Leaky Shaft Seals

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Scenario

Half of the gearboxes on a brand-new conveying system on the varnishing line for a manufacturer of high-end kitchen cabinets were leaking. Oil was dripping on the cabinet parts—ruining the finish. Why were half of the gearboxes leaking?

Background

I commonly hear people complain that shaft seals always leak. Many accept leaking seals as normal gearbox operation. But properly done, shaft seals do not leak, and will last for a long time. Remember when new cars—say 1970s and prior—always left a puddle of oil on the garage floor under the hood area? Like putting down newspaper for a new puppy, our family kept a sheet of cardboard on the ground under the engine to keep the garage clean. Cars don't do that anymore—their seals take years to wear out.

Improper shaft surface is the leading cause for shaft seals leaking prematurely; shaft surface factors are surface finish (i.e., roughness) and machine lead.

The most common shaft seal (Fig. 1) is spring-energized and typically referred to as a "lip seal." A garter spring adds light radial tension (energizes) to the rubber (elastomer) lip to assure good contact with the shaft. This article is based around application of lip seals, but works equally well for applications using O-rings and quad-rings sealing rotating shafts.

Surface Finish

If the shaft surface finish is too rough, it will prematurely abrade the lip. If extremely rough, the lip will not be able to conform to the surface and leak immediately. If the surface is too smooth the seal will overheat; this is because the lip rides on a thin film of lubricant. Fluid surface tension prevents it from escaping. A small amount of shaft roughness forms tiny lubrication pockets.

There are a number of recommended ranges of surface finish, depending on the seal manufacturer, industry association, or engineer. They span from 6 to 32 micro-inch R_a, with a common overlap from 10 to 20 micro-inch R_a. Personally, I specify 8 to 20 micro-inch R_a; it has worked well in applications as diverse as steel mill equipment to light-duty pumps.

Starting and stopping causes more wear than continuous operation. When running, hydrodynamics maintains the oil film between the seal lip and the shaft. When stopped, the oil film collapses. On start-up the lip seal rubs the shaft until the oil film is generated. This causes increased wear. During long, idle periods the seal elastomer begins to bond to the shaft surface. Anybody who has left something with rubber feet on a glass table for a few days has witnessed the effect. When this happens a small layer of rubber can be torn off the seal, causing leaks. Often, in the first few idle re-starts, the seal will wear back into sealing again. I have an antique car that experiences this problem every spring when I start using it again. Sitting in the garage, no leaks—even the seals below the oil level. When I back it out for its spring cleaning, it leaks in the driveway. After a long drive on the freeway the seals stop leaking. Within a few years I have to replace the seals as they will not stop. But if you look at both mileage and calendar days between having to replace them, it is a fraction of my daily vehicle.

Inspection showed all of the shafts had excessive lead. The mirrored arrangement of the conveyor meant half the gearboxes rotated clockwise, and half counter-clockwise. One rotation the shaft lead pumped the oil in, and the other rotation pumped it out.

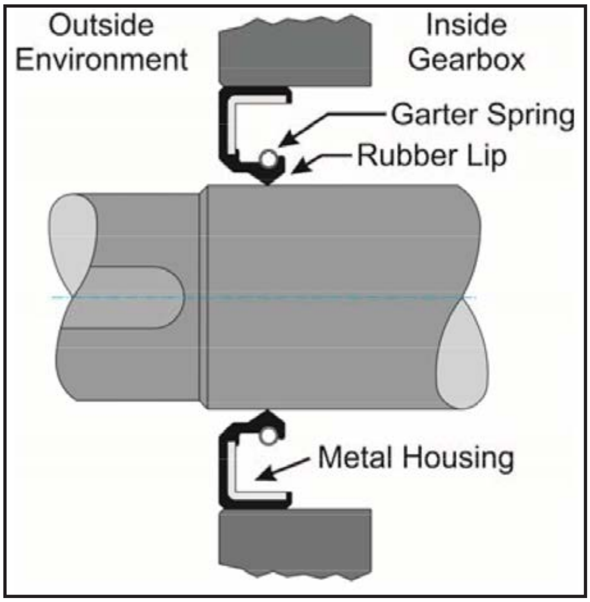


Figure 1 Typical elastomeric lip shaft seal.

