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A QUASI-STATIC METHOD FOR THE CALCULATION OF LOCK-UP SPEED IN FLOATING-RING OIL SEALS

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ABSTRACT

A quasi-static method for calculation of lock-up conditions in floating ring oil seals is presented. The analysis considers a compressor rotor mounted on tilt-pad bearings and floating ring seals. The seal-rotor-bearing interactions during the compressor star-up are simulated to determine if the seals lock and to obtain the final operating eccentricities of seals and tilt-pad bearings. Results from the analysis identify the seal lapped area as a major factor for determination of the seal lock-up operating eccentricities, since to prevent lock-up the seal must generate al all speeds a fluid force greater than the current maximum friction force available at the seal lapped face.

NOMENCLATURE

A_F	Area of the seal lapped face (m^2)
A_G	Area of the annular section of the groove at the seal exit (m^2)
C, C_B	Radial clearance (m)
D	Journal diameter (m)
e_x, e_y	Journal eccentricities in X and Y directions respectively (m).
F_B	Tilt-pad bearing load (N).
F_F	Friction force at the seal lapped face (N).
F_S, F_{SP}	Oil seal load, and seal axial preload (N).
F_X, F_Y	Fluid film forces in X and Y directions (N).
L	Total seal axial length (m).
M	Rotor mass (kg).
M_X, M_Y, M_Z	Moments in X, Y and Z directions respectively (N m).
N_L, N_d	Operating and design rotational speed (RPM).
P, P_d, P_L	Fluid pressure, design and operating total pressure drop across the seal (Pa).
S	$(\eta_s L^3 D N_d)/(60 W C^2)$ Modified Sommerfeld Number for seals and tilt-pad bearings (rad).
W_R, W_S	Rotor weight and seal package weight (N).
X,Y,Z	Inertial coordinate system (m).
XB, YB	Tilt-pad bearing eccentricity ratios in X and Y directions.
XS, YS	Oil seal eccentricity ratios in X and Y directions.
ΔP	Total pressure drop across the seal (Pa).
ε	e/C. Dimensionless eccentricity.
μ	Fluid viscosity (Pa-s).
μ_c	Friction coefficient at seal-lip interface.

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INTRODUCTION

Oil seals are commonly used in multistage compressors to minimize the leakage of the process fluid while allowing for limited lubricant flow. Oil seat rings are floating bearings that when locked due to friction forces at the seal lapped face, operate as hydrodynamic bearings and generate substantial cross-coupled dynamic force coefficients. If cross-coupled stiffnesses are large enough, the oil seal package may become a source of subsynchronous vibration and prevent the machinery from operating at rated conditions. Floating ring seals are designed to follow journal radial motions. Under these operating conditions, the dynamic forces induced on the journal by the floating ring are small. However, when oil seals are prevented from moving they can be the source of subsynchronous vibration due to the large dynamic forces generated by the lubricant film. This condition is usually identified as seal lock-up (Emerick et al., 1982).

Emerick (1982) reported an operating experience of a compressor which presented an interesting history of sporadic increases in shaft vibration. He concluded that stable operation with the design seal clearances would be possible only if the seals did not lock and operated at the concentric position. Even if lock-up occurs, stable operation is possible provided that the seal clearances are large or that the seal effective length is reduced by grooves. Allaire et al. (1986) presented aquasi-steady-state method for calculating the seal lock-up position. Numerical predictions of cross-coupled stiffnesses based on a finite element model were found to be four times larger than those obtained from the simple formulae proposed by Emerick (1982). The results showed conclusively the importance of flow turbulence, axial pressure drop and entrance inertial pressure drop on the dynamics of the oil seals. Tanaka et al. (1986) studied the effect of floating ring oil seals in turbocompressor rotordynamics. The subsynchronous vibration present due to the seal lock-up was suppressed by replacing the single-land oil seal by a four-land oil seal with the same total length. The seal lockup condition were calculated by considering only the static loads on the rotor bearing system.

