# VIBRATION AND DESTABILIZING EFFECTS OF FLOATING RING SEALS IN COMPRESSORS

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## I. INTRODUCTION

Operating experience on a compressor commissioned 12 years ago has presented an interesting history of sporadic increases in shaft vibration. Initial operation was satisfactory with low levels of vibration. However, after some time the shaft vibration level increased to several mils. Initially this was believed to be due to rotor unbalance from deposits formed in the passages due to process upsets. After cleaning up the rotor, operation was again satisfactory. In time the vibration level again increased. It was then found that the rotor vibration was primarily subsynchronous. Further investigation revealed that the original seal design was subject to wear and was no longer properly pressure balanced. A modified seal design was installed and it has operated successfully for the past six years.

Subsequent analysis has provided a better understanding of the seal destabilizing effects on the rotor and motion of the seal which has been confirmed by test data on the current seal design. These will be briefly presented.

## NOMENCLATURE

| | |
|---|---|
| MW | molecular weight |
| P | pressure |
| Kyx Kxy | cross coupled stiffness damping properties for seals |
| Cxx Cyy | principle damping properties for seals |
| $\omega$ | angular velocity (rad/sec) |
| D | diameter |
| $\mu$ | oil viscosity |
| L | seal or bearing length |
| c | clearance |
| $F_p$ | force, radial |
| $F_A$ | force, axial |

$\mu$ coefficient of friction

Q aerodynamic or seal destabilizing

$\delta$ logarithmic decrement

## II COMPRESSOR DESIGN

The subject machine is a 5-stage, vertically-split, centrifugal compressor with floating ring oil film seals in refinery service. (See Figure 1) The gas is a diesel distillate (MW = 8.2) and the process conditions are as follows:

| | | |
|---|---|---|
| Inlet Pressure | $P_1$ | = 2896. K Pa (420 PSIG) |
| Discharge Pressure | $P_2$ | = 4178 K Pa (606 PSIG) |
| Flow | | = 322 Kg/min (739 LB/MIN) |
| Driver Size | | = 1118.6 KW (1500 HP) |

The compressor has 38.1 cm (15") diameter impellers and operates at 12,320 RPM in 8.89 cm (3.5") diameter tilting pad journal bearings. The seal diameter is 11.43 cm (4.5 in.)

## III OPERATING HISTORY

Commissioned in 1970, the machine had a history of minor, occasional vibration problems. A pattern developed which was noted by the Allis-Chalmers Field Service and Repair Group:

- The machine generally operated smoothly following service or maintenance.

- Increased shaft vibration would develop over time (6 months). Seal oil flows would sometimes increase substantially, resulting in operation of the auxiliary seal pump to keep up with the increased flow.

- Upon disassembly, the unit would be fouled with ammonium chloride deposits in the aerodynamic passages. (See Figure 2) The presence of the deposits and the resulting unbalance was initially believed to be the cause of the vibration.

  The floating bushing would be quite worn on the axial face resulting in high axial forces on the seal housing. (See Figure 3 for arrangement.)

Finally, in 1975 the operator reported that vibration levels had become unacceptable and noted that the machine behaved differently with each of the two rotors (main and spare). One rotor performed smoothly with enlarged radial seal clearances (8 mils vs. 2-3 mils design), even though it was fouled and a balance check indicated it was out of balance. The other rotor ran rough with design seal clearances, despite a touch-up balance. The apparent contradiction between vibration experience and the machine balance condition strongly suggested that the vibration problem was non-synchronous in nature. The Allis-Chalmers service group discussed the problem with the Compressor Engineering group and a study was initiated.

Field vibration spectra were obtained by Allis-Chalmers on the balanced rotor at several locations using displacement probes and accelerometers. See Figure 4a.

The data showed:

1. A subsynchronous vibration signal was present at all locations checked. (See Figure 4b & c)

2. The frequency which would change with a slight variation in bearing and seal oil temperature, varied slightly from 80.6 to 81.6 hz, (4836 to 4906 CPM).

3. Accelerometer data (integrated to yield displacements) showed casing motions which were significantly lower than shaft amplitudes indicating that the probes were measuring actual shaft motion, and not a foundation resonance.

Vibration amplitudes at various locations are shown in Table 1.

TABLE I. - VIBRATION AMPLITUDES AT 4850 CPM (FILTERED)

| AMPLITUDE | $A_1$ | $IB_1$ | $A_2$ | $OB_1$ | $OB_2$ |
|---|---|---|---|---|---|
| mm | .0127 | .051 | .009 | .009 | .074 |
| (mils) | (.5) | (2.0) | (.35) | (3.5) | (2.9) |

The synchronous component of shaft vibration was less than .0127 mm (.5 mils). This coincided with the customers comments about the performance of the two rotors. An increase in the synchronous vibration of the out-of-balance rotor could still result in lower overall vibration levels if the sub-synchronous component present in the well balanced rotor were eliminated.

## IV ANALYSIS OF PROBLEM

At the time the machine was designed, analysis was limited to undamped critical speeds, so an updated rotor dynamics analysis was performed using improved rotor dynamics programs in use at Allis-Chalmers in 1975 which included:

- Undamped critical speed map showing intersection of undamped critical speeds with bearing stiffness curves.
- Mode shapes.
- Elliptical orbit synchronous response analysis.
- Stability analysis.

The critical speed analysis indicated that the machine was operating between the 2nd and 3rd modes, see Figure 5a. The mode shapes show substantial motion at the bearings for the second mode indicating it should be well damped (see Figure 5b). This is confirmed by the response analysis (Figure 5c), note that the 2nd mode is well damped and that the amplitudes produced by an unbalance distribution based on the API residual unbalance limit are quite low (less than .002 mm, [.8 mils]). The response analysis showed a 1st resonance at 4100 RPM, and the 1st critical speed on test was 4086 RPM.

Baseline (no destabilizing) stability analysis showed acceptable stability with a log decrement of .169 at 3845 CPM for the 1st Y-mode in forward precession. (See Figure 5d.)

Since the machine was stable under baseline conditions it was then desired to evaluate the rotor's sensitivity to destabilizing forces. To approximate their destabilizing effect the stiffness and damping properties of the seal ring were estimated by assuming that the seal stops tracking the shaft ("locks up") and behaves as a non-cavitated concentric plain sleeve bearing.

Under these assumptions the properties are given approximately by:

$$Kxy = Kyx = \frac{\pi}{4} \omega D \mu \left(\frac{L}{c}\right)^3$$

$$\omega C_{xx} = \omega C_{yy} = 2\ Kxy$$

From this calculation the properties developed for design conditions are:

- Kxy = Kyx = 1077 N/CM ($6.148 \times 10^5$ lb/in.)
  Cxx = Cyy = 1681 N-S/CM (960 lb-sec/in.)

The principle stiffness terms Kxx and Kyy for a concentric seal are negligible. Note that these properties are highly sensitive to variations in clearance (inversely proportional to $C^3$), and that quadrupling the clearance reduces the properties by a factor of 64. Thus the enlarged clearances at the seals found on disassembly could have allowed operation even if the seals were locked up. This explains why the unbalanced rotor with enlarged clearances operated with less vibration than the balanced rotor with design clearances which had bounded whirl.

More sophisticated calculation schemes exist to develop the seal complete stiffness and damping matrices for various assumptions about the seal lock-up eccentricity. However, the results are sensitive to the assumptions about whether lock-up results in increased or decreased journal loading. Reduced journal loading will change the natural frequency of the rotor because the effective bearing span changes with transfer of the load to the seals. Due to the uncertainties associated with the assumption of a lock-up eccentricity, the simple concentric seal properties were used in this case.

