Influence of the installation environment on the operating behavior of mechanical seals

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Mechanical seals have significant internal power losses inside the sealing gap on the axial surfaces because of friction conditions. Even with pure liquid friction with a complete hydraulic balance of the sealing gap, friction losses of several kW can occur on high-performance seals. The heat must be dissipated through the seal's components and transferred to the liquid to be sealed for cooling. In practice on the cylindrical surfaces, carbonization layers might occur in oil seals and evaporation processes might occur in water seals. The cause is to be found in insufficient heat transfer to the liquid to be sealed. Heat transfer is significantly influenced by the geometry of the seal's cavity beside the physical properties of the fluid. The overall responsible designer of the machine must be aware of this fact.

1 Introduction

Figure 1 shows a typical horizontal process pump according to API 610 with a double-acting mechanical seal. The focus of the pump designer is on hydraulics. The mechanical seal is a subordinate component comparable to the bearings. The present double-acting mechanical seal – installed in a ready-to-install sealing cartridge (yellow) – serves to seal the pump shaft.

The real primary seal is formed by two axially rubbing very fine and at the same time hard, often ceramic surfaces. The technical friction caused by the rotational movement causes heat during operation, which must be dissipated to the liquid to be sealed to cool the sealing rings. The liquid heats up. If there is not enough coolant available in the flow through the sealing cartridge, the friction of the mechanical seal heats the coolant too much until carbonization of hydrocarbons or evaporation of water-based liquids. If carbonization occurs at the outer cylindrical surface acting like a thermal isolator the heat transfer is influenced negative, the seal heats up further. A carbonization layer at the seal face will be created due to overheating when sealing oil influencing the leakage. If dry running occurs during friction because of evaporation processes, the sealing rings are damaged, and friction and wear continues to escalate drastically due to a lack of lubrication also influencing the leakage.

Although there is a generally valid standard EN 12756, according to which different installation spaces for mechanical seals are standardized (Figure 2 for unbalanced type component seals and Figure 3 for balanced type component seals), creating a narrow radial space $d_4 - d_3$, the ultimate responsibility for the reliability of the mechanical seal as part of the pump lies with the pump designer.

Excellent heat transfer is considered in modern pump designs (typically shown in Figure 4) by conically stepwise expanding the rear end back cover plate of the hydraulics typical for ANSI pumps deviating from the EN standard.

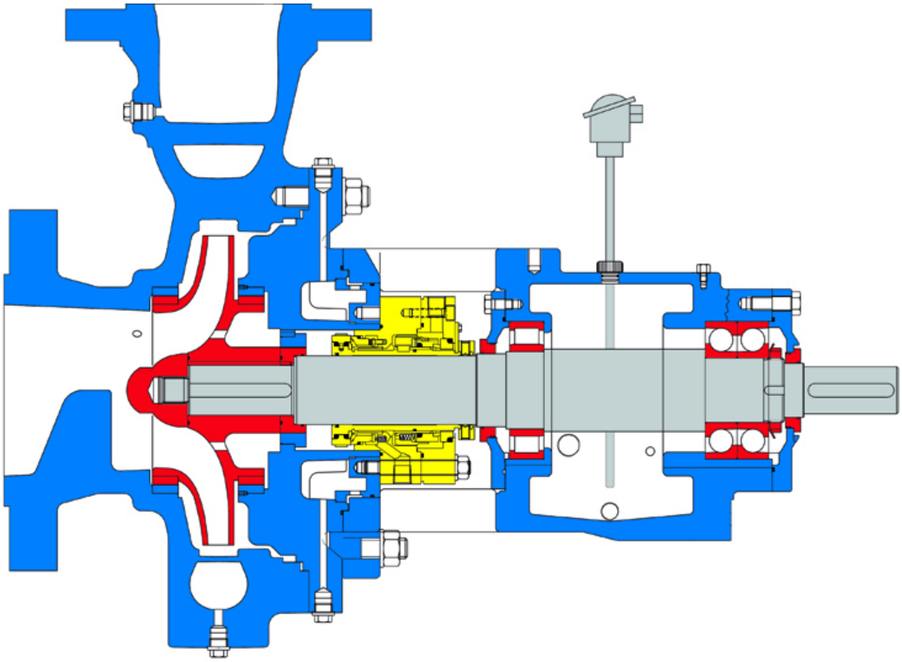

Figure 1: Apollo Pumpen, typical pprocess pump API 610, OH2 (typical example for a single stage pump)

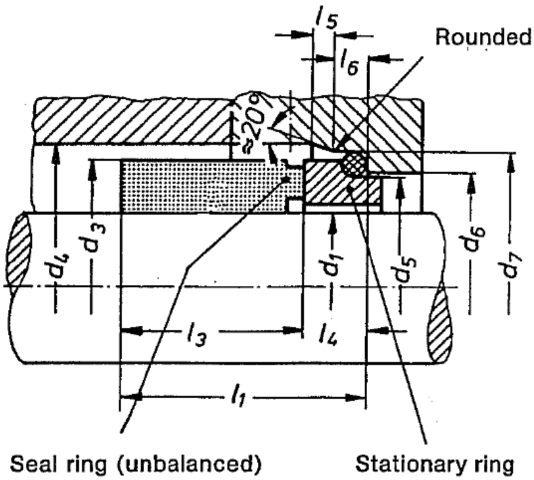

Figure 2: Single mechanical seal, arrangement U (unbalanced) according to EN 12756 former DIN 24960: 1992-06 (withdrawn)

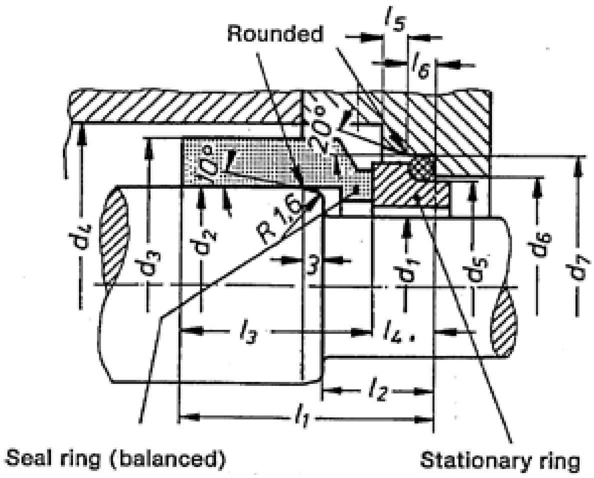

Figure 3: Single mechanical seal, arrangement B (balanced) according to EN 12756 former DIN 24960: 1992-06 (withdrawn)

