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The Efficiency of a Simple Spur Gearbox - A Thermally Coupled Lubrication Model

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

A thermally coupled efficiency model for a simple dip-lubricated gearbox is presented. The model includes elastohydrodynamic (EHL) friction losses in gear teeth contacts as well as bearing, seal and churning losses. An iterative numerical scheme is used to fully account for the effects of contact temperature, pressure and shear rates on EHL friction. The model is used to predict gearbox efficiency with selected transmission oils whose properties were first obtained experimentally through rolling-sliding tribometer tests under representative contact conditions.

Although the gearbox was designed using standard methods against a fixed rating, the model was used to study efficiency over a much wider range of conditions. Results are presented to illustrate the relative contribution of different sources of energy loss and the effect of lubricant properties on the overall gearbox efficiency under varying operating conditions.

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Introduction

The efficiency of drivetrain components is quickly becoming a significant research area pushed by the ever intensifying quest for improved fuel economy in automotive vehicles. Emission-controlling regulations are becoming stricter, owing mainly to environmental issues such as air contamination but also due to the depletion of oil deposits and the resulting high fuel prices. Pursuing augmented efficiency of a drivetrain or its components can also have the positive side effect of decreasing the frictional heat that is generated inside the gearbox, differential or axle component therefore improving scuffing or pitting behavior.

The drive train in passenger cars absorbs around six per cent of the total fuel energy in combined city-highway driving - equivalent to about thirty per cent of the mechanical energy delivered to the wheels [1]. The biggest part of this energy loss ends up as heat in the axle or transmission lubricant resulting from friction, windage and churning. As a result the reduction of lubricant-related losses in a vehicle's drivetrain can lead to significant improvements in both the fuel economy and the environmental. Thus, this efficiency increase is a much sought-after goal for lubricant suppliers of OEM factory fill and aftermarket lubricants.

Background

The efficiency of spur gears is started receiving significant attention since about 1980's with the work of Anderson [2] [3]. Recent years have seen a boom in such publications owing mainly to the energy crisis. Li and Kahraman [4] and more recently Chang and Jeng [5] focused on a spur gear pair while Michaelis [6] considered a more integrated approach which included churning losses as well as bearing and seal losses. Churning losses play a very important role in the prediction of a dip lubricated component's efficiency and recent studies from Changenet and Velez [7] [8] have shown that the accurate determination of churning losses and how these are affected by design parameters is a challenging problem.

Thermal behavior of the components and thermal response of lubricants are dominant factors of efficiency enhancement. There are several published models of spur gear pairs that analyze the thermal behavior of the pair like those developed by Long and Lord [9] and Taburdagitan [10] which use finite element methods to predict the overall and surface temperatures. A more integrated approach by Changenet and Velez [11] considered lump thermal elements to study and model a six-speed gearbox, but with no experimental validation. In addition, the thermal response of transmission lubricants was extensively studied by Olver [12] who also developed a comprehensive model to predict traction in Elastohydrodynamic (EHL) contacts which included thermal effects [13]. However, there are currently no efficiency models that consider full spur gearbox including the all-important thermal coupling, taking into account the mesh and bulk temperature rise of all drivetrain components or that of the oil surrounding them as well as all sources of energy loss including bearings, seals, churning and EHL traction.

Additionally, the type of lubricant itself is a crucial factor in determining the efficiency of a drivetrain. Despite this, there is currently very limited ability to predict the relative fuel economy arising from the use of different drive train lubricants. Petry-Johnson [14] have included more than one type of lubricants in their studies in an attempt to pinpoint the possible effect that a specific combination of lubricant, component design and operating condition may have on the overall gearbox efficiency. The results from their studies showed a linear relationship between the gear power loss and the rotational speed of the gears and also highlighted the effect of surface finish on the efficiency of the gearbox. Their lubricant comparison used three different lubricants to indicate that there is a possible change in overall efficiency depending on the lubricant type. However, no complete method has been developed that is able to simultaneously account for specific lubricant characteristics under EHL contact conditions and relevant

gearbox parameters. Such an approach should be able to provide a more accurate estimation of the possible efficiency gain.

The limitations of current approaches include limited treatment of gear churning losses and not accounting for the transient conditions arising from variable vehicle duty cycle. Kolekar and Olver [15] have recently worked on this issue concentrating on hypoid axles. A transient thermal model coupled to a quasi-steady state lubricant traction and churning formulation has been used together with lubricant bench tests in order to predict the energy that is dissipated during specified drive cycles. The results highlight the high influence that the properties of axle oils have and that the ranking order of lubricant composition and properties depends greatly on the specified duty cycle, with high viscosity, friction modified oils being favored for high power use. In contrast, lower viscosity fluids than are currently in use provide lower losses for city and light highway duty.

In addition to treating only the axle, this work has other significant shortcomings including the lumped mass axle temperature that does not capture the bulk lubricant temperature, the lack of a change gearbox model as well as the lack of provision to determine the effect of drive train efficiency on the engine behavior and therefore on fuel consumption.

The present work is designed to offer significant improvements in the accuracy of efficiency predictions for simple spur gearboxes including the effects of lubricant rheology on gearbox efficiency. The model accounts for EHL friction losses in gear teeth contacts, bearing and seal losses, and losses due to oil churning. The EHL friction losses are calculated based on lubricant properties that have been obtained through extensive measurements on a ball-on-disk tribometer, while also fully accounting for the effect of thermal coupling so that any increase in lubricant temperature (and the corresponding change in lubricant properties) due to power losses, is fed back to EHL friction calculations. The ambient temperature of the lubricant in the gearbox is calculated using a multi-physics finite element model which includes all conductive and convective heat transfers within the gearbox. Such a holistic approach, particularly the inclusion of thermal coupling, will enable the model to account for the transient conditions due to particular usage history and therefore predict the efficiency for a given drive-cycle.

Methods

This section first outlines the basics of the EHL model used to predict tooth frictional losses, followed by the methods to predict bearing, seal and churning losses. Finally the FEM approach used to calculate the ambient (oil bath) temperature rise, which is fed back to the EHL model, is outlined.

The experimental method used to extract lubricant characteristics that are used as input to the EHL model is also described.

Basic EHL model

A thermally coupled EHL model was devised adopting the approach of Olver and Spikes [13] which predicts the friction coefficient in the rolling-sliding EHL contact of a lubricated disc pair.

The model uses the EHL regression equations developed by Chittenden [16] to calculate the minimum and central film thickness:

$$\frac{h_0}{R'_x} = 3.63 \bar{U}^{0.68} \bar{G}^{0.49} \bar{W}^{-0.073} \quad (1)$$

$$\frac{h_c}{R'_x} = 4.30 \bar{U}^{0.68} \bar{G}^{0.49} \bar{W}^{-0.073} \quad (2)$$

where

- h_0 is minimum EHL film thickness, m;
- R'_x is reduced radius of contact, m;
- \bar{U} is entrainment velocity, m/s;
- \bar{G} is gravitational acceleration, m/s²;
- \bar{W} is total contact load, N;
- h_c is central EHL film thickness, m.

Where the non-dimensional parameters

$$\bar{U} = \frac{U\eta_0}{E'R'_x} \quad (3)$$

$$\bar{G} = \alpha E' \quad (4)$$

$$\bar{W} = \bar{W}_L = \frac{W}{E'R'_x L} \quad (5)$$

are the speed, load (L = line contact) and material parameter respectively.

where

- η_0 is inlet viscosity (at $p = 0$), Pa.s;
- E' is reduced modulus of elasticity, Pa;
- α is pressure-viscosity coefficient;
- \bar{W}_L is total contact load in line of contact;
- L is length of EHL contact, m.

The mean shear stress is calculated based on the Ree-Eyring approach as adapted by Evans and Johnson [17], and the traction regime is decided using the following three-stage process described by Olver and Spikes which is based on the non-dimensional Deborah and S numbers (Figure 1).

The friction coefficient is then predicted by means of a convergence loop; the loop is initiated by assuming an initial friction coefficient μ_0 and then the temperature rise due to shear heating of the lubricant $(\Delta T_{oil})_{av}$ and contact of asperities $(\Delta T_i)_{av}$ are calculated using the approach developed by Olver [13].

$$(\Delta T_i)_{av} = \frac{1.06 \alpha_h \dot{q}}{A k_1} \left(\frac{\chi_1 \alpha_0}{U_1} \right)^{1/2} = 1.06 B_1 \alpha_h \dot{q} \quad (6)$$

Question	Criterion	Mean shear stress formula
Stage 1		
Is the response viscoelastic?	i.e. $D_0 > 1$	
Is there shear thinning?	i.e. $S > \sinh 1$	
Stage 2		
If the response is <i>not</i> viscoelastic <i>and</i> there is <i>no</i> shear thinning, then		$\bar{\tau}^* = S$
If there is shear thinning <i>but</i> the response is <i>not</i> viscoelastic, then		$\bar{\tau}^* = \sinh^{-1} S$
If the response is viscoelastic <i>but</i> there is <i>no</i> shear thinning, then		$\bar{\tau}^* = S[1 - D_0(1 - e^{-1/D_0})]$
If the response is viscoelastic <i>and</i> there is shear thinning with $\tau^* > 2$, then		$\tau^* \approx \ln(2S) - \ln(1 + 2 S - \frac{1}{2} e^{-S^*/(2D_0)})$ (see text)
Stage 3		
Is the plastic limit reached?	i.e. $\bar{\tau}^* > \tau_c^*$	
If so, then discard previous value and put		$\bar{\tau}^* = \tau_c^*$

Figure 1. The rules for determination of the mean shear stress [13]

(D_0 is Deborah number ($= \eta_0 U / (2\alpha G e)$), S is non-dimensional strain rate ($= \dot{\gamma} \eta_0 / \tau_E$), τ^* is non-dimensional shear stress ($= \tau / \tau_E$) and τ_c is the limiting shear stress)

$$(\Delta T_{oil})_{av} = \frac{\mu W U_s h_c}{8 A k_{oil}} = \frac{U_s \bar{\tau} h_c}{8 k_{oil}} \quad (7)$$

where

- α_h is heat partition;
- \dot{q} is heat generated by sliding in the contact, W;
- A is EHL contact area, m^2 ;
- k is thermal conductivity, W/m-K;
- χ_1 is thermal diffusivity of material, m^2/s ;
- α_0 is EHL semi contact width, m;
- B is transient thermal resistance;
- U_s is sliding velocity, m/s;
- τ is shear stress, Pa;
- k_{oil} is thermal conductivity of the oil, W/m-K.

