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OIL SEAL EFFECTS AND SUBSYNCHRONOUS VIBRATIONS IN HIGH-SPEED COMPRESSORS

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Oil seals are commonly used in high speed multistage compressors. If the oil seal ring becomes locked up against the fixed portion of the seal, high oil film crosscoupled stiffnesses can result. This paper discusses a method of analysis for determining if the oil seals are locked up or not. The method is then applied to an oil seal in a compressor with subsynchronous vibration problems.

NOMENCLATURE

Symbol	Definition
A_r	Unbalanced Seal Ring Area
c_j	Tilting Pad Bearing Clearance
c_r	Oil Seal Ring Clearance
D	Shaft Diameter
e_j	Shaft Eccentricity in Bearing
e_r	Shaft Eccentricity in Oil Seal Ring
F_{rmax}	Maximum Available Radial Friction Force Acting on Oil Seal Ring
L	Seal Length
n	Pressure Law Power
N	Shaft Speed (RPM)
N_d	Shaft Speed at Design Conditions
O_s	Shaft Centerline
O_r	Oil Seal Ring Centerline
P	Pressure Drop Across Oil Seal
P_d	Pressure Drop Across Oil Seal at Design Conditions
Re	Reynolds Number Based on Clearance
W_r	Oil Seal Ring Weight
x_j, y_j	Horizontal and Vertical Positions of the Shaft Centerline (Relative to the Bearing Centerline
x_r, y_r	Horizontal and Vertical Positions of the Oil Seal Ring Centerline (Relative to the Bearing Centerline)
ε_j	Shaft Eccentricity Ratio ($\varepsilon_j = e_j/c_j$)
ε_r	Oil Seal Ring Eccentricity ($\varepsilon_r = e_r/c_r$)
ϕ_j	Shaft Attitude Angle (Relative to the Load Vector)
ϕ_r	Oil Seal Ring Attitude Angle (Relative to the Load Vector)
μ_s	Coefficient of Static Friction On Oil Seal Ring Face
ω	Shaft Angular Velocity (rad/sec)
μ	Viscosity

INTRODUCTION

Oil seals are frequently used in compressors to minimize the leakage of the process gas into the atmosphere. The type of design discussed in this work has a seal ring which permits a limited axial flow under the ring to produce a pressure drop.

If the desired pressure drop across the seal ring is adequate, only one seal ring is employed. The seal ring is part of the non-rotating component of the compressor seal assembly. The clearances involved are small - on the order of bearing clearances - so the flow can be analysed by Reynolds equation in one form or another. It differs from conventional bearing analysis because there is an axial pressure gradient which can have a significant effect on the oil film properties.

It is observed in the field that some compressors with oil seal rings are subject to subsynchronous vibration while others are not. The key feature of the oil seal ring is the stiffness and damping coefficients developed by the fluid film between the rotating shaft and non-rotating seal ring. Often the seal ring is designed to orbit with the shaft as it vibrates (rotation is prevented by some pin or other device which still allows radial ring motion). If the seal ring orbits with the shaft, the oil film force, and thus the dynamic coefficients, between the shaft and ring are necessarily very small. However, the compressor may still be unstable due to effects other than the oil seals such as labyrinth seals between the compressor wheels in a multistage compressor or aerodynamic crosscoupling associated with the flow near the wheels themselves. Alternatively, the seal rings may be prevented from moving (called seal ring lockup) due to one effect or another. This in turn causes large oil film forces to act between the shaft and ring which may produce large stiffness and damping coefficients. If the crosscoupled stiffness coefficient is high enough, the oil seals can cause subsynchronous vibration by themselves or contribute to it with other effects.

Kirk and Miller [1] reported on the effect of oil seals on instability in compressors. They presented a method of analysis for low Reynolds number axial flow seals where the axial flow is neglected. Stiffness and damping coefficients were developed for short cavitated and noncavitated lubricant films. Also a time transient analysis method was shown for evaluating the operating eccentricity of such seals. Kirk and Nicholas [2] extended the analysis to include a heat and flow balance.

Emerick [3] discussed a compressor case history where shaft vibrations were a problem. Over time the rotor vibrations increased until the machine had to be shut down. After extensive measurements and calculations, it was found that large subsynchronous vibrations were caused by wear to one of the seal components which eventually lead to seal ring lockup. Modifications were made to reduce the wear and to pressure balance the seal to permit shaft tracking. Other references on compressor subsynchronous vibration [4,5] discuss possible effects of oil seals. Usually such seals orbit (float) with the shaft so that lockup problems do not occur. In some cases, seals which appeared to cause the problems were changed with no reduction in vibration.

This paper discusses the action of oil seal rings in a compressor subject to subsynchronous vibration problems. While the vibration problems were not severe, it was desired to remove the subsynchronous component of vibration if possible to prevent future problems. The main purpose of the paper is to discuss the operation of the oil seal and its dynamic coefficients. The effects of seal ring lockup, determination of stiffness and damping coefficients, interaction of seals with bearings (tilting pad), and the final influence on the compressor are discussed. Some of the differences between this and previous works include the quasi-steady state method of determining the lockup conditions, the effect of axial pressure gradient, and the seal ring/bearing interactions.

