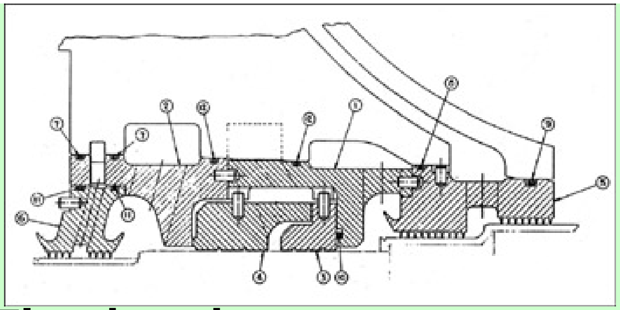
Modern Lubrication Theory FALL 2014 TAMU



Date:

Today's material

Notes 11a — **Applications of oil seals in turbomachinery Floating ring seals. Effect of eccentricity**

Notes 11b — **Applications of oil seals in turbomachinery Long seals**

Observations/Announcements

Read

Childs & Vance., 1997, TurbSymp, pp.201, 220.

Notes 11.
High Pressure Floating Ring Oil Seals

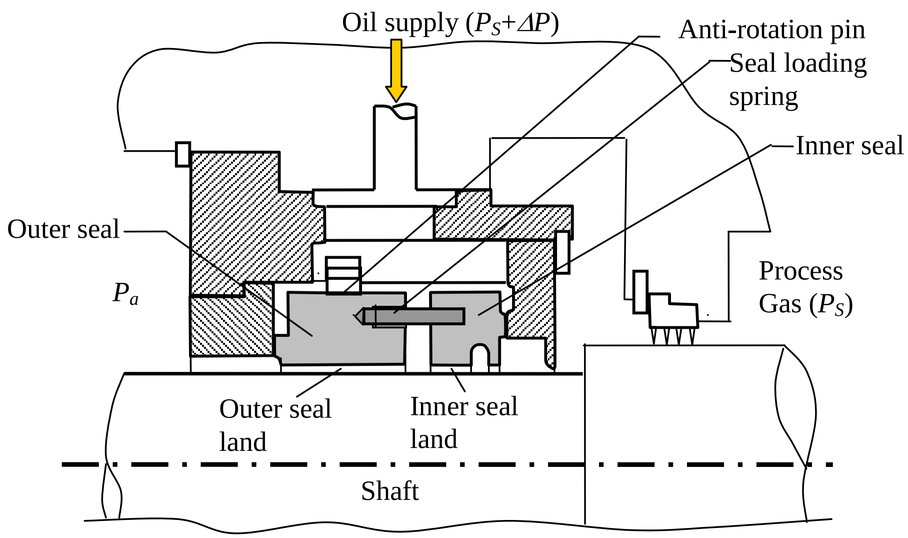


Fig. 1 Typical oil seal multi-ring assembly

Bushing oil seals and mechanical dry-gas (buffer) seals are the final sealing elements in compressors keeping the process gas within. Oil bushings, also known as floating ring seals, can have a major detrimental effect on the rotordynamic stability characteristics of compressors; and in some cases act as additional support bearings, i.e., they generate radial loads.

Oil seal rings **minimize** process product leakage while allowing a limited lubricant flow rate accompanied by a pressure drop. Oil seal rings are of the mechanical face type, with rotating and stationary faces, as well as with a carbon face in between the two.

