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Investigation of Friction Power Losses in Automotive Journal Bearings

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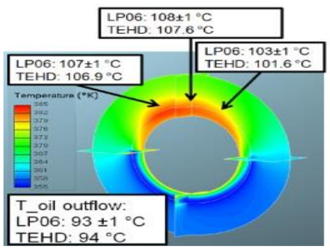
Graphical abstract



Abstract

Journal bearings are used in a large number of critical positions in automotive internal combustion engines (ICE) and contribute a major contribution to the total friction power losses in these engines. These reasons motivate in addition the accurate and reliable simulation of the operating conditions and friction power losses in journal bearings. In this work the lubrication of journal bearings is investigated in detail using detailed rheological lubricant models that include the piezoviscous effect. To describe mixed lubrication realistically in the simulation, a sophisticated contact model is employed together with measured surface roughness data from journal bearings. Starting point is an extensive thermo-elastohydrodynamic (TEHD) simulation, which yields important insights into the thermodynamical behavior of the lubricant film in journal bearings. From these results, a powerful isothermal elastohydrodynamic (EHD) simulation model is derived that calculates the oil temperature for the simulation from two easily accessible experimental temperatures. The capabilities of the presented simulation methods are compared to extensive experimental measurements performed on a journal bearing test-rig, which show excellent agreement.

Keywords: Friction; engines; simulation; testing; validation; journal bearings

1.0 INTRODUCTION

Friction in internal combustion engines (ICE) is a steady topic within the automotive industry, be it to lower emissions and fulfill demands by legislation or to stay longer on the track with a single filling of the fuel tank as in motor sport applications. Friction can also affect component lifetime and causes maintenance costs. All of these reasons add also an economical relevance to the scientific interest for a better understanding of the processes behind friction in ICEs.

However, before any efficient measures to reduce friction in engines can take place, the main friction sources need to be known. At the Virtual Vehicle Research Center, we use our friction test-rig (Figure 1) to investigate the sources of friction for a typical four cylinder gasoline engine; exemplary results for this engine are shown in Figure 2. The chart confirms the commonly regarded main sources of friction: the piston-liner contact is the cause for about 60% of the total mechanical losses, while the journal bearings in the crank train (main and big end bearings) contribute together about 25%. Finally, the valve train represents the third main source of friction and typically causes losses that are about half of the power losses in the journal bearings, or, more precisely, 15% of the total power losses shown in Figure 2. These results differ significantly from the results for special single cylinder research engines [6], which show the importance of measuring the actual engine to be investigated. Summarizing, the results indicate that a considerable amount of friction is caused by the journal bearings.

In addition, innovative engine concepts such as downsizing and ever increasing peak cylinder pressures lead to rising operational demands of the crankshaft main and big end journal bearing. At the same time low viscosity lubricants are used to reduce the friction losses in the engine and thereby also the specific fuel consumption of the engine. Low viscosity lubricants, however, reduce the load carrying capacity of journal bearings. Despite of all these measures, bearing reliability must not be affected through occurring mixed lubrication. Therefore, a powerful simulation model is needed that accurately describes the lubrication conditions in highly stressed journal bearings.

2.0 LUBRICANT RHEOLOGY

Modern hydrocarbon based lubricants generally show a complex rheological behavior with the dynamic viscosity strongly depending on temperature and pressure [4,7]. Multi-grade lubricants can also show strong Non-Newtonian behavior [7]. To be able to simulate accurately the tribological working conditions in the journal bearings, these lubricant properties need to be taken into account [4-5]. In this work we focus on mono-grade lubricants that do not exhibit the strong shear rate dependency of multi-grade lubricants. Therefore, only the temperature and pressure dependence of the lubricant viscosity needs to be considered [2-3]. These data are shown for the investigated lubricants SAE20 and SAE40 in Figure 3 and reveal as expected a strong dependence on both pressure and temperature. Specifically, already a hydrodynamic pressure of about 60 MPa doubles the lubricant viscosity.