DESCRIPTION OF COMPRESSOR AND OIL SEAL GEOMETRY

The example centrifugal compressor considered in this paper is driven by a steam turbine through a flexible gear type coupling. The compressor has six closed face impellers as shown in Figure 1. The shaft is mounted between bearings in a straight through design with the bearings located at the ends. It also has oil seals inside the bearings and a balance piston at the stepped region of the shaft. Rotor weight is 1620 lbs and the bearing span is 68 inches. The bearing configuration is a 5 pad, tilt pad design with load on pad. The journal diameter is 4.5 inches and the bearing L/D ratio is 0.44. Bearing preload ranges from 0.45 to 0.55 and installed diametral clearance ranges from 0.005 inches to 0.007 inches. The rotor is designed to run at 9800 RPM and has a rigid bearing first critical speed at 3900 RPM.

At speeds above 9000 RPM, subsynchronous whirl was observed at a frequency of 4200 RPM. Figure 2 shows the subsynchronous and synchronous vibration components at a running speed of 9210 RPM. The amplitude of the subsynchronous vibration was erratic and occasionally, without warning, would spike at two to three times its usual level. A waterfall diagram could not be obtained for this compressor because of the startup and shutdown procedures.

The oil seals are located just inboard of the bearings and are each composed of two rings mounted axially on the shaft with close running clearance. The outboard side of each ring forms a face seal with the stationary seal housing and is kept from rotating by an anti-rotation pin, figure 3.a. Oil is introduced between the rings at a pressure slightly above compressor suction pressure. Part of the oil flows through the shaft-to-ring clearance on the process gas side and is carried off by the process gas. The remainder of the oil flows through the shaft-to-ring clearance on the ambient side and is returned to the oil reservoir. The inner ring seals against process gas leakage while the outer ring limits seal oil leakage to the outside.

QUASI-STATIC SEAL RING LOCKUP ANALYSIS

The purpose of this section is to describe the method of calculating seal lockup conditions. As the oil film crosscoupled stiffness coefficients vary strongly with shaft position relative to the oil seal ring, it is important to have a method of estimating the positions during actual lockup and then during running conditions. A quasi-static method, now presented, is used.

Because the Reynolds number is quite small for these lubricant flows, it is assumed here that the inertia forces are small compared to the viscous forces. The transient motions of the seal ring and shaft (up to the threshold of instabilty at least) are relatively quickly damped without inertia forces. Thus the shaft has a small orbit around the steady state operating eccentricity of either the seal ring or tilting pad bearing. Relative sizes may be found in Emerick [3] where the shaft vibration (after removal of the subsynchronous instability) was about 0.00254 mm (0.1 mils) as compared to the seal ring clearance of 0.0508-0.0762 mm (2-3 mils). A typical lockup eccentricity ratio of 0.5 gives a ratio of shaft vibration to steady state eccentricity of approximately 0.08. Thus the analysis can be reasonably approximated as a quasi-static analysis rather than a transient analysis such as described in [1]. This provides a first evaluation to determine the dynamic coefficients. If desired, the transient analysis can still be carried out to verify the quasi-static approach. The steps involved in the quasi-static analysis are:

1. Before shaft rotation begins, the oil ring is resting on the top of the shaft. The eccentricity ratio is 1.0. As the rotational speed increases,the oil seal fluid

film supports the weight of the oil seal ring W. The force in the oil film acting on the seal ring exactly matches the weight.

2. The pressure drop across the seal ring increases with machine speed. This in turn increases the axial force due to unbalanced pressures, and correspondingly, the radial friction force acting on the seal ring (given by the coefficient of friction times the axial force). When the radial friction force acting on the seal exceeds the oil film radial force, the seal will lock up.

3. As the shaft speed continues to increase, the tilting pad bearing oil film forces tend to force the shaft upward in the bearing. The seal ring oil film forces tend to oppose this because the the shaft is now taking an eccentric position within the fixed seal ring.

4. Unbalances are always present in rotors. These produce forces which make the shaft undergo dynamic orbits. These dynamic orbits in turn produce dynamic oil film forces in the lubricant between the shaft and fixed seal ring. If the combination of static and dynamic forces is larger than the radial friction force, the oil seal ring will move. If the dynamic forces are not large enough to overcome the available radial friction force at a given speed, the seal ring is likely to lockup.

There are several possibilities to be explored for any particular seal geometry and operating conditions.

INITIAL LOCKUP POSITION

Figure 4 shows the shaft, seal ring inner radius, and bearing pad circle at the initial position for zero speed. The clearances are exaggerated for clarity in the illustrations. Note that the seal ring clearance c_r at 0.097485 mm (3.838 mils) is larger that the bearing pivot clearance c_j at 0.07620 mm (3.0 mils). Relative to the bearing center (which is assumed to be the only fixed point as both the shaft and oil seal ring move), the shaft position is $x_j = 0.0$ and $y_j = -0.07620$ mils (-3.0mils) while the seal ring position is $x_r = 0.0$ and $y_r = -0.173685$ mm (-6.838 mils).

As the speed increased, the pressure drop across the seal was assumed to follow a power law curve of the form

$$\frac{P}{P_d} = \left(\frac{N}{N_d}\right)^n \tag{1}$$

Usually the power n is in the range of 1.5 to 2.0. For illustration purposes here the value n=2.0 is used. The maximum available friction force is given by the expression

$$F_{rmax} = \mu_s \, P A \tag{2}$$

With Equation (1) this becomes

$$F_{rmax} = \mu_s \, P_d \, A \left(\frac{N}{N_d}\right)^n \tag{3}$$

The oil film radial force is equal to the maximum available radial friction force at lockup. It has also been noted that during start up the oil film radial force must equal the seal ring weight W. Finally the ring weight equals

$$W_r = \mu_s \, P_d \, A \left(\frac{N_{lock}}{N_d}\right)^n \tag{4}$$

This is easily solved for the desired ring lockup speed with the result

$$N_{lock} = N_d \left(\frac{W_r}{\mu_s P_d A}\right)^{\frac{1}{n}} \tag{5}$$

Note that the dynamic forces due to shaft orbitting are assumed small at the low speeds where lockup takes place.

