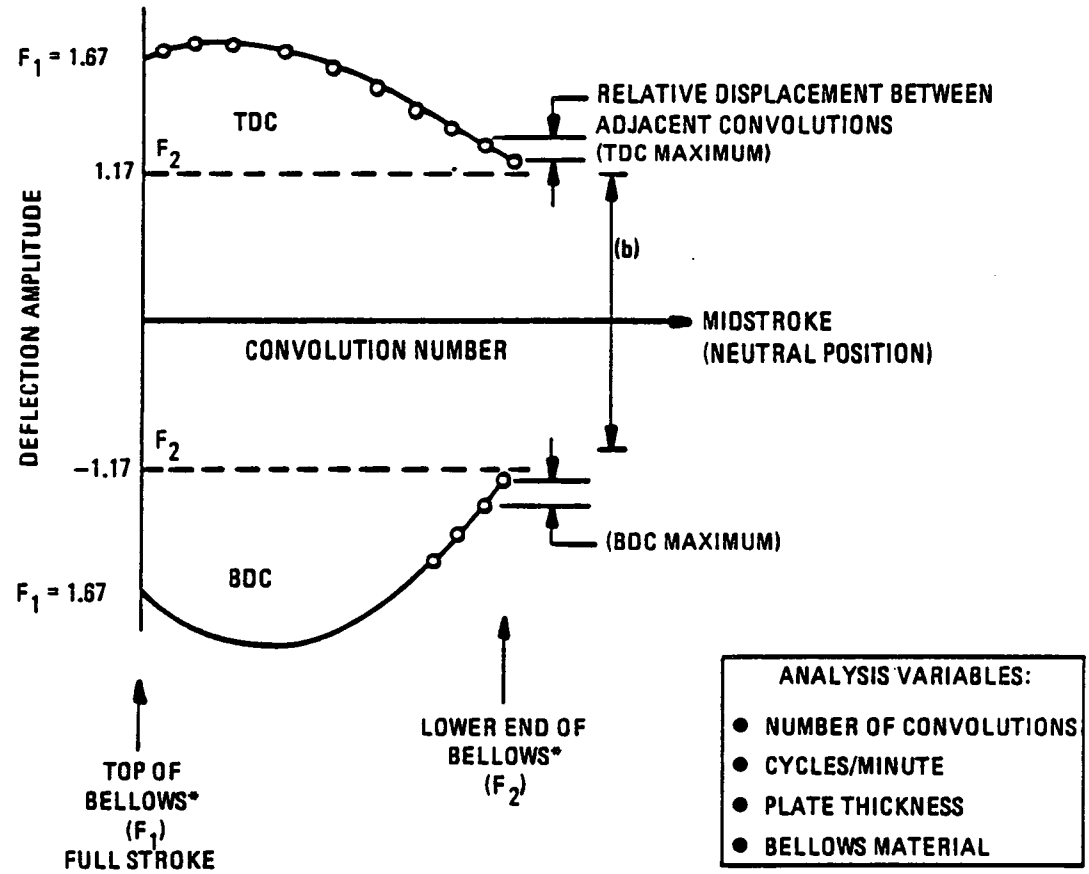


Figure 5. Seal Convolution Displacement for Single-Seal Amplitudes Under Dynamic Conditions (Analytical Results)



*CONSTRAINED TO DEFLECTIONS SHOWN (e.g., $F_2 = 0.7\ F_1$)
CASE 4 OF GENERAL CASE $F_2 = KF_1$

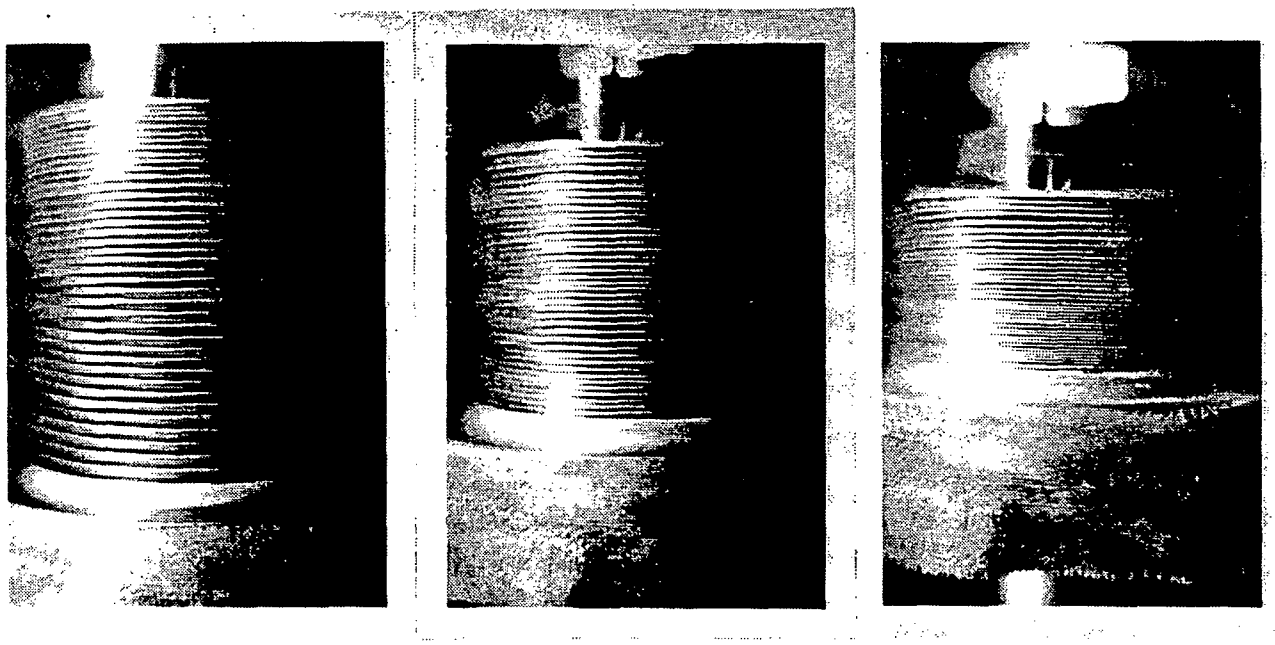
Figure 6. Single Bellows Seal Analysis Representation (Case 4)