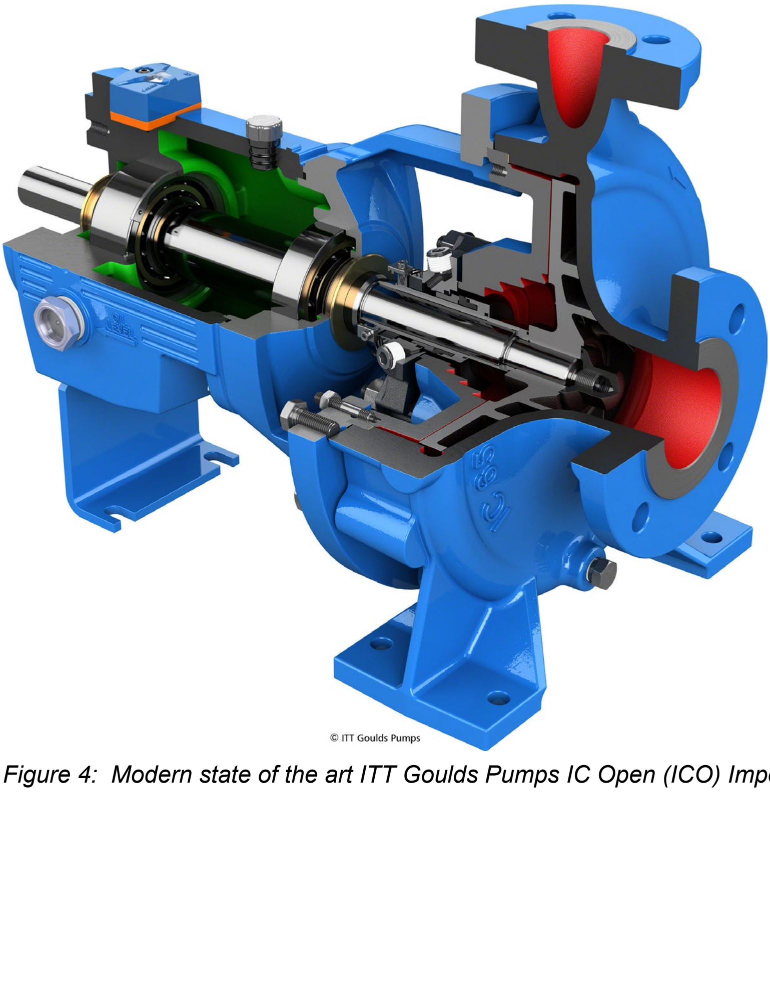

Figure 4: Modern state of the art ITT Goulds Pumps IC Open (ICO) Impeller i-FRAME pump

The conically enlarged seal cavity (Figure 5) for single seal arrangement seems to be optimized regarding the hydraulic conditions for the seal. The fluid exchange and thus the heat dissipation from the cylindrical surface of the seal is supported by toroidal vortex formation. Nevertheless, the heat dissipation is significantly influenced by the mechanical decoupling of the rotating seal ring from the shaft sleeve via isolating air gaps and the elastomer secondary seal. The resulting remaining heat transfer area is significantly reduced.

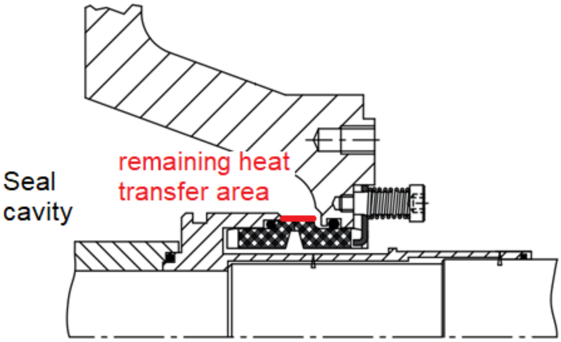

Figure 5: Seal installation space for single seal with conically open geometry

The seal cavity (Figure 6) for the inner seal of a double seal is comparable to the cavity of a single seal shown in Figure 5. The heat transfer apparently also takes place to the fluid to be sealed, but there is no forced liquid exchange of the buffer fluid (orange) at the inner diameter of the inner (left-side) seal. The seal works dead end concerning the buffer fluid. The outer seal is cooled by the circulating buffer fluid. The heat transfer at the surface under consideration depends on the design of the seal's internal installation space. This might be under the exclusive responsibility of the seal maker.

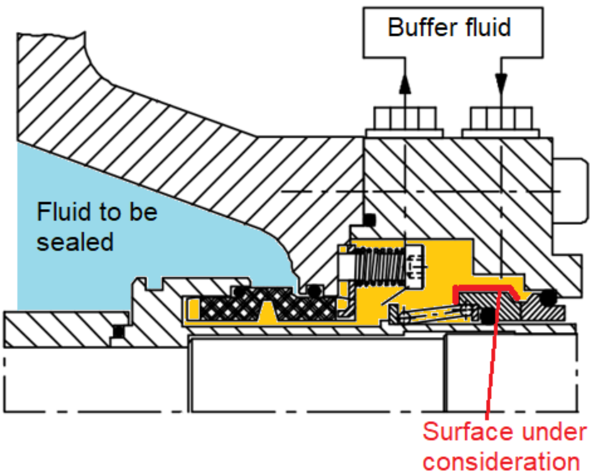

Figure 6: Seal installation space for dual seal with conically open geometry

The heat transfer in mechanical seals with their rotating components is characterized by forced convection, in which the fluid to be sealed is moved by rotational drag flows because of the surface adhesion condition. The fluid also adheres to the stationary components such as housing parts. In between, a complex three-dimensional coiling flow is formed from the predominant circumferential component and the superimposed axial flow. The dissipating heat transfer phenomenon consists primarily of heat conduction within the laminar boundary layer superimposed with the intensive mixing within the turbulent swirling range. Overall, the heat dissipation is thus controlled by thermodynamic and hydrodynamic processes in the system.

Heat convection within the turbulent fluid proceeds very quickly and occurring temperature gradients within the convective laminar flow are usually very small. Such gradients are therefore often deliberately neglected or overlooked. Namable temperature gradients between laminar surface layers and pronounced mostly turbulent flow layers, on the other hand, are known. These phenomena are characterizing the heat transfer.

2 Conservation of energy

The first law of thermodynamics teaches us the conservation of energy.

$$\delta Q + \delta W = const. \tag{1}$$

With Q – heat and W – dissipated work (the friction), which means that all kind of friction as the heat sources (Figure 7) must be converted into heat, conducted through the parts and removed to the heat sinks – the fluid to be sealed.

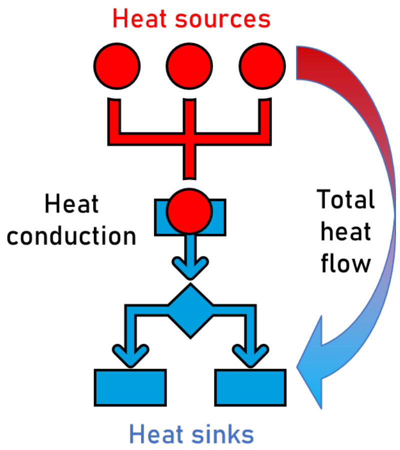

Figure 7: The total heat flow and heat conduction – simplified model

The basis of the heat flow is therefore:

$$\dot{Q} = K \cdot A \cdot \Delta T \tag{2}$$

With Q – the heat flux (heat per time and thus the power consumption by the friction of the seal), K – the total heat transfer coefficient, A – the cross-sectional area (or a relevant model replacement cross-sectional area), and ΔT – the average temperature difference between the heat sources (the sealing gap) and the heat sinks (the cavity of the seal). For the determination of the temperature difference, the gap temperature and the average liquid temperature must be known. The temperature levels and the relevant replacement cross-sectional heat transfer area of the seal face are not scope of this paper, but it can be estimated, or the temperature allocation can be calculated by FEA.