Figure 1 Picture of the friction test-rig FRIDA at the Virtual Vehicle Research Center with engine under test

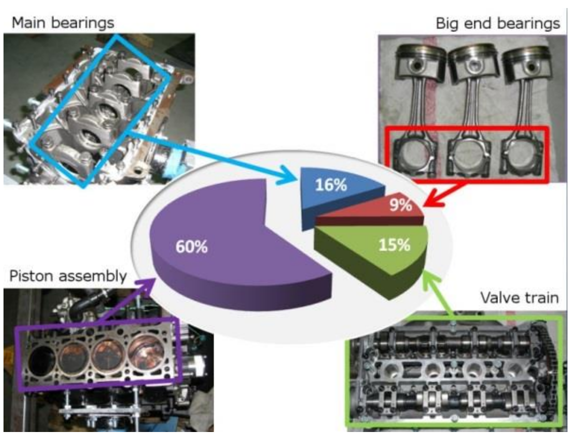


Figure 2 Exemplary results obtained from the friction test-rig for an inline four cylinder gasoline engine (see also [1])

3.0 MIXED LUBRICATION

To be able to describe mixed lubrication, a contact model is employed to describe metal-metal contact. In the following the Greenwood and Tripp approach [9] is used together with roughness data that were obtained from profilometer roughness measurements of the tested journal bearings (see [3] for a detailed discussion). The measured surface roughness is exemplary shown for one of the tested journal bearings in Figure 4. For the calculation of the Greenwood/Tripp parameters a 1D-profilometer trace was used that was performed on a run-in part of the bearing shell along the axial direction. From this measured data, an asperity roughness of 0.45 µm and a mean summit height of 0.38 µm was calculated. However, in the calculations a mean summit height of 0.4 µm was actually used for the bearing shell. For the journal an asperity roughness of 0.2 µm and a mean summit height 0.1 µm were used due to its smooth surface finish. The latter is also confirmed by surface roughness measurements of the shaft that show an Rpk~Ra~0.14 µm.

Modern engine oils include friction modifying additives like zinc dialkyl dithiophosphate (ZDTP) or Molybdenum based compounds to lower friction and wear in case metal-metal contact occurs [8]. For the Greenwood and Tripp contact model [9] we employed, therefore, in the following a boundary friction coefficient of µBound = 0.02.

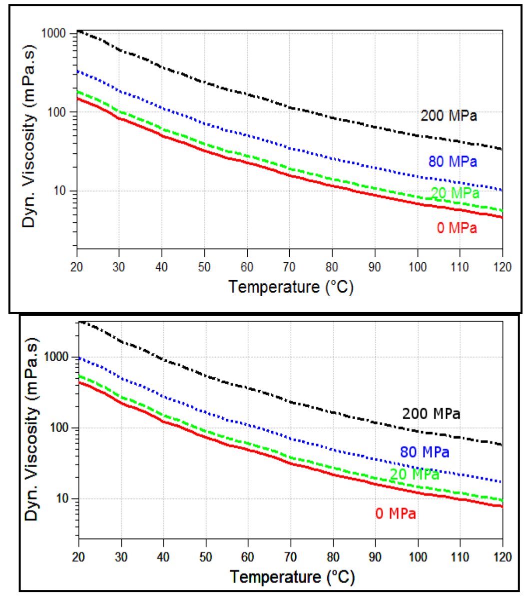


Figure 3 Dynamic lubricant viscosity of the used SAE20 (top) and SAE40 (bottom) lubricant for different temperatures and hydrodynamic pressures. As can be seen, a temperature increase of about 15°C halves the lubricant viscosity. It can also be seen that the SAE20 lubricant has close to half of the SAE40 viscosity only above 70°C. Below this temperature the SAE20 is considerably thinner and shows a viscosity considerably less than half the SAE40 viscosity