The parameter of interest from this analysis is the relative displacement of adjacent convolutions, with the maximum such value for a given case being subsequently used in stress calculations. This maximum value will occur where the slope of the deflection curve is steepest.

To further illustrate the derivation of deflection values, reference is made to an actual test sequence illustrated in Figure 7. This figure presents strobe photos of an actual seal at top and bottom dead centers and at mid-stroke. Referring to Figures 6 and 7, which represent the same test case (No. 4), the following points are evident:

- At the bottom of the bellows (last several convolutions, e.g., 35 to 39, Figure 6), the bellows convolution spacing can be seen to go through the widest variation through the full stroke. Quantitatively, the convolute spacing at the F_2 end (TDC and BDC MAX) is several times the convolute spacing of the midconvolutes at all strokes, except at the neutral midstroke position. The cause of this is the delay in momentum transfer (inertia effect) as the flexible convolute sections move out of phase with each other and pile up as the direction of motion changes. The displacement quantities plotted in Figure 6 are the incremental displacements between midstroke and TDC/BDC. Thus, since TDC and BDC data are symmetrical, the total deflection throughout the stoke between the last convolution segments is twice the plotted value of the TDC value.

- A second interesting correlation between the analytical results and the actual test is made by examining seal behavior at about the fourth and fifth convolution from the top of Figure 7. Figure 5 shows that there is no relative displacement (zero slope) between the convolutions. The statements in the preceding paragraph imply that the spacing between those elements remains constant throughout the stroke. Qualitatively, this behavior is evident on the photos of Figure 7.



i) at TDC ii) at mid-stroke iii) at BDC

Figure 7. Typical Photographs of the 39 Convolution Seal at Different Points of Stroke (BR-420 Drive)
Data: $F_2 = 0.7\ F_1$, Case 4
RPM = 1250
Seal Plate Thickness = 0.003 Inch

Thus, it is possible to qualitatively assess the validity of the analysis in predicting seal behavior. Quantitative comparisons also were made by measuring actual displacements from the stroke photos and comparing results with those predicted, as shown in Table 3. Agreement between test and analysis shows good correlation, considering the margin for error when measuring small distances from the strobographic photos.

Table 3. Maximum Displacement per Convolution for the 39 Convolution Seal (0.003-Inch Material), Predicted and Measured Values in Sinusoidal Motion

CASE NUMBER	RPM	F_2	MAXIMUM DISPLACEMENT (DEFLECTION) PER CONVOLUTION	
			PREDICTED (INCHES)	MEASURED (INCHES)
3	1250	0.8 F_1	0.048	0.046
4	1250	0.7 F_1	0.054	0.052
5	800	0.8 F_1	0.029	0.028
6	800	0.7 F_1	0.036	0.041

Stress Analysis

The maximum convolution displacement for each seal case, measured as well as calculated, was used to calculate maximum seal stress levels. The resultant stress levels are presented in the form of Goodman diagrams[1] wherein the endurance and life of the seal can be exhibited.

Figure 8 shows the resultant Goodman stress diagram for the four cases listed in Table 3. The endurance limit envelope is representative of the steel used in the manufacture of the tested bellows seal. "Infinite" seal life is assumed to be demonstrated when all stress points lie within the endurance