Various values of seal destabilizing were input into the stability analysis to evaluate the system sensitivity. See Figure 5e. The stability analysis shows zero log decrement with aerodynamic destabilizing of 2329 N/CM (1330 lb/in) distributed among the impellers and 38,530 N/CM (22,000 lb/in) at each seal. This is substantially less destabilizing than would be produced by the locked up seals with original design clearance. Thus the seals can produce sufficient destabilizing to drive the rotor into bounded whirl under lock-up conditions.

The frequency of the analytically predicted unstable mode was 3850 CPM as opposed to 4850 CPM in the field. Phase information from the field test data indicated that the shaft ends were in phase, so it was concluded from the limited available information about frequency and mode shape that the 1st 'Y' mode of the rotor was unstable (bounded whirl) and was being driven by the seals. Several other conclusions can be drawn from the operating experience and analysis:

1. Stable operation of this rotor with the design seal clearances is only possible if the seal "tracks" the shaft and doesn't lock-up. This implies that any destabilizing produced under tracking conditions must be substantially less than that present under lock-up, in fact less than 38,530 N/CM (22,000 lb/in).

2. Stable operation is possible if the seal locks up, if the seal clearances are abnormally large (resulting in much smaller hydrodynamic destabilizing forces).

3. The difference in the frequency of the unstable mode between analysis and field data is possibly due to the development of principle stiffness terms (Kxx, Kyy) at the seals due to an eccentric lock-up of the seal ring which transfers bearing load to the seals, thus reducing the effective bearing span of the rotor, and raising its natural frequency from 3850 to 4850

4. The frictional force ($F_R$) which restrains the seal from moving radially (and determines lock-up eccentricity) is a function of the pressure induced axial forces ($F_A$) on the seal ring and the coefficient of friction ($\mu$) between the seal and its housing.

$$F_R = \mu F_A$$

Review of the axial forces ($F_A$) on the seal at the design pressure with no seal wear show a relatively small value. See Figure 6A.

$$F_A = P\left(\frac{\pi}{4}\right) \quad (D_2^{\,2} - D_1^{\,2}) = 792 \text{ N } (178 \text{ lbs.})$$

However, as wear occurred on the axial face of the seal, the outer diameter of the contact face ($D_2$)increased. For example, if axial wear on the ring was .0254mm (.001") one fourth of the chamfer would be removed, and $D_2$ would increase from 11.53 cm (4.54 in.) to 13 cm (5.118 in.).

At this point the axial force would be 8985 N (2020 lbs.) or 11.3 times the original seal design value. The radial force could increase by more than this if the coefficient of friction increased with wear.

Thus the original design was highly sensitive to both the friction coefficient and wear so that following some initial wear, the wear rate would accelerate until lock-up occurred.

4. Since the seal parts had shown substantial wear during earlier maintenance and service inspections, the compressor performed well after maintenance, and the stability analysis showed good correlation with experience, seal modification was selected as the best method of resolving the problem.

V <u>SOLUTION</u>:

Such a seal-induced instability can be solved by two types of seal modifications:

1. Allow the seal to lock up, but <u>reduce the hydrodynamic forces</u> produced by changing the geometry of the seal in the following ways:

   a. <u>Reduce seal effective length</u> - reducing the effective length by adding grooves to the seal surface reduces the stiffness and damping produced but this is at the expense of reduced film thickness and therefore seal centering capability which can increase the possibility of seal rubs.

   b. <u>Increase seal clearances</u> - This reduces the stiffness and damping properties but increases oil flows dramatically which is undesirable.

   Increased clearance due to wear allowed operation in this case despite a locked-up bushing. The auxiliary seal pump ran all the time. In addition the customer added a third pump to keep up.

2. <u>Balance the axial forces</u> on the seals as closely as possible so that the seal doesn't lock up to begin with. Reduction in axial forces makes the seal less sensitive to the coefficient of friction between the seal and its housing (which can increase with wear) and reduces the seals tendency to wear.

The second type of modification was used to solve this problem. Figure 6a and 6b show the original and modified seal bushings.

Note that the following modifications were made.

1. The residual axial force on the bushing was reduced by balancing the pressure induced axial force.

a. O-ring was removed from bushing end reducing friction.

b. Face relief was remachined to better control pressure equalization.

c. Pressure balancing axial hole added.

d. The taper was removed to make the new design insensitive to wear.

2. These changes don't affect the hydrodynamic performance (i.e., leakage or film thickness) of the seal, only the force required to move it radially (FR).

Since the new seal has been installed, seal wear has been negligible and the sub-synchronous vibration problems have been eliminated. The modified seal has performed successfully for over 6 years.

Allis-Chalmers has over 20 years experience in the design and application of oil film seals. The current Allis-Chalmers standard "Trapped Bushing Seal" features:

- A "dual" bushing which encompasses both the inner and outer seal in one ring for reduced axial length. (See Figure 7)

- Low residual axial force on the seal which effectively reduces the potential for lock-up and seal sensitivity to friction and wear. (See Figure 8)

## VI SEAL VIBRATION TEST PROGRAM

A test program was subsequently conducted in the Allis-Chalmers test facility to verify the motion of the A-C standard dual bushing at design pressure level to insure that the bushing tracks the shaft without lock-up. All seal vibration data were provided by Mr. P. G. Shay, the Supervisor of the Allis-Chalmers Compressor Test Facility. Two displacement probes were mounted 90° apart in the cage surrounding the bushing to determine the amplitude of bushing motion and the relative phase lag between the shaft motion and bushing motion. (See Figure 9, note the epoxy-filled relief around the probe tips.) The oscilloscope traces show the seal orbit to be circular with a phase lag of 45° behind shaft motion and amplitudes slightly less than the shaft amplitude (See Figure 10.) The vibration spectra show the seal motion to be predominently synchronous with only small traces of non-synchronous motion. Shaft vibration is entirely synchronous. (See Figures 11A-D.)

Based on this information, it may may concluded that this arrangement results in minimal destabilizing effects as the bushing is able to freely track the shaft motion. Seal induced hydrodynamic forces are dissipated in seal motion and not applied to the shaft. The seal is also insensitive to wear on the axial faces.

## VII CONCLUSIONS:

1. Residual axial forces in seals can influence seal and shaft vibration. Some small level should be present. However, seal lock-up should be avoided.

2. Restraint ("lock up") of bushing results in high levels of destabilizing forces and it is therefore better to err on the low side with respect to axial (pressure induced) forces in the event that seal wear increases the friction coefficient significantly.

3. Normally tracking seals exert only minor destabilizing effects.

4. The seal design should be relatively insensitive to wear on its axial face to prevent accelerating wear rates.

## REFERENCES:


1. Rouch, K. E., Kao, J. S., "Reduction in Rotory Dynamics by the Finite Element Method". ASME Paper 79-DET-70.

2. Alford, J. S., "Protecting Turbomachinery From Self-Excited Rotor Whirl", Journal of Engineering and Power, Trans. ASME Series A, Vol. 87, October 1965, pp. 333-339.

3. Lund, J. W., "Stability and Damped Critical Speeds of a Flexible Rotor in Fluid Film Bearings", ASME Paper 73-DET-103 - 1973.

4. Kirk, R. G., Miller, W. H. "The Influence of High Pressure Oil Seals on Turbo-Rotor Stability ASLE Preprint 77-LL-3A-1.


