Theoretical and Experimental Study of the Frictional Losses of Radial Shaft Seals for Industrial Gearbox

Michel Organisciak, Pieter Baart, Stellario Barbera, Alex Paykin and Matthew Schweig

The improvement of the energy efficiency of industrial gear motors and gearboxes is a common problem for many gear unit manufacturers and end-users. As is typical of other mechanical components, the radial lip seals used in such units generate friction and heat, thus contributing to energy losses of mechanical systems. There exist today simulation tools that are already helping improve the efficiency of mechanical systems — but accurate models for seal frictional losses need to be developed. In this paper SKF presents an engineering model for radial lip seal friction based on a physical approach.

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Introduction

Industrial gear units are widely used in power transmission systems. They are composed of shafts, gears, rolling elements bearings and dynamic lip seals. The performance of the seals is critical for the proper functioning of the system. The primary functions of the seals are to prevent the leakage of oil to the environment and to avoid the ingress of water or other contaminants into the mechanical system. Both can lead to a premature failure of the gear unit. In addition, the seals influence the system by generating friction and heat. The heat generated by the friction of the seals has an impact on the operational temperature of the gear unit as well as on the viscosity of the lubricant inside the unit. Moreover the seals contribute to the total energy losses of the mechanical system.

The improvement of the energy efficiency of industrial gear motors and gearboxes is a common challenge for OEMs and end-users. For instance energy efficiency classes are defined for electrical motors and gear motors. Moreover the power losses of gear units and seals can impact the total energy bill of an industrial installation. Therefore understanding seal friction generation and reducing it are essential challenges for seal manufacturers.

Simulation tools are commonly used to design mechanical components and systems. For the prediction of specific parameters like seal temperature or friction torque, specific models and calculation tools need to be developed. In this paper SKF presents an engineering model for the prediction of radial lip seal friction based on a physical approach. The friction model includes the generation of friction due to rubber dynamic deformation and lubricant viscous shear between the surfaces of a seal and a shaft. The friction model is coupled with a heat generation and seal thermal model. Indeed, seal friction and seal temperature are closely related: the heat generated in the sealing lip is conducted through the seal and shaft and dissipated into the environment. This changes for instance the lubricant viscosity.

The model is verified step by step in an extensive experimental study. Measurements of seal friction, seal temperature and lubricant film thickness have been performed for various dynamic lip seals. The analyzed parameters are: surface speed, oil viscosity, seal material, seal size, seal lip style and duty cycles. The correlation between model predictions and experimental friction measurements can therefore be verified.

This unique modelling capability allows selecting or developing shaft seals which would meet and exceed the demands of modern gearbox applications. It also enables gearbox manufacturers to bring to the market better performing and more reliable gearboxes.

Seal Friction Modeling

Physical phenomena influencing seal friction. The friction force, F_T, is the force resisting the relative motion of two bodies when a normal force, F_N, is applied to the contact between these bodies. The coefficient of friction, μ, can be defined as:

$$\mu = \frac{F_T}{F_N} \quad (1)$$

The coefficient of friction is not constant for radial shaft seals, which makes the prediction of seal frictional torque much more complicated. This has been demonstrated in various studies. Plath (Ref. 1) in 2005 developed a seal friction model based on finite element analysis. They assumed initially a constant coefficient of friction for the seal-shaft contact. However this led to inaccurate results and they demonstrated that it was necessary to take into account the variation of temperature of the seal due to the generated frictional heat to accurately predict seal friction.

More recently, the studies from Haas (Refs. 2–3) have revealed the influence of surface roughness and of the duty parameter G (representing the lubricant viscosity, angular speed and contact pressure) on the friction coefficient. Their papers show that the friction coefficient follows a Stribeck-like curve (Fig. 1). A transition between mixed and fully lubricated regime is clearly shown in the evolution of the friction coefficient.

