A test procedure to investigate lubricant-surface combination for high performance racing transmissions

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A test procedure to investigate Lubricant-Surface combination for high performance racing transmissions

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Abstract

Compact light weight and dry sump (significantly reduced volume of lubricant) are the desired attributes for high performance racing transmissions, whilst improving upon efficiency and reliability remain paramount objectives. The complex multi-objective nature of this task points to an integrated approach to lubricant-mechanical system optimisation. The extreme operating conditions in racing transmissions, such as contact kinematics and thermal loading present significant tribological challenges. Thin lubricant films in non-Newtonian shear are subjected to mixed thermo-elastohydrodynamic regime of lubrication. Under these conditions boundary active lubricant species often determine the contact tribological performance rather than the bulk rheological properties of the lubricant itself. Therefore, the interaction of lubricant additive package with the contacting solid surfaces is the key to an optimised solution.

The paper investigates the lubricant-surface interfacial effect upon frictional characteristics in contact conditions which are representative of gear teeth meshing conditions in high performance transmissions. The study uses pin-on-disc tribometry. As the contact conditions are mainly governed by the formation of surface-adhered tribo-films, their effect upon frictional characteristics is further investigated through use of atomic force microscopy (AFM) in lateral force mode (LFM). A test procedure is presented to benchmark lubricant additive package-surface combinations for improved tribological performance. The investigation takes into account surface material, surface topography and lubricant additive package, all of which affect the tribo-chemical absorption or bonding of a thin film to the contacting surfaces. The test protocol also includes surface chemical spectrometry and Scanning Electron Microscopy (SEM). The presented methodology has not hitherto been reported in literature.

1. Introduction

Transmissions of high performance racing vehicles are routinely subjected to extreme contact loads, generated pressures, temperatures and variations in contact velocity of the meshing gear teeth pairs. These variations can have significant effects upon transmission efficiency. To improve upon the functional performance and reliability of the system, it is crucial to understand the lubricant-surface system.

Owing to the extreme operating conditions and the compact nature of racing transmissions, the loaded gear teeth experience contact pressures of the order of 1 – 3 GPa, coupled with contact velocities in the range 0-30 m/s [1]. Thus, the contact is subjected to a wide variety of transient lubrication conditions, including thin films of the order of surface roughness.

The predominant regime of lubrication is elastohydrodynamic in all loaded gearing systems. This mode of lubrication is found in all highly stressed non-conformal machine elements such as gear pairs. The development of EHL theory during the 1950s-70s [2], [3] explained the inconsistencies and discrepancies between the previous understandings and experimental evidence.

The current trend for transmission engineering is to develop light weight and compact systems, whilst improving upon transmission efficiency, noise and vibration refinement [4] as well as system reliability. Therefore, often with high performance transmissions an air-oil mist environment with a vapour spray system is utilised in preference to an oil sump. This reduces additional fluid sloshing which can affect the cornering or braking manoeuvers of racing vehicles. Due to the fact that the vapour spray lubrication system has a significant void fraction and liberated vapour, there is a greater incidence of partial lubrication, starvation and cavitation. Therefore, the conditions within the transmission induce a mixed regime of lubrication. This causes the contacting loads to be transmitted through a combination of asperity interactions on the counter face surfaces of gear teeth, as well as through lubricant piezo-viscous response. Owing to the thinness of the lubricant film a significant investment is made into the development of hard wear-resistant surface coatings and lubricant additives to improve upon tribological performance in all such contacts [5]–[7]. Effort has also been expended in the development of lubricant and additives packages to mitigate some of the adverse effects of boundary interactions, lubricant degradation and improve upon its thermal stability [8], [9].

This paper highlights a methodology to investigate the salient parameters of the lubricant–surface combination. The developed method is used to improve the understanding of lubricant–surface system and the formation and retention of ultra-thin low shear strength films adsorbed onto the contacting surface, and finally benchmark and characterise a series of lubricants for a range of surfaces. This methodology is required due the complex nature of interactions of lubricant additives with the real rough contacting surfaces of various surface material or coatings, including in the case of meshing gear teeth [10]–[12].

