Chapter 4

1. Lubrication

2. Wear
The science of friction, wear and lubrication is called tribology. The word is derived from the old Greek word “tribos”, which means “rubbing”.

As described in Chapter 1, the seal rings of a mechanical shaft seal rub against each other with a very thin lubricating film.

Tribology is a very old science. An old Egyptian inscription similar to fig. 4.1. shows how 172 slaves were able to pull a large statue on a sled.

Fig. 4.1: Assuming the slege is made of wood sliding against wood, it can be calculated that this only is possible when lubricated with water. The man standing on the sledge is pouring water under the sledge to lubricate, and three more slaves are bringing water to the tribologist.
1. Lubrication

The pressure distribution in the lubricating film is composed of a hydrostatic and a hydrodynamic contribution. The hydrostatic contribution arises due to the pressure difference between the pumped medium side and the atmospheric side. The hydrodynamic pressure is generated as a pumping action due to the sliding motion of the surfaces. The different lubrication regimes for hydrodynamic pressure are often described by means of a so-called Stribeck curve. See fig. 4.2.

At high velocities and not too high loads, the hydrodynamic pressure completely separates the sliding parts, allowing the formation of full-fluid-film lubrication. At lower velocities or higher loads, the hydrodynamic pressure is not sufficient to completely separate the sliding parts. In this situation, a mixed lubrication regime exists where part of the load is supported directly by the contact points of the surfaces.

The topography of the surfaces affects where the mixed lubrication regime is reached. At even lower velocities or higher loads, the generated hydrodynamic pressure becomes insignificant. This lubrication regime is called boundary lubrication.

The thickness of the lubricating film of the mechanical shaft seal must be very small to avoid excessive leakage. Consequently, the seal is always in the mixed- or boundary lubrication regime.
**Duty parameter**

The mechanical shaft seal lubrication number on the x-axis in fig. 4.2 is called the duty parameter, \( G \), defined by this formula:

\[
G = \frac{\eta \omega}{k \Delta p + p_s}
\]

- \( \eta \): dynamic viscosity
- \( \omega \): angular velocity (\( 2\pi \nu \) [sec\(^{-1}\)])
- \( k \): balancing ratio of the seal
- \( \Delta p \): pressure difference across the seal face
- \( p_s \): pressure in the seal gap caused by the spring.

More details about duty parameter can be found in [1].

For small duty parameter values, the leakage is very low and the seal operates in the boundary lubrication regime. For large duty parameter values, even full-film lubrication can be achieved.

**Examples:**
The duty parameter, \( G \), in boiler feed water is typically \( 10^{-9} \), in cold drinking water \( 10^{-7} \) and in crude oil \( 10^{-5} \).

The description of the lubrication regime with the duty parameter is not based on the calculation of a physical phenomenon, but more on empirical studies/methodes based on common practice. The following section provides further physically- based model descriptions of hydrodynamic lubrication.

**Hydrodynamic pressure distribution**

In the full-fluid-film lubrication regime, the friction between the surfaces with the relative velocity, \( v_0 \), is determined by the “internal friction” in the lubricating film. The shear resistance of a fluid is called “viscosity”, represented by the symbol, \( \eta \) (eta).

In the case of two flat surfaces, moving relative to each other at the velocity \( v_0 \) and separated by a fluid with the viscosity, \( \eta \), the molecules of the fluid normally adhere to the surfaces. Consequently, the velocity of the fluid near a surface is almost identical with the velocity of the surface. When the distance between the surfaces is small, the fluid flow is laminar (no turbulence). In this case, the velocity increases linearly between the two surfaces; the force, \( F \), required to keep the surface moving is proportional to the area of the surface, \( A \), and to the shear strain, \( v_0 / h \), where \( h \) is the distance between the surfaces. See fig. 4.3.
Thus, the shear stress $F/A$ is proportional to the change of shear strain, $v_0/h$:

$$F/A = \eta \frac{v_0}{h}$$

Or more generally with $\tau$ as shear stress:

$$\tau = \eta \frac{\partial v}{\partial h} \quad \text{(Newtonian fluids)}$$

In the case of parallel faces shown in fig. 4.3, the velocity distribution does not cause any pressure increase. If one of the surfaces is tilted slightly, the fluid will be forced into a smaller cross-section and therefore compressed. This will cause the pressure to increase and create a pressure distribution between the surfaces. See fig. 4.4.