## ACKNOWLEDGEMENTS:


1. P. G. Shay, Supervisor, Compressor Test, Compressor Engrg.,C&CPD

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3. J. H. Hudson, Supervisor, Mechanical Design Engineering, Compressor Engineering, C&CPD.

4. Keith Rouch, Advanced Technology Center, Allis-Chalmers.

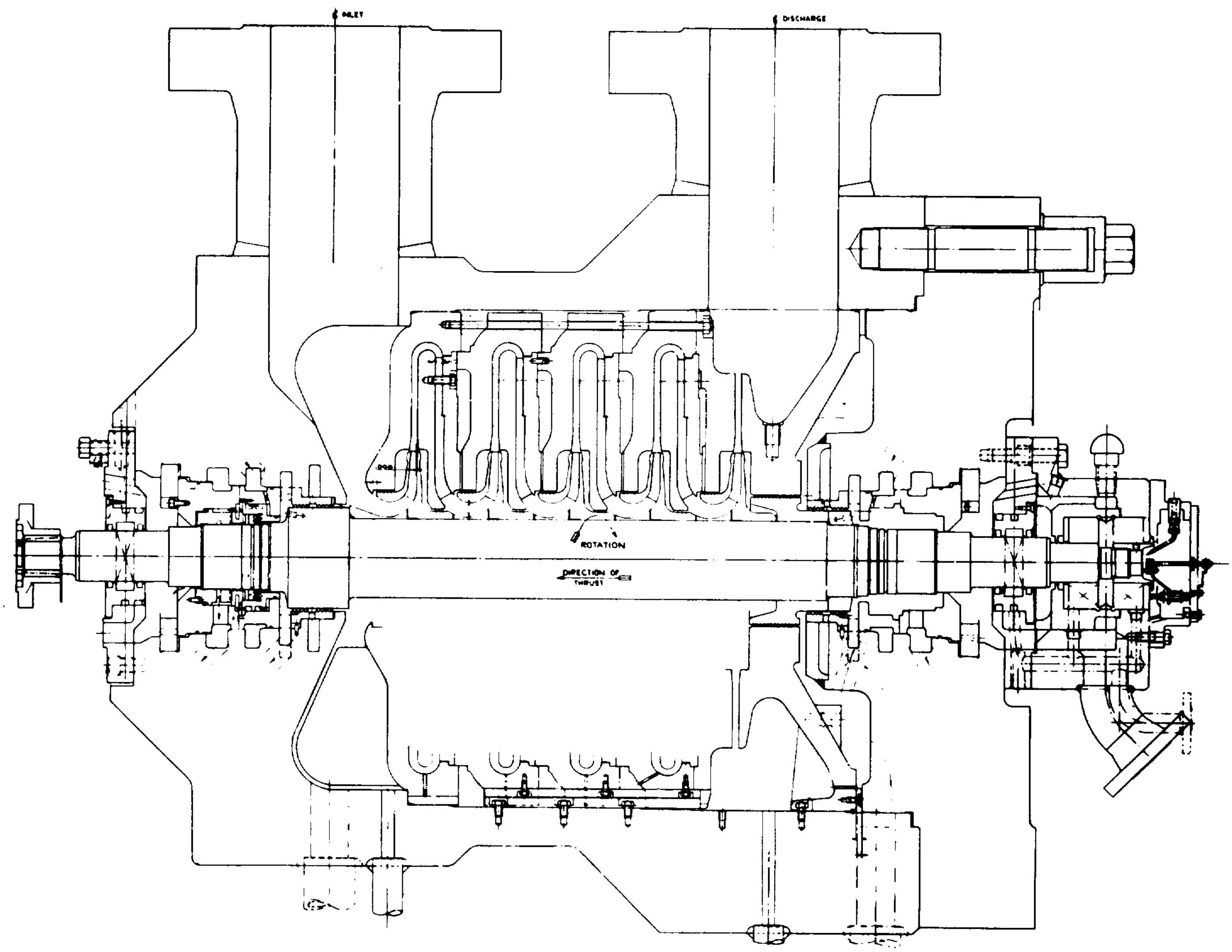



Figure 1. - 5 stage barrel compressor.

Figure 2. - Fouling deposits in compressor.

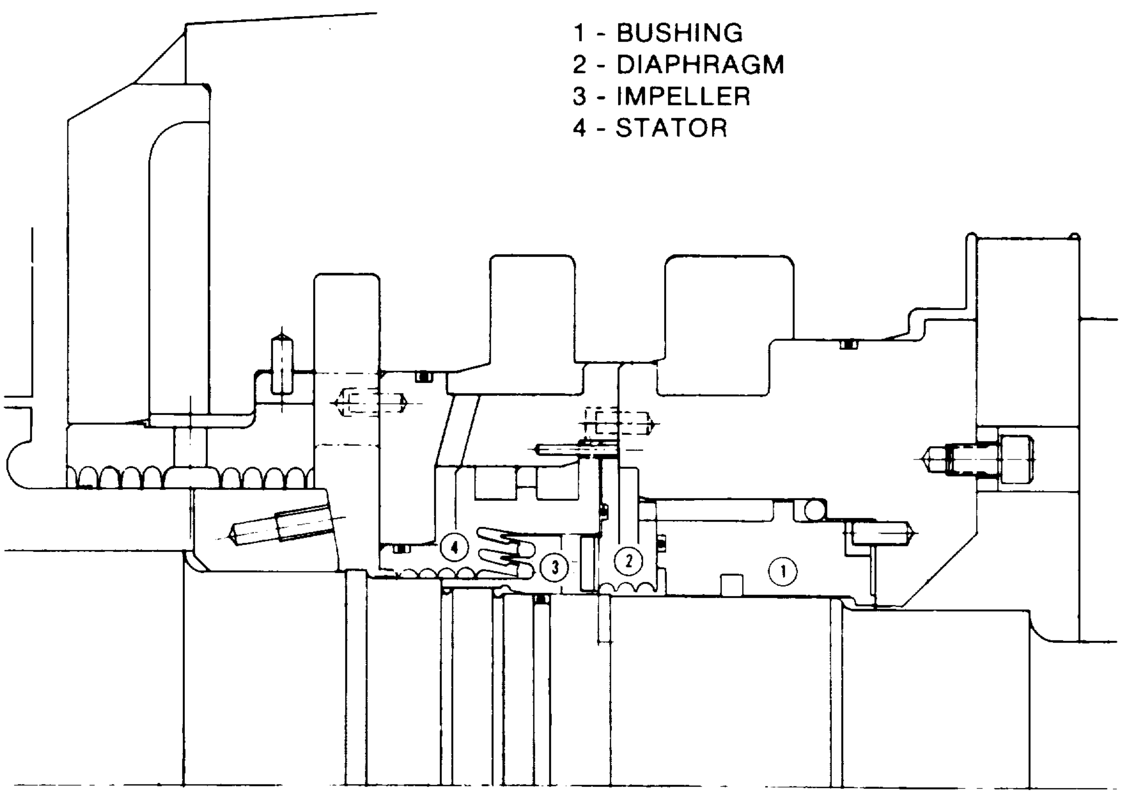



Figure 3. - Seal arrangement.

