Mechanical shaft seals for pumps

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Mechanical shaft seals for pumps

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Contents

Preface 5

Chapter 1. Introduction 7
- 1. Types of shaft seals 8
- 2. Mechanical shaft seals 10
- 3. Operating principle 12
- 4. Historical development 22

Chapter 2. Mechanical shaft seal types and sealing systems 25
- 1. Mechanical shaft seal types 26
- 2. Sealing systems 31
- 3. Selecting a mechanical shaft seal 42

Chapter 3. Materials 45
- 1. Seal face materials 46
- 2. Seal face material pairings 51
- 3. Testing of shaft seals 55
- 4. Secondary seals 59
- 5. Materials of other shaft seal parts 61

Chapter 4. Tribology 63
- 1. Lubrication 65
- 2. Wear 72

Chapter 5. Failure of mechanical shaft seals 75
- 1. Introduction to failures 76
- 2. Lubrication failures 77
- 3. Contamination failures 78
- 4. Chemical, physical degrading and wear 80
- 5. Installation failures 84
- 6. System failures 86
- 7. Shaft seal failure analysis 88

Chapter 6. Standards and approvals 93
- 1. European Standard EN 12756 94
- 2. Approvals 97

Index 102

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

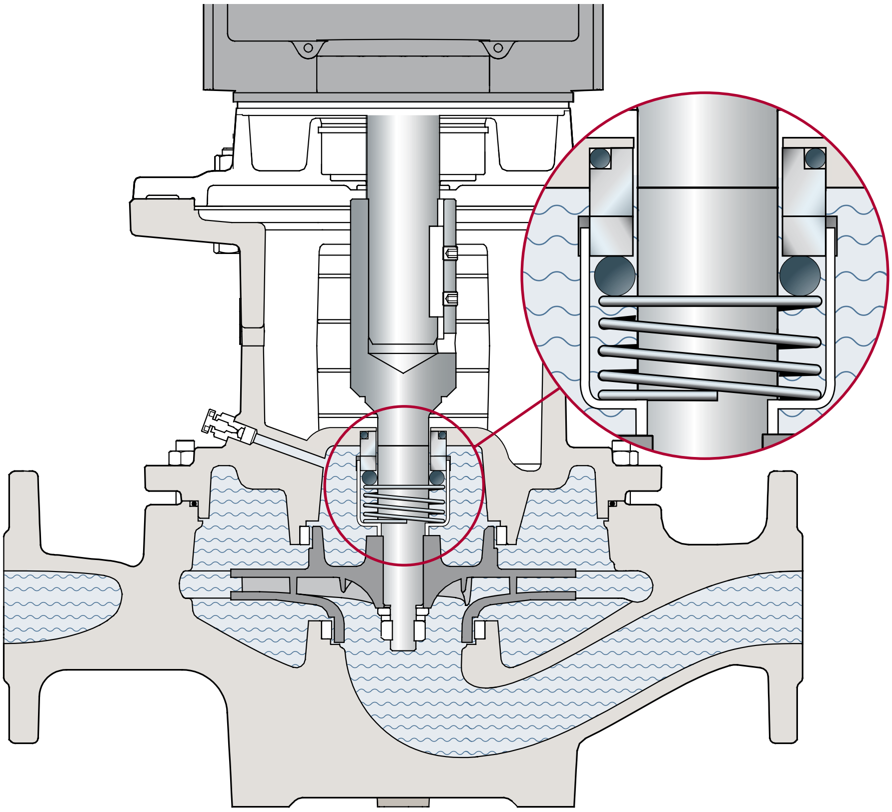


Fig. 1.1: Position of shaft seal in pump

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.

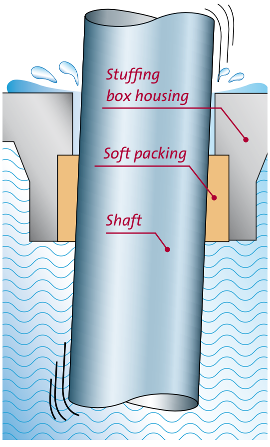


Fig. 1.2: Braided stuffing box packing with housing

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.

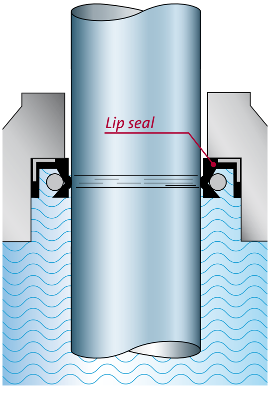


Fig. 1.3: Lip seal

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.

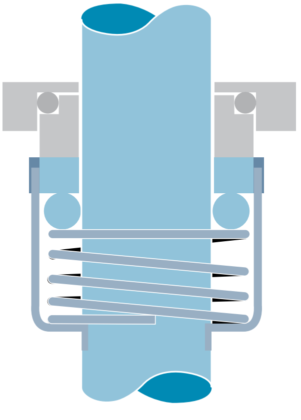


Fig. 1.4: Mechanical shaft seal

Stationary part *Rotating part*

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.

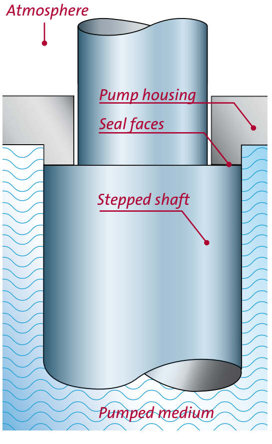


Fig. 1.5: Two axial surfaces acting as a shaft seal

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.

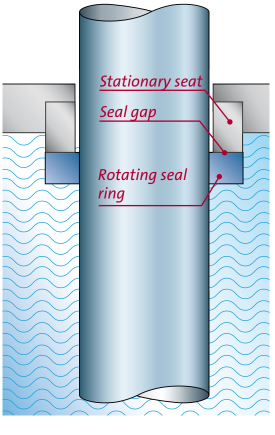


Fig. 1.6: Mechanical shaft seal with rotating seal ring and stationary seat

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.

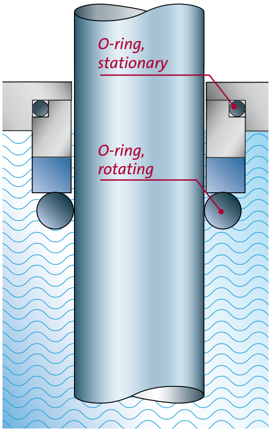


Fig. 1.7: The secondary seals confine leakage to the atmosphere

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.

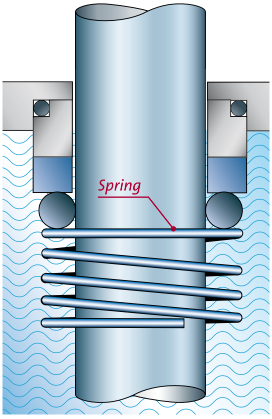


Fig. 1.8: A spring presses the rotating seal ring against the stationary seat

Spring

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

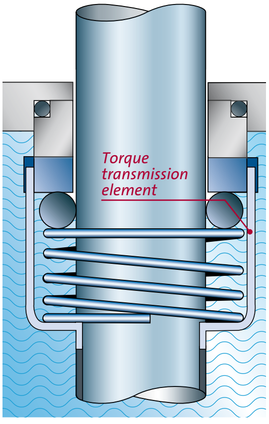


Fig. 1.9:The torque transmission element completes the mechanical shaft seal

Torque transmission element

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

All compoments 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 and the stationary part. See fig. 1.10.

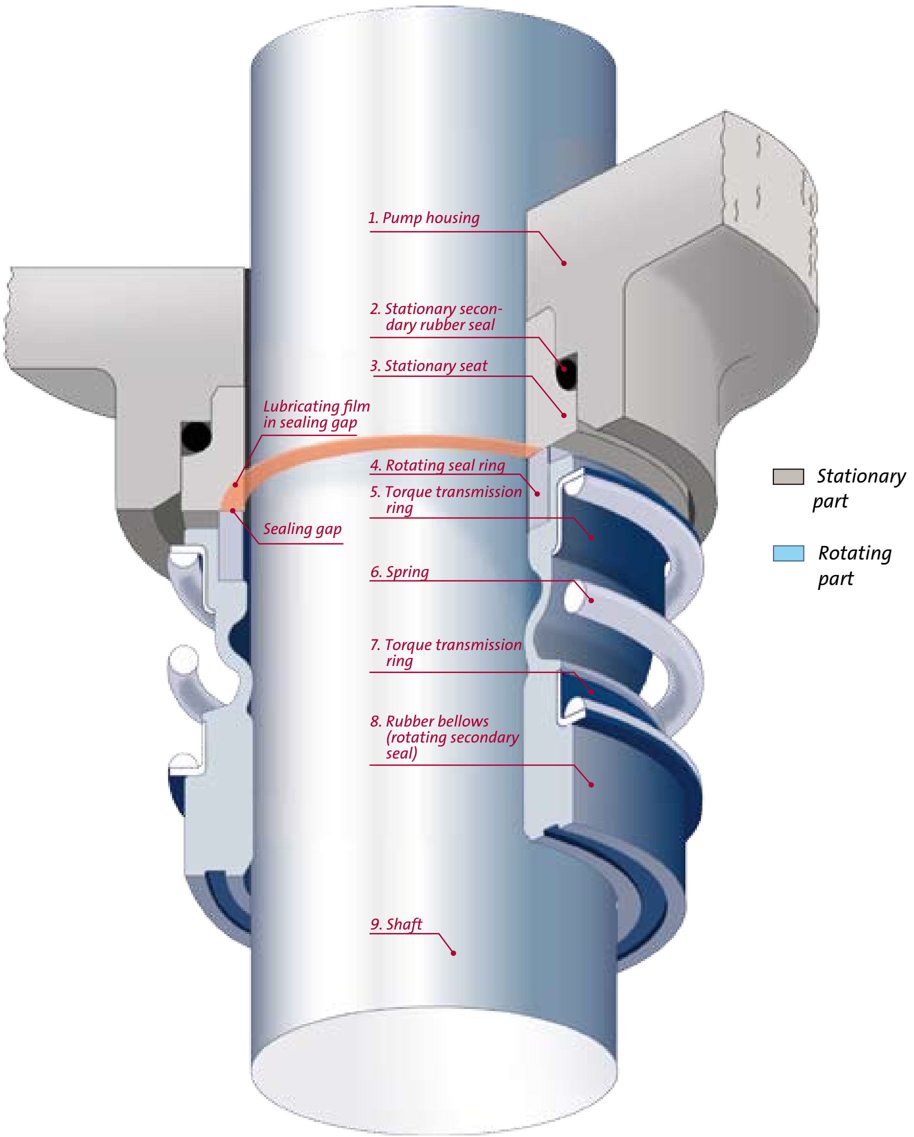


Fig: 1.10: Mechanical bellows shaft seal

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.

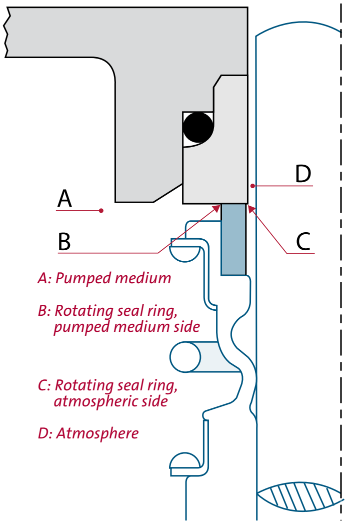


Fig. 1.11: Indication of sealing gap positions

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.

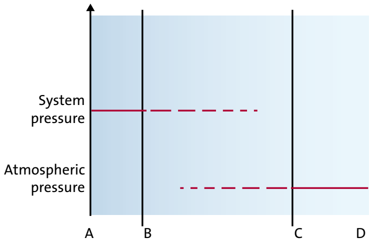


Fig. 1.12a: Pressure at standstill is either system pressure or atmospheric pressure

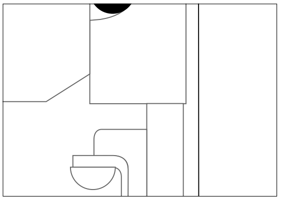


Fig. 1.12b: At standstill, there is only direct contact between the seal faces

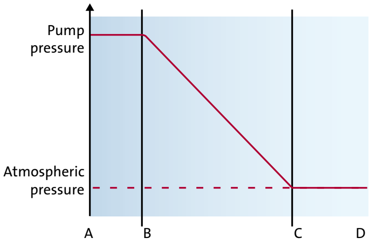


Fig. 1.13a: Hydrostatic pressure distribution for seal with parallel seal faces

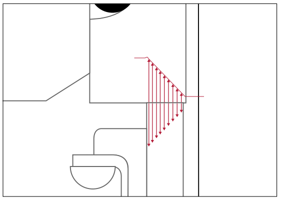


Fig. 1.13b: Opening forces from hydrostatic pressure distribution

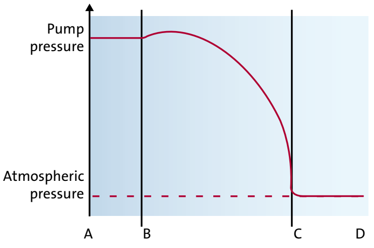


Fig. 1.14a: Pressure distribution in the sealing gap when the hydrostatic and hydrodynamic pressures are added

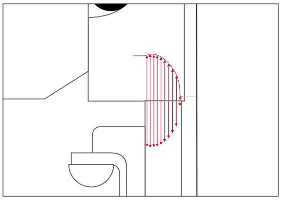


Fig. 1.14b: Opening forces from combined hydrostatic and hydrodynamic pressure distribution

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.

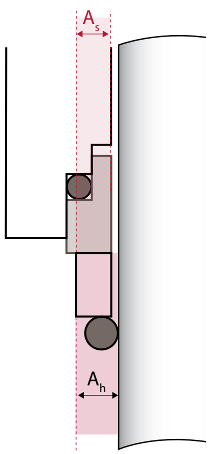
Closing force

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.



Formula 1:

$$k = \frac{\text{Hydraulically loaded area}}{\text{Sliding face area}} = \frac{A_h}{A_s}$$

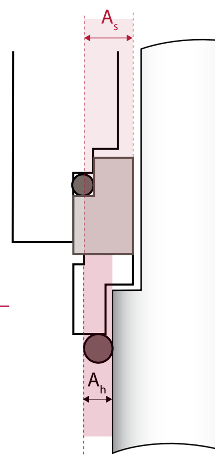


Fig. 1.15a: An unbalanced shaft seal, k>1

Fig. 1.15b: A balanced shaft seal, k<1

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 low pressure range of the pumped medium, unbalanced mechanical shaft seals are sufficient. See fig. 1.15a.
The area, A_h, of a balanced mechanical shaft seal is smaller than the area, A_s, and the balancing ratio, k, is smaller than 1. The area, A_h, can be decreased by reducing the diameter of the shaft on the atmospheric side. See fig. 1.15b.
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.

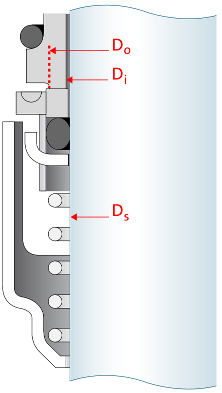


Fig. 1.16: Unbalanced Grundfos type A shaft seal

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:

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

Sliding face area:

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

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:

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

For a balanced Grundfos type H shaft seal for a Ø16 shaft, the calculation is as follows:

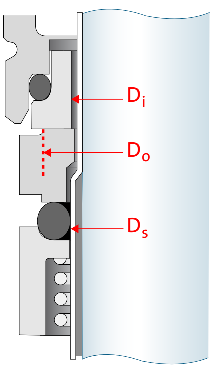


Fig. 1.17: Balanced Grundfos type H shaft seal

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:

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

Sliding face area:

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

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.

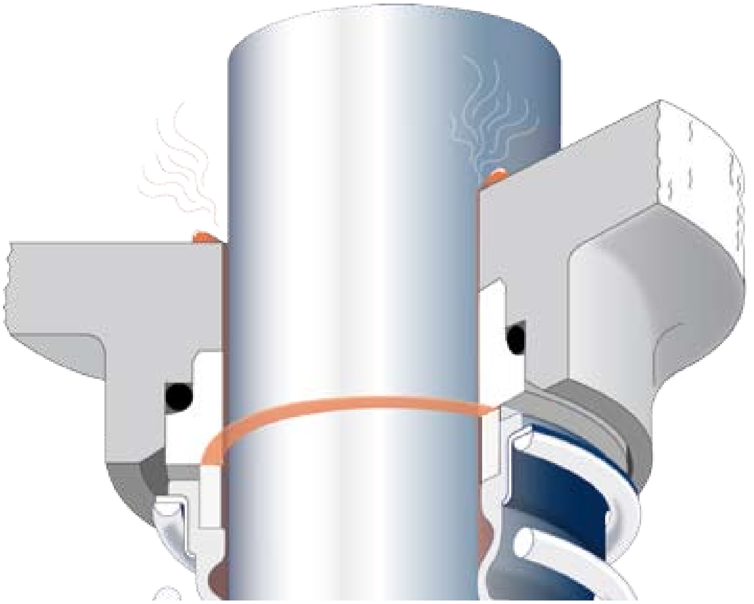


Fig. 1.18: Seal with excessive leakage

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)
Δp = differential pressure to be sealed
η = 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, Δp. 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, η, 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 μm is assumed.

Δp = 10 bar = 1 MPa = 1 x 10^6 N/m^2
D_o = 22 mm
D_i = 17 mm
Viscosity = 1 cst = 0.001 N x s/m^2
h = 0.0002 mm = 0.2 x 10^{-6} m

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

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

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

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.

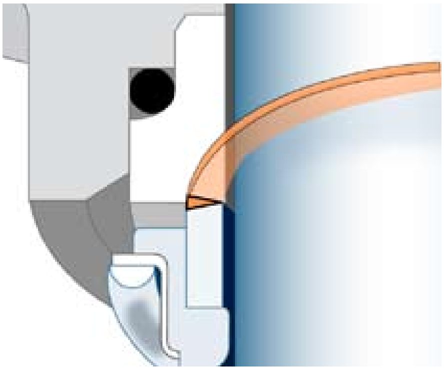


Fig. 1.19: Converging sealing gap

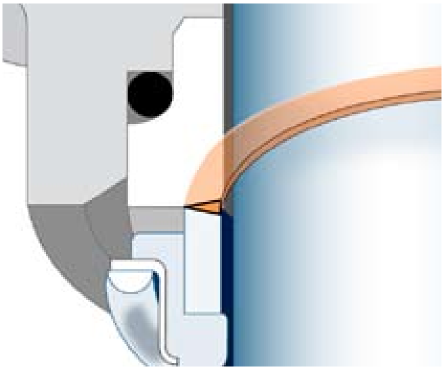


Fig. 1.20: Diverging sealing gap

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.

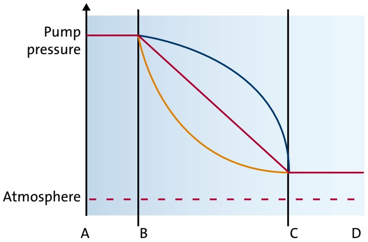


Fig. 1.21: Hydrostatic pressure distribution for different sealing gap geometries

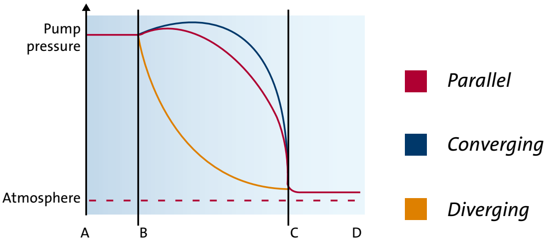


Fig. 1.22: Hydrostatic and hydrodynamic pressure distribution for different sealing gap geometries

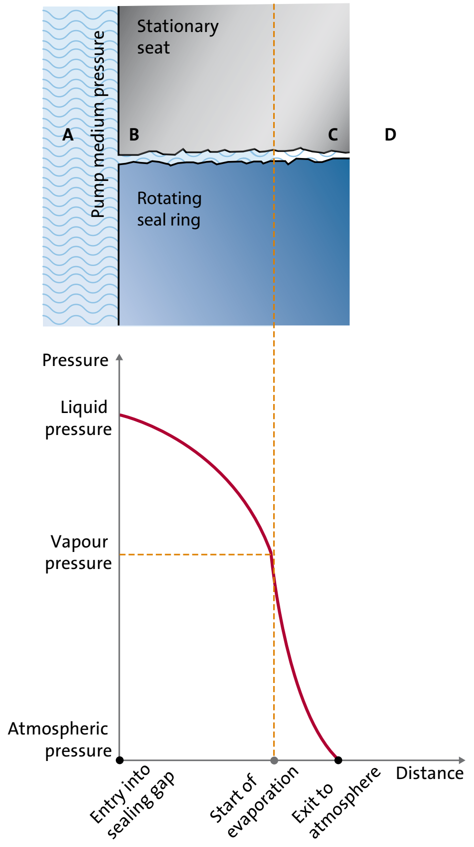


Fig. 1.23: Pressure distribution in a sealing gap with hot water

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.

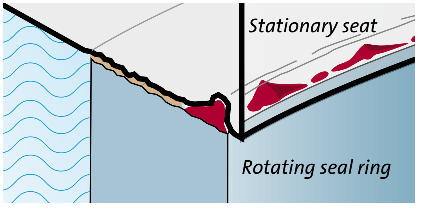


Fig. 1.24a: Development of wear tracks due to hard deposits

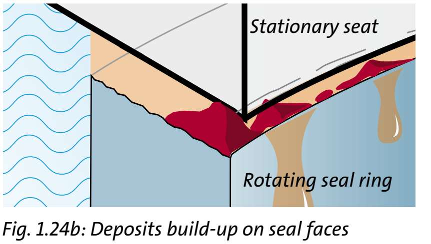


Fig. 1.24b: Deposits build-up on seal faces

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.

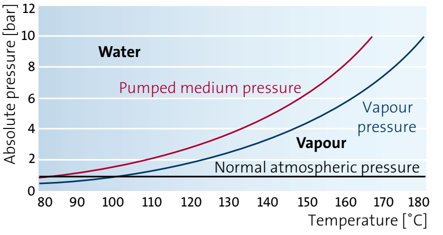


Fig. 1.25: Vapour pressure curve for water

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 N, f = 0.04, v = 3.0 m/s

$$P = F_c \times f \times v = 195\ [N] \times 0.04 \times 3.0\ [m/s] = 23.4\ [W]$$

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.

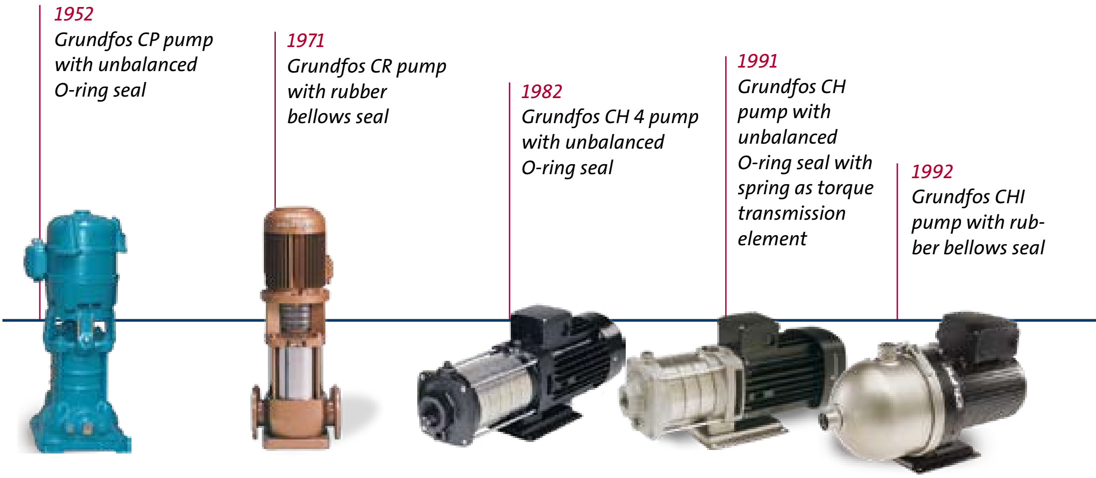


Fig. 1.26: Grundfos shaft seal development

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.

