

# EHL friction mapping - the influence of lubricant, roughness, speed and slide to roll ratio

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## Abstract

A friction test is conducted in a WAM ball on disc test rig. The output from the test is friction coefficient versus entrainment speed and slide-to-roll ratio presented as a 3D friction map. A number of parameters are varied while studying the friction coefficient; surface roughness, base oil viscosity and EP additive package. Entrainment speed, slide to roll ratio and oil temperature are also varied. The results show that the mapping is efficient in showing the different types of friction that may occur in an EHL contact. The results also show that the friction behaviour can be strongly influenced by changing surface roughness as well as base oil viscosity, EP additive content and operating temperature.

*Keywords:* EHL, roughness, friction

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## 1. Introduction

Reducing losses in transmissions has become a priority in the automotive market during the latest years, mainly due to environmental aspects leading to regulations on the automotive industry to drive the development of cars with lower fuel consumption. Rising fuel prices and increasing environmental concern also makes customers more prone to purchase more fuel efficient vehicles.

In addition to the fuel savings that could be done by increased efficiency of transmissions there are other benefits as well. A more efficient transmission will in general generate less heat, and experience less wear. This will lead to fewer failures, longer lifetime of components, and possibly longer service intervals. Furthermore this implies a possibility to reduce coolant components, thus reducing the total weight of the system, leading to further decrease in consumption and a lower impact on nature due to a reduction of material usage. A low weight design is also beneficial for vehicle dynamics and handling.

In some cases a substantial part of the losses in an automotive transmission is attributed to gear contact friction due to sliding and rolling between the gear teeth. The total transmission losses due to gear contact are ranging between 4.5 and 50 percent [1–3]. Generally gearboxes running at low speeds and high loads have a substantial part of gear contact losses, whereas high speed applications usually dominates by churning losses. Since churning losses mostly are related to the oil viscosity, the best system performance ought to be obtained by optimizing the gear contact to work with as low viscosity as possible to minimize churning losses while still keeping a low wear of the system.

Much research has been conducted on the topics of gear contacts covering contact geometry, materials, surface topography, coatings as well as lubricant properties and rheological effects. Several authors have presented papers describing and modeling the gear contact behaviour. Already 1966 Dowson and Higginson [4] used their EHL theory to predict the film thickness between two gear teeth at the pitch point. This work was extended by Gu [5] to include the whole line of action in an involute

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gear mesh. This study also included an approximate thermal model. The complexity of the models has increased over the years to consider more parameters and gives more accurate results. Recently Akbarzadeh and Khonsari [6] presented a model for calculating the friction in spur gears considering shear thinning and surface roughness. Their model uses the scaling factors of Johnson *et al.* [7], to predict friction from the hydrodynamic respectively asperity contact part, and Greenwood and Tripp [8] formula for contacts between two rough surfaces. They later extended their model to incorporate thermal effects in the analysis [9].

Parallel with the theoretical research, experiments have been carried out for different purposes. One is of course for validation of mathematical models, but it is also a way to test how altering different factors, such as lubricant properties and surface roughness influences for instance the efficiency of the system. Experiments are conducted in various tests rigs, for instance the FZG back-to-back, as well as in twin disc, and ball on disc configurations. All of these has their own benefits and disadvantages which must be considered together with the purpose of the actual experiment. Experiments with real gears are of course closest to the real application, and in some cases the actual gear box is mounted in a test rig and the experiments are performed. This approach gives very good indications on how the real system performance is influenced by changing certain parameters, and standardised methods like the FZG should make it easy to replicate results [10–14]. However, testing with real gears is generally the most expensive way of testing, and it is also hard to make any detailed conclusions since there are many components influencing the results and the output generally is an average friction value even though contact load, radii, entrainment speed and slide to roll ratio (SRR) are continuously changing along the line of action.

Many authors have used twin-disc test devices to simulate power loss and wear behaviour of gear contacts [10, 14–16]. The benefits compared to gear testing are lower costs, and the possibility to in detail study parameters like friction coefficient at specific entrainment speeds and slide to roll ratios. Furthermore it is possible to simulate the gear contact without other friction losses present in gears (like churn-

ing in dip lubricated gears and bearing losses).