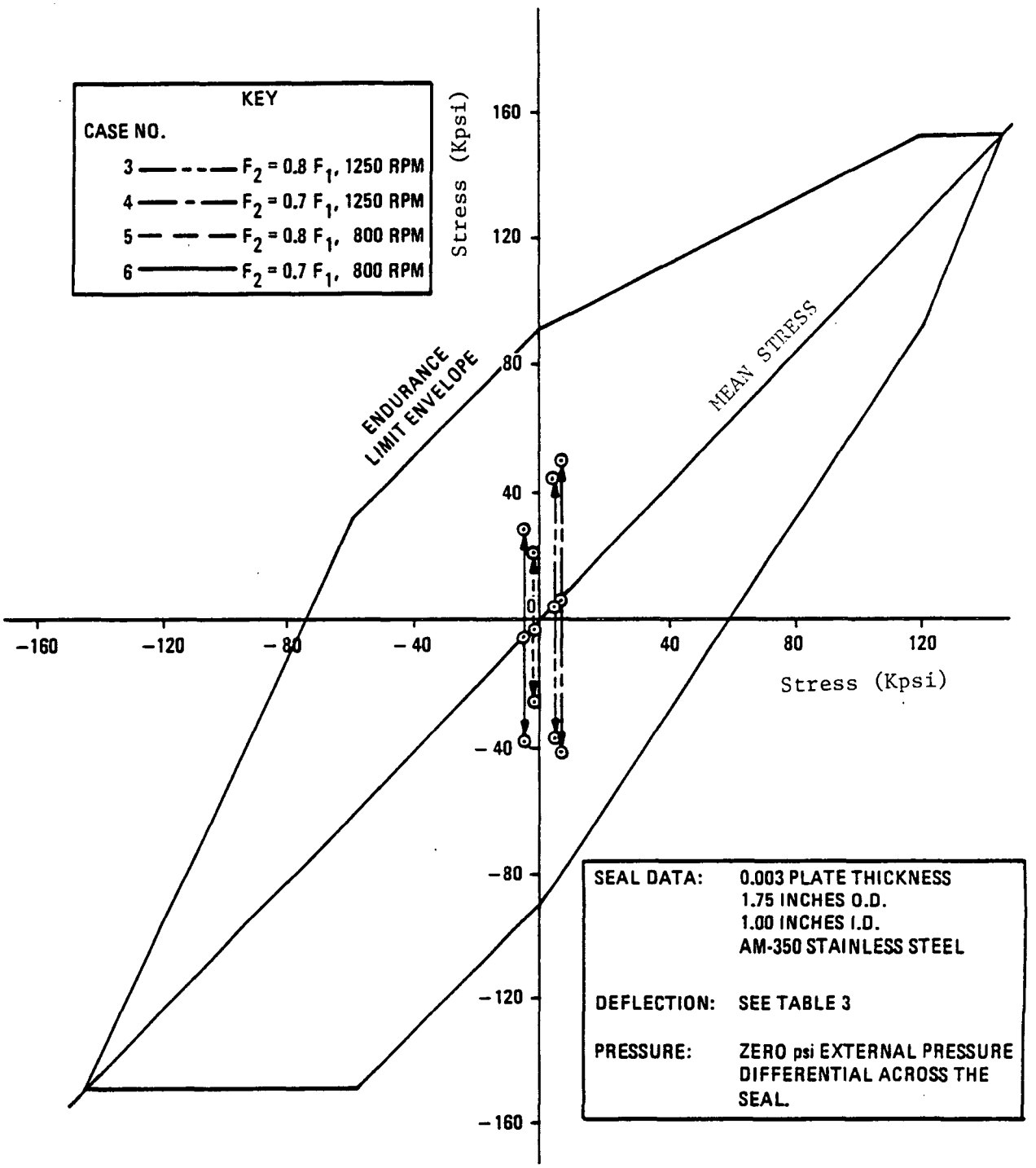


Figure 8. Effect of Speed on Stress Levels in a 39-Convolution Seal

limit envelope. While the four test cases do meet this criterion, they do not represent a "solution" since an actual seal installation is not represented. For such an actual seal installation (e.g., case 7 of Table 2), the maximum deflection is beyond the range of the manufacturer's charts for stress calculation.

From this early work, the measured maximum deflections were felt to be acceptable to meet stress criteria. Using that deflection, an analysis was made to determine the influence of seal plate thickness and pressure differential across the seal on stress levels. Figures 9 and 10 show the stress levels in the Goodman diagram of different plate thicknesses and pressures across the seal. This parametric analysis led to concentration on one specific plate for the seals.

Up to this point the analysis and tests had been directed toward developing an understanding of the general behavior and dynamics of linear seals. Analysis procedures were improved and exercised to provide the framework to extend the seal analysis toward a practical solution for the BR-105 engine-compressor.

A basic philosophy applied to this applications effort was to utilize existing, available bellows stock materials, as opposed to a direct seal design effort. This approach was felt to offer the strongest probability of a timely seal solution, as well as providing the availability of existing manufacturer's stress data based on extensive field experience.

For a fixed machine stroke, a single-seal application was investigated analytically (i.e., case 7). A seal with the selected plates, which would result in a certain nominal nondynamic deflection per convolution, was assumed. The resultant dynamic deflection is plotted in Figure 11, a diagram equivalent to Figure 5 (discussed earlier). The maximum slope gives an equivalent of the maximum convolution displacement. This deflection is well beyond the range of the manufacturer's design stress data and would result in "immediate" seal failure. This has been demonstrated.

