Mechanical shaft seals for pumps
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Preface

Technology and using technology in our products is the very core of Grundfos’ success. It has been like that since the start of Grundfos, and this is also how it is going to continue in future.

But this position doesn’t just come to us, and many of our colleagues in the pump business would be happy to take over this position. However, this is not going to happen – as we at Grundfos want to continue our tradition for long-range technology and material development.

For most pumps a decisive element for the quality of the pump during its lifetime is a good and robust shaft seal. Grundfos has many years of experience with the development, production and use of mechanical shaft seals in pumps, and our solutions in this field are contributing significantly to our leading position within pump technology.

I am pleased to introduce this book which I encourage you to use in our organisation. Looking ahead and working together, it is important that we systematically apply the knowledge which we have gained, and which has now been set down in writing in this book.

Enjoy the reading!

Carsten Bjerg
Group President
Chapter 1

Introduction

1. Types of shaft seals
2. Mechanical shaft seals
3. Operating principle
4. Historical development
1. Types of shaft seals

Almost everywhere where pumps with rotating shafts are used, a shaft seal is involved. The shaft seal forms a barrier between what is inside the pump and the atmosphere.

A pump with a through-shaft is not completely sealed. It is a challenge to the entire pump industry to minimise leakage.

There are countless variants of shaft seals, reflecting the diversity of the pump industry, and the need for specific solutions for individual situations. In its most basic form, a shaft seal combines a rotating part with a stationary part. When properly designed and installed, the rotating part rides on a lubricating film, only 0.00025 mm in thickness. Should the film become too thick, the pumped medium will leak. If the film becomes too thin, the friction loss increases and the contact surfaces overheat, triggering seal failure.

Seal performance greatly influences pump performance. When functioning correctly, the seal remains unnoticed. As soon as it starts to leak, however, significant problems can arise, either with the pump or the surrounding environment. The importance of the shaft seal must never be underestimated during pump design, operation, or maintenance.
Stuffing box
A braided stuffing box packing is the simplest type of shaft seal. The packing is placed between the shaft and the pump housing. See fig. 1.2.
In the stuffing box housing used in fig. 1.2, a soft packing ring is axially compressed until it makes contact with the shaft. After the soft packing has been exposed to wear, the stuffing box must be further compressed to prevent excessive leakage.

Vibrations and misalignment will cause this seal type to leak.

Lip seal
A universal lip seal type is a rubber ring sliding against the shaft. See fig. 1.3. This type of seal is primarily used in connection with a low differential pressure and low operating speed.

Mechanical shaft seal
A mechanical shaft seal consists of two main components: a rotating part and a stationary part. See fig. 1.4. The rotating part is axially pressed against the stationary part.
In the following, we shall focus on the mechanical shaft seal and its many construction possibilities and applications.
2. Mechanical shaft seals

This section briefly describes the design and elements of the mechanical shaft seal.

As previously stated, a pump with a through-shaft is not leakproof. The mechanical shaft seal is essentially a throttle arranged around the shaft. It reduces leakage between the pump and the surroundings to an absolute minimum. The clearance between the stationary and rotating part of the seal must be small in order to reduce leakage.

Mechanical shaft seal with two axial seal faces
The best possible way of making a seal with a minimum of clearance and thus a minimum amount of leakage is by pressing two axial surfaces against each other. These axial surfaces can be obtained with a stepped shaft, running against a flat surface on the pump housing. See fig. 1.5.

The shaft and pump housing must be highly wear resistant and well aligned.

Mechanical shaft seal with rotating seal ring and stationary seat
A more practical solution is obtained by fitting a rotating seal ring on the shaft and a stationary seal ring (seat) in the pump housing. The tiny space between the seal faces is called the seal gap. See fig. 1.6.

This design allows the use of a wide selection of materials for the rotating seal ring and stationary seat.
Secondary seals
Secondary seals consist of rubber parts such as O-rings or bellows, used to avoid leakage between the shaft and the rotating seal ring as well as between the stationary seat and the pump housing.