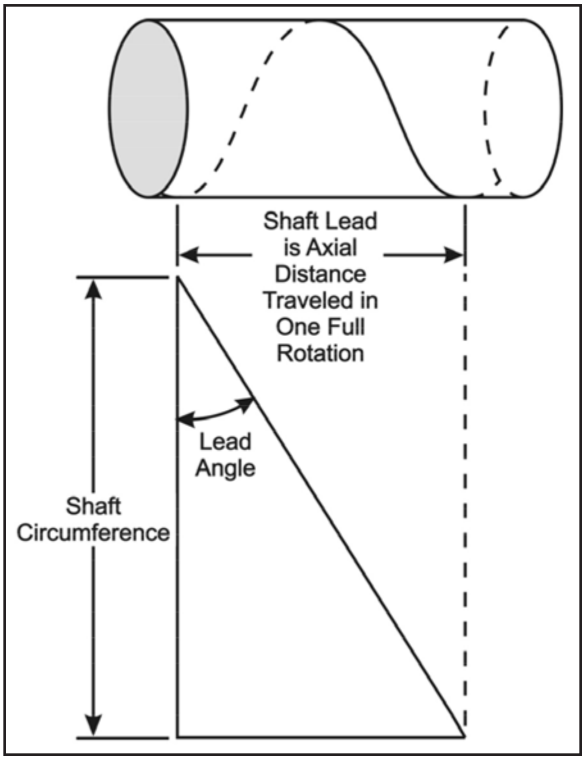


Figure 2 Shaft lead.

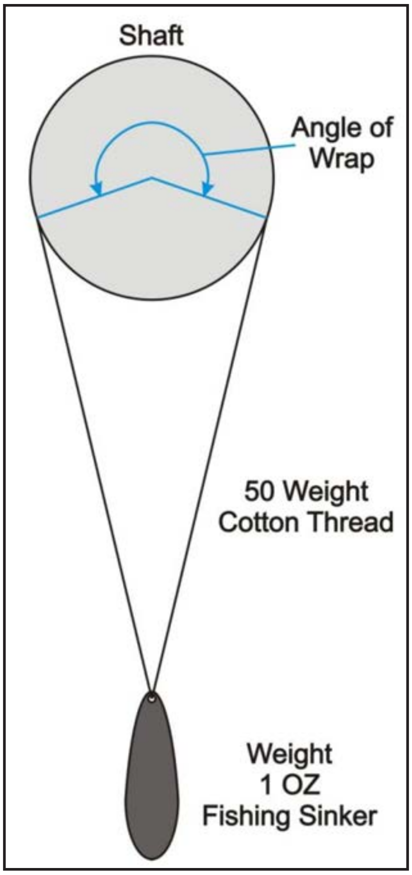


Figure 3 Lead test diagram.

Machine Lead

Manufacturing a shaft generally is done by 'turning'; removing material by rotating the part on its axis and moving a cutting tool axially. Turning leaves a helix pattern similar to a screw thread, commonly called machine lead or shaft lead (Fig. 2). Beyond the visible lead on the surface, the cutting tool also burnishes (*to move material by plastic deformation*) shaft in a helix. Subsequent finishing operations, ex. plunge grinding (*the grinding wheel is brought into contact with the shaft radially with no axial movement*), may not remove this burnished lead. The helix may remain even if the surface finish is correct and there is not a visible pattern.

The best test for shaft lead is not very high-tech, but it works. All of the special equipment that is not already in a typical machine shop can be purchased at a Wal-Mart.