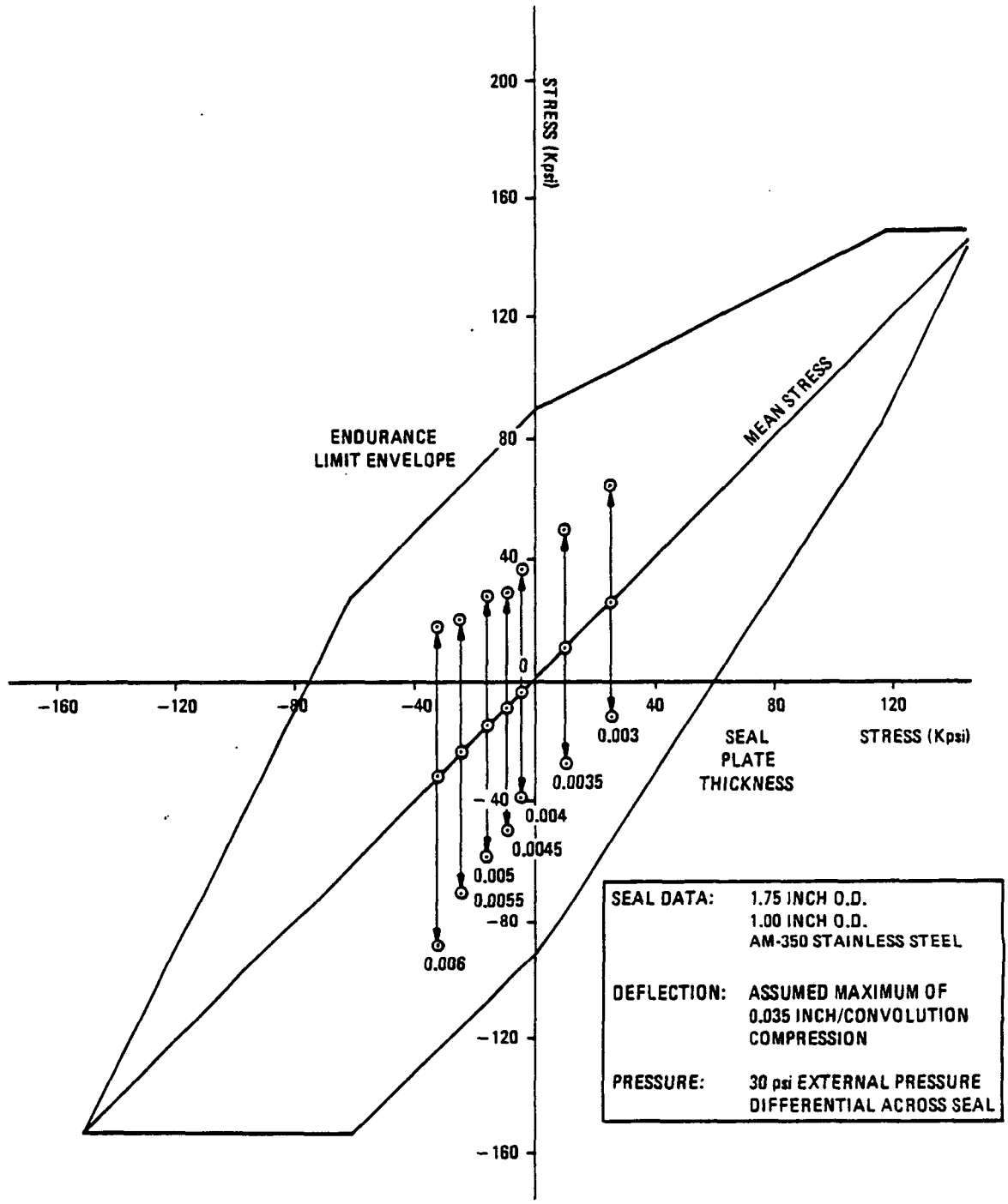


Figure 9. Effect of Plate Thickness on Stresses in Metal Bellows Seals at 30 psi Differential Pressure

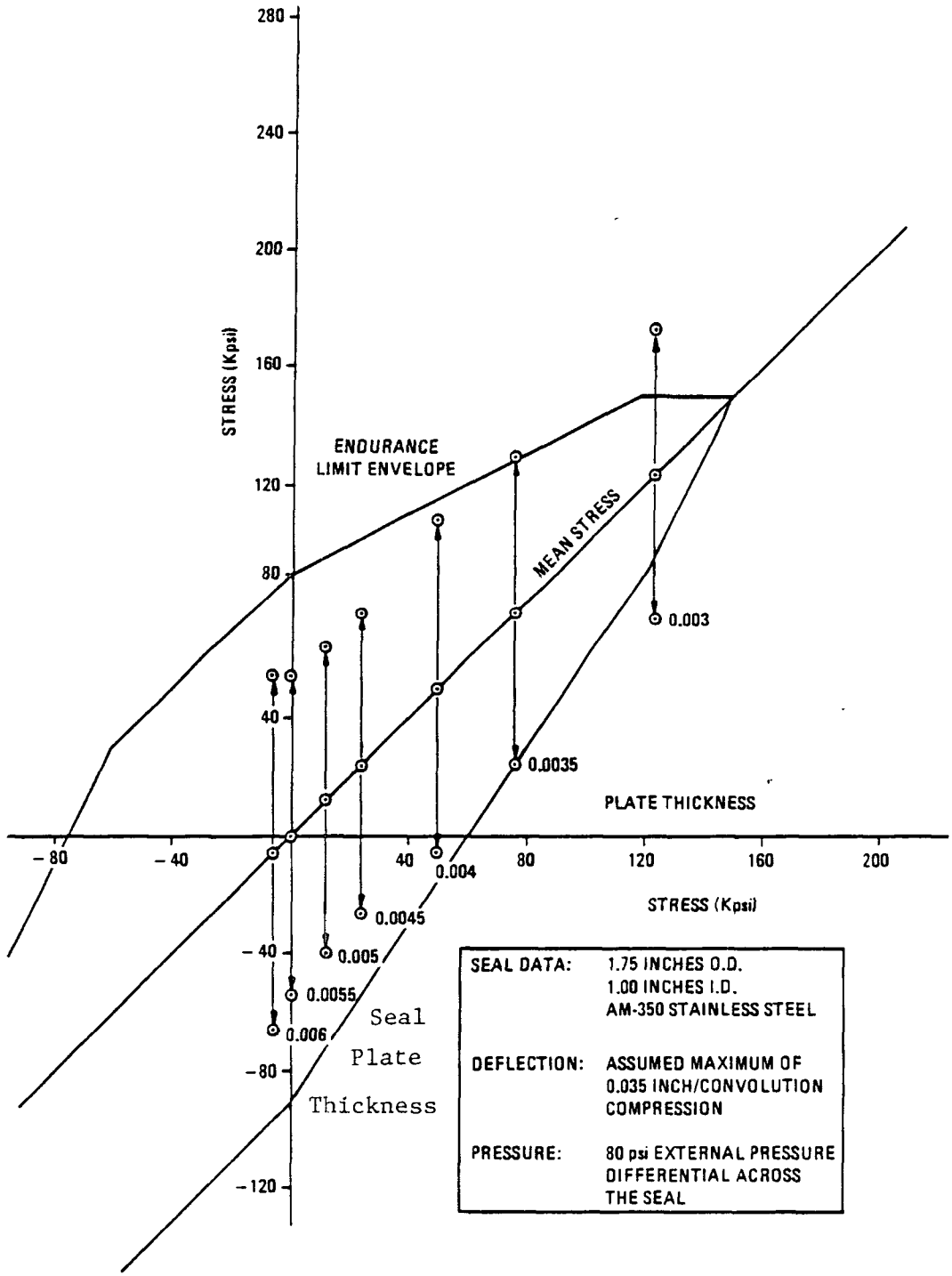


Figure 10. Effect of Plate Thickness on Stresses in Metal Bellows Seals at 80 psi Differential Pressure