To minimise leakage, the rotating seal ring must be pressed against the seat. Therefore the rotating seal ring must be able to move axially on the shaft. To obtain axial flexibility, the secondary seal must either be a bellows or an O-ring sliding on the shaft.

The secondary seal that seals between the rotating seal ring and the shaft rotates together with the shaft. The secondary seal that seals between seat and pump housing is static. See fig. 1.7.

Spring
The rotating spring presses the rotating seal ring against the seat and the rotating O-ring along the shaft. See fig. 1.8.

Torque transmission element
A torque transmission element ensures that the rotating seal ring rotates together with the shaft. See fig. 1.9.

All components of a complete mechanical shaft seal have now been introduced.
3. Operating principle

This section describes how the lubricating film is generated in the sealing gap in a liquid-lubricated mechanical bellows shaft seal. The design differs slightly from the O-ring seal shown in fig. 1.9.

In its simplest form, the mechanical shaft seal consists of two main parts:
- The rotating part
- The stationary part

See fig. 1.10.
The rotating part
The rotating part of the seal is fixed on the pump shaft and rotates in the liquid during pump operation.

The compression of the rubber bellows (8) between the shaft (9) and one of the two torque transmission rings (7) fixes the rotating part to the shaft. See fig. 1.10.

The spring (6) transfers the torque between the torque transmission rings (7 and 5). The rotating seal ring (4) is mounted together with the rubber bellows (8). The torque transmission ring (5) compresses the rubber bellows (8) to the rotating seal ring (4). The rubber bellows prevents leakage between the shaft (9) and rotating seal ring (4) and ensures axial flexibility despite contamination and deposits.

In a rubber bellows seal, as shown in fig. 1.10, axial flexibility is obtained by elastic deformation of the bellows. However in an O-ring seal, as shown in fig. 1.9, the O-ring slides along the shaft.

The compression force from the spring keeps the two seal faces together during pump standstill and operation thanks to the flexibility of the bellows or the O-ring. This flexibility also keeps the seal faces together, despite axial movements of the shaft, surface wear, and shaft run-out.

The stationary part
The stationary part of the seal is fixed in the pump housing (1). It consists of a stationary seat (3) and a stationary secondary rubber seal (2).

The secondary seal prevents leakage between the stationary seat (3) and the pump housing (1). It also prevents the seat from rotating in the pump housing. See fig. 1.10.

The pumped medium to be sealed (A) is generally in contact with the outer edge of the rotating seal ring (B). See fig. 1.11. When the shaft starts to rotate, the pressure difference between the pumped medium (A) in the pump housing and the atmosphere (D) forces the medium to penetrate the sealing gap (from B to C) between the two flat rotating surfaces. The lubricating film is generated.

The pressure in the sealing gap is reduced from B to C, reaching the pressure at D. Leakage from the seal will appear at C.

The pressure at B is equal to the pressure at A. The pressure drop in the sealing gap during pump standstill is shown in fig. 1.12a. The closing force is only supported by direct contact between the seal faces.

The opening forces from the pressure in the lubricating film are shown by the red arrows in fig. 1.13b and 1.14b.

The parts of the seal inside the pump are subjected to a force emanating from the pressure within the pump. The axial component of this force, together with the spring force, creates the closing force (Fc) of the seal.

During pump standstill, the pressure at the outer edge of the ring (B) is equal to the system pressure (A). See fig. 1.12a.
When the shaft starts to rotate, the seal rings will separate and the pumped medium will enter the sealing gap. The pressure decreases linearly from pump pressure $B$, to atmospheric pressure $C$. See fig. 1.13a.

**Note:** In this book, pump pressure means pressure in the seal chamber.

The linearly decreasing pressure is known as the hydrostatic pressure in the sealing gap. The opening force is shown with red arrows in fig. 1.13b.

When the pump runs, see fig. 1.14a, a pressure builds up in the lubricating film. This is similar to a car hydroplaning on a wet road. This pressure is known as the hydrodynamic pressure in the sealing gap.