The ball on disc configuration shares the benefits of the twin disc, and is also easier to control since there are not the same alignment issues. In the present study a WAM ball on disc configuration is used to study the friction behaviour in various entrainment speeds and slide to roll ratios. Additional parameters studied includes: surface roughness, base oil type, base oil viscosity, oil temperature and additive packages. The output from the test is friction coefficient versus entrainment speed and slide-to-roll ratio presented as a 3D friction map. Ball on disc friction experiments have earlier been carried out to investigate EHL film formation and friction behaviour during rolling and sliding [17, 18].

## 2. Method

The following sections cover a description of the ball on disc test rig, the test specimens and lubricants, and an overview of the test procedure.

### 2.1. Ball on disc tribotester

The experiments are conducted in a Wedeven Associates Machine (WAM) ball on disc test device, model 11, where the contact is shown in detail in Fig. 1. WAM 11 utilizes advanced positioning technology for high precision testing under incipient sliding conditions. The ball and the disc are driven by separate electric motors, the former to a speed up to 25000 rpm and the latter up to 12000 rpm. Each motor is adjustable on-line to change entrainment speed and slide to roll ratio. The standard ball specimen has a diameter of 20.637 mm and the disc has a diameter of 101 mm. With standard sized test specimens an entrainment speed of up to 27 m/s is possible under pure rolling conditions, and the maximum load of 1000 N which gives a maximum circular Hertzian contact pressure of 2.91 GPa.

The test device contains a built in cooling and heating system allowing for lubricant test temperatures between 5 and 100 degrees Celsius. A closed loop system supplies the ball on disc contact with new lubricant.

Load cells are used to measure the force on the three principal axes where the machine operates, X, Y and Z. The test device also measure shaft rotating

speeds, oil pump speed and values from up to twelve thermocouples. In the current setup three thermocouples are used. One is located in the oil bath, one in the outlet of the oil supply and one measures oil film temperature very close to the inlet region in the ball on disc contact.

The lubricant is supplied to the contact trough the oil dispenser in the middle of the disc in Fig. 1. The supply to the dispenser is secured by a hose pump delivering approximately 60 ml/min.



Figure 1: WAM ball on disc test device

## 2.2. Test specimens and lubricants

Two different pairs of test specimens were used in the test. The first pair, referred to as "smooth" is made from AISI 52100 bearing steel, where the balls are direct from factory and the disc are processed the same way as raceway material. These specimens both have a hardness of  $HRC = 60$  and very smooth surfaces (approximately 30 nm  $S_a$  for the ball and 80 nm  $S_a$  for the disc) whereas the second pair, referred to as "rough" is made of AISI 9310 gear steel for both ball and disc providing a rougher grind closer to gear roughness, approximately 220 nm  $S_a$  for the disc and 200 nm  $S_a$  for the ball. The 9310 disc is case carburized to a depth of about 0.8 mm and has a hardness of  $HRC = 63$ . Both discs have a circumferential grind. The roughness of the discs was measured with a Wyko NT1100 optical profiling system from Veeco. Measurements were done using 10x magnification and 0.5x field of view (FOV). The

Table 1: Mean roughness values of unworn 9310 disc

Diameter [mm]	$S_a$ [nm]	$S_q$ [nm]
95	220	282
81	214	260
60	221	283

Table 2: Test oil data

Type	SL211	SL212	SL326
Additives	None	EP	None
Kinematic Visc @ 40 °C, cSt	30.8	30.7	109.3
Dynamic Visc @ 40 °C, mPas	27.1	27.1	94.9
Kinematic Visc @ 100 °C, cSt	5.3	5.3	11.98
Dynamic Visc @ 100 °C, mPas	4.46	4.46	9.97
Density @ 15 °C, kg/m <sup>3</sup>	872	872	885
Viscosity Index	104	104	99
Type	Mineral	Mineral	Mineral

measurements were made at different diameters of the discs. For each diameter, mean values of seven measurements on different positions of the 9310 disc are presented in Table 1.

Three different lubricants were used in the study. Two pure mineral base oils with the same viscosity, 30 cSt at 40 °C, one of them with a two % EP additive content, and one pure mineral base oil of the same type but with a viscosity of 100 cSt at 40 °C. The lubricant data is presented in Table 2.