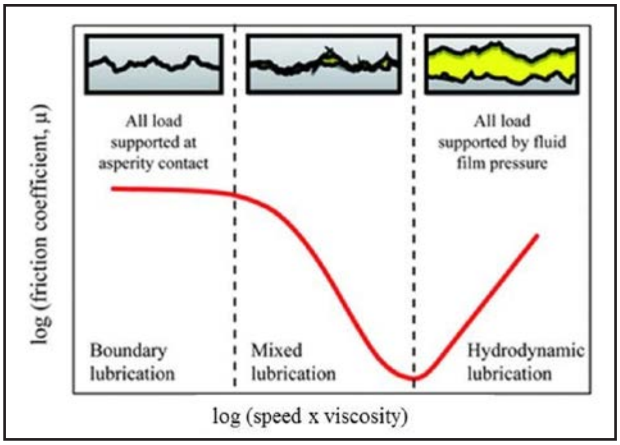

Figure 1 Stribeck curve: coefficient of friction as a function of contact speed and lubricant viscosity.

The variations of coefficient of friction in a radial lip seal contact can be attributed to three phenomena:

1. ***The variation of lubricant viscosity as a function of temperature***. Typical curves for standard gearbox oils are shown (Fig. 2).
2. ***The variation of the coefficient of friction between rubber and steel.*** As shown by Grosch (Ref. 4) and Hermann (Ref. 5), the coefficient of friction varies significantly — between 0.1 and 3 in extreme cases — as a function of temperature, sliding speed and pressure in dry and lubricated conditions. This is due to the fact that for rubbery material, friction is essentially governed by the dissipation of energy during the dynamic deformation of the rubbery material on the counter-face.
3. ***The variation of rubber modulus with temperature.*** A typical curve is shown in Figure 3. The prediction of seal friction is a complex task and requires a model being able to predict the temperature in the seal and in the contact and to take into account the variations mentioned in the previous paragraph.

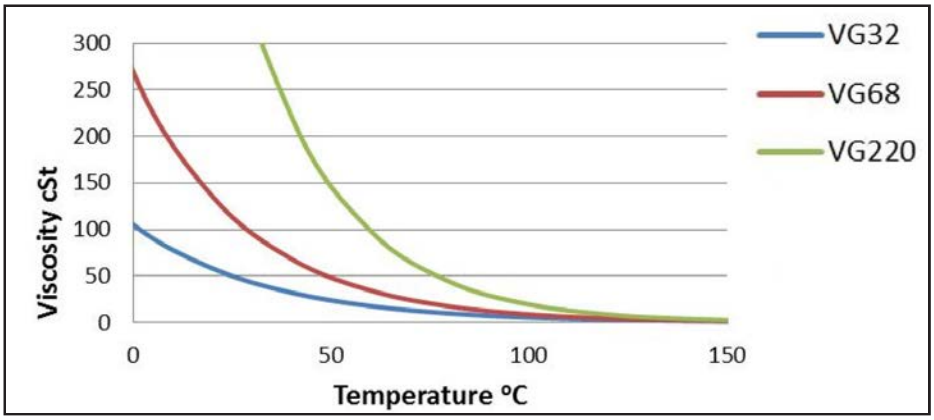

Figure 2 Lubricant viscosity as a function of temperature for VG32, VG68 and VG220 oils.

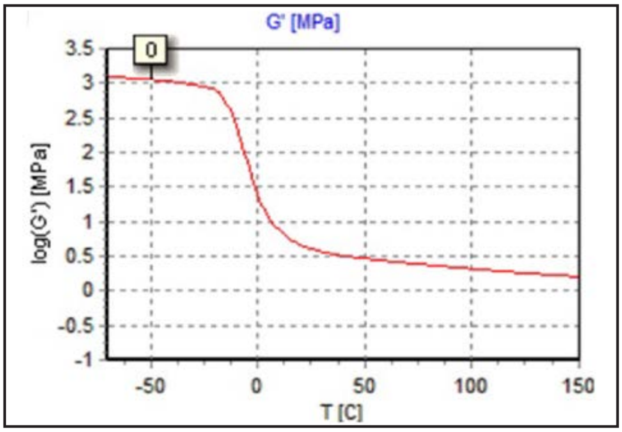

Figure 3 Modulus as a function of temperature of a typical NBR material.

