ISSN 2176-5480

22nd International Congress of Mechanical Engineering (COBEM 2013)
November 3-7, 2013, Ribeirão Preto, SP, Brazil

INVESTIGATION OF LUBRICATION PERFORMANCE IN FOUR-BALL TESTS OF LUBRICANTS FOR INDUSTRIAL GEAR TRANSMISSIONS

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Abstract. *The viscosity is the most important parameter to increase the lubricant film thickness in order to protect the gear tooth surfaces against wear, pitting and micropitting. Industrial gear transmissions that demand high viscosity lubricants use to perform nicely with lubricants composed by mineral base stocks and binders. However the current trend is to use synthetics in these formulations since it is possible to blend lubricants in many different viscosity ranges and they may be less aggressive to human health. New methodologies for four-ball tribometer have been developed to investigate the performance of gear lubricants under elastohydrodynamic conditions. The results show that mineral lubricants with higher viscosity have lower frictions coefficients while the synthetic blends with lower viscosity showed a higher friction coefficient. The synthetic blends depend strongly on EP-additives to reduce wear.*

Keywords: *high viscosity, four-ball, elastohydrodynamic, EP-additives*

1. INTRODUCTION

The current development industry has been demanding the use of gearboxes transmitting more power with smaller components. This tendency causes the gear teeth to operate under higher contact pressures. Moreover, the need to reduce energy and lubricants consumption due to economic and environmental reasons has required the development of less viscous lubricants associated with smaller reservoirs applied in gear boxes. Thus, the lubricants are required to operate under more severe conditions, where the teeth must be protected against wear with lower energy losses. The energy losses are mainly due to the friction between teeth, but there are also losses due to the movement of the lubricant, diving gear in baths or pumping up the lubrication points (Höhn *et al.*, 2012). The trend is to develop formulations of lubricant with lower viscosities, but with additive packages capable of protecting the gear teeth, with thinner lubrication films. The lubricant between the teeth of a gear are commonly subjected to pressures above 1 GPa and shear rates between 10^6 e 10^7 s^{-1} (Matos, 2004), occurring mainly in the foot and head of the teeth. Such conditions are referred to as elastohydrodynamic lubrication and illustrated in Fig. 1 (Stachowiak and Batchelor, 2006).

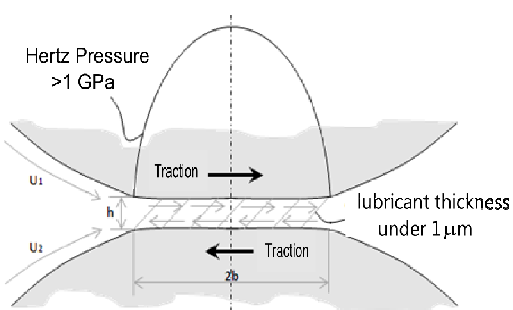

Figure 1. Lubricant film and friction in a contact EHD (Stachowiak and Batchelor, 2006)

Dowson (1995) states that the amount of lubricant required to sustain the proportions of micrometric films formed in elastohydrodynamic lubrication is very small and there is a proportional relation between viscosity and film thickness. So increasing the initial viscosity of the lubricant, the film thickness also increases, especially when increasing the sliding velocity (Luo *et al.*, 1996). Hoglund (1999) proposes that lubricant properties play a significant role in lubricating film formation to reduce the friction between the contact surfaces, especially in elastohydrodynamic lubrication, for example, in gears, bearings and cams. Lubricants with extreme pressure additives have been developed to operate under high loads and high sliding speeds that coexist in contact, being responsible for the formation of a protective film that adheres to surfaces. The chemical reaction occurs primarily between the EP-additive material and the surface reducing film shear stress and reducing friction and wear (Stachowiak and Batchelor, 2006).

In order to investigate the tribological behavior of formulations under elastohydrodynamic conditions, a Four Ball Tester was used, since it has reduced cost and requires small amounts of lubricants for the test. The four-ball tribometer was Plint TE92 equipment that enables to apply normal loads up to 1000N and rotational speeds between 60 rpm and 3000 rpm. In a four-ball test a rotating ball is pressed against three fixed balls in a lubricant bath, as illustrated in Fig. 2, adapted from ASTM D2783-88 (1998) standard. After the tests each, one of the fixed balls have a circular scar.