For a given geometry, the pressure profile can be calculated using the Reynolds equation:

$$\frac{\partial}{\partial x} \left(h^3(x) \frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y} \left(h^3(x) \frac{\partial p}{\partial y}\right) = 6v \cdot \eta \frac{\partial h(x)}{\partial x}$$

The lubricating film calculated depends on velocity, $v_0$, and load, $F_N$. However, in all cases the pressure distribution generated between the surfaces will only be able to separate the surfaces by a distance comparable to the wedge height $(h_2 - h_1)$. See fig. 4.4.

More details about lubrication theory can be found in [2].
Waviness of seal rings
To minimise leakage, the surface of the mechanical shaft seal rings must be flat.
Consequently, no hydrodynamic pressure should be generated between the relatively rotating seal faces.
Flat seal rings are normally obtained by lapping. However, even very accurately machined surfaces are not completely flat. Some surface waviness of the order of 1/10000 mm always persist. When there is a relative rotation between seal rings, the small waviness generates a hydrodynamic pressure. This pressure increases the lubricating film thickness, resulting in a higher leakage rate. See fig. 4.5.

Waviness also appears as a result of mechanical and thermal distortion, but in most cases the resulting hydrodynamic pressure is not sufficient to completely separate the surfaces.
The effect of waviness on the hydrodynamic pressure distribution is further discussed in [3]. The conclusion is that the safest compromise between lubrication and leakage is obtained by lapping the surface as flat as possible.

Hydrodynamic tracks
In shaft seals for very low-viscosity fluids like hot water and gases, the hydrodynamic lubrication can be increased by making tracks in the seal ring or seat. See figures 4.6 and 4.7.
By thermal distortion, a wedge is created on the seal face near the tracks. See fig. 4.6. This type of tracks in the seal face pushes the evaporation zone closer to the atmospheric side of the seal [4]. Following each track, an area with increased pressure is created. This design allows the pumped medium to enter the seal gap very easily; a sealing zone still remains at the atmospheric side of the seal.

A more efficient way of increasing hydrodynamic pressure is to machine small grooves in the seal face, making a wedge into the seal gap. This design is common in gas seals where a hydrodynamic pressure is desired even with an extremely low viscosity. See fig. 4.7.

**Roughness of seal rings**

Friction and wear depend on the actual area of contact and therefore on the surface topography. Roughness parameters such as $R_a$ values, indicate the average size of the roughness but not the shape of the topography. To describe the friction, wear and lubrication (tribological) properties of surfaces, the “bearing area curve” (BAC) is more suitable. The BAC describes the contact area with an imaginary plane as a function of the distance. This plane is pulled down in the surface, see fig. 4.8. The desired area in a certain depth is called the “relative material ratio” (Rmr) value in the relevant depth.

Fig. 4.8 shows a bearing surface Rmr of 5 %, 40 % and 80 % for different depths. The percentages are calculated as the thick line in percentages of the total length.

![Fig. 4.8: Cross-section of surface showing how a BAC is obtained](image1)

Different machining processes normally provide different BACs. See fig. 4.9.

![Fig. 4.9: Examples of BACs for a grinded and a lapped surface](image2)
The lapped surface has a plateau with some valleys. Consequently, the bearing area rapidly increases with the depth until a large area has been reached. As opposed to the lapped surface, the area of the grinded surface is slowly increasing with depths indicating a more even distribution of valleys and peaks.

Fig. 4.10 shows how the leakage rate differs according to the direction of the scratches on the surface. The arrows indicate the direction of rotation of the seal ring. According to fig. 4.10, the lubricating film can be pumped to the pumped medium side or to the atmospheric side, depending on the direction of the scratches on the surface.

The typical surface topography of seal rings is a statistic distribution of scratches in all directions obtained by means of a lapping process. A shiny surface with a small roughness can be produced by lapping. However, where both seal rings are made of hard materials, one of the seal rings should have a dull finish to prevent the seal rings from sticking together during standstill. For a dull surface finish lapped to an $R_a$ value of 0.2 the running-in period may last several days.
Surface texturing

Surface texturing is one way of increasing the lubricating film in a seal running above the boiling point of the liquid (for example water above 100 °C). No significant increase of leakage will take place when the seal runs below the boiling point, [7]. Pockets in the seal faces are filled with the pumped medium and therefore act as an extra reservoir, preventing the lubricating film from evaporating completely. Surface texturing can be achieved by laser machining or etching.

If the seal rings are made of a material with closed pores, the seal faces appear as a textured surface. The advantage of porous seal rings is that the surface remains textured even when the seal rings are worn.

Hydrostatic lubrication

As described in Chapter 1, see the figures on page 14, hydrostatic pressure has a linear decrease through the seal gap with parallel faces, whereas the decrease is non-linear with a diverging or converging seal gap. See figures 1.21 and 1.22, page 19.