Table 1 gives the parameters used for calculating the lockup speed of 611 rpm. with the 0.2 coefficient of friction, the required radial friction force is only 147 N (33.2 lbf) which is easily attained when the seal ring is not pressure balanced. The large face area of 8540 mm^2 (13.24 in^2) requires only a small fraction of the design pressure drop across the seal to produce this friction force.

Perhaps the least well known value is the friction coefficient. Standard tables give a value of about 0.1 for carbon to steel surfaces while a measured value for lubricated face seals of 0.3 has been published by Metcalf [6]. Thus an average value of 0.2 was chosen for use here. Actually a range of highest to lowest possible initial lockup speeds was considered but space considerations prevent presentation of all of the calculations made. Overall, the range of possible lockup speeds for this seal design did not produce major differences in the calculated oil seal ring stiffness and damping coefficients evaluated at running speed.

As the shaft speed increases, it lifts off of the bottom pad of the tilting pad bearings. The calculated shaft eccentricity ratio e_j [7,8,9] relative to the bearing is 0.731 at 611 rpm. Note that the rotor weight carried by one bearing of 3600 N (809 lbf) is very large compared to the oil seal ring weight of 29.5 N (6.64 lbf) so the seal ring oil film force has little effect on the bearing load at this low speed. The shaft position relative to the bearing center line is x_j = 0.0 mm (0.0 mils) and y_j = -0.0556 mm (-2.19 mils).

The oil seal ring eccentricity and attitude angle were calculated for the load capacity of 29.5 N (6.64 lbf). A finite element method due to Schmaus [7], described in Appendix A, was used to evaluate the pressures around the shaft including the axial pressure gradient. The calculated ring eccentricity ratio relative to the shaft was 0.463 while the attitude angle for a fully cavitated oil film (as the lubricant film in the seal was taken to be due to the high seal pressure) is 90.0 degrees relative to the load vector. Thus the oil seal ring moves horizontally relative to the shaft and the final oil ring position expressed in terms relative to the bearing center line is x_r = -0.0452 mm (-1.78 mils) and y_r = -0.0556 mm (-2.19 mils). Figure 5 illustrates the final shaft and seal ring positions at lockup.

OIL SEAL RING PROPERTIES AT STABILITY THRESHOLD SPEED

The compressor was observed to have the onset of subsynchronous vibrations at approximately 9000 rpm. The purpose of this section is to determine the seal ring stiffness and damping coefficients at that speed. It is assumed (and verified) that the seal ring remains fixed at the location just calculated as the initial lockup position. As the speed increases from the 611 rpm lockup speed to 9000 rpm, the tilting pad bearing oil film forces make the shaft rise relative to the bearing centerline. This creates an eccentric position of the shaft in the oil seal ring which in turn generates fairly large oil film forces in the seal ring. Both steady state load capacity and dynamic coefficients are developed in the seal ring.

If the shaft had only the shaft weight of 3600 N (809 lbf) acting on it, the calculated bearing eccentricity ratio would be 0.235. The shaft center would then be located at x_j = 0.0 mm (0.0 mils) and y_j = -0.0179 mm (-0.705 mils). The shaft is about 45 degrees to the right and above the ring center. However, the oil film forces acting on the seal ring are important if the seal ring is locked up in the eccentric position described here. These contribute to the total load seen by the bearing.

Table 2 gives the calculated load value and attitude angle for a range of eccentricity ratios using the method of Schmaus [7]. Here the shaft position vector was taken temporarily along the negative y axis and the resulting fluid film forces had the attitude angle shown (relative to the positive x axis). It is easily seen that the load capacity varies from values small compared to the weight carried by each bearing 3600 N (809 lbf) to values above that. However in the mid range of interest here (forces around 890 N (200 lbf)), the attitude angle is nearly constant at about 45 degrees. This means that the relative angle between the oil ring position vector and the oil film force is about 45 degrees. The force acting on the shaft will be about 45 degrees on the counter-clockwise side of the shaft position vector, assuming counter-clockwise shaft rotation. In the range of forces generated in the ring oil film, the force is nearly vertical and can be added directly on to the static bearing load.

The shaft eccentricity in the bearing vs. load is given in Table 3. Matching the vertical position of the shaft y_j from Table 3 to the to the corresponding eccentricity in the oil seal ring from Table 2 gives the shaft position y_j = -0.216 mm (-0.85 mils). This corresponds to a shaft/bearing eccentricity ratio of 0.283 and a shaft/ring eccentricity of 0.583. The downward load imposed by the oil seal ring on the shaft is 756 N (170 lbf) for a total bearing load of 4356 N (979 lbf). Figure 6 indicates the final shaft and oil seal ring positions at 9,000 rpm.

The stiffness and damping coefficients were calculated for the seal ring assuming lockup at the 611 rpm speed. Initially they were obtained from Schmaus [7] using an attitude angle of -90 degrees. The coordinates were then rotated to an attitude angle of -135 degrees to represent the approximate relative position of the shaft relative to the center of the oil seal ring (illustrated in Figure 3). Table 4 gives the calculated values vs. shaft/ring eccentricity. Field engineers noted that the measured axial flow rates varied over approximately the same range as the calculated values, 10-20 gal/min, over a period of time including a number of startups. This gives some confirmation of the analysis.