Friction model. The friction between the seal and shaft is considered to be generated by two main governing phenomena:

1. ***Lubricant viscous shearing***. This takes place in the contact between the lip and the shaft surface. The frictional force produced in this manner is defined as F_{lub}.
2. ***Viscoelastic losses***. This is due to dissipation in the rubber as its surface is dynamically deformed by the shaft roughness asperities. The frictional force generated by the rubber material is referred as $F_{material}$.

Taking both these effects into account, the total frictional torque T_{Torque} can be expressed as:

$$T_{Torque} = (F_{lub} + F_{material}) \frac{D_{shaft}}{2} \tag{2}$$

Where
T_{Torque} is seal frictional torque Nm
F_{lub} is seal lip force N
$F_{material}$ is contribution of the material to the seal frictional force N
D_{shaft} is shaft diameter m

The material contribution is calculated following the relation below:

$$F_{material} = \mu_{dry} F_{tip} f(A_c) \tag{3}$$

Where
μ_{dry} is the coefficient of friction between the rubber and steel surface
F_{tip} is seal lip force N
f is a function of given variables
A_c is real contact area at the surface roughness level, which is calculated from contact mechanics, m^2

The lubricant contribution can be written as:

$$F_{lub} = \frac{\eta u}{h_e} S_{contact} \tag{4}$$

Where
η is lubricant viscosity in the contact, $P_a\, s$
u is surface speed, m/s
h_e is effective film thickness depending on the lip tip style (i.e., wave or plain), m

$S_{contact}$ is surface area where the lubricant is sheared, $m2$

The effective film thickness is based on elastohydrodynamic lubrication theory (Ref. 6) and can be written as:

$$h_e = f(\eta u)^{0.66} \tag{5}$$

The combination of these equations allows the calculation of the seal friction torque at any given speed and temperature.

Thermal dissipation model. The friction between a rotating shaft and a seal lip generates heat that is dissipated by the different components of the system. The power dissipated q_{disp} by the sliding contact can be written as:

$$q_{disp} = T_{Torque} \frac{2}{D_{shaft}} u \tag{6}$$

Where
q_{disp} is power dissipated in the sealing contact, W.

The generated heat flux in the seal/shaft contact is integrated into the heat conservation equation for the lip contact. The heat is then diffused in the shaft and seal according to the energy equation:

$$\frac{\rho C_P}{k} \frac{\delta T}{\delta t} = \frac{\delta^2 T}{\delta X} + \frac{\delta^2 T}{\delta Y} + \frac{\delta^2 T}{\delta Z} + f(q_{disp}) \tag{7}$$

Where
ρ is material density kg/m^3
C_P is heat capacity, J/K
k is heat conductivity, $W/(mK)$
T is temperature, K
t is time, s

The complete computational algorithm is indicated in Figure 4. Here, the friction model is combined with the thermal model. The effects of temperature change on lip force and oil viscosity are also included. The algorithm is transient, allowing computations for different speed cycles.

Experimental Techniques Used for Model Validation

The validation of the model is conducted for three parameters:
1. Lubricant film thickness in the contact (to validate Eq. 5)
2. Frictional torque
3. Seal temperature

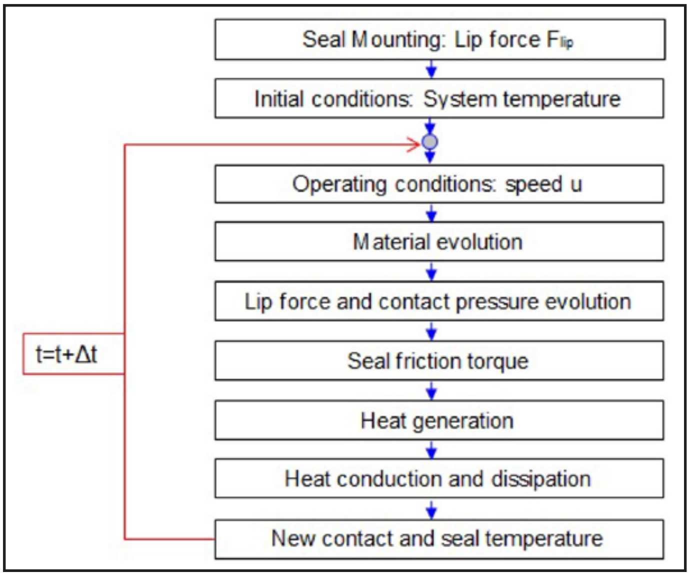

Figure 4 Calculation algorithm.