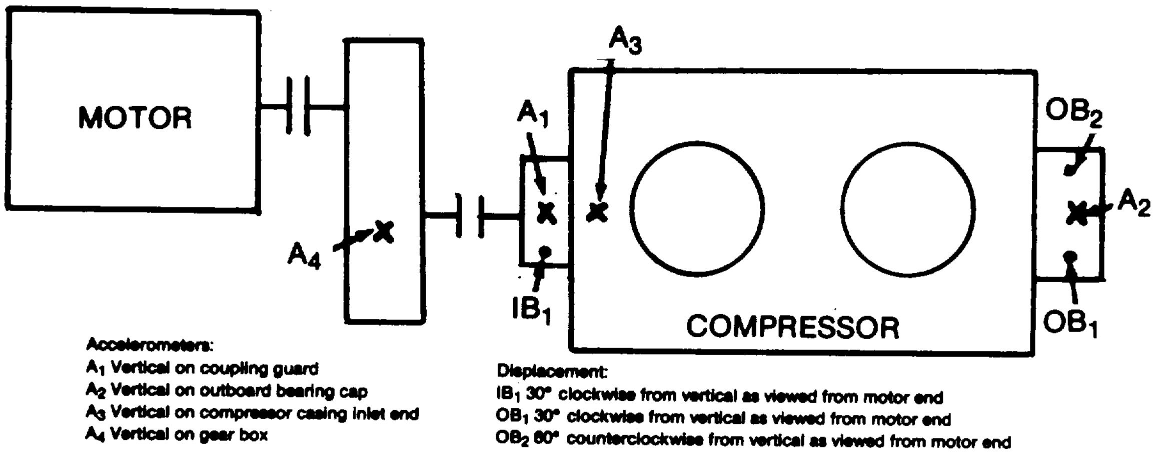



Figure 4(a).

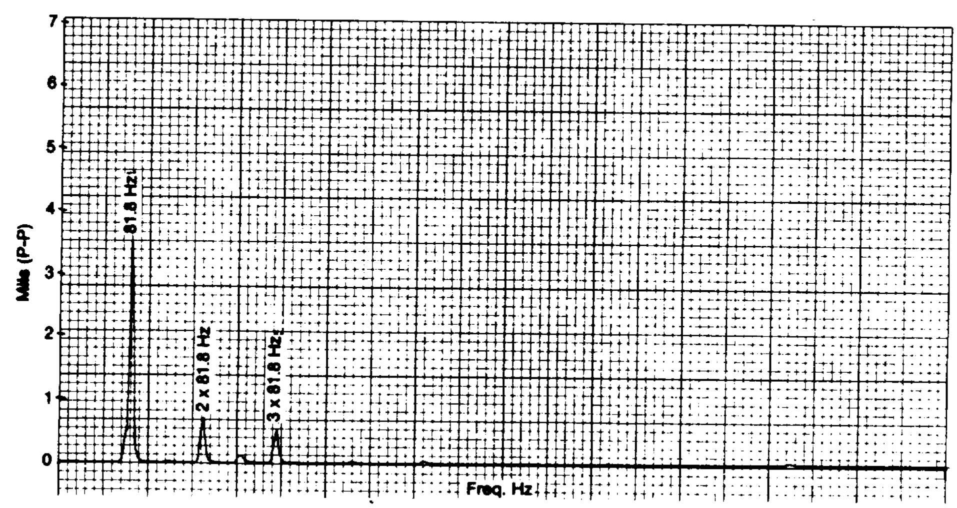



Figure 4(b). - Vibration spectrum inboard probe I.

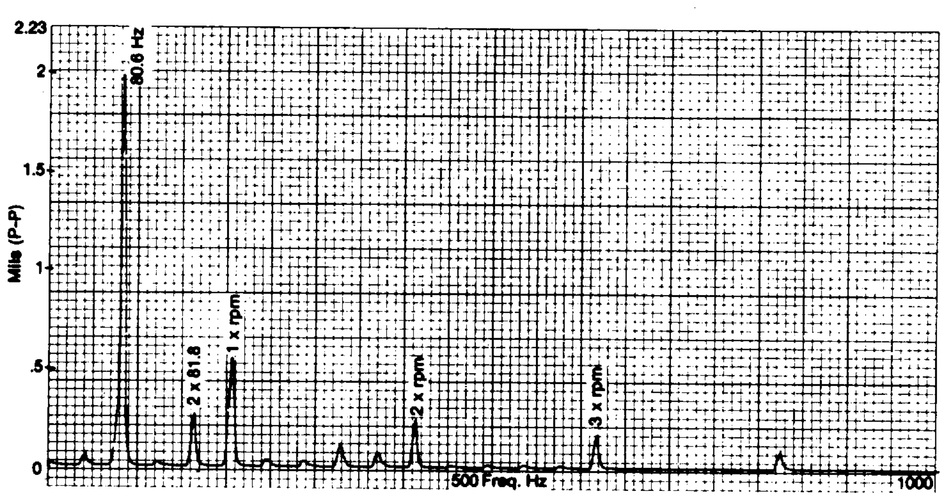



Figure 4(c). - Vibration spectrum outboard probe I.

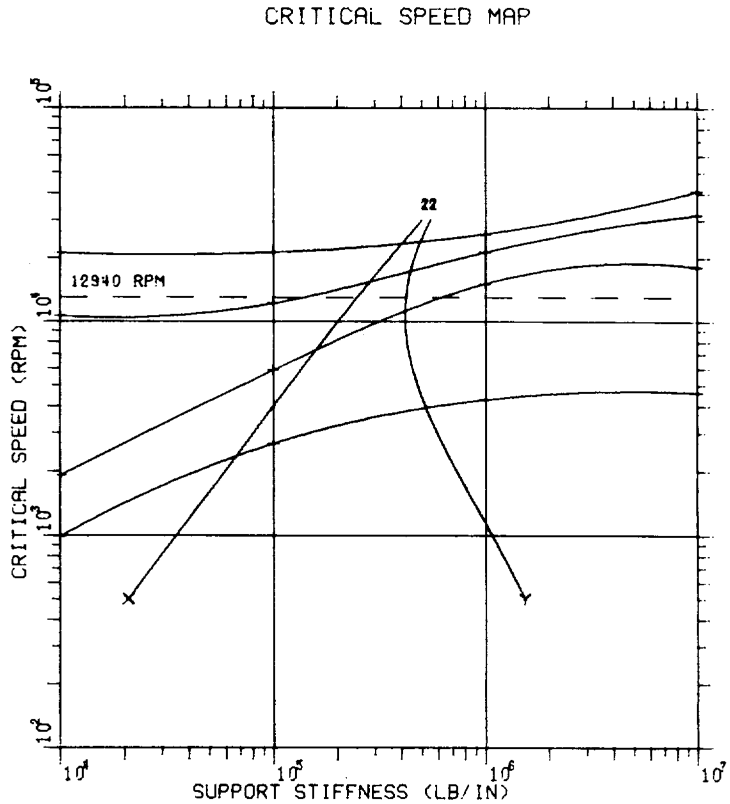



Figure 5(a). - Critical speed map.