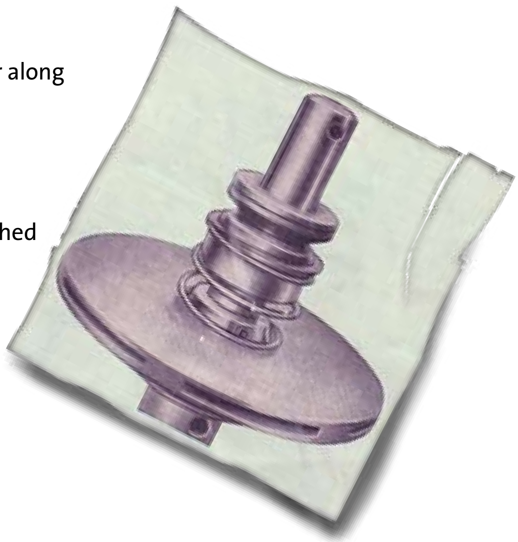


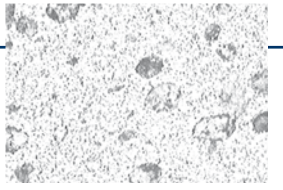
Fig. 1.27: Original illustration of CP pump shaft seal from the "Grundfos pump magazine", 1956

1993
Rubber bellows seal introduced in Grundfos CR pumps

1998
Unbalanced O-ring seal in cartridge design for large CR pumps

2000
Balanced O-ring seal in cartridge design for CR pumps

2004
Silicon carbide introduced as common seal ring material in CR pumps



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.

Chapter 2

Mechanical shaft seal types and sealing systems

1. Mechanical shaft seal types
2. Sealing systems
3. Selecting a mechanical shaft seal

1. Mechanical shaft seal types

In this chapter, the basic working principles for single mechanical shaft seals will be put into a practical context.
The chapter describes mechanical shaft seals used in Grundfos pumps as examples of the variety of shaft seal solutions for different applications.

Type A

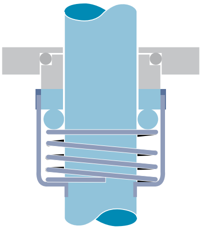
Unbalanced O-ring seal with rigid torque transmission system

Robust O-ring seal featuring a rigid torque transmission design required for hard material pairings (WC/WC or SiC/SiC), even where lubrication is poor.
The dynamic secondary seal is an O-ring. This involves a risk of wear on the shaft under the O-ring and of seal hang-up (blocking of axial movement of the rotating seal ring).

Type B

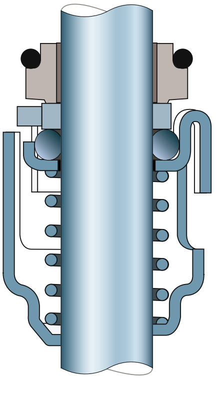
Rubber bellows seal

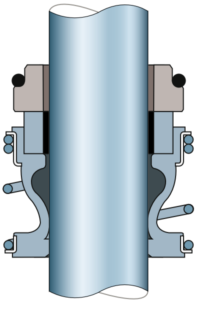
Bellows seal with torque transmission across the spring and around the bellows. Therefore it is not designed for hard material pairings in applications with poor lubrication.
Due to the bellows, the seal does not wear the shaft, and the axial movement is not prevented by deposits or seizure on the shaft.



Stationary part

Rotating part



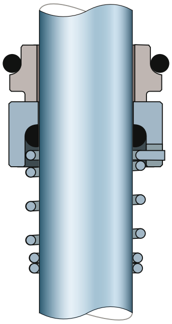


Type C

Unbalanced O-ring seal with spring as torque transmission element

Low-pressure, simple O-ring seal with the spring acting as torque transmission element. Therefore the seal is dependent on the direction of shaft rotation. The shown seal is for a counter-clockwise shaft rotation.

The seal type is excellent for low-temperature, clean-water applications with a ceramic/carbon seal face pairing.

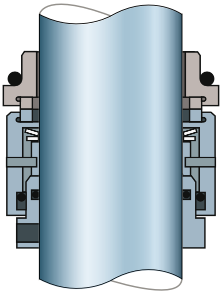


Type D

Balanced O-ring seal with spring on the atmospheric side

Due to the balancing, this O-ring seal type is suitable for high-pressure applications.

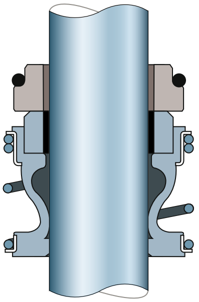
The seal is excellent for high-viscosity, dirt- and fibre-containing liquids becauce the spring is located on the atmospheric side.
The seal features rigid torque transmission design.



Type G

Rubber bellows seal with reduced seal face

Rubber bellows seal like type B but with a narrow seal face. Due to the narrow seal face, the seal performs well in high-viscosity and anti-freeze liquids.



Type H

Balanced, cartridge O-ring seal unit with rigid torque transmission system

This seal type is assembled in a cartridge unit which makes replacement safe and easy.

Similar to the type D seal, the balancing makes this O-ring seal type suitable for high-pressure applications.

Type K

Balanced, rolled-metal bellows cartridge seal unit

The metal bellows acts both as spring and torque transmission element.

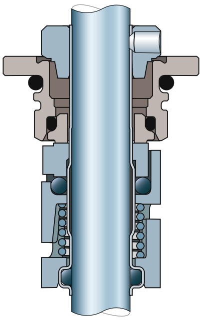
This seal type has only static rubber parts, with reduced risk of hang-up similar to type B.

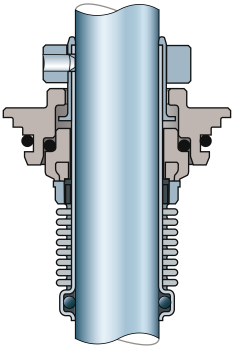
Type R

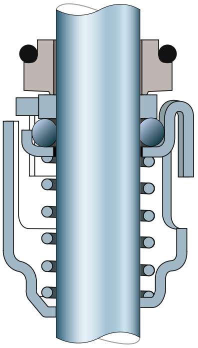
Unbalanced O-ring seal, type A, with reduced seal face

O-ring seal like type A but with a narrow seal face. Due to the narrow seal face of the unbalanced design, the balancing ratio exceeds that of seal type A. This reduces the pressure and temperature operating limits of the seal.

Similar to type G, the seal performs well in high-viscosity and anti-freeze liquids.



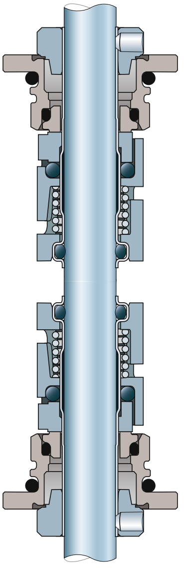




Type O

Two seals mounted in a "back-to-back" arrangement

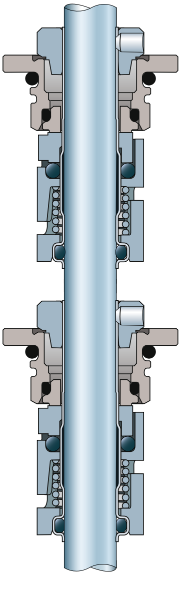
This seal arrangement incorporates a clean barrier fluid with a higher pressure than the pumped medium.
This totally prevents leakage from the pumped medium to the environment and the clean barrier fluid secures a good lubrication of the seal faces of both seals.
See descriptions on page 32.



Type P

Two seals mounted in a "tandem" arrangement

This seal arrangement incorporates a clean flushing fluid with a lower pressure than the pumped medium.
This cools the seal rings of the seal in the pumped medium and prevents precipitation from leakage.
See descriptions on page 36.



2. Sealing systems

Some of the shaft seals described previously can be combined with specially designed pumps and in double seal arrangements. See the principles described below.

Circulation

Sometimes it is necessary to cool the seal faces of single mechanical shaft seals or remove deposits in the seal chamber. In such cases a circulation pipe from the pump discharge side to the seal chamber can be fitted. The cooling liquid flows from the seal chamber back to the pumped medium. This ensures a good exchange of liquid in the seal chamber. A pipe dimension of Ø10/Ø8 is sufficient.

Internal circulation from the pressure side to the seal chamber can also be integrated in the pump design with the same result. See fig. 2.1.

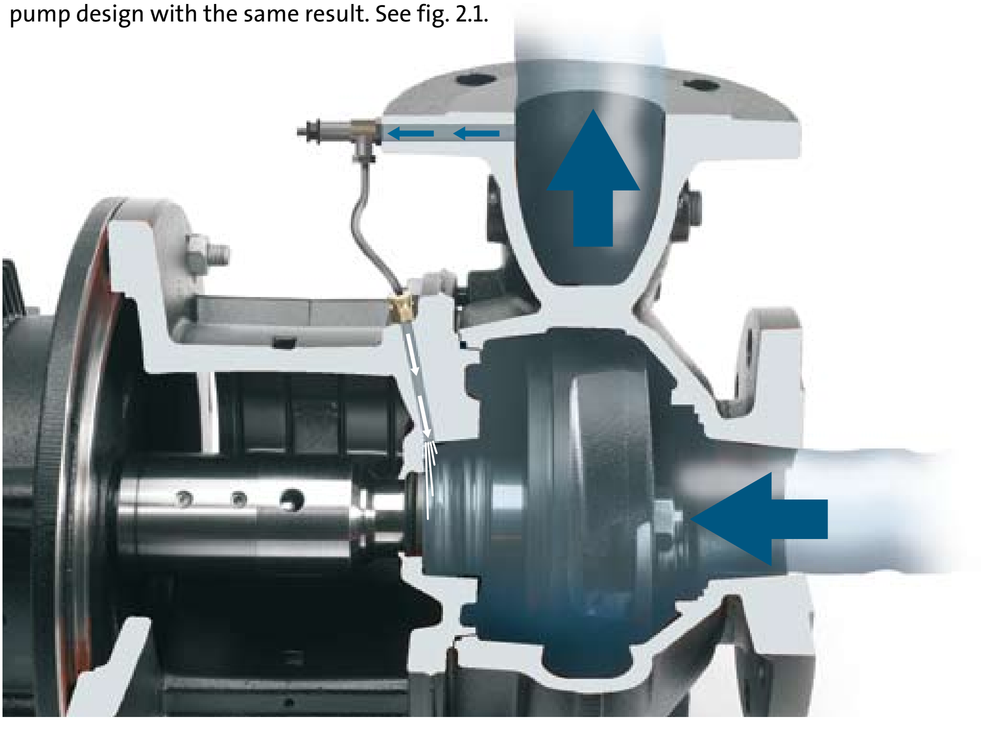


Fig 2.1: Circulation circuit for cooling a single mechanical shaft seal

Double seals can be arranged in tandem with the seats in the same direction on the shaft, or back-to-back with the seats in the opposite direction on the shaft.

The purpose of these designs is, among other things, to control temperature, pressure or flow in the cooling/heating lubricating liquid.

Back-to-back arrangement with barrier fluid, seal type arrangement O
This term is commonly used in sealing engineering to describe an arrangement with two shaft seals mounted in opposite directions. Between the two seals is a pressurised barrier fluid. The barrier fluid has several advantages to the product-side seal as compared to a single shaft seal. See. fig. 2.2.

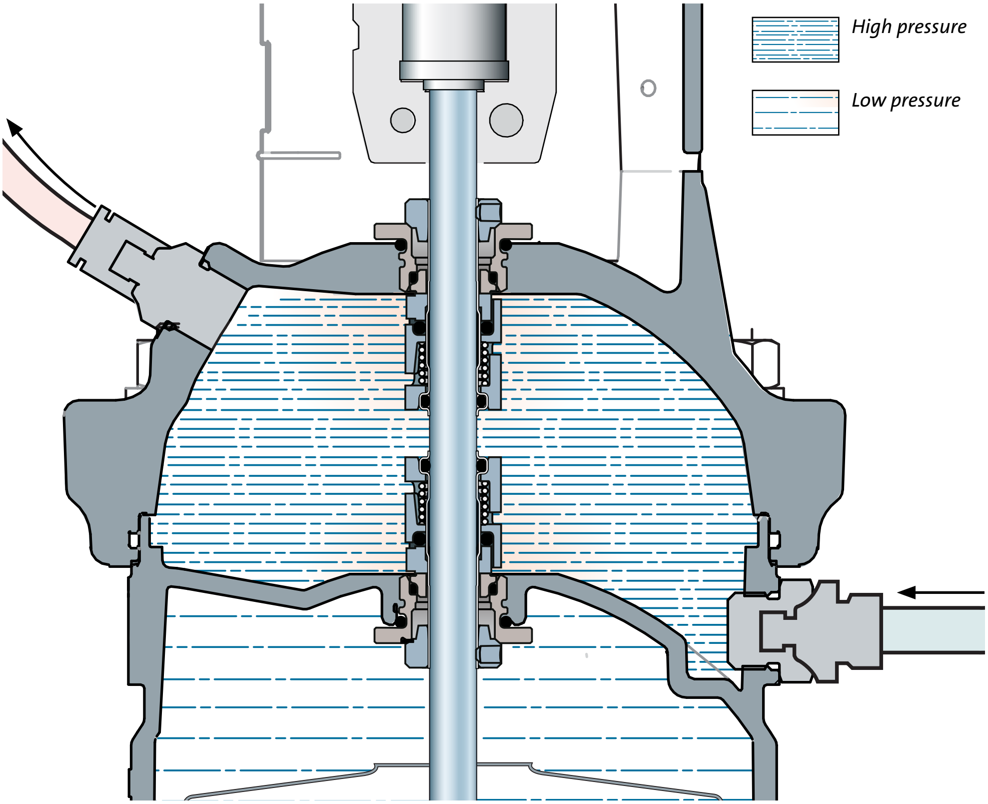


Fig. 2.2: Grundfos CR pump with back-to-back seal arrangement

The seal arrangement is suitable for poisonous and explosive liquids when no leakage from the pumped medium to the atmosphere can be accepted.

The barrier fluid pressure is higher than the pump pressure, as a result of which any leakage will pass from the barrier fluid to the pumped medium. The barrier fluid pressure must be minimum 2 bar or 10 % above the pump medium pressure close to the seal. As the clean barrier fluid has a higher pressure, it also serves as lubricating liquid for all seal faces.

The back-to-back shaft seal arrangement is particularly suitable for sticky media and/or liquids with many abrasive particles. The seal arrangement prevents the pumped medium from entering the seal gap and consequently prevents excessive wear.

Water or water mixed with glycerine is the most common liquid in closed pressurized back-to-back arrangements because it is non-poisonous and compatible with many types of pumped media. The barrier fluid chosen must always be compatible with the pumped medium.

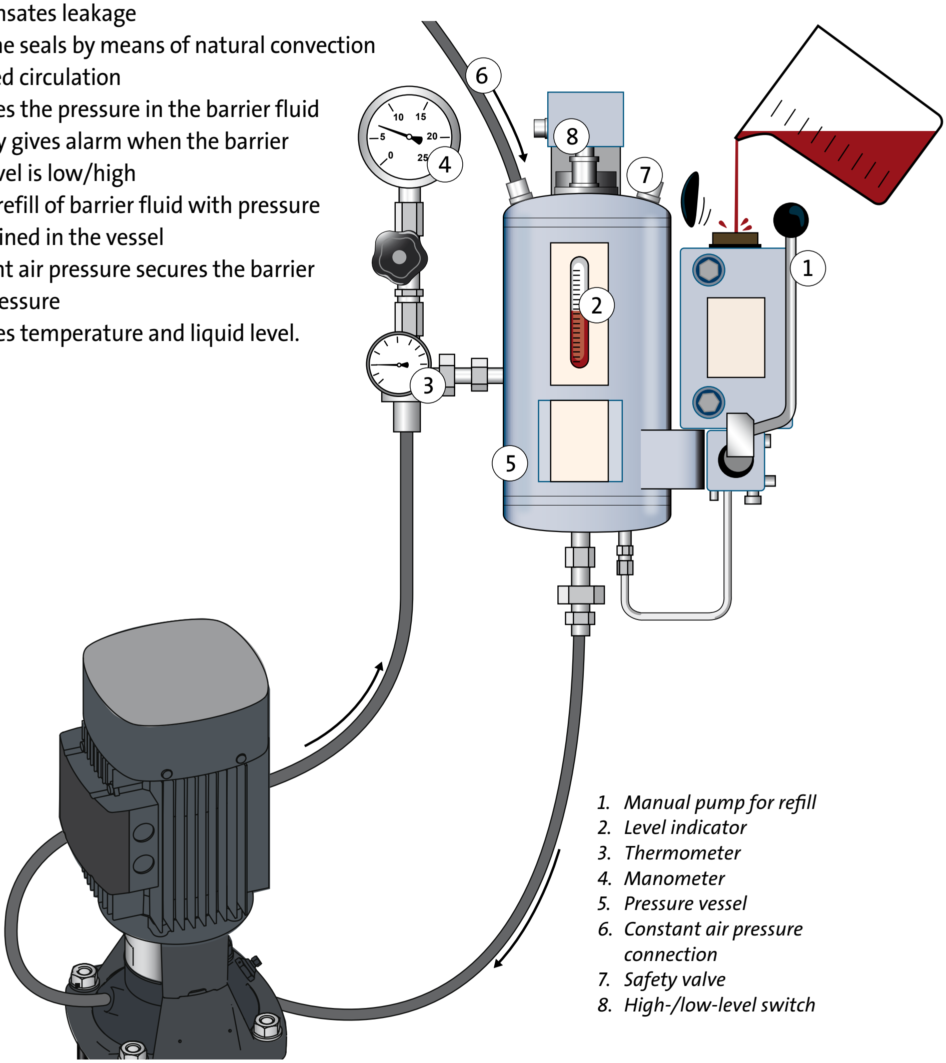
To maintain the overpressure in the barrier fluid in relation to the pumped medium pressure, various pressure sources can be used as described in the following sections.

Fixed pressure

A pressure vessel with fixed pressure in the barrier fluid with 10 % or 2 bar higher than the pressure in the pumped medium. See fig. 2.3.

The advantages are as follows:

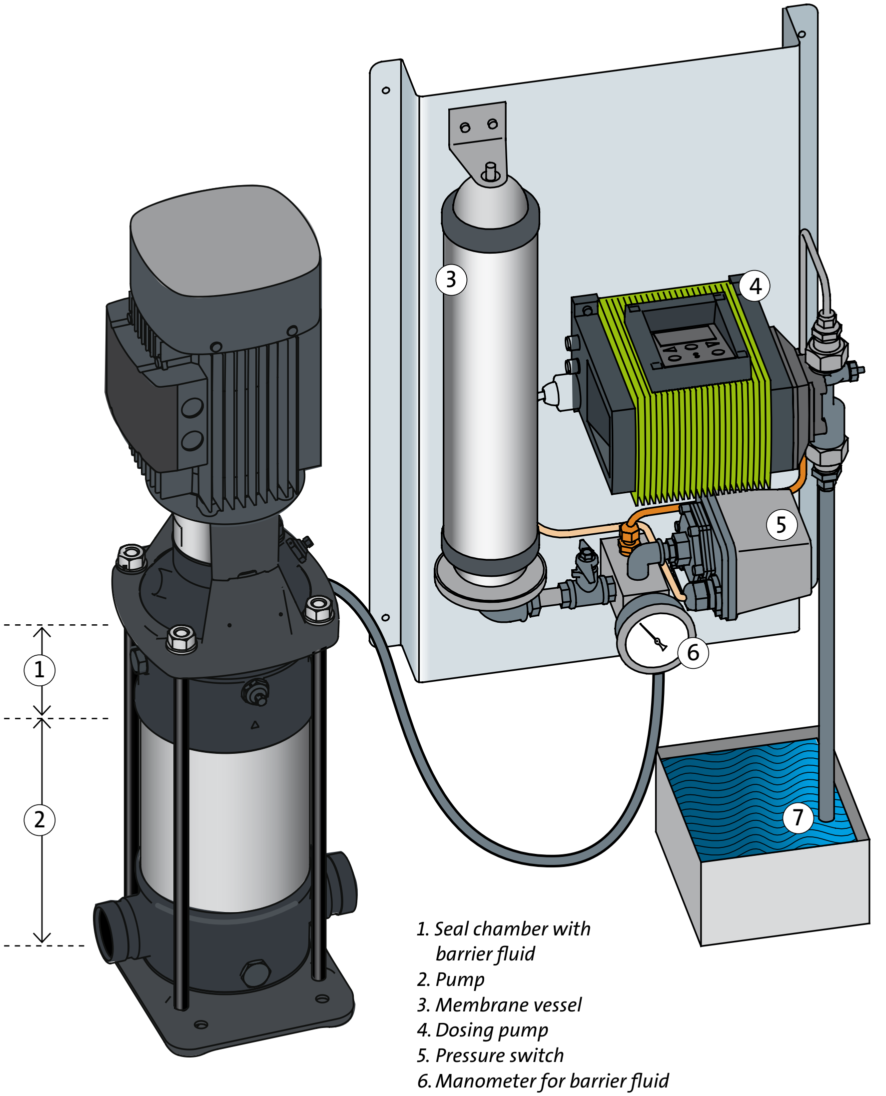
- compensates leakage
- cools the seals by means of natural convection or forced circulation
- indicates the pressure in the barrier fluid
- possibly gives alarm when the barrier fluid level is low/high
- allows refill of barrier fluid with pressure maintained in the vessel
- constant air pressure secures the barrier fluid pressure
- indicates temperature and liquid level.



1. Manual pump for refill
2. Level indicator
3. Thermometer
4. Manometer
5. Pressure vessel
6. Constant air pressure connection
7. Safety valve
8. High-/low-level switch

Fig. 2.3: Pressure vessel with fixed pressure connected to a Grundfos CR pump with a back-to-back seal arrangement

Fixed pressure obtained by means of a dosing pump
Another way of obtaining a fixed pressure in the seal chamber is by means of a dosing pump. The pump automatically keeps the level set for the overpressure. This solution is mainly used in dead-end applications where cooling from the seal chamber is sufficient. See fig.2.4.