2. Method

A detailed method is presented to analyse a set of pre-determined parameters for the evaluation of lubricant–surface combinations, within representative contact conditions. To achieve this, the first step is to use the Greenwood chart [13]–[15] to determine the operating regime of fluid film lubrication for the transmission system. This is in order to create the same conditions with pin-on-disc experiments. The dimensionless viscous; G_v and elastic; G_e parameters are obtained as:

$$G_v = \frac{G^* {W^*}^3}{{U^*}^2} \qquad G_e = \frac{{W^*}^{8/3}}{{U^*}^2} \tag{1}$$

where, the dimensionless load, speed (rolling viscosity) and materials' parameters and the lubricant used are:

$$W^* = \frac{W}{E\prime R^2}, U^* = \frac{U\eta}{E\prime R}, G^* = E'\alpha \tag{2}$$

where, W is the applied contact load, U is the sliding velocity, R is the equivalent radius of the meshing teeth pair during a meshing cycle (for the case of gears), η is the dynamic viscosity of the lubricant used and E' is the effective (reduced) Young's modulus of elasticity of the contact:

$$\frac{1}{E\prime} = \frac{1}{\frac{E_1}{1-\vartheta_1^2}} + \frac{1}{\frac{E_2}{1-\vartheta_2^2}} \tag{3}$$

where, $E_{1,2}$ and $\vartheta_{1,2}$ are moduli of elasticity and Poisson's ratios for the pin and disc materials respectively.

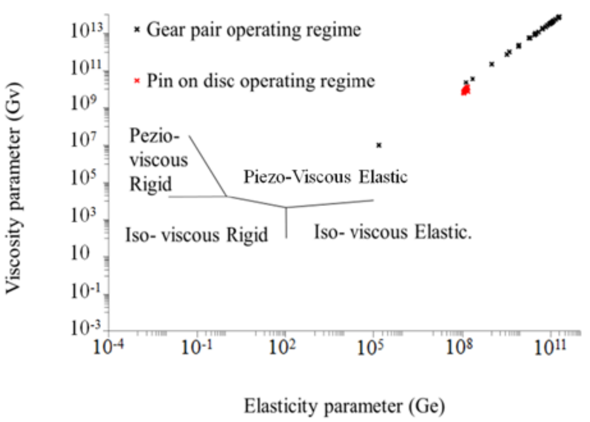

Figure 1 - A comparison between lubricant operating regimes for the transmission and the pin on disc tribometer

The conditions for the meshing gear teeth pair are shown in Figure 1. The analysis shows that the contact condition remains within Piezo-viscous Elastic (i.e. elastohydrodynmic) regime of lubrication. These conditions are then created, as far as possible in the pin-on-disc tests, with attainable conditions also indicated in Figure 1. With a small Hertzian circular footprint, representative pressures are generated. However, due to the high entrainment velocities in the high performance transmissions, only a limited proportion of the transmission meshing cycle can be accurately replicated. High values of viscous parameter indicate high shear conditions, present under non-Newtonian conditions. Therefore, the second step is a comparison made between the test conditions and those in situ in the transmission gear meshing, based on the Deborah Number [16] as this would indicate the regime of traction and extent of non-Newtonian behaviour. The Deborah Number is defined as the ratio of lubricant relaxation time to the time of its passage through the contact [14], [16]:

$$D = \frac{\eta U}{2aG} \tag{4}$$

where, a is the radius of the Hertzian circle in the case of the pin-on-disc experiments and G is the lubricant shear modulus. A value of $D \gg 1$ is obtained, indicating non-Newtonian traction. For this purpose, the lubricant dynamic viscosity η is corrected for the generated pressures and contact temperature using Roeland's equation, [17] while the elastic shear modulus was corrected using a linear relationship given by Dalmaz et al [18]. Good conformance is noted between the pin-on-disc condition and that in the meshing of the transmission gearing teeth. As with the Greenwood chart analysis, the major limiting factor with tribometry is the lubricant entrainment velocity.