A separate page of the test procedure was created so it can be given to an inspector to perform as a work instruction. At the end is the math (Eqs. 1 – 5), should you want to wade through it to better understand the theory behind the procedure. Seeing the math makes the procedure less like magic.

Theory. A loop of thread is used to act as a stylus that rides in the lead grooves (Fig. 3). A light weight is attached to the thread to hold it in contact and prevent it from rotating with the shaft. The shaft is coated with a thin film of oil and rotated at a consistent speed to mitigate problems with friction and inertia. By knowing the shaft rotation speed and the time it takes the thread to travel a distance, the lead angle can be determined (Fig. 4).

As the angle of wrap increases towards 360°, the tension in the thread and its grip on the shaft approach infinity. The thread length is sized so that the tension in the loop is kept low. The maximum wrap should be less than 255° and preferably less than 240°. Typically, less wrap (i.e., longer thread) is better.

The standard test speed is considered to be 60 rpm. However this is based on the diameters of standard shaft seals available, and does not account for extremes in shaft diameters. With large diameters the speed must be reduced because of the high surface speed. High surface speed will prematurely wear out the thread, or make it begin to travel with the shaft rotation, which is evidenced by the weight bouncing. With small diameter shafts the speed must be increased because the surface speed is too slow to mitigate problems with friction and inertia. This is evidenced by the weight swinging like a pendulum. The low rubbing velocity makes the coefficient of friction vary. Thus the weight should remain relatively stable in its motion.

This is not something where deviation from the rules will invalidate the test. Depending on your application you may be required to use trial and error to find the best combination of thread wrap, shaft speed, oil, and weight.

Every company where I implemented this test thought it was a joke—until they saw the results. At one company the VP of operations disliked the concept so much he spent thousand dollars on a sophisticated surface finish tester to prove it would predict shaft seal leaks, and the piece of thread would not.

He lost!

Residual machine lead test procedure

Equipment

- Stopwatch
- Cotton quilting thread (i.e., 50 weight cotton thread) or un-waxed dental floss; thread length 5 times the shaft diameter
- One-ounce fishing sinker
- 3-In-One machine oil (*a brand produced by WD-40 and generically is a severely hydro-treated, heavy naphthenic oil*), sewing machine oil or silicone oil viscosity 5 to 10 cps
- Lathe or other device to hold the shaft horizontally and rotate consistently at a set speed
- Distance indicator; piece of metal or cardboard to mark two lines a distance apart

Steps

1. Make a thread loop by knotting both ends at the weight
   a. Check that the free loop perimeter is 4 or more times the shaft diameter
2. Place the loop around the shaft
3. Mount the shaft in the lathe
4. Thinly coat the surface to be tested with oil
5. Position distance indicator near the shaft in the sealing area; mark 2 lines near the limits of the seal area and record the distance
6. Place the string to one end of the sealing surface outside of the indicators
7. Set the lathe rotation speed; nearest available speed to $w = 130/\text{diameter}$ in inches, or $w = 5/\text{diameter}$ in mm
8. Start the lathe; if the thread is moving away from the first position mark, reverse the lathe
9. Start the stopwatch when the thread reaches the first position mark
10. Stop the stopwatch when the thread reaches the second position mark; record the time
11. Check the thread for wear; replace as needed — every 50 to 100 tests

The Math

Producing Shafts for Successful Sealing

As harder surfaces are better for shaft seals, traditional practice for shaft calls for hardening and plunge-grinding the seal areas. Both are expensive operations. As previously noted, plunge-grinding may not resolve the problem with shaft lead. To not create the problem when turning the shaft, the feed rate can be set to the allowable for seal shaft lead. With old manual machines this was not very practical, as it would slow production. CNC machines can be programmed to reduce the feed rate just in the seal area. Roller burnishing the seal area is useful as it can improve the surface finish and the hardness.