The hydrostatic pressure combined with the hydrodynamic pressure produces the pressure distribution in the sealing gap. The opening force is shown with red arrows in fig. 1.14b.

Full-fluid-film lubrication can be obtained if the pressure in the sealing gap is sufficiently high to balance the closing force of the seal.
Closing force
The parts of the seal inside the pump are subjected to an axial force from the pressure in the pumped medium. Together with the spring force, the axial force creates the closing force on the seal faces.

If the differential pressure between the pumped medium and the atmosphere is above approximately 20 bar, the closing force becomes so strong that it prevents the formation of an adequate hydrodynamic lubricating film. The seal faces begin to wear. Wear can be avoided by reducing the area where the hydraulic pressure affects the axial force on the shaft seal. The hydraulic force of the primary seal faces as well as the closing force of the seal are reduced.

Unbalanced and balanced mechanical shaft seals
The balancing ratio, k, is the ratio between the hydraulically loaded area, \( A_h \), and the sliding face area, \( A_s \).

\[
F_c = A_h \times P + F_s = 179 \text{ mm}^2 \times 1 \text{ MPa} + 45 \text{ N} = 224 \text{ N}
\]

\[
F_c = A_h \times P + F_s = 150 \text{ mm}^2 \times 1 \text{ MPa} + 45 \text{ N} = 195 \text{ N}
\]

\[
k = \frac{A_h}{A_s} = \frac{179}{153} = 1.17
\]

In the low pressure range of the pumped medium, unbalanced mechanical shaft seals are sufficient. See fig. 1.15a.

The pump pressure acting on the area, \( A_h \), causes a closing force to be exerted on the seal. The area, \( A_h \), of an unbalanced mechanical shaft seal is larger than the area, \( A_s \), and the balancing ratio, \( k \), is larger than 1. The contact pressure in the sliding face area exceeds the pumped medium pressure. The spring force further increases the contact pressure. The balancing ratio is often chosen to be around 1.2.

In the high pressure range of the pumped medium or at high speed, the balanced mechanical shaft seal is used. The contact pressure in the sliding face area can be smaller than the pumped medium pressure. The balancing ratio is often chosen to be around 0.8.

Balancing a mechanical shaft seal gives a thicker lubricating film in the sealing gap. A low k value can cause a higher leakage rate or can even cause the seal faces to open up.
Calculation example, unbalanced and balanced shaft seal

In this example, we shall look at the closing force of a liquid-lubricated mechanical shaft seal. The data below apply to an unbalanced Grundfos type A shaft seal. For more details on this shaft seal type, see Chapter 2, type A, page 27.

**Introduction**

**Fig. 1.16: Unbalanced Grundfos type A shaft seal**

Shaft diameter, $D_s = 16$ mm
Sliding seal face, inner diameter, $D_i = 17$ mm
Sliding seal face, outside diameter, $D_o = 22$ mm
Spring force, $F_s = 45$ N

This gives the following results:

*Hydraulically loaded area:*

$$A_h = \frac{\pi}{4} (D_o^2 - D_s^2) = \frac{\pi}{4} (22^2 - 16^2) = 179 \text{ mm}^2$$

*Sliding face area:*

$$A_s = \frac{\pi}{4} (D_o^2 - D_i^2) = \frac{\pi}{4} (22^2 - 17^2) = 153 \text{ mm}^2$$

*Balancing ratio, according to formula 1, page 15:*

$$k = \frac{A_h}{A_s} = \frac{179}{153} = 1.17$$

The closing force, $F_c$, at a 10-bar pressure ($P = 1$ MPa) is calculated as follows:

$$F_c = A_h \times P + F_s = 179 \text{ mm}^2 \times 1 \text{ MPa} + 45 \text{ N} = 224 \text{ N}$$

**Fig. 1.17: Balanced Grundfos type H shaft seal**

For a balanced Grundfos type H shaft seal for a $\Phi 16$ shaft, the calculation is as follows:

Sleeve diameter, $D_s = 17.1$ mm
Sliding seal face, inner diameter, $D_i = 17$ mm
Sliding seal face, outside diameter, $D_o = 22$ mm
Spring force, $F_s = 45$ N

*Hydraulically loaded area:*

$$A_h = \frac{\pi}{4} (D_o^2 - D_s^2) = \frac{\pi}{4} (22^2 - 17.1^2) = 150 \text{ mm}^2$$

*Sliding face area:*

$$A_s = \frac{\pi}{4} (D_o^2 - D_i^2) = \frac{\pi}{4} (22^2 - 17^2) = 153 \text{ mm}^2$$

*Balancing ratio:*

$$k = \frac{A_h}{A_s} = \frac{150}{153} = 0.98$$

The closing force, $F_c$, at a 10-bar pressure ($P = 1$ MPa) is calculated as follows:

$$F_c = A_h \times P + F_s = 150 \text{ mm}^2 \times 1 \text{ MPa} + 45 \text{ N} = 195 \text{ N}$$
In the examples above, where the areas of the sliding faces and the spring force are equal, the closing force is reduced from 224 N to 195 N by reducing the balancing ratio from \( k = 1.17 \) to \( k = 0.98 \). A smaller closing force gives less wear on the sliding faces because improved lubrication is obtained. The result is also a higher leakage rate.

**Leakage**

The lubricating film formed in the sealing gap during pump operation results in the escape of some of the pumped medium to the atmospheric side. If the mechanical seal works well and no liquid appears, the lubricating film has evaporated due to heat and pressure decrease in the sealing gap. Therefore, no liquid seeps out of the seal.

![Fig. 1.18: Seal with excessive leakage](image)

Note that evaporation of water can take place at temperatures below 100 °C, unless the surrounding atmosphere is saturated with vapour. Think of how you can dry your clothes outside on a clothes line.

The leakage rate of a mechanical shaft seal depends of a number of factors such as:

- surface roughness of seal faces
- flatness of seal faces
- vibration and stability of pump
- speed of rotation
- shaft diameter
- temperature, viscosity and type of pumped medium
- pump pressure
- seal and pump assembly.
Calculation of leakage rate

The leakage rate of a liquid-lubricated mechanical shaft seal with parallel seal faces through the sealing gap can be calculated by means of this approximate formula:

Formula 2: \[ Q = \frac{\pi \times R_m \times h^3 \times \Delta p}{6 \times \eta \times b} \]

Where
- \( Q \) = leakage rate per unit of time
- \( R_m \) = average radius of the sliding face
- \( h \) = gap height between the sliding faces (thickness of the lubricating film)
- \( \Delta p \) = differential pressure to be sealed
- \( \eta \) = dynamic viscosity of the pumped medium
- \( b \) = radial extension of the sealing gap (sliding face width).

The leakage rate, \( Q \), is then linear to the radius, \( R_m \), sliding face width, \( b \), and pressure difference, \( \Delta p \). The gap height, \( h \), however, is extremely important. Note that twice the height causes eight times as much leakage, with all other conditions remaining the same.

It seems as if the leakage decreases when viscosity, \( \eta \), increases. But when viscosity increases, the lubricating film and thus the sealing gap increases, which may result in an increase in the leakage rate. The increase in sealing gap height with an increase in viscosity is not linear. This makes it difficult to predict whether or not an increase in viscosity results in a higher or lower leakage rate.

The roughness and flatness of the two sliding faces affect the height of the sealing gap and thus the leakage. The hydrodynamic pressure increases with the speed. This can cause an increase of the gap height and thus the leakage rate.

A gap height between the sliding faces of 0.2 micron is typical for a mechanical shaft seal running in water. Consequently, the seal faces have to be very smooth and flat.