At the subsynchronous vibration threshold speed, the above results indicate that the lowest possible stiffness and damping values which can occur are the zero eccentricity values if the oil seal ring is locked up. Table 5 shows these values. Note that even if the shaft is centered in the locked up seal ring, the crosscoupled stiffness values are 52,000 lbf/in. This may be enough to drive the compressor into subsynchronous whirl. A more typical value of eccentricity is likely to be about 0.62 which was used in this study for the actual values. Then the crosscoupled stiffness value is 163,000 lbf/in or about three times the centered value. The value of 0.62 was taken to be somewhat on the conservative side in the final design calculations.

The principal stiffness and damping coefficients are not as important as the crosscoupled coefficients because the seals are close to the bearings. Bearing principal stiffness and damping coefficients are much larger and tend to dominate the seal coefficients. Tilting pad bearings have no crosscoupled coefficients when symmetric loading occurs, as it does in this case, so the oil seal cross coupled coefficients are very significant. The bearing coefficients are presented later.

Here the calculated seal ring dynamic coefficients can be compared to a formula presented in Emerick [3]. The oil seal crosscoupled stiffness and principal damping coefficients are

$$
\begin{aligned}
K_{yx} &= -\frac{\pi}{4}\omega D \mu\left(\frac{L}{c_r}\right)^3 \\
K_{xy} &= \frac{\pi}{4}\omega D \mu\left(\frac{L}{c_r}\right)^3 \\
C_{xx} &= \frac{2}{\omega} K_{xy} \\
C_{yy} &= \frac{2}{\omega} K_{xy}
\end{aligned}
\tag{6}
$$

These formulas assume that there is no axial pressure gradient, the seal ring is locked up with the shaft centered and all cavitation is suppressed. Numerical values for the example seal here give

$$-K_{yx} = K_{xy} = 39{,}000 \text{ lbf/in}$$

$$C_{xx} = C_{yy} = 82 \text{ lbf-sec/in}$$

These are compared to the theory used in this paper in Table 5. For the eccentric case, Equation (6) would estimate only one fourth of the crosscoupled stiffness as compared to the full quasi-steady method described here.

SEAL RING BREAKAWAY LOADS

It must be verified that the seal ring is indeed locked up. This can be done by comparing the available radial friction force to the static plus dynamic force exerted on the seal ring by the oil film. The axial pressure drop across the seal is about 600 psi at 9,000 rpm. With an unbalance seal face area of 13.24 in^2, the axial force on the seal ring is 7944 lbf. Assuming a friction factor of 0.2, the available radial friction force is 1589 lbf. Even with a friction factor of 0.1, the friction force is 794 lbf.

The static force on the ring is about 250 lbf even at the largest expected eccentricity of 0.62. An estimate of the dynamic force can be obtained by multiplying the stiffness times the displacement. Field measurements of the orbit size indicate a vibration level of approximately 0.8 mils. For a crosscoupled stiffness of 163,000 lbf/in, the dynamic force would be about 130 lbf. The total of static plus dynamic forces is well below the smallest expected available friction force tending to lockup the seal. A comparable analysis can be done for any speed from the lockup speed on up to operating speed and the conclusion is that the seal ring will be locked up.

STABILITY ANALYSIS

A full rotor dynamic analysis was carried out. This included an undamped critical speed analysis, stability and unbalance response. Due to length restrictions only the results of the stability analysis, which are relevant to the oil seal, are presented. The overall rotor dynamics analysis will be reported in a later paper.

The rotor was examined to determine the sensitivity to different destabilizing mechanisms. These included oil seals, balance piston and aerodynamic crosscoupling at the compressor wheels. Again the volume of analysis done on these effects requires that only partial results be given here. Table 6 gives the logarithmic decrement and damped natural frequencies for the various cases. First, the shaft was taken alone. The logarithmic decrement was zero as expected due to no system damping. When the bearings were added, the system has a good log decrement of 0.19. Next, a value of 19,800 lbf/in was added for the balance piston at one end of the machine near the bearings [10]. Now the log decrement is reduced to 0.14 or a change of about 0.5. The locked up seal with stiffness of 163,000 lbf/in was added (without the balance piston) and the value of log decrement dropped to 0.09. The two effects combined dropped the log decrement to 0.006. The major effect is seen to be the oil seals at both ends of the shaft which produce a change in log decrement of 0.10.

Overall, the balance piston crosscoupling and the seal crosscoupling are not enough to make the rotor unstable. Compressor wheel aerodynamic crosscoupling is often added to complete the list of instability mechanisms [5]. A typical value of 10,000 lbf/in (1667 lbf/in per wheel) placed at the center of the rotor gives the final log decrement discussed in this paper of -0.06. The rotor is finally unstable (at least the calculations indicate instability). The calculated whirl speed is 3780 rpm while the observed whirl speed is 4200 rpm. The calculated value is lower than the measured value by about 10% which is typical of compressor instability calculations with many machines.

DESIGN CHANGE RECOMMENDATIONS

Several possible design changes were proposed to the existing seal which have been known to improve seal performance from a stability viewpoint. The objectives used in assessing the proposed changes were

1. Reduce cross coupled stiffness of the seal while maintaining the flow rate.
2. Reduce the radial friction force (by pressure balancing).

The analysis of the proposed seal design modifications assumes a shaft eccentricity of 0.5 and an attitude angle of -135° for the seal. A smaller eccentricity was selected to represent the largest expected lock up condition for a balanced seal. Also the speed at which the analysis was performed was increased to the actual operating speed of 9800 rpm.

Several possible design modifications were considered.

