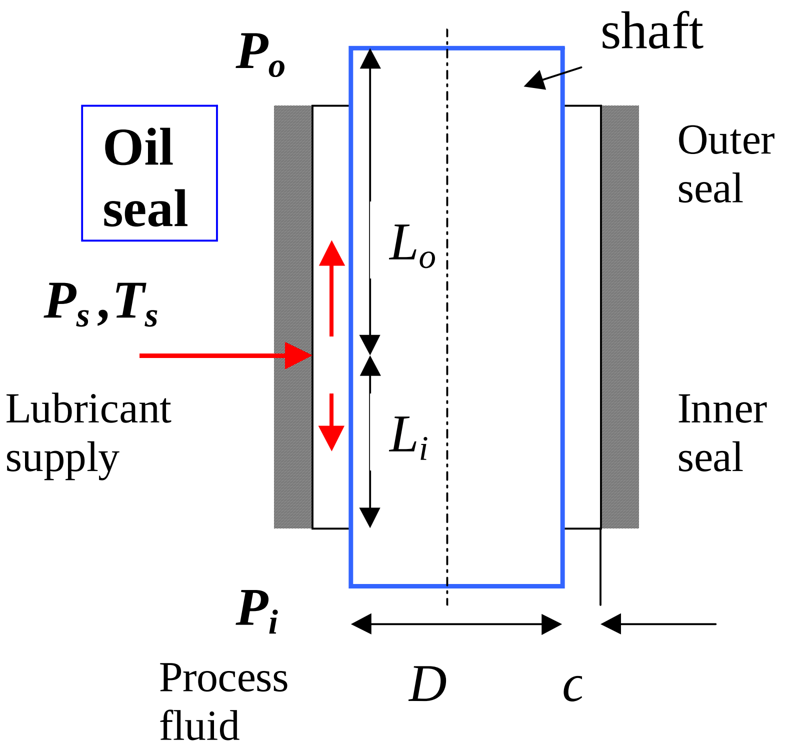
Notes 11.

High Pressure Long Oil Seals



Small clearance oil seals minimize process gas leakage while allowing a limited lubricant flow accompanied by a pressure drop. Annular seals are used in agitators and mixers, balance pistons in centrifugal pumps, etc.

Long annular seals (L/D>> 1), however, have a major impact on the stability of rotating machinery since they introduce large cross-coupled forces. The imposed axial pressure drops prevents liquid cavitation, and hence full film conditions prevail for mot operating conditions.

In practice, long seals integrate two sealing sides, each with a small clearance, and fed with some mineral lubricant at a pressure (P_s) slightly higher than the pressure (P_i) of the process product.

The **inner side seal** faces the process fluid and has a lubricant flow towards it, thus providing some degree of product contamination.

The **outer side seal** faces the support bearing (also oil lubricated) and has a larger pressure drop (usually) to atmospheric conditions (P_a) with a return to the main oil reservoir or sump.

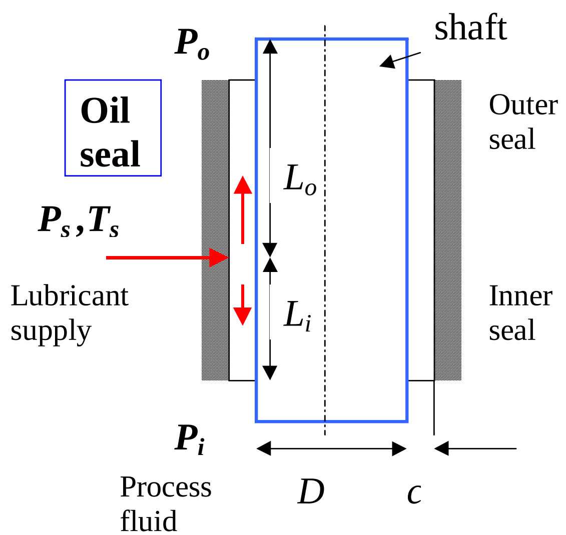
Typical sealing pressures can be as high as 30,000 psig (2,040 bar). In these applications, the lubricant viscosity depends greatly on the local pressure (oil viscosity can increase to three orders of magnitude its value at ambient pressure).

Oil seals have very small clearances (tighter than the support bearings) and **dissipate** large amounts of mechanical energy (**lubricant heats**) shear-induced by shaft rotation and by the extrusion of lubricant due to the pressure drop. In some cases, oil seal assemblies contain a **water-cooling jacket** to remove the heat generated and to provide an accurate control of the seal clearances. The large pressure drops also produce elastic deformations of the seal bushing and shaft with enlarged clearances on the sides of the high pressure condition.