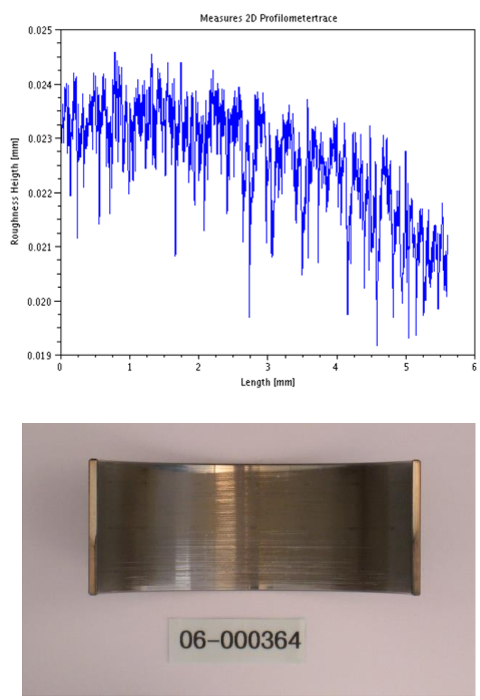


Figure 4 (Top): Measured surface roughness of one of the tested journal bearings. (Bottom): one of the journal bearings after running the test with the SAE40 lubricant

4.0 SIMULATION AND EXPERIMENTAL VALIDATION

The Reynolds equation is the central differential equation to describe the lubricant flow and calculate the properties of the lubricating film separating the elastic bodies. In its basic form the Reynolds equation treats the lubricating film as isothermal and does not account for locally different temperatures (elastohydrodynamic simulation, EHD). Combining the Reynolds equation with the energy equation together with suitable thermal boundary conditions yields the thermoelastohydrodynamic approach. TEHD is considerably more complex due to the involved thermal boundary conditions and is, therefore, also numerically much more demanding. Of course a considerable number of previous works exist in literature concerning these methods and journal bearings in general. An extensive list is not reproduced here and the authors would like to direct the reader to the literature cited in the previous works [1-4].

In the following, only results are discussed as the mentioned methods together with lubricant data, thermal boundary conditions, contact model parameters and surface contours are published in detail in [1-4].

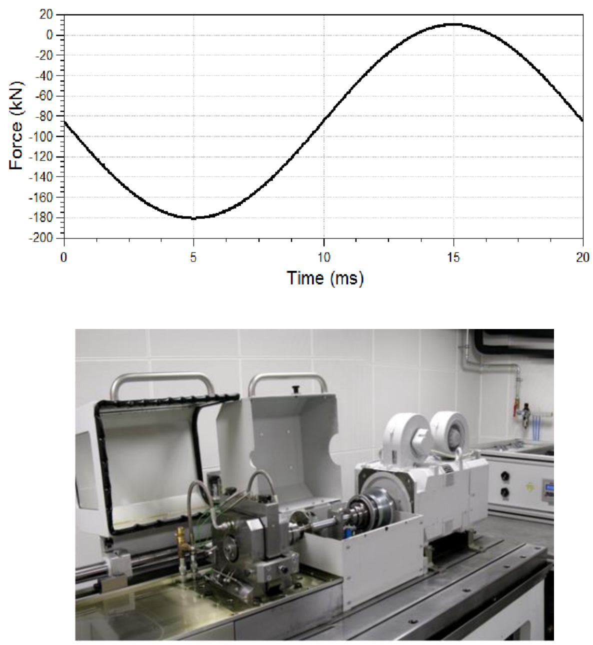


Figure 5 (Top): Plot of the dynamic load cycle with a peak load of 180 kN discussed in this work. The picture on the bottom shows the journal bearing test-rig of MIBA[1] that was used to obtain the experimental data; it consists of three journal bearings: the actual test bearing on which the shown dynamic load is applied and the two support bearings that carry the journal (see also [1-4])

Following the aim of this work to directly compare the results from simulation to experimental measurements from the journal bearing test-rig LP06 (see Figure 5), consequently, journal bearings with a diameter of 76 mm and a width of 34 mm are considered. A dynamic (sinusoidal) load with a peak load of 180 kN is applied to the test bearing with a fixed frequency of 100 Hz.