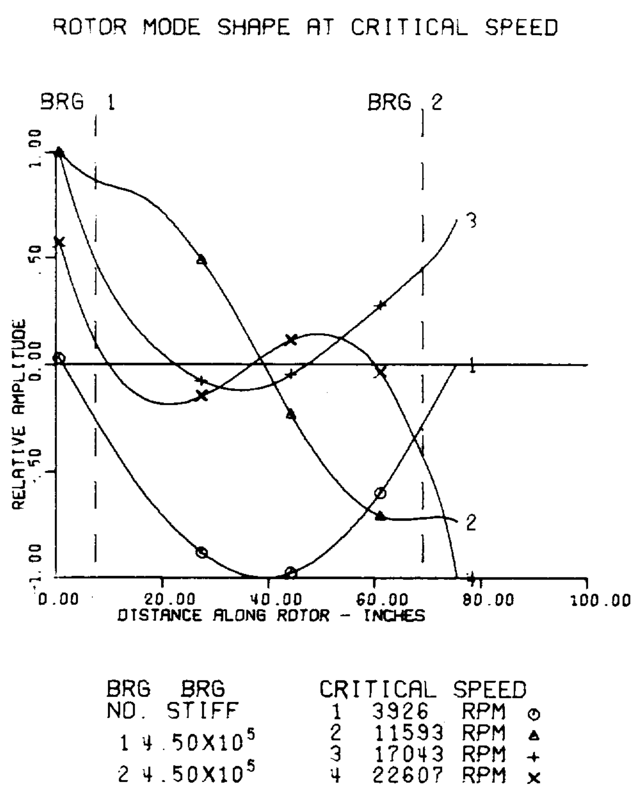
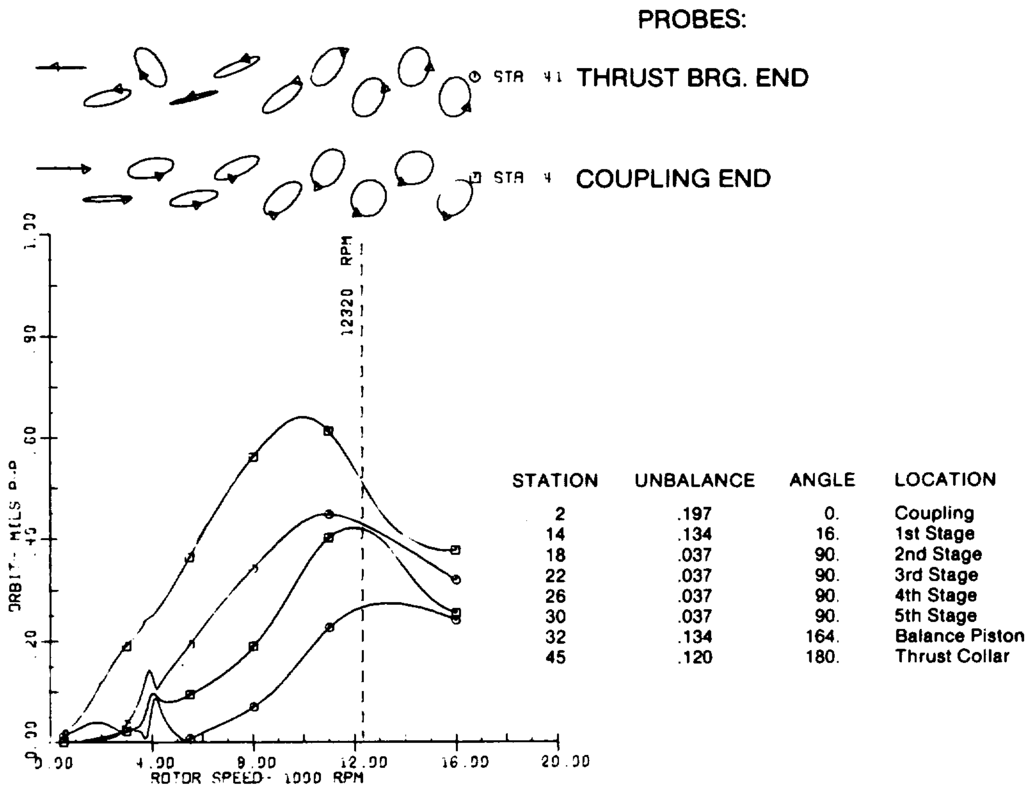



Figure 5(b). - Mode shapes.




| STATION | UNBALANCE | ANGLE | LOCATION |
|---|---|---|---|
| 2 | .197 | 0. | Coupling |
| 14 | .134 | 16. | 1st Stage |
| 18 | .037 | 90. | 2nd Stage |
| 22 | .037 | 90. | 3rd Stage |
| 26 | .037 | 90. | 4th Stage |
| 30 | .037 | 90. | 5th Stage |
| 32 | .134 | 164. | Balance Piston |
| 45 | .120 | 180. | Thrust Collar |

Figure 5(c). - Synchronous response.

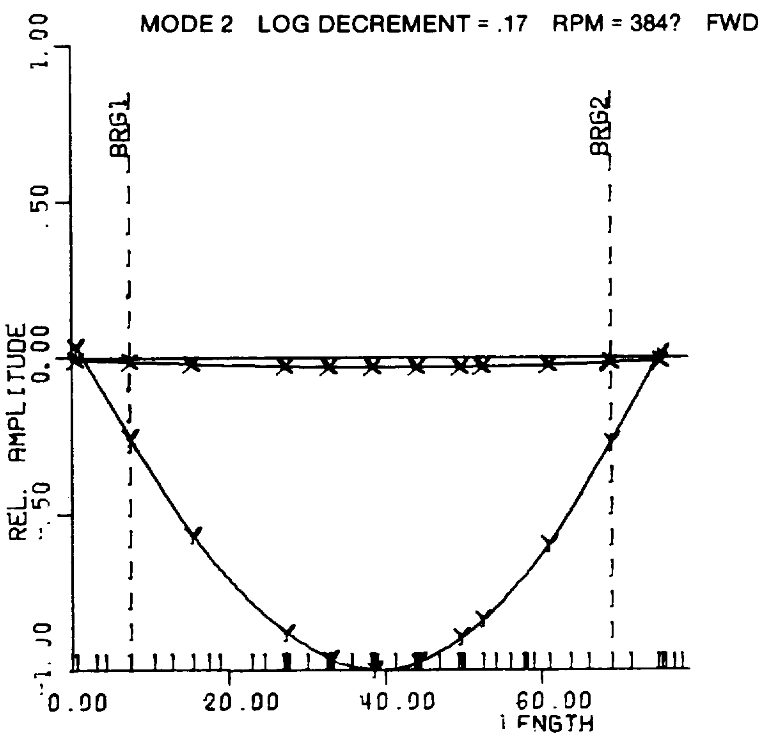



Figure 5(d). - Shaft whirl mode.