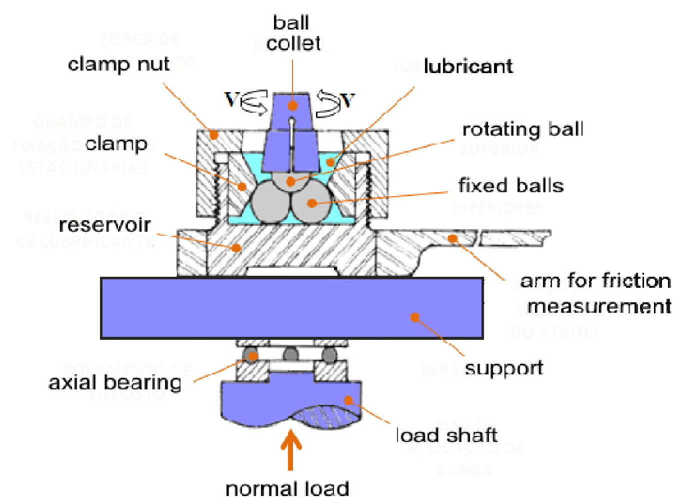

Figure 2. Four-ball test (ASTM D2783-88, 1998)

The formulations with good lubricating performance in four-ball test as well as good coefficient of friction and wear protection, become candidates for additional tests in other tribometers, with more expensive specimens and lubricant consumption, such as disc-disc and FZG (gear geometry) equipments.

2. EXPERIMENTAL METHODOLOGY

For each lubricant tested the friction coefficients were measured under different loads, temperatures, and sliding speeds. Furthermore, the wear protection was evaluated with a microscope and a profilometer. The results obtained with this experimental methodology may be used to pre-qualify formulations for field tests.

2.1 Lubricant formulations

The tested lubricants were developed for application in gear systems used in industrial speed reducers and typically have high viscosity. Lubricants were compared with two different basic stocks: one with mineral asphalt base and the other with a synthetic one. Each lubricant base was also formulated with one extreme pressure additive (EP). The oils formulations used in this investigation are showed in the Tab. 1.

Table 1. Lubricant formulations

Lubricant	Basis	EP - Additive
A1EP	Mineral (asphaltic)	Yes
A1S	Mineral (asphaltic)	No
S1EP	Synthetic	Yes
S1S	Synthetic	No

The dynamic viscosities and densities were measured at 40ºC and 100ºC. The results are showed in the Tab. 2.

Table 2. Viscosities of the lubricant formulations

Lubricant	Dynamic Viscosity [Pa·s]		Density [adim.]	
	40°C	100°C	40°C	100°C
A1EP	3.26	0.0800	0.931	0.891
A1S	4.31	0.0954	0.941	0.901
S1EP	2.07	0.0733	0.910	0.871
S1S	2.14	0.0758	0.917	0.878

The purpose is to compare the tribological behavior between a synthetic blend and a highly viscous lubricant used in gear boxes with high torque transmission. The EP-additive contribution is also investigated and the blends containing this additive have lower viscosities than the formulations without it.

2.2 Friction coefficient tests

In order to achieve all the lubrication regimes, the tests in the Four-Ball tribometer were designed with four different normal loads. Under each normal load, the rotation speed was increased from 60 rpm (0.02 m/s, sliding speed) to 2870 rpm (1.10 m/s, sliding speed). Each rotation speed was maintained for 10 minutes, enough to stabilize the measure of the coefficient of friction, which was related to lubricant sample, normal load, sliding speed and temperature. This procedure presented good repeatability. The oil temperature was controlled and the temperatures of 40ºC and 100ºC were chosen to reproduce the same ones of the viscosity measurements. In the Tab. 3 the sliding speeds and Hertz pressure of each normal load can be observed.