With a given calculated or measured friction power consumption Q of the seal and a surface A specified by the designer for a specified estimated heat transfer coefficient K, a defined temperature difference ΔT and thus a sealing gap temperature will occur. The inside seal gap temperature as well as the outside surface temperature of the seal face or the seal carrier can be reversed calculated. Since the specified goal must therefore be the minimization of the sealing gap temperature, the overall heat conduction K must therefore be maximized.

3 Heat conduction through a multi-layer wall – the relevant overall heat transfer coefficient K

The total heat transfer coefficient K can be estimated by the model of the heat conductivity through a multi-layer wall (Figure 8). Based on the design of the mechanical seal the components of the model are defined as follows:

- Fluid A – the fluid inside the sealing gap with its local properties depending on temperature T_A
- With the boundary layer 1 with the thickness δ_1 and the surface temperature T_1
- Material 2 – the seal's face material (e.g., Carbon or SiC)
- Material 3 – the tiny air gap between the seal face and the seal carrier
- Material 4 – the seal carrier
- The boundary layer at the OD of the seal carrier with the boundary layer thickness δ_5 and the surface temperature T_5
- Fluid B – the fluid to be sealed with its mean or local properties depending on temperature T_B

The heat transfer inside the sealing gap (the boundary layer 1) is normally neglected because of the dimension of the total gap height of less than 1 µm – much less compared to any boundary layer thickness. The surface temperature T_1 is equivalent with the mean seal gap temperature. The material 3 (a supposed air gap) represents the reduced sum of the surface roughness of the seal face and the carrier – reduced because of elastic and plastic deformation during shrink fitting assembly – or any real existing gap. With solid construction, material 1 to 3 is reduced to the seal's face material. All of that is well known and under the responsibility of the seal maker.

The boundary layer 5 represents the relevant mostly laminar Prandtl boundary layer at the interface from the seal maker's responsibility to the pump designer's scope of control.

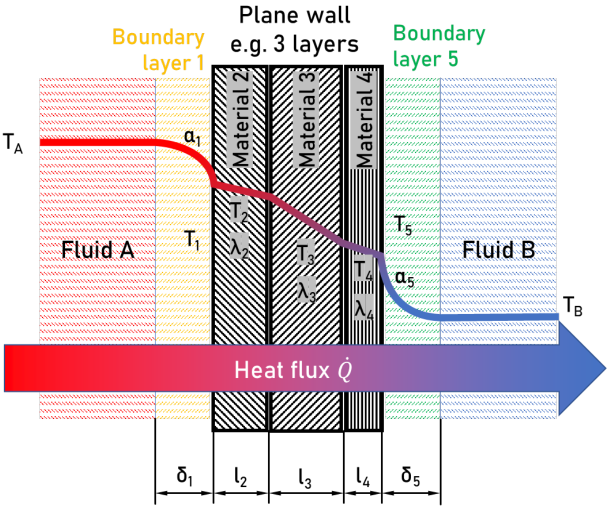

Figure 8: Heat conduction through a multi-layer wall – model

The relevant total heat transfer coefficient K is now as follows:

$$K = \frac{1}{\frac{1}{\alpha_1} + \frac{l_2}{\lambda_2} + \frac{l_3}{\lambda_3} + \frac{l_4}{\lambda_4} + \frac{1}{\alpha_5}} \tag{3}$$

With α – the heat transfer coefficient, l – the corresponding length (mostly in that case any wall thickness or an axial length), and λ – the specific heat conductivity. As mentioned before α_1 is infinitely high and the boundary layer δ_1 is infinitely thin because of dimensions. For the presented consideration for the design of the sealing cavity, the heat transfer α_5 and thus the thermal boundary layer δ_5 is relevant.

Now the goal for the optimization is to increase the heat conduction K by reducing any geometrical lengths l of the heat flux, or to influence the heat conductivity of the material λ, or to influence the heat transfer α as well as the surface area by increasing.

3.1 The influence of the circumferential flow

Exclusively the laminar Prandtl boundary layer with its linear velocity drop dragged along by the rotating components with the outside radius R in the circumferential direction is responsible for the direct heat transfer between the rotating surface and the liquid to be sealed in the direct vicinity zone r/R of the cylindrical surface.

The fluctuation movement generated by impulse exchange, on the one hand side removes energy from the average circumferential drag flow of the fluid by heat conduction via a vortex cascade, on the other hand side the additional swirling losses of the rotor are converted into heat by internal dissipation. With a good approximation under the consideration of temperature distribution, the circumferential speed first drops sharp (Figure 9) radially based on the viscose shear strain creating shear stress and then drops reduced smooth based on the swirling impulse exchange reaching a constant circumferential speed at a certain distance r - R. For the limited cavity (EN 12756) it will reach zero-speed with a comparable sharp drop near the housing wall vice versa.

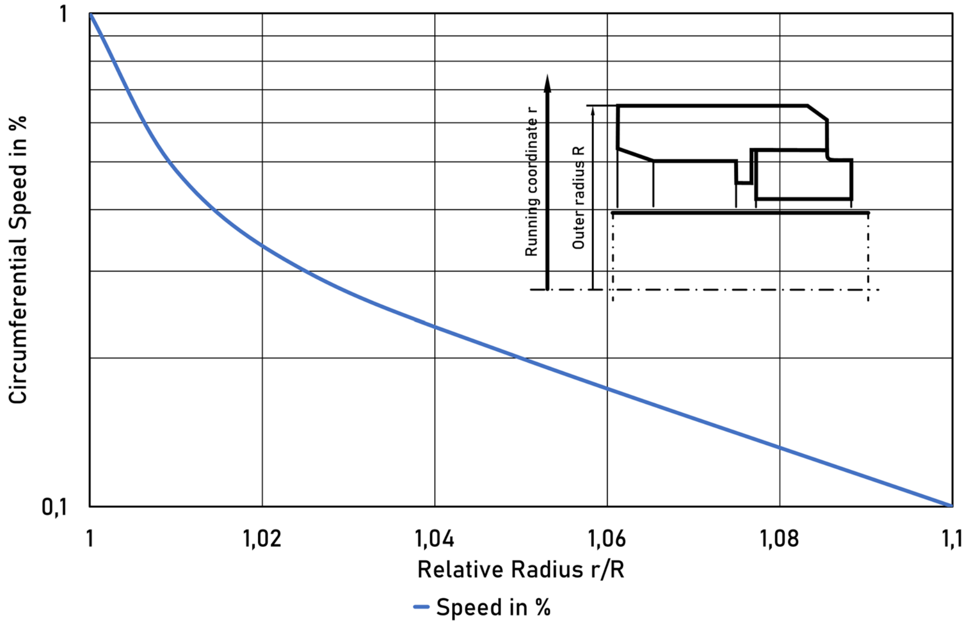

Figure 9: Circumferential speed by viscous shear relative to the radius (Example)

The resulting viscos friction affects the heat transfer and leads to additional heating of the fluid in the direct vicinity zone of the carrier's surface again influencing mostly viscosity and heat conductivity depending on the temperature. The influence of the dissipation swirling elements based on the viscose friction, which are mostly neglected in the energy conservation equations, can even lead to a local reversal of the global heat transfer in the case of high surface velocities, high viscosities, such as for oil sealing, caused by a local temperature maximum. It can be disastrous.