For low volume I use traditional practice. With high volume applications, where I have the opportunity for production trials, turned and roller burnished seal areas offer large savings without sacrificing performance. **PTE**

D = Shaft Seal Diameter
X = Distance Between Position Indication Marks
t = Time to Travel X Distance (sec)
ω = Shaft Rotation Speed (rpm)
λ = Lead Angle

$$\lambda = \tan^{-1}\frac{60 \times X}{\pi \times D \times \omega \times t}$$

Equation 1 **Lead formula.**

Diagnosis

Observed Thread Movement	Interpretation
Stationary in both rotations	No lead present
Thread travels axially with rotation. Reversing rotation reverses axial travel.	Lead present Maximum Allowable Shaft Lead 0.05° (3.5 arc minutes)
Results where shaft lead cannot be determined	
Thread travels away from the center for both rotations	Crowned shaft; Barrel
Thread travels toward the center for both rotations	Cupped shaft; Hour Glass
Thread travels in same direction for both rotations	
Remount the shaft end-for-end and retest	
Reverses direction of thread movement	Tapered shaft
Does not reverse direction of thread movement	Machine holding shaft not level

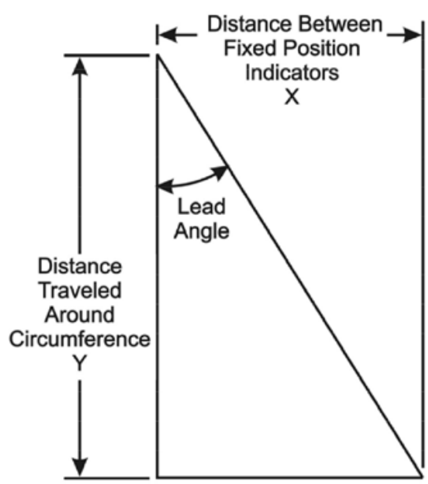


Figure 4 **Test diagnostics, interpretation.**

D = Shaft Seal Diameter
C = Shaft Circumference
Y = Distance Traveled Around Circumference
X = Distance Between Fixed Position Indicators
t = Time to Travel X Distance (sec)
ω = Shaft Rotation Speed (rpm)
n = Number of Shaft Rotations
λ = Lead Angle

$$C = \pi \times D$$
$$n = \frac{\omega \times t}{60}$$

60 converts w in rev/min and t in seconds

$$Y = C \times n \Rightarrow Y = \frac{\pi \times D \times \omega \times t}{60}$$

$$\tan\lambda = \frac{X}{Y} \text{ € } \lambda = \tan^{-1}\frac{X}{Y}$$
$$\lambda = \tan^{-1}\frac{60 \times X}{\pi \times D \times \omega \times t}$$

Equation 2 **Calculating lead angle.**

D = Shaft Diameter
C = Shaft Circumference
s = Surface Speed
ω = Shaft Rotation Speed (rpm)

$$C = \pi \times D$$
$$s = \frac{C \times \omega}{12} \Rightarrow s = \frac{\pi \times D \times \omega}{12}$$

12 Converts inches to feet

$$\omega = \frac{s \times 12}{\pi \times D}$$
$$s = 20 \text{ fpm} \Rightarrow \omega = \frac{240}{\pi \times D} \Rightarrow \omega = \frac{76.4}{D}$$
$$s = 50 \text{ fpm} \Rightarrow \omega = \frac{600}{\pi \times D} \Rightarrow \omega = \frac{191.0}{D}$$

Select mid-range to create a rule of thumb

$$\omega = \frac{130}{D} \text{ for } D \text{ in inches or } \omega = \frac{5}{D} \text{ for } D \text{ in mm}$$

Equation 3 **Calculating shaft rotation speed.**

General Concepts of Lubrication

Background

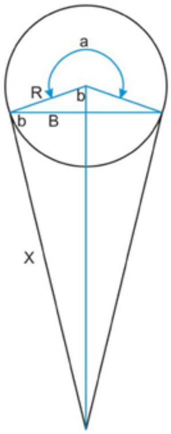
Lubricants simultaneously perform many functions. Most obviously they reduce friction. They also transfer heat, prevent surface contact, dampen vibrations, inhibit rust, and spread the contact load over larger area. Improper lubrication is the most common cause of reducer failure. This can be improper oil selection, property break down or contamination.