Table 3. Friction coefficient tests conditions

<table>
<tr><th colspan="10">Temperatures [ºC]</th></tr>
<tr><td colspan="5">40</td><td colspan="5">100</td></tr>
<tr><th colspan="10">Normal Loads [N] (Hertz Pressure [GPa])</th></tr>
<tr><td colspan="3">98 (2.17)</td><td colspan="4">196 (2.73)</td><td colspan="3">588 (3.94)</td></tr>
<tr><th colspan="10">Sliding speeds [m/s]</th></tr>
<tr><td>0.02</td><td>0.05</td><td>0.10</td><td>0.20</td><td>0.30</td><td>0.40</td><td>0.60</td><td>0.80</td><td>1.00</td><td>1.10</td></tr>
</table>

The friction coefficient measures at 40ºC with different sliding speeds are shown in the Fig. 3. Similar data measured at 100ºC are shown in the Fig. 4.

The charts in Fig. 3 and Fig. 4 indicate that mineral oil with EP-additive has lower coefficients of friction in all test conditions. It can also be observed that the synthetic oil without EP-additive has shown higher coefficients of friction in almost all test conditions. It is interesting to note that the coefficients of friction were not significantly different by varying the temperature, with an interesting exception of mineral oil with EP-additive, which showed friction coefficients lower under the highest load (588N) and at 100ºC. Maybe the EP-additive works better reducing friction in mineral oil formulations under high contact pressures and high temperatures. This behavior was not observed in the synthetic oil. The test results indicate that a substantial reduction of the lubricant viscosity may not necessarily result in reduced friction.

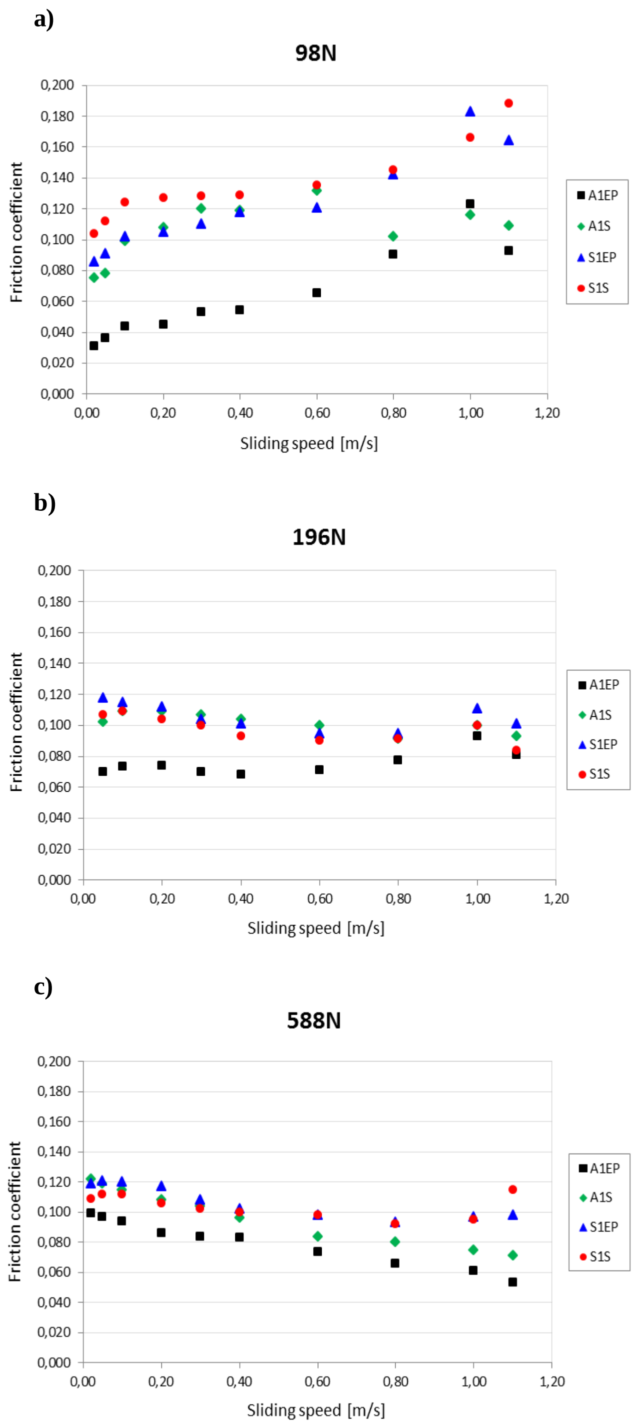

Figure 3. Friction coefficient behavior at 40ºC - Loads: a) 98N ; b) 196N ; c) 588N