At high speed, the flow component is dominant in the circumferential direction compared to the axial direction. Occurring three-dimensional secondary flows are strongly geometry dependent.

- Subordinate is the flow in the axial direction

The superordinated respective equation for the total heat dissipation Q for the mechanical seal's axial flow is:

$$\dot{Q} = \dot{m} \cdot c_p \cdot \Delta T \quad (4)$$

With m – the mass flow, c_p – the specific heat capacity of the fluid, ΔT – the temperature difference between inlet and outlet. API recommendations are shown in Figure 10 without any reference to the geometry.

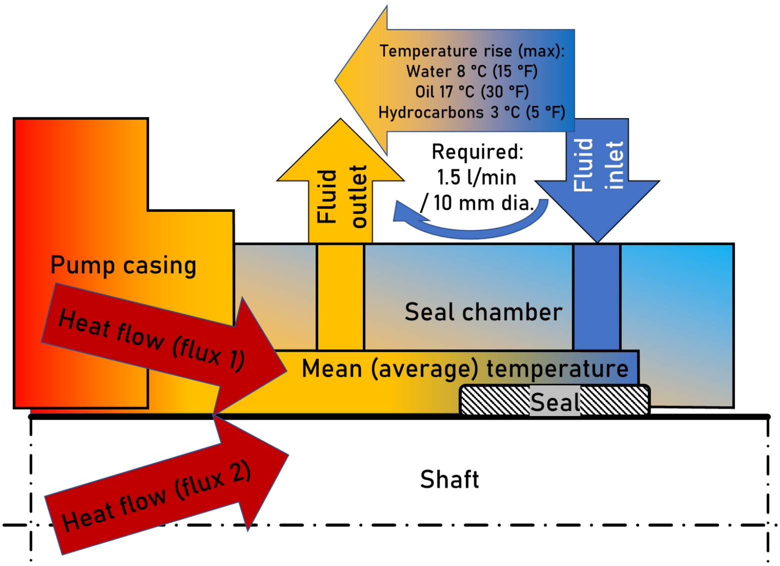

Figure 10: Heat transfer to mechanical seals according to API rules

- Exclusively the circumferential drag flow is decisive for frictional heat transfer of the seal (the purpose of this paper)

Laminar Couette flow is the fluid dynamics flow of a viscous fluid in the space between two flat relative moving surfaces with the distance – the gap height – h(r) based on the Navier-Stokes equation. The relative motion of the surfaces imposes a shear stress on the fluid and induces circumferential flow in the space in between. A basic requirement of this flow is that viscosity and thus shear stress is constant throughout the space. The first derivative of the velocity u(r), du/dh is constant. According to Newton's Law of viscosity (for Newtonian fluid only), the shear stress τ is the product of du/dh and the (constant) fluid viscosity η. One-dimensional flow is valid when both plates are infinitely long in the tangentially and perpendicular directions. Taylor–Couette flow (Figure 11) is a flow between two rotating, and thus infinitely long in the circumference, coaxial cylinders with a certain distance in the radii. Assuming the inner cylinder (with the radius R_i) rotates at a constant angular velocity ω relative to the outer stationary cylinder (with the radius R_o), then the velocity u(r) in the radial direction is non-linear:

$$u(r) = \frac{\omega \cdot \left({R_o}^2 \cdot {R_i}^2\right)}{{R_o}^2 - {R_i}^2} \cdot \frac{1}{r} - \frac{\omega \cdot {R_i}^2}{{R_o}^2 - {R_i}^2} \cdot r \qquad (5)$$

Taylor's development of hydrodynamic stability theory of Couette flow demonstrated that the adhesive boundary condition was correct for viscous flows at a solid surface.

When the angular velocity of the inner cylinder is increasing above a certain threshold, the flow becomes unstable, and a secondary steady state characterized by axisymmetric toroidal Taylor vortices with alternating direction of rotation occur due to centrifugal forces. The threshold rotational speed depends beside the circumferential speed from the gap height. With any superposition of an axial flow spiral vortex flow arises depending on the speed of the axial flow. Upon further increased angular speed the flow system becomes more unstable leading to states with higher complexity, with a so-called wavy vortex flow with global turbulences finally ending up in total chaotic swirl at further increased speed which is best for heat dissipation.

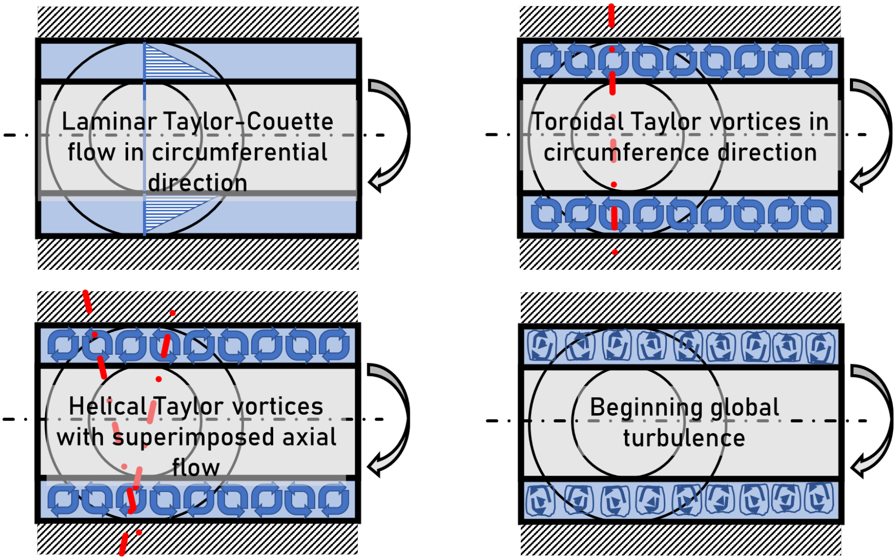

Figure 11: Couette flow, Taylor-Couette vortices, global turbulences

Figure 12 demonstrates the model of Prandtl's boundary layer theory of a fluid flow along a flat wall. After a certain length x_u the laminar boundary layer becomes instable, and it collapse to a thin boundary laminar viscous layer.

This model of Prandtl in combination with Couette flow and Taylor's vortex theory is now to be transferred to rotating cylinders and thus to mechanical seals. For the consideration of the circumferential flow in mechanical seals, with good approximation Taylor's theory can be assumed, because the gap between rotating and stationary parts is normally small compared to the diameter. In the drag flow, the flow itself meets again after each circumference. The flow thus becomes "infinite", and the flow thus becomes mostly chaotic turbulent swirling with a very thin laminar viscous ground flow layer on the rotating surface exclusive responsible for heat transfer.