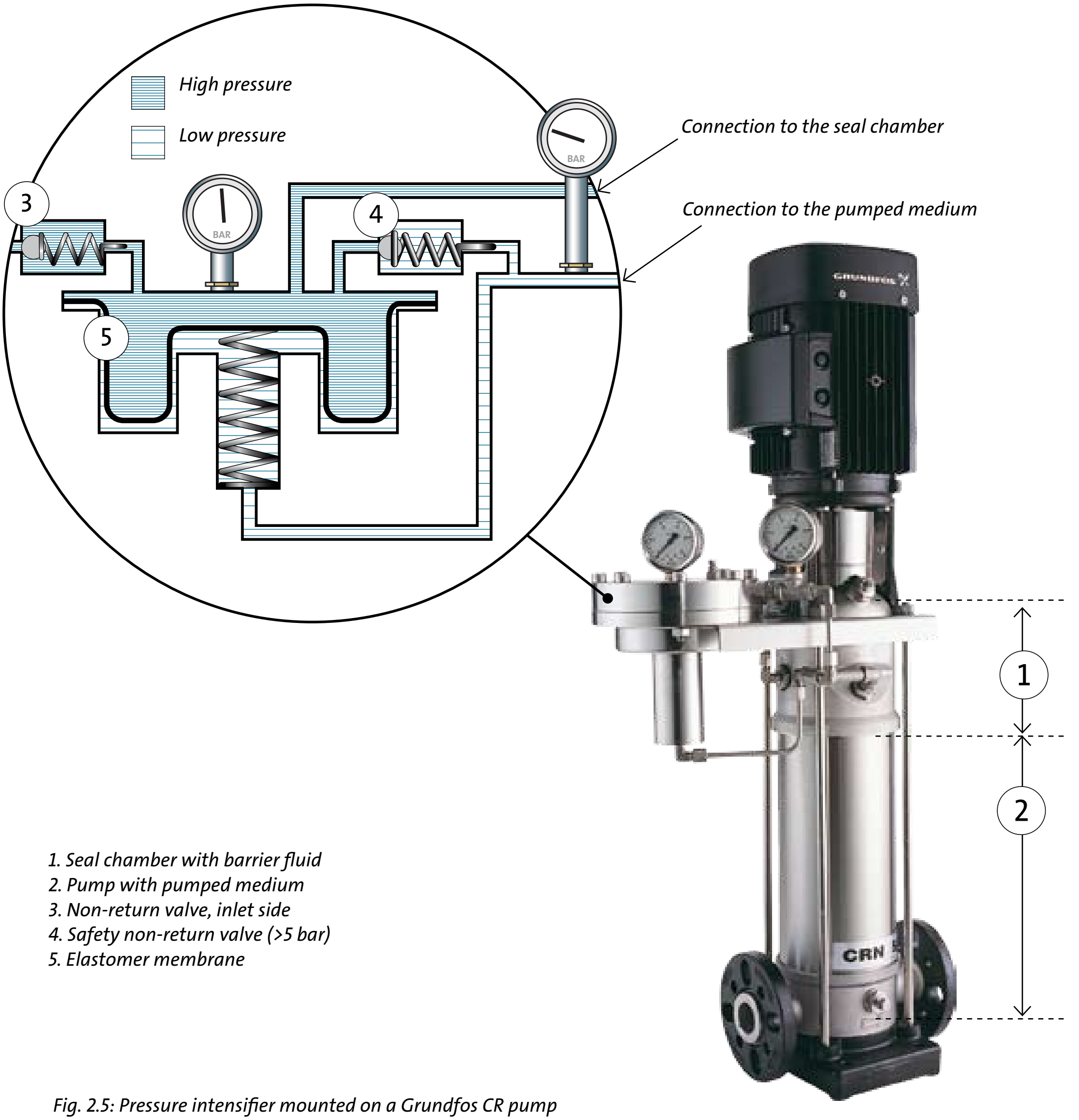


1. Seal chamber with barrier fluid
2. Pump
3. Membrane vessel
4. Dosing pump
5. Pressure switch
6. Manometer for barrier fluid
7. Reservoir with barrier fluid

Fig. 2.4: Dosing pump maintaining a fixed pressure for back-to-back seal in a Grundfos CR pump

Pressure intensifier

The Grundfos pressure intensifier automatically creates a pressure that is 2 bar higher than the pump medium pressure, independent of the specific pump medium pressure.
The system maintains the overpressure automatically until it is empty. The intensifier requires a discontinuous working cycle, as it has to be refilled.
The barrier fluid inlet must be fitted with a non-return valve to avoid back pressure to the source. See. fig.2.5.



1. Seal chamber with barrier fluid
2. Pump with pumped medium
3. Non-return valve, inlet side
4. Safety non-return valve (>5 bar)
5. Elastomer membrane

Fig. 2.5: Pressure intensifier mounted on a Grundfos CR pump

Tandem seal arrangement with flushing fluid, Seal type arrangement P
The system contains a seal chamber with two shaft seals mounted in the same direction. The flushing fluid between the two seals has lower pressure as compared to the pumped medium and offers several advantages to the product-side shaft seal such as following:

- There is no evaporation in the sealing gap. This prevents the formation of deposits as well as crystallisation on the flushing fluid side.

- The flushing fluid lubricates and cools even when the pump runs dry or runs with vacuum.

See fig. 2.6.

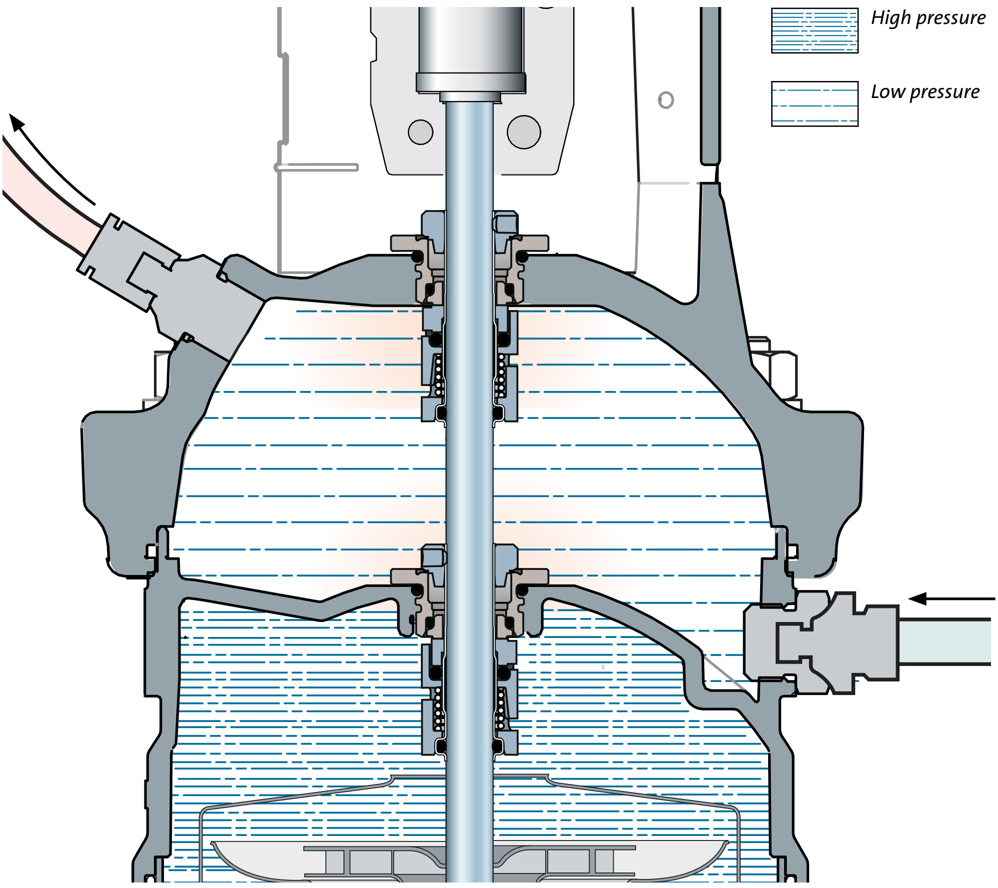


Fig. 2.6: Grundfos CR pump with a tandem seal arrangement

There are several ways of connecting the flushing fluid from an elevated reservoir to the seal chamber such as:

Seal chamber with circulation from a reservoir

Connect the seal chamber to a reservoir with circulation. The flushing fluid circulates by natural convection or a separate pump, lubricates and cools the seal faces. The flushing fluid in the reservoir must be replaced after a period of time due to contamination from the pumped medium.

Seal chamber with dead end connection from a reservoir

Connect the reservoir with a single pipe to the seal chamber. The flushing fluid lubricates the seal faces, but cools them less than by circulation. The flushing fluid in the reservoir must be re-placed after a period of time due to contamination from the pumped medium.

Seal chamber with external flushing fluid

Allow the flushing fluid to circulate through the seal chamber to a drain. The flushing fluid cools and lubricates the seal faces effectively and makes it possible to monitor the seal leakage.

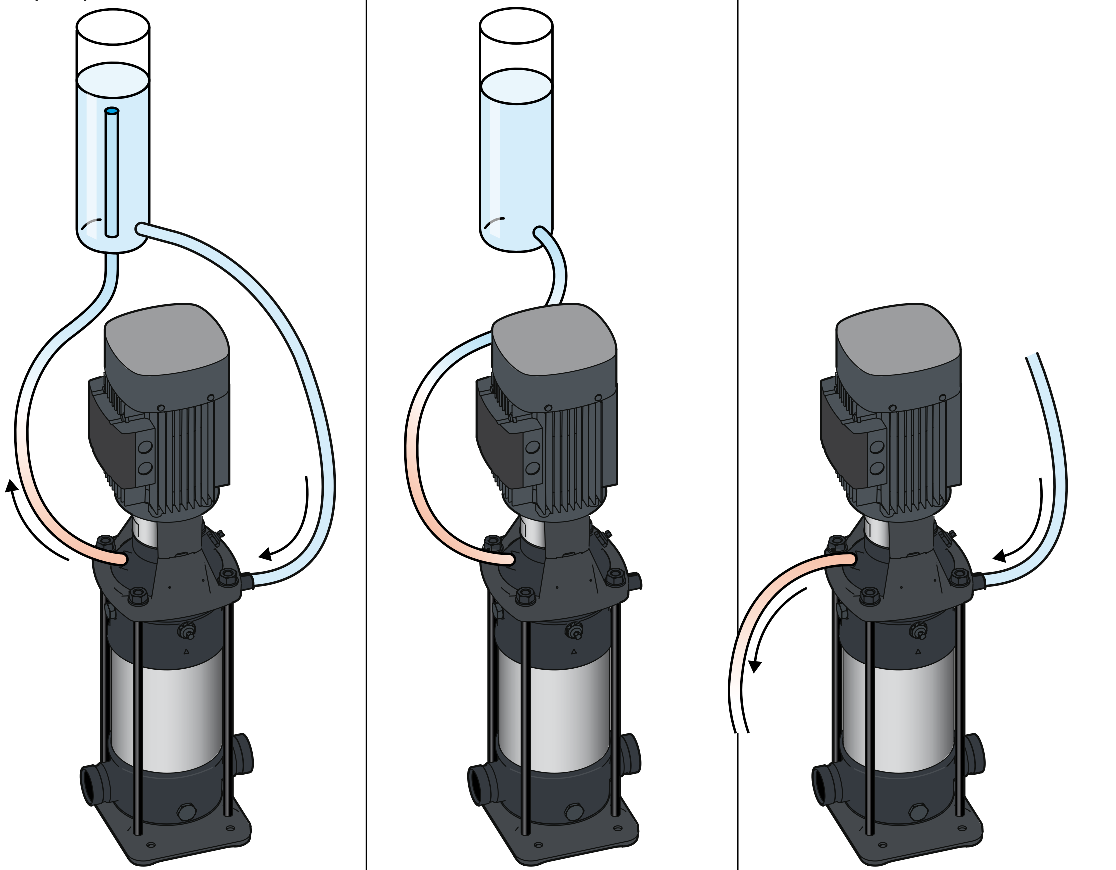


Fig. 2.7: Flushing fluid examples

Other sealing systems

Sanitary shaft seals

The demands on shaft seals in pumps designed for sterile and sanitary applications differ entirely from those made on other seals.

Often the seal needs to comply with standards and regulations. Some of these are summerized in Chapter 6.

In some instances the seal materials must comply with guidelines for cleanability and resistance to the pumped media and be capable of CIP, cleaning-in-place, and SIP, sterilisation-in-place. In addition, low roughness values and electro polished surfaces, marked yellow, are required on medium side components.

Special attention must be paid to the elastomer components of the shaft seal. Elastomer components must withstand the pumped media and temperatures in the cleaning processes. The purpose of these requirements is to ensure that all shaft seal surfaces in contact with the pumped media can be cleaned.
See figures 2.8, 2.9 and 2.10.

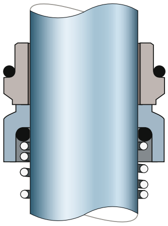


Fig. 2.8: Grundfos shaft seal type C for low cleaning requirements

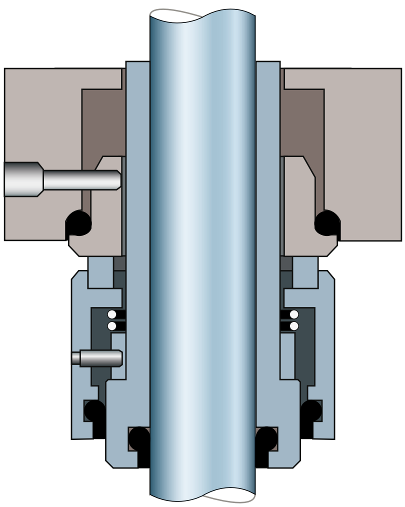


Fig. 2.9: Grundfos shaft seal type D for moderate cleaning requirements. Secondary seals have been modified, leaving no gaps

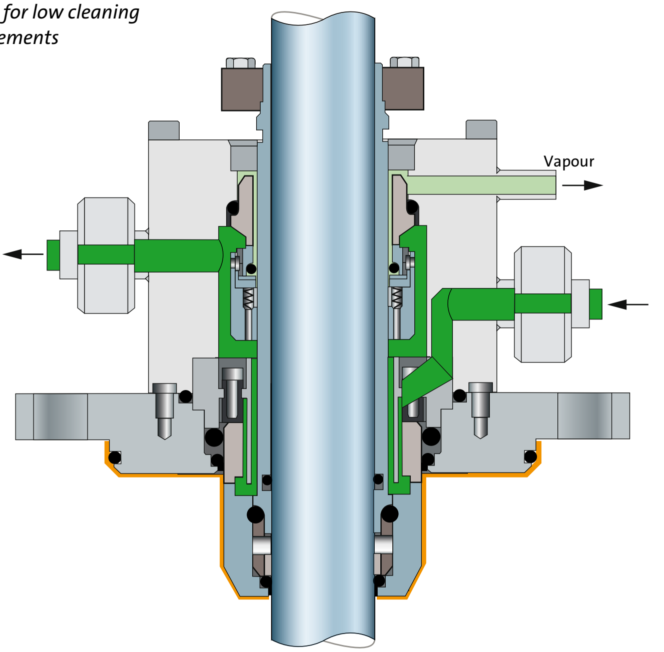


Fig. 2.10: Example of complex sanitary agitator seal subject to the highest sterilisation and cleanbility requirements. The barrier fluid (green) can be steam condensate. Surfaces marked with yellow are electro-polished. Secondary seals on medium side have been modified, leaving no gaps

High-speed mechanical shaft seals
Where speeds exceed 15-20 m/sec, the seat must be the rotating part to reduce unbalance of the seal. See fig. 2.11.

Other advantages of the rotating seat are that misalignment of the shaft causes the springs to adjust only once and prevent fretting of the sleeve under the O-ring.

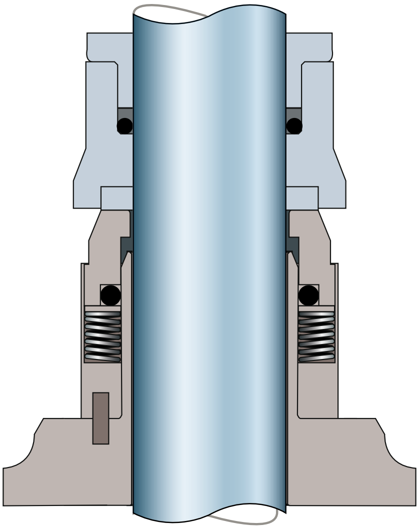


Fig. 2.11: (To the right) Example of a high-speed shaft seal for Grundfos BME pumping system

Air cooled top for high temperatures
For applications in woling high temperatures of the clean pumped medium such as hot water or thermal oil, it can be advantageous to extend the length of the pump.

As a result, and air chamber will be formed below the seal chamber. Thanks to this arrangement, the standard shaft seal is located at a distance from the hot pumped medium, allowing the shaft seal to generate a stable lubricating film in the sealing gap. The exchange of pumped medium with pumped medium from the seal chamber is very low due to a throttle around the shaft.

An automatic air vent valve is required to vent the seal chamber. This stand-alone sealing arrangement is independent of external connections. See fig. 2.12.

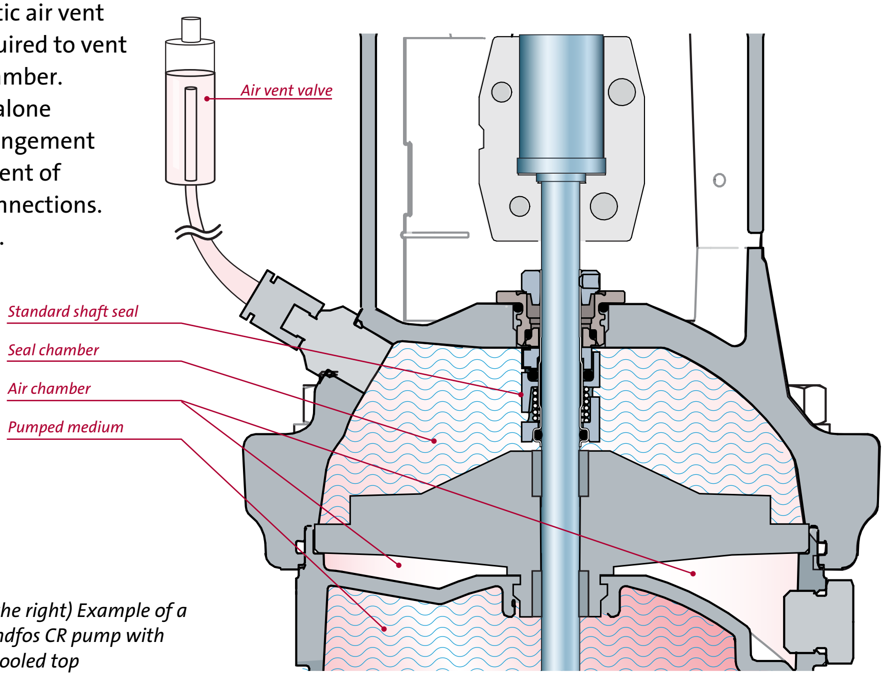


Fig. 2.12: (To the right) Example of a Grundfos CR pump with air-cooled top

Magnetic-drive system
The magnetic-drive system constitutes an entirely different type of sealing of a rotating shaft. For applications where it is absolutely necessary to avoid leakage from the shaft seal, an alternative to a back-to-back arrangement is a can that separates the pumped medium side from the atmospheric side. The magnetic-drive system incorporates an outer and inner rotor with magnets, separated by the can. The magnetic-drive system transfers the torque from the motor to the pump shaft. The system only has static O-rings to seal the clean pumped medium, free from magnetic particles.
This stand-alone sealing arrangement is independent of external connections.
See fig. 2.13.

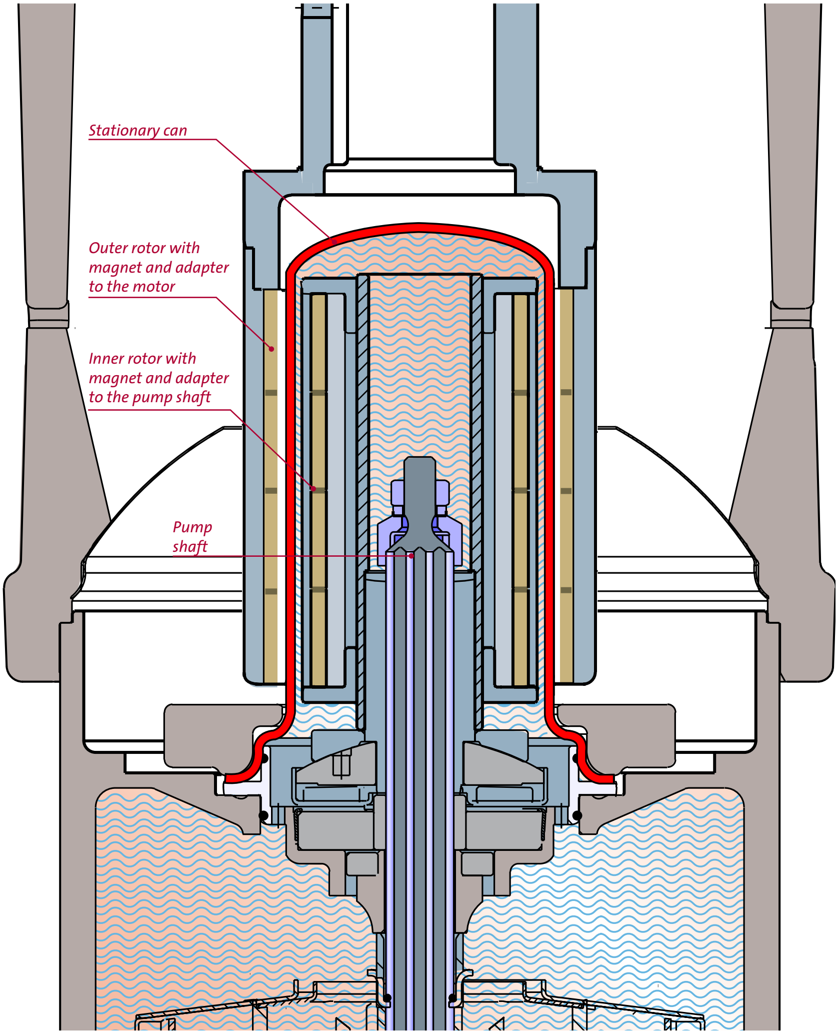


Fig. 2.13: Hermetically sealed system with magnetic-drive system

External-seal arrangement
For the pumping of some types of clean and very aggressive but non-poisonous media, it can be an advantage to place the rotating part of the seal with the springs and seal driver outside the pumped medium.

This type of balanced seal requires internal overpressure to keep the seal faces together. The clearance between shaft and seat is so large that an exchange of liquid to cool the seal faces can take place.
See fig. 2.14.

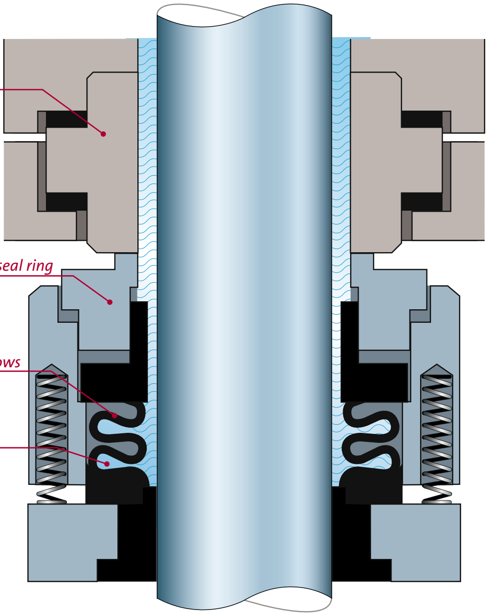


Fig. 2.14: Balanced external shaft seal for corrosive media

Submersible motors
The differential pressure between the inside and the outside of the submersible motor is small. Therefore mechanical shaft seals as well as lip seals can be used. However, the life of a mechanical shaft seals is much longer.

Special arrangements are made to prevent excessive overpressure inside the motor.
See. fig. 2.15.

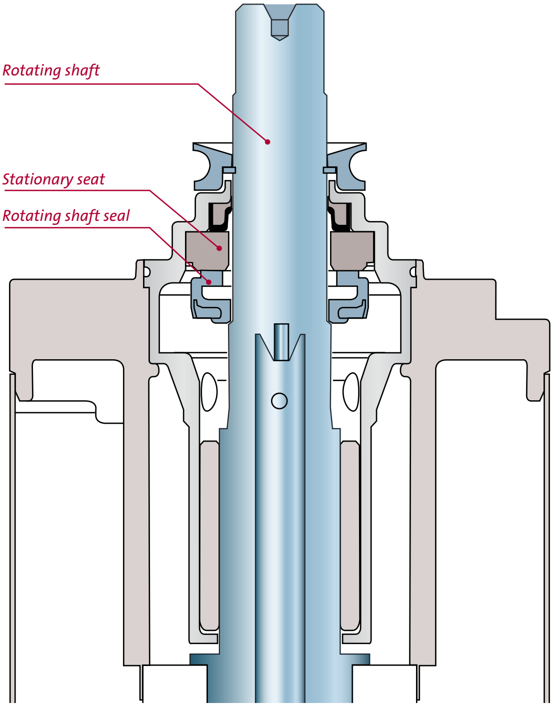


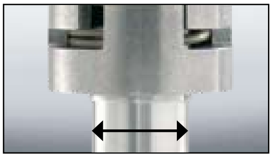
Fig. 2.15: Submersible motor with mechanical shaft seal

3. Selecting a mechanical shaft seal

The mechanical shaft seal should be selected according to the operating conditions at the shaft seal location.