Type 1: Clearance Region Modifications
Two Groove Seal with c = 3.838 mils (40 mil groove width)
Three Groove Seal with c = 3.838 mils (30 mil groove width)
Two Groove Seal with c = 5.0 mils (40 mil groove width)

Type 2: Ring Balance Modifications
Balanced Ring Face

Figure 7 illustrates the geometries of the grooved and balanced oil seals.

Cutting circumferential grooves, as in type 1, in the seal land region has the effect of reducing the crosscoupled stiffness while keeping the flow rate low. A case with an increased seal clearance is also considered. While this should reduce the cross coupled stiffness, it will also increase the flow rate across the seal.

The existing oil seal's properties were recalculated at 9800 rpm and serve as a comparison for the proposed seal design modifications. Table 7 contains the operating properties of the existing and grooved seals. As expected the grooves greatly reduce the crosscoupled stiffness, by an order of magnitude in the three groove seal over the existing design. The load capacity of the seal is also reduced. It should be noted that the two groove seal with the expanded clearance has crosscoupling terms ten times less than the existing seal but with a flow rate twice as great.

The objective of balancing the oil seal, as mentioned earlier, is to decrease the maximum available friction force tending to lock up the seal ring. This will prevent the seal ring from locking up at a very large eccentricity which could produce large crosscoupling stiffnesses in the seal.

Figure 7.c shows the proposed modified seal face to decrease the axial loading on the seal. The unbalanced axial force is determined from the area of the face shown as 1.047 in^2. This is much less than the face area of the original seal (over 13 in^2). The maximum frictional force at the operating speed is then 202 lbf. This is much more reasonable as a friction force than the 1700-2500 lbf range for the original seal.

Finally, the three grooved seal was selected as the best design. It had the lowest crosscoupled stiffness while maintaining the same flow rate as the existing seal. Balancing of the three groove seal was recommended to assure that highly eccentric seal operation would not occur.

REDUCTION OF SUBSYNCHRONOUS VIBRATIONS

To reduce the subsynchronous vibrations two possible machine changes have been studied: seals and bearings. Only the changes in stability due to seal modifications are presented. The bearing effects will be presented in another paper.

As previously mentioned, stability at the operating speed of 9800 rpm is desired. To predict the stability at this speed, the aerodynamic crosscoupling has been assumed to vary linearly with speed. Namely, at 9800 rpm the aerodynamic crosscoupling is 8490 lb/in. The balance piston stiffness was assumed to be constant with speed.

With the existing seals the system has a log-decrement of -0.075 at 9800 rpm (unstable). By replacing the original seals with 3-groove seals the log decrement increases to 0.12 (stable).

CONCLUSIONS

An unbalanced seal design such as the one considered here is likely to produce large available friction forces acting on the seal ring. These will tend to lock the seal ring against the fixed components at some speed well below running speed. As the shaft increases in speed, the oil film forces in the bearings will lift the shaft in the bearing. This will create an eccentric position of the shaft relative to the locked up oil seal ring. This in turn will produce large crosscoupling stiffness terms acting on the shaft due to the oil film in the seal ring. In the case considered here, the crosscoupled stiffness could be as large as 163,000 lbf/in for each oil seal.

A quasi-steady method of determining the seal ring lockup conditions has been presented. It involves matching the radial friction force to the seal ring weight. While no direct evidence of its accuracy, such as ring measured orbits are available for comparison, indirect verification by matching calculated instabilty thresholds with measured ones is shown here. The method appears to be much easier to use than the time transient method decribed by other authors [1,2].

If a seal ring does lockup, the possible range of dynamic coefficients goes from the centered position to the largest likely eccentricity. The centered position can produce fairly large crosscoupled stiffness values for the seal ring oil film. In this case the value was about 52,000 lbf/in. The eccentric values are typically several times this value, in this case about three times as large.

Use of the centered-no axial flow formula presented in Emerick [3] would result in an oil seal crosscoupled stiffness of 39,000 lbf/in. Compared to the value of 163,000 when the eccentric case with axial flow is considered, Equation (6) does not appear to be a good approximation. Also the principal stiffness terms are neglected by Equation (6). The centered-no axial flow formula probably should not be employed particularly as it seems to strongly underestimate the oil seal effects.

The oil seal ring fluid film will generate additional loads acting on the bearing. In this case, the additional load was about 25% of the steady bearing load due to shaft weight. It appears likely that the seal ring added load will normally be in the vertically downward direction although no general studies were carried out to verify this. The change in the bearing dynamic coefficients was found to be only about 10% or less with the additional load. Thus the effect of the bearing on moving the shaft eccentrically in the locked up seal ring was the major effect. Put another way, the change in eccentricity in the seal ring produced a large effect in the oil seal ring dynamic coefficients while the change in eccentricity in the bearing did not modify the bearing dynamic coefficients very much.

The oil seal lockup was found to be the major effect in driving the compressor into subsynchronous vibration. It produced a lowering of logrithmic decrement of 0.10 or more as compared to balance piston (0.05) or aerodynamic crosscoupling (0.066). Calculation of a negative log decrement to drive the rotor unstable required all three mechanisms of instability. The calculated value of whirl speed at 3780 rpm compares fairly well with the measured value of 4200 rpm.

To reduce the subsynchronous vibrations, modifications were recommended to the oil seals. The existing seal was modified to have three grooves cut in the land region. A groove width of 0.7616 mm (30 mils) and a groove depth of about 2.54 mm (0.1 in) was recommended. The three groove design reduced the crosscoupled stiffness from 109,000 lbf/in to an average value of 6,000 lbf/in. Further, grooving did not increase the seal leakage significantly. Balancing of the seal was recommended to prevent the seal ring from locking up at large eccentricities. Thus very large crosscoupled stiffness terms were prevented. Cutting a relief on the face of the seal reduced the friction force (tending to lock the seal up) from about 1700-2500 lbf in the existing seal to about 200 lbf in the proposed seal design.