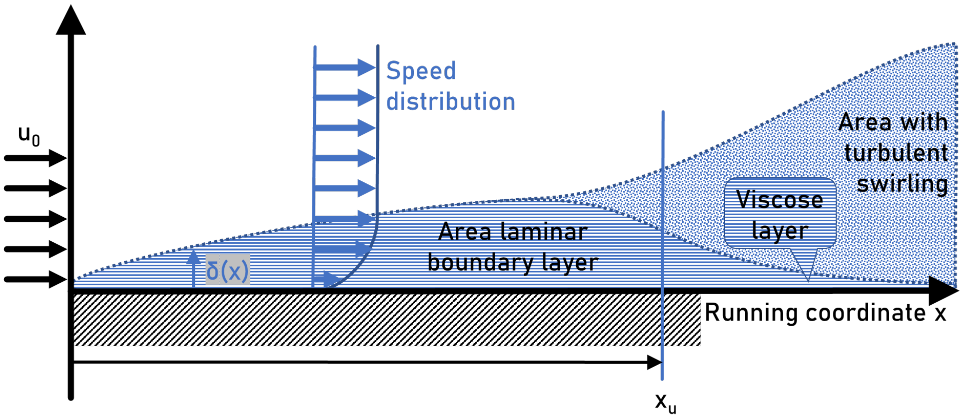

Figure 12: The viscous boundary layer based on Prandtl's Boundary Layer Theory

The laminar or turbulent swirling flow ensures the total heat dissipation to the buffer fluid system mentioned above.

3.2 Rotational Reynolds Number

The non-dimensional rotational Reynolds Number is used in problems of rotating parts in viscous fluids. It is equal to the product of its angular speed ω [1/s] the radius R_i, the gap hight $h = R_o – R_i$ and divided by the kinematic viscosity v of the fluid.

$$Re_r = \frac{\omega \cdot R_i . h}{\nu} \tag{6}$$

The threshold rotational Reynolds Number is mostly indicated with 10,000 to 80,000 for the transition from laminar to turbulent flow depending beside the geometry of the rotor and stator itself on different lead-in-area geometrical parameters.

Figure 13 shows exemplarily a calculation of the heat transition coefficients α (turbulent and laminar) for a usual 50/50 Water/Glycol mixture and the corresponding rotational Reynolds Number Re_r for mechanical seals. At the intersection of the heat transfer coefficients, the transition point is recognizable at 20,000 at a circumferential speed of 9 m/s.

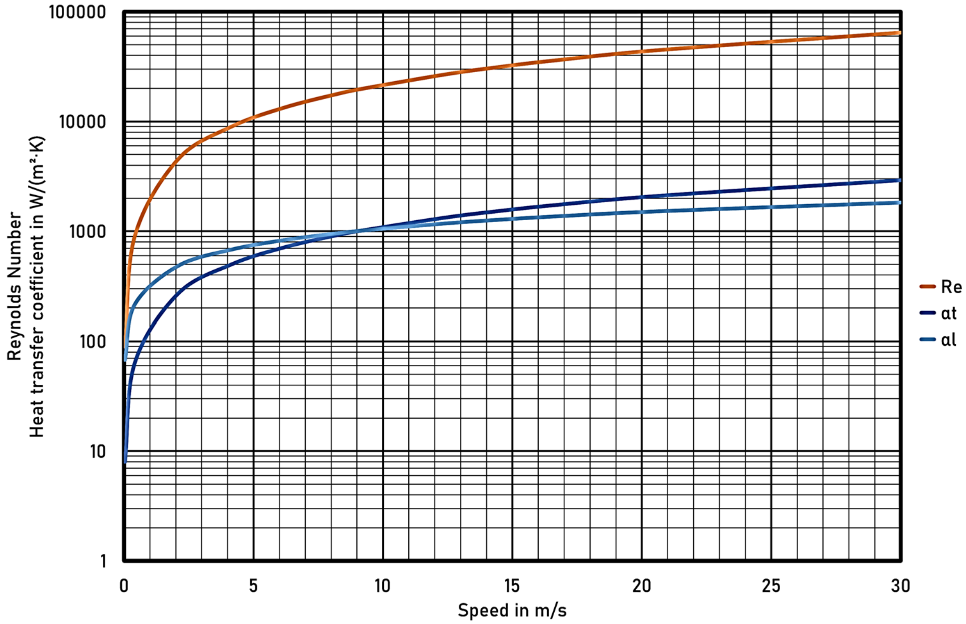

Figure 13: Estimation of the threshold rotational Reynolds Number for mechanical seals (exemplary)

3.3 The estimation of Prandtl's viscose boundary layer

From the perspective of the near surface heat transfer in mechanical seals and the infinite flow the viscous boundary layer thickness δ as a function of the circumferential length can be calculated. The derivation is not discussed in more detail due to the high effort.

The mean viscous laminar boundary layer is initially evenly applied to the entire rotating body. Vortices do not arise inside the layer. The viscous laminar boundary layer thickness δ_l is calculated as:

$$\delta_l = 5 \cdot \frac{x_u}{\sqrt{Re}} = 5 \cdot \frac{\pi \cdot d_m}{\sqrt{Re}} = \frac{10 \cdot \pi \cdot R_i}{\sqrt{\frac{\omega \cdot R_i \cdot h}{\nu}}} \tag{7}$$

If any critical diameter d depending on the circumferential speed (based on the threshold rotational Reynolds Number) is exceeded and the flow thus is transacted into complete turbulence, the viscous turbulent boundary layer thickness is δ_t of the drag flow swirling is calculated as:

$$\delta_t = 0.37 \cdot \frac{x_u}{\sqrt[5]{Re}} = \frac{0.74 \cdot \pi \cdot R_i}{\sqrt[5]{\frac{\omega \cdot R_i \cdot h}{\nu}}} \tag{8}$$

The constructive gap height h between the rotating seal and the stationary machine part should exceed the critical value δ_t if possible.

When considering, the widely described transition range of the Reynolds number must be evaluated. A clear separation cannot take place due to the strong geometry influences, e.g., surface roughness or any geometrical deviation from the smooth cylindrical surface. Instabilities in the transition area must be expected.

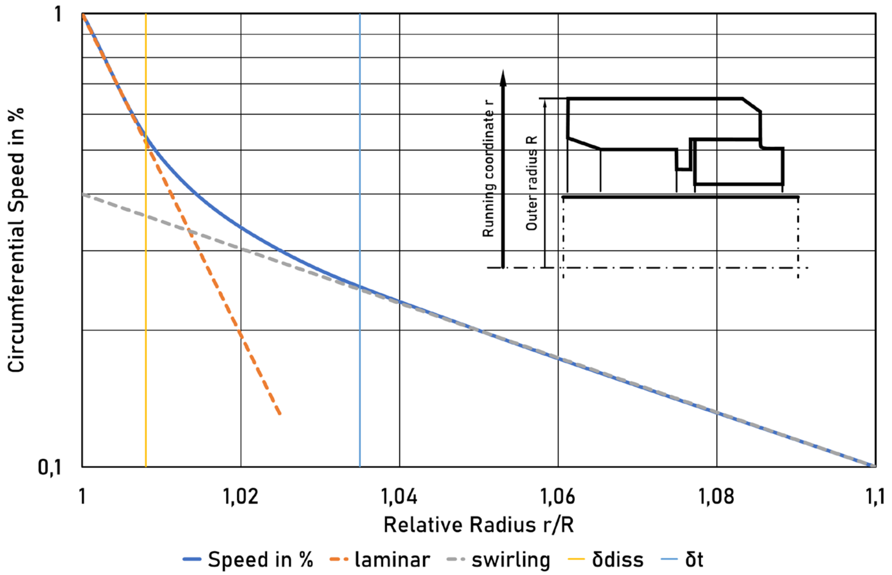

Figure 14: Circumferential speed and the influence on the thermal viscous layer thickness and the turbulent boundary layer (exemplary)