ISSN 2176-5480

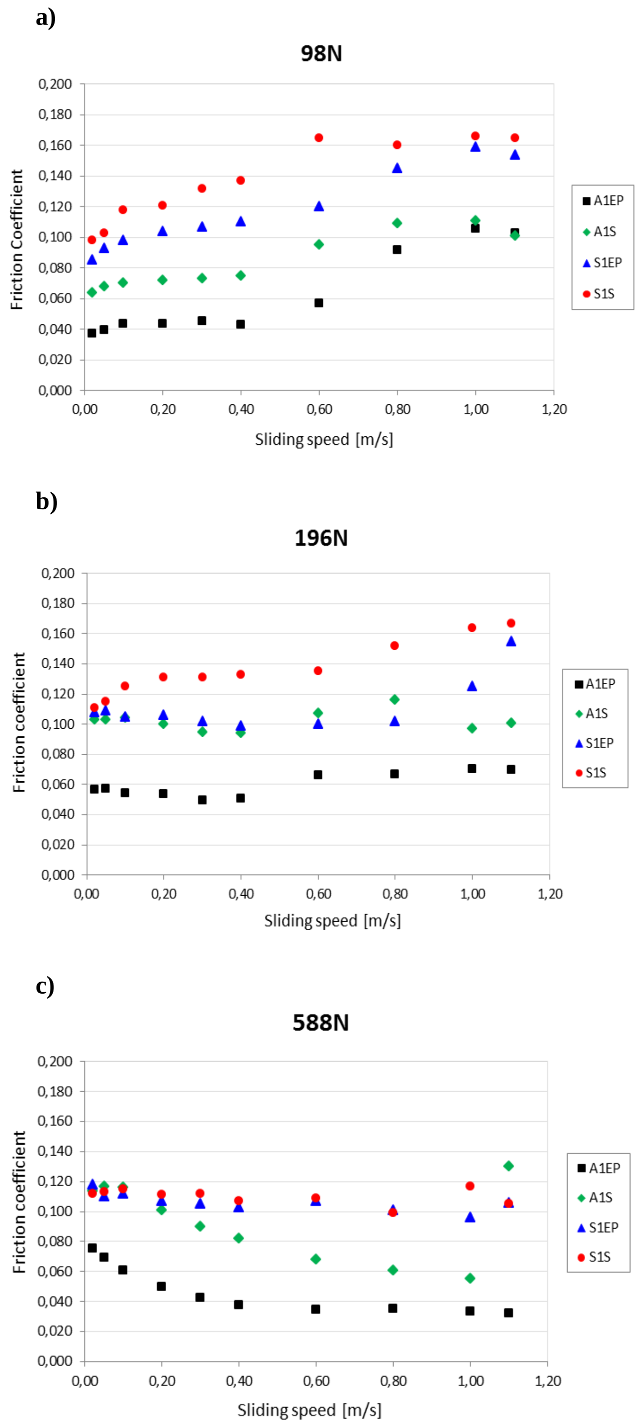

Figure 4. Friction coefficient behavior at 100ºC - Loads: a) 98N ; b) 196N ; c) 588N

2.3 The wear tests

In order to complement the study of tribological behavior of the formulations, wear tests were conducted in the four-ball tribometer. The test conditions were chosen from ASTM 4172-94 (2004) standard, that establishes wear tests with sliding speed of 0.47 m/s (1.200 rpm) during 1 hour with normal loads of 392N or 147N an oil bath temperature of 75ºC.