Using these temperatures in conjunction with the skin (boundary) temperature rise, T_B , calculated using a heat partition α_h between the two surfaces, the mean film temperature, \bar{T} , can be calculated.

$$\alpha_h = \frac{1.06 B_2 + \frac{h_c}{2 k_{oil} A} + M_2}{1.06 (B_1 + B_2) + \frac{h_c}{k_{oil} A} + (M_1 + M_2)} \quad (8)$$

$$T_B = \frac{U_1 T_{B1} + U_2 T_{B2}}{U_1 + U_2} \quad (9)$$

$$\bar{T} = T_B + (\Delta T_f)_{av} + (\Delta T_{oil})_{av} \quad (10)$$

where

- M is steady state thermal resistance of contacting body.

The temperatures are repeatedly evaluated until the desired convergence is achieved (usually within 0.1 °C or less). Once these are known, the dissipated power in the form of heat can be calculated by simply multiplying the load with the friction coefficient and the sliding speed.

The EHL model assumes that the basic lubricant properties do not change within the contact and are calculated either in respect to the inlet conditions or to the mean film temperature, based on the assumption by Evans [18]. The lubricant properties that are used are based on a synthetic 75W90 type gear lubricant and they are calculated using a linear approach for the variation of density and pressure viscosity coefficient with temperature. The shear modulus is calculated using an exponential approach quoted by Muraki [19]. The ASTM equation based on the ASTM D341-722 standard is used to calculate the low shear rate viscosity of the lubricant, based on measurements at 40 °C and 100 °C.

$$\log \log (v + 0.7) = b - c \log T \quad (11)$$

where

- v is kinematic viscosity, m^2/s ;
- b is tooth face width, m;
- c is constant.

The model can successfully predict the traction regime and friction coefficient of a combination of materials and lubricants. Roughness of the surfaces is also taken into account by means of the non-dimensional lambda (λ) value which necessitates the use of a boundary modified friction coefficient of the form shown below, suggested by Smeeth and Spikes [20] for $m = 2$.

$$\mu_{\text{eff}} = \mu_f + \frac{\mu_b - \mu_f}{(1 + \lambda)^m} \quad (12)$$

where

- μ_{eff} is effective or mixed friction coefficient;
- μ_f is fluid friction coefficient;
- μ_b is boundary friction coefficient.

The model converges in less than 5 iterations for all but the most extreme conditions and can be used as a base to simulate a lubricated gear pair. The basic algorithm described here is summarized in the flowchart of Figure 2.

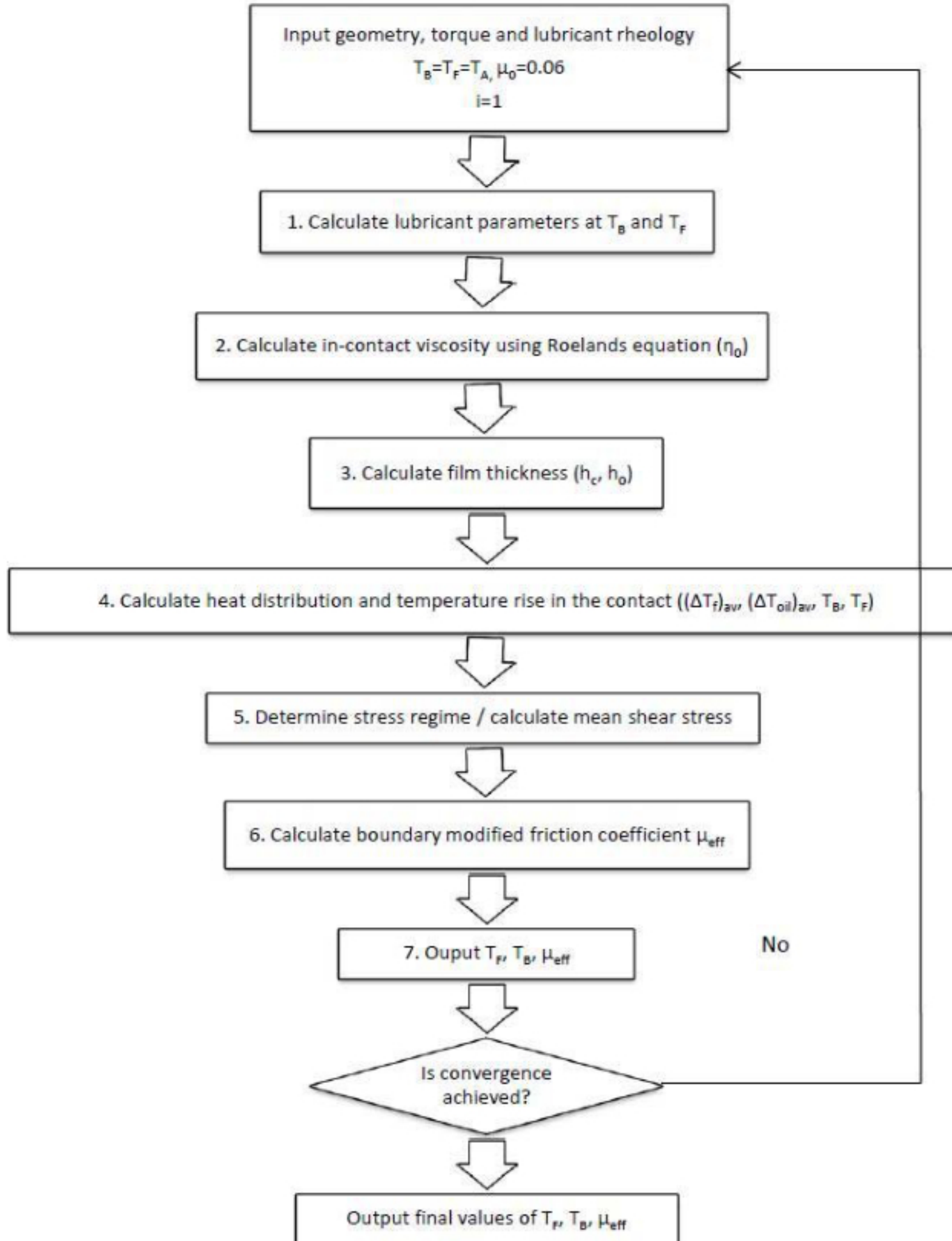


Figure 2. Flowchart of the EHL model algorithm

Extracting lubricant rheology coefficients

The rheological characteristics of the lubricant play a crucial role in determining the thermal and frictional behavior. Previous studies [13] have shown that EHL contacts can operate at temperatures that are significantly higher than the ambient oil bath temperature. It is therefore necessary to produce an extended lubricant rheology database which will include the typically high temperatures that are encountered in practice.

As stated earlier, the core of the EHL model is the Eyring model [21]; it is therefore assumed that the lubricant is following the Eyring behavior where the relationship between the shear rate and the shear stress is described by the equation:

$$\dot{\gamma} = \frac{\tau_0}{\eta} \sinh \left(\frac{\tau}{\tau_0} \right) \quad (13)$$

The viscosity in the second part of the equation is the high pressure, in-contact viscosity which is in turn described by either the Barus [22] or the Roelands equation. In this case, the complete equation originally proposed by Roelands [23] which is both pressure and temperature corrected was used:

$$\frac{\eta_P}{\eta_\infty} = e^{\left(\log e \left(\frac{\eta_r}{\eta_\infty} \right) \left(1 + \frac{p}{p_r} \right)^z \left(\frac{T_r + 135}{T + 135} \right)^{s_0} \right)} \quad (14)$$

where

η_P is in-contact viscosity, Pa.s;

η_∞ is viscosity constant (= 0.0000631 Pas);

η_r is measured viscosity at atmospheric pressure;

p is pressure;

p_r is reference pressure

T_r is reference temperature.

135 is temperature constant where the viscosity becomes infinitely high (-135°C);

The atmospheric slope index S_0 is calculated using the Roelands chart [23] (p. 58) and the reference low shear rate viscosity at two (2) temperatures. The measured values of η_r are listed in Table 1.