## 2.3. Test procedure

The test cycle covers entrainment speeds between 0.34-9.6 m/s and slide to roll ratios from 0.0002 to 0.49, or 0.02 to 49 % slip as used in the present paper. In all cases of slip the ball rotates faster than the disc. SRR, or slip is defined as the speed difference divided with the mean entrainment speed. After the test, surfaces were measured in the Wyko to observe eventual changes in surface topography. Before each test the device and specimens were thoroughly cleaned with heptane and ethyl alcohol, and the test device warmed up approximately 60 minutes before starting the test with lubricant circulation to ensure temperature stability. During the warm up sequence the entrainment velocity is set to 2.5 m/s and there is no load applied, but the ball is positioned very close to the disc so that lubricant is circulated over the ball to ensure warm up. When temperature stability is reached a 200 N load, equivalent to 1.7 GPa Hertzian pressure is applied and the machine

Table 3: Test cases

Track diameter[mm]	Oil	Temp[°C]	Material (AISI)	$S_a$ [nm] (After test)
70	SL326	90	9310	187
74	SL211	90	9310	188
77	SL212	90	9310	186
80	SL212	40	9310	183
82	SL211	40	9310	191
94	SL326	40	9310	194
68	SL212	90	52100	85
74	SL326	90	52100	83
76	SL326	90	52100	76
90	SL211	40	52100	78
94	SL212	40	52100	78

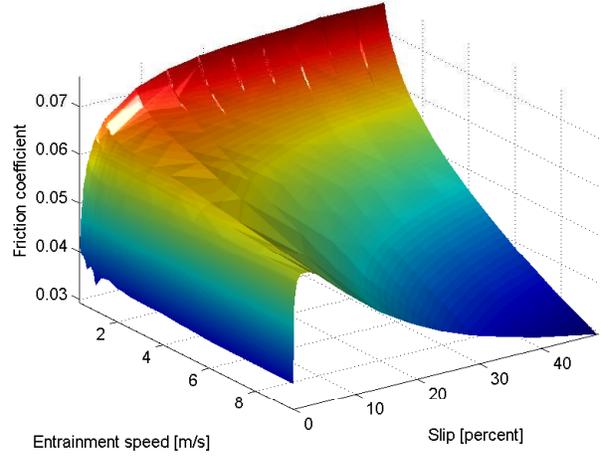


Figure 2: 3D friction map - SL211, 40° C, smooth

calibrated for pure rolling. The machine is run 20 minutes with these settings to ensure a mild run-in. The test cycle is then started which contains several loops where the slip is held constant for each loop and the entrainment speed is varied from 9.6 to 0.34 m/s. In the first loop the slip is held at 0.02 % and is increased with each loop until it reaches 49 %. The test cycle is then repeated in the same track for both ball and disc until the absolute traction coefficient does not vary more than a maximum of 0.002 from the previous test cycle, excluding slip below 0.16 % where the machine scatters a bit.

The logged data from each test is processed separately. All measured values from a specific running condition is averaged, and a triangle based linear interpolation is used between the data points. The result is either plotted as a 3D map, or as a 2D contour plot.

### 3. Results and discussion

The different test cases are shown in Table 3, also containing surface roughness information after completed tests. Figure 2 shows a 3D friction map of one of the test cycles, where friction coefficient is displayed versus entrainment speed and slip. Here one can see the high gradient in friction coefficient when little slip is induced from pure rolling, as well as the general decrease in friction coefficient with increased entrainment speed. In addition to the 3D map, a contour map is usually more suitable for comparisons.

Figure 3 shows a schematic 2D friction map divided into four different regimes, with reasoning

from Johnson and Tevaarwerk [19]. In the linear region "L", shear stress is proportional to shear rate, and are in the presented maps barely visible. The non-linear region, "NL", is dominated by shear thinning effects. In the thermal region, "T" the shear stress decreases with increasing shear rate. Finally one region is marked "M", where asperity contact occurs between the surfaces, which is the mixed lubrication regime. The boundaries of these regimes are exclusive for each system, depending on running conditions as well as on the material and lubricant parameters.