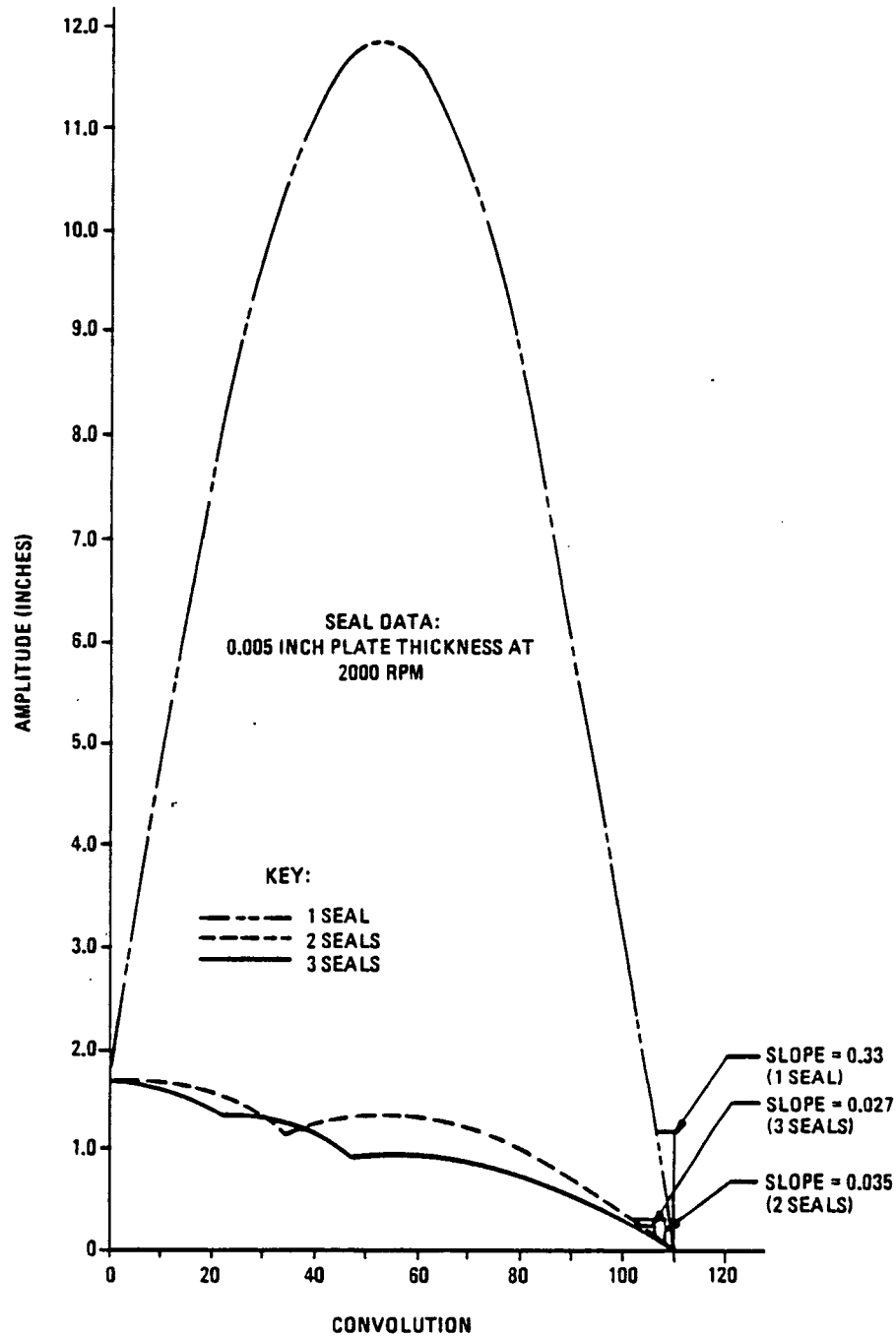


Figure 11. Seal Convolution Amplitudes for Single Seal, Two Seals, and Three Seals in a Series

The previous work with the Braun seal was then applied to determine its potential benefits. The results are significant. The schematic seal arrangements evaluated are shown in Figure 12. In a multiple arrangement, the bellows seal segments, each with a given number of convolutions, are series-connected end to end. One end of the combination is fixed, the other end driven by the primary drive shaft, and the intermediate connection points controlled by the mechanism. The total number of convolutions is 110, as in the single-seal arrangements.

Parametric analyses were made relative to the number of convolutions per seal segment and drive arrangement to minimize maximum displacement for the whole seal system. The resultant dynamic deflection amplitudes are shown in Figure 11 for the two- and three-segment seal arrangements in comparison with the single seal. The resultant maximum displacement amplitudes* are 0.07 inch for the dual-seal, and 0.054 inch for the triple-seal arrangements. The maximum displacement, as discussed earlier, is twice the slope, since the calculated amplitudes represent the TDC position relative to the position at stroke midpoint. For this investigation, the speed was increased to 2000 RPM to more closely match the anticipated acceleration environment of the BR-105 engine.