Once locked, the seals act as journal bearings and may introduce large static and dynamic forces into the rotordynamic system. Semanate and San Andres (1992), presented an analysis for multi-land high pressure oil seals operating on the laminar/turbulent flow regimes with moderately high values of the axial Reynolds numbers. Bearing surface roughness and fluid inertia and viscous effects at the seal entrance were accounted for Rotordynamic characteristics for one-, two-, and three- land seals showed that multi-land seal geometries reduce I simultaneously the seal cross-coupled stiffness and direct damping forces preserving the limited dynamic stability characteristics of the locked seal. Semanate and San Andres (1993) extended the previous analysis to account for thermal effects on oil seals. A thermal bulk-flow analysis for short-length oil seals is coupled with the continuity and momentum equations to describe the lubricant flow in multi-land seals. Comparisons among results obtained from this thermal model and the classical effective viscosity method (Pinkus, 1990) show good agreement for concentric operation. However, as eccentricity increases, the comparisons strongly suggest that an isothermal model based on an effective viscosity could be misleading. An important outcome of the analysis presented by Semanate et al. (1993) is the prediction of he maximum seal temperature. For the seal cases studied, a maximum temperature of more than three times the inlet temperature was found. This maximum temperature can not be predicted by an isothermal analysis and constitutes an important design tool that may be used to avoid seal seizure, localized seal overheating, etc.

This paper presents a method to calculate the seal lock-up speed by a quasi-static simulation of a compressor start-up. A commonly used arrangement of a compressor rotor mounted on tilt-pad bearings and floating ring oil seals is considered to illustrate the method for lock-up calculation. Figure 1 shows a schematic view of the system studied. The compressor rotor, represented by a rigid rotor of modal mass M, is supported on tilt-pad bearings. While process gas leakage to the environment is prevented by the oil seals shown next to each tilt-pad bearing. Oil seal lock-up occurs as a result of the seal-bearing-rotor dynamic interactions that take place during the turbomachinery start-up. However, a quasi-static approach can be used to estimate the seal lock-up because it happens at relatively

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small rotational speeds where dynamic forces do not playa primary role (Allaire et al, 1986). Additionally, the fluid flow in oil seals is characterized by small Reynolds numbers, and thus bearing inertia forces are small compared to viscous forces. This implies that journal transient motions are quickly damped out prevailing only a small orbit around the journal equilibrium position due to rotor unbalance. The radius of this steady-state orbit is usually less than 5% the value of the seal clearance (Emerick, 1982).

QUASI-STATIC ANALYSIS FOR A COMPRESSOR ROTOR MOUNTED ON TILT-PAD BEARINGS AND FLOATING RING OIL SEALS

The static forces present on the rotor and on the floating ring are represented in figure 2. The system equilibrium is dictated by Newton's second law as follows:

a) For the rotor supported on bearings:

$$F_{Bx} - F_{Sx} = 0 \tag{1}$$

$$F_{By} - F_{Sy} = \frac{W_R}{2} \tag{2}$$

$$\sum M_X = \sum M_Y = 0 \tag{3}$$

where F_B and F_S are the forces generated by the tilt-pad bearing and by the floating ring respectively, and W_R is the rotor weight.

b) For the floating rings:

$$F_{Sx} - F_{Fx} = 0 \tag{4}$$

$$F_{By} - F_{Sy} = W_S \tag{5}$$

$$F_Z = F_{SP} + \Delta P\left(\frac{A_F}{2} + A_G\right) \tag{6}$$

where F_Z is the total axial force acting on the seal package, F_{SP} is the force produced by the seal spring, F_F is the friction force generated at the seal lapped face, W_S. is the seal weight, A_F is the area of the seal lapped face, and A_G is the area of the annular section of the groove at the seal exit, as indicated in figure 2.

The maximum friction force available at the lapped seal face is dictated by the Coulomb law for dry friction as:

$$F_{F\max} = \mu_c F_Z \tag{7}$$

where μ_c is the static coefficient of friction at the lapped face. Typical values for steel-steel contact are 0.1 or 0.2 (Allaire et al, 1986).

For compressor applications, the total pressure drop across the seal is related to the rotor rotational speed by means of a power law of the form (Allaire et ai, 1986):

$$\left(\frac{P}{P_D}\right) = \left(\frac{N}{N_D}\right)^n \tag{8}$$

where $1.5 \le n \le 2$ are typical values.

A description of the tilt-pad bearings and seals used as example for this analysis is given on the discussion section.