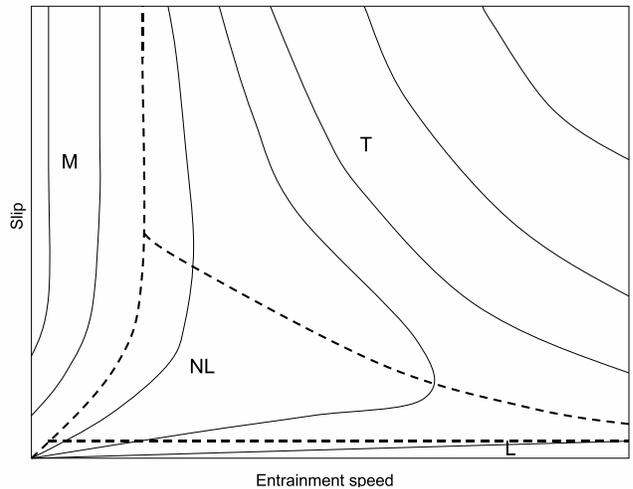
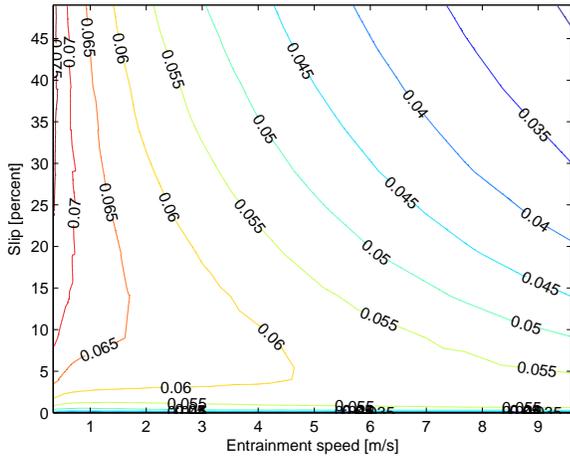
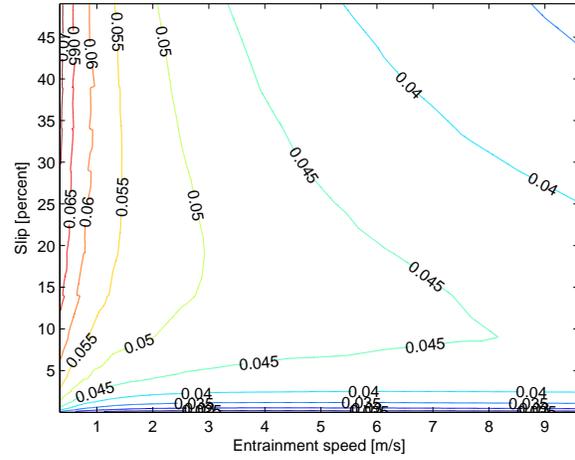


Figure 3: 2D map - Regimes

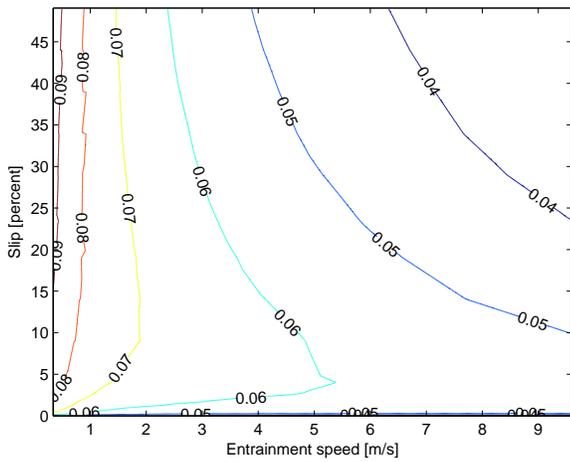
All the results in the present work are presented in 2D contour maps Figs. 4(a)- 6(d).