Stress levels corresponding to the above deflections were determined for the multiple-seal arrangements, including the effect of pressure differential across the seal. No stress calculations could be made for the single seal because the maximum displacement per convolution was beyond the range of the manufacturer's charts. The Goodman stress diagrams are given in Figures 13, 14 and 15. In all cases, the three-segment seal would provide infinite life, and the two-segment seal would provide limited life.

*of the convolution stretched the most

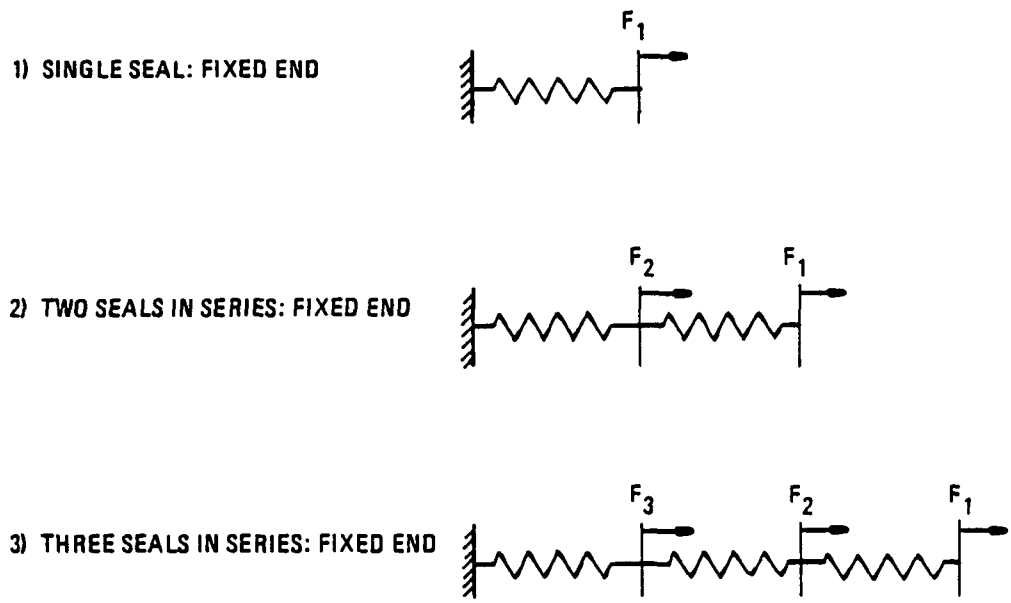


Figure 12. Schematic of Seals in Series With Driving Functions Corresponding to Displacement

These analytical results with the multiple-seal arrangement have been supported by demonstrations on the test rigs.

Conclusions

The work performed on this project has produced two significant results:

- Analytical techniques have been improved and modified to allow precise evaluation of the dynamic performance of complex linear metal bellows seals.