The stability of the existing compressor was then examined at the operating speed of 9800 rpm. Replacing the original oil seals with the three groove design increased the log decrement to 0.12 (stable). Due to plant shutdown, the proposed seal redesign has not yet been implemented.

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Table 1. Oil Seal Ring Parameters For Initial Lockup Condition

W_r = 29.5 N (6.64 lbf)
μ_s = 0.2
P_d = 1.85 x 10^6 N/mm^2 (644 lbf/in^2)
A_r = 8,540 mm^2 (13.24 in^2)
N_d = 9800 rpm

Table 2. Load Capacity For Original Seal With Shaft Along Negative y Axis at 9,000 RPM

ECC	W_x (LB)	W_y (LB)	LOAD (LB)	ANGLE
0.00	.00	.00	.00	-89.93
.10	20.21	22.04	29.90	-42.52
.20	41.79	44.06	60.72	-43.48
.30	66.43	65.91	93.58	-45.22
.40	96.85	87.96	130.83	-47.76
.50	138.07	109.93	176.49	-51.47
.60	200.35	131.78	239.81	-56.66
.70	309.57	153.40	345.49	-63.64
.80	516.45	261.65	578.95	-63.13
.90	1019.89	1393.99	1727.24	-36.19

Table 3. Tilting Pad Bearing Load vs Eccentricity and Shaft Position

Load (lbf)	Bearing Eccentricity ε_j	Shaft Position In Bearing y_j (mils)
784	0.229	-0.69
827	0.240	-0.72
834	0.242	-0.73
945	0.270	-0.81
1071	0.300	-0.90
1205	0.330	-0.99
1350	0.360	-1.08
1508	0.390	-1.17

Table 4. Subsynchronous Threshold Stiffness and Damping for Existing Seal

ECC	LOAD (lbf)	KXX	KXY	KYX	KYY
			(lbf/in)		
0.00	1.	57401.	52065.	-52063.	57400.
0.10	30.	57968.	53201.	-53209.	56820.
0.20	61.	59842.	56880.	-56917.	54889.
0.30	94.	63843.	64289.	-63806.	51183.
0.40	131.	71161.	76957.	-76772.	43574.
0.50	176.	85577.	100239.	-100344.	28850.
0.60	240.	116701.	146478.	-146928.	-2744.
0.70	345.	195577.	263713.	-254970.	-82789.
0.80	579.	698264.	781308.	-112971.	140389.
0.90	1727.	5334498.	5226771.	2383265.	3082203.

a) Stiffness at -135° Attitude Angle and 9,000 RPM

ECC	LOAD (lbf)	CXX	CXY	CYX	CYY
			(lbf-sec/in)		
0.00	1.	110.	0.	0.	110.
0.10	30.	110.	-1.	-1.	110.
0.20	61.	119.	3.	3.	119.
0.30	94.	134.	12.	12.	134.
0.40	131.	162.	28.	28.	163.
0.50	176.	212.	60.	60.	214.
0.60	240.	313.	130.	129.	315.
0.70	345.	548.	306.	306.	553.
0.80	579.	1176.	704.	692.	841.
0.90	1727.	4170.	2681.	2567.	2124.

b) Damping at -135° Attitude Angle and 9,000 RPM

Table 5. Stiffness and Damping For Existing Seal At Subsynchronous Vibration Analysis (At 9,000 RPM and -135^{o} Angle)

Case		K_{xx}	K_{xy} (lbf/in)	K_{yx}	K_{yy}	C_{xx}	C_{xy} (lbf-sec/in)	C_{yx}	C_{yy}
Centered with Axial Flow	0	57,400	52,060	- 52,060	57,400	110	0	0	110
Eccentric with Axial Flow	0.62	128,000	163,000	-163,000	147,000	360	165	164	363
Centered -No Axial Flow-Eq.(6)	0	0	39,000	- 39,000	0	82	0	0	82

Table 6. Sensitivity of Current Design to Different Destabilizing Mechanisms at 9,000 RPM

	Logarithmic Decrement δ	Whirl Speed (cpm)
Shaft Only (0.5 x 10^{6} lb/in Supports)	0.00	3070
Shaft, Bearings	0.19	3555
Shaft, Bearings, Balance Piston (19.8 Klb/in)	0.14	3550
Shaft, Bearings, Seals	0.09	3770
Shaft, Bearings, Seals, Balance Piston	0.006	3770
Shaft, Bearings, Seals, Balance Piston, Aerodynamic Cross Coupling at Wheels (10 Klb/in)	-0.06	3780

Seal Design	Load (lbf)	Flow Rate (gal/min)	Stiffness (lbf/in)				Damping (lbf-sec/in)			
			K_{xx}	K_{xy}	K_{yx}	K_{yy}	C_{xx}	C_{xy}	C_{yx}	C_{yy}
Existing	186	13.5	88,200	109,300	-109,600	26,000	213	61	61	215
Two Groove (1A)	92	12.8	46,300	6,680	-13,910	41,400	20	5	5	20
Three Groove (1B)	84	12.1	40,500	1,750	-9,930	37,900	11	3	3	11
Two Groove Large Clearance (1C)	46	21.6	50,300	-360	-11,800	48,100	11	2	2	11