The calculation example below applies to a Grundfos type H seal running in water at 20 °C at a pressure of 10 bar. A sealing gap of 0.2 \( \mu \)m is assumed.

\[ \Delta p = 10 \text{ bar} = 1 \text{ MPa} = 1 \times 10^6 \text{ N/m}^2 \]
\[ D_o = 22 \text{ mm} \]
\[ D_i = 17 \text{ mm} \]
\[ \text{Viscosity} = 1 \text{ cst} = 0.001 \text{ N x s/m}^2 \]
\[ h = 0.0002 \text{ mm} = 0.2 \times 10^{-6} \text{ m} \]

Thus, \[ R_m = \frac{(22 + 17)}{4} = 9.75 \text{ mm} \] and \[ b = \frac{(22 - 17)}{2} = 2.5 \text{ mm} \]

Using formula 2, the leakage rate, \( Q \), is as follows:

\[ Q = \frac{\pi \times 9.75 \times 10^{-3} \text{ m} \times (0.2 \times 10^{-6} \text{ m})^3 \times 1 \times 10^6 \text{ N/m}^2}{6 \times 0.001 \text{ N x s/m}^2 \times 2.5 \times 10^{-3} \text{ m}} = 1.63 \times 10^{-11} \text{ m}^3/\text{s} = 0.06 \text{ ml/h} \]

If the roughness of the seal faces is higher, resulting in a sealing gap of 0.3 micron, the leakage rate is 0.2 ml/h.
Non-parallel seal faces
In practice, the seal faces become distorted due to temperature and pressure gradients. The most typical deformation is a tapered seal face.

For non-parallel seal faces, the hydrostatic pressure no longer decreases linearly from the pump side to the atmospheric side. In this situation formula 2 is no longer valid for calculating the leakage rate.

Converging sealing gap
When the sealing gap opens towards the pumped medium, as shown in fig. 1.19, the hydrostatic pressure increases. This is called a converging sealing gap. It appears as the blue curve in fig. 1.21.

Diverging sealing gap
When the sealing gap opens towards the atmospheric side, as shown in fig. 1.20, the hydrostatic pressure decreases. This is a called a diverging sealing gap. It appears as the orange curve in fig. 1.21.

The pressure distribution in the sealing gap is obtained by adding the hydrostatic pressure and the hydrodynamic pressure. This is shown in fig. 1.22. Note the similarity with fig. 1.14 a, page 14.
Evaporation
The absence or inadequate formation of lubricating film frequently causes damage to the seal faces. Evaporation of the pumped medium in the sealing gap occurs where the pressure is below the vapour pressure of the pumped medium.

The frictional heat in the seal faces increases the temperature of the medium resulting in an increase of the vapour pressure. This moves the start of evaporation point to the pumped medium side. See fig. 1.23.

For seals in cold water, the lubricating film extends through the entire sealing gap. For a well-functioning seal, the only leakage escaping on the atmospheric side is vapour. The evaporation will occur even in cold water due to leakages through the very narrow sealing gap, i.e. 0.0002 mm.

A partial lack of lubricating film often occurs in the sliding seal faces towards the atmospheric side when pumping water above 100 °C. This is due to evaporation of the lubricating film.

Deposits and wear tracks
When the lubricating film in the sealing gap evaporates, dissolved solids are left deposited on the seal faces.

If the thickness of deposits exceeds the necessary thickness of the lubricating film, the seal starts to leak.

In case of hard deposits, wear tracks can develop in one of the seal rings, see fig. 1.24a. In case of soft and sticky deposits, a build-up can cause the seal faces to separate, see fig. 1.24b.
Vapour pressure curve
In order to secure a proper liquid lubrication in the major part of the seal gap, it is recommended to keep the temperature around the seal at 10 to 15 °C from the vapour pressure curve. The curve for water can be seen in fig. 1.25.

Frictional heat
A mechanical shaft seal generates frictional heat. If the lubrication is poor, the heat generated can be as high as 100 watts/cm². Compared to this, a cooking plate generates around 10 watts/cm² at maximum power. To minimise the temperature increase in the sealing gap, it is important to remove the heat. The amount of heat removed is determined by these factors:

- liquid flow in the seal chamber
- thermal conductivity of the machine parts
- convection to the atmosphere.