Evaporation of the liquid in the seal gap also affects the pressure distribution. Vapour has a much lower viscosity than liquid, and therefore the evaporated liquid quickly escapes from the seal gap. On the other hand, the density of the evaporated liquid is much lower than the density of the liquid, which means that the volume increases by vaporisation.

Thus vaporisation can increase the hydrostatic pressure above the linear decrease and push the evaporation zone closer to the atmospheric side. See fig. 4.11.

Calculations of the hydrostatic pressure distribution in seal gaps with evaporation can be seen in [5].

Fig 4.11: Evaporation in the seal gap can increase the pressure in the gap because the pumped medium expands when it evaporates.
Dry running
Mechanical shaft seals for liquids must be lubricated and cooled by a liquid. The shaft seal will be damaged if it is allowed to run without a liquid. In the absence of a lubricating film in the seal gap, frictional heat is dissipated in the seal rings. The frictional heat causes the temperature of the seal rings to increase up to several hundred degrees Celsius after few minutes of dry running.
The high temperature damages the elastomeric secondary seals.
The temperature reached and the time it takes to reach this temperature depend to a large extent of the materials of the seal rings and the design of the seal. Shaft seals with one carbon seal ring might be capable of running dry for several hours without severe damage to the seal components.

2. Wear
Wear is an undesired removal of material from a surface.
A number of processes may lead to wear of a surface. These processes are categorised into four common types of wear [6]:

- Adhesive wear
- Abrasive wear
- Corrosive wear
- Fatigue wear.

Adhesive wear
Even macroscopic smooth surfaces are rough on an atomic scale. When such two surfaces are brought together, contact is made at relatively few isolated asperities. When a normal load is applied, the local pressure at the asperities becomes extremely high. In the absence of lubricating films on the surface, the surfaces adhere to each other. However, very small amounts of contaminants prevent adhesion. See fig. 4.12.
Tangential motion of one surface relative to another might cause the surface film to disperse at the point of contact; cold welding of the junction can take place.

Continued sliding causes the junctions to be sheared and new junctions to be formed. This is the adhesive wear process.
The surface topography is very important to prevent adhesive wear as it determines the contact stress in asperities.
The materials chosen for seal rings should not be easily welded together to prevent adhesive wear.
**Abrasive wear**

Abrasive wear or abrasion is the ploughing of a tip from one material in another material. When asperities in one surface remove material in another surface, it is a two-body abrasion. The result of this wear is a surface with regular grooves. The abrasion process is called erosion, when a particle impinges the surface and the kinetic energy of the particle is used to remove material from the surface. In this case, a more random surface similar to a grit blasting can be observed.

Hard particles trapped between two sliding surfaces may cause severe damage. This is called three-body abrasion. See fig. 4.13.

![Three-body abrasion](image)

**Fig. 4.13: Three-body abrasion**

Three-body abrasion also appears as regular grooves in the surface. See fig. 5.12, page 84.

The resistance against abrasion depends on the ductility of the surface as well as the hardness of the surface compared to the hardness of the tip causing the abrasion on. The more ductile the surface, the more tendency to plastic deformation instead of debris removal, this results in less wear.

**Corrosive wear**

When surfaces rub against each other in corrosive environments, reaction products may be formed on the surface. These products often have a low adherence to the surface and can be removed by the rubbing and eventually cause abrasive damage. Corrosive wear can be observed on shaft seals with hard seal faces in a corrosive medium. This can be due to corrosion of a binder phase releasing hard grains from the material.

**Fatigue wear**

Surfaces repeatedly subjected to large stresses might wear on account of fatigue. Stresses can be caused by the mechanical load which is typical for roller bearings. The largest stresses to seal rings are caused by thermal gradients produced by frictional heat and evaporation.

Wear on SiC faces may occur in hot water. It may look like abrasive wear because SiC grains are pulled out due to thermal fatigue of SiC. The grains pulled out cause abrasion of the seal faces, leaving the impression that abrasive wear is the cause of the wear. This type of wear is only seen above the pressure and temperature limit for stable friction. The thermal fatigue of SiC may be a complex process involving evaporation, cavitations and corrosion.
Summary
This section describes how pressure can be established in the lubricating film and how different mechanisms can lead to wear. Pressure in the lubricating film is increased when there is a wedge in the seal face in the direction of the moving faces or if evaporation occurs between the seal faces. A wedge will appear with seal ring waviness, hydrodynamic tracks or with surface texture. The most common types of wear are: adhesive wear with seal faces sticking together, abrasive wear due to ploughing, corrosive or fatigue wear, or a combination of these wear types.