Film thickness measurements. The measurement of an absolute value of film thickness in the sealing contact has always been a challenge. For instance in 1992, Poll and Gabelli (Ref. 7) developed a method where they use magnetic fluid as a lubricant and measure the magnetic resistance through the lubricant film in the sealing contact. In the same period, Poll (Ref. 8) used the fluorescent technique: a fluorescent dye is added to the oil and is excited with a laser. The intensity of the light can be related to the film thickness in the contact. However, both techniques require complex calibration and specific equipment.

In this work, a capacitance technique using the SKF Lubcheck set-up is applied to measure the evolutions of lubricant film thickness in a radial lip seal/shaft contact. Seals molded from a special conductive rubber compound have to be used to realize the experiments. This compound is part of the SKF compound portfolio and has similar mechanical proprieties as standard sealing materials.

Figure 5 shows the electric schematic of the measurement system. V_{mx} is the maximum voltage applied to the system; C_{ref} is a reference capacitance added to the system; C_m is the capacitance of the sealing contact; and R_m is the electrical resistance of the seal itself. After calibration using lubricants with different viscosities and simultaneous friction torque measurements, the measured voltage V_{cap} can be related to the capacitance of the sealing contact and therefore to the lubricant film thickness. The system is implemented on the test rig shown in Figure 6.

Friction torque and seal temperature measurement. Seal friction measurements are performed on a specialized SKF test rig (Fig. 6). The shaft is driven by an electrical motor allowing a very wide, programmable, range of rotational speed. The central part of the test rig is the air bearing spindle, onto which the stationary seal specimen is mounted and the friction torque sensing unit is connected. The air bearing ensures that the measured friction is only due to the seal. The seal is lubricated with an oil bath and different oil sump volumes are possible.

In addition to the frictional torque, seal temperature is constantly recorded during the tests. Thermal measurements are made using a thermocouple placed in the spring groove of

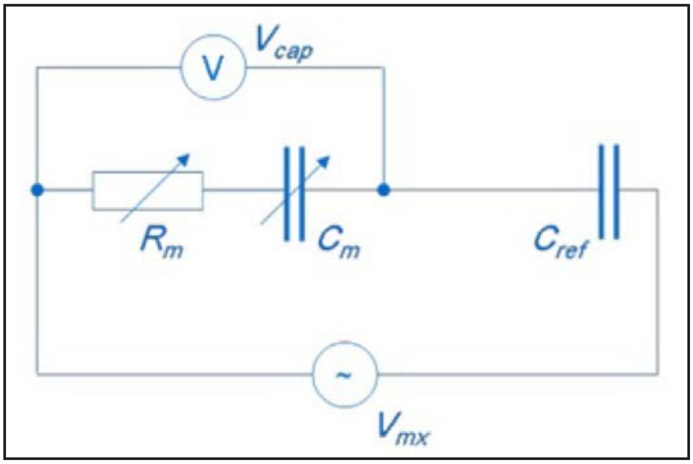

Figure 5 Electrical schematic for Lubcheck measurement.

the seal. The analysis of temperature changes is used in combination with frictional torque to validate the model.