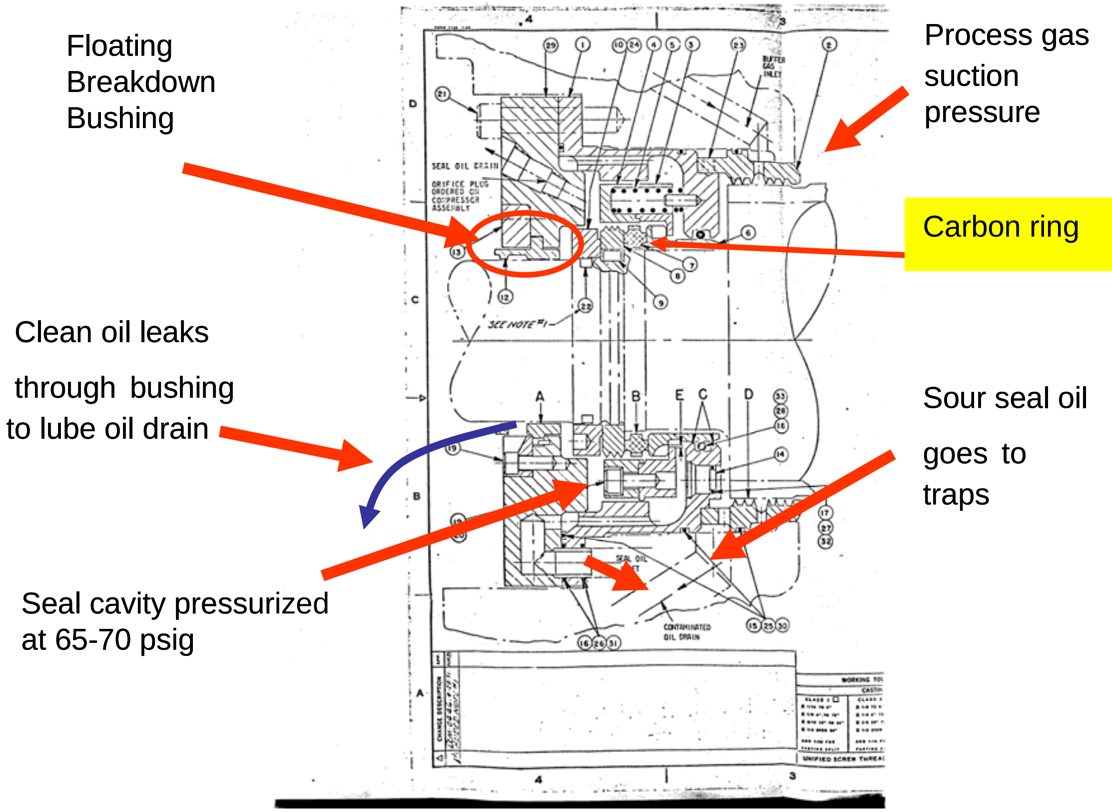
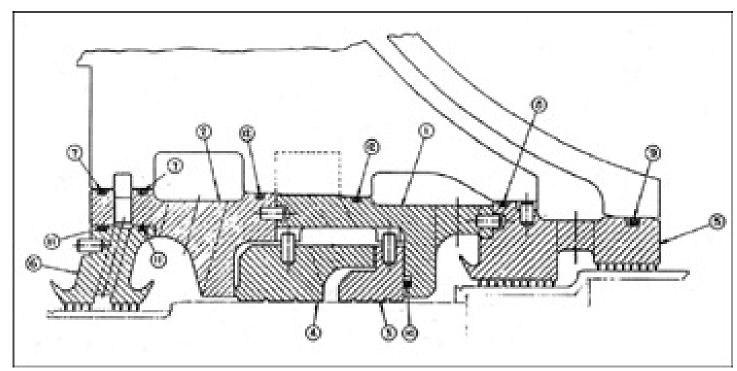


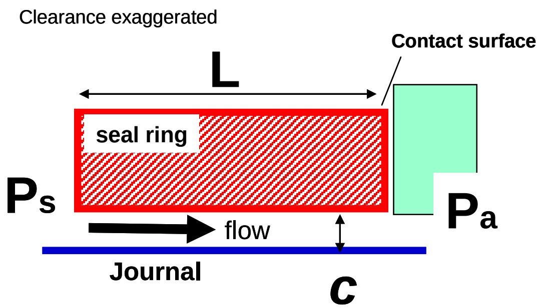
Fig 2. Oil seal ring cartridge in a compressor

Oil seal rings come as an assembly cartridge with a preload spring. The cartridge contains two seals (low pressure and high pressure) with small radial clearances. The seals operate with some mineral oil supplied at a pressure slightly higher than the (gas) sealing pressure.

The inner seal faces the process fluid, with lubricant flow (leakage) towards the process gas side, thus providing some degree of product contamination. The outer seal faces



the rotor support bearings, and is subject to a larger pressure drop, from supply towards atmospheric condition. The oil flow rate returns to the main oil reservoir or sump.