Elasto-Hydrodynamic Lubrication

When a speeding car hydroplanes on a rain puddle, the car's tires float on a wedge of water. In general engineering terms this phenomenon is called "elasto-hydrodynamic lubrication (EHL)" — also known as "thick film lubrication." For the components not to contact, the film thickness must be greater than the surface roughness of the contacting parts. Film thickness is proportional to the relative surface velocity and lubricant viscosity, and inversely proportional to the unit load. Meaning, viscosity is the oil's contributing factor to film thickness.

Viscosity and velocity are the predominant factors influencing the lubricant film thickness, whereas load has less importance. Analytical rela-

a= Angle of Wrap in Degrees
R= Shaft Radius
A= Thread Length in Cotact with Shaft
X= Length of Thread to Weight
T= Total Length of Thread (not including length knotted at weight)



$$T = A + 2 \times X$$

$$R = \frac{D}{2}$$

$$A = \pi \times D \times \frac{a}{360}$$

$$b = \frac{360 - a}{2} \Rightarrow b = 180 - \frac{a}{2}$$

$$B = R \times \sin b$$

$$B = X \times \cos b \Rightarrow X = \frac{B}{\cos b}$$

$$X = R \times \frac{\sin b}{\cos b} \Rightarrow X = \frac{D \times \tan b}{2}$$

$$\tan(180 - \theta) = -\tan\theta$$

$$X = \frac{-D \times \tan\frac{a}{2}}{2}$$

$$T = \pi \times D \times \frac{a}{360} + 2 \times \frac{-D \times \tan\frac{a}{2}}{2}$$

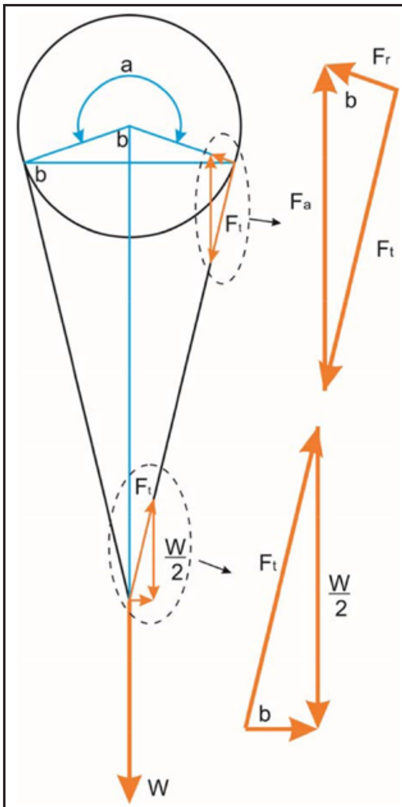
$$T = D \times \left[\pi \times \frac{a}{360} - \tan\frac{a}{2}\right]$$

$$a = 240° \Rightarrow T = D \times 3.83$$

5 times the shaft diameter easily gives enough extra thread length to tie the knot at the weight

Equation 4 Calculating thread length.

a= Angle of Wrap in Degrees
R= Shaft Radius
A= Thread Length in Cotact with Shaft
W= Weight Hanging in Thread
F_t= Tension Force in Thread
F_r= Radial Force on Shaft from Thread
F_a= Apparent Force