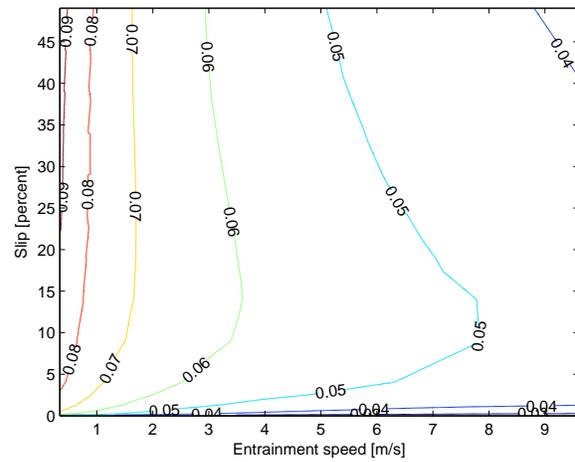
(a) Smooth surfaces, 40°C



(b) Smooth surfaces, 90°C



(c) Rough surfaces, 40°C



(d) Rough surfaces, 90°C

Figure 4: 2D friction maps - SL211, low viscosity oil without EP additive

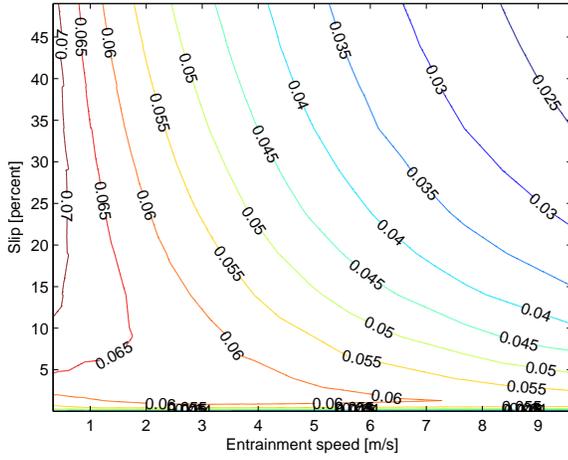
The location of the mixed lubrication boundary is assumed to be where the coefficient of friction is no longer decreasing with increasing slip for a certain entrainment speed, which would imply incipient asperity interactions. However, this is a floating boundary controlled by several parameters, among others the balance between increasing asperity interactions, and the decrease in limiting shear strength and increased shearability with the increase in temperature associated with increased slip, both affecting coefficient of friction, but in opposite ways.

The location of the thermal boundary is assumed to be where the limiting shear strength is reached for the lubricant, and thus entering the region where thermal effects dominates the coefficient of friction.

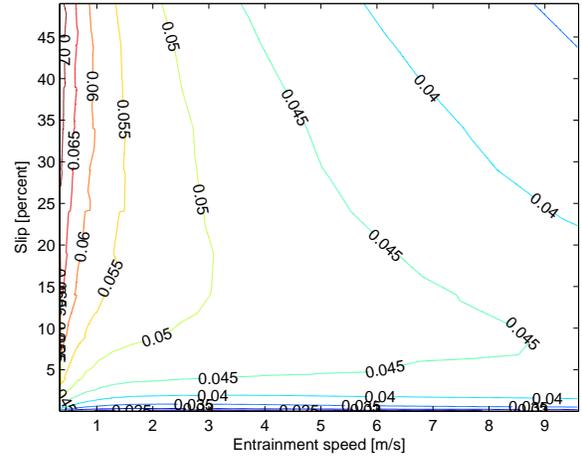
### 3.1. Surface roughness

It is evident from the test results that the surface topography has a rather big influence on friction characteristics. In all cases where oil type and temperature are kept at the same level the test cycles performed with the smoother surfaces gives lower coefficients of friction. Furthermore, the mixed lubrication regime is larger when rough surfaces are used, not surprising since the transition from full film to mixed lubrication is reached at higher entrainment speeds. This could be examined in pairs, for instance, Figs. 4(a) and 4(c) or Figs. 4(b) and 4(d).

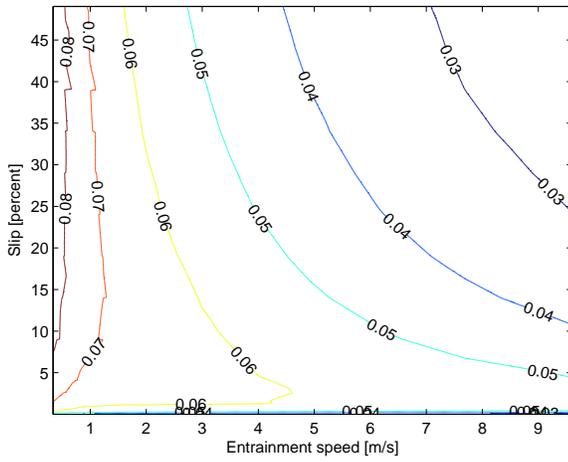
In case of the smooth disc and ball pair, the observations made after the test cycles showed no sign of wear on the disc surface, and a very small wear