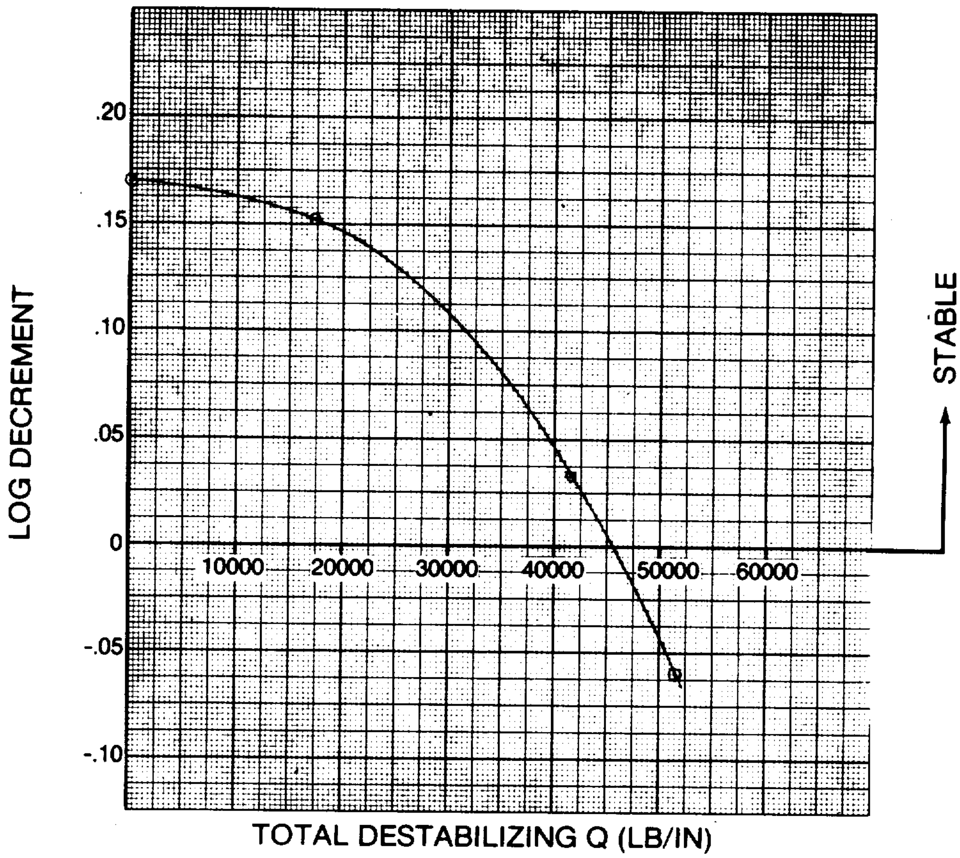



Figure 5(e). - Log decrément vs. destabilizing.

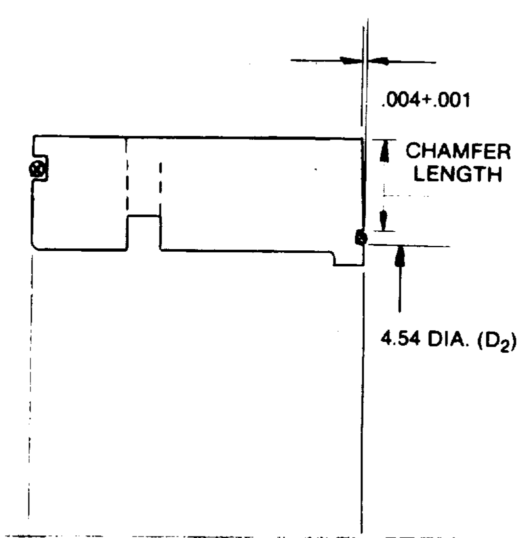



Figure 6(a). - Original bushing.

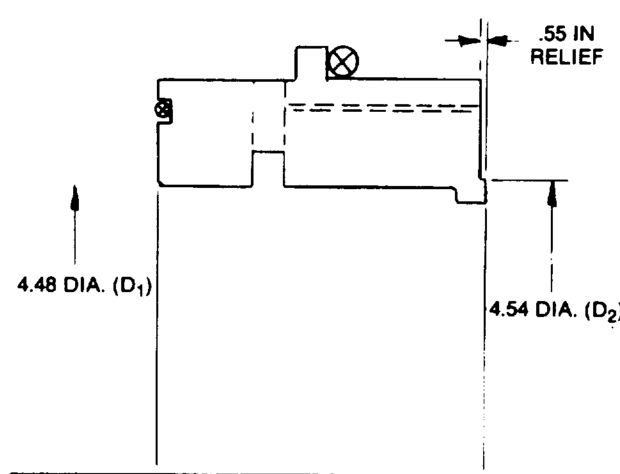



Figure 6(b). - Modified bushing.

**Compact Design** — allows shorter bearing spans for higher critical speeds of the compressor rotor.
**Sleeve** (impeller) with interference fit under bushing — protects shaft and simplifies assembly and disassembly. Requires only a jack/puller bolt ring.
**Spacer** fit at initial assembly — no field fitting of parts.

| ITEM | DESCRIPTION |
|---|---|
| 1 | Shaft |
| 2 | Impeller |
| 3 | Stator |
| 4 | Stepped Dual Bushing |
| 5 | Bushing Cage |
| 6 | Nut |
| 7 | Shear Ring |
| 8 | Oil/Gas Baffle |
| 9 | Spacer Ring |

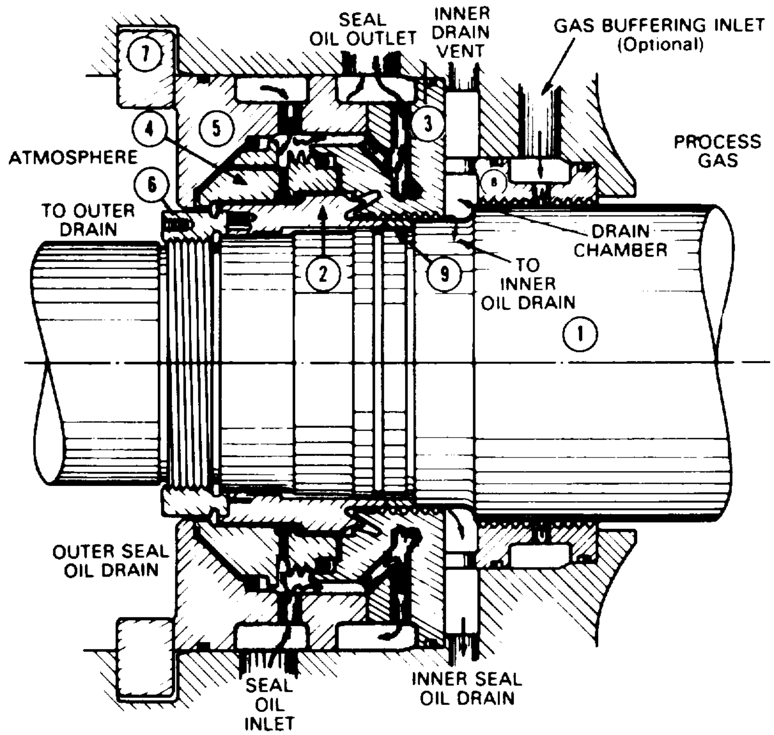



Figure 7. - Trapped bushing seal arrangement.