THE NUMERICAL SOLUTION PROCEDURE

The set of equations (1) to (6) are used for the simulation of the compressor start-up and further determination of the rotational speed and eccentricity at which the seal locks. The numerical procedure is as follows:

At zero speed, the shaft rests on the tilt-pad bearings and the floating rings rest on top of its journals (shaft). As the rotational speed increases, two effects take place. First, hydrodynamic forces are generated both at the oil seals and at the tilt-pad bearings. Second, the total pressure drop across the seal increases as dictated by equation (8). This, in turn, produces larger friction forces at the seal-case interface. When the seal hydrodynamic forces can not overcome the friction forces, the seals lock and become the potential source of subsynchronous instability.

An iterative algorithm, based on the quasi-static method described above, is developed to track the journal motion while testing for lock-up conditions. The calculation begins with a small rotational speed, typically 200 RPM. At this speed the seal and bearing eccentricities are calculated assuming that the seals support only their own weight, and that the tilt-pad bearings sustain the rotor weight. The rotational speed is then increased to a N+ Δ N value (Typically A ΔN=50RPM) and the bearing eccentricity is recalculated. As the maximum permissible friction force at the seal lapped face is speed dependent, it is also updated by using equations (6) to (8). The rotor position relative to the tilt-pad bearing and to the floating ring are coupled. Therefore, when the bearing eccentricity changes, the seal eccentricity and fluid forces need to be recalculated. The vectorial summation of the fluid force and the seal weight determines the net radial force that needs to balance the friction force at the lapped face. If the net radial force is larger than the maximum available friction force, the seal slides on its seat until the new equilibrium position is reached; otherwise, the seal absolute position is unchanged and the seal is said to be 'locked' at that speed. The process continues until the system equilibrium position at that speed is found. System equilibrium is declared when the bearing and the seal forces changes between two successive iterations are smaller than 1%. The system speed is then increased to a new N=N+ΔN velocity and the same procedure explained above is repeated. The numerical procedure continues until the design rotational speed is reached. Note that for each rotor speed, seal and bearing operating eccentricities and load capacities are obtained by interpolating the corresponding Sommerfield numbers. Eccentricity Ratio curves that are provided as input data for the numerical procedure.

RESULTS AND DISCUSSION

A quasi-static method is used to simulate the start-up of a compressor rotor mounted on tilt-pad bearings and floating ring oil seals. The seals are disposed as shown in figure 1, with a clearance of 97.5µm, a sealing length of 0.02 m, and a weight of 3 kg per seal. The seals are mounted on a journal of 0.1143 m of diameter and the design rotational speed is 9,800 RPM. At this velocity, the compressor generates a pressure drop of 4.45MPa across the seals. Table 1 gives the values of the Modified Sommerfeld Number (see Nomenclature) and attitude angle versus eccentricity ratio (journal eccentricity over nominal radial clearance) for the oil seals considered in the present example. This table was obtained by using a computer program developed earlier (Semanate and San Andres, 1993).

The bearing configuration is a 5 pad, tilt-pad design with load on pad. The journal diameter is 0.01143 m and bearing L/D ratio is 0.44. The bearing preload is about 0.5 and the assembled clearance is 761µm. The rotor, which is fully supported by the tilt-pad bearings has a weight of 736 kg. Table 2 gives the Modified Sommerfeld Number versus Eccentricity Ratio for the tilt-pad bearings (Allaire et al., 1986).

The numerical model uses the data given in tables 1 and 2 to determine the equilibrium position of the system. The oil seals are locked when the fluid film radial force and the seal weight added vectorialy can not surpass the maximum friction force available at a particular instant. Therefore, the lock-up rotational speed is determined by tracking these two forces as the rotor speed increases. For clarity in the discussion that follows, the first force mentioned above is called "fluid radial force", white the second one is named 'friction force". Two cases are considered. In the first one, the seal lapped face has an area of 0.00854 m^2, while in the second one this area is reduced to 0.000675m^2.

Figure 3 shows the evolution of the bearing force, one half the rotor weight, the fluid radial force and the friction force for the case of zero spring force and area of 0.00854 m^2 (13.237 in^2). For rotor velocities smaller than 1000 RPM the maximum friction force is not large enough as to definitely overcome the fluid radial force. However, at a velocity of approximately 1000 RPM the seal locks. After this point the friction force available at the lapped seal face is always larger than the actual radial force. As rotational speed increases, the tilt-pad bearings support not only the rotor weight, but also the load imposed by the oil seals. Figure 4 shows the seal and bearing eccentricities as the rotational speed increases. The floating ring absolute position as measured from the center of the tilt-pad bearing is also included in this figure. Note that after a speed of about 1000 RPM, the seal ring absolute; position does not change since it is locked. On the other hand, the seal and bearing eccentricities continue their Variations as the compressor rotor accommodates itself to the new fluid forces generated by the increasing speeds. Finally, at

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9800 RPM, the seal reaches a steady state dimensionless eccentricity of 0.75 while the journal bearing final dimensionless eccentricity is 0.32. Large cross-coupled stiffnesses are expected since the seal equilibrium eccentricity is fairly large. This will reduce the stability of the rotor-seal-bearing system.