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5.0 INSIGHTS FROM TEHD

With carefully chosen thermal boundary conditions on the ground of physical arguments ([2] and references therein, in particular [10]), it is indeed possible to accurately simulate the different local temperatures in journal bearings. Figure 6 shows an exemplary result from TEHD simulation in direct comparison to temperatures measured on the journal bearing test-rig LP06. It is notable that not only the peak temperature in the high load area of the journal bearing, but also the oil outflow temperature is accurately calculated [2]. This is of great relevance as the oil outflow temperature is directly related to the friction power losses in the bearing. Therefore, this agreement is a further indication that the simulation accurately describes the lubricating film in the journal bearing.

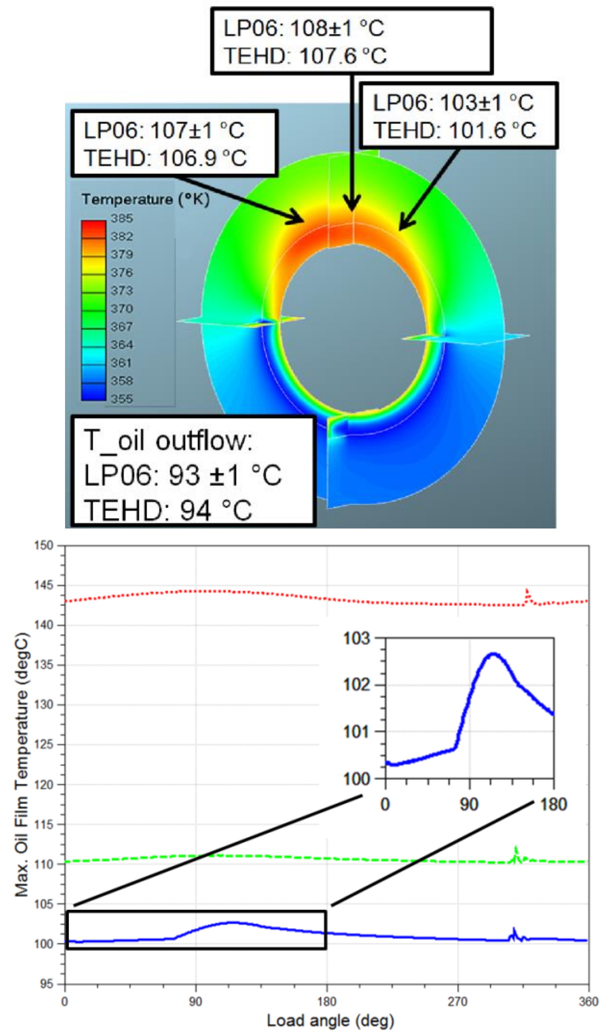

Figure 6 (Top): Plot of the computed and measured local temperatures of the oil film and of the surrounding structure (separated by a thin white line) of the journal bearing for a rotational speed of 2000rpm, lubricated with SAE40 and a dynamical load with a peak force of 180kN; LP06 denotes the temperatures measured on the journal bearing test-rig LP06 as shown in Figure 5. TExp of Eq. (1) corresponds to 108°C and Toil supply to 83.2°C in this case, which yields an equivalent temperature TEHD of 101.8°C for the EHD-simulation. (Bottom): calculated peak temperatures over an entire load cycle for three different journal bearing operating conditions, shown as green dashed line for a journal speed of 2000 rpm and lubricated with SAE40, as red dotted line with SAE40 and a journal speed of 4500 rpm and, finally, shown as blue solid line for a journal speed of 2000 rpm and lubricated with SAE20

As one can see from Figure 6, a significant temperature gradient is present in the oil film ranging from 82°C from the incoming lubricant (cold parts shown in blue color representing the oil supply groove) to 108°C of the oil film in the high load zone.