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The higher normal load, 392N, was chosen to investigate the lubricant behavior under high contact pressures, since the transmission gears are increasingly required to transmit more power with the same geometrical dimensions, resulting in higher contact pressures and the need of better materials and lubricants to protect their surfaces. The objective of the wear tests was to investigate the ability of the lubricants to protect the surfaces against the wear under high contact pressures.

After the tests in the tribometer, the scar diameters were measured in all the fixed balls, in optical microscopic with x-y table, which movement is performed by two micrometers with resolution of 0.01 mm. Each scar diameter was measured in two orthogonal directions. The results, presented in the Tab. 4, are the mean value of the measured scar diameters. The scar profiles were measured with the profilometer Taylor Hobson, Talysurf 50, as it is shown in Fig. 5 and the scar pictures are shown in Fig. 6.

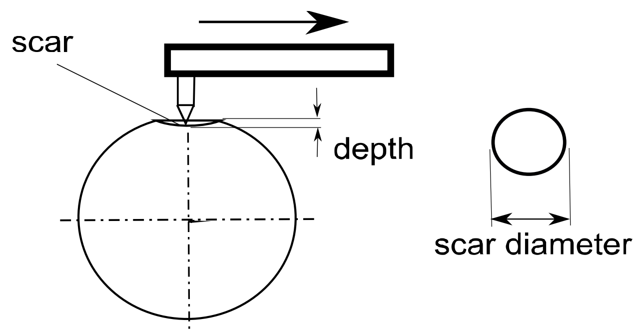

Figure 5. Scar depth measuring in a fixed ball with a profilometer

Table 4. Geometric wear results

Lubricant	Friction Coefficient	Scar diameter [mm]	Wear depth [µm]	Roughness Ra [µm]	Roughness Rq [µm]
A1EP	0.074	0.53	5.94	0.283	0.403
A1S	0.078	0.56	4.84	0.507	0.846
S1EP	0.075	0.33	0.59	0.055	0.077
S1S	0.108	0.90	22.65	0.452	0.717

The wear test results show that the synthetic lubricant with EP-additive has the best performance compared to the other formulations. The wear was so low, confirmed by the small scar depth, that it was difficult to measure its diameter. On the other hand, the same synthetic lubricant without EP-additive showed the highest wear compared to the other formulations. The high wear and high coefficient of friction presented by synthetic lubricant without additive indicates that this formulation must not be used under elevated contact pressures. For the mineral lubricants, the results obtained were similar in magnitude, but the formulation without EP-additive showed higher wear and friction coefficient than the formulation with additives. The lack of EP-additive reduces performance, but this effect was more significant in the synthetic formulations. The high viscosity of mineral formulations produces a film thick enough to protect the surfaces even in the absence of EP-additive. So the synthetic based lubricant is not recommended for use under elastohydrodynamic conditions with high Hertz pressures. It is also interesting to note that despite the better wear protection performance of synthetic based lubricant with EP-additive, its friction coefficient was higher than the one measured in the mineral oil formulation with EP-additive. Therefore, tests have shown that higher friction coefficients not always indicate that there will be more material removal.