Table 7 Summary of Seal Properties At 9,800 RPM with Attitude Angle -135^{o} and Eccentricity 0.5

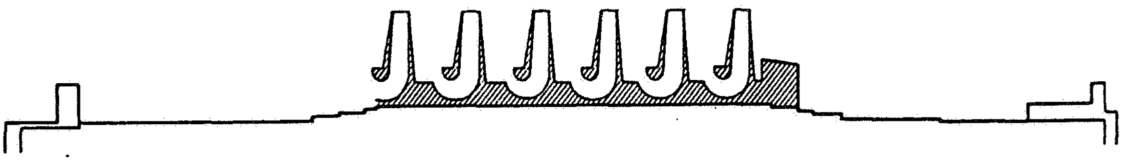


Figure 1. Compressor Geometry

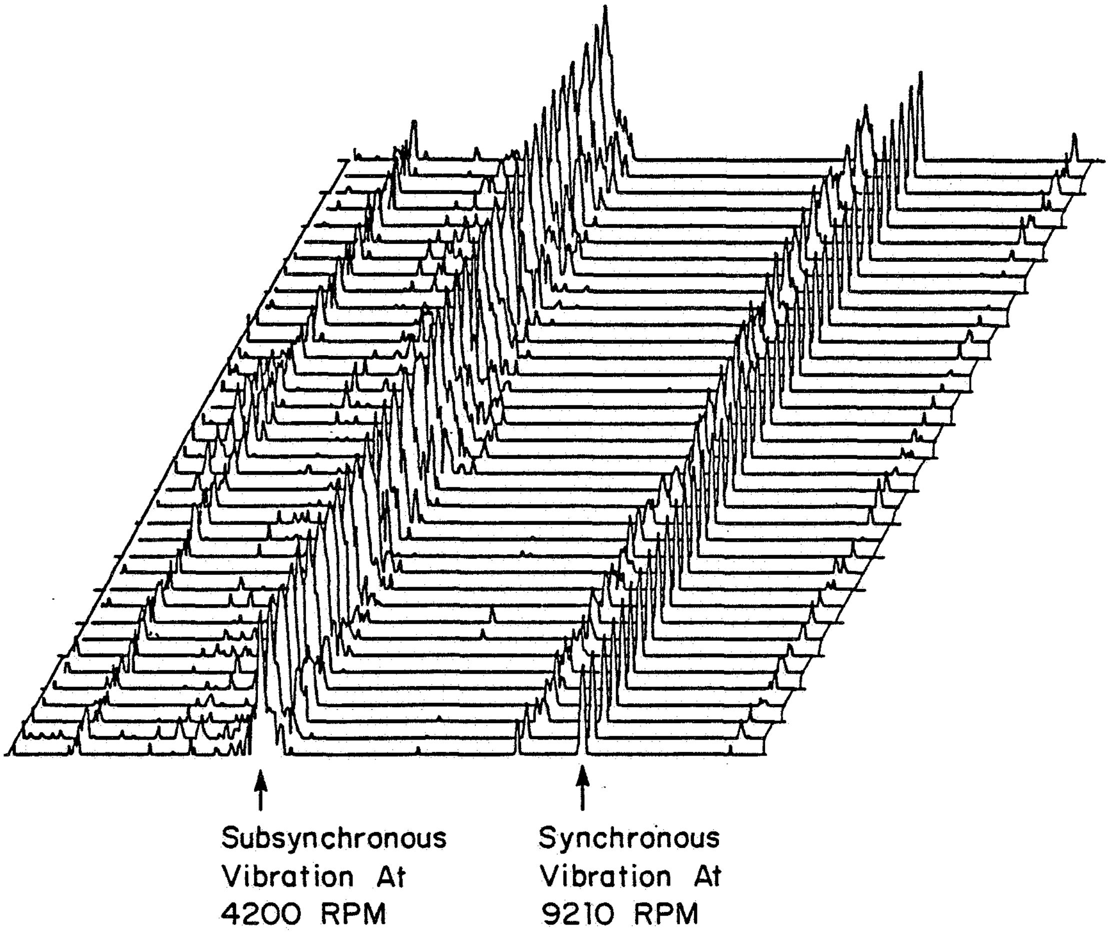


Figure 2. Vibration Pattern At 9210 RPM

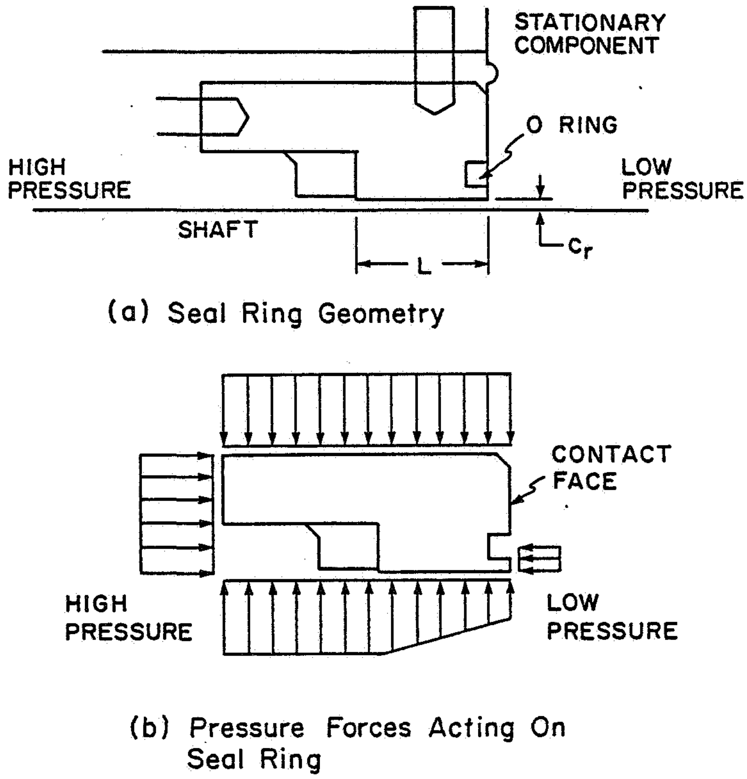


Figure 3. Seal Ring Geometry and Pressure Forces

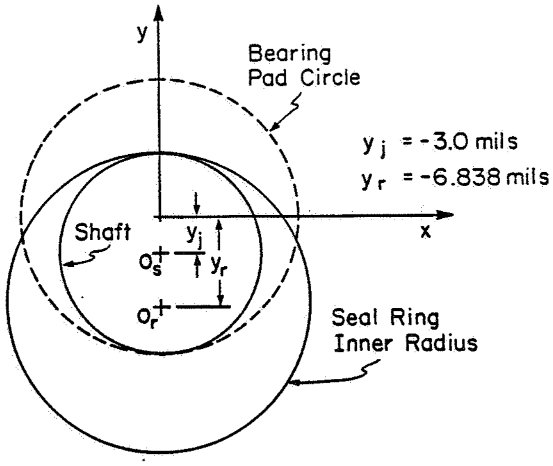


Figure 4. Bearing and Seal Ring Clearance Circle Plots At Zero Speed

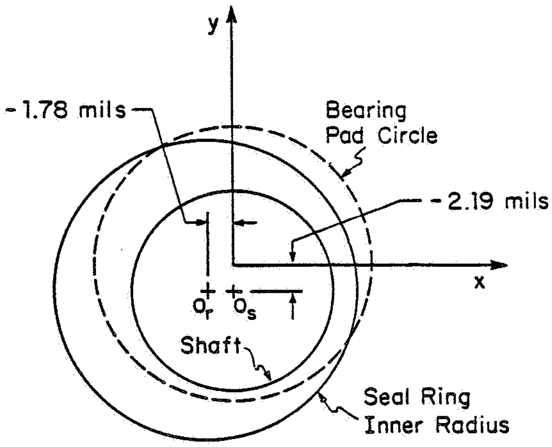