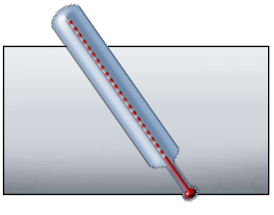
These important factors must be considered when selecting a mechanical shaft seal:

- Shaft seal diameter
- Type of pumped medium
- Temperature
- Sealing pressure
- Shaft speed of rotation.

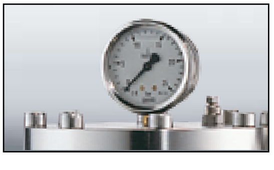


Shaft seal diameter
The shaft seal diameter must be selected to fit the pump shaft. If no seal with the required diameter is available, the shaft diameter can be changed with a bushing.

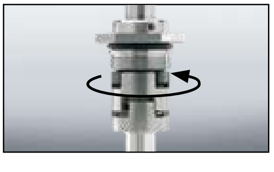
Type of pumped medium
The chemical resistance of the shaft seal materials to the pumped medium has to be considered.
The viscosity of the pumped medium affects the lubrication and leakage of the seal. The viscosity of most media depends on the temperature.
A single shaft seal can be used for a dynamic viscosity below 2500 cP (centipoise). For a higher viscosity, a back-to-back seal arrangement should be used.



Temperature
The elastomeric parts of the seal must be able to withstand the temperature of the medium around the seal. This might be different from the temperature of the pumped medium. If the temperature is above the boiling point of the pumped medium, lubrication is poor. This must be considered when selecting seal design and materials.



Sealing pressure
The sealing pressure is the pressure around the seal. For high pressures, a balanced seal should be used.



Shaft speed of rotation
If the speed of rotation is low, shaft seals with hard/hard material pairings might produce noise because the lubricating film in the seal gap is extremely thin. At speeds above 15 m/sec, a balanced seal with a rotating seat must be used to reduce seal unbalance.

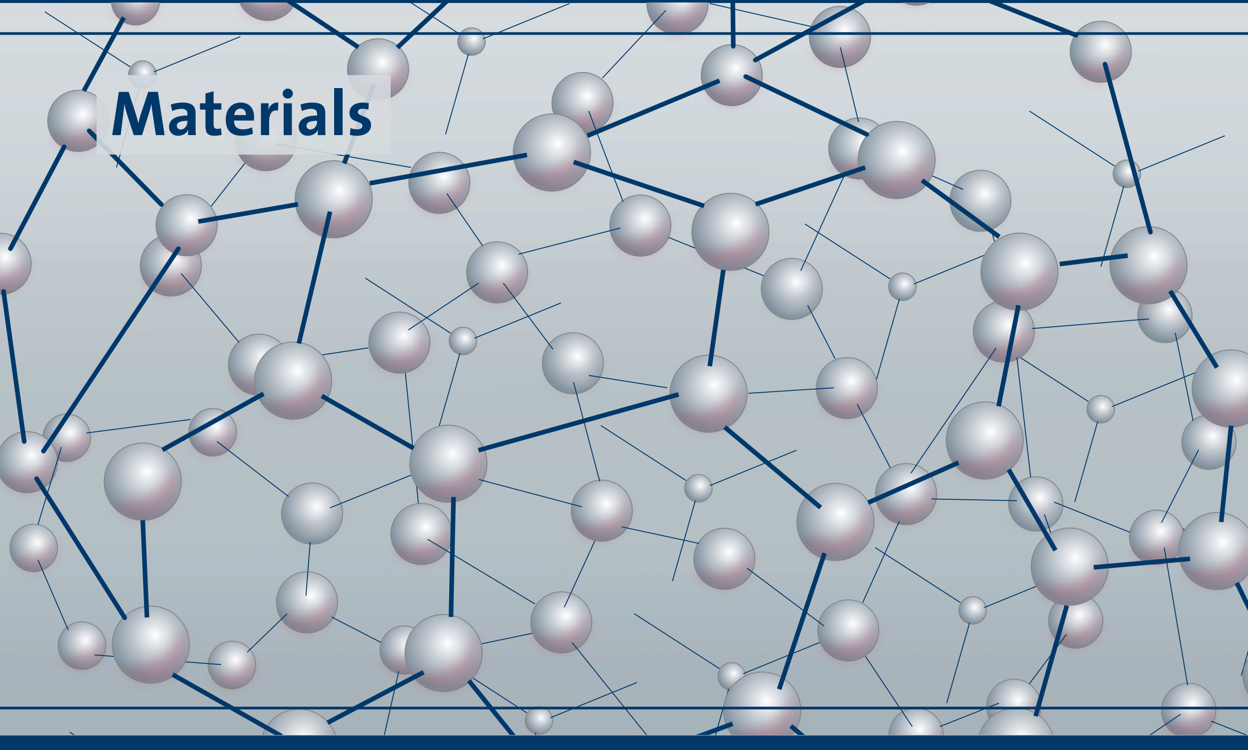
In addition to these operating conditions, the content of abrasives and additives in the pumped medium might be relevant to consider when selecting seal ring materials. In some instances, the space available for the shaft seal is also an important factor. When selecting the right sealing arrangement around the mechanical shaft seal, also take into account the content of abrasives and the risk of build-up of wearing particles on the atmospheric side as well as the health hazards, explosion risk and toxicity of the pumped medium.

Summary

Mechanical shaft seals can be composed in many different ways with different performances. The type of seal selected depends on the pumped medium, temperature, pressure and speed. For harsh applications, single mechanical shaft seals can be combined using tubes, membranes, springs, valves and separate cooling liquid systems.

Chapter 3

Materials



1. Seal face materials
2. Seal face material pairings
3. Testing of shaft seals
4. Secondary seals
5. Materials of other shaft seal parts

The preceding chapters have explained the composition and principle of operation of mechanical shaft seals. This chapter describes commonly used materials for the various parts of the mechanical shaft seal, including a number of tests of seals with different seal face materials.

1. Seal face materials

Few materials are suitable for seal faces. To keep leakage as low as possible, the seal gap must be very small. As a result, the lubricating film is very thin. Consequently, the seal face materials must be able to withstand rubbing against each other at high load and speed.
The best seal face materials have low friction, high hardness, good corrosion resistance and high heat conductivity.

The choice of seal face materials is decisive of the function and life of the mechanical shaft seal. In the following, commonly used seal face materials will be described.

Carbon graphite

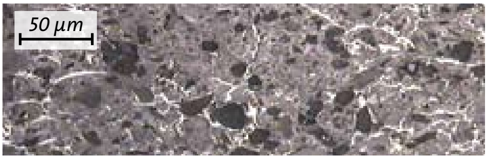


Fig. 3.1: Micrograph showing the material structure of antimony-impregnated carbon graphite

Fig. 3.2: Micrograph showing the material structure of resin-impregnated carbon graphite

Carbon graphite is a widely used seal face material thanks to its anti-friction properties.
The material is suitable as counter face material to many other types of materials.
Carbon graphite is a mixture of hard carbon and graphite.

Impregnated carbon graphite
Each carbon graphite manufacturer offers their own carbon graphite grades, depending on the source of the hard carbon, the graphite content, the grain size, mixing and baking.
After pressing and baking, the carbon graphite contains 5–20 % porosities. To obtain a leak-proof product, the carbon graphite must be impregnated, using metals or resins as impregnating agents.

The metals used for metal-impregnation are low-melting-point types such as antimony (Sb), tin (Sn), lead (Pb) or alloys of these products. See fig. 3.1.
According to EN 12756, the material code for this group is named A. See page 96.

Resin-impregnation often involves a phenolic resin. See fig. 3.2.
According to EN 12756, the material code for this group is named B.

For special purposes, resin-impregnated carbon graphite can be further heat-treated to convert the resin to carbon. It might prove necessary to repeat the impregnation and heat treatment process several times to obtain a leak-proof carbon-impregnated carbon.

Resin-bonded carbon graphite
Resins containing up to 70 % carbon-graphite fillers can be injection moulded and used without baking. The material is called “resin-bonded carbon”. The resin-bonded carbon has a lower wear and chemical resistance than the resin-impregnated carbon.

Properties
In vacuum, the friction of graphite is high whereas it is low under normal atmospheric conditions. In hot water applications (> 100 °C), metal-impregnated carbon graphite has a lower friction and higher wear resistance than similar types of resin-impregnated carbon graphite. The disadvantage of metal impregnation is the limited corrosion resistance.
In addition, a drinking water approval cannot be obtained with metal-impregnated carbon graphite, see Chapter 6.

The typical dry coefficient of friction value for carbon graphite against a hard seal face material is 0.1 - 0.2 under normal atmospheric conditions. The stiffness and toughness of carbon graphite is low. These properties must be taken into consideration when designing and mounting mechanical shaft seals. In cold, clean water, a mechanical shaft seal with one carbon graphite seal face has a lifetime of several years. However, if the seal is used in hot water or solids-containing water, the seal must be changed at regular intervals.

Aluminium oxide (alumina)

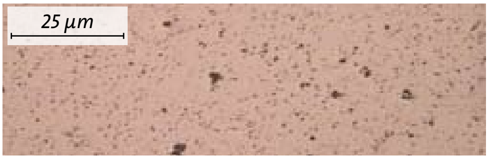


Fig. 3.3: Micrograph showing surface of alumina

Fig. 3.4: Micrograph showing etched surface of alumina

Aluminium oxide is a ceramic material, also known as "alumina". Alumina is commonly used as seal face material due to its good wear resistance and low price.
Each supplier offers his own grades of alumina with different compositions of glass phase and various grain sizes. See figures 3.3 and 3.4.
According to EN 12756, the material code for this group is named V.

Properties
The corrosion resistance in water is limited to a certain pH range, depending on the composition of the glass phase as well as on the purity. The best corrosion resistance is obtained with a 99.99 % alumina. However, the price of the material increases drastically with the purity. Alumina is only suitable for low-load applications due to its low thermal conductivity as compared to tungsten carbide and silicon carbide. Alumina is mostly used as counter face to carbon graphite. The stiffness of alumina is high, but the thermal shock resistance is limited.

Tungsten carbide (WC)

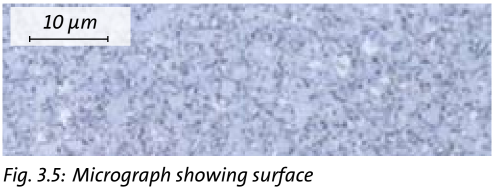


Fig. 3.5: Micrograph showing surface of tungsten carbide

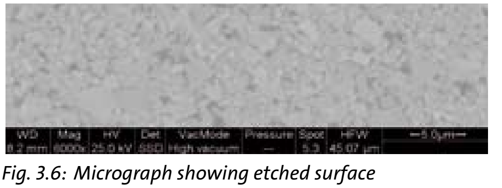


Fig. 3.6: Micrograph showing etched surface of tungsten carbide

Tungsten carbide (WC) is the designation of the type of hard metals based on a hard tungsten carbide phase and usually a softer metallic binder phase. The correct technical term of tungsten carbide is "cemented tungsten carbide". However, the abbreviated term "tungsten carbide" is often used for convenience, "cemented" being understood. See figures 3.5 and 3.6.
According to EN 12756, the material code for this group is named U.

Properties
The hardness of WC is below that of most ceramics, whereas the wear resistance of the material is superior, mainly due to its high toughness. WC is a heavy material with a density of approx. 14 g/cm^3. Cobalt-bonded (Co) WC is only corrosion-resistant in water if the pump is made of a non-inert material such as cast iron. The corrosion resistance of some chromium-nickel-molybdenum-bonded WC types is similar to stainless steel EN 1.4401 (AISI 316). WC with less than 0.5 % binder phase has the highest resistance to corrosion, although the material is not resistant in media such as water containing hypochlorite. Due to its extremely high wear resistance, WC is the preferred seal face material for applications involving abrasive particles.

Silicon carbide (SiC)

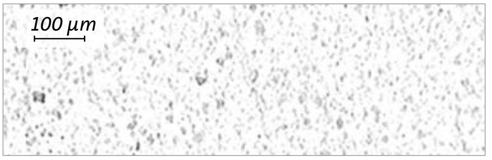


Fig. 3.7: Micrograph showing surface of dense silicon carbide

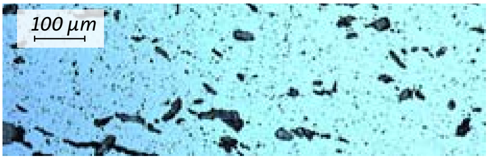


Fig. 3.8: Micrograph showing surface of graphite-loaded silicon carbide

SiC ceramics can be manufactured in many ways giving different properties.
According to EN 12756, the material code for this group is named Q. See figures 3.7 and 3.8.

The main SiC types are as follows:

- Direct-sintered.
  This SiC type is the most commonly used type for seal faces.
- Reaction-bonded.
  This SiC type has limited corrosion resistance in alkaline water due to the content of free silicon.
- Liquid-phase sintered.
  This SiC type has limited corrosion resistance in alkaline water due to the content of glass phase.
- Converted carbon graphite.
  This SiC type is manufactured from carbon graphite. It can be made as a thin SiC layer on the surface of the carbon graphite.

Properties

The direct-sintered SiC is brittle and requires careful handling. The material is light weight with a density of slightly above 3 g/cm³. The resistance to wear and corrosion is superior.
The direct-sintered SiC has a typical porosity below 2 %, but also grades with pores have been developed. The pores are discrete, non-interconnected and dispersed in a controlled manner throughout the body of the material.
The spherical pores act as fluid or lubricant reservoirs, helping to promote the retention of a fluid film at the interface of sliding component surfaces. This pore-based lubrication mechanism allows porous SiC to outperform conventional reaction-bonded and sintered SiC types in hot water.

Sophisticated sintering or the addition of different fillers can imply variations in these standard SiC grades. Fillers can be added to obtain improved electric conductivity, more toughness or lower friction.

Carbon or graphite inclusions can be used as dry lubricant to reduce friction. To use graphite inclusions successfully as lubricant, it is necessary to optimise the bonding between the SiC and the graphite as well as the size and amount of the graphite inclusions.

Diamond coatings

Diamond is the best known material for wear parts. Diamond has the highest hardness and thermal conductivity of any known material. In addition, it has an excellent corrosion resistance and a low friction. These properties make diamond the ideal material for seal faces. The major drawback of diamond is the price.

Diamond coatings have been commercialised during the last decade. Coatings can be made as polycrystalline diamond and as a more amorphous carbon called diamond-like carbon (DLC). The polycrystalline diamond has the lattice structure of diamond, where each carbon atom has four neighbour carbon atoms equally spaced (Sp^3 bonds). See fig. 3.9. In DLC coatings, some of the carbon atoms are located in structures similar to the diamond lattice.
Other carbon atoms are located in a structure similar to the lattice of graphite, which is hexagonal. See fig. 3.10.

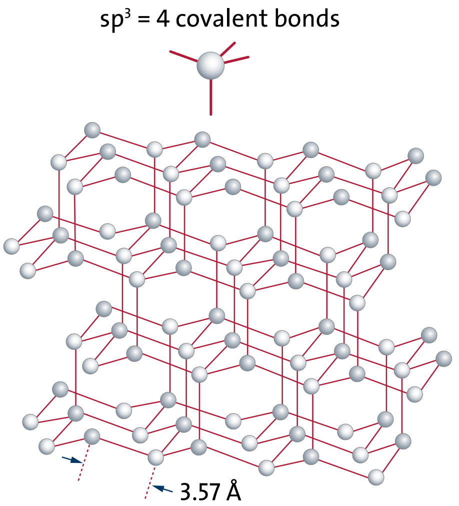


Fig. 3.9: Carbon atoms in the lattice structure of diamond (Sp^3 bonds)

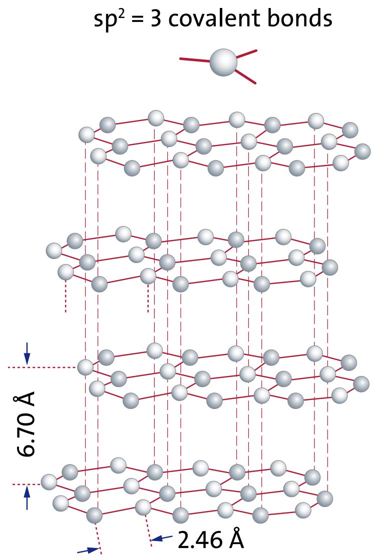


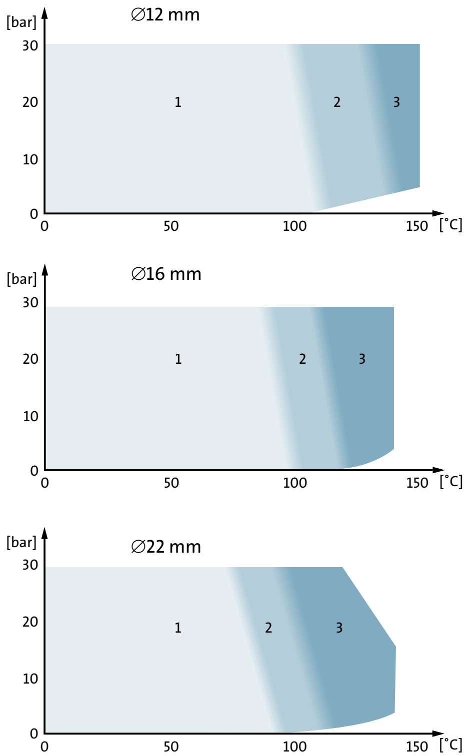
Fig. 3.10: Carbon atoms in the lattice structure of graphite (Sp^2 bonds)

Different variants of DLC coatings can be made, ranging in hardness from 1000 to 4000 HV.
The DLC coating thickness ranges from 0.1 to 10 μm and affects the production costs to a great extent. When the coating thickness is small, the adhesion to the substrate must be very strong to prevent delamination when the DLC coating is used on a seal face.
The best properties are obtained with thick polycrystalline diamond coatings on a hard substrate. However, if the counter face does not have a similar coating, it may suffer from wear.

2. Seal face material pairings

Carbon graphite against WC

Carbon graphite against WC is a widely used seal face material pairing.
The carbon graphite/WC pairing withstands dry running for several minutes without causing damage to the mechanical shaft seal. The corrosion resistance depends on the carbon graphite grade as well as on the alloying elements of the WC binder.
If the pumped medium contains hard particles, wear on the seal faces must be expected.
Due to the favourable lubricating properties of carbon graphite, the seal is suitable for use even under poor lubricating conditions, such as hot water. However, under such conditions, wear on the carbon graphite face reduces seal life.
The level of wear depends on factors such as pressure, temperature, pumped medium, seal diameter, carbon graphite grade and seal design. See fig. 3.11.
All materials pairrings performance diagrams in Chapter 3, refer to 3000 RPM.



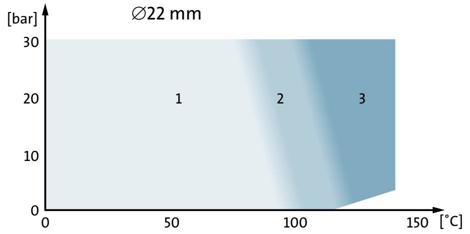
Pos.	Operating hours before wear-out
1	More than 14,000
2	8,000 - 14,000
3	4,000 - 8,000

Fig. 3.11: Pressure/temperature diagrams showing operating life of Grundfos type H carbon graphite/WC shaft seals in water at three different shaft diametres

Low speeds reduce lubrication between seal faces. This could have resulted in increased wear. However, due to the shorter running distance, the level of wear is unaltred in most cases.

Carbon graphite against direct-sintered SiC
Carbon graphite against SiC is another widely used seal face material pairing.
The corrosion resistance of the carbon graphite/SiC pairing is very good.
The dry running properties are similar to those of carbon graphite/WC.
The use of the carbon graphite/SiC pairing for hot-water applications may cause heavy wear on the SiC face, depending on the grade of the carbon graphite and the water.

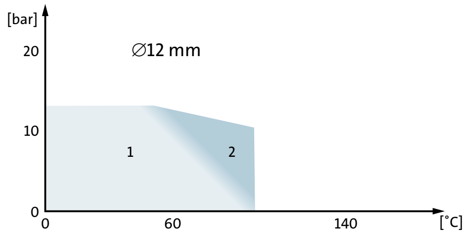
The use of porous or graphite-loaded SiC against carbon graphite causes far less wear than with dense SiC. See fig. 3.12.



Pos.	Operating hours before wear-out
1	More than 14,000
2	8,000 - 14,000
3	4,000 - 8,000

Fig. 3.12: Pressure/temperature diagrams showing operating life of Grundfos type H carbon graphite/SiC shaft seal in water for a ∅22 shaft

Carbon graphite against alumina
Carbon graphite against alumina is a widely used seal face material pairing for mass-production low-cost seals.
The corrosion resistance is often limited in water to a range between pH 5 and pH 10, depending on the alumina grade used.
The dry-running properties are similar to those of carbon graphite/WC, but the performance in hot water is much poorer. See fig 3.13.

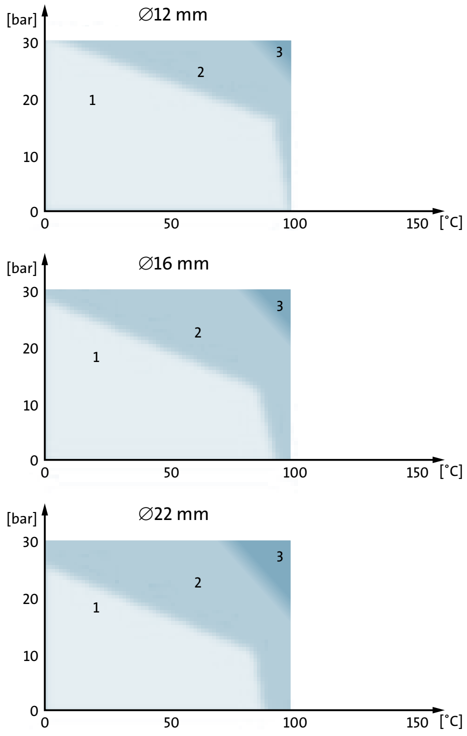


Pos.	Operating hours before wear-out
1	More than 14,000
2	8,000 - 14,000

Fig. 3.13: Pressure/temperature diagrams showing operating life of Grundfos type C carbon graphite/alumina shaft seal in water for a ∅12 shaft

WC against WC
A shaft seal with WC seal faces is extremely wear resistant. Being very robust, WC resists rough handling.
The dry friction of WC against WC is high. Consequently, the WC/WC shaft seal material pairing has poor dry-running properties. A shaft seal with WC/WC seal faces running completely dry may be damaged within less than one minute of dry running.

If certain pressure and temperature limits are exceeded, the seal may generate noise. The noise is an indication of poor lubrication, causing wear of the seal in the long term. The limits of use depend on seal diameter and design.
The pressure/temperature diagrams of the various seals show areas where noise may occur. See fig. 3.14.



Pos.	Range
1	Good performance
2	Risk of periodical noise in connection with start-up and variations in pressure and temperature
3	Periodical noise

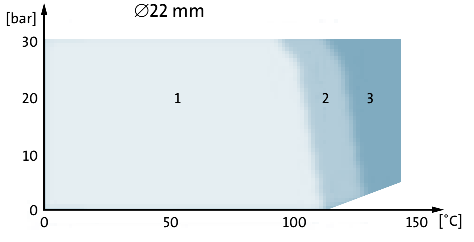
Fig. 3.14: Pressure/temperature diagrams of Grundfos type H WC/WC shaft seals in water showing performance range for three different shaft diameters

Note: The running-in wear period with noise of a WC/WC seal face material pairing is up to four weeks. However, typically, no noise occurs during the initial operating days due to higher leakage

SiC against SiC

Being an alternative to WC/WC, the SiC/SiC material pairing is used where higher corrosion resistance is required. This material pairing has good resistance against abrasive particles due to the high hardness. The friction is high, but for some SiC grades containing solid lubricants, the friction is only half, giving some improvement of the dry-running properties. Seals incorporating these SiC grades may be capable of running several minutes without being lubricated by the pumped medium.