The fluid **flow** in the seal is typically **laminar** since the (mineral oil) lubricant viscosity is large and the radial clearance is rather small. Operation without allowance for liquid cavitation makes the seal not able to generate direct stiffness coefficients, i.e. **the seal cannot support radial loads**.

SEAL MASS FLOW RATE

In a **centered** long seal, the mass flow rates towards the inner $(\dot{m}_i)$ and outer $(\dot{m}_o)$ sides of the seal are given by the simple formulas:



$$\dot{m} = \dot{m}_i + \dot{m}_o \ ; \ \dot{m}_o >> \dot{m}_i$$

$$\dot{m}_i = \frac{\pi D \rho_i c_i^3}{12 \mu_i} \frac{(P_s - P_i)}{L_i}; \qquad (1)$$

$$\dot{m}_o = \frac{\pi D \rho_o c_o^3}{12 \mu_o} \frac{(P_s - P_o)}{L_o}$$

where $(\rho,\mu)_{i.o}$ are the lubricant effective density and viscosity, $(L,D,c)_{i,o}$ are the seal axial length, diameter and radial clearance; and $(P_s\text{-}P_o) > (P_s\text{-}P_i)$ are the pressure differentials across the outer (o) and inner (i) sides of the seal.

Note that tight (small) seal clearances *(c)* have a pronounced effect on the seal leakage, i.e. $\dot{m} \sim c^3$. Furthermore, the **journal rotational speed *(Ω)*** has no effect on the seal leakage for **laminar flow** conditions.

SEAL POWER LOSS AND LUBRICANT TEMPERATURE RAISE

The **mechanical power loss** $\wp_S$ due to **shear** (drag of the lubricant by shaft rotation) is given by

$$\wp_s = \wp_{si} + \wp_{so} = 2\pi\Omega^2 \left(\left[\mu \left(\frac{D}{2}\right)^3 \frac{L}{c} \right]_i + \left[\mu \left(\frac{D}{2}\right)^3 \frac{L}{c} \right]_o \right) \qquad (2)$$

Also mechanical power $\wp_E$ is required to **extrude** (push) the lubricant through the thin film clearances, i.e.

$$\wp_e = \wp_{ei} + \wp_{eo} = \left(1 - \beta_A T_s\right)\left[F_i v_i + F_o v_o \right] \qquad (3)$$

where

$$F_i=\left(P_s-P_i\right)\left(\pi D c_i\right); \quad F_o=\left(P_s-P_o\right)\left(\pi D c_o\right) \tag{4}$$

are axial forces pushing fluid into the inner and outer sides of the seal,

$$v_i=\frac{\dot{m}_i}{\rho_i \pi D c_i}; \quad v_o=\frac{\dot{m}_o}{\rho_o \pi D c_o} \tag{5}$$

are the lubricant axial velocities on each side of the seal; and β_A is the oil volume-thermal expansion coefficient. Note that

$$F_i v_i=\left(P_s-P_i\right)\frac{\dot{m}_i}{\rho_i}; \quad F_o v_o=\left(P_s-P_o\right)\frac{\dot{m}_o}{\rho_o} \tag{6}$$

A portion of the total mechanical power (shear + extrusion), $\wp=\left(\wp_s+\wp_e\right)$ is conducted into the seal cartridge or housing and into shaft. The other portion is carried away (convected) by the lubricant, thus increasing its temperature.

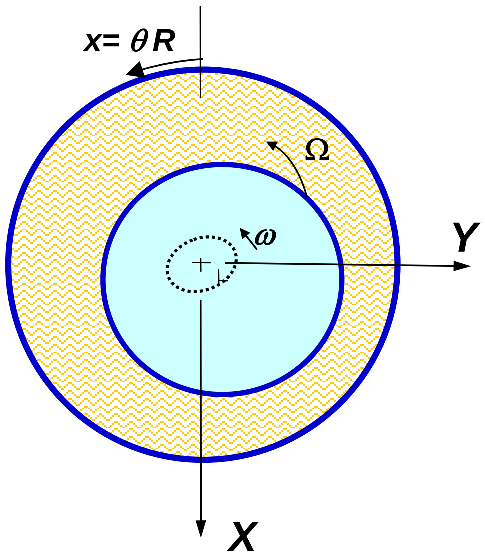
The **temperature rise *(ΔT)*** on the lubricant (on each side of the seal) is determined from the balance of heat transported by the lubricant, i.e.

$$\Delta T_i = \left(T_i - T\right)_s \sim \kappa_i \frac{\wp_i}{\dot{m}_i\, C_{p_i}}; \qquad \Delta T_o = \left(T_o - T_s\right) \sim \kappa_o \frac{\wp_o}{\dot{m}_o\, C_{p_o}} \tag{7}$$

where C_p is the lubricant specific heat and κ represents a fraction of the mechanical power convected into the fluid flow. Note that lubricant temperature rises can be large since the fluid flow rates are (by necessity) small. κ=0.8 - 1.0 when the seal walls are insulated. With a cooling jacket (typically water flow) then $\kappa << 1$.