Figure 5 shows the bearing force, the rotor weight and the seal forces acting at the lapped surface for the same seal but with the addition of an axial spring force of 500 N. In this case the seal is locked from start-up because the holding friction force is large even for zero rotational speed due to the addition of the axial spring. However, as rotational speed increases, the fluid film forces become large quite rapidly and overcome the friction forces at a rotational speed of 700 RPM. At this point, the seal slips to another equilibrium position and stays locked at that place until it reaches the operating speed. A very interesting conclusion obtained from this figure is that, contrary 1.0 the former case, the seals support some of the rotor: weight. Therefore no final conclusion can be withdrawn regarding the effect of oil seals on the distribution of static forces in the system.

Figure 6 shows the rotor loci respect to the seal and to the tilt-pad bearing. Note that the seal and tilt-pad bearing: eccentricity ratios (XS, YS,XB, VB) are defined as, $XS=e_x/C$, $YS=e_y/C$ and $XB= e_x /C_B$, $YB= e_y/C_B$ ' where C and C_B are the seal and tilt-pad bearing nominal clearances, respectively. The rotor center in the tilt-pad bearings, as expected, travels in the direction of the load similarly as in the former case. For the floating rings, the rotor travels, in the direction of the load too, since the seal locks form start-up. However, at 700 RPM, the fluid forces balance the friction forces at the lapped face and the floating seals jump. The rotor continues to be lifted by the tilt-pad bearings, but in this case, due to the new position acquired by the seal after the slipping, the rotor upward motion produces a centering effect in the oil seals. Finally, at the design rotational speed, the seal eccentricity is 0.467 and the bearing eccentricity is 0.205. Therefore, a much smaller destabilizing effect than before is expected because smaller cross-coupled stiffnesses are generated by the seal.

The effect of a reduction in the seal lapped surface is analyzed next. The lapped area is reduced to 0.000675 m^2 while keeping the spring force equal to N. Figure 7 shows the one-half rotor weight and illustrates the evolution of the bearing force, of the fluid radial force and of the friction force. Because of the spring force, the seals are locked at a very low rotational speed. However, at approximately 600 RPM, the fluid radial forces surpass the friction forces and the seal slides. As a consequence of the reduction of the seal lapped area, the floating rings do not lock. However the fluid friction forces are only slightly larger than the maximum friction forces, which for this case have become relatively small. Figure 8 shows the seal and bearing loci. As in the other cases studied, the rotor travels aligned with the tilt-pad bearing load. In the case of the oil seal, a large jump is produced the first time the fluid forces overcome the friction forces. After that, as the seal is not locked, the rotor path in the oil seals is not vertical. The final equilibrium eccentricity at the design is speed is 0.426 for the oil seals and 0.248 for the tilt-pad bearings. An important observation here is that although the seal is not locked, however its final eccentricity is only 8.6% smaller than when the seal is locked. It means that preventing the seal from locking-up does not eliminate the possibility of subsynchronous instability problems in the system. To guarantee satisfactory operating conditions, it should be checked that the friction forces at the operating speed are small enough to determine small.

CONCLUSIONS

The start-up of a compressor rotor mounted in tilt-pad bearings and oil seals, is simulated by means of a quasistatic method. The results show that axial forces generated by the spring in the seal assembly prevent seal wear during start-up since they impede displacements at small rotational speeds. The seal lapped face area is identified as a major factor in the determination of the seal lock-up velocity and equilibrium eccentricity. A careful selection of this parameter can prevent the seal lock-up and diminish the possibility to produce rotordynamic instabilities. Nevertheless, seal operating eccentricities can be large since to prevent the lock-up, the seal must generate at all speeds a fluid force greater than the current maximum friction force available at the seal lapped face.