The performances in hot water of seals incorporating porous SiC grades or SiC grades contaning solid lubricants can be seen in fig. 3.15.



Pos.	Range
1	Good performance
2	Risk of periodical noise in connection with start-up and variations in pressure and temperature
3	Periodical noise

Fig. 3.15: Pressure/temperature diagram of Grundfos type H SiC/SiC shaft seal in water showing performance range for a ∅22 shaft

3. Testing of shaft seals

Various types of simple testing configurations, such as ring-on-ring or even pin-on-disc, can be used to evaluate whether a material is suitable for a machanical shaft seal. Such tests give information about the tribological performance of materials and may even reveal wear processes in the tribological system. To get an accurate picture of the performance of a shaft seal, the tests must be made under conditions resembling the application for which the seal is intended.

Seal performance in hot water

The lubrication of the seal faces in hot water is limited. This is due to the low viscosity of water at high temperatures as well as to the evaporation in the seal gap. The temperature and pressure limits for shaft seals can be obtained by means of extensive testing.

Above these limits, noise from the seals may be expected and fatigue wear may occur.

The pressure-temperature diagram, fig. 3.16, shows how the limits of good performance change with the velocity. At lower velocities, the limits shift towards lower temperatures.

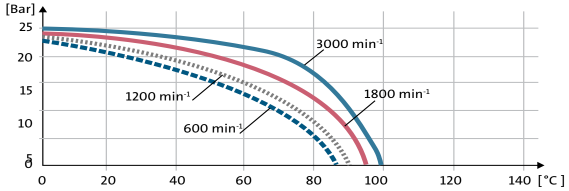


Fig. 3.16: Example of limits of stable friction of a shaft seal at different velocities

The hot-water tests are performed in tap water. At pressures and temperatures below the relevant curve with stable friction, the seal faces are exposed to a minimum of wear. Some wear may be expected above the relevant curve. See fig. 3.16.

Another way of showing temperature limits is to plot the wear rate as a function of the temperature at a fixed pressure. See fig. 3.17.

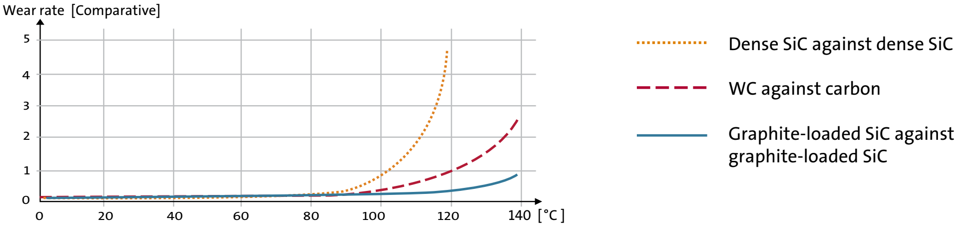


Fig. 3.17: Comparative wear rate of seal faces with three different material pairings

Dry running

Dry running may cause serious damage to the seal.
As it may be difficult to avoid dry running altogether in some applications, it is important to test the dry running performance of the seal. This can be done in a very simple way by running the shaft seal completely dry with a thermocouple attached to the seat or with thermographic equiment. The results obtained are slightly affected by the relative humidity of the air in the test room.
Fig. 3.18 shows the temperatures measured on the seat of various dry-running seals.

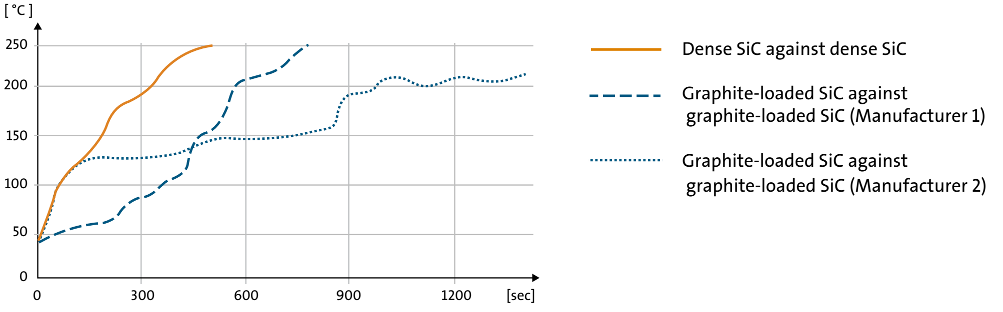


Fig. 3.18: Temperature on seat at dry running of seals with different SiC grades

As will be seen from fig. 3.18, dense SiC against dense SiC and graphite-loaded SiC aginst ifself (manufacturer 1) show poor dry-running performance, similar to WC against WC. The graphite-loaded SiC against itself (manufacturer 2) shows better dry-running performance.
Dry-running tests show large variations, even within the same grade of SiC.

Seal performance in water containing abrasive particles

If both seal faces are made of hard materials such as ceramics, wear on the seal faces caused by abrasives are rarely observed. The seal gap in a mechanical shaft seal is typically below 0.3 micron. Theoretically, this means that only particles below 0.3 micron can enter the seal gap. In practice, the edge of a seal face is not completely sharp. Consequently, particles measuring a few microns are able to enter the seal gap. Normally, such small particles only cause a polishing wear on a hard/hard seal face material pairing.

When one of the faces is a carbon ring, the edge of the seal face will wear and permit larger particles to enter the seal face.
Such larger particles can be trapped in the carbon seal face and cause wear on the hard counter face.

Seal performance in water containing glycol

Water containing glycol may cause problems with leaking seals. The problems often arise due to additives such as inhibitors, antioxidants, alkalines, etc. Some additives such as silicates and phosphates may crystallize in the seal gab as hard particles. These hard crystallites cause wear on seals with one carbon face. See fig. 1.24a, page 20.

Organic film binders, so-called inhibitors, adhere to all surfaces in contact with the liquid, including a major part of the seal face. Many inhibitors may build up sticky layers in the seal gap, resulting in leakage. Seals with WC/WC or SiC/SiC seal faces have better self-cleaning properties than seals with a carbon/SiC seal face material pairing. A high closing force and a narrow width of the seal face reduces the risk of build-up of deposits. See type G, page 28, and type R, page 29.
Fig. 3.19 shows the results of tests made with various seal face material pairings in water containing glycol with a high content of additives.

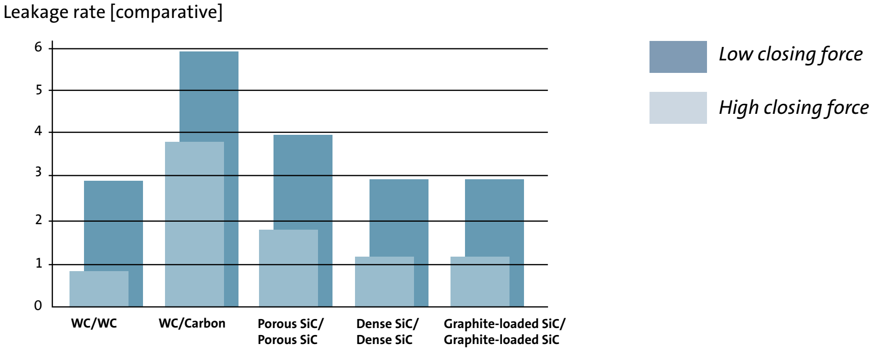


Fig. 3.19: Leakage of seals with different material pairings running in water-based anti-freeze liquid

To prevent a large seal gap with excessive leakage, a smooth surface finish is preferred. On the other hand, if the surface finish is too smooth, seizure of the seal faces may occur. Consequently, a compromise is often made with a different surface roughness of the two seal faces.
The leakage rate of hard/hard material pairings is elevated until the seal faces have become smooth as a result of the running-in.
Seals with one carbon seal face often have a lower, accumulated leakage during the running-in period because this period is shorter as compared to a seal with hard/hard material pairings.
Seals with a high closing force have a shorter running-in period because the lubricating film is thinner.

Seal performance in pure water
Pure water can be aggressive to many ceramics. As far as direct sintered SiC is concerned, the grain boundaries containing sinter additives may be attacked in pure water.
Damage is only observed on seal faces where high shear stresses may be achieved in asperity contacts.
By controlling the sintering process, it is possible to achieve SiC grades that are more resistant in pure water.

Fig. 3.20 shows the result of tests with dense SiC grades in 40°C demineralised water with a conductivity of 2 µS/cm. Special corrosion-resistant SiC grades show no failure during 11,000 hours of testing under these conditions.

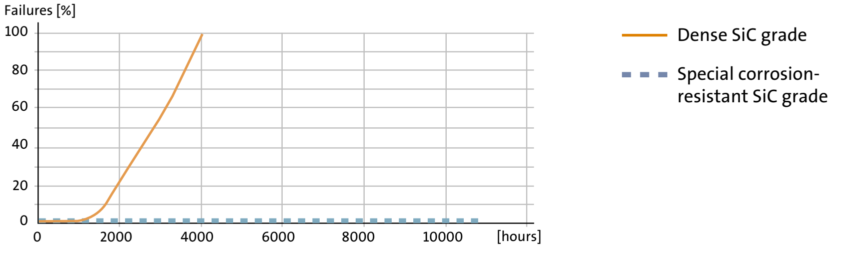


Fig. 3.20: Failure of SiC seals in demineralised water with a conductivity of 2 µS/cm

Sticking of seal faces

Very smooth and flat seal faces easily adhere to each other. In extreme cases, the adhesion is so strong that it prevents the motor from starting. Alternatively, it might cause the stationary seat to rotate in the secondary seal.

Various mechanisms act on the adhesion between the seal faces.

Physical adhesion

Vacuum may occur when two flat and smooth surfaces are pressed tightly together. Consequently, a large axial force is required to separate the two surfaces, while a lower shear force is required to rotate the surfaces. The size of the shear force at start-up is equal to the force required for a very low rotational speed. See fig. 3.21.

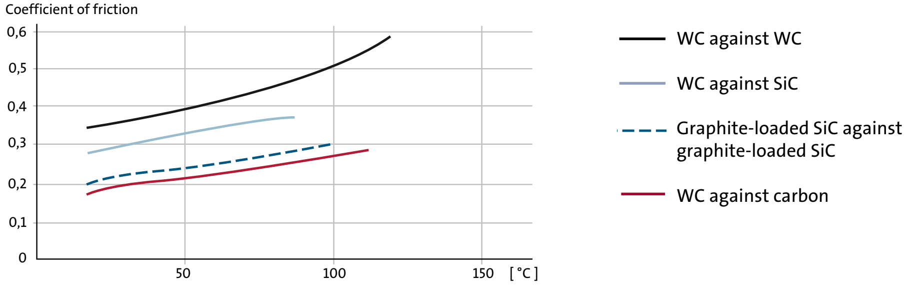


Fig. 3.21: Coefficient of friction of different seal face material pairings in water, at low rotational speed

Chemical adhesion of surfaces
All surfaces subjected to the atmosphere have an oxide layer. See fig. 4.12, page 72.
The equilibrium of the oxide layer may change when the surface gets into close contact with another surface or when it is exposed to the pumped medium. This change in equilibrium may involve chemical bindings to oxides from other surfaces. The more inert the oxide layer is to the surroundings, the weaker are the bindings to the counter surface.
If the medium is aggressive to the seal face material, the corrosion products from the seal faces may form chemical bonds, resulting in high adhesion forces.
To prevent such adhesion mechanisms, highly inert seal face material types such as SiC are preferred.

Chemical adhesion involving adhesive agents
If the pumped liquid contains ions that may precipitate on the seal face, the precipitations may act as glue between the seal faces.
This adhesion mechanism may occur in hard water and liquids containing fugitive elements and can be reduced by using a carbon/SiC seal face material pairing.
Solid lubricant-loaded SiC materials also reduce adhesion because the solid lubricant is smeared in a thin layer on the seal faces, providing low shear forces.

4. Secondary seals

As mentioned, it is important to choose the most suitable seal face material pairing to obtain the longest seal face life. Likewise, the secondary seals such as O-rings and bellows made of elastomer are essential for the right functioning and overall life of the mechanical shaft seal.
Elastomers refer to polymers with a high degree of elasticity. The material is also known under the term “rubber”.
Elastomers are the preferred choice of material for secondary seals due to their elastic properties. All these materials remain flexible within the operating range of temperature for the chosen mechanical shaft seal. The choice of elastomer is mainly based on the chemical composition and the temperature of the pumped medium. Besides, product approvals should be considered, see page 97.
For an overview of temperature and chemical resistance of elastomeric materials, see fig. 3.22.

The below-mentioned elastomers are used in mechanical shaft seals:

NBR
Acrylonitrile-butadiene rubber (NBR) belongs to the family of unsaturated copolymers.
Varying the composition with more acrylonitrile increases the resistance against oil, but reduces the flexibility. Compared to natural rubber, NBR is more resistant to oil and acids.
According to EN 12756, the material code for this group is named P. See page 96.

HNBR
Hydrogenerated acrylonitrile-butadiene rubber (HNBR) has the same good oil resistance as NBR and also good resistance to ozone, alkalis and amines. HNBR has a higher temperature limit in water than NBR.

MVQ
Silicone rubber covers a large group of materials in which methyl vinyl silicone (MVQ) is the main material. Silicone elastomers as a group have relatively poor tensile strength and poor wear and tear resistance. However, they have many special properties. Silicone in general has good heat resistance up to +230 °C and good cold flexibility down to –60 °C and good weather resistance. According to EN 12756, the material code for this group is named S.

EPDM
Ethylene-propylene diene monomer (EPDM) can be compounded to give many specific properties by varying the content of dicyclopentadiene, etylidene and vinyl norbornene. Compared to NBR, the material has very poor resistance to mineral oil, but excellent resistance to hot water. EPDM has a good resistance to polare liquids and poor resistance to apolare liquids. According to EN 12756, the material code for this group is named E.

FKM
Fluoro-carbon monomer (FKM) belongs to a family of rubbers designed for very high temperatures in many different liquids, due to the degree of fluorination. The material has poor resistance to hot water, but excellent resistance to oils and chemicals. FKM has poor resistance to polare liquids and good resistance to apolare liquids.
According to EN 12756, the material code for this group is named V.

FXM
Flourinated copolymer (FXM) has a good chemical resistance and withstands a wide temperature range in hot water applications.

FFKM
Perflouroelastomer (FFKM) has the best chemical resistance of any known elastomeric material. The chemical resistance of FFKM resembles that of polytetraflouretylene (PTFE), and the elastic properties resemble those of rubber. The material solves many difficult sealing problems.
FFKM is very expensive and can only be made in relatively simple geometries.
According to EN 12756, the material code for this group is named K.

Pumped medium	Elastomer						
	NBR	HNBR	MVQ	EPDM	FKM	FXM	FFKM
Water, max. temp. [°C]	80	100	120	140	90	275	230
Mineral oils, max. temp. [°C]	100	150	120	-	200	275	230
Acids	+/-	+	+/-	+/-	+/-	+/-	+
Alkalis	+	+	+/-	+	-	+	+
Glycols	+	+	+	+	-	+	+
Oils, fuel	+/-	+/-	+/-	-	+	+/-	+
Solvents	-	-	-	-	+/-	-	+
Abrasive particles	+/-	+	-	+	+/-	+/-	-

Legend: + = excellent   +/- = good under certain conditions   - = poor   - = disastrous

Fig. 3.22: Overview of temperature and chemical resistance of elastomeric materials

5. Materials of other shaft seal parts

Besides seal rings and elastomeric parts, the other parts of the mechanical shaft seal must also be selected according to the application. The number of parts of the mechanical shaft seal depends on the complexity of the seal design.

Torque transmission parts
Metal or polymer parts can be used to transfer the torque from the seal faces to the shaft and pump housing. This is of particular importance in case of hard/hard material pairings producing a large friction torque. Metal parts are often made of stainless steel with a corrosion resistance similar to or above the level of the other pump parts. Polymers or formed sheet metal is often used for mass-produced mechanical shaft seals. Powder-metal parts can be used for minor series and machined parts for small quantities.

The mechanical shaft seal can be fastened to the shaft in different ways, but the most common is by means of small screws made of stainless steel or compression fitting.

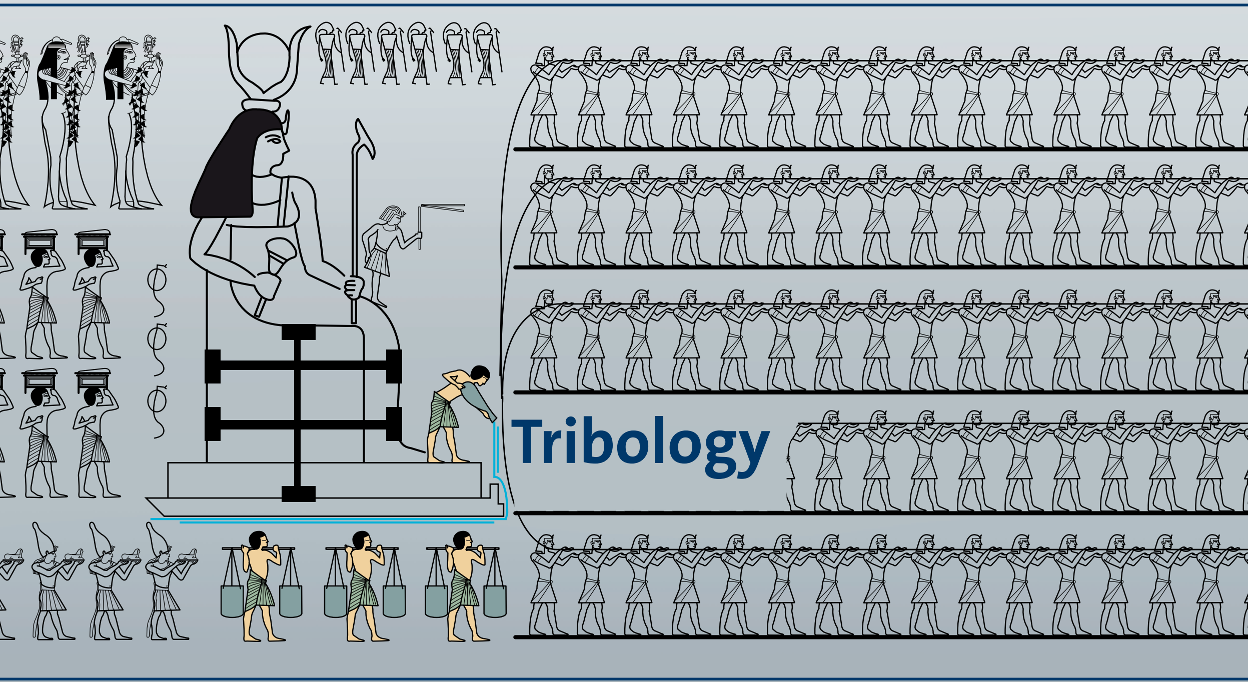
Springs and bellows
Metal springs are used to press together the seal faces of O-ring shaft seals and rubber-bellows seals. Alloys of various levels of corrosion resistance are available.
The bellows of the bellows seals can be used to provide the force that presses the seal faces together. This is very common for metal bellows, but also applies to polymer bellows and rubber bellows. Metal bellows are made of very thin sheet material. They are often cold-worked to obtain high yield strength. The material grain size must be small compared to the thickness of the bellows.
The corrosion resistance grade of materials used for the bellows of above types must exceed that of other pump parts.

Guiding elements
High-pressure mechanical shaft seals may incorporate polymer or metal discs to minimise the gap between the rotating seal face and the shaft/sleeve. This reduces the risk of extrusion of the O-ring. See fig. 5.19, page 86.
Bellows seals may incorporate polymer or metal guiding elements to centre the rotating seal ring on the shaft.

Tubes, plugs and holders
Tubes and plugs for cartridge seals can be made of metal or rigid polymers. This also applies to holders for O-rings and clamping rings for rubber parts. The material chosen depends on the corrosion resistance, strength and dimension stability required as well as the number of identical seals to be made.

Summary
Materials for mechanical shaft seals must be chosen according to the applications.
Chemical resistance, temperature range and approvals must be considered.
For seal face materials, the friction and wear properties are very important.

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.

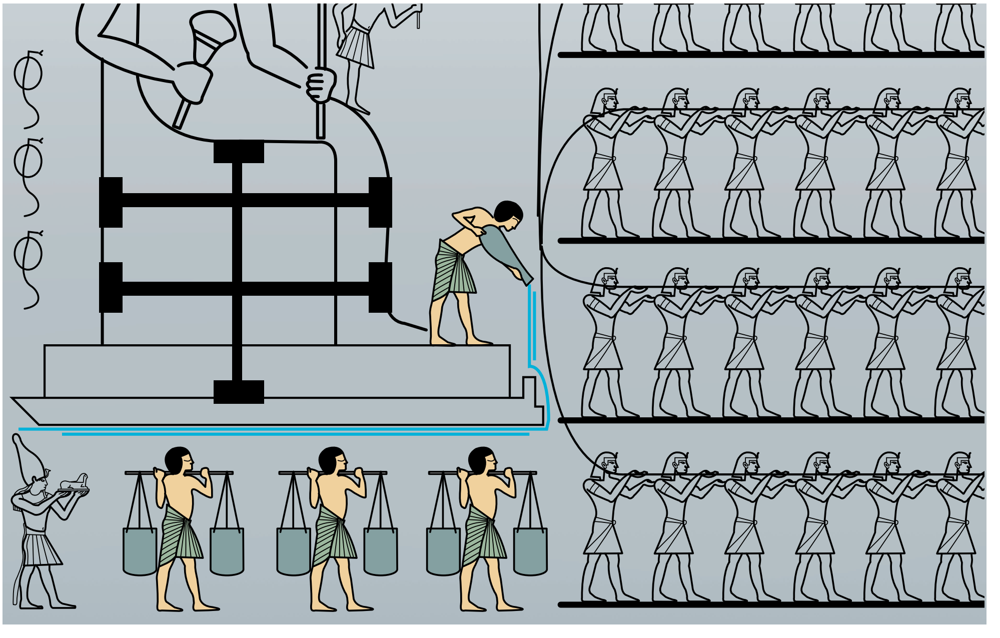


Fig. 4.1: Assuming the slege is made of wood sliding against wood, it can be calculated that this only is possible when lubricated with water. The man standing on the sledge is pouring water under the sledge to lubricate, and three more slaves are bringing water to the tribologist.

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.