Sometimes the influence of these factors is not sufficient, causing the lubricating film in the sealing gap to evaporate. This results in dry running of the seal.

The power loss, P, due to friction can be calculated by means of the following formula:

\[ P = F_c \times f \times v \]

Where:
- \( F_c \) = Closing force
- \( f \) = Coefficient of friction
- \( v \) = Sliding speed

The coefficient of friction (COF) depends on the lubrication and the pairing of the seal face materials. For well-lubricated seal faces, the factor is between 0.03 and 0.08.

In case of poorly lubricated seal faces, the COF depends on the seal face materials. Thus if the two seal faces are made of hard materials such as tungsten carbide, a COF up to 0.4 is possible in hot water.

For a balanced Grundfos type H shaft seal for a Ø16 shaft at 2900 min⁻¹ and 10 bar, assuming \( f = 0.04 \), the situation is as follows. See page 16:

\[ F_c = 195 \text{ N}, \quad f = 0.04, \quad v = 3.0 \text{ m/s} \]
\[ P = F_c \times f \times v = 195 \text{ [N]} \times 0.04 \times 3.0 \text{ [m/s]} = 23.4 \text{ [W]} \]

Turbulence loss in the seal chamber generates small amounts of heat when the sliding speed is below 25-30 m/s.

Sometimes a narrow seal chamber requires additional precautions to remove the heat, for example increased circulation of the pumped medium around the seal. See Chapter 2, page 31.
4. Historical development

At the beginning of the nineteenth century, many endeavours were made to develop a replacement for the conventional, braided packing used for piston pumps and rotating shafts. A more reliable system for different kinds of liquid-conveying rotating machinery was desired.

By the 1930’s, the James Walker Group came up with a mechanical shaft seal for refrigeration compressors. At the same time, the John Crane company invented the first automotive mechanical shaft seal. In the early 1940’s, the company developed and introduced the patented elastomer bellows axial shaft seal, today known as “Type 1”.

After this breakthrough in sealing technology, other types of mechanical shaft seals were developed. With several types of mechanical shaft seals, the John Crane company adopted the tagline, “The right seal for the right application”.

Today, John Crane is still a leading seal manufacturer along with Grundfos, Burgmann, Flowserve, etc.

The first Grundfos mechanical shaft seal
The first Grundfos mechanical shaft seal was launched in 1952. The seal was introduced in the CP, the first vertical multistage pump in the world. It consisted of an O-ring seal type with tungsten carbide seal faces.
The Grundfos unbalanced O-ring seal with tungsten carbide seal faces was used with success in abrasive liquids. It soon led to the development of seals for other Grundfos pumps, including the BP deep-well pumps, CR multi-stage pumps, UPT single-stage pumps, LM and LP inline pumps.

The tungsten carbide/tungsten carbide seal faces proved to be a very successful material pairing for cold-water applications. This pairing did not turn out to be as successful in hot-water applications on account of very noisy operation.

**Tungsten carbide against carbon graphite**

In the early 1990’s, Grundfos developed a rubber bellows seal with tungsten carbide against carbon graphite seal faces. This soon became the common material choice. The rubber bellows is ideally suited for seals with a carbon seat. This bellows seal was developed for CR pumps and also introduced in LM/LP single-stage pumps, CHI, AP and UMT/UPT single-stage pumps.

Later on a generation of cartridge seals facilitating mounting and service was developed.

**SiC against SiC becomes the common material pairing**

Since 2004, silicon carbide against silicon carbide (SiC/SiC) became the common material pairing for Grundfos cartridge shaft seals. This pairing has an excellent abrasive resistance and good performance in hot water.

**Summary**

This section has described the design and composition of a mechanical shaft seal. We have learned that a lubricating film is very important in order to obtain good performance. Balancing the seal can increase the thickness of the lubricating film. However, to prevent excessive leakage, the lubricating film must remain thin.