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The examples presented show that oil seals may generate loads larger than 60% the nominal tilt-pad bearing load capacity calculated on basis of the rotor weight. Therefore, tilt-pad bearings need to be designed accounting for seal loads to prevent the damage that could arise from bearing overheating, deformation or wearing.

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APPENDIX ANALYSIS OF FLOATING RING OIL SEALS

Floating ring seals handle high viscosity lubricants and operate with total pressure drop across the seal up to 5.51MPa. These operating conditions induces in the lubricant flow two important characteristics. First, fluid inertia effects within the thin oil film are small and can be regarded as negligible (Semanate and San Andres, 1992). Second, the flow regime may be laminar, turbulent, or it can be located in the transition region. Under these premises, the fluid flow in the thin annular gap of oil seals can be described by the lubrication Reynolds equation. When turbulence is included, the Reynolds equation for incompressible fluids takes the form (Constantinescu, 1973)

$$\frac{\partial}{\partial x}\left(\frac{H^3}{kx\mu}\frac{\partial}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{H^3}{kx\mu}\frac{\partial P}{\partial y}\right)=\frac{k_j}{k_x}\frac{\Omega R}{2}\frac{\partial H}{\partial x}+\frac{\partial H}{\partial t} \tag{1}$$

where the x,y coordinate system is shown in figure A.1 , P is the fluid pressure, H is the film thickness, μ is the fluid viscosity, Ω is the journal rotational speed, R is the journal radius, and k_j and k_y are shear parameters. (They are equal to 12 for laminar flow (Semanate and San Andres, 1992).

Thermal effects have recently been identified as important parameters in floating ring oil seal analysis. In fact, large temperature rises are expected in oil seals due to the typically high lubricant viscosity, large rotational speed and pressure drop, and tight clearance. The energy equation needs to be solved in conjunction with the Reynolds equation to properly account for thermal effects in oil seals. For inertialess lubrication problems, the energy equation is (Constantinescu, 1973)

$$\rho C_p H\left[\frac{\partial T}{\partial t}+U_0\frac{\partial T}{\partial x}+V\frac{\partial T}{\partial y}\right]=-Q_s+\Omega R_r^H{}_{xz}-V.H\frac{\partial P}{\partial y} \tag{2}$$

where T is the fluid temperature, ρ and C_p are the fluid density and specific heat respectively, Q_s is the heat transferred to the seal surroundings, and $\tau^H{}_{xz}$ is the shear stress at the floating ring surface.

Equation (1) and (2) along with appropriate boundary conditions fully describe the lubrication problem in oil seals. Finite volume, finite difference and finite element based method have been devised to solve this system of partial differential equations. (See, for example, Semanate and San Andres, 1992). Once the pressure field is obtained, fluid film forces are easily extracted by

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integrating the pressure field on the journal surface as follows,

$$F\alpha = \int_0^L \int_0^{2\pi R} P_0 H_\alpha dxdy \qquad (3)$$

$$\alpha = X, Y$$

Where Hα a is equal to cos(x/R) for α = X, and it is equal to sin(x/R) for α=Y.

Oil seal rotordynamic force coefficients are found by perturbing the flow variables about a static equilibrium position. The expression for these coefficients is,

$$K\alpha\beta + i\omega C\alpha\beta = 2\int_0^L \int_0^{2\pi R} P\beta H\alpha dxdy \qquad (4)$$

$$\alpha, \beta = X, Y$$

where k$\alpha\beta$ are the seal induced dynamic stiffness, C α β are the seal dynamic damping, ω is the excitation frequency, P α is the perturbed pressure, and X and Y are the coordinates for the journal based inertial coordinate system, as shown in figure A.1.