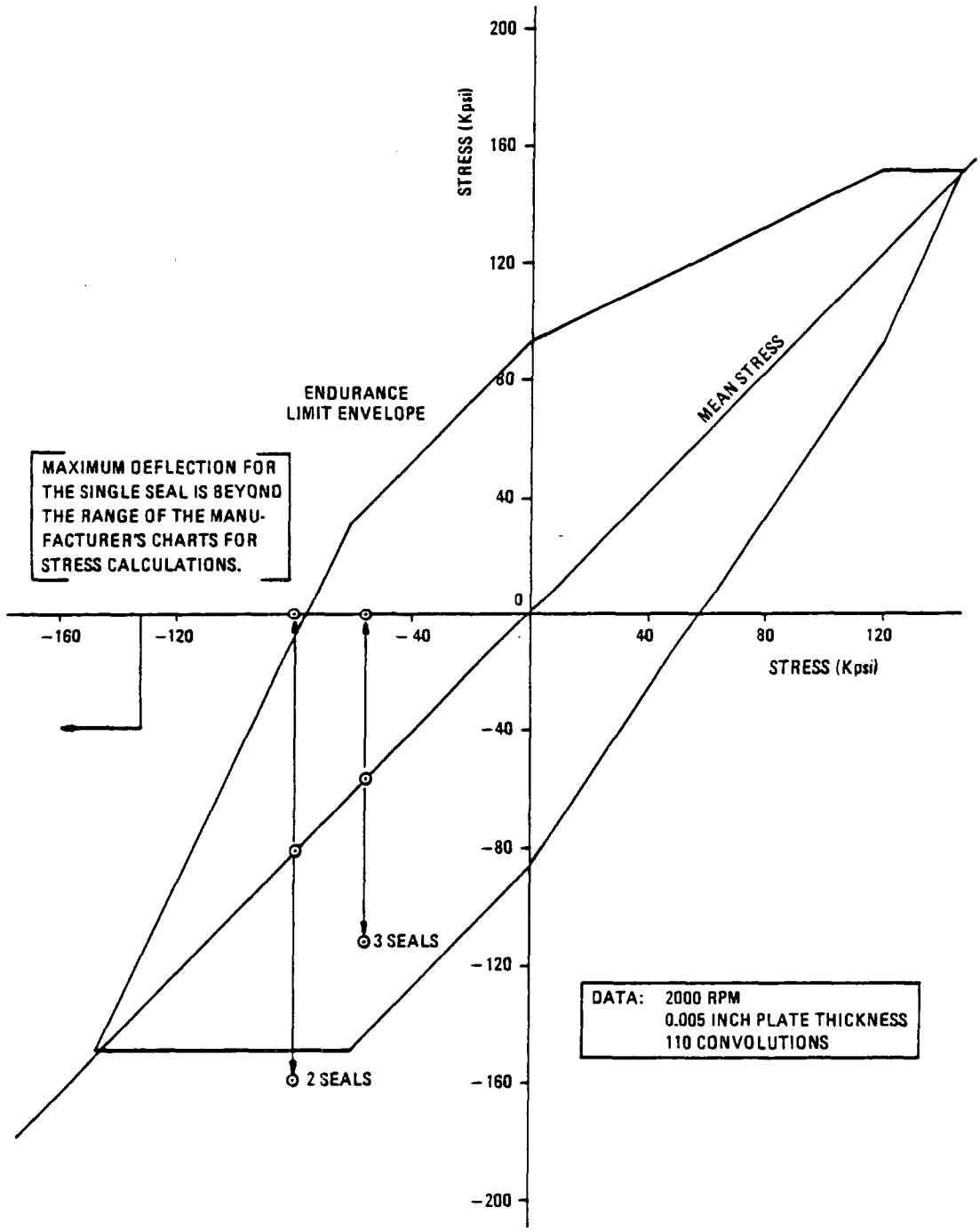


Figure 13. Maximum Stress Levels in Single-Seal, Two-Seal, and Three-Seal Arrangements; Zero psi Pressure Differential Across Seal Wall

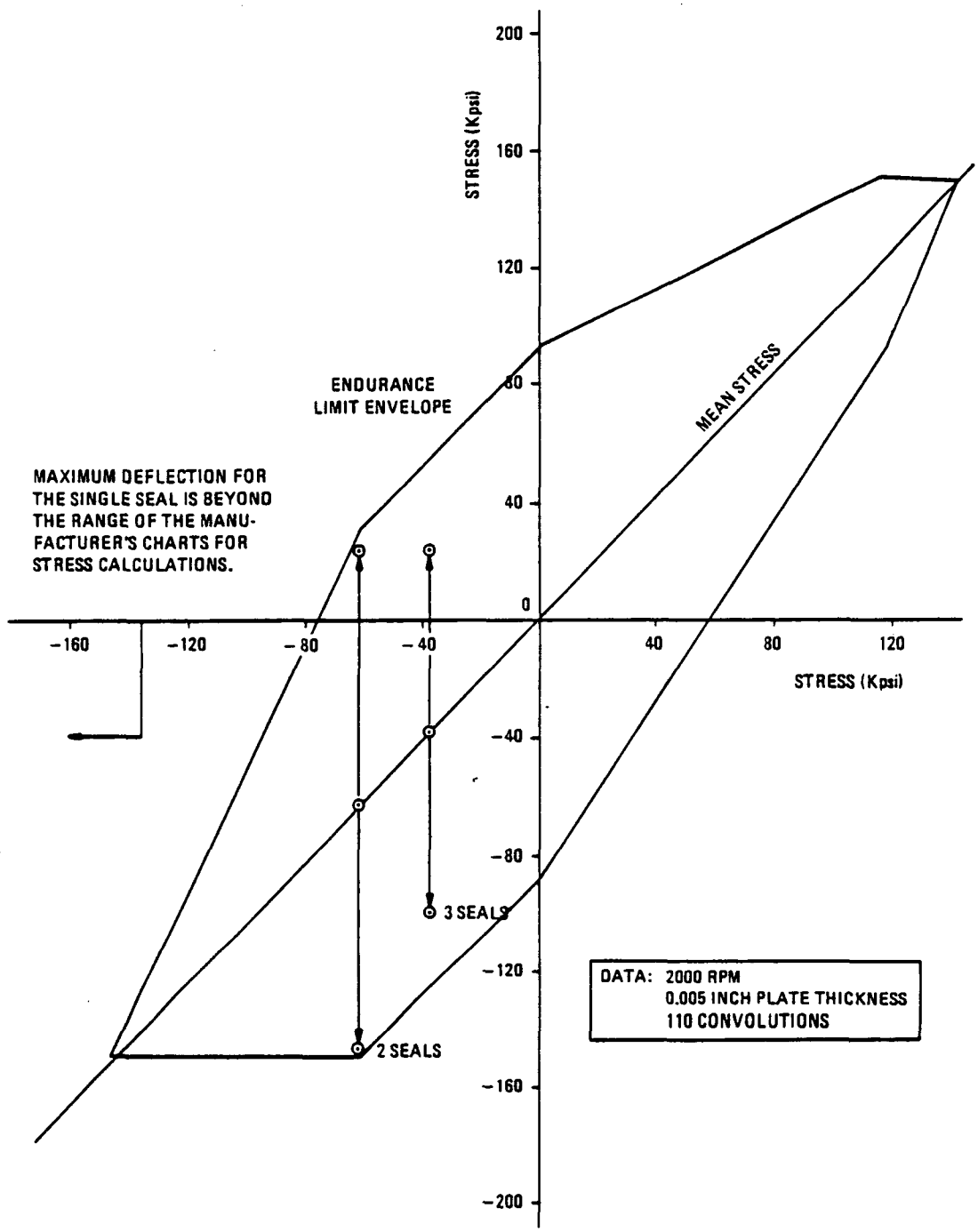


Figure 14. Maximum Stress Levels in Single-Seal, Two-Seal, and Three-Seal Arrangements; 30 psi Pressure Differential Across Seal Wall