Smooth land seal ring

At low rotor speeds, an oil bushing acts as a floating ring and <u>follows the shaft motions</u>. The oil seal reaction force is small, equal to the contact area force (F_C), of Coulomb or dry-friction type. See Fig. 3 for a balance of forces acting on the seal ring. The contact force:

$$F_C = \mu_S N$$

where N is the normal (wall) force and μ_S is a friction coefficient, which depends on the materials of the seal face (typically an ISO carbon ring with a lapped surface) and the mating stationary casing. The balance of static forces in the axial direction gives the normal N force

$$N = F_S + \Delta P\, Area_{contact}$$

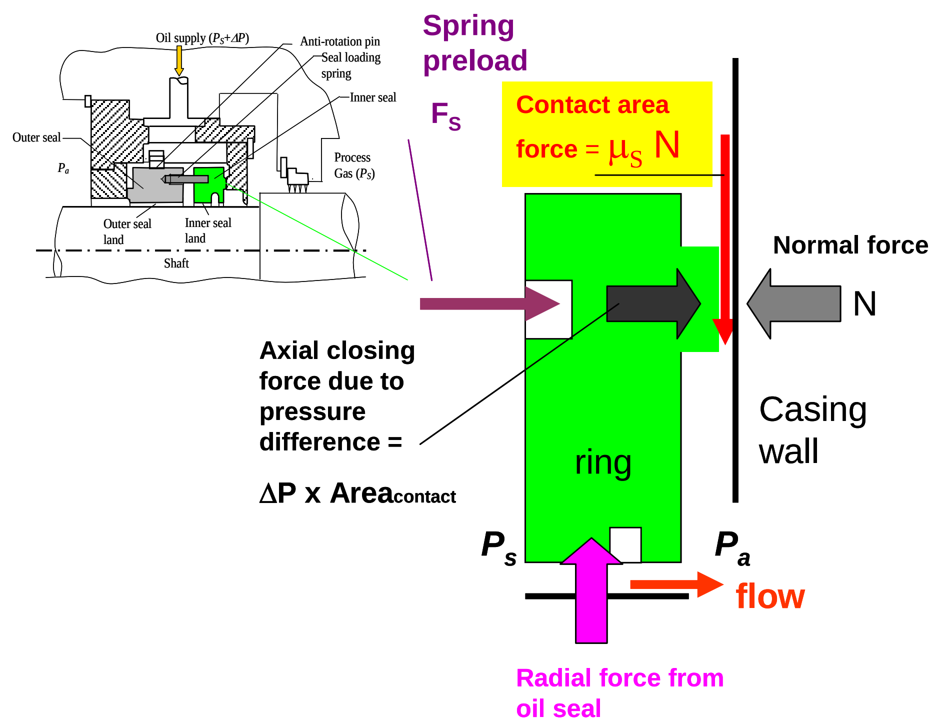


Fig 3. Forces acting on a floating ring seal

where F_S is the spring preload force and ΔP is the pressure drop across the contact face.

Note: The friction coefficient varies with time and with operation as the seal ages since the contact area wears out.

As the rotor speed increases so does the process pressure. The increase in pressure generates larger normal forces, and thus larger friction forces result at the seal-lap surface in contact with its mating stationary surface. Too large forces induced by the pressure differential eventually cause the seal to **lock up,** and thus the seal behaves as a hydrodynamic plain journal bearing. That is, the oil seal ring becomes a load path.

Engineering facts about floating ring oil seals

- The spring preload force prevents seal wear at low rotational speeds by impeding seal (carbon ring) displacements.
- The seal lapped (contact) surface area is a major factor in determination of the **lock-up speed** and the ensuing equilibrium seal off centered position (eccentricity).
- Seal operating eccentricities can be large, since to prevent lock-up, seals must develop film forces just equal or greater than the dry-friction force induced by the pressure drop across the seal.

- Oil seals, when locked at an off-centered position, can generate reaction forces of the same magnitude as the reaction forces from the tilting-pad bearings that support the whole rotor. At times seals may share rotor static load or weight with the support bearings; while at other times, oil seals can actually overload the primary bearings. See Figure 4 for a graphical description of the (possible) lock up conditions.