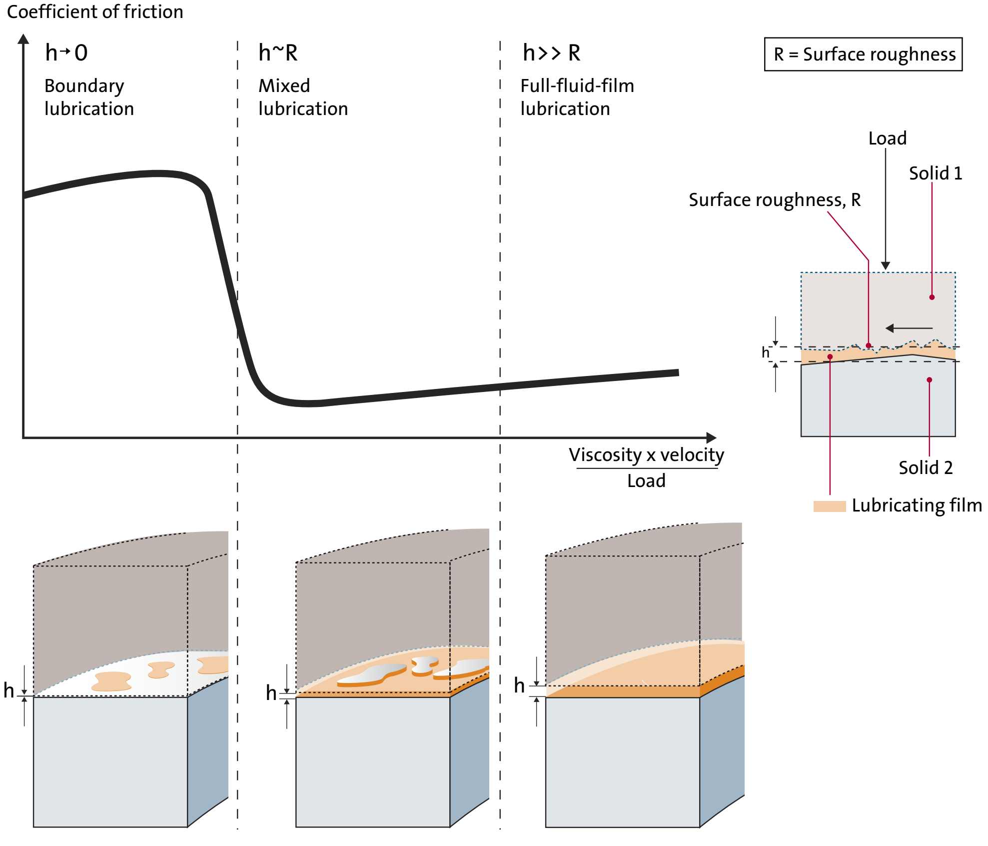


Fig. 4.2: Stribeck curve showing different lubrication regimes

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:

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

η : dynamic viscosity
ω : angular velocity ($2\pi\ \nu$ [sec^{-1}])
k : balancing ratio of the seal
Δp : 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^{-8}, 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, η,(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, η, 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.

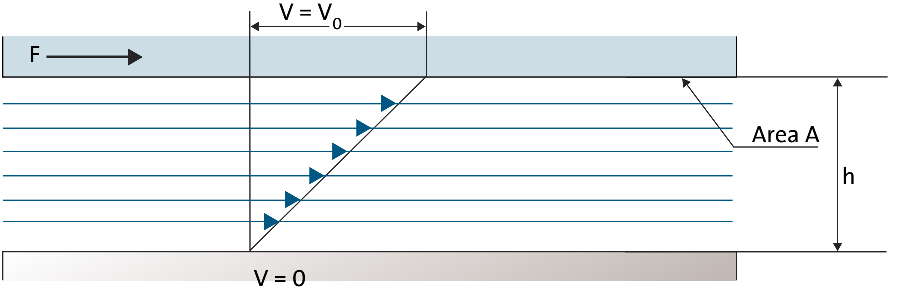


Fig. 4.3: Velocity distribution and shear resistance, F, of a fluid film between two surfaces, h being the distance between the surfaces

Thus, the shear stress F/A is proportional to the change of shear strain, v_0/h:

$$F/A = \eta\, v_0/h$$

Or more generally with τ as shear stress:

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

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.

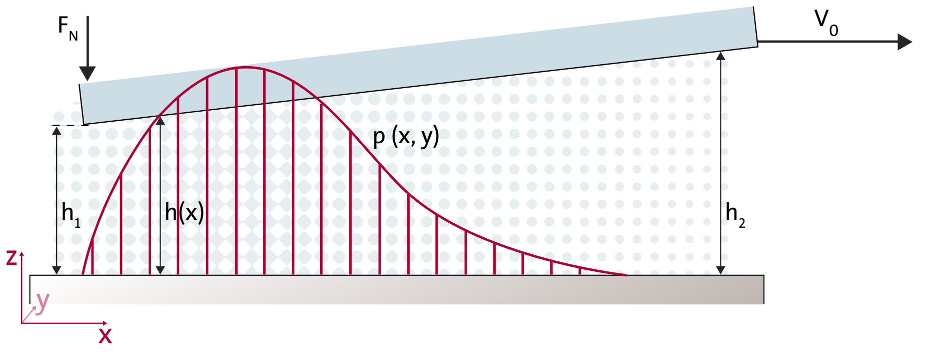


Fig. 4.4: Slightly tilted moving surface creating a pressure profile

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.

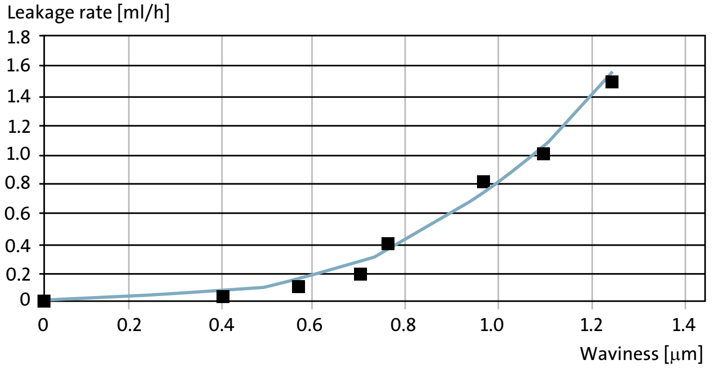


Fig. 4.5: Example of measured leakage as a function of waviness

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.

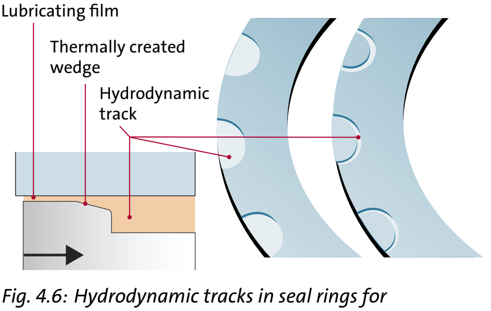


Fig. 4.6: Hydrodynamic tracks in seal rings for hot water

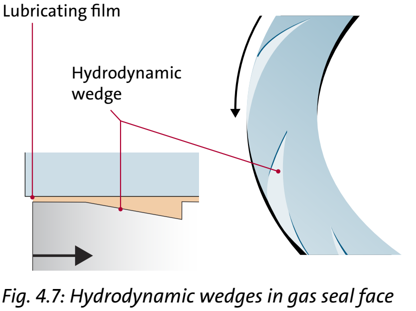


Fig. 4.7: Hydrodynamic wedges in gas seal face

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.

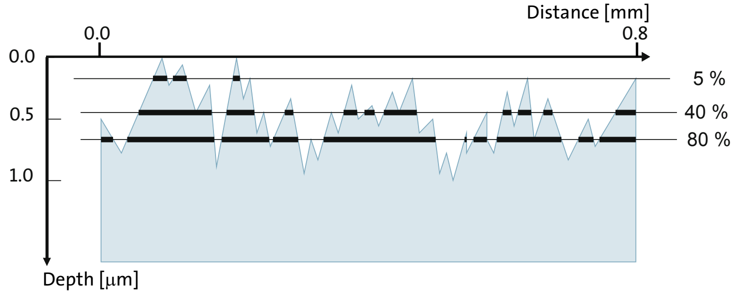


Fig. 4.8: Cross-section of surface showing how a BAC is obtained

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

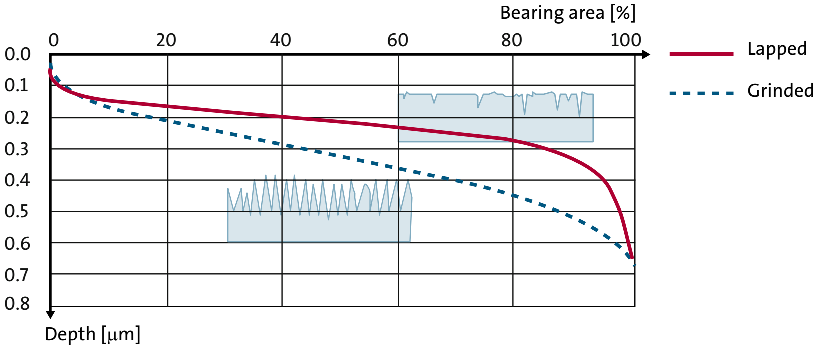


Fig. 4.9: Examples of BACs for a grinded and a lapped surface

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.

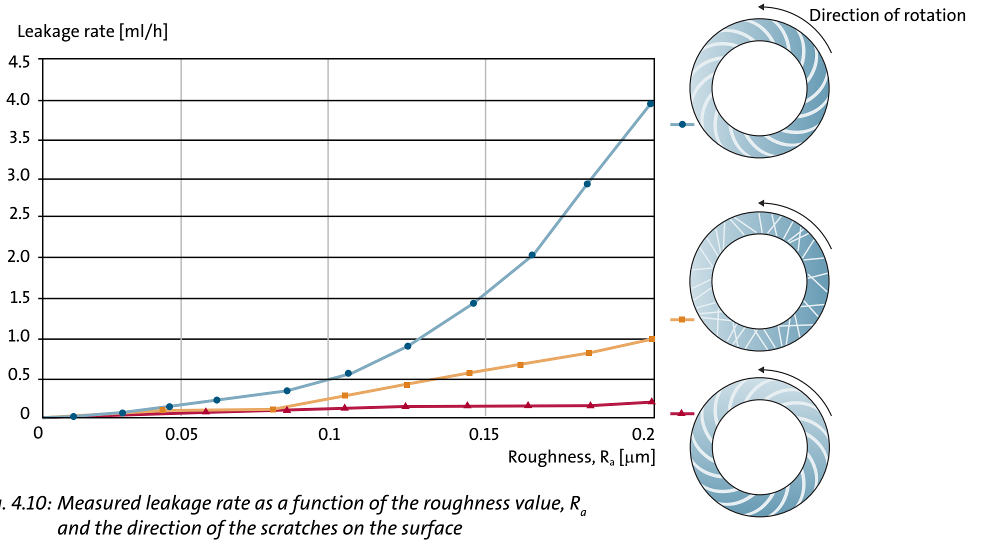


Fig. 4.10: Measured leakage rate as a function of the roughness value, R_a and the direction of the scratches on the surface

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].

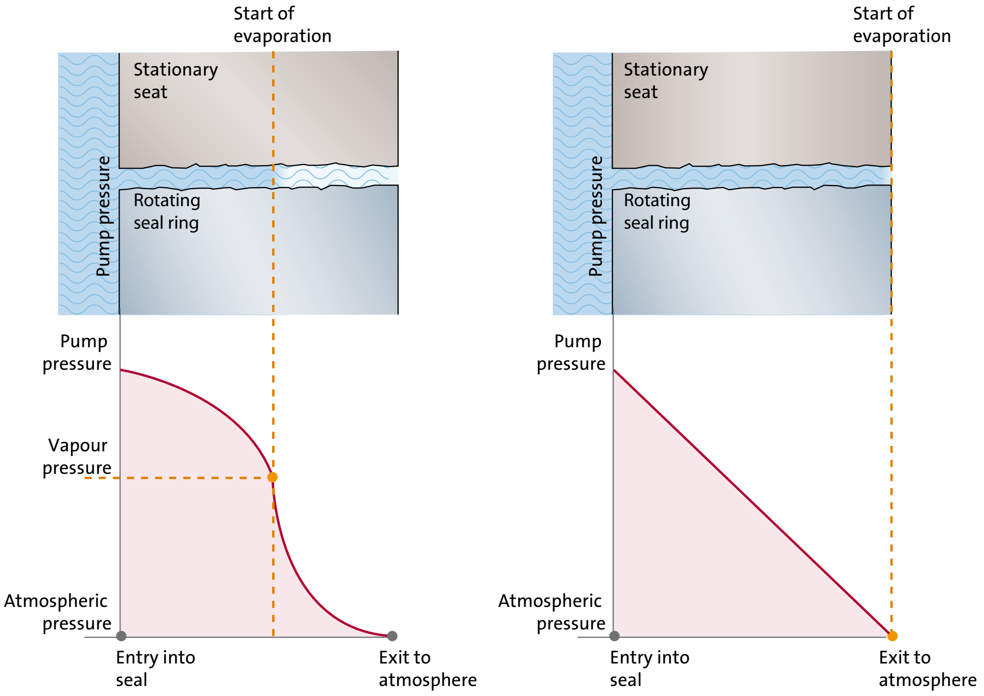


Fig 4.11: Evaporation in the seal gap can increase the pressure in the gap because the pumped medium expands when it evaporates.

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.

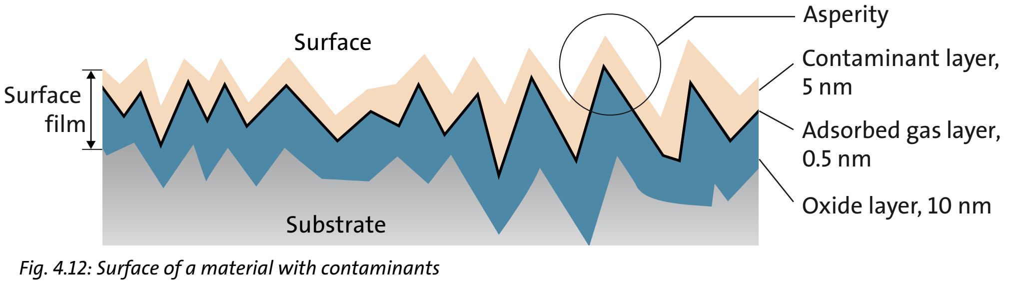


Fig. 4.12: Surface of a material with contaminants

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.

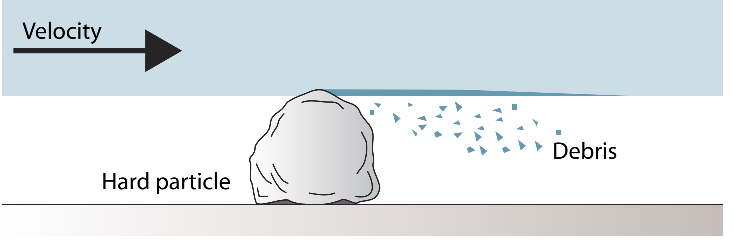


Fig. 4.13: Three-body abrasion

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.

[1] B.S. Nau: " Hydrodynamic Lubrication in Face Seals", 3th Int. Conf. on Fluid Sealing (1967).

[2] Bernard J. Hamrock: "Fundamentals of Fluid Film Lubrication".

[3] A.O. Lebeck: "Face Seal waviness, prediction, measurement, causes and effects", 10th Int. Conf. on Fluid Sealing (1984).

[4] L.E. Hershey: "Extending Mechanical Seal Service Life when operating in hot Water", 7th Int. Conf. on Fluid Sealing (1975) .

[5] P. Waidner: "Evaporation in the gap of face seals – Theoretical calculations and results for hot water applications", 12th Int. Conf. on Fluid Sealing (1989).

[6] J.T. Burwell: "Survey of possible wear mechanisms", Wear 1 (1957), p 119 – 141.

[7] I. Etsion: " A Model for Mechanical Seal with Regular Microsurface Structure", Tribology Transactions, vol. **39** (1996), p 677-683.

Chapter 5

Failure of mechanical shaft seals

1. Introduction to failures

2. Lubrication failures

3. Contamination failures

4. Chemical, physical degrading and wear

5. Installation failures

6. System failures

7. Shaft seal failure analysis

1. Introduction to failures

Failure of the mechanical shaft seal is the most common cause of pump downtime. The shaft seal is exposed to widely varying operating conditions. Sometimes operating conditions change to become quite different from the specific conditions for which the seal was intended.

The diagrams below show that shaft seal failure is by far the most common cause of pump system failure. See figures 5.1 and 5.2.

Causes of pump failure **Pump repair costs**

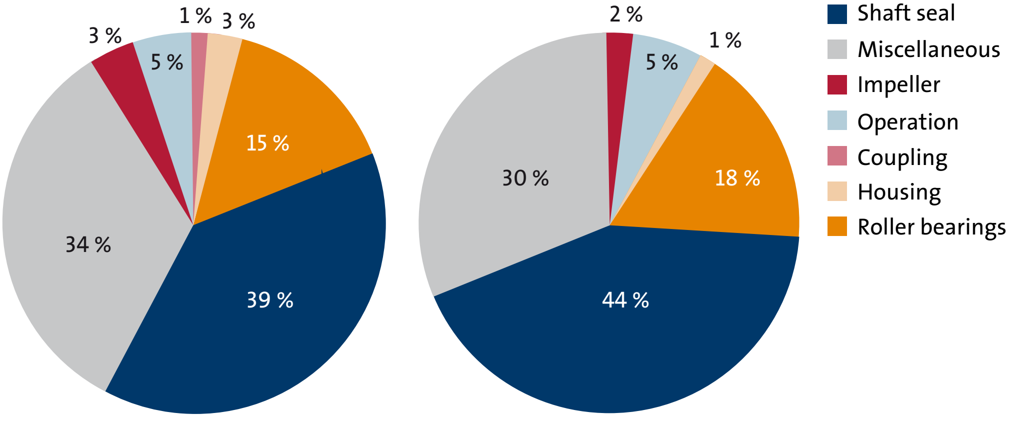


Fig. 5.1: Analysis of pump failure. Mechanical seals account for 39 % of pump failures. [1]

Fig. 5.2: Analysis of pump repair costs. Mechanical seals account for 44 % of pump repair costs. [1]

Typical shaft seal failures depend highly on the seal type and material pairing. O-ring-type shaft seals with dynamic O-ring and one seal ring in carbon-graphite typically have problems with wear on seal faces and seal hang-up, which prevents axial movement of the dynamic O-ring and seal ring. Mechanical shaft seals with hard/hard seal face material pairings usually experience problems associated with dry running.
Detailed information is required to make a failure analysis of a damaged shaft seal and subsequently prevent new failures. On the damaged shaft seal, it is possible to observe what is damaged, but the reason for the damage must often be found on the basis of knowledge of the pump and the application. This information should be recorded in a damage report, including details on the operating conditions of the equipment and components around the shaft seal.
See shaft seal failure analysis, pages 89-91.

In the following, examples of common causes of failure of mechanical shaft seals will be discussed.

2. Lubrication failures

Proper functioning of mechanical shaft seals with hard/hard seal face material pairings depends on lubrication by the pumped medium. Dry running and poor lubrication can produce the results described below.

Dry running

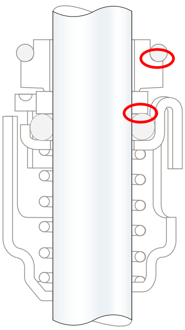
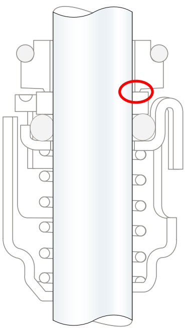


Fig. 5.3: Surface damage due to high temperature on EPDM and FKM secondary seals

Dry running occurs when there is no liquid around the seal, either due to the absence of pumped medium in the pump or poor venting, resulting in the formation of air around the seal. The absence of lubricating film causes the friction between the seal faces to increase. Consequently, the temperature rise dramatically.
As there is no pumped medium in contact with the seal rings, the heat must be transported away through the seal. Many seals with hard seal faces reach a temperature of several hundred degrees Celsius on the seal faces within few minutes. The typical damage caused by dry running is burnt elastomeric parts. The damage occurs where the O-ring is in contact with the hot seal ring. See fig. 5.3.

Poor lubrication



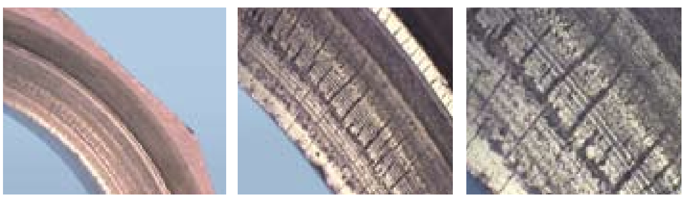


Fig. 5.4: Severe thermal cracks in WC seal face caused by poor lubrication

Similar to dry running, the frictional heat generated on the seal faces as a result of poor lubrication may also cause problems. Poor lubrication may occur when the viscosity of the pumped medium is very low or if the temperature is well above the boiling point at atmospheric pressure.
Under these conditions, the frictional heat dissipating in small areas on the seal face can be very high. The alternating local heating and cooling of the seal faces may cause small, radial, thermal cracks in the seal faces. See fig. 5.4.

Noise

When lubrication is poor or totally absent, shaft seals with seal rings made of hard materials tend to generate a loud noise. Depending on the seal design, the hard materials used and the system, the noise can be at a constant level of intensity and frequency or be more random.
When noise is generated from the mechanical shaft seal, some parts of the seal vibrate. This may reduce the life of the seal. Metal bellows seals in particular have a tendency to fatigue on account of vibrations.

3. Contamination failures

The pumped medium is often a mixture of miscible liquids and a solution of solids, in addition to small suspended insoluble particles.
The lubricating film in the sealing gap is subjected to large gradients in temperature, pressure and velocity. This increases the risk of precipitation and sedimentation in or near the sealing gap.

Hang–up

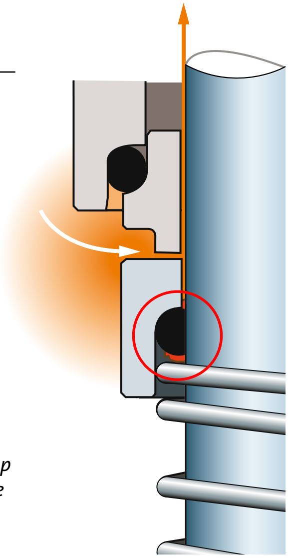
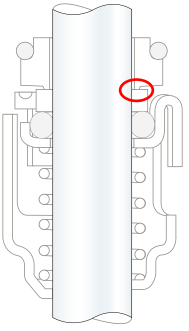


Fig. 5.5: Sketch of seal hang-up due to blocking of the axial movement of the dynamic O-ring

Hang-up of a mechanical shaft seal means that the axial movement of the rotating part of the shaft seal is blocked. Hang-up mainly occurs in connection with O-ring-type seals, but is also seen in connection with bellows seals, although the underlying mechanism is different.
In connection with O-ring-type shaft seals, settlements or precipitations may build up on the shaft beside the O-ring, preventing the O-ring from sliding freely. When the temperature or pressure in the system change, the dimensions of pump parts change likewise. As a result, the O-ring must be able to slide freely on the shaft or sleeve to continue to function correctly. See fig. 5.5.

If a rubber bellows seal is operated at temperatures close to the permissible maximum limit of the rubber type and close to the maximum permissible pressure limit of the seal, the inner surfaces of the bellows tend to stick to the shaft. This results in hang-up. The same failure mechanism results in hang-up of O-ring-type shaft seals, as the dynamic O-ring may stick to the shaft. Some rubber types such as FKM have a high tendency to stick to stainless steel.
A hang-up failure is not always possible to observe during an analysis, as the seal has already been disassembled.

Opening of the sealing gap



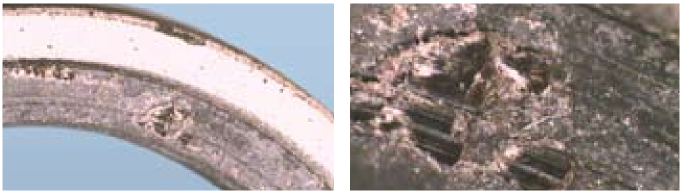


Fig. 5.6: Deposits on carbon graphite seal face

Some suspensions and solutions tend to cause build-up of scattered deposits on the seal faces.
As the deposits only cover part of the seal faces, the sealing gap opens correspondingly. The result is a leaking shaft seal. The leakage is small at the beginning, accellerating as more liquid passes the sealing gap. The settlement accelerates, because the temperature is higher on the surface of the already anchored deposits. See fig. 5.6.

Clogging

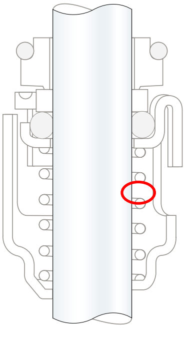
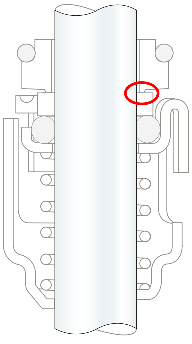


Fig. 5.7: Metal bellows seal clogged by lime scale build-up

When the pumped medium has a high content of suspended particles and fibres, the seal can fail due to precipitation or consolidation of the particles and fibres on or at springs, bellows, seal-drivers or O-rings.
The sedimentation rate is affected by the pumped medium and the flow conditions around the seal.
In extreme cases, the sedimentation on metal bellows shaft seals may prevent the axial spring action of the bellows. Subsequently, the seal can open when the operating conditions are changed. See fig. 5.7.