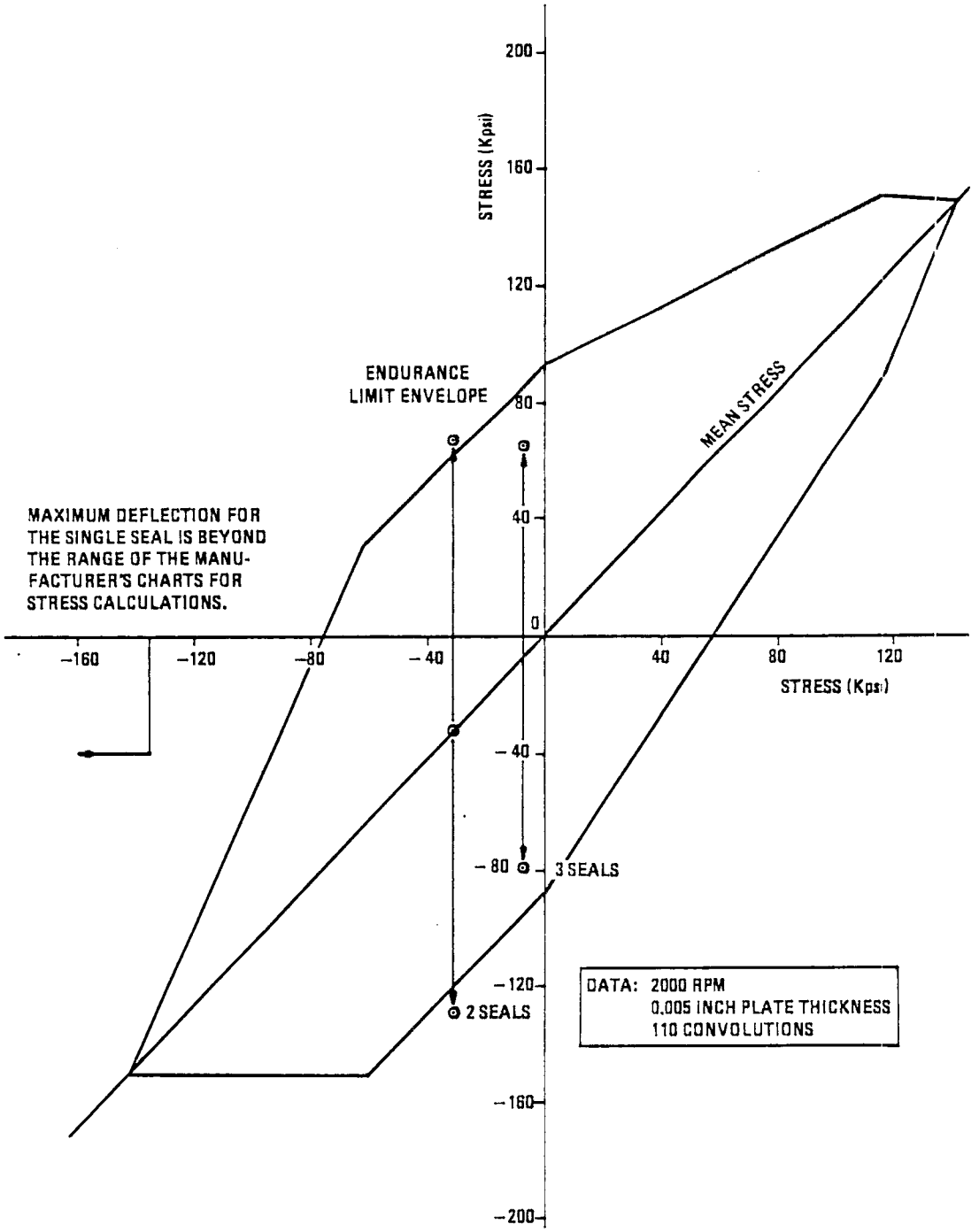


Figure 15. Maximum Stress Levels in Single-Seal, Two-Seal, and Three-Seal Arrangements; 80 psi Pressure Differential Across Seal Wall

- A seal solution has been identified, analyzed, and demonstrated to meet the general requirements set forth for a hermetic refrigerant seal.

The ability to achieve the seal solution is made possible by the application of the proprietary Braun seal solution. It provides limits on the stress levels and "infinite" design life of the seal when used under conditions of high dynamic loading of the bellows.

Again, the work was accomplished within the framework of existing, readily available seal components whose specifications were derived from early preliminary analyses. As the program progressed in results and insights, the relation of BR-105 seal requirements to specific seal characteristics began to emerge. The program concluded with a seal solution optimized for a specific stroke utilizing those available seals.

However, the BR-105 engine has a nominal maximum stroke of 5.000 inches and higher accelerations for a given cycling rate. This stroke is greater than the BR-420 machine on which most of the tests were performed. The seal solution, nevertheless, has been dynamically demonstrated on the BR-105 engine. Confidence therefore exists that an eventual mating of the seal solution concept with the operational characteristics of the Braun-105 engine/compressor will be achieved.

As work progresses on the engine/compressor and seals in the Heat Pump Demonstration program, it is expected that the final seal solution application will involve trade-offs between engine/compressor design and seal design. In turn, this trade-off will focus on two related efforts:

- Continuing coordination with the seal manufacturer to explore alternate seal configurations--still remaining within manufacturer's existing tooling capabilities;

- Continuing analysis and design work on the engine/compressor to evaluate such parameters as stroke and accelerations and their impact on system performance.

Both of these efforts have been initiated. For example, computer performance analyses and design layouts were already made for BR-105 machines with two different strokes. Early results are promising. With only a limited reduction in power, the modified machines could lead to a lower cost, simpler, and safer seal design, as well as a more compact, lower cost machine.

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