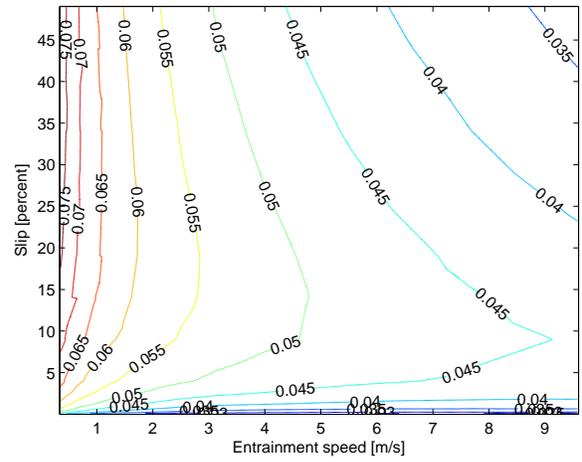
(a) Smooth surfaces, 40°C



(b) Smooth surfaces, 90°C



(c) Rough surfaces, 40°C



(d) Rough surfaces, 90°C

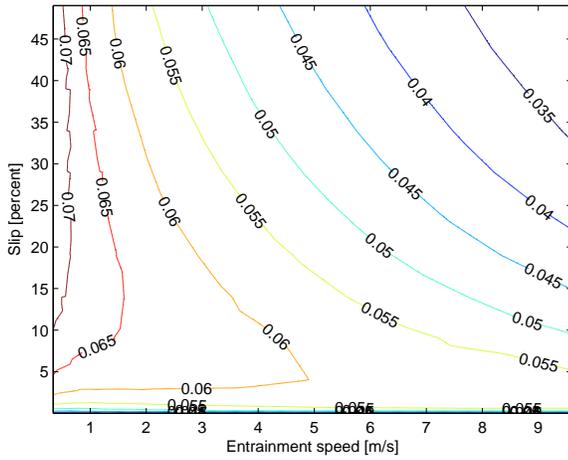
Figure 5: 2D friction maps - SL326, high viscosity oil without EP additive

track on the balls, suggesting a mild running in of the surfaces. The rough balls and disc on the other hand show more pronounced wear, where the  $S_a$  values, in the wear track on the disc, generally are 80 nm smoother compared to the unworn disc. It is, however, difficult to draw any conclusions from the difference in wear when using the various oil and temperature combinations. This is because the resulting surface measurements show very similar results, within 20 nm in  $S_a$  for all wear tracks, and that the difference could as well be attributed to different amounts of roll-overs, since the tracks are located on different diameters on the disc, and that tests have not been run the same number of times for all combinations. Furthermore is the surface roughness varying

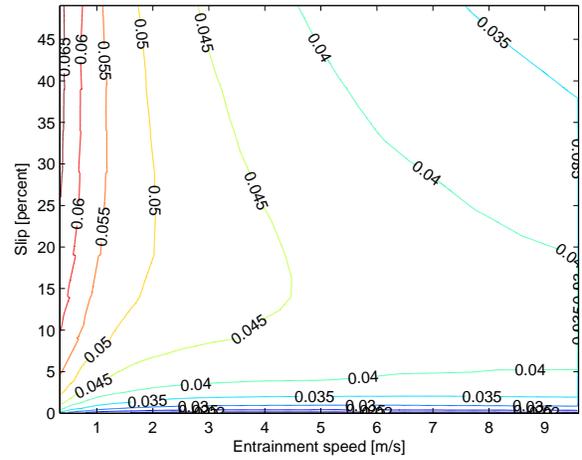
at different locations on the disc even before the tests are performed as shown by Table 1.

### 3.2. Temperature and viscosity

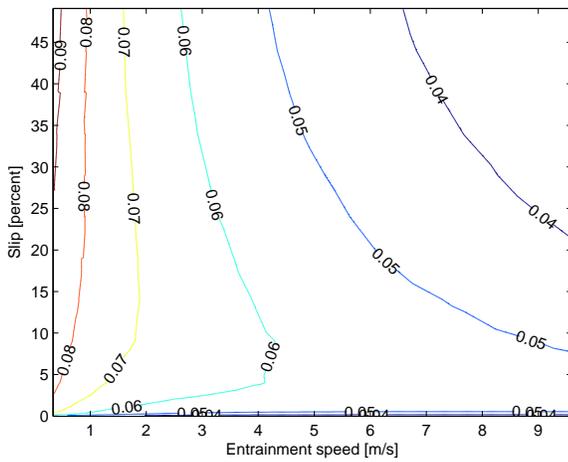
With increasing temperature, pressure viscosity coefficient, limiting shear strength and viscosity are decreasing. [20, 21] As a consequence of this the lubricant, and therefore also the friction characteristics will have different behaviours at 40 and 90°C. At the lower temperature the friction coefficients will increase faster with increased slip, and also reach higher values due to a higher limiting shear strength. However, after the peak value, the friction coefficients will decrease faster as well. This behaviour can be seen in Figs. 4(a) and 4(b) or Figs. 5(a) and