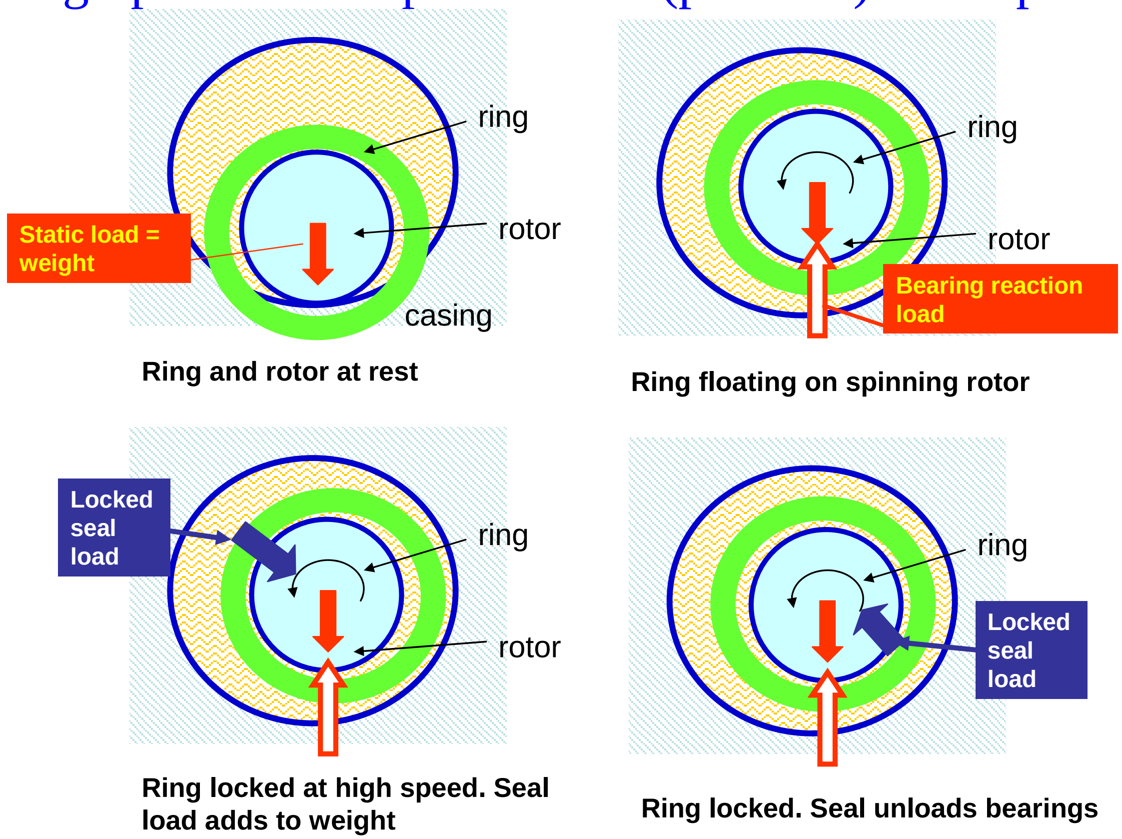


Figure 4. Operation of seal ring: rest, floating and locked

- Tilting pad journal bearings must be designed to prevent damage from bearing overload, overheating, deformation and wear produced by the oil seals.

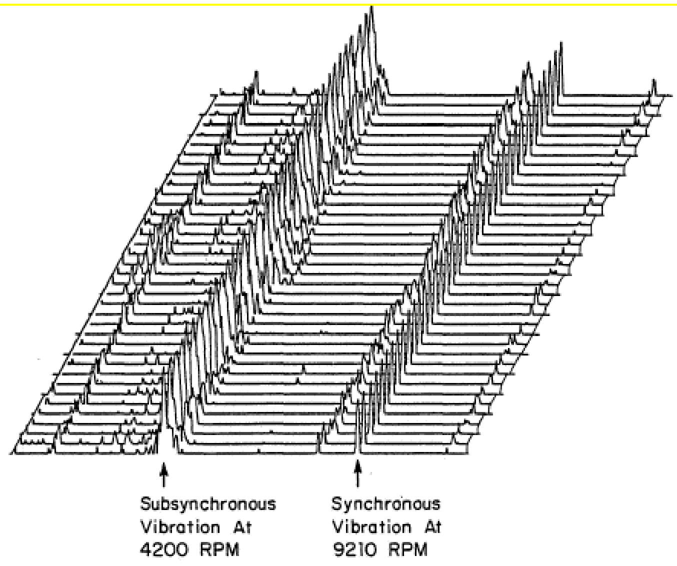


Figure 2. Vibration Pattern At 9210 RPM

Operation under a locked and off-centered condition causes the oil seal ring to generate significant levels of cross-coupled stiffnesses that could produce large amplitude rotor subsynchronous response and (severe) rotor stability problems (See inset figure taken from Allaire et al., 1985)

The most important consideration in oil seal bushings is to determine the rotor speed at which the oil seals lock-up and act as destabilizing elements on the dynamic response of the rotor-bearing system.