3.4 The local dissipative surface heat transfer coefficient α_{diss} for forced convection

In reality, it must be considered, that the viscosity ν itself is temperature dependent and thus the corresponding Reynolds and Prandtl Number are three-dimensional functions of the position. Local values of the heat transfer coefficient α_{diss} (dissipative) are important for computer simulations and for theoretical considerations. Mostly an average value is sufficient.

In the viscous boundary layer on the rotating wall surface, the flow is laminar or turbulent at any time (depending on the rotational Reynolds Number) and the heat transport takes place mainly by the heat conduction λ_F of the fluid. In this case, the local dissipative heat transfer coefficient α_{diss} for the drag flow is simplified as:

$$\alpha_{diss} = \frac{Nu \cdot \lambda_F}{\pi \cdot D_i} \tag{9}$$

With λ_F – the mean thermal conductivity of the fluid at the mean wall temperature T_m:

$$T_m = \frac{T_F + T_W}{2} \tag{10}$$

With T_F – the mean fluid temperature in the area with turbulent swirling, mostly as the average between inlet and outlet temperature, T_W – the mean surface temperature of the wall. The gap temperature and the wall temperature mostly are unknown. They can be calculated reverse based on the friction power loss and the mean temperature T_m of the fluid to be sealed.

The decisive factor seems primarily the thermal conductivity λ_F of the liquid at the surface temperature of the wall in combination with the Nusselt Number which applies for laminar flow:

$$Nu_l = k^* \cdot 0.664 \cdot Re^{1/2} \cdot Pr^{1/3} \tag{11}$$

and for turbulent flow:

$$Nu_t = k^* \cdot \frac{0.037 \cdot Re^{4/5} \cdot Pr}{1 + 2.443 \cdot Re^{-0.1} \cdot (Pr^{2/3} - 1)} \tag{12}$$

and finally, with the Prandtl Number (appropriate for the fluid's properties):

$$Pr = \frac{\nu \cdot \rho \cdot c_p}{\lambda} \tag{13}$$

and the relation k* between the Prandtl Number at the fluid and at the wall

$$k^* = \left(\frac{Pr_F}{Pr^W}\right)^{1/4} \tag{14}$$

The Prandtl Number must be calculated twice for fully trained fluid flow Pr_F and surface wall Pr_W conditions.

3.5 Findings

Beside geometrical effects, the heat transfer depends mainly on the thermal conductivity of the fluid to be sealed, and strongly on the flow character represented by the Reynolds Number, as well as the fluid properties represented by the Prandtl Number. With the general thickness of any thermal boundary layer δ_{diss} with its sharp drop of the circumferential speed and the local temperature (Figure 15) it is now apparent that the heat flux Q to be dissipated is limited, but in addition to the heat transfer surface A (the component surface) and the surface temperature difference ΔT will be influenced.

One can now see the pronounced dependence on the partly strongly temperature-dependent material values listed in Table 1 exemplary.

Table 1: Relevant mean properties of selected fluids

Fluid	Thermal conductivity λ in W/(m·K)	Kinematic viscosity ν in mm²/s	Density ρ in kg/m³	Specific heat capacity c_p in kJ/(kg·K)
Water	0.60	1.00	1000	4.18
Water / Glycol	0.39	4.11	1072	3.31
Glycol	0.26	16.00	1122	2.45
Ethanol	0.17	1.51	789	2.40
Benzene	0.14	14.00	720	2.20
Hydraulic Oil	0.13	5 … 46	± 950	± 2.00
Air	0.03	0.000015	1.2	1.01

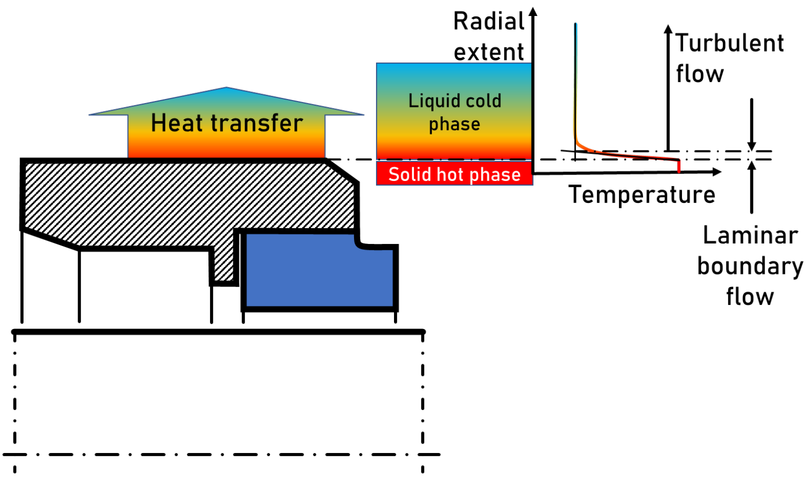

Figure 15: Heat transfer at the typical cylindrical surface of a mechanical seal

Figure 16 shows exemplarily the strong effect of the fluid properties on the heat transfer coefficient. Figure 17 shows exemplarily the effect of the gap height between the rotating cylindrical outer surface of the mechanical seal to the stationary wall of the seal's cavity.

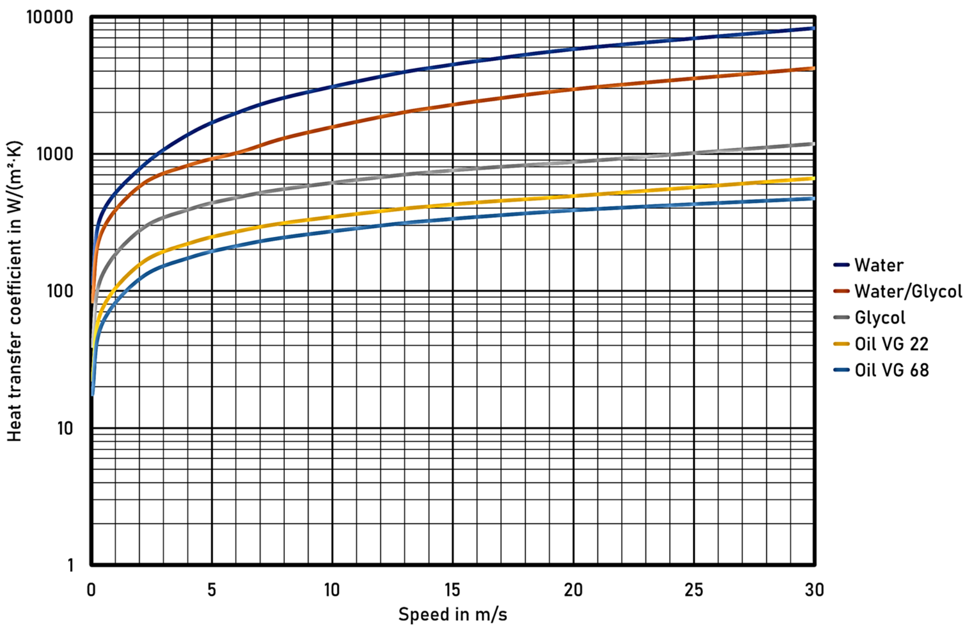

Figure 16: Influence of the fluid to be sealed on the heat transfer coefficient (gap height 6 mm)