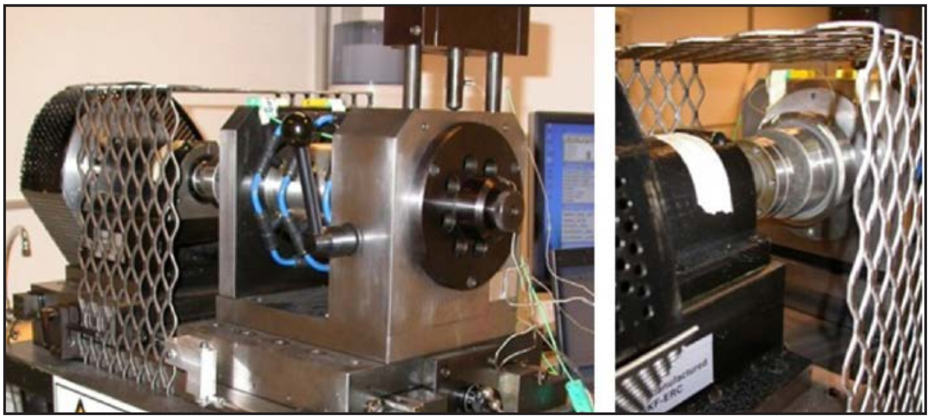

Figure 6 Seal friction measurement test rig.

Model Validation: Correlation Between the Model and Experimental Results

Film thickness. Using the set-up described earlier, the film thickness is measured for a seal with different oils having different viscosities and for different rotating speeds. Figure 7 shows the film thickness as a function of the product sliding speed u times lubricant viscosity η at the running temperature. The results can be fitted with a power law function:

$$\text{film thickness} = 3.45\,(u\eta)^{0.68} \tag{8}$$

displaying an R^2 value of more than 0.95.

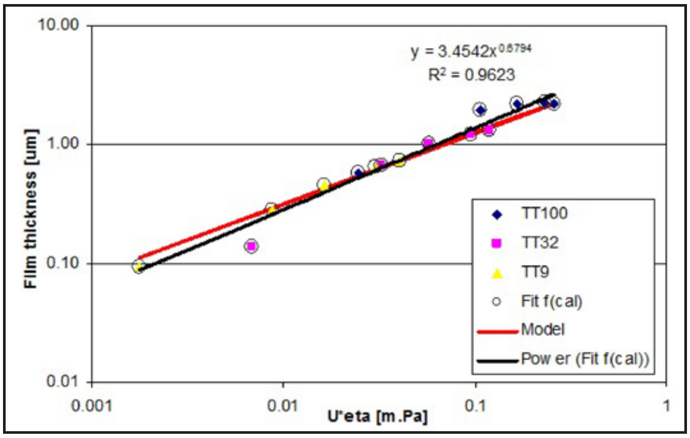

Figure 7 Measured film thickness for different oils with different viscosities and sliding speeds (points). Power fit of the experimental results (in black) and prediction by Equation 5 (in red).

The result from Equation 5 used in the model is added to the figure (in red). Equation 5 assumes a power 0.66 applied to the product ($u\eta$), which is very close to the numerical fit (Eq. 8). This shows a very good agreement qualitative between the theoretical formula and the measured film thickness, validating the approach in the model.

Model validation: seal friction and temperature. Measurements and seal friction and temperature calculations are performed for molded wave seals and trimmed plain lip seals (HMS 5 RG and V seals; Fig. 8). The two seal types are standard seals used in industrial applications, such as in gearboxes.

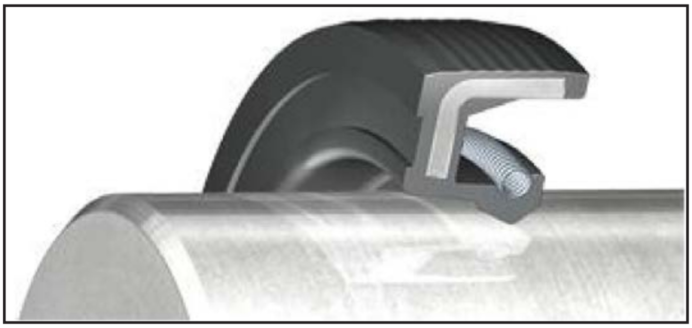

Figure 8 Typical cross-section of a trimmed plain lip HMS5 seal.