Figure 5. Bearing and Seal Ring Clearance Circle Plots At Lockup Speed (611 RPM)

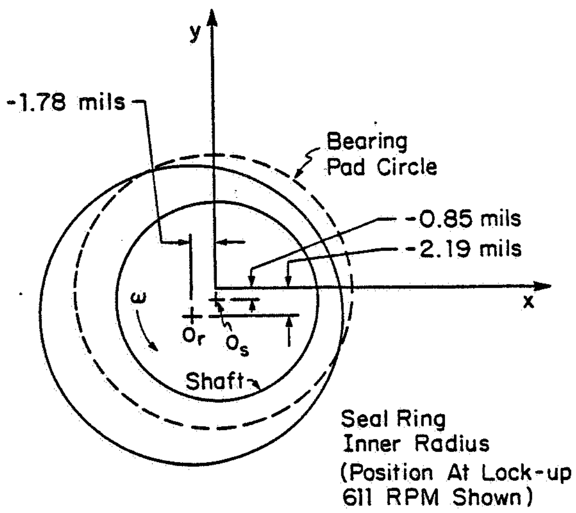
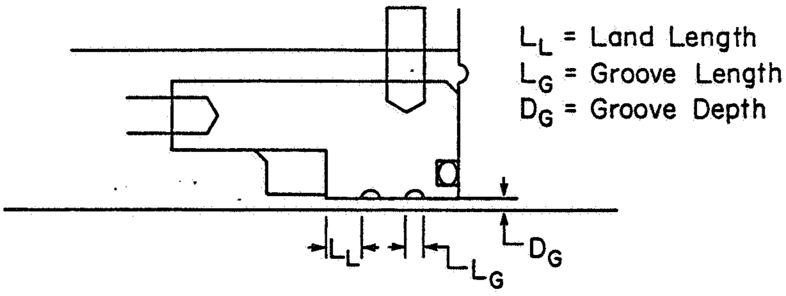
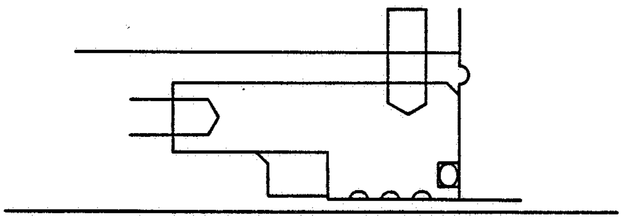


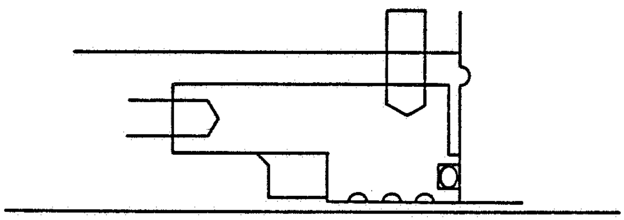
Figure 6. Bearing and Seal Ring Clearance Circle Plots At Instability Threshold Speed (9,000 RPM)



(a) Two Groove Design



(b) Three Groove Design



(c) Balance Ring Face Design

Figure 7. Geometry For Circumferential Groove And Balancing Seal Modifications