LONG SEAL ROTORDYNAMIC FORCE COEFFICIENTS

For a centered seal (vertical configuration or unloaded condition), the direct and cross-coupled stiffness *(K_{XX}, K_{YY}* and *K_{XY}, K_{YX})* coefficients and damping *(C_{XX}, C_{YY}* and *C_{XY}, C_{YX})* coefficients are:



View of rotating journal whirling about housing center

$$\begin{aligned} K_{XX} &= K_{YY} = 0 \\ C_{XY} &= C_{YX} = 0 \\ C_{XX} &== C_{YY} = C_{XX_i} + C_{XX_o} \\ K_{XY} &= -K_{YX} = C_{XX}\frac{\Omega}{2} \end{aligned} \qquad (8)$$

where Ω= RPM(π/30) is the rotor speed in (rad/s). Note that the direct stiffnesses are zero and the **cross-coupled stiffnesses increase (linearly) with shaft speed Ω.** The **direct damping coefficients** from the inner and outer sides of the seal are

$$C_{XX_i} = 12\pi\ \mu_i L_i \left(\frac{D}{2c}\right)_i^3 \left[1 - \frac{\tanh\left(L_i/D_i\right)}{\left(L_i/D_i\right)}\right];\ \ C_{XX_o} = 12\pi\,\mu_o L_o \left(\frac{D}{2c}\right)_o^3 \left[1 - \frac{\tanh\left(L_o/D_o\right)}{\left(L_o/D_o\right)}\right] \qquad (9)$$

And the whirl frequency ratio is simply $WFR = \dfrac{K_{XY}}{\Omega C_{XX}} = 0.50$, i.e. identical to that of a conventional (unloaded) cylindrical journal bearing. Thus, **long annular seals are potential sources of rotordynamic instability!**

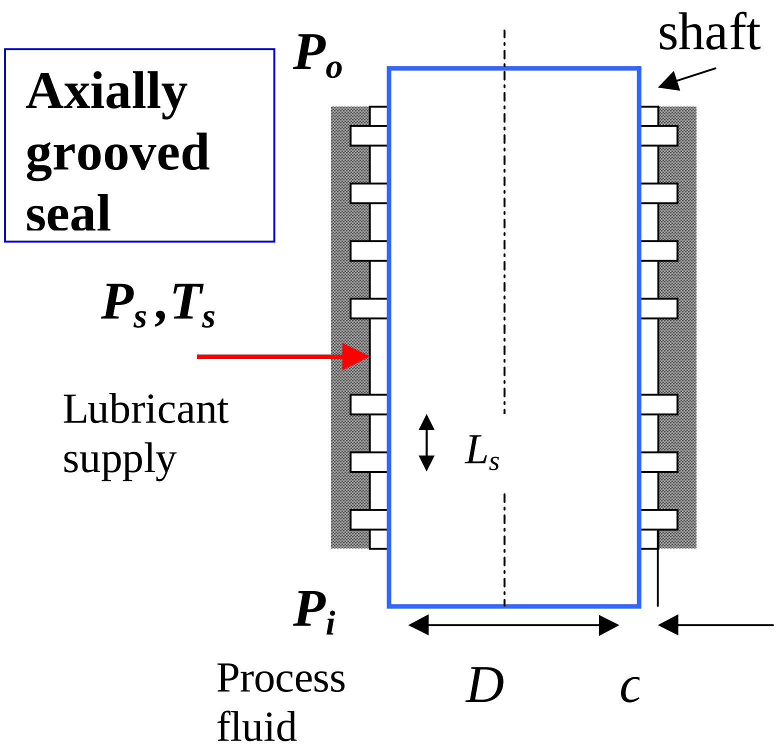
Note the cross-coupled stiffnesses are proportional to rotor speed *(Ω)* and to the **third power** of the seal diameter over clearance ratio $(D/2c)^3$.

In lieu of recent test data by Childs et al. (2006, 2007) that shows oil seal rings develop large added mass coefficients (smooth land and grooved configurations), formulas for **direct** added mass coefficients (M_{XX}=M_{YY}) are (Reinhardt, E., and Lund, J., 1975)

$$M_{XX_i} = \pi \frac{\rho_i L_i}{c_i}\left(\frac{D_i^3}{8}\right)\left[1-\frac{\tanh\left(L_i/D_i\right)}{\left(L_i/D_i\right)}\right];\quad M_{XX_o} = \pi \frac{\rho_o L_o}{c_o}\left(\frac{D_o^3}{8}\right)\left[1-\frac{\tanh\left(L_o/D_o\right)}{\left(L_o/D_o\right)}\right] \qquad (10)$$

for the inner and outer sides of the seal, respectively.