A typical example of experimental results and model predictions is shown in Figure 9. The graph on the left displays the used speed cycle, with different steps of speed between 10 and 1,000 rpm. The graph in the middle displays the predicted and measured frictional torque. The graph on the right shows the predicted and measured garter spring groove temperature, with additionally the predicted temperature in the contact. The graphs show that the model predictions are close to the measured friction and temperature.

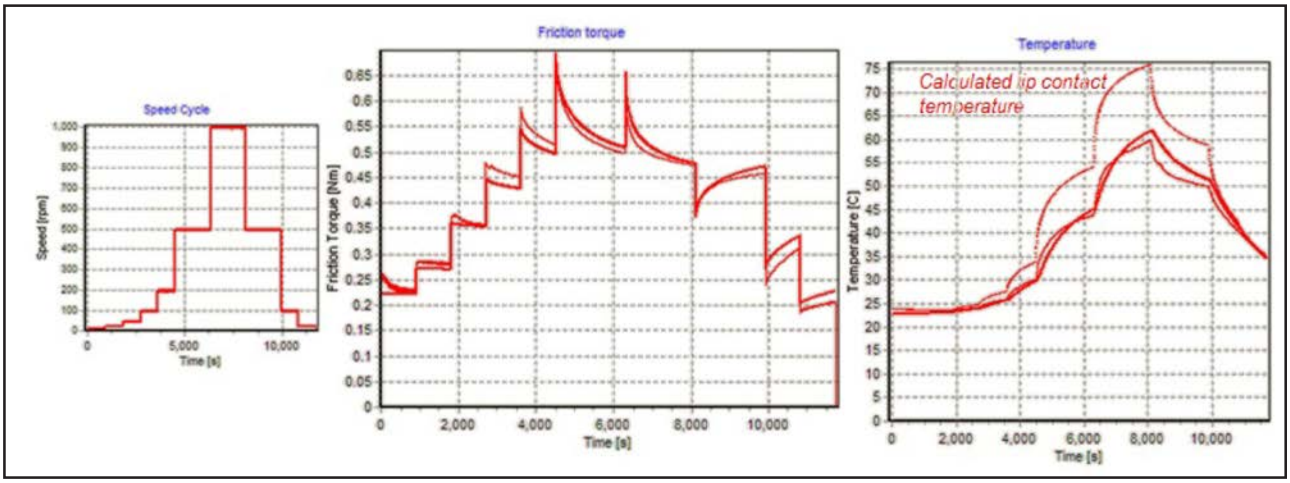

Figure 9 Speed cycle (left), friction torque (middle) and temperature (left) for a typical study case. Measurements are displayed with thin lines. Model predictions are displayed with bold red lines. The dashed line in the temperature plot (right) is the predicted contact temperature.

An extensive number of test conditions, different compounds (NBR and FKM) and lubricants have been used to validate the model. The left graph in Figure 10 depicts the correlation between the model predictions and the measurements for all tests for molded wave seals. The right graph in Figure 10 depicts the correlation for trimmed plain lip seals. For the purposes of the comparison, only the average friction obtained in the last 30 seconds at the end of each speed step is considered. The correlation plot shows that all the predictions are within 20% of the measurements results. The resulting correlation is high, showing an R^2 value of more than 95%.

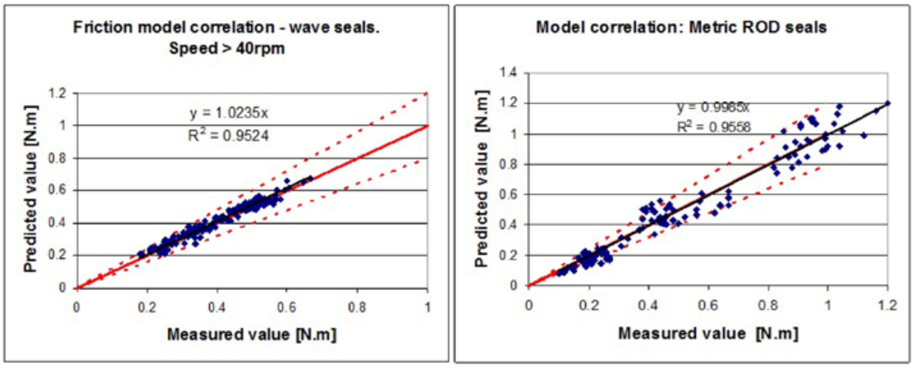

Figure 10 Predicted friction torque as a function of the measured friction torque for wave seals (left) and plain seals (right). The red dashed lines represent the interval at ± 20%.