However, it is a crucial finding that the local temperatures change only very weakly during the entire load cycle. This property allows to approximate the local temperatures as being essentially constant over time. This thermal behavior can be seen in Figure 7, where the peak temperatures in the journal bearing are shown over the entire load cycle (as shown in Figure 5) for different operating conditions. The peak temperature in the high load area of the bearing changes only a few single degrees Celsius during the load cycle. This stable thermal behavior can be used in the following to derive a considerably simpler isothermal EHD-simulation model that still describes accurately the friction power losses including metal-metal contact for a large range of operating conditions from full film lubrication up to weak mixed lubrication.

From the TEHD-results following simple relation for an equivalent temperature can be derived [2] that allows to include this temperature gradient in the EHD-simulation,

$$T^{EHD} = T^{Exp} - \frac{T^{Exp} - T_{\text{oilsupply}}}{4} \tag{1}$$

Where T^{Exp} denotes the measured bearing shell back temperature in the high load area of the journal bearing and Toil supply is the temperature of the cool oil supplied to the journal bearing. This simple formula yields a suitable oil film temperature to be used in the isothermal EHD-simulation of the journal bearing. It is conveniently calculated from only two measured temperatures of the test-rig. As can be seen from the equation, the dominating temperature is the temperature in the high load area of the journal bearing which is corrected by a small amount to consider the cool lubricant in the oil supply groove with its considerably larger viscosity. It is important to note that above relation is applicable to journal bearings with a 180° oil supply groove only. Such journal bearings are commonly used as e.g. main bearings in ICEs to carry the crank shaft. Journal bearings without a 180° groove - like they are commonly used in big end bearings in ICEs - show generally very consistent temperatures across the journal bearing [11] and do not need, therefore, such a correction.

6.0 AN EQUIVALENT ISOTHERMAL EHD-SIMULATION

In a next step, the oil film is simulated by the EHD approach using the basic, isothermal form of the Reynolds equation [2] employing the same contact model. For the isothermal oil temperature Equation (1) is used to calculate an equivalent lubricant temperature. From Figure 6 the required temperatures can be obtained, in particular TExp of Equation (1) corresponds to 108°C and Toil supply to 83.2°C in this case, which yields an equivalent temperature TEHD of 101.8°C used in the EHD-simulation.

In the following the average torque needed to motor the three journal bearings (the test bearing and two support bearings) at a given journal speed is used to compare the results from simulation to the measured data; this so called friction moment is averaged over the entire load cycle (as shown in Figure 5) due to experimental limitations.

Figure 7 and Figure 8 show the comparison of the calculated friction moment with the measured friction moment on the journal bearing test-rig LP06 for two different journal speeds and two different lubricants. From Figure 9 and Figure 10 it can be seen that the SAE40 case with 4500 rpm journal speed is purely full film lubricated, while for the SAE20 case weak mixed lubrication occurs. The SAE40 case with 2000 rpm journal speed lies right between these two results and shows within EHD simulation less than 1 W (rounded to 0 W) and within TEHD simulation about 1 W power loss due to metal-metal contact.

The results demonstrate clearly that the simulations are not only able to accurately calculate the friction power losses in journal bearings, but also that an EHD-simulation with the discussed equivalent temperature is a valid and very accurate approximation for a full TEHD-simulation at least for full film and weak mixed lubrication.