Recall that seal lock up occurs when the friction force at the seal lapped interface is larger than the oil seal film reaction force. The lock-up condition should occur at small seal ring eccentricities to reduce the magnitude of the seal reaction forces (and cross-coupled force coefficients).

Seal lockup known to promote rotor-bearing system instability is prevented by redesigning the seals to

a) reduce the cross-coupled force coefficients (K_{XY} and K_{YX}), while maintaining the same leakage rate,

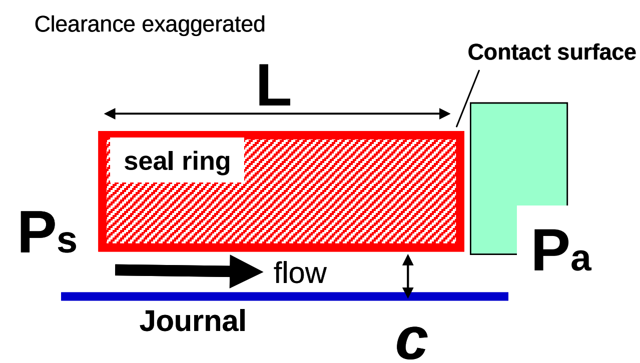
b) ensure concentric operation to avoid excessive radial forces,

c) reduce the contact dry-friction force ($F_C = \mu_S N$) induced by the pressure drop across the seal face. Since $N = F_s + AP\ Area_{contact}$
   - lower the friction coefficient μ_S
   - reduce the area of contact of the lapped surfaces
   - limits spring preload F_S

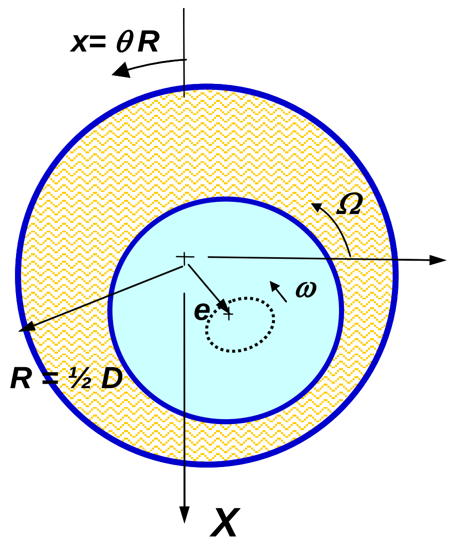
Most oil seals operate under laminar flow conditions and at very low flow Reynolds numbers (axial and circumferential). Therefore**, floating ring pressure seals show little direct stiffness** **(K_{XX}=K_{YY}=0)** even when locked at the concentric position (null eccentricity).

The flow rate (Q), cross-coupled stiffness (K_{XY}=-K_{YX})[1] and direct damping (C_{XX}=C_{YY}) coefficients are

$$Q = \frac{\pi D c^3}{12\mu}\frac{\Delta P}{L}; \quad K_{XY} = \frac{\pi \mu \Omega D L^3}{4c^3}; \quad \frac{K_{XY}}{\Omega C_{XX}} = 0.5$$



Smooth land seal ring

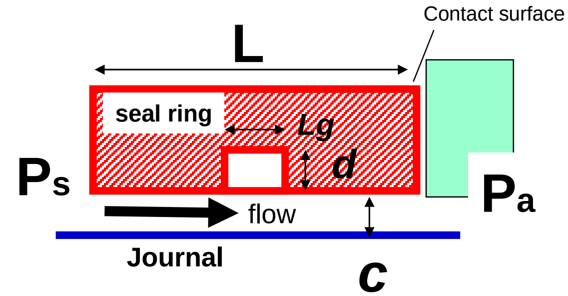


View of eccentric journal

where μ is the lubricant viscosity, (L, D, c) are the seal ring length, diameter and radial clearance, ΔP is the pressure differential across the seal, and Ω= RPM (π/30) is the rotor speed (rad/s). Note that the whirl frequency ratio of a centered (e=0) locked oil seal is 0.50 as for a plain cylindrical journal bearing [**WFR=$K_{XY}/\Omega C_{XX}$**]. Also note that the cross-coupled stiffnesses are proportional to the **third power** of the seal length (L).