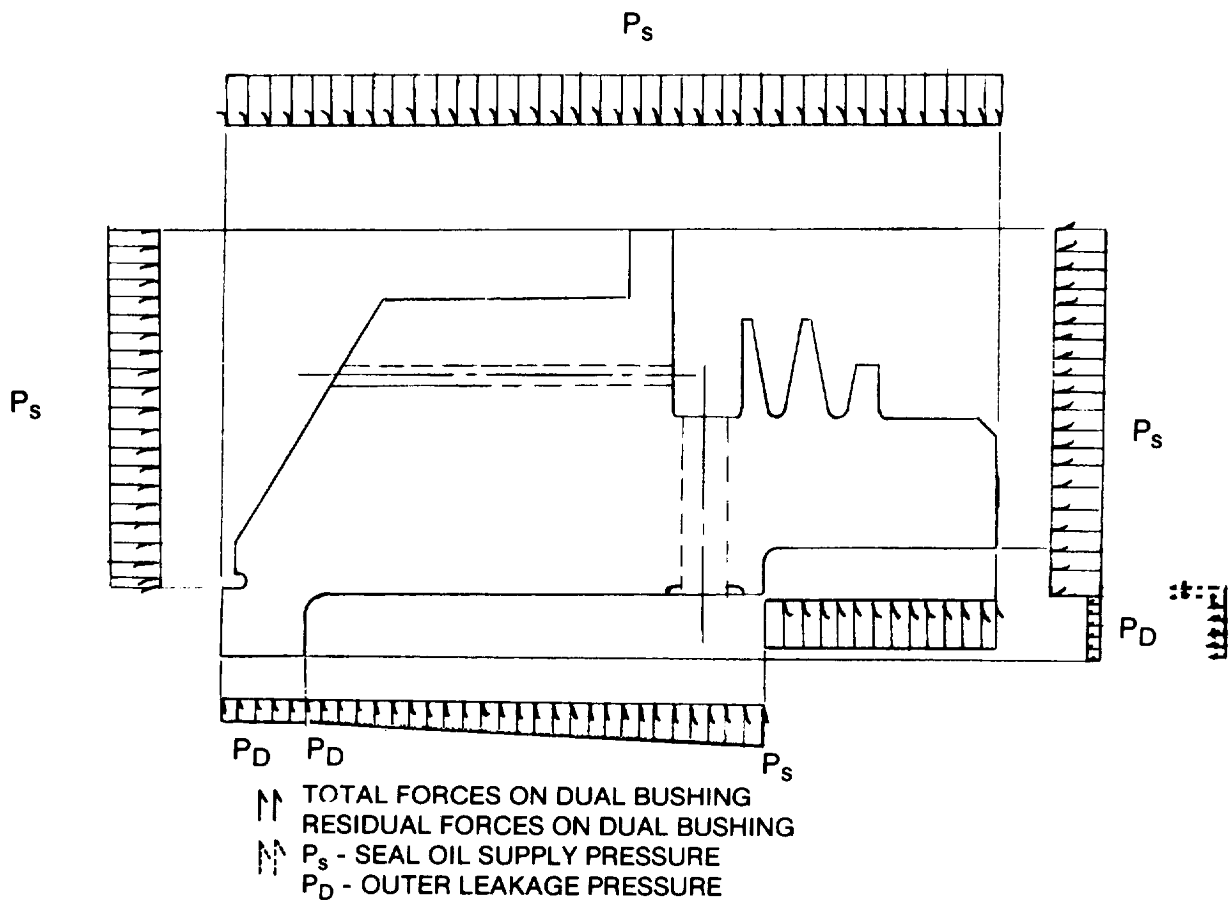



Figure 8. - Pressure induced forces on dual bushing.

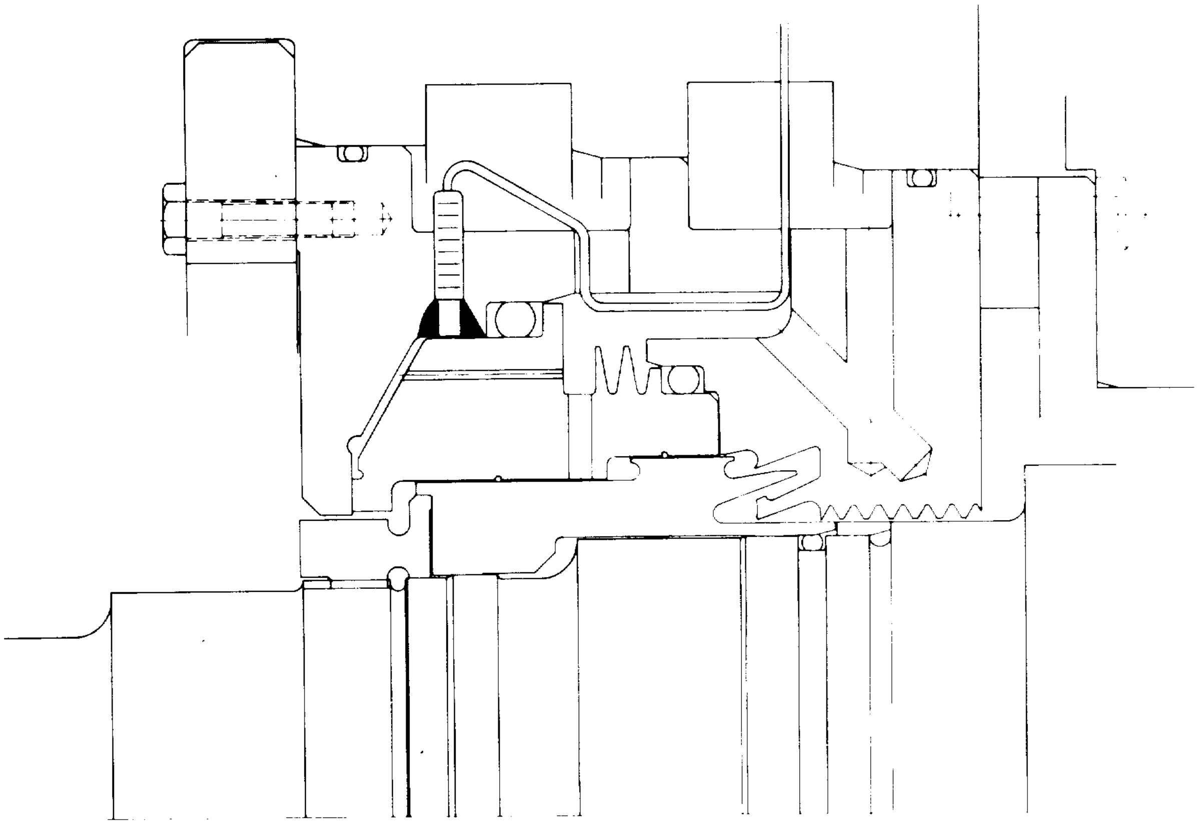
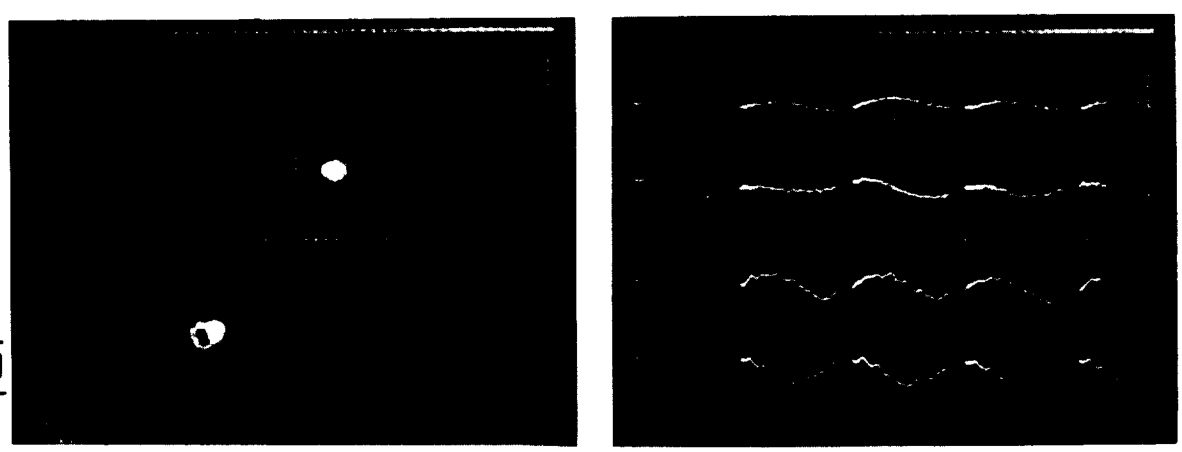


Figure 9. - Probe arrangement for trapped bushing seal vibration test.