Both techniques used have shown that the Pin-on-disc tribometry would be a suitable tool for benchmarking the lubricant-surface combinations and replicating a portion of the conditions experienced in the high performance transmission. Therefore, analysis of created tribo-films can be carried out with a good degree of confidence.

The following parameters are assessed through the devised testing procedure:

1. Presence of adsorbed/bonded lubricant additives to the contacting surfaces
2. Determination of boundary and viscous friction
3. Coefficient of friction

A testing protocol is developed to evaluate the above stated parameters. Extensive testing is carried out to determine the state of solid contacting surfaces and the lubricant prior to and after experimentation as in-situ analysis is not possible. Comparisons are made for pre and post testing conditions through use of Atomic Force Microscopy (AFM), and X-ray Photoelectron Spectroscopy (XPS). White light interferometry is used to monitor the evolving surface topography. AFM in lateral force mode is employed to obtain friction characteristics of the specimen.

A 3 stage testing procedure is used: 1) Virgin sample – this is the initial testing stage with a new surface sample and without the introduction of a lubricant. Measurements with a new dry sample form the baseline against which the frictional performance of subsequent tested samples in the presence of a lubricant, are ascertained. The hypothesis is that under combined pressure, shear and thermal loading, boundary active lubricant species reach their activation energy barriers and adsorb or bond to the contacting surfaces. 2) Wet sample – this is the second stage in the process. Pin-on-disc experiments determine the coefficient of friction of contacting pairs under both dry and wet (lubricated) conditions. The effect of any formed tribo-film can be noted. 3) Wet samples are suitably cleaned of hydrocarbon residue (Dry specimen post lubricated tribometry) and coefficient of friction is measured by AFM in lateral force mode, and compared with the same measurement of the original dry (virgin) sample. Care is taken to ensure identical surface roughness topography for both samples.

This approach provides measurement of friction at micro-scale (tribometry) and meso/nano-scale (AFM-LFM). The experimental set up for pin-on-disc tribometry, includes a signal generator to enable accurate control of the rotational speed of the disc sample. A copper heated plate with a thermo-couple feedback loop is used to control the bulk surface temperature of the sample disc surface. A digital microscope is used to record the inlet and outlet conjunctional menisci, ensuring repeatable lubricant availability in the case of lubricated conditions. A Wheatstone bridge strain gauge set-up on the measurement arm allows for direct measurement of generated contact friction. The contact is loaded with a counter-lever loading arm. A Gravity lubricant feed system and a roller wiper is used to maintain a controlled inlet meniscus ahead of the contact.

3. Results

A series of experiments are conducted to obtain the coefficient of friction at different contact pressures, sliding speeds and bulk disc temperatures. Testing is carried out for discs made of EN36C gear steel, case carburized to a depth of 1.2mm and tempered to a hardness of approximately 700HV. The pin is a 10mm 316 bearing steel. The test conditions varied the sliding from 0.9m/s – 5.5m/s for a contact pressure of 1.2GPa, contact pressure was varied from 1.2GPa to 2.2GPa for a sliding velocity of 1.28m/s and bulk temperature of disk was altered from room temperature to 100^{O}C.

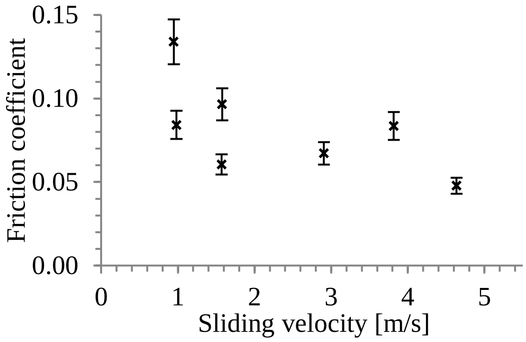

Figure 2 - Friction coefficient variation with sliding velocity for 1.2GPa contact pressure, 60^{o}C bulk surface temperature for a lubricated contact

Figure 2 shows an example of coefficient of friction variation with sliding velocity. The characteristics are assembled from a series of tests at different sliding velocities with the same contact load, disc bulk surface temperature and lubrication condition. All