In order to extract the required coefficients for Roelands equation, bench tests were carried out using an MTM ball-on-disc test rig to measure the friction and an EHD rig to measure the film thickness of the lubricant under varying loads and temperatures. The former uses a steel ball pressed against a steel disc to measure the traction coefficients under given conditions while the latter uses the optical interferometry technique to measure the film thickness of the lubricant through a transparent glass disc. The two rigs are shown in Figure 3.

The rheology coefficients namely the Eyring stress τ_E and the Roelands parameter Z_R were then derived from the measured data using an approach similar to that outlined by LaFountain [24]. This involves fitting the Eyring model to the measured traction curves obtained from the MTM tests and then extracting the coefficients once the appropriate fit is achieved. The coefficients were then incorporated into the EHD traction model to improve the accuracy of the calculations. The lubricants modelled here are both fully synthetic 75W90 gear oils and are believed to be of similar composition but they come from different manufacturers.

Table 1. Measured values of η_r

Lubricant	Type	40 °C	100 °C
A	Synth 75W90 gear oil	0.0751	0.013
B	Synth 75W90 gear oil	0.0926	0.0135

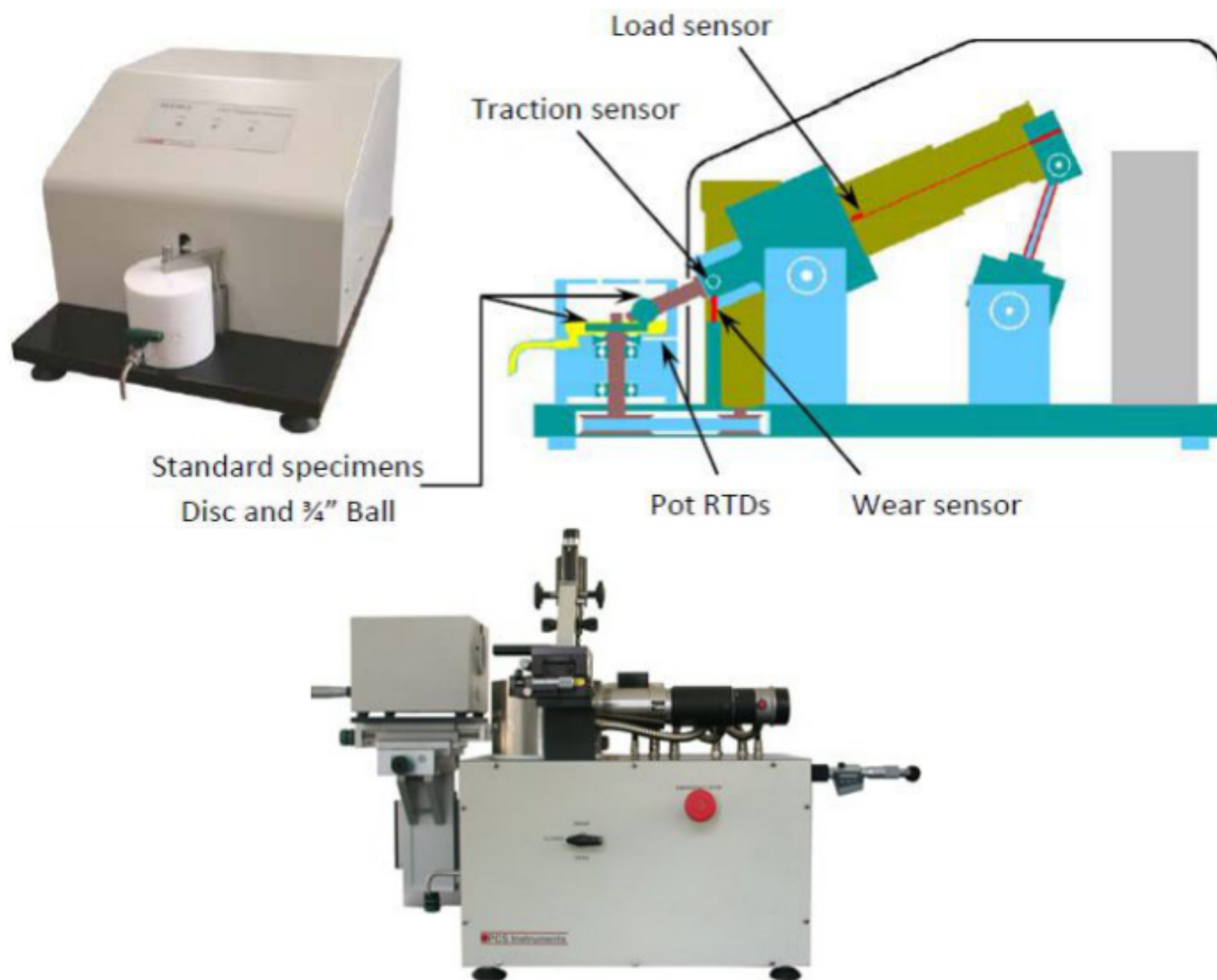


Figure 3. The MTM2 (above) and the EHD2 rig (below) by PCS Instruments that were used to measure friction and film thickness respectively

Spur gear pair design and modelling

With the rheology adequately modelled, the next step of the process involves designing a spur gear pair for the required transmitted torque, desired life and reliability. The gear design procedure that is used is compliant with the British Gear Association and in accordance with international design standards [25]. The gears and the bearings were designed for 12 months of continuous operation which is equivalent to around 9000 hours. For simplicity, a 1:1 gear ratio was chosen.

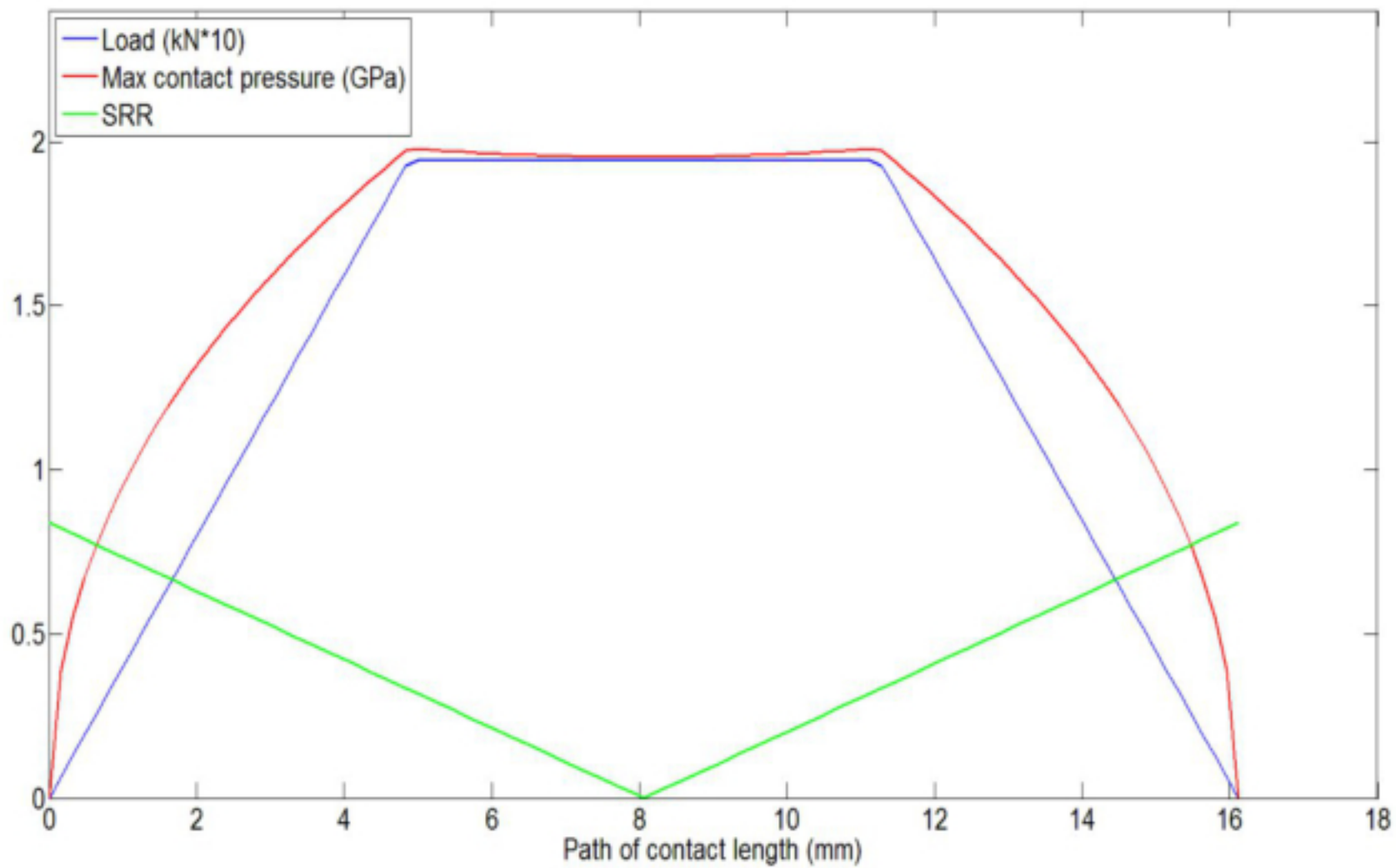
After the rough design process is complete and the basic parameters of the gear pair are defined, the geometrical parameters of the pair can be calculated according to appropriate standards [26]. The basic specifications of the selected gears are shown in Table 2. The gears that comprise the gear pair are identical and are similar to the 23T gears mentioned in Petry-Johnson [14]. The full model covers the geometry of spur and helical gears, however in the presented study only spur gears have been considered. The load profile and the resulting pressure distribution of the gear pair are shown in Figure 4. Note that the contact pressure for this application is in the high end for a gear region at almost 2 GPa and that the slide roll ratio varies drastically along the path of contact and is zero at the pitch point.