With the very good correlations for the film thickness, frictional torque and seal temperature, we can conclude that the developed approach is validated and can be used for the prediction of seal frictional torque in an application.

Applications of the Model

Comparison between molded wave and trimmed plain lips seals. The model presented in this paper can be applied to trimmed plain lip seals or to molded wave seals (Ref. 9). As shown in Figure 11, the wave seal has a special sinusoidal contact patch on the running counter face. This enables a better lubricant flow at the vicinity of the lip and a higher lubricant film in the contact. This also enables a better heat exchange between the lip and shaft.

In order to study the difference between plain and wave seals, seals with the same cross-section and material are used for the experimental study. Seal friction and temperature are calculated in parallel with the model described in this paper. Figure 12 represents an example of the results for the speed cycle displayed in Figure 9. There is a clear difference in the friction between the two lip geometries. The wave lip reduces by about 20% the frictional torque during the tests. The friction reduction has a direct influence on the temperature, with a self-induced temperature decreasing by more than 10°C.

Figure 12 further illustrates the good match between the measured values of frictional torque and temperature and the model predictions, thus confirming the quality of the model. This also enables the usage of this approach to predict the seal torque and temperature in a mechanical system.

Influence of oil sump volume on seal friction. The volume of oil in a gear unit can vary for different applications. It influences the friction of different mechanical components and the temperature of the mechanical system. The model is used in order to study the influence of the oil sump volume on seal temperature and seal friction. The volume of oil has an impact on the dissipation of the heat generated in the sealing contact and therefore on the friction and temperature of the seal.

Seal friction measurements are also carried out with an oil sump of a volume 0.2 and 3 liters but maintaining the same oil level relative to the center of the shaft. The results of the model and of the measurements are displayed (Fig. 13). First they show a very good agreement between the model and the measurements. Secondly, both the model and the experiments confirm that the oil sump volume has an influence on the frictional losses. A system with more oil has a lower operating temperature because it is able to better dissipate the heat from the sealing contact. Consequently, the oil viscosity is higher, which results in a higher friction. Therefore it is very difficult to give an absolute value of seal friction in operations since it

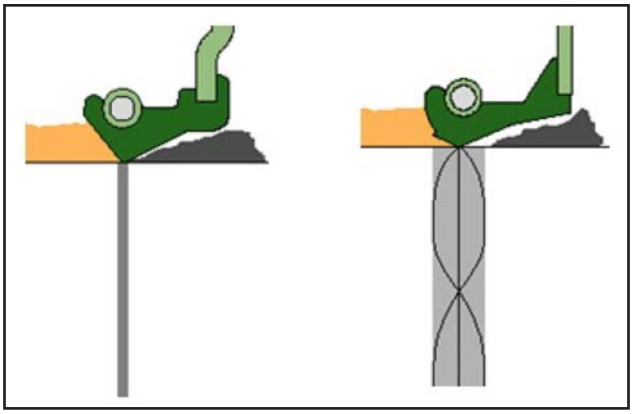

Figure 11 Difference of the contact patch between a plain seal (left) and a wave seal (right).