1 Mil



| | Upper: | Bushing | Orbit |
|---|---|---|---|
| 1 Mil | Lower: | Shaft | Orbit |

| | | |
|---|---|---|
| Upper: | Bushing | X |
| | Vibration | Y |
| Lower: | Shaft | X |
| | Vibration | Y |

13050 rpm 1 MIL Per Division

Figure 10. - Shaft and dual bushing vibration data.

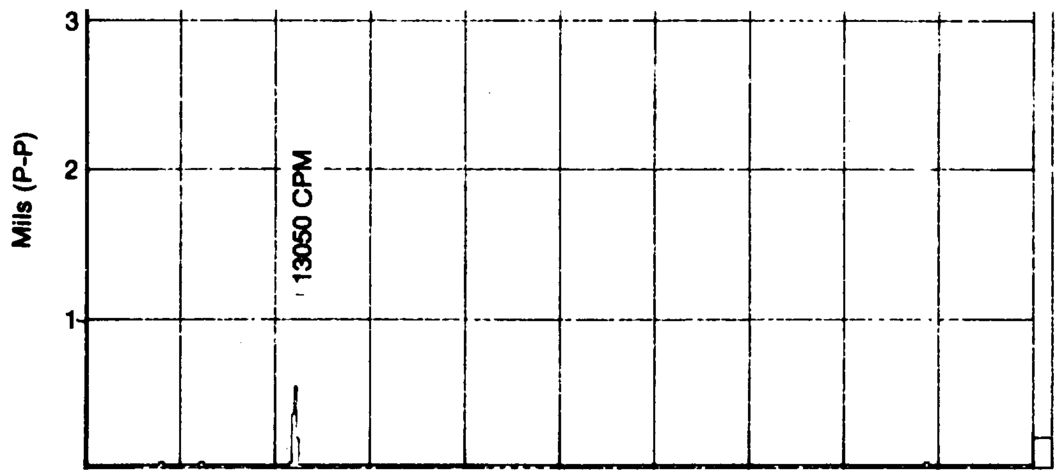



Figure 11(a). - Vibration spectrum - shaft (horizontal).

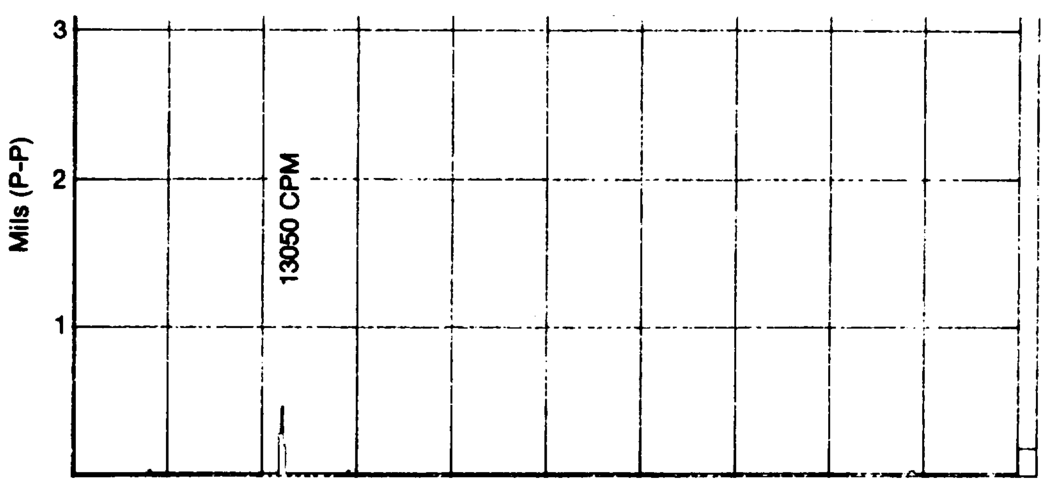



Figure 11(b). - Vibration spectrum - shaft (vertical).

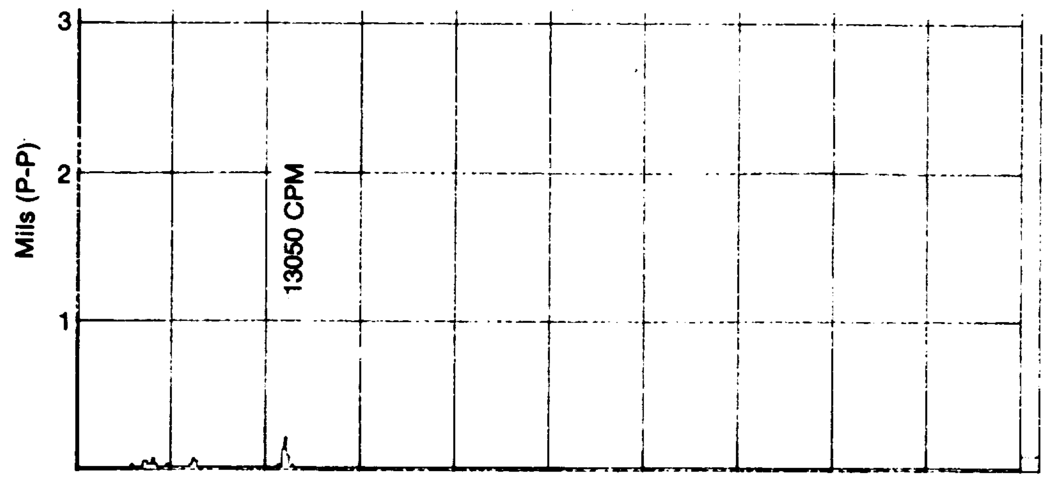



Figure 11(c). - Vibration spectrum - seal (horizontal).

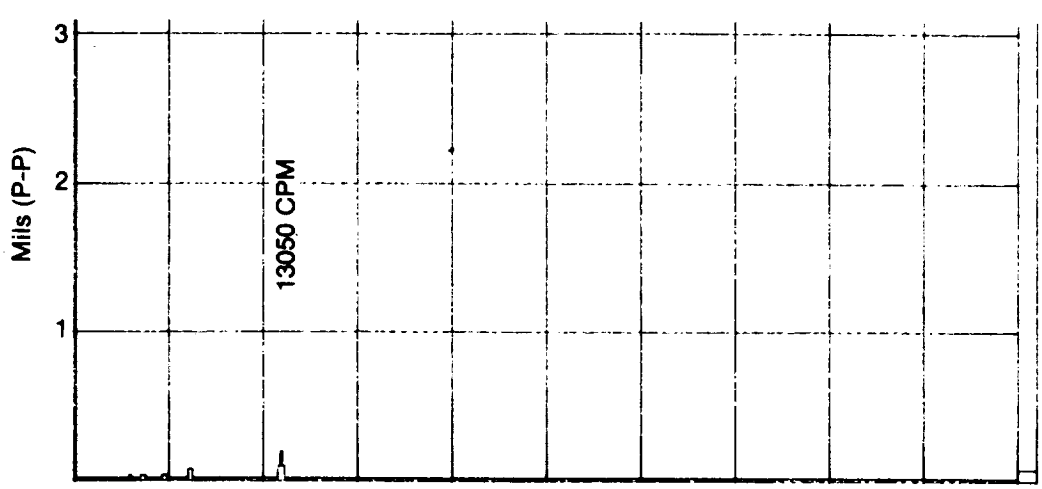



Figure 11(d). - Vibration spectrum - seal (vertical).