With the geometrical parameters defined, the EHL friction model can be applied to the gear mesh to predict the coefficient of friction and the thermal parameters during the variable conditions of a gear cycle.

The current tooth mesh cycle was used as a basis for the development of the model, and in each point of the line of contact, a sub-routine was applied to calculate the converging friction coefficient and the resulting temperature rise inside the contact (mean film temperature). As the roughness of the gears has also been taken into account, a boundary friction coefficient of 0.11 was assumed in order to also consider the boundary/mixed lubrication regime. The inputs that have been used for the design and calculation of the gearbox are roughly based on a 2013 Dodge Ram 3500 Semi Truck and are shown in Table 3.

Table 2. The basic design parameters of the gear pair

Parameter	Value
Normal module (mm)	3.95
Number of teeth	23
Pitch Diameter (mm)	90.9
Base Diameter (mm)	82.3
Root Diameter (mm)	81.0
Outside Diameter (mm)	98.8
Normal Tooth Thickness (mm)	6.205
Start of active profile (mm)	85.9
Face width (mm)	19.5
Tight mesh center distance (mm)	90.9
Centre distance (mm)	91.4
Pressure Angle (deg.)	25

**Figure 4. The load and contact pressure distribution along the path of contact when input torque is at the maximum of 800 Nm****Table 3. The basic inputs for the design process and the model**

Input	Value
Engine	6.7 L (408 in ³) Cummins Diesel I6
Maximum power	350 (hp)/260 (kW)
Maximum torque	660 lbf. (900 Nm)
Design torque	1050 (Nm)
Gear roughness	0.32 (μ m)
Average lambda ratio	0.6
Ambient temperature of the oil bath	70 ($^{\circ}$ C)

Modelling of losses due to bearing and seal friction and churning

The remaining losses in a gearbox are those due to bearing and seal friction and the churning of the oil. Churning losses are the largest of these and they are calculated using the set of equations developed by Changenet and Vex [7]. Once the churning torque is calculated, the churning losses are predicted using the rotational speed of the gears.

The series of equations used to calculate the churning losses based on Changenet's method are shown below:

$$C_{ch} = 0.5 \rho \Omega^2 R_p^3 S_m C_m \quad (15)$$

$$C_m = 1.366 \left(\frac{h}{D_p} \right)^{0.45} \left(\frac{V_0}{D_p^3} \right)^{0.1} Fr^{-0.6} Re_c^{-0.21}, \quad Re_c < 6000 \quad (16)$$

$$C_m = 3.644 \left(\frac{h}{D_p} \right)^{0.1} \left(\frac{V_0}{D_p^3} \right)^{-0.35} Fr^{-0.88} \left(\frac{b}{D_p} \right)^{0.85}, \quad Re_c > 9000 \quad (17)$$

where

C_{ch} is churning torque, Nm;

ρ is density, kg/m³;

Ω is rotational speed, r/s;

R_p is gear pitch radius, m;

S_m is submerged surface area, m²;

C_m is dimensionless torque

D_p is gear pitch diameter, m;

h is gear immersion depth, m;

V_0 is oil volume, m³;

Fr is Froude number dependent on the gear parameters ($\Omega^2 R_p/g$)

Re_c is critical Reynolds number ($\Omega R_p b/\nu$)

A linear interpolation between the two formulae is used when $6000 < Re_c < 9000$

The calculation of the churning torque necessitates the definition of a gearbox casing, where the hypothetical gear pair will operate. This is essential, so that the oil level inside the gearbox and the oil volume that is being displaced by the gears can be calculated and taken into account. Therefore, an arbitrary gearbox casing was defined, based on proper clearances between the gears and the casing. Using Changenet [27] as a guide, the value of $2 m_n$ has been chosen for the radial clearance and $1.5 m_n$ for the axial clearance, where m_n is the normal module of the gear. Once the casing is defined, the gearbox churning losses can be modelled. Although adequate clearances were defined, it should be noted that in terms of size and shape the casing defined here differs from that normally used in the automotive applications. This is inevitable as the current gearbox consists of a single stage only, although the transmitted torque is representative of automotive applications.

Bearing and seal losses were calculated using the SKF bearing calculator tools [28] by considering the selected bearing type and then calculating the seal losses from the difference in losses between the plain and the sealed bearings of the same type. Bearing selection is an important design parameter in a gearbox. A common design procedure selects the bearings for each axle based on the maximum load of the most heavily loaded axle; keeping in mind that the various axles in a gearbox (two in the current simple single stage application) rotate at drastically different speeds, it is obvious that such an approach will lead to one of the bearings to be over-designed and that the system would be less efficient as a result. Therefore, in order to improve the efficiency of the system, ideally one should apply a procedure that considers each component of the drivetrain individually to ensure that while every component has the same operational life expectancy, there are no over-designed components. For simplicity, in this particular application a 1:1 ratio was chosen and therefore there is no need to design the bearings individually.

Thermal coupling of the gearbox

To account for ambient temperature rise and incorporate thermal coupling into the presented efficiency model, the gearbox system was modelled using a multi-physics finite element software tool. This FEM model calculates the conductive and convective heat transfers within the gearbox itself and between the gearbox and its surroundings. For the sake of simplicity, the gears were approximated by rotating discs. The discs were modelled in such a way so as to sweep through the oil sump as they would do in a real gearbox. The vehicle to which the gearbox is attached is represented by a lump mass while air is allowed to flow around the assembly at specified velocity causing forced convection and therefore simulating the moving vehicle. The required inputs for the FEM model are obtained from the numerical model presented above and include the total dissipated heat from the teeth as well as the friction-induced heat from the bearings. Figure 5 shows the modelled FEM geometry and an example output of temperatures within a gearbox.

Results and discussion

This section shows example results obtained through the application of the model described above. All results are for oil A, at the condition of 90% of maximum torque (800Nm). First the EHL conditions in the gear contacts at specified conditions are shown followed by magnitudes of individual losses in gear contacts, bearings and those due to churning. The total losses and the relative contribution of each loss source are shown. Finally, the model is used to compare the relative efficiency of the whole gearbox for the two selected oils.

Friction in gear teeth contacts

The predicted friction coefficients, minimum film thickness and the mean film temperatures along the path of contact for oil A and conditions specified in Table 2 are shown in Figure 6. Friction reaches a maximum value of 0.0593 at around 6 mm along the contact path and is minimized at the pitch point. It is also evident that the minimum film thickness remains almost constant throughout the contact. The temperature quickly rises as the slide roll ratio increases and is minimized at the pitch point as may be expected.