OIL SEALS WITH IMPROVED STABILITY

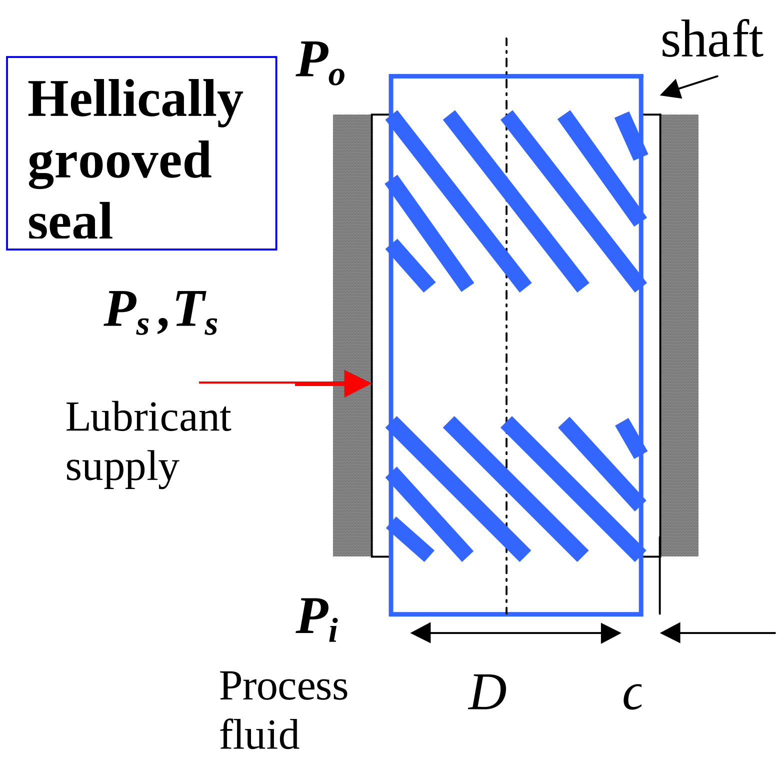


San Andrés (for Nova Corp., 1997) recommended the substitution of a long annular oil seal with a helically grooved seal to fix a (quite destructive) rotordynamic instability in a large size chemical mixer. The replacement was highly successful.

A seal with axial grooves divides the overall seal length (*L*) into *N*-small portions, each of length L_s (<<*D*). In this case the seal cross-coupled stiffnesses reduce to:

$$K_{XY} = -K_{YX} = \sum_{s=1}^{N} \frac{\pi\,\mu\,\Omega\,D\,L_s^3}{8\,c^3}$$

Note that the direct damping coefficients are also reduced and the whirl frequency ratio remains unchanged at WFR=0.50.



An engineered **helically grooved or spiral grooved seal** pumps the lubricant backwards (towards the high pressure zone), thus producing a more effective sealing with a larger film clearance and reducing substantially the lubricant temperature raise.

Furthermore, grooved seals provide direct stiffness (K_{XX}, K_{YY}) and reduce considerably the cross-coupled stiffnesses (K_{XY}, K_{YX}) while preserving the same leakage rate.

FURTHER NOTES

- Using large clearance seals is not recommended due to the excessive leakage rates of the sealing lubricant.

- Spirally grooved seals have consistently smaller load capacity, cross-coupled stiffnesses and direct damping coefficients than smooth land seals.
- Using a cooling jacket prevents overly large lubricant temperature rises and provides a way to control tight clearances.

References

Reinhardt, E., and Lund, J., 1975, “The Influence of Fluid Inertia on the Dynamic Properties of Journal Bearings,” ASME J. Lubr. Technol., April, pp. 159–165.

Childs, D. W. Rodriguez, L. E., Cullotta, V., Al-Ghasem, A., and Graviss, M., 2006, “Rotordynamic-Coefficients and Static (Equilibrium Loci and Leakage) Characteristics for Short, Laminar-Flow Annular Seals,” J. Tribol., **128**(2), pp. 378-387.

Childs, D. W., Graviss, M., and Rodriguez, L. E., 2007, “The Influence of Groove Size on the Static and Rotordynamic Characteristics of Short, Laminar-Flow Annular Seals,” ASME J. Tribol, **129**(2), 398-406

Zirkelback, N., and L. San Andrés, "Finite Element Analysis of Herringbone Grooved Journal Bearings: A Parametric Study,” ASME Journal of Tribology, Vol. 120, pp. 234-240, 1998 (ASME Paper 97-Trib-14).