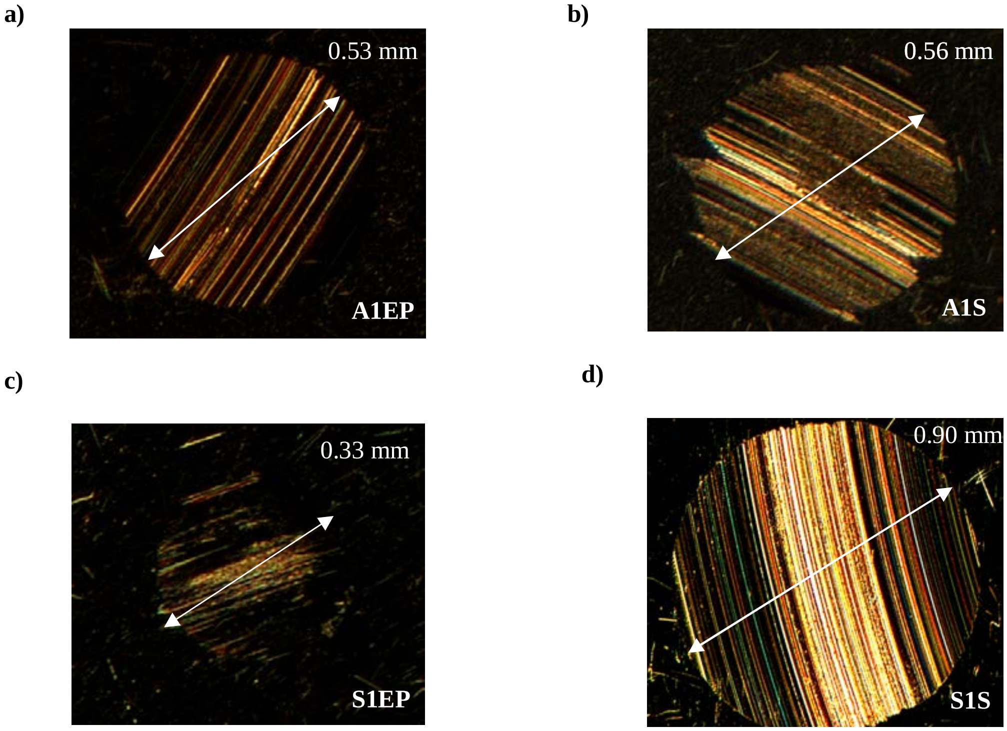

Figure 6. Images in the scars obtained from wear tests – fixed balls

3. CONCLUSIONS

Four different lubricant formulations have been tested in the Four-ball tribometer to compare the performance between synthetic and mineral (asphaltic) based lubricants, with and without EP-additive. Friction coefficients were measured under different loads, temperatures and sliding speeds, and also the wear protection according to ASTM 4172-94 (2004) standard. The results show that the mineral lubricant with high viscosity and EP-additive presented lower friction coefficients in all tested conditions. However, the synthetic formulation with EP-additive had a better performance than the others in the wear test under high contact pressures. Even not having a coefficient of friction as low as mineral oil, its lower viscosity may reduce energy losses associated with the movement of the gears in gearboxes. The synthetic formulation without EP-additive showed a very low performance on the wear protection, practically making its use not recommended. Therefore, the synthetic blend with EP-additive can be considered a candidate to replace asphaltic oils of high viscosity in industrial gearboxes.

4. REFERENCES

Höhn, B. R., Stahl, K., Michelis K. and Otto, H. P., 2012. "*Surface resistance of gear teeth and energy losses lubrication with minimal lubricant quantity (in German)"*, Tribologie und Schmierungstechnik, vol.2, pp. 8-14.

Matos, T.F, 2004. *"Rheological properties of lubricant fluids in elastohydrodynamic conditions (in Portuguese)"*, M.Sc. Thesis , COPPE/UFRJ, Rio de Janeiro, RJ, Brasil.

Stachowiak, G. W., Batchelor, A. W., 2004. *Engineering Tribology (Third Edition)*, Ed. Elsevier.

Dowson, D., 1995. *"Elastohydrodynamic and micro-elastohydrodynamic lubrication"*, W*ear*, Volume 190, Issue 2, December, Pages 125-138.

Luo, J., Wen, S., Huang, P., 1996. *"Thin film lubrication Part I - Study on the transition between EHL and thin film lubrication using a relative optical interference intensity technique"*, Wear, Volume 194, Issues 1-2, June, Pages 107-115.

ISSN 2176-5480

Hoglund, E., 1999. "*Influence of lubricant properties on elastohydrodynamic lubrication*", Wear, Volume 232, Issue 2, October, Pages 176-184.

ASTM International D-2783-88, Revisão1998. "*Standard test method for measurement of extreme pressure properties of lubricating fluids (four-ball method)"*.

ASTM International D-4172-94, Revisão 2004. "*Wear preventive characteristics of lubricating fluid (four-ball method)"*.

5. RESPONSIBILITY NOTICE

The authors Costa Jr., J. B. da, Oliveira, S. J. R de, Lázaro, L. M. S. M. and Lastres, L. F., are the only responsibles for the printed material included in this paper.