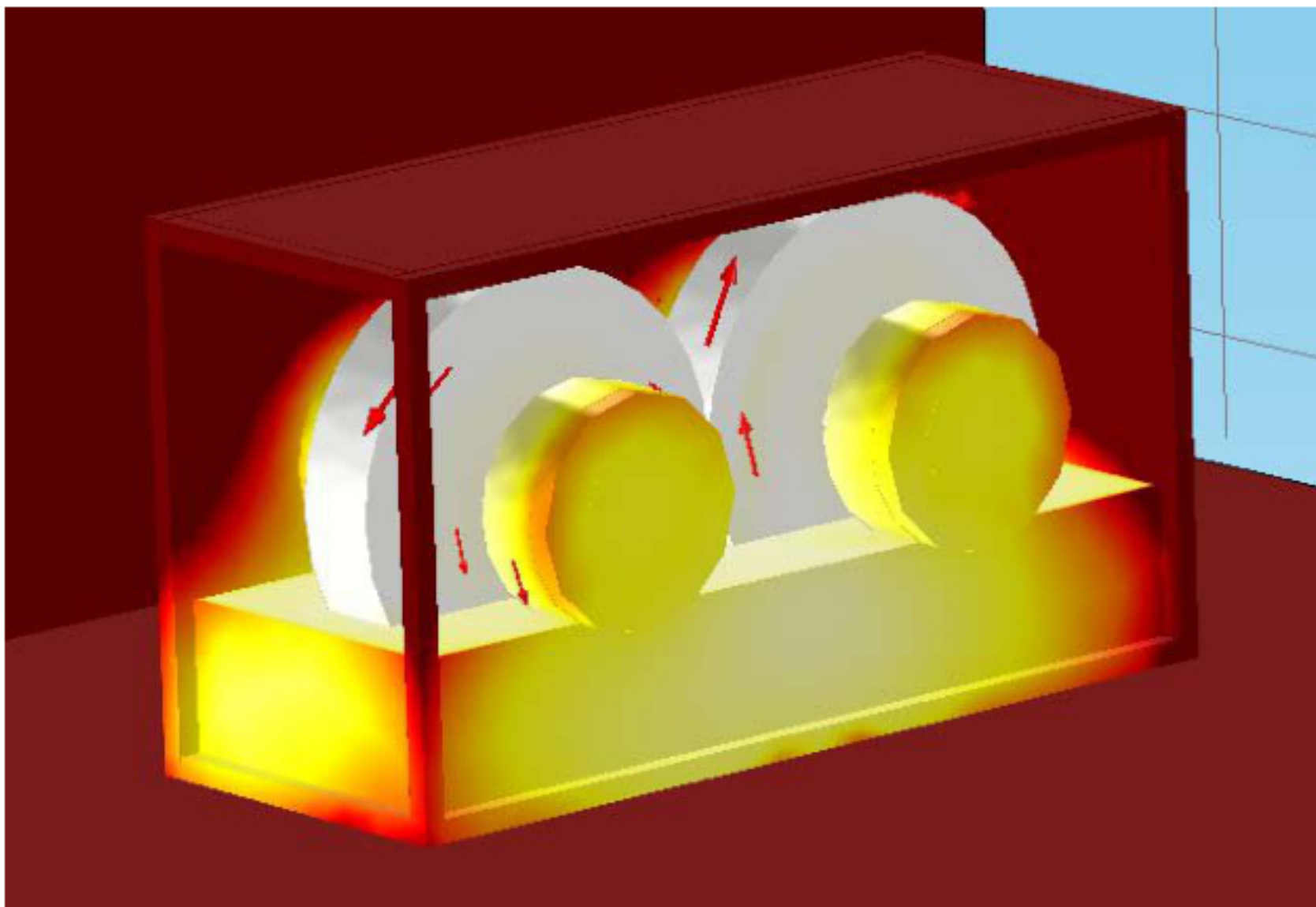


Figure 5. The multiphysics model of the simple gearbox.

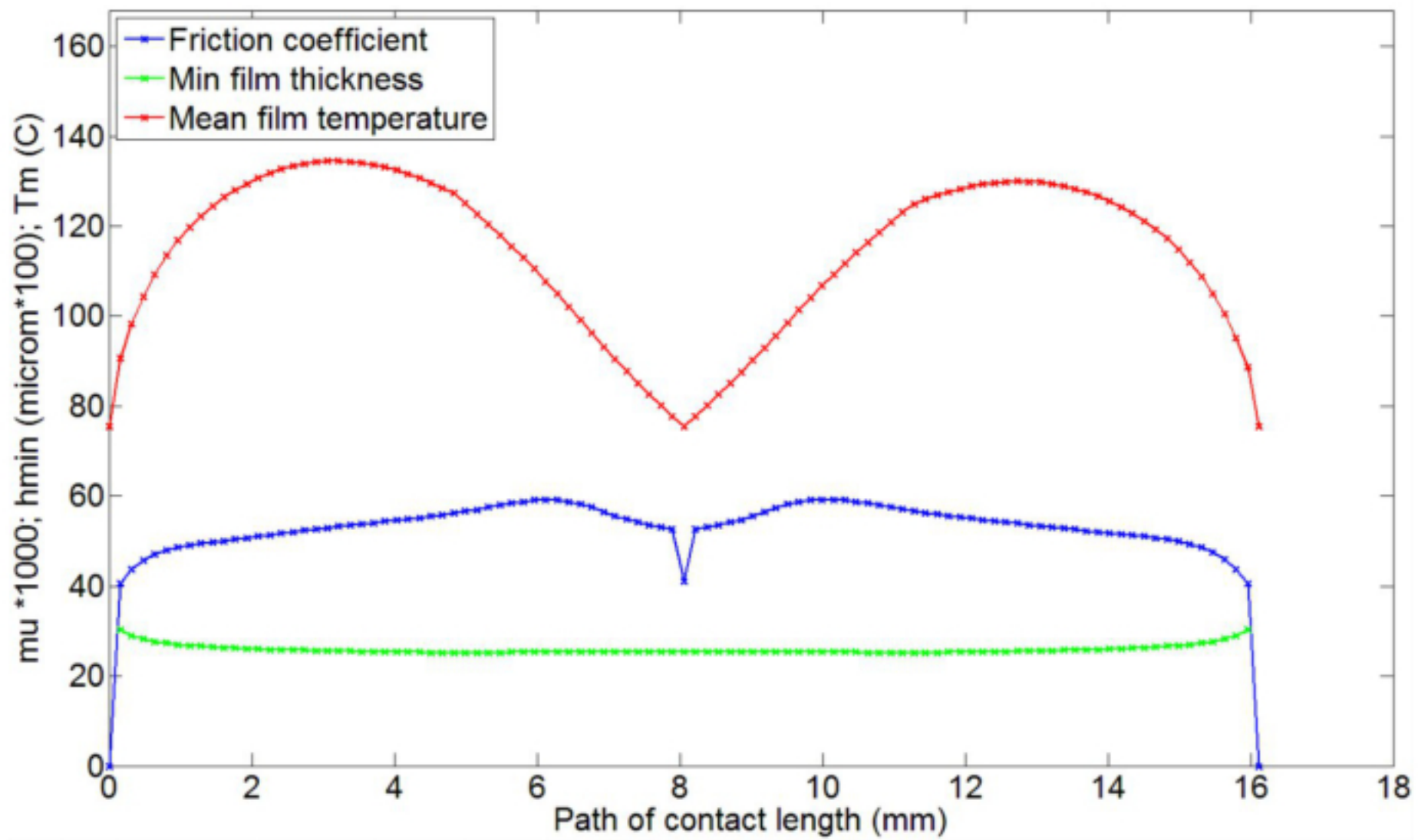


Figure 6. The coefficient of friction, minimum film thickness and mean film temperature along the path of contact at 1600 rpm

Efficiency

Since it is assumed that the losses of the gear pair are predominantly in the form of dissipated heat the calculation of the heat combined with the actual transmitted power can provide the efficiency of the gear pair in each step. Figure 7 shows example efficiency calculations along the tooth contact path for oil A at gearbox conditions listed in Table 2. As expected, the efficiency within the mesh cycle varies according to the slide roll ratio with the values in the overall range of 98 to 100%.

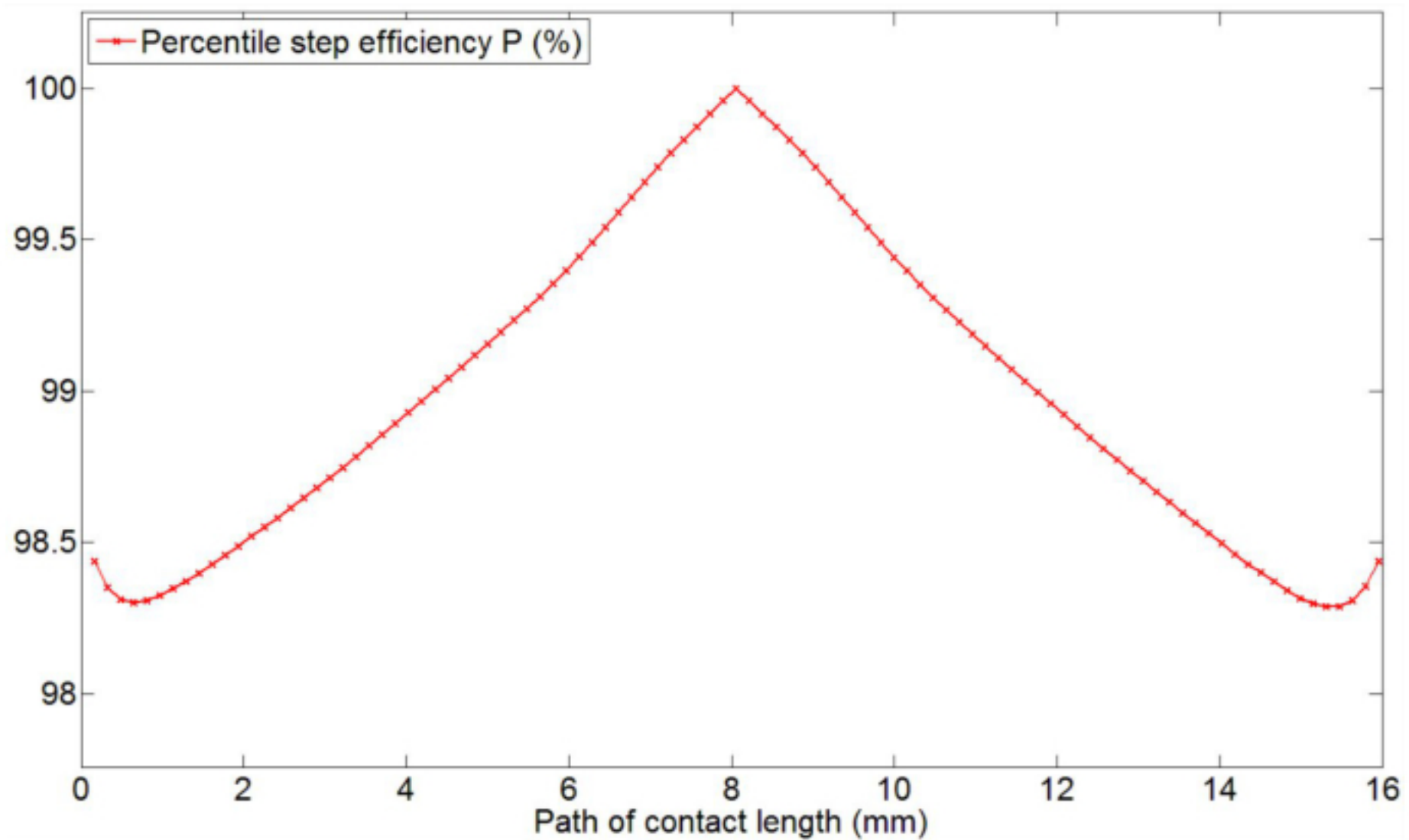


Figure 7. The efficiency of the gear pair along the path of contact at 1600 rpm

Churning

Figure 8 shows how the churning losses vary with the rotational speed and the immersion depth of the gear. As expected, the larger the portion of the gear that is submerged the higher the churning losses are. It is evident that increasing speed also increases churning losses. The rate of increase in losses with speed depends strongly on the immersion depth itself. In the example of Figure 8 it is clear that the speed has a much bigger influence on losses for the bigger immersion depth of $12\ m_n$.