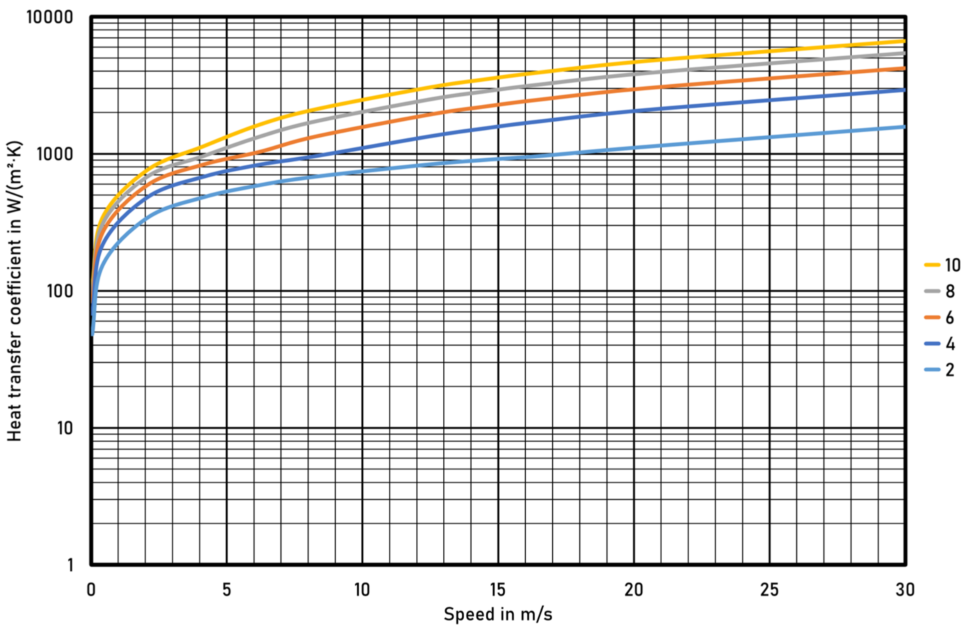

Figure 17: Influence of the cylindrical gap height between rotating and stationary part of the mechanical seal (example for Water/Glycol Mixture)

4 Summary and Conclusion

4.1 The heat balance of a mechanical seal

The entire heat balance of a mechanical seal is characterized by the following components:

- Sliding axial surface friction influenced by
  - friction condition and sliding speed
- Swirling losses influenced by
  - the liquid to be sealed, in particular its viscosity
  - the geometric installation conditions, in particular gap heights
  - the surface geometry of the rotating components, in particular grooves, holes, protruding components such as screws or pins creating additional turbulences
- Heat soak from the machine to be sealed (operating conditions) influenced by
  - the temperature differences
  - the heat exchange surface
  - the thermal conductivity of a possibly multi-layered wall of the housing
- Ambient conditions influenced by
  - heat dissipation or supply to or from the environment, by heat radiation, free or forced convection (ventilation)

The frictional heat dissipation to the fluid to be sealed (flush, buffer or barrier fluid) takes place via the usual considerations of heat transport. In the case of double seals, the heat dissipation is divided between the possibly existing product-side flushing F and the liquid buffer or barrier system LB. The division of frictional heat dissipation is complex.

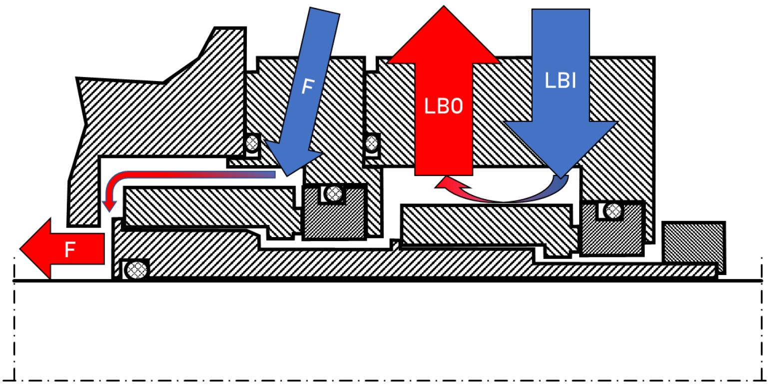

Figure 18: The fluid flow through a dual mechanical seal – tandem arrangement

Figure 18 (for example, flow according to API) is of interest. For the total heat balance mostly, the flushing is neglected because of additional safety, or it is considered only for the internal seal separately. Especially for API Plans 01, 11, or 12 (internal flushing without heat exchanger) the temperature of the flushing liquid is only essential for cooling of the internal seal. The purpose of flushing in double seals is mostly cleaning the internal seal. For hot liquids, there are Plans 21, or 22 with coolers for the flushing available. The most common API Plan for the outer seal is Plan 52 (non-pressurized buffer), or 53 (pressurized barrier) with closed cooling loops.

Internal and external seals can be viewed separately without restriction following the above-mentioned considerations. Only the control of the friction performance of the axial surface is to be dealt with in detail.

Decisive for the required amount of cooling liquid is the value of the specific heat capacity c_p (Table 1). This specific heat capacity provides information about the ability of a substance to store heat and thus thermal energy. It marks the energy needed to heat the mass 1 kg of a given substance by 1 degree Kelvin (ΔT).

Finally, the required volume flow V per unit of time is derived from the mass flow via the density ρ:

$$\dot{V} = \dot{m} \cdot \rho \tag{15}$$

Figure 10 shows the maximum permissible (API recommendations only) temperature differences for water (8 °C or 15 °F) and oil (17 °C or 30 °F) as well as highly volatile hydrocarbons (3 °C or 5 °F) due to the risk of evaporation such as alcohol as a typical example of sealing fluids for double mechanical seals for pharmaceutical applications derived from empirical values. The different API values for water and oil are based on the almost inversely proportional specific heat capacities c_p.

4.2 Heat conduction

The seal surfaces friction power dissipation must first be directed via heat conduction to the component cylindrical or axial surface as shown in Figure 19 schematically. The details cannot be discussed at this point. The two-dimensional temperature distribution can be calculated by FEA in detail. This is not scope of this paper. Nevertheless, a simplified one-dimensional model with reduced cross-sectional area and uniform length of conduction meets the results with good accuracy, according to personal experience. A highly simplified model presentation shows Figure 20. In any case, a temperature difference between the gap temperature and the carrier surface temperature must be assumed based on power loss. Depending on various parameters such as component geometry and specific thermal conductivity λ of the materials, a three-dimensional real surface temperature profile is created. Even on the backside of the carrier a reversed temperature difference between cylindrical surface and fluid to be sealed can occur due to swirling losses.