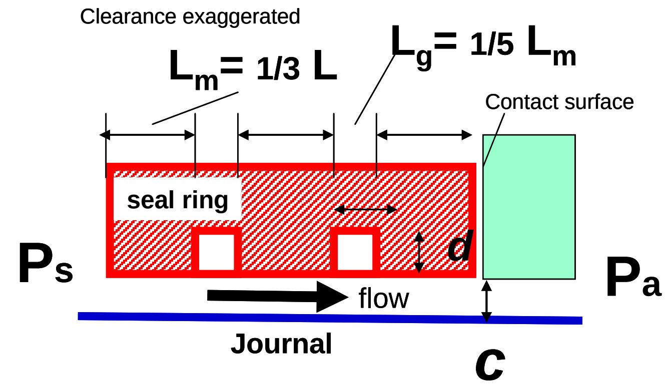
[1] These coefficients presume a full film (uncavitated oil) condition.

Allaire et al. (1985, 1987) recommend the substitution of long bushing seals with a series of short length sections separated by deep grooves. The grooved (multiple thin land) seal effectively divides the original seal land and reduces considerably the cross-coupled coefficients while preserving the same leakage rate.



grooved seal ring

For example, **modify** the seal above into one with three lands, each of length $L_m=L/3$, and separated by **deep & narrow grooves**, say of length $L_g= L_m/5$, then the flow rate and force coefficients are:



Three-groove seal ring

$$Q_m = \frac{\pi D c^3}{12\mu}\frac{\Delta P}{L};\quad K_{XY m} = 3\frac{\pi \mu \Omega D L_m^3}{4 c^3};\quad \text{WFR} = \frac{K_{XY_m}}{\Omega C_{XX_m}} = 0.5$$

i.e., the leakage rate is maintained, while the cross-coupled coefficients are reduced by nearly an order of magnitude since $3(L_m/L)^3=3\,(1/3)^3=1/9$. However, the direct damping coefficients are also reduced and thus, the whirl frequency ratio remains unchanged at 0.50.

Knowledge gained from analysis and practice

- Operation of truly floating rings at large eccentricities is not encouraged because when locked the seal will produce very large reaction forces.
- Note that the use of large clearance oil seals is not recommended due to excessive leakage rates of the sealing lubricant.
- Seals with multiple inner grooves (separating film lands) have consistently smaller load capacities, cross-coupled stiffnesses, and direct damping coefficients than smooth land (groove less) seals.
- Thermal effects (lubricant temperature rise and reduction in operating clearance) are typical in seals locked at high eccentricities. Mechanical energy dissipation is quite large.

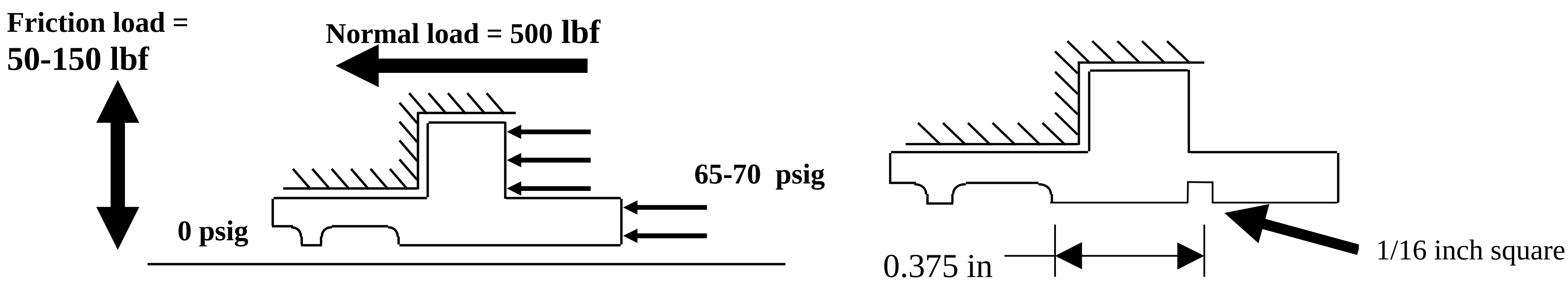
Example from a typical compressor (Ed Wilcox, Conoco, 1998)

In the following example, an oil bushing seal (L/D= 0.1875), can float 0.030 inch diametrically in the housing, but has only 3 mil axial travel. The bushing was originally designed with a very low diametrical clearance of 5 to 7 mil. Note that this clearance is less than the support fluid film bearing diametrical clearances, typically 6 to 8 mil.