The churning losses for this particular application, which can be seen in Figure 4.4 were calculated for $T = 70^\circ\text{C}$ and immersion depth equal to $6\ m_n$ which is relatively shallow. Since the most important parameter for churning losses is the volume of the gear that is submerged in the oil and the gears in this application are relatively small the churning losses are predictably only a small portion of the overall losses.

Bearings

The bearings that were selected for the application and the power loss that results from their rotation at maximum load and at the corresponding rotational speed are shown in Table 4. Note that single sealed bearings were chosen because these use the internal lubricant and are only sealed to prevent leakage to the outside of the shell.

Combined losses

The combined losses in the gearbox across the power range of the engine can be seen in Figure 9. The calculations are carried out through a set of conditions corresponding to actual spots on the torque-power curve of the Diesel engine of the reference vehicle at roughly 80-90% of accelerator pedal input.

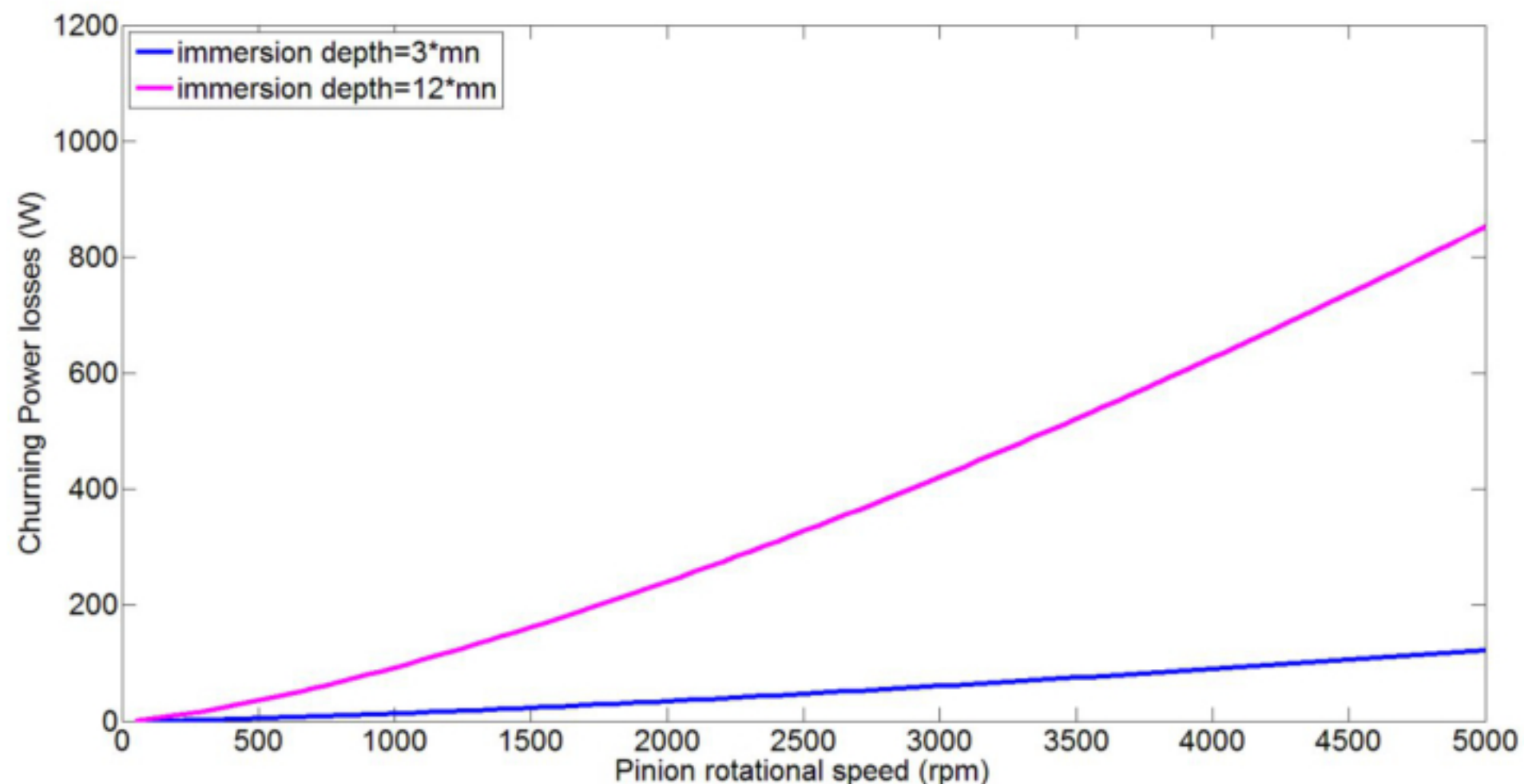


Figure 8. The churning losses of the gear pair relative to the immersion depth of the gear in a range of speeds typical of automotive applications

Table 4. The bearings of the single-stage gearbox and their respective power loss at 90% torque for oil A

Shaft	Designation	Power loss (W)
1 (1600 rpm)	6212	74×2
	6212 RS1 (sealed)	$99.3 (+ 25.3) \times 2$
2 (1600 rpm)	6212	74×2
	6212 RS1 (sealed)	$99.3 (+ 25.3) \times 2$
Total losses (bearings + seals)		397.2 (296 + 101.2)

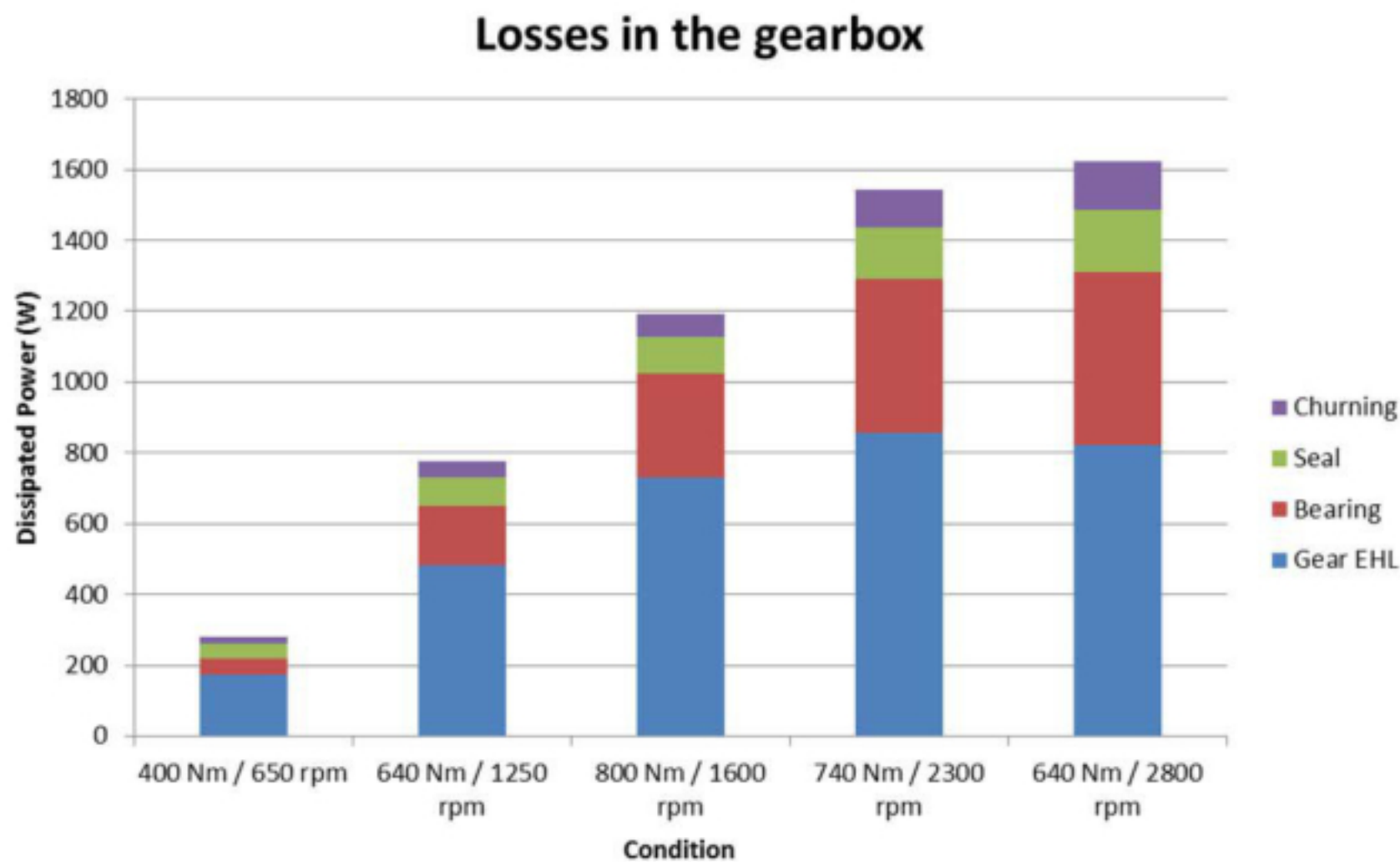


Figure 9. The individual and total losses of the modelled gear pair at 70 °C

It is evident that the EHL friction losses in the gear teeth contacts provide by far the largest contribution to overall losses. However, the relative magnitudes of each contribution differ for different operating speed/torque conditions. In particular the power loss in bearings and those due to churning only become significant at high rotational speeds, although they are still smaller than losses in gear teeth contacts.