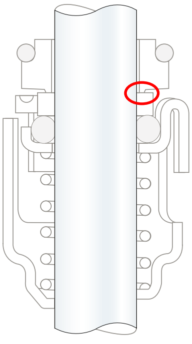
If a change in operating conditions necessitates an axial compression of a bellows clogged by sediments, the closing force of the seal can be extremely high. This may result in excessive mechanical stress on the seal components or failure due to poor lubrication.

Particles and deposits



Small amounts of hard particles on the seal faces result in increased wear, especially when using hard/soft seal face material pairings. In such cases, small, hard particles might be squeezed into the soft seal ring from where they act as a grinding tool on the hard seal ring.
Impurities between seal faces result in a high leakage rate, permanently or until the impurity has been grinded and flushed away. As an example, a human hair is 50 to 100 µm thick, as compared to the 0.3 µm (height) of a sealing gap operating under normal operating conditions. Thus, a human hair of 60 µm is 200 times the width of a normal sealing gap. The leakage rate is proportional to the sealing gap raised to third power, see formula 2, page 18. If a human hair is trapped between seal faces, the leakage rate will be 200^3 = 8,000,000 times the leakage rate of a shaft seal with clean seal faces.

Sticking/seizure



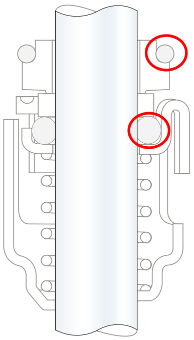
Sticking (British) and seizure (American) are two different terms with the same meaning. In the following, we shall use the term “sticking”.
Sticking occurs when the two seal rings are locked or partially welded together. The locked state results in a failure if the interconnection is higher than the starting torque of the motor. It may also result in mechanical damage of seal parts.
Sticking can have different causes. Mainly hard/hard seal face pairings have a tendency to sticking. The main causes of sticking are precipitation of sticky materials from the pumped medium on the seal faces or corrosion of the seal faces.
Sticking is only possible on shaft seals of pumps with start/stop operation. The period it takes for the seal rings to stick together ranging from a few hours and up, depending on the pumped medium. The process accellerates at elevated temperatures.

4. Chemical, physical degrading and wear

All parts of the mechanical shaft seal must have adequate resistance to the chemical and physical environment to operate properly during the expected working life. Elevated temperatures and severe chemical or mechanical loading reduce the expected working life of the seal. This applies up to certain limits; above these limits, degeneration and malfunction will rapidly occur.

The polymer elastomeric parts may exhibit many signs of degradation, including blisters, cracks, voids or discoloration. In some cases, the degradation can be ascertained only by measuring the physical properties. Incompatibility with the chemical and/or thermal environment contributes to the degradation.

Swelling of rubber parts



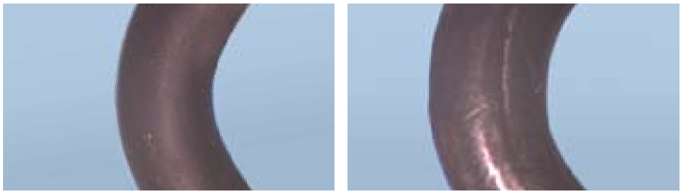


Fig. 5.8: A fresh EPDM O-ring (left) and a swollen EPDM O-ring (right) exposed to water containing mineral oil

Swelling of rubber is an increase in volume and a decrease in hardness due to absorption of a solvent. The volume increase depends on the type and grade of rubber, the type and concentration of the solvent as well as the temperature and time of exposure. In extreme situations, the linear dimensions of a swollen rubber part can be doubled.

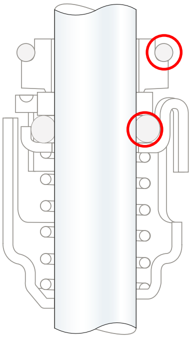
The function of many types of shaft seals depends to a great extent on the geometry of the rubber parts. Consequently, even tiny changes in dimensions can be critical. The reduction in hardness is associated with changes in other mechanical properties of the rubber.

The commonly used EPDM rubber material shows a high degree of swelling when exposed to mineral oil. EPDM swelling takes place irrespective of the concentration of mineral oil in water, however, the lower the concentration level, the longer the period of operation before failure. See fig. 5.8.

Other types of rubber also show swelling when exposed to non-compatible liquids.

Volume swelling, as described above, is only one indication of incompatibility with the pumped medium and may be based on the solubility parameter alone. In addition, attack by the pumped medium on the back-bone and/or cross-link system of the elastomer may appear as a change in physical properties such as tensile strength, elongation at break and hardness. Elevated temperatures and extended exposure times may create more aggressive conditions.

Ageing of rubber parts



In addition to the above degradation process, an ageing process takes place. Ageing often results in a change in physical parameters such as tensile strength and hardness. Ageing is divided into these two categories:

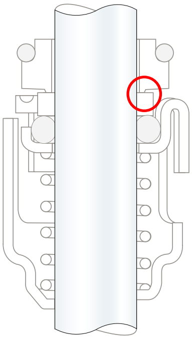
- shelf-ageing
- atmospheric ageing.

Shelf-ageing is basically oxidative degradation. Apart from the obvious influence of oxygen, the catalytic effects of heat, light, internal and external stresses or strains and pro-oxidant metals should be considered.
As opposed to shelf-ageing, atmospheric ageing is characterised by the attack of ozone on the rubber. It is essential to know that this is not merely another form of oxygen-induced degradation the mechanism is quite different. If the rubber is exposed to any sort of stress, the result is atmospheric cracking in which the cracks are perpendicular to the direction of elongation. Atmospheric ageing is well-known from old bicycle or automotive tires. Here several concentric cracks can be observed where the tire has the highest deflection during operation.

Explosive decompression

Explosive decompression occurs on polymeric parts as blisters, pits or pocks on its surface. When the pumped medium has a high partial pressure of a gas-phase, the gas diffuses into the rubber. If the pressure is decreased and the absorbed gas in the material cannot escape as fast as the pressure is decreased, an internal overpressure builds up. If this pressure exceeds the strength of the material, an explosion takes place due to the decompression. An elastomer with high gas solubility and low mechanical strength is exposed to explosive decompression.

Corrosion



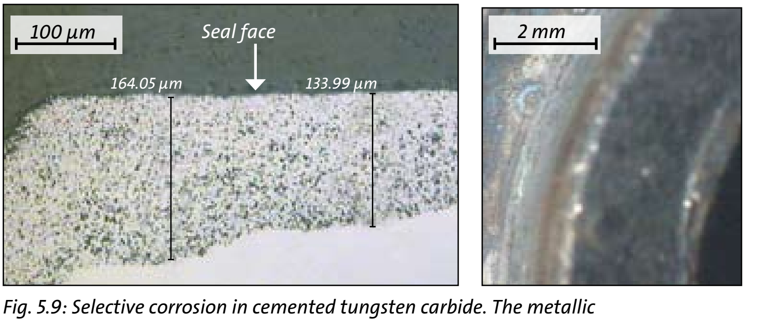
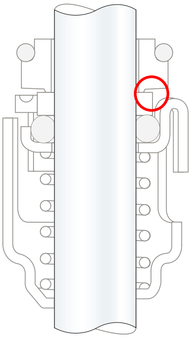


Fig. 5.9: Selective corrosion in cemented tungsten carbide. The metallic binder phase is corroded resulting in reduced wear resistance

Most seal rings materials are made of composite materials. In order to prevent selective corrosion, all phases of the seal ring material must be resistant to the pumped medium. Corrosion in cemented tungsten carbide is typically seen as an attack on the metallic binder phase. The result of this attack is loss of the



mechanical properties, including decrease in wear resistance. When the metallic binder phase disappears, the surface becomes matted. The selective corrosion of the binder phase may induce stresses, leading to cracks in the seal rings. See fig. 5.9.

On other surfaces of the seal rings, heavy erosion can occur where the binder phase is corroded. See fig. 5.10.
In stainless steel pumps and pipe systems, tungsten carbide with cobalt binder corrodes in tap water.

Fig. 5.10: Corrosion of the binder phase followed by severe erosion on a tungsten carbide seal ring with cobalt binder phase

In ceramic materials such as aluminium oxide, the process of a corrosion attack often dissolves or oxidises the glass phase, resulting in a decrease in wear resistance.
When the glass phase has disappeared from the surface of the seal ring, the porosity increases. This affects the mechanical strength of the seal ring. The strength can be reduced with a few percentages or it can totally crumble, depending on the material grade and pumped medium.

Corrosion seldom occurs on the metal parts of the seal in stainless steel pumps. The reason is that seal metal parts are commonly made of a higher grade of stainless steel than other pump metal parts.

Wear

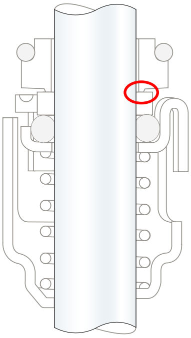


Fig. 5.11: Normal wear on surface of tungsten carbide seal ring

Because the thickness of the lubricating film is of the same order of magnitude as the surface roughness, the seal faces will wear to some extent. This normal wear on well-performing seals will be so small that the seal will be able to survive for many years. See fig. 5.11. In special cases, wear can cause problems, but often seals work perfectly with severe wear up to 0.5–1 mm, even with heavily grooved seal faces, as long as the axial flexibility of the seal ring is maintained.

Wear, continued

Fig. 5.12: Deep seal face grooves close to the pumped medium side

Fig. 5.13: Grooves on seal faces at the evaporation zone

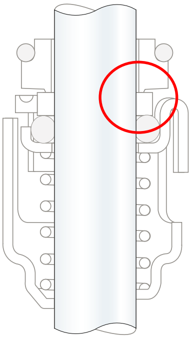
Deep seal face grooves close to the pumped medium side indicate that hard particles from the pumped medium have entered the sealing gap. See fig. 5.12.

Deep grooves close to the atmospheric side indicate that hard precipitates from the pumped medium have been created where the lubricating film evaporates. See fig. 5.13.

5. Installation failures

Some mechanical seal failures come from wrong mounting and handling. Examples can be shaft misalignment, seats not mounted perpendicular to the shaft, axially moving shaft and wrong assembly length, etc.

Misalignment



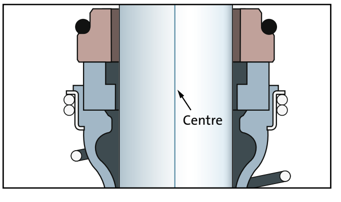


Fig. 5.14: Well-aligned shaft seal

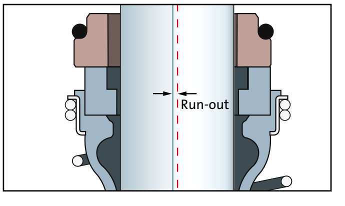
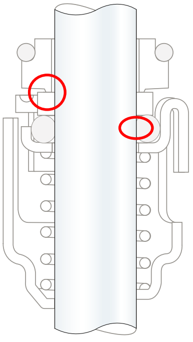


Fig. 5.15: Shaft seal with radial run-out

The position and width of wear tracks on the seat indicate various problem areas.
If the width of a wear track on the seat is the mirror of the sliding face of the opposite seal ring, the shaft seal seems to be well aligned with no run-out of the shaft. See fig. 5.14.

If the sliding face of the seat is broader than the rotating sliding face, a wider track on the seat all way around indicates a high run-out of the shaft. See fig. 5.15.
This can also be seen if, for some reason, there is an unbalance of the rotating mass.

An uneven depth of the wear track around the seat indicates a tilted/crooked mounting of the seat. See fig. 5.16.



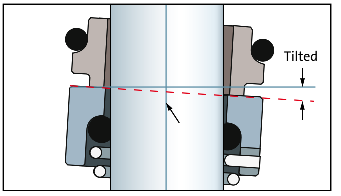


Fig. 5.16: Uneven depth of wear track on the seat caused by tilted/crooked mounting

Fig. 5.17: Wear on dynamic O-ring at the point where it seals against the shaft

Abnormal wear on the dynamic O-ring is observed, if the seat of an O-ring seal is tilted. Wear on the dynamic O-ring is followed by axial scratches on the inner surface. See fig. 5.17.

Wear on the dynamic O-ring is caused by a movement between the shaft or the sleeve and the O-ring. The movement can be an axial movement of the shaft (vibration) or a misalignment of the stationary seat (the seat face is not perpendicular to the shaft).

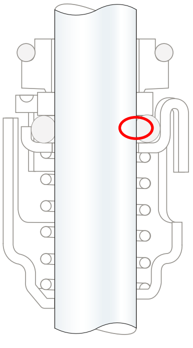


Fig. 5.18: Corrosion on shaft below dynamic O-ring caused by axial movement of the shaft

Axial movement of the shaft caused by vibration or a tilted seat, can cause wear on the shaft or sleeve below the dynamic O-ring. The O-ring rubbing on the shaft continually removes the protecting oxide layer on the shaft, causing corrosion. See fig 5.18.

Assembly

Where a mechanical shaft seal is installed in rotating equipment, the axial movement of the shaft must not exceed the shaft seal flexibility.
Axial movement of the shaft exceeding the permissible flexibility of the rotating part of the mechanical shaft seal may cause extended wear on the seal rings or permanent damage to individual parts of the shaft seal.

Fitting

Many shaft seals have assembly lengths according to standards. This allows users to change from one type of seal to another with improved performance for the actual application. Even if two different seals have the same total length, the sliding faces are not necessarily placed at the same height. If components from two different seals are mixed, the result can be inferior or excessive compression of the seal.

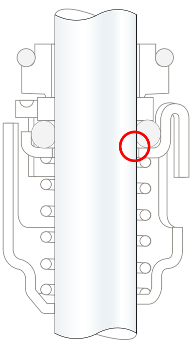
6. System failures

When operating pump systems, the operating parameters may differ slightly from what the system was designed for. A change in operating conditions may affect the seal performance. These parameters affect the performance of a mechanical shaft seal:

- the pressure in the seal chamber
- the temperature around the shaft seal in the seal chamber
- the pumped medium
- the speed
- the shaft seal dimensions.

If the above parameters are not correct for the application in question, the result may be malfunction or damage to the shaft seal.

Pressure



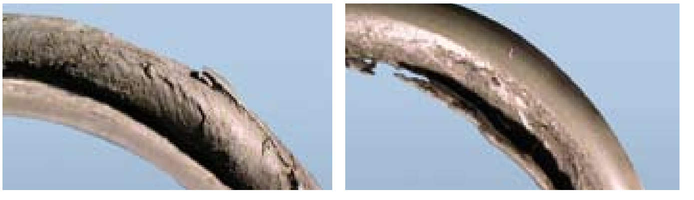


Fig. 5.19: O-rings extruded on account of high pressure

The pressure at the pumped medium side of the seal must be within limits defined by the seal design, the materials and the pumped medium.
When the pumped medium pressure on the shaft seal exceeds the level it was designed for, various failures may occur:
The friction between the seal rings may increase and cause damage either directly due to the friction forces or to the secondary seals in the seal.
Extruded O-rings is a commonly known failure mechanism. If the temperature approaches the maximum operating limit of the rubber, the rubber material becomes softer and thus susceptible to extrusion. See fig. 5.19.

Temperature

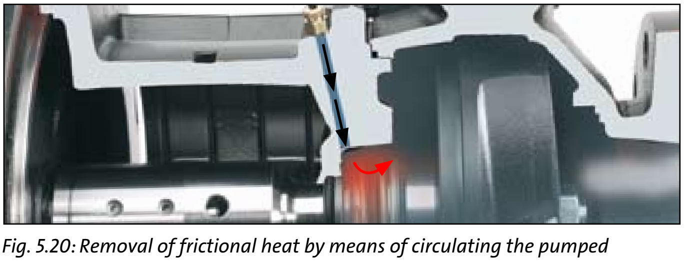


Fig. 5.20: Removal of frictional heat by means of circulating the pumped medium across the shaft seal in the seal chamber

The friction between seal faces in normal operating conditions generates heat. Consequently, the temperature in and close to the seal is higher than the temperature of the pumped medium.

This temperature increase is often 10 to 20 K.
The temperature specification of a shaft seal is given for the actual temperature of the seal. This must be taken into account by the system designer. See fig. 5.20.
Exceeding the maximum temperature of the seal may cause damage to elastomeric parts. Another typical failure due to elevated temperatures is poor lubrication resulting in a higher wear rate. See lubrication failures, page 77.

No or insufficient flow

No flow occurs when the pump contains pumped medium and the pump is running against closed valve. The heat generated by friction in the shaft seal and the heat generated on account of turbulence around the impellers result in a high temperature increase in the entire pump. The increased temperature can damage the elastomeric parts of the shaft seal in particular.
In addition to the temperature increase, the risk of dry-running is also increased, when there is no or insufficient flow through the pump and the seal chamber.

Poor venting

Under certain conditions, it is possible to collect air around the shaft seal, depending on the design of the shaft seal chamber, the operating conditions and the pumped medium. Poor venting resulting in continuous or periodic dry running has disastrous consequences to the life of most mechanical shaft seals.

Vibration

Mechanical vibrations result in higher forces on each part of a mechanical shaft seal. The result is extended wear on all parts, chipped seal rings and possibly opening of the sealing gap.
Vibrations can be generated from friction between seal faces if the operating conditions are exceeded. Otherwise, vibrations are often associated with worn bearings or special liquid flow conditions.
The width of the wear track on the seat will be extended in case of radial vibrations of the shaft occur. See fig. 5.15

Summary
The main causes of failures on mechanical shaft seals used in pumps are described.
It is difficult to state causes of shaft seal failures exactly, even with knowledge of the pump system.
A detailed failure analysis is needed to reduce future failures on shaft seals.

[1] OMV AG, Pumps in Process Engineering 2002

7. Shaft seal failure analysis

The previously described failures have been collected into a failure analysis scheme.
See pages 89 - 90.
The purpose of the scheme is to assist in troubleshooting and give input to possible improvements of seal usage and manage shaft seal parameters involved.

We shall refer to these main shaft seal components:

- complete shaft seal
- seal rings
- elastomers
- metal parts
- shaft/sleeve.

The first column contains the main component.
The second column shows the result of the inspection.
The figures indicate possible causes of failure.

See the “Key to failure analysis scheme”, page 91.

Shaft seal failure analysis scheme

		Possible cause					
		Plant/process influence				Poor thermal control	
Component	Result of visual inspection	Poor lubrication	Cleaning (treatment)	Conta-mination	Corrosion (chemical attack)	Cooling	Heating
Complete shaft seal	Clogged			15/17/41			
	Good condition			1		1	1
	Hang-up			1		1	1
	Noise	13				20	15
	Sticking				11		
Seal rings (1) - (2)	Chipped edges				11		
	Cracked/fractured	3			11		
	Deposits outside seal faces			1/17/38			
	Etched/decomposed		5		11		
	Pitted		5		11		
	Scored/scuffed/galled	13				20	15
	Wear track:						
	No track						
	Blistered/flaked	13/24		23/18	31		
	Deposits			8/9/17			
	Excentric						
	Grooved	20		8			
	Incomplete track				39		
	Matted				11/21		
	Narrow or wide						
	Normal						
	Thermal cracks	13/20				20	15
	Wear on atmospheric side	13		8			
	Wear on pumped medium side						
Elastomers	Burnt	3			11		
	Cuts/nicks						
	Decomposed		11		11		
	Extruded						
	Fractured (bellows)						
	Lost flexibility	3					12
	Ruptured						
	Swollen		11		11		
	Wear, dynamic (3)						
	Wear, stationary (4)	3		34	34		
Metal parts	Cracked/fractured			34	34		
	Deformed			34	34		
	Discoloured	3	11		11		
	Etched/decomposed		5/11		11		
	Pitted		11		11		
	Worn	13					
Shaft/sleeve	Cracked/ruptured	13	11				
	Pitted		11		11		
	Worn			8	11		

(1) Rotating seal ring (2) Stationary seat (3) Dynamic O-rings (4) Stationary O-rings

s of shaft seal failure

Incorrect installation			System faults					
Assembly	Fitting	Misalign-ment	Flow	Pressure	Tempera-ture	Poor venting	Vibration	Other
2/19	2							
				1		1	1	
	7			15/35	13	36	16	
	10						16	
	10		4			3	16	37
	5		11					
	7			35	13	36		
1	2							
	22				15			
		26						16
19	7					4		
38		40						
		19/40						16
			4	15		36		
			4		13	3		
10/19	25							
					12	3		
19	10			13	13			
	7	26						
		28			12	3		
		28		27/30				
19/26		19/26/40		29			16/32	
						3		
19		26		13/29	13		16	
7/37	7			13	13			
19	32					3		
					33			
		26					14	
10	10			29			16	
	6			29			16	
	6	26		29			32	

Key to failure analysis scheme

	Possible causes of failure
1	Seal faces open due to locking of axial movement of the dynamic seal ring. Axial movement is needed to compensate for thermal and pressure-induced axial expansion.
2	Seal is not built into the correct assembly length, resulting in no contact between seal ring and seat.
3	Seal faces are running dry, resulting in overheating. High torque on seal faces made of hard materials can generate heat that can be transmitted to elastomers, resulting in hardening and burning.
4	Closed pump outlet valve results in excessive temperature or failure no 3.
5	Chemical reaction with an oxidizing agent, e.g. nitric acid.
6	Seal or sleeve is not fastened properly.
7	Overcompression due to incorrect assembly length.
8	Solid particles, such as sodium hydroxide crystals, precipitate out of liquid across sliding faces.
9	Volatile elements of liquid evaporate in seal gab, leaving highly viscous sticky layers on sliding faces.
10	Damage due to mishandling (or overcompression).
11	Seal material not chemically resistant to liquid or contaminant.
12	(Elastomer) material has decomposed due to temperature beyond limitations.
13	Seal has been exposed to pressure/temperatures in excess of limits.
14	Continual removal of passive film due to relative movement.
15	System pressure is below or close to vapour pressure.
16	Bearings possibly worn.
17	Liquid is saturated with scale.
18	Solid particles in pumped medium.
19	Wrong assembly of main shaft seal components.
20	Viscosity too low for the actual shaft seal.
21	Corrosion makes tungsten carbide appear mat grey or green.
22	By electrochemical deposition, metals such as copper may form on the sliding faces due to missing or poor electrial grounding on the pump.
23	In water of a conductivity below 5 micro µS/cm, some SiC grades can corrode.
24	Excessive heat dissipation may cause blistering of resin-impregnated carbon.
25	Elastomer fitted on uncleaned surfaces.
26	Seat is misaligned.
27	Explosive decompression due to fast pressure release.
28	Heat build-up may take place due to misalignment.
29	Start/stop at excessive system pressure with hard seal faces.
30	Extrusion of rubber at high pressure and/or temperature.
31	Swelling of impregnation can cause blistering of impregnated carbon/graphite.
32	Undesired axial movement of shaft and shaft seal.
33	Cavitation in seal chamber.
34	Sticking of hard seal rings (SiC/SiC or WC/WC).
35	Partly dry running due to atmospheric pressure exceeding liquid pressure around seal.
36	Partial dry running due to air, air boubles around seal faces.
37	Extreme handling/drop of pump or shaft seal.
38	Seal ring or seal chamber is unsymmetrical, deformed by pressure or other forces.
39	Corrosion under a (soft) seal ring that is supported by a stiff corrosive material.
40	Misalignment of seal rings.
41	Hair or mud contamination.

Chapter 6

Standards and approvals

1. European Standard EN 12756

2. Approvals

This chapter describes the standards, approvals and guidelines governing the use of mechanical shaft seals in industry.