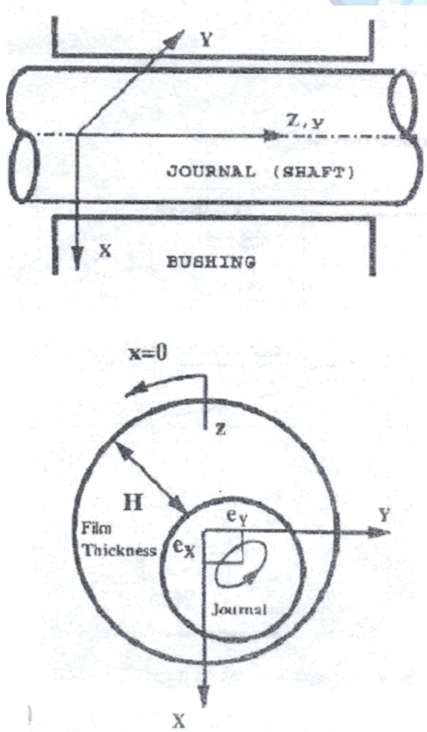

Figure A.1. Annular gap geometry

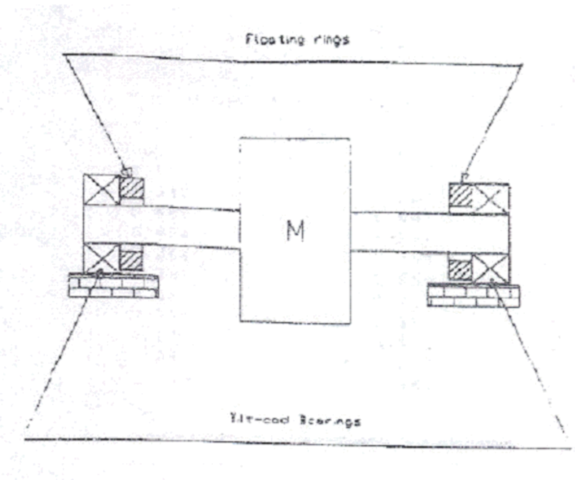

Figure 1. Compressor rotor mounted on tilt-pad bearings and floating ring oil seals.

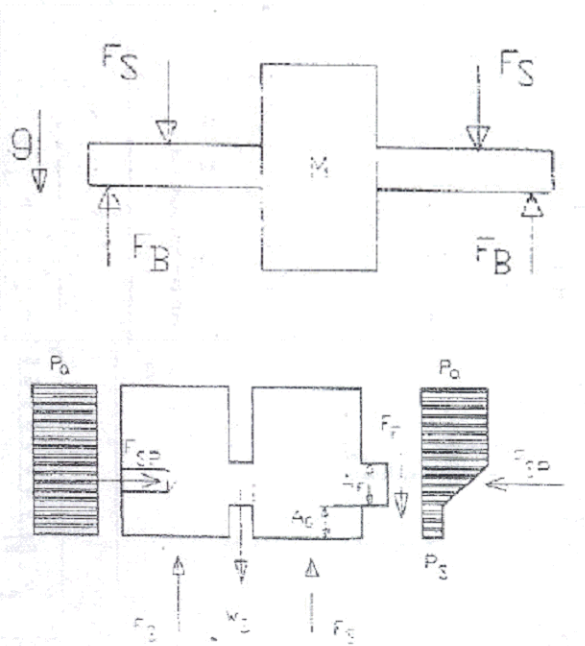

Figure 2. Static forces acting on the rotor and on the floating rings.

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Table 1.- Modified Sommerfeld Number, attitude angle and eccentricity ratio for the floating ring oil seals.

Eccentricity Ratio e/C[illegible]	Modified Sommerfeld Number*	Attitude Angle (rad)
0.01	18.809	1.558
0.05	2.342	1.507
0.1	0.889	1.444
0.2	0.438	1.317
0.3	0.284	1.190
0.4	0.203	1.064
0.5	0.151	0.937
0.6	0.111	0.808
0.7	0.077	0.676
0.8	0.046	0.532
0.9	0.015	0.364

Table 2.- Modified Sommerleld Number, attitude angle and eccentricity ratio for the tilt-pad bearings.

Eccentricity Ratio e/C[illegible]	Modified Sommerfeld Number*	Attitude angle (rad)
0.05	3.153	0.0
0.1	2.296	0.0
0.2	1.218	0.0
0.3	0.646	0.0
0.4	0.343	0.0
0.5	0.182	0.0
0.6	0.096	0.0
0.7	0.051	0.0
0.8	0.027	0.0
0.9	0.014	0.0