The overall efficiency, including all additional losses for lubricant A under the specified conditions is shown in Table 5. The values are in the range of 99-99.15% and broadly agree with the results of the experimental study of Petry-Johnson¹⁴. Relative contributions of different loss sources shown here are also generally comparable to those shown in the same study. However, these previous results are not directly comparable to the current study because they were obtained on a temperature controlled, jet lubricated gearbox which is significantly different than the current dip lubricated application.

Lubricant comparison

The described approach is capable of comparing different lubricants in terms of their effect on the overall efficiency of the gearbox. To illustrate this, the efficiency was calculated for the two lubricants described in Table 1, whose rheology was extensively characterized following the method described earlier. The model was run in the range of 650 - 2800 rpm and torque settings of 400 to 800 Nm, corresponding to the particular automotive application chosen. Figure 10 shows predicted gear friction power losses for the two lubricants under these test conditions. It should be noted that thermal coupling was not used when calculating the losses in this particular case. The results clearly show that the predicted gearbox efficiency for the two lubricants varies by about 1 to 2% depending on the operating conditions.

These preliminary results are encouraging as they confirm that the devised methodology is able to differentiate between different lubricants in terms of their effect on transmission efficiency and that different lubricant rheology is indeed an important factor in the overall transmission efficiency. Therefore the model will now be further refined and used to provide a more extensive set of results.

Table 5. The overall efficiency across the power range for lubricant A

Condition	Efficiency (%)
400 Nm / 650 rpm	98.98
640 Nm / 1250 rpm	99.07
800 Nm / 1600 rpm	99.11
740 Nm / 2300 rpm	99.14
640 Nm / 2800 rpm	99.13

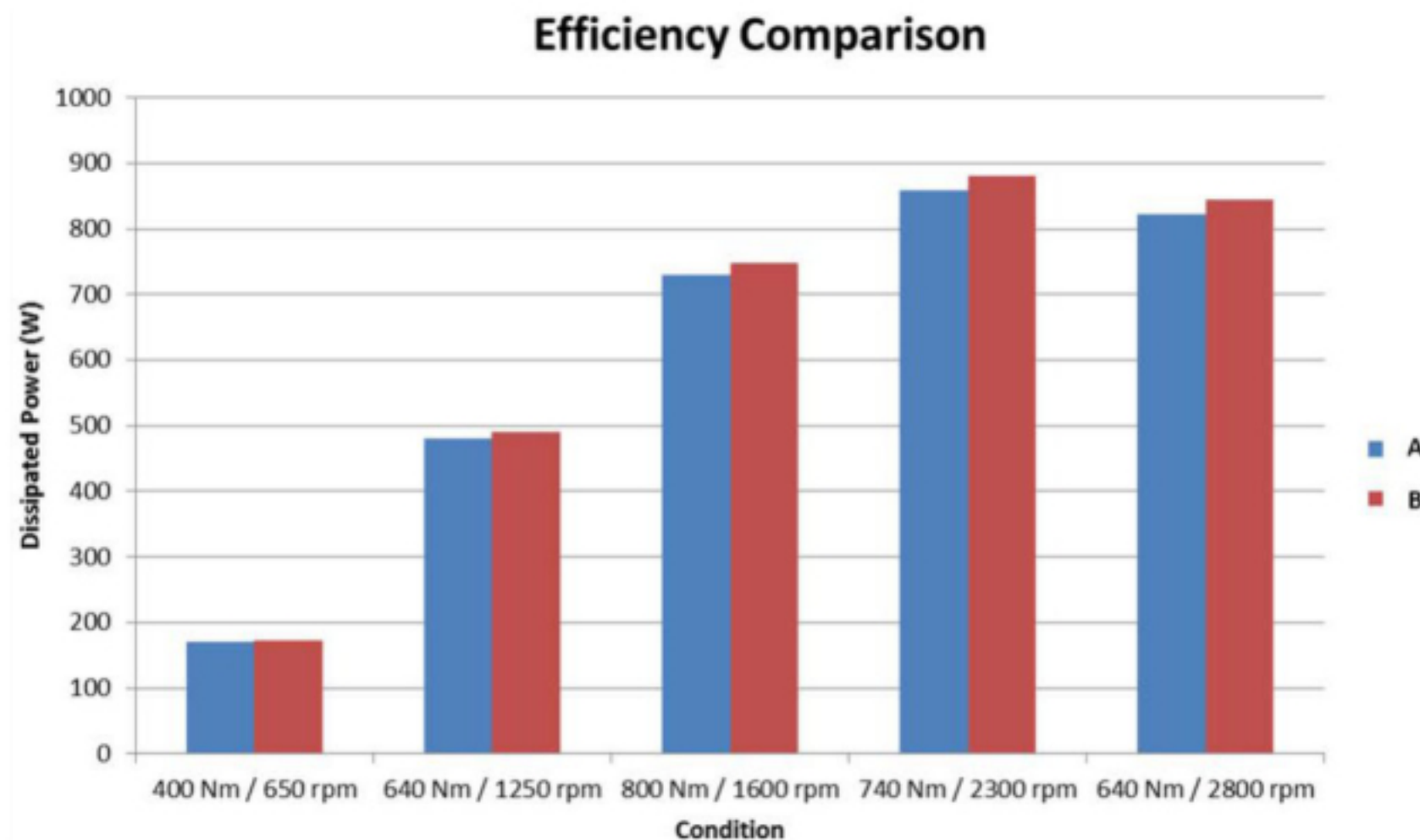


Figure 10. The total dissipated heat due to gear friction for lubricants A and B

Thermal coupling

The model described above is able to calculate friction using an EHL algorithm and can predict the ambient temperature rise resulting from the heat loss in the contact. However, if the model is not thermally coupled the accuracy of the method is suffering significantly because even though the contact temperatures are predicted using a feedback loop as illustrated earlier, the ambient temperature receives no thermal feedback from the code. However, the addition of the multiphysics simulation addresses this drawback by adding the missing link of feedback.

As soon as the temperature rise in the contact is calculated, it is being fed back at the inlet of the model; this loop continues until the model reaches a state of thermal equilibrium where the heat that is generated in the contact is being dissipated away, resulting in zero temperature rise. The model converges relatively quickly due to its simple layout. A comparison of the thermally coupled ambient temperature prediction and the non-thermally coupled prediction highlights the significance of the coupling as can be seen in Figure 11; the initial value of ambient temperature was 70° C in both cases and the simulated condition was that of 90% of maximum torque at 800 Nm and 1600 rpm.