1. European Standard EN 12756

Dimensions according to EN 12756

European Standard EN 12756 defines the principal dimensions for the installation of single and multiple mechanical shaft seals into pump housings.
The standard also describes the shaft seal designation and material codes to be used. This facilitates the exchange of shaft seal information between shaft seal suppliers and users.

Type designation

The type designation according to EN 12756 is based on this coding system for single shaft seals:

Example **N U 012 S 0**

Pos. 1: ***Assembly length:***
N = Normal
K = Short

Pos. 2: ***Balancing:***
U = Unbalanced
B = Balanced

Pos. 3: ***Nominal shaft diameter:***
The diameter in mm, indicated by a three-digit number

Pos. 4: ***Direction of rotation of the seal:***
R = Right-hand rotation (clockwise when viewed from the seat)
L = Left-hand rotation (counterclockwise when viewed from the seat)
S = Capability of rotation in either direction

Pos. 5: ***Retention against rotation of the seat***
0 = without retention
1 = with retention

Example of designation for a mechanical shaft seal with normal assembly length: **NU043S0.**
The code represents a shaft seal with normal assembly length, unbalanced, for a Ø43 shaft, capable of rotation in either direction, without retention against rotation of the seat.

Comparable example for a mechanical shaft seal with short assembly length: **KU043S0.**

The EN 12756 specifies dimensions of the seat housing and a recommended maximum outside diameter of the rotating part.

Most seal manufacturers insert a type/product code in front of the seal designation.

In addition to the dimensions, EN 12756 also specifies the materials of the shaft seal components.

Material key according to EN 12756
The materials of the shaft seal components are indicated by means of code letters.
The code for single seals has five letters. In the following, only single seals will be described.

Material key positions 1 2 3 4 5

Material of the rotating seal ring

Material of the stationary seat

Material of the secondary seals

Material of the spring

Other construction materials in the seal

Example: EN12756-NU043S0-QQEGG

The code represents a shaft seal with normal assembly length, unbalanced, for a Ø43 shaft, capable of rotation in either direction, without retention against rotation of the seat, and with

1. rotating seal ring made of SiC
2. stationary seat made of SiC
3. secondary seals made of EPDM
4. spring made of CrNiMo steel
5. other shaft seal materials made of CrNiMo steel.

Many seal suppliers use this standard with some additions to describe the seals, but some also use their own codes.

Material key according to EN 12756, edition 2000

Position 1 and position 2	Position 3	Position 4 and position 5
Material for the rotating seal ring (1) and the stationary seat. (2)	Material for secondary seals.	Material for other construction materials in the seal.
Manufactured carbons A Carbon, metal-impregnated B Carbon, resin-impregnated C Other carbons **Metals** D Carbon steel E Cr steel F CrNi steel G CrNiMo steel H Metals with carbide coatings K Hard coatings, metallic M High-nickel alloy N Bronze P Grey cast iron R Alloyed grey cast iron S Cr-cast steel T Other materials **Carbides** U Tungsten carbide Q Silicon carbide J Other carbides **Metal oxides** V Aluminium oxide W Chromium oxide X Other metal oxides **Plastics** Y PTFE glass-fibre, reinforced Z Other plastics	**Elastomers** B Butyl rubber (IIR) E Ethylene-propylene rubber (EPDM) K Perfluoro rubber (FFKM) N Chloroprene rubber (CR) P Nitrile rubber (NBR) S Silicon rubber (MVQ) V Fluorocarbon rubber (FKM) X Other elastomers **Elastomers, sheathed** M Elastomers/ PTFE-sheated **Non-elastomers** G Graphite T PTFE Y Other non-elastomers **Various materials** U Various materials for flexible elements	D Carbon steel E Cr steel F CrNi steel G CrNiMo steel M High-nickel alloy N Bronze T Other materials

Numerous variants of the above materials can be seen in the literature of the various shaft seal manufacturers. **Note:** The same letter will be used in different positions.

2. Approvals

Specific approvals of shaft seals are sometimes required. Below you will find some examples.

Drinking water approvals and local approvals on a global scale

Materials which come into contact with drinking water during supply, treatment and distribution to the tap, may release substances into the water. This may have adverse effects on the general quality of the drinking water or pose a health risk to consumers.
Common to all drinking water approval schemes is therefore an assessment of the suitability of the materials for use with drinking water. The purpose is to prevent an unacceptable deterioration of the quality of the drinking water. The suitability of materials intended for use with drinking water can be determined by means of migration/leaching tests. The materials are tested independently or together in the assembled product.

The table below shows a number of important national approval bodies responsible for materials intended for use with drinking water.

Country	Name of scheme	Legal framework	Approval body/ institute	What is evaluated for leaching/ migration?	Mechanical testing of the complete product?
France	ACS (Attestation de conformité sanitaire). See [1]	Ministerial decree of 29 May 1997	Any one of four laboratories within France, authorised by the French Ministry of Health to perform and award ACS certification.	Either individual materials or a complete multiple material product can be awarded an ACS certificate	No
United Kingdom	WRAS / DWI BS 6920. See [2]	Water Supply (Water Fittings) Regulations 1999	Water Regulations Advisory Scheme/ Drinking Water Inspectorate	All non-metallic materials in contact with water in the product must pass British Standard 6920:2000, tested independently	Yes
Unites States	NSF 61. See [3]			Either individual materials or the complete product can be approved	No
Germany	DVGW UBA / KTW. See [4]			Individual materials/ components	No – not for pumps at this time

For further information, see [1] to [4].

Sanitary approvals

The use of pumps in hygienic and sanitary applications, such as plants for the pharmaceutical, food and biotechnology industries as well as sterile processes, is subject to higher design standards, in terms of cleanability and sterilisation, than the use of pumps in other applications.

The design, materials used and material surface finish are subject to a variety of national and international rules and regulations, guidelines and laws, such as:

- FDA (Food and Drug Administration) regulations
- EHEDG (European Hygienic Equipment Design Group) recommendations and certification
- 3A Sanitary Standards
- QHD (Qualified Hygienic Design) criteria.

FDA

The FDA (Food and Drug Administration) is a scientific, regulatory, public health agency within the United States Department of Health and Human Services.

FDA is responsible for the safety of the United State's foods, cosmetics, drugs, biologics, medical devices, and radiological products. It is one of the United States' oldest consumer protection agencies.

For further information, see FDA's homepage [5].

EHEDG

The EHEDG (European Hygienic Engineering & Design Group) was founded in 1989. The members of the organisation come from food industries, equipment manufacturers, research institutes and public health authorities. EHEDG is a non-profit organisation and has no legislative authority.
EHEDG develops guidelines and testing methods for the safe and hygienic processing of food.

Two EHEDG guidelines are important for assessing the hygienic design and cleaning of process equipment and components. See [6] and [7] from http://www.ehedg.org.

The guideline on Hygienic Equipment Design Criteria [6] describes the design criteria to be met for hygienic and aseptic process equipment. It gives guidance on how to construct food processing equipment and components so that it does not affect the bacteriological growth and quality of the food product in any adverse manner.

The guideline on in-place cleanability (CIP) of food processing equipment and components [7] describes a test procedure to indicate areas of poor hygienic design where food products and/or microbes can accumulate.
The degree of cleanliness is based on the removal of a bacteria-containing soil. The cleaning is performed using a mild detergent to leave some soil in the reference pipe. This facilitates cleanability comparisons between the test object and the reference pipe with a known surface roughness. If the test object has a cleanability equal to or better than the reference pipe, an EHEDG certificate can be issued.

As food safety does not end at the borders of Europe, the EHEDG actively promotes global harmonisation of guidelines and standards.

The EHEDG symbol is used by manufacturers to indicate compliance with the EHEDG criteria.

For further information, see EHEDG's homepage [8].

3-A Sanitary Standards

The 3-A Sanitary Standards, Inc. is the American counterpart of the European EHEDG.
The 3-A Sanitary Standards have no testing schemes. Consequently, only 3-A certificates of compliance can be issued. The US-based organisations NSF and 3-A have agreed to co-operate in the development of EHEDG Guidelines, and in turn EHEDG co-operates in the development of 3-A and NSF standards.

The 3-A symbol is used by manufacturers to indicate compliance with the 3-A Sanitary Standards

For further information, see the 3-A Sanitary Standards homepage [9].

QHD

QHD (Qualified Hygienic Design) is a testing system for the hygienic design and cleanability of components, machinery and plant for aseptic or sterile applications. The system is for self-certification under the German industry association, VDMA (Verband Deutscher Maschinen- und Anlagenbau e.V.). This ensures that all surfaces can be cleaned in place (CIP).

The manufacturer attests the relevant regulations for the designed product from the QHD manual.
Tests for good cleanability are carried out in a test laboratory either by the manufacturer himself or by independent approval bodies.

The QHD symbol is used by manufacturers to indicate compliance with the QHD criteria

For further information, see the VDMA homepage [10].

Other guidelines for use of mechanical shaft seals:

ATEX

ATEX is a French abbreviation for "ATmosphère EXplosible" (explosive atmosphere). ATEX is the name given to a set of European directives dealing with equipment intended for use in potentially explosive atmospheres.

ATEX certification is based on meeting the requirements of these two EU Directives:

1. **Directive 94/9/EC,** also known as ATEX 95 or the **ATEX Equipment Directive.** The directive applies to equipment and protective systems intended for use in potentially explosive atmospheres. The directive places responsibilities on the manufacturer of these products. The main responsibility of the manufacturer is to prevent the formation and ignition of explosive atmospheres.

2. **Directive 99/92/EC,** also known as ATEX 137 or the **ATEX Workplace Directive**. The directive sets out minimum requirements for improving the health and safety protection of workers potentially at risk from exposure to explosive atmosphere. This directive is concerned with the health and safety of workers with relation to potentially explosive atmospheres. It places responsibilities on the employer.

As previously described, mechanical shaft seals develop heat in the sealing gab during operation. The ATEX directives state whether special protections are required. Explosive atmospheres can be caused by flammable gases, mists or vapours or by combustible dusts, mixed with air. Areas with explosive atmospheres are classified into hazardous zones. The classification given to a particular zone, and its size and location, depends on the likelihood of an explosive atmosphere occurring and its persistence if it does.

Equipment is classified into categories, depending on the level of protection.

For further information, see [11].

API 682 and ISO 21049

API (American Petroleum Industry) is the only national trade association that represents all aspects of America's oil and natural gas industry. The API 682 and ISO 21049 standards deal with mechanical shaft seals and systems around the seal used in the oil and gas industry.

For further information, see [12].

Summary
Different applications require different standards or approvals. The most important standard for mechanical shaft seals is EN 12756. In addition, approvals or guidelines for drinking water, food, cleaning or protection against explosion may be relevant.

Reference list

[1] http://www.sante.gouv.fr

[2] http://www.wras.co.uk

[3] http://www.nsf.org

[4] http://www.dvgw.de/104.html

[5] http://www.fda.gov

[6] The Hygienic Equipment Design Criteria, Document Guideline No. 8 (2004)

[7] A method for the assessment of in-place cleanability of food processing equipment, Document No. 2 (2000).

[8] http://www.ehedg.org

[9] http://www.3-a.org

[10] www.vdma.org

[11] www.ce-mark.com/atexdir.html

[12] Overview of API 682 and ISO 21049, Proceedings of the Twenty-First International Pump Users Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 131-137, 2004

A

abrasive particles 32, 56
abrasive wear 72, 73, 74
abrasives and additives 43
absorption 81
acrylonitrile-butadiene rubber 59
ACS 97
adhesive wear 72
ageing of rubber parts 82
aggressive media 41
air pressure 33
air vent valve, automatic 39
air-cooled top 39
alarm 33
aluminium oxide 48
American 3-A sanitary standards 99
anti-freeze liquids 28, 29
antimony 47
API 682 100
approvals 94, 97
asperities 72
asperity contacts 57
assembly 85
assembly length 85, 94
ATEX 100
ATEX equipment directive 100
ATEX workplace directive 100
atmosphere 10
atmospheric pressure 14, 20
atmospheric side 84
atmospheric ageing 82
automotive mechanical shaft seal 22
average radius, R_m 18
axial flexibility 13, 83
axial force 15
axial movement 13, 27, 76, 78, 85
axial surfaces 10
axially moving shaft 84

B

back-to-back 31, 32
back-to-back seal arrangement 34
back-bone system 81
bacteria-containing soil 99
balanced cartridge O-ring seal 29
balanced mechanical shaft seal 15
balanced O-ring seal 28
balanced metal bellows cartridge seal 29
balancing 94
balancing ratio 15, 16
barrier 8
barrier fluid 30, 32
barrier fluid pressure 32
bearing area curve 69
bellows 11
bellows shaft seal 12
boiling point 42
boundary lubrication 65
braided packing 22
braided stuffing box packing 9
burnt elastomeric part 77

C

can 40
carbon graphite 23, 47
carbon graphite against alumina 52
carbon graphite against SiC 52
carbon graphite against WC 51
cartridge design 23
cartridge unit 29
causes of pump failure 76
centipoise 42
chemical adhesion of surfaces 59
chemical bonds 59
chemical resistance 42
chemical resistance of elastomer 60
chipped seal rings 87
CIP, cleaning-in-place 38, 99
circulation 31, 37
cleanability and resistance 38
cleaning-in-place 38
cleaning processes 38
clearance 41
clogged by sediments 79
clogging 79
closing force 13, 15, 57, 79
cobalt binder 83
coefficient of friction 21, 47
cold welding 72
compression force 13
concentric cracks 82
conductivity 58
consolidation 79
construction materials 95

contact points 65
contact surface 8
contamination 13, 37
contamination failure 78
converging sealing gap 19, 71
cooking plate 21
cooling liquid 31
corrosion 80
corrosion of tungsten carbide 82
corrosive media 41
corrosive wear 72, 73
counter-clockwise 28
cross-link system 81
crystallisation 36

D
damage report 76
dead-end application 34
debris removal 73
deep seal face grooves 84
degeneration 80
degrading and wear 80
demineralised water 58
deposits 13, 20, 36, 79
deposits, hard 20
deposits, sticky 20
diamond coatings 50
differential pressure, Δ p 18
dimensions 94
direct sintered SiC 49
direction of rotation 94
directive 94/9/EC 100
directive 99/92/EC 100
dirt-containing liquids 28
diverging sealing gap 19, 71
DLC 50
dosing pump 34
double-seal arrangements 31
drinking water approvals 47, 97
dry running 56, 72, 76, 77
dull finish 70
duty parameter 66
DVGW 97
dynamic O-ring 76
dynamic secondary seal 27
dynamic viscosity 18, 42

E
EHEDG 98, 99
elastic deformation 13
elastomer components 38
elastomer membrane 35
elastomeric parts 42, 80
electro polished 38
elevated reservoir 37
elongation at break 81
EN 12756 47, 94
EPDM 60, 81
equilibrium 59
erosion 73
etching 71
evaporation 17, 20, 36, 71, 73, 74
evaporation point 20
evaporation zone 69
excessive compression 85
excessive over-pressure 41
exchange of liquid 41
explosive atmosphere 100
explosive decompression 82
explosive liquids 32
external-seal arrangement 41
extruded O-rings 86
extrusion 86

F
failure analysis 76
fatigue wear 72, 74
FDA 43, 98
FFKM 60
fibre-containing liquids 28
fitting 85
fixed pressure 33, 34
FKM 60, 78
flat surface 10
flatness of seal faces 17, 18
flourinated copolymer rubber 60
fluorocarbon monomer rubber 60
flushed-fluid side 36
flushing fluid 30, 36, 37
fretting 39
friction forces 86
friction loss 8
frictional heat 21, 72, 73, 77
fugitive elements 59

full-fluid-film lubrication 14, 65
FXM 60

G
gab height 18
glycerine 33
grade of rubber 81
graphite inclusions 49
grinded surface 70
grinding tool 80
Grundfos BME pumping system 39
Grundfos cartridge shaft seal 23
Grundfos CH/CHI pump 22
Grundfos CP pump 22
Grundfos CR pump 22, 33
Grundfos first mechanical shaft seal 22
Grundfos pump magazine 22
Grundfos type A shaft seal 16, 27
Grundfos type B shaft seal 27
Grundfos type C shaft seal 28
Grundfos type D shaft seal 28
Grundfos type G shaft seal 28
Grundfos type H shaft seal 16, 21, 29
Grundfos type K shaft seal 29
Grundfos type O shaft seal 30
Grundfos type P shaft seal 30
Grundfos type R shaft seal 29
guiding elements 61

H
hang-up 78
hard particles 80
hard precipitates 84
hardness 81
harzadous zones 100
hermetically sealed system 40
high-level switch 33
high-pressure applications 28
high-speed shaft seals 39
high-temperature pumped medium 39
high-viscosity liquids 28, 29
HNBR 59
hydraulically loaded area 15, 16
hydrodynamic 65
hydrodynamic lubricating film 15
hydrodynamic lubrication 66
hydrodynamic pressure 14, 19, 66
hydrodynamic tracks 68, 74
hydrodynamic wedge 68
hydrostatic 65
hydrostatic lubrication 71
hydrostatic pressure 14, 19
hygienic and sanitary application 98
hygienic equipment design 98
hypochlorite 48

I
impregnated carbon graphite 47
impurities 80
inhibitors 57
inner rotor with magnets 40
installation failure 84
insulating air 39
internal circulation 31
internal friction 66
internal overpressure 41
introduction to failures 76
ions 59
ISO 21049 100

K
k = balancing ratio 15, 16

L
lack of lubricating film 20
laminar 66
lapped surface 70
lapping 68
laser machining 71
lead 47
leakage 10,17, 30, 40
leakage rate 18, 57, 68, 70, 80
level indicator 33
lip seal 9
liquid level 33
liquid pressure 20
liquid-lubricated 12
low-level switch 33
low-viscosity fluids 68
lubricating film 12, 13, 14, 20, 68
lubricating liquid 31, 32
lubrication 30, 65
lubrication failure 77
lubrication regime 65

M

magnetic-drive system 40
malfunction 80
manometer 33, 34
material code 94, 95
material code A 47
material code B 47
material code E 60
material code K 60
material code P 59
material code Q 49
material code S 60
material code U 48
material code V 48, 60
material surface finish 98
materials 46
maximum temperature 87
mechanical properties 81
mechanical shaft seal 9
mechanical stress 79
mechanical vibration 87
membrane vessel 34
metal bellows seal 78
metal parts of the seal 83
misalignment 39
mixed lubrication 65
monitor seal leakage 37

N

narrow seal face 28, 29, 57
natural convection 33, 37
Newtonian fluids 67
no or insufficient flow 87
noise 78
nominal shaft diameter 94
non-parallel seal faces 19
non-poisonous 33
non-return valve 35
normal assembly length 95
normal wear of tungsten carbide 83
NSF 61 97

O

opening force 13, 14
opening of the seal gab 79
operating conditions 42, 76
operating parameters 86
operating principle 12
O-ring, rotating 11
O-ring, rubbing on the shaft 85
O-ring, stationary 11
O-ring-type-seal 78
outer edge 13
outer rotor with magnets 40
outside shaft seal 41
overpressure 33
oxide layer 59
oxides 83

P

parallel faces 67, 71
parallel seal faces 19
particles and deposits 80
particles and fibres 79
perflouroelastomer rubber 60
permissible flexibility 85
perpendicular 84
pressure-temperature diagram 55
physical adhesion 58
piston pump 22
poisonous liquid 32
poor lubrication 77, 87
poor venting 87
possible causes of failure 91
precipitation 30, 78, 79
pressure difference 13
pressure distribution 14
pressure in the seal chamber 86
pressure intensifier 35
pressure profile 67
pressure sources 33
pressure switch 34
pressure vessel 33
pressureless 36
PTFE bellows 41
pump downtime 76
pump housing 10, 12, 13
pump pressure 14
pump repair cost 76
pump standstill 13
pump system failure 76
pumped medium 13, 30, 42, 78, 86
pumped medium pressure 33
pumped medium side 84

Q

QHD 98, 99

R

reduced seal face 28, 29
relative material ratio 69
relative rotation 68
reservoir with barrier fluid 34
resin-bonded carbon graphite 47
retension against rotation 94
Reynolds equation 67
risk of dry-running 87
rolled-metal bellows 29
rotating part 8, 9, 12, 13
rotating seal ring 10, 12, 20, 41, 95
rotating seat 39, 42
roughness 69
roughness of seal faces 18
roughness values 38
rubber 59
rubber bellows 12
rubber bellows seal 23, 27
rubber parts 11
running-in period 70
running-in wear period 53
run-out 84

S

safety non-return valve 35
safety valve 33
sanitary applications 38
sanitary approvals 98
sanitary shaft seals 38
sanitary standards 3A 98, 99
seal arrangement type P 36
seal chamber 31
seal face material pairings 51
seal face materials 46
seal faces 10
seal gap 10, 12, 14, 56, 57
seal hang-up 27, 76
seal performence in hot water 55
seal performence in pure water 57
seal performence in water abrasives 56
seal performence in water glycol 56
seal type arrangement O 32
seal chamber with circulation 37
seal chamber with dead end 37
seal chamber with external flushing 37
sealing gap height, h 18
sealing pressure 42
sealing systems 31
sealing technology 22
sealing zone 69
seat 41
secondary seals 11, 59, 95
secondary seals, modified 38
sedimentations 79
seizure of seal faces 57, 58
selecting a mechanical shaft seal 42
selective corrosion 73, 83
settlements 78
shaft 12
shaft diameter 17
shaft misalignment 84
shaft run-out 13
shaft seal components 95
shaft seal designation 94
shaft seal diameter 42
shaft seal dimensions 86
shaft seal failure 76
shaft seal surfaces 38
shaft speed of rotation 42
shear force 58
shear resistance 66, 67
shear strain 67
shear stress 67
shelf-ageing 82
shiny surface 70
short assembly length 95
SiC against SiC 54
silicon carbide 23, 49
silicone rubber 60
SIP, sterilisation-in-place 38
slide freely 78
sliding face 84
sliding face area 15, 16
sliding face width, b 18
solids dissolved 20
solid lubricant 59
solubility parameter 81
speed 86
speed of rotation 17
spring 11, 12, 95

spring force 13, 15
springs and bellows 61
stable friction 55, 74
stable lubricating film 39
stand-alone magnetic-drive system 40
stand-alone system 39
standard shaft seal 39
standards 94
standards and regulations 38
starting torque 80
static O-rings 40
static rubber parts 29
stationary part 8, 9, 12, 13
stationary seat 10, 12, 13, 20, 95
stationary secondary rubber seal 12, 13
steam condensate 38
stepped shaft 10
sterile 38
sterile processes 98
sticking/seizure 80
sticky materials 80
sticky media 32
Stribeck curve 65
stuffing box 9
submersible motor 41
surface damage 77
surface roughness 17, 83
surface texturing 71
surface topography 69, 70
surface wear 13
suspensions 79
swelling of rubber parts 81
system faults 86
system pressure 14

T

tandem seal arrangement 31, 36
tangential motion 72
tapered seal faces 19
temperature 17, 42, 86
temperature around the shaft seal 86
temperature damage 72
temperature increase 86, 87
temperature specification 87
tensile strength 81
test procedure 99
testing of mechanical shaft seals 55
thermal cracks 77
thermal distortion 68, 69
thermal fatigue 73
thermal shock resistance 48
thermocouple 56
thermometer 33
three-body abrasion 73
through-shaft 8
tilted seal 85
time of exposure 81
tin 47
topography 65
torque-transmission parts 11, 27, 61
torque-transmission ring 12
tribology 64
tribological performence 55
tubes, plugs and holders 61
tungsten carbide, WC 23, 48
two-body abrasion 73
type designation 94
type/product code 95

U

unbalanced mechanical shaft seal 15
unbalanced O-ring seal 27, 28

V

vaporisation 71
vapour 71
vapour pressure 20, 21
velocity distribution 67
vibration 17, 87
viscosity 17, 42, 66

W

waviness 68, 74
WC against WC 53
wear 15, 72
wear on the dynamic O-ring 84
wear rate 55
wear track 20, 84, 87
wedge height 67
width of wear tracks 84
working life 80
worn bearings 87
WRAS 97

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