By Symetry, Force Applied to Each Thread Strand is Half the Total Weight

$$b = \frac{360 - a}{2} \Rightarrow b = 180 - \frac{a}{2}$$

$$\sin b = \frac{\left(\frac{W}{2}\right)}{F_t} \Rightarrow F_t = \frac{W}{2 \times \sin b}$$

$$\tan b = \frac{F_t}{F_r} \Rightarrow F_r = \frac{F_t}{\tan b}$$

$$F_r = \frac{W}{2 \times \sin b \times \tan b}$$

Trigonometric Identities

$$\frac{\sin\theta}{\cos\theta} = \tan\theta$$

$$\sin\theta \times \tan\theta = \sin\theta \times \frac{\sin\theta}{\cos\theta} = \frac{\sin^2\theta}{\cos\theta}$$

$$\sin(180 - \theta) = \sin\theta$$

$$\cos(180 - \theta) = -\cos\theta$$

$$\sin\frac{\theta}{2} = \pm\sqrt{\frac{1 - \cos\theta}{2}} \Rightarrow \sin^2\frac{\theta}{2} = \frac{1 - \cos\theta}{2}$$

Substituting into F_r

$$F_r = \frac{W}{2 \times \sin b \times \tan b} \Rightarrow F_r = \frac{W \times \cos b}{2 \times \sin^2 b} \Rightarrow F_r = \frac{-W \times \cos\frac{a}{2}}{2 \times \sin^2\frac{a}{2}} \Rightarrow F_r = \frac{-W \times \cos\frac{a}{2}}{1 - \cos a}$$

$$a = 202° \Rightarrow F_r \cong \frac{W}{10} \qquad a = 240° \Rightarrow F_r = \frac{W}{3} \qquad a = 283° \Rightarrow F_r \cong W$$

$$a = 207° \Rightarrow F_r \cong \frac{W}{8} \qquad a = 255° \Rightarrow F_r \cong \frac{W}{2} \qquad a = 320° \Rightarrow F_r \cong 4 \times W$$

$$a \to 360° \Rightarrow F_r \to \infty$$

Equation 5 Calculating thread grip.

tionships for calculating the minimum and the average film thickness have been developed. The equations are shown here (EQs. 1 and 2) are only for illustration of the variables' relative effects.

Boundary Lubrication

High loads can collapse the oil film, allowing the surfaces to contact. This is called "boundary lubrication." In this mode lubricity (slipperiness) and other properties of the lubricant become more important than the viscosity. Mixed lubrication, sometimes referred to as thin film lubrication, is where the oil film has not entirely collapsed, but the surfaces contact on the higher points of surface roughness.

Extreme pressure oils (EP oils) are a type of lubricant formulated to improve performance in boundary or mixed lubrication. At the high pressures and temperatures that occur in the contact area of gears and bearings, a chemical reaction forms a protective skin. Most EP oils use sulfur, phosphorus, and/or chlorine additives, and are designed to work in steel-on-steel applications. Note that care should be taken when using EP oils in applications using bronze components.

Synthetic lubricants are becoming very common; they reduce wear, increase efficiency, reduce friction, and lower sump temperatures — all of which increase component life. Their viscosity index is much higher than mineral oils. This allows one lubricant to provide adequate service over a broader temperature range. And they have longer service life, thus reducing the number of oil changes required. Efficiency increases of 20% of lost power are possible. Under severe conditions, properly selected synthetic oils are outstanding. Many companies have found cost advantages using the more expensive synthetic oil.

James K. Simonelli, inventor on multiple patents, is a Licensed Professional Engineer with over 30 years' experience designing and troubleshooting machine automation, heavy-duty equipment and industrial products. He has a broad background, with leading roles in engineering, project management, quality, business development, and division management in companies varying from start-ups and turnarounds to Fortune 100 corporations. Simonelli has served on committees developing industrial standards for the American Gear Manufacturers Association and the Hydraulics Institute. Mobile: (404) 702-3050; Email: *j.simonelli@att.net*; Skype: jim.simonelli; Linkedin: *www.linkedin.com/in/jsimonelli.*