The 65-70 psi oil pressure drop pressure exerts approximately 500 lb_f of normal force, pressing the bushing into the outer ring of the seal housing. With a contact surfaces dry friction coefficient μ_S =0.1 to 0. 3 (typical for smooth and rough or worn outstell surfaces), the contact force force F_C is ~ 50-150 lb_f . This force can readily "lock up" the bushing in an eccentric position. The radial load carrying capability of these seals is not enough to lift the rotor off the bearings. However, the bushings can affect the rotor stability in two additional ways:

a) Decreasing the radial load on the bearings would reduce the bearings stiffnesses and potentially cause the bearing to "whirl".
b) Act as an additional bearing support with a high cross-coupled stiffness

<u>Recommendation:</u> Cut a 1/16" square groove in the middle of the inside diameter of the bushing land. Increasing the clearance and cutting the groove in the bushing breaks up the hydrodynamic effect which produces the high cross-coupled stiffness. It also reduces the radial load capacity of the seal.

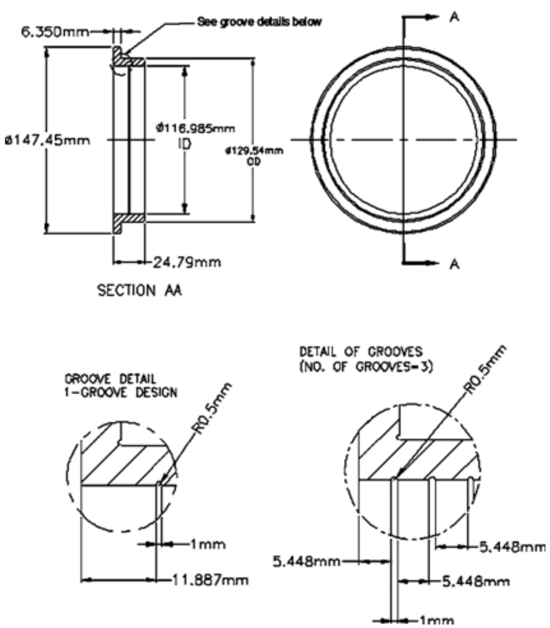


Cd = 0.005 inch (125 microns)
Axial float = 0.003 inch (76 microns)
Radial float = 0.030 inch