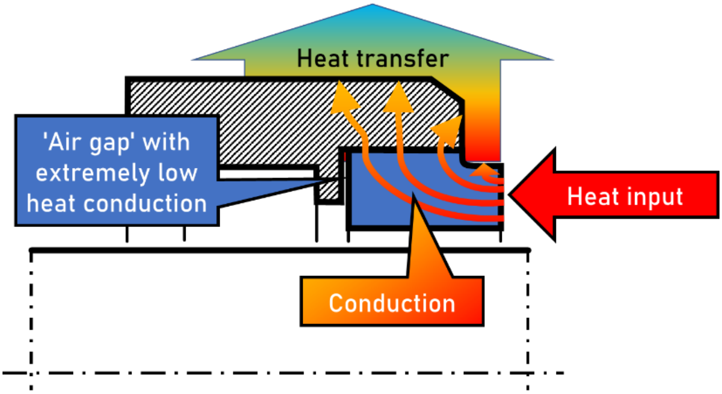

Figure 19: Heat conduction through mechanical seals from the sealing gap to the carrier surface

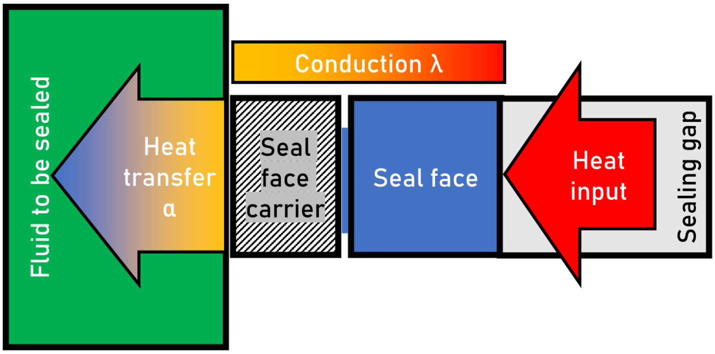

Figure 20: One-dimensional model presentation for heat conduction through mechanical seals

The mechanism for dissipating thermal energy is heat conduction. According to the law of thermodynamics, heat flows from warm to cold. Thermal energy is not lost – the system is adiabatic – the rate for energy conservation applies. The dissipated heat output Q per unit of time is described by Fourier's law[1], which is as follows for the simplified case of a solid body with two parallel wall surfaces already shown in Figure 8 earlier.

$$\dot{Q} = \frac{\lambda}{l} \cdot A \cdot \Delta T \qquad (16)$$

The corresponding parameters are

- λ – the specific thermal conductivity
- l – the average conducting length – as a substitute the length of the seal face + the cylindrical thickness (approx. the face width)
- A – the average cross-sectional area – as a substitute, the sliding surface

[1] 1822 Jean Baptiste Joseph Fourier

- ΔT – the temperature difference – the potential

With appropriate composite design consisting of several components (for example, a shrink fitted construction), the model for a multi-layer wall (Figure 8) can be used. The equation of Fourier's law then expands to n layers because of the conservation of energy.

$$\dot{Q} = \frac{\lambda_1}{l_1} \cdot A_1 \cdot \Delta T_1 = \frac{\lambda_2}{l_2} \cdot A_2 \cdot \Delta T_2 = \cdots = \frac{\lambda_n}{l_n} \cdot A_n \cdot \Delta T_n \tag{17}$$

If the considered cross-sections A are equal or one uses any respective replacement area, each local temperature, e.g., the gap or surface temperature can be calculated, or with the replacement area A_r it can be summarized to:

$$\dot{Q} = \frac{1}{\frac{l_1}{\lambda_1} + \frac{l_2}{\lambda_2} + \cdots + \frac{l_n}{\lambda_n}} \cdot A_r \cdot \Delta T_{total} \tag{18}$$

If one considers the heat conduction both over the seal face and the mating ring and over different components and heat transfer surfaces, Kirchhoff's laws of mesh and knot rule generally apply.

4.3 Heat transfer

The heat dissipation refers to the heat transfer between the solid phase of the rotating component of a mechanical seal and the liquid phase of the fluid to be sealed, resulting in a significant temperature drop. Ideally, the phase state is stable without vaporization, and the heat is dissipated exclusively via heat conduction in the liquid phase. However, material properties change depending on temperature. Decrease in density at elevated temperature of the surface layer immediately leads to self-convection due to centrifugal gravity. Convectively moving particles dissipate increased heat energy, which is much higher than that of heat conduction. The hot component surfaces are cooled much more strongly. If the self-convection is now superimposed by forced flow, the heat transfer increases even more, the fluid property values also change more, the temperature drop becomes even higher.

The clear recommendation is to use the seal face material with the better heat conductivity for the rotating part because of extended heat transfer.

Heat transfer between different phases (solid to liquid) is more complicated than heat conduction within a particularly solid phase. In addition to the pure material properties, partially transient flow processes also play a role. Phase changes, for example during evaporation, can occur.

Fluids with high viscosity, such as, e.g., oils, have high Prandtl numbers up to Pr = 1,000 and higher. It seems that the high viscosity of the fluid at constant temperature thus ensures that the velocity gradient from the wall edge condition can only form slowly.

Nevertheless, due to the strong temperature dependence of the viscosity of oils the gradient of the speed in the immediate vicinity of the wall is very high. The thermal conductivity of the fluid is low. Therefore, a strong temperature gradient is formed at a very short distance from the component. The circumferential velocity gradient of the fluid with its sharp drop is similar.

The heat transfer coefficient α_{diss} depends on

- the flow processes in the immediate vicinity of the wall (represented by the rotaional Reynolds Number)
  - circumferential speed
  - gap height between rotating and stationary part
  - axial ribs or grooves (for torque transmission) as well as fixing screws or holes increase turbulence and thus heat transfer
  - rough surfaces increase thicker laminar boundary layers to a certain extent
- the material properties of the fluid (represented by the Prandtl Number)
  - viscosity changes, especially of hydrocarbons because of temperature gradients
  - specific heat capacity
  - density changes due to temperature gradients
  - thermal conductivity
- the temperature fields
  - destruction of the laminar boundary layer because of incipient pool boiling mode up to the critical heat flux or Leidenfrost phenomenon can occur at water sealing or light volatile hydrocarbons

4.4 Special flow profiles

Taylor-Couette flow refers to the flow of an incompressible viscous liquid located in the space between two coaxial cylinders rotating relative to each other based on drag flow. The flow between the cylinders depends not only on the rotational speed, but also on whether the inner or outer cylinder rotates. With rotating inner cylinders, as is common with mechanical seals, superimposed local Taylor vortices are self-forming under certain geometric conditions, or other macro-vortices are build depending on the geometry for seal’s cavity (the stuffing box) which can significantly influence local heat dissipation.

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seal-ing – the sealing doctor

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