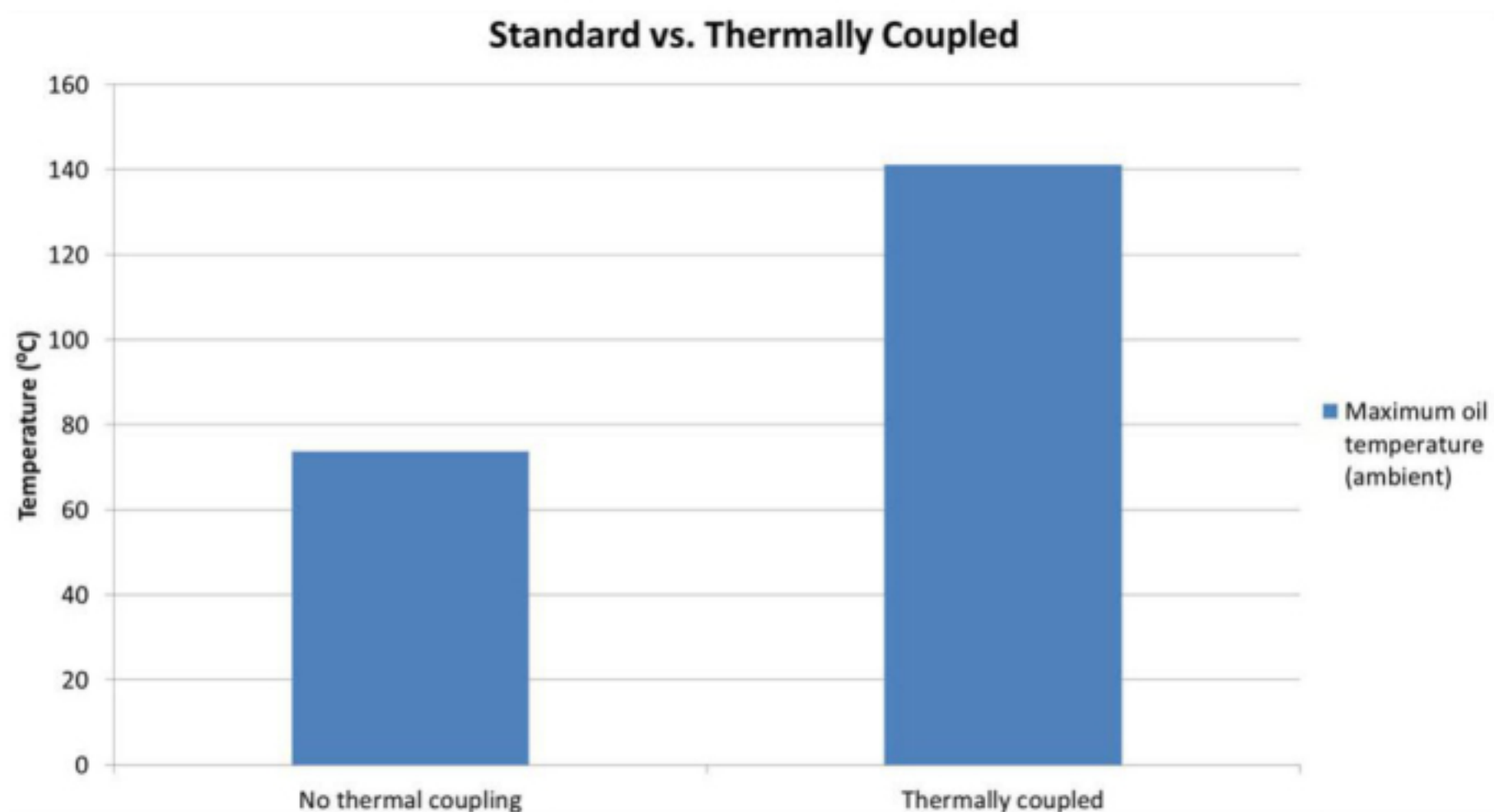


Figure 11. The effect of thermal coupling on the oil temperature prediction

The temperature predicted using the thermally coupled model is almost double compared to the non-thermally coupled. This is expected as the EHL code on its own is essentially an “in-contact” code used to calculate thermal gradients and temperature rise within the contact while it lacks any information about the thermal parameters outside the contact such as the heat capacity of the assembly, the outside air flow and the complex thermal network involved in the heat path. Past efficiency studies [14] omit the temperature rise of the oil sump and instead focus on efficiency and power loss keeping the temperature controlled at a specific value via thermocouples. However, this may not be the case in real life applications, where there is a constant heat exchange between the gearbox and the environment.

Conclusions

A simple thermally coupled one stage gearbox has been simulated using an EHL model to calculate the friction within the teeth contact and the resulting frictional losses. The present model was based upon viscoelastic rheology, according to the Eyring Maxwell theory and, crucially, uses well defined lubricant characteristics obtained from extensive experimental tests. The EHL traction is calculated using a fast, iterative method. The model calculates the traction, the temperatures and the film thickness within the contact as well as the ambient temperature rise. The model includes full thermal coupling by accounting for ambient temperature rise of the oil in the sump and the effect that this has on oil properties and consequently on gear teeth contact friction. The ambient temperatures were calculated through application of a multiphysics FEM simulation which considers all conductive and convective heat transfers in the gearbox. Bearing, seal and churning losses are also considered. Preliminary results show that the efficiency of the gearbox is strongly dependent on the lubricant rheology as well as on the operating conditions. Changes in the lubricant composition can therefore provoke changes in the relative fuel economy of a vehicle. The high temperatures that have been predicted even for relatively mild operating conditions highlight the need for accurate lubricant data at such high temperatures. The current model has been successfully used to highlight the differences between two lubricants with similar formulation but slightly differing rheology. Work is underway to elucidate the effects of lubricant properties on the overall gearbox efficiency.

References

1. Baglione, M., Duty, M., Pannone, G., *Vehicle system energy analysis methodology and tool for determining vehicle subsystem energy supply and demand*, SAE Technical Paper 2007-01-0398, 2007, SP-2072, <http://www.fueleconomy.gov/feg/atv.shtml>.
2. Anderson, N.E., and Loewenthal, S.H., *Spur-Gear Efficiency at Part and Full Load*, NASA Technical Paper 1622, AVRADCOM, 1980.
3. Anderson, N.E., and Loss, P., *Effect of Geometry and Operating Conditions on Spur Gear System*, no. 80, 1981.
4. Li, S., and Kahraman, A., *Prediction of Spur Gear Mechanical Power Losses Using a Transient Elastohydrodynamic Lubrication Model*, Tribology Transactions, 53:4, 554-563, 2010.
5. Chang, L., and Jeng, Y.R., *Modeling and Analysis of the Meshing Losses of Involute Spur Gears in High-Speed and High-Load Conditions*, Transactions of the ASME, Journal of Tribology, Vol 135, 2013.
6. Höhn, B.-R., Michaelis, K., and Hinterstoißer, M., *Optimization of Gearbox Efficiency*, Gear Research Centre FZG, Technische Universität München, Germany, 2009.
7. Changenet, C., and Vexlex, P., *A Model for the Prediction of Churning Losses in Geared Transmissions - Preliminary Results*, Journal of Mechanical Design, 129(1), pp. 128–133, 2007.
8. Changenet, C., and Vexlex, P., *Housing Influence on Churning Losses in Geared Transmissions*, Journal of Mechanical Design, Vol 130, pp. 128–133, 2008.
9. Long, H., Lord, A.A., Gethin, D.T., and Roylance, B.J., *Operating temperatures of oil-lubricated medium-speed gears: numerical models and experimental results*, Proceedings of the I MECH part G: Journal of Aerospace Engineering, 217 (2). pp. 87-106, 2003.
10. Taburdagitan, M., and Akkok, M., *Determination of surface temperature rise with thermo-elastic analysis of spur gears*, WEAR 261 pp. 656-665, Elsevier B.V., 2006.

11. Changenet, C., Oviedo-Marlot, X., and Velez, P., *Power Loss Predictions in Geared Transmissions Using Thermal Networks-Applications to a Six-Speed Manual Gearbox*, Journal of Mechanical Design, vol. 128, no. 3, p. 618, 2006.
12. Olver, A.V., *Testing transmission lubricants: the importance of thermal response*, Proceedings of the Institution of Mechanical Engineers, Part G, Journal of Aerospace Engineering, 1991, 206(G1), 35-44.
13. Olver, A.V., and Spikes, H.A., *Prediction of traction in elastohydrodynamic lubrication*, Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology 1998 212: 321.
14. Petry-Johnson, T., Kahraman, A., Anderson, N.E., and Chase, D., *An Experimental Investigation of Spur Gear Efficiency*, Journal of Mechanical Design, DOI: 10.1115/1.2898876, 2008.
15. Kolekar, A., Olver, A.V., Sworski, A.E., and Lockwood, F.E., *The efficiency of a hypoid axle - a thermally-coupled lubrication model*, Tribology International, Volume 59, March 2013, pp 203–209.
16. Chittenden, R.J., *A Theoretical Analysis of the Isothermal Elastohydrodynamic Lubrication of Concentrated Contacts, Direction of lubricant entrainment*, Proceedings of the Royal Society A: Mathematical, Physical and Engineering Sciences, 397, 271–294. 1985.
17. Evans, C.R., Johnson, K.L., *The rheological properties of elastohydrodynamic lubricants*, Proceedings of the Institution of Mechanical Engineers. C200, pp. 303-312, 1986.
18. Evans, C.R., Johnson, K.L., *Regimes of traction in elastohydrodynamic lubrication*, Proc. Institution of Mechanical Engineers, Part C, Journal of Mechanical Engineering Science, 1986, 200(C5), pp 313-324.
19. Muraki, et al., *Influence of temperature rise on non-Newtonian behavior of fluids in EHD conditions*, In Proceedings of the Fifth International Congress on tribology (Eds K. Holmberg and I. Nieminen), Espoo, Finland, 1989, Vol. 4, pp. 226-231.
20. Smeeth, M. Spikes, H.A., *The influence of slide roll ratio on the film thickness of an EHD contact operating within the mixed lubrication regime*, Presented at the Twenty-second Leeds-Lyon Symposium on Tribology, *The Third Body Concept*, Lyon, France, 5-8 September 1995.
21. Eyring, H., *Viscosity, Plasticity, and Diffusion as Examples of Absolute Reaction Rates*, The Journal of Chemical Physics 4, 236.
22. Barus, C., *Isothermals, Isopiestic and Isometrics in Relation to Viscosity*, American Journal of Science, 3rd Ser. 45, pp. 87-96, (1893).
23. Roelands C.J., *Correlational aspects of the viscosity-temperature-pressure relationship of lubricating oils*, University of Delft, 1966.
24. LaFountain, A.R., Johnston, G.J., and Spikes H.A., *The Elastohydrodynamic Traction of Synthetic Base Oil Blends*, Tribology Transactions, 44, 648-656 (2001).
25. Hoffman, Dieter A., British Gear Association Teaching Pack, Module 4.
26. Hoffman, Dieter A., British Gear Association Teaching Pack, Module 3.
27. Changenet, C., and Velez, P., *Housing Influence on Churning Losses in Geared Transmissions*, Transactions of the ASME, Journal of Mechanical Design, Vol. 130, 062603-1, 2008.
28. <http://www.skf.com/group/knowledge-centre/engineering-tools/skfbearingcalculator.html>