* See Nomenclature

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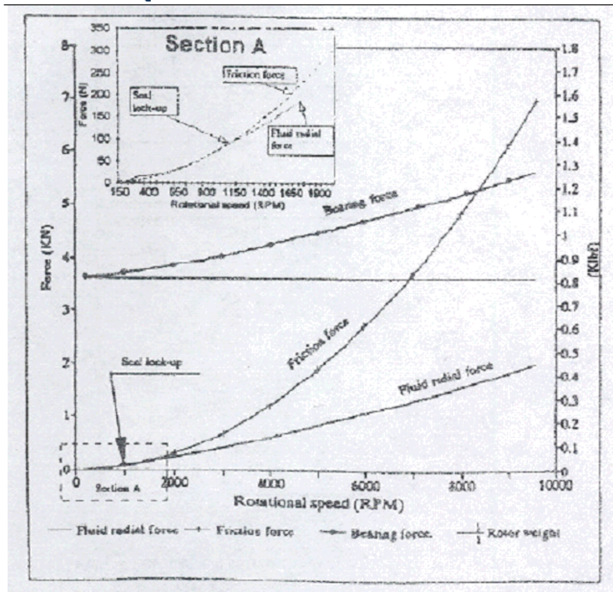


Figure 3. System forces vs. rotational speed. Spring force = 0. N. seal lapped area = 0.0085m^2.

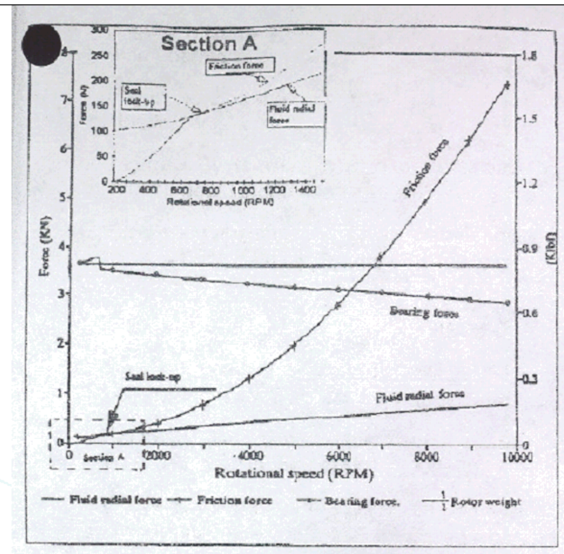


Figure 5. System forces Vs. rotational speed. Spring force =500 N, seal lapped area=0.0085m^2.

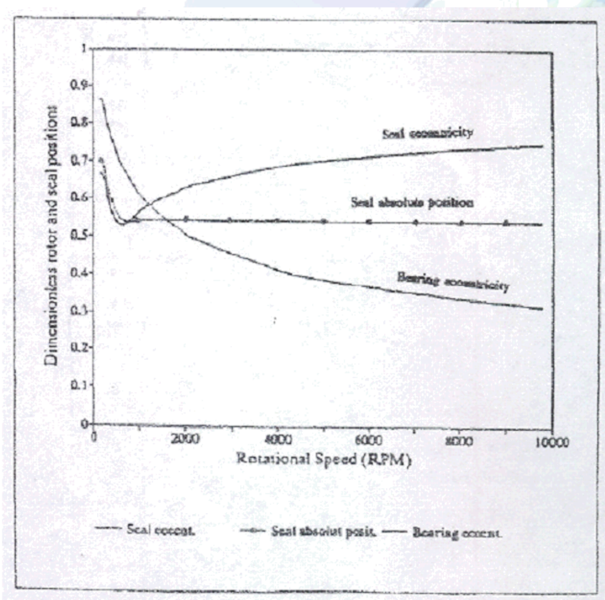
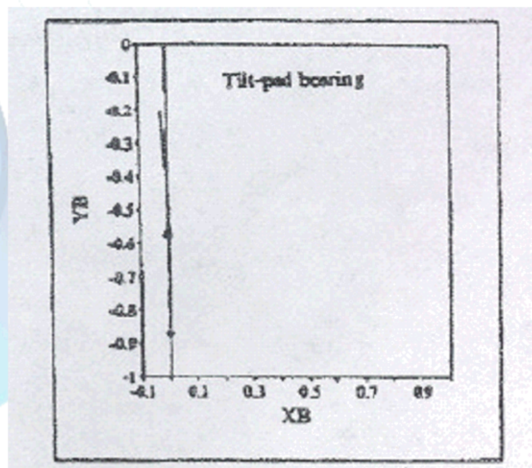


Figure 4. Oil seal and bearing relative positions. Spring force = 0. N, seal lapped area = 0.0085m^2.