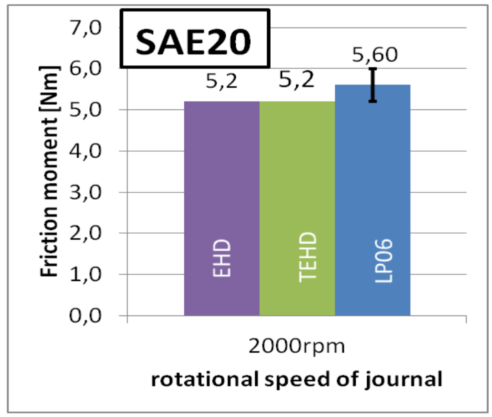

Figure 7 Plot of the friction moment averaged over the entire load cycle for a dynamic peak load of 180kN and lubricated with SAE20

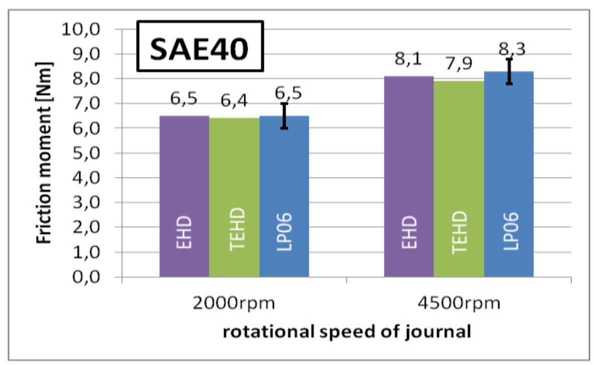

Figure 8 Plot of the friction moment averaged over the entire load cycle for a dynamic peak load of 180kN and lubricated with a SAE40 lubricant. LP06 denotes the value measured on the friction test-rig together with the measurement uncertainty shown as bars. TEHD and EHD denote the values obtained from simulation by either the TEHD approach or the EHD approach with the temperature calculated from Equation (1), respectively

As SAE20 oil has about half the viscosity of SAE40 (see Figure 3), the operating conditions in the journal bearings are distinctly different. The difference in viscosity affects the load carrying capacity of the journal bearing which is consequently reduced for lubrication with SAE20 and weak mixed lubrication occurs. This can be seen in Figure 9 where the friction power losses due to hydrodynamic lubrication and due to metal-metal contact are shown separately for the test bearing. As only during a short time of the load cycle metal-metal contact occurs, the load cycle averaged contribution of metal-metal contact to the total friction power losses appears rather low with 5 W despite that significant amounts of metal-metal contact occur for a short time of the load cycle.

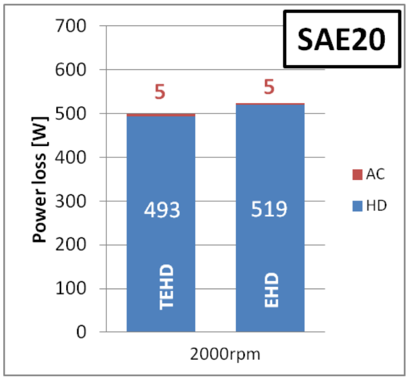

Figure 9 The contributions of hydrodynamic lubrication (denoted as HD) and of metal-metal contact (denoted as AC) to the total friction power losses in the test journal bearing for a journal speed of 2000 rpm and lubrication with SAE20

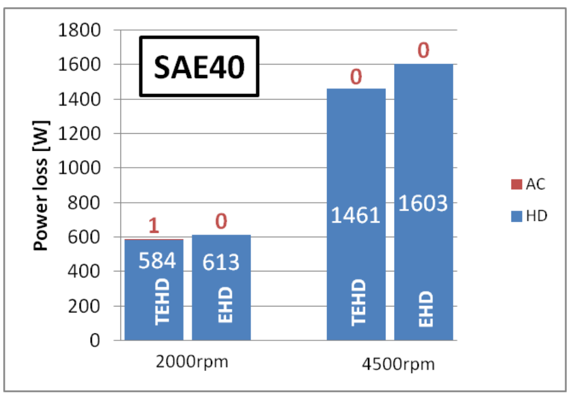

Figure 10 The contributions of hydrodynamic lubrication (denoted as HD) and of metal-metal contact (denoted as AC) to the total friction power losses in the test journal bearing for a journal speed of 2000 rpm/4500 rpm and lubrication with SAE40. TEHD and EHD denote the values obtained from simulation using the TEHD approach and the EHD approach using the equivalent temperature from Equation (1), respectively

■7.0 SUMMARY

With the aid of detailed experimental data it is shown that a TEHD-simulation model with suitable thermal boundary conditions is able to accurately predict not only local temperatures, but also the oil outflow temperature and the average friction power losses in the oil film and in the journal bearing. From these results it is found that the thermal behavior of the oil film is very stable and evens the peak temperature in the journal bearing changes only by very few degrees Celsius during the load cycle despite the very high and dynamic load. From these insights, a relation for a global oil film temperature to be used for the isothermal EHD simulation of journal bearings with a 180° oil supply groove is presented. In direct comparison to experimental data this approach shows excellent agreement and the validity of the approach for full film lubrication and for weak mixed lubrication is confirmed. For operating conditions with further increased amounts of mixed lubrication, however, it is expected that local heating due to metal-metal contact will become significant and could make a full TEHD-simulation necessary. For the investigated operating conditions with only small amounts of mixed lubrication, the presented approximation by using an EHD-approach with the equivalent temperature is indeed very accurate and allows studying the tribological conditions in journal bearings.

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References

[1] Allmaier, H., Priestner, C., Sander, D. E., Reich, F. M. 2013. Friction in Automotive Engines, Book Chapter in Book Tribology in Engineering, Intech. ISBN 978-953-51-1126-9.

[2] Allmaier, H., Priestner, C., Reich, F. M., Priebsch, H. H. 2013. Novotny-Farkas, F.. Predicting Friction Reliably and Accurately in Journal Bearings-Extending the EHD Simulation Model to TEHD. *Tribology International*. 58: 20–28.

[3] Allmaier, H., Priestner, C., Six, C., Priebsch, H. H., Forstner, C., Novotny-Farkas, F. 2011. Predicting Friction Reliably and Accurately in Journal Bearings-A Systematic Validation of Simulation Results with Experimental Measurements. *Tribology International*. 44: 1151–1160.

[4] Allmaier, H., Priestner, C., Reich, F.M., Priebsch, H. H., Forstner, C., Novotny-Farkas, F. 2012. Predicting Friction Reliably and Accurately in Journal Bearings-The Importance of Extensive Oil-Models. *Tribology International.* 48: 93–101.

[5] Kasolang, S., Ahmad, M. A., Dwyer Joyce, R., Tai, C. F. M. 2012. Preliminary Study of Pressure Profile in Hydrodynamic Lubrication Journal Bearing. *Procedia Engineering*. 41: 1743–1749.

[6] Mufti, R. A. and Priest, M. 2009. Technique of Simultaneous Synchronized Evaluation of the Tribological Components of an Engine Under Realistic Conditions, Proceedings of the Institution of Mechanical Engineers. *Part D: Journal of Automobile Engineering*. 223: 1311–1325.

[7] Bair, S., Jarzynski, J. and Winer, W. O. 2001. The Temperature, Pressure and Time Dependence of Lubricant Viscosity. *Tribology International*. 34: 461–468.

[8] Morina, A. and Neville, A. 2007. Understanding the Composition and Low Friction Tribofilm Formation/Removal in Boundary Lubrication. *Tribology International*. 40: 1696–1704.

[9] Greenwood, J. A. and Tripp, J. H. 1970. The Contact of Two Nominally Flat Rough Surfaces, Proceedings of the Institution of Mechanical Engineers, June 1970. 185(1): 625–633.

[10] Wang, Y., Zhang, C., Wang, Q., Lin, C. 2002. A mixed-TEHD Analysis and Experiment of Journal Bearings Under Severe Operating Conditions. *Tribology International*. 35(6): 395–407.

[11] Suzuki, S., Ozasa, T., Yamamoto, M., Nozawa, Y., Noda, T., O-Hori, M. 1996. Temperature Distribution and Lubrication Characteristics of Connecting Rod Big End Bearings. *SAE Transactions*. 104: 2025–3.