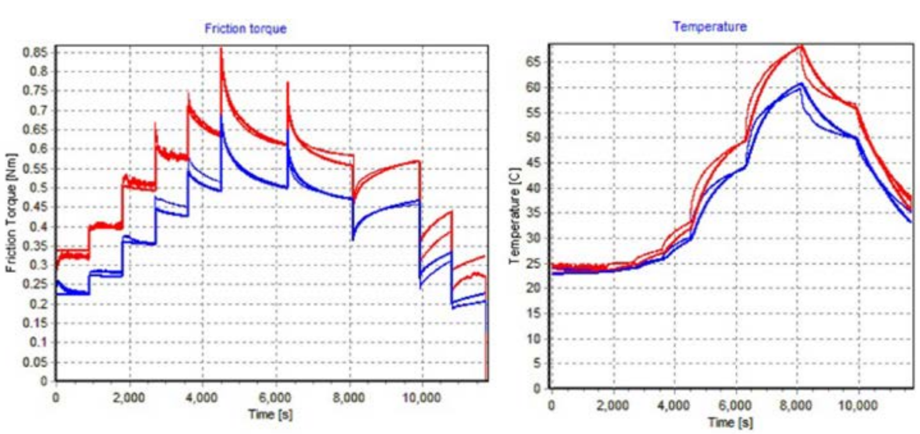

Figure 12 Friction torque (left) and temperature (right) measurements as a function of time for a wave seal (blue) and plain (red) seal. The thin lines are measurements; the thick lines are model predictions.

is highly dependent on the environment in which the seal operates. Only a combined seal frictional and thermal model is able to predict seal friction and frictional losses in an application.

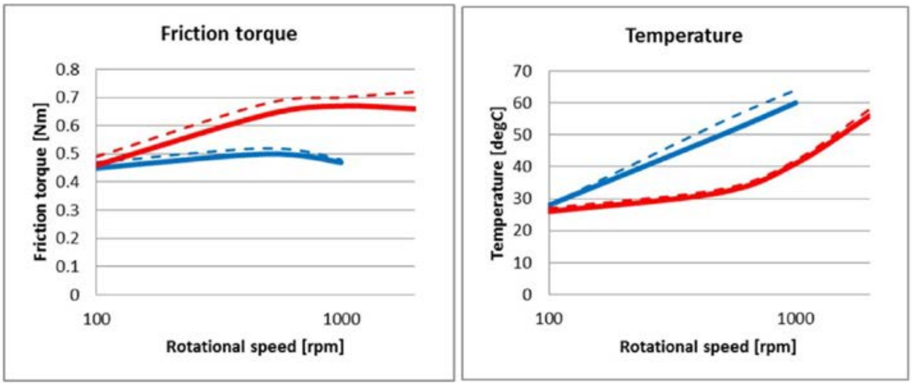

Figure 13 Measured (dotted line) and predicted (solid line) frictional torque (left) and seal temperature (right) for a 0.2 L oil sump (blue) and 3 L oil sump (red).

General Conclusion

The paper has presented a physical model to analyze and predict frictional torque and temperature of radial lip seals with a plain or wave lip geometry. A good correlation between experimental and theoretical results has been obtained. It has been shown that the reduction of friction has a direct effect on the self-induced temperature in the seal. Lowering seal friction decreases the operating temperature which in its turn can have a positive impact on other performance parameters such as material life and lubricant life.

The results have also shown the importance of considering the effect of operating conditions and temperature in the prediction of seal frictional torque in any environment and system. The heat induced by the friction of the sealing contact needs to be dissipated in the other elements of the mechanical system. Therefore the real operating temperature and frictional losses of a seal can only be accurately predicted if the friction model is coupled to a heat generation and heat dissipation model. This modelling approach is complementary to the simulation techniques for a complete gear unit presented by Wemekamp (Ref. 10).

In conclusion, the approach can therefore be used confidently to:

- Predict the seal friction in an application
- Optimize seal design by acting on the parameters influencing the friction and prediction the final outcome
- Together with other SKF simulation tools, analyze the performance of the seal in the application

These unique modelling capabilities will allow selecting and developing shaft seals which would meet and exceed the demands of modern gearbox applications. They enable also the design of better performing and more reliable gear units.

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Matthew Schweig is involved in global research and the advanced development of rotating lip seals at SKF Sealing Solutions. He joined SKF Sealing Solutions in 2005 as a product development engineer. In the last few years Schweig has supported all R&D activities at Sealing Solutions, including performance simulation tools development at the SKF Engineering and Research Center in the Netherlands.