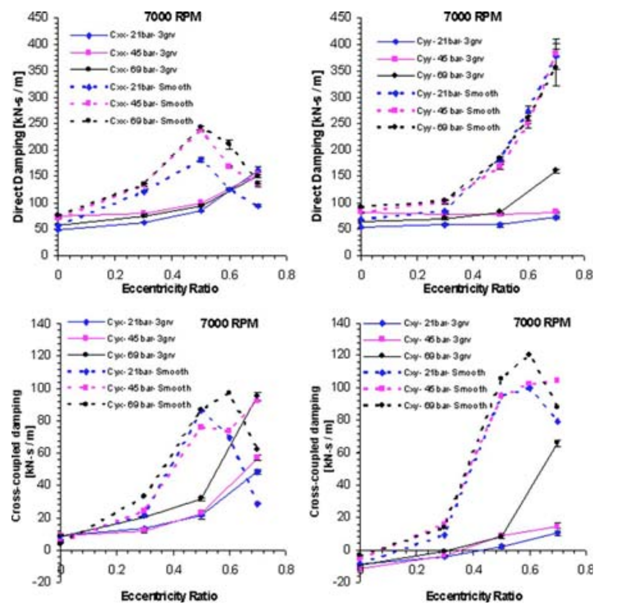
Original and modified oil seal bushings

CLOSURE

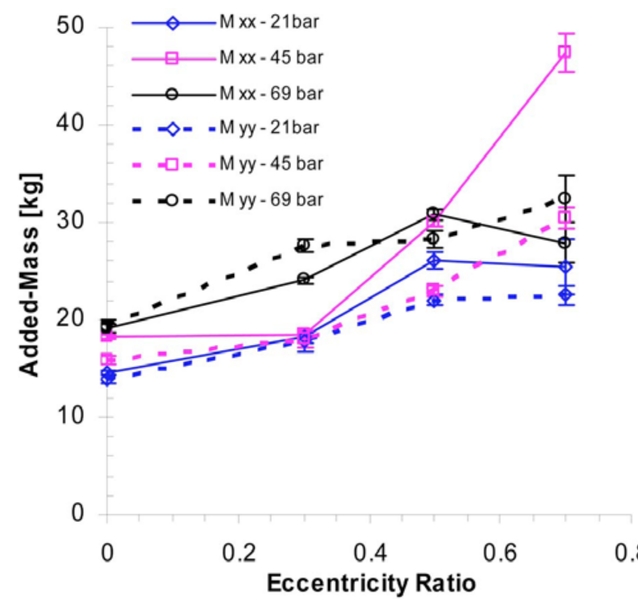
Test data from Childs et al. (2005-2007) show that narrow inner land grooves with depths as large as 15 times the thin land clearance **DO NOT** effectively reduce the oil seal cross-coupled stiffnesses. Most importantly, the tests also reveal very large added mass coefficients, much higher than predictions based on the classical formula of Reinhart and Lund (1975) for a smooth land seal.



Seals tested by Childs et al. (2006)



Damping coefficients for the smooth-3-groove seals at 7000 rpm from Childs et al. (2006)



Added-mass coefficients for 3-groove seal at 7000 rpm from Childs et al. (2006)

A simple formula for prediction of the added mass coefficients (M) in a cylindrical bearing or seal whirling around its centered condition is (Reinhart and Lund, 1978)

$$M_{XX} = M_{YY} = \pi \frac{\rho L}{c}\left(\frac{D^3}{8}\right)\left[1 - \frac{\tanh\left(L/D\right)}{\left(L/D\right)}\right]; \quad M_{XY} = M_{YX} = 0$$

where ρ is the fluid density. The formula is applicable to a full film condition (no liquid cavitation) in a smooth land (groove less) seal or bearing with length L, diameter D, and uniform radial clearance c.

For very long seals, $L/D >> 1$; $M_{XX} = M_{YY} \Rightarrow \pi \frac{\rho L}{c}\left(\frac{D^3}{8}\right)$ since $\left[1 - \frac{\tanh\left(L/D\right)}{\left(L/D\right)}\right] \rightarrow 1$

For short length seals, $L/D << 1$; $M_{XX} = M_{YY} \Rightarrow \pi \frac{\rho L^3 D}{24 c}$ since $\lim_{s \rightarrow 0}\left[1 - \frac{\tanh(s)}{(s)}\right] \rightarrow \frac{s^2}{3}$

Note that the mass of fluid within the seal thin annulus is just $M_f = \rho\left(\pi D L c\right)$

Hence, the ratio of added or apparent mass to the fluid annulus mass, for the long and short length seals, is

$$\frac{M_{XX}}{M_f} = \frac{\pi \frac{\rho L}{c}\left(\frac{D^3}{8}\right)}{\pi \rho L D c} = \frac{1}{2}\left(\frac{D}{2c}\right)^2 ; \quad \frac{M_{XX}}{M_f} = \frac{\pi \frac{\rho L^3}{c}\left(\frac{D}{24}\right)}{\pi \rho L D c} = \frac{1}{24}\left(\frac{L}{c}\right)^2$$

That is, the fluid inertia coefficient is orders of magnitude larger than the physical mass in the seal annulus.

In addition, note the mass of a solid journal is $M_J = \rho_J \left(\frac{\pi D^2}{4} L\right)$ where ρ_J is the journal material density (typically made of steel). Hence, for a long seal, the ratio

$$\frac{M_{XX}}{M_J} = \frac{\pi \frac{\rho L}{c}\left(\frac{D^3}{8}\right)}{\pi \rho_J L \frac{D^2}{4}} = \left(\frac{\rho}{\rho_J}\right)\left(\frac{D}{2c}\right)$$

shows that fluid inertia coefficients (apparent mass) could be larger than the (solid) journal mass.

Recent advances in flow analysis by Delgado and San Andrés (2010-12) introduce a novel model and show predictions that reproduce with great accuracy the unusual experimental results.

A presentation on the most recent developments in oil-seal analysis follows.

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