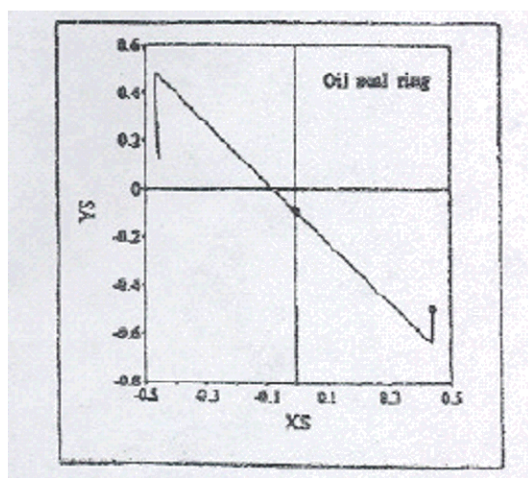


Figure 6. Tilt-pad bearing (XB,YB) and oil seal (XS,YS) loci. Spring force = 500. N,seallapped area=0.0085 m2.

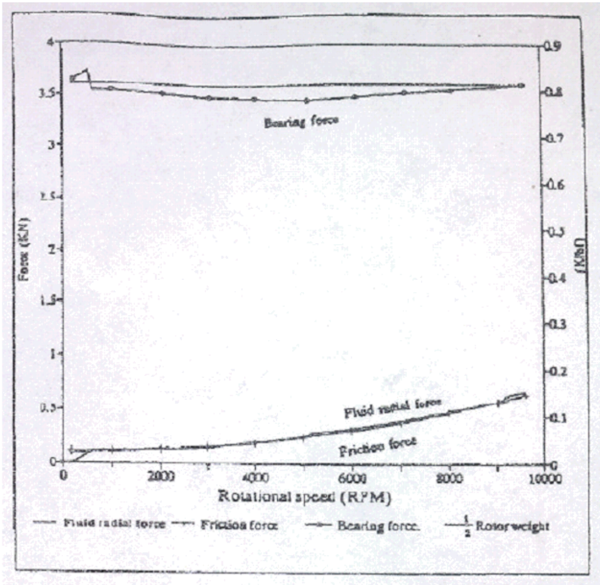


Figure 7. System forces vs. rotational speed. Spring force=500 N, seal lapped area=0.0006755m^2.

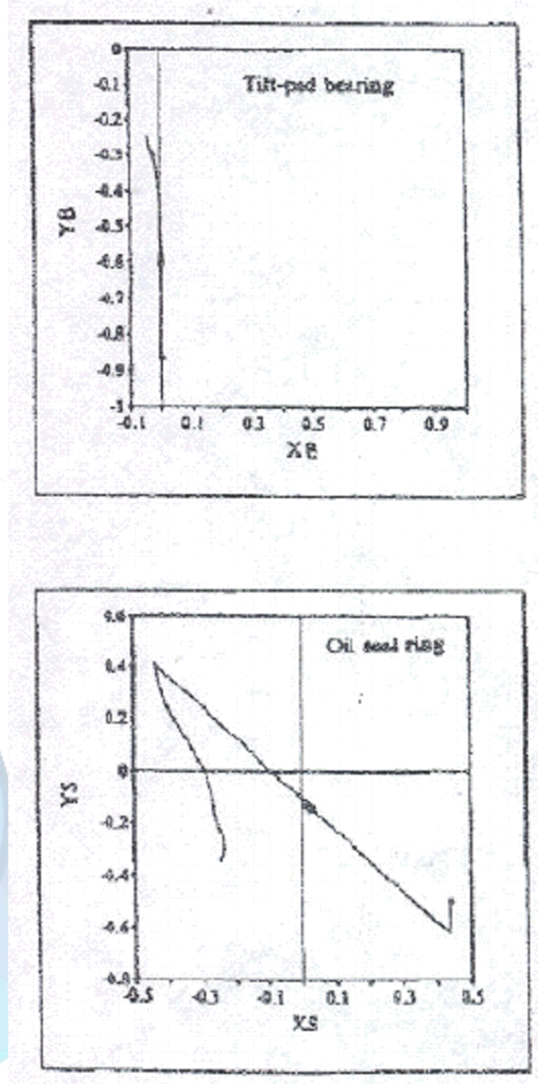


Figure 8. Tilt-pad bearing (XB,YB) and oil seal (XS,YS) loci. Spring force=500 N, seal lapped area=0.0006755m^2.