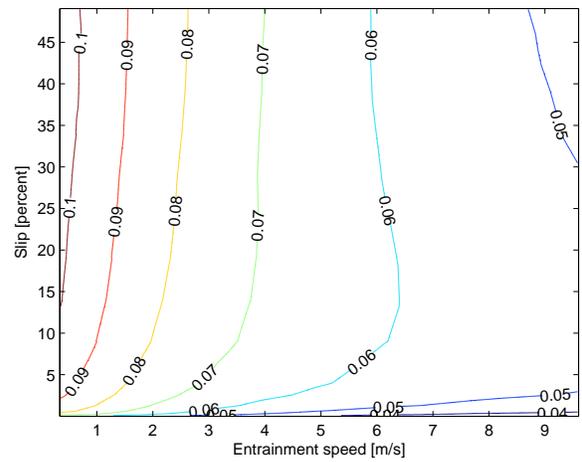
(a) Smooth surfaces, 40°C



(b) Smooth surfaces, 90°C



(c) Rough surfaces, 40°C



(d) Rough surfaces, 90°C

Figure 6: 2D friction maps - SL212, low viscosity oil with EP additive

5(b), where the friction coefficients increases in the thermal region, and decreases in the Linear and Non Linear regions when the lubricant temperature is increased from 40 and 90°C for cases with smooth surfaces.

Cases with rough surfaces follows the same trend, however showing higher friction coefficients in average due to a transition from full film to mixed lubrication regime at higher speeds, thus increasing the area of the mixed lubrication regime, see Figs. 4(c) and 4(d) or Figs. 5(c) and 5(d).

An interesting observation is between the high viscosity oil SL326 at 90° C Fig. 5(b), and the low viscosity oil SL211 at 40° C, Fig. 4(a) tested with smooth surfaces. In this case the SL211 has three

times as high viscosity compared to SL326 since they are run at different temperatures. Despite this the latter show lower coefficients of friction with the exception of a region with high speeds and high slip. This comparison also indicates that the film formed in both cases are sufficient, and that the lower limiting shear strength, and more easily sheared films at higher temperature is beneficial for the friction coefficients.

Changing the lubricant viscosity from low to high with smooth surfaces show small differences, Figs. 4(a) and 5(a) or Figs. 4(b) and 5(b), suggesting that the film thickness in both cases are sufficient for full film lubrication in most parts of the map. However, for the test cases with rough surfaces, a change from

the lower viscosity oil, SL211 to the higher viscosity oil, SL326 show a general decrease of the friction coefficient and a transition from full film to mixed lubrication at lower speeds, Figs. 4(c) and 5(c) or Figs. 4(d) and 5(d).

### 3.3. Additives

When adding EP additive to the low viscosity oil, the friction coefficient in the low temperature tests show very little difference, Figs. 4(a) and 6(a), or Figs. 4(c) and 6(c). At 90°C and with the smoother surfaces, Figs. 6(b) and 4(b), the SL326 EP oil shows a slightly lower friction coefficient, but with the rougher surfaces the friction coefficient is higher compared to the pure base oil, Figs. 4(d) and 6(d). Similar results were reported in [22] where higher friction coefficients were found for rough surfaces with added EP, and lower friction coefficients with added EP with smooth surfaces, compared to the same base oil without EP additive, for measurements made at 90°C. These results suggest that the EP additive becomes more reactive with higher temperature and an increased number of asperity collisions.

## 4. Conclusions

A method for evaluating and presenting contact friction behaviour in EHL tribological systems with respect to surface roughness, temperature and oil parameters under various running conditions are presented. The method gives a good overview, a system finger print, of the frictional behaviour in a broad operating range, and the performed tests show the differences in friction coefficient with respect to different surface topographies as well as viscosity and temperature.

It is shown that a decreased base oil viscosity is non beneficial for contact friction, introducing a transition from full film to mixed lubrication at higher entrainment speeds. It is also shown that an increased operating temperature is beneficial in terms of friction coefficients in the Linear and Non Linear regions and the opposite for the high speed-high slip thermal region.

By using smoother surfaces the transition from full film to mixed lubrication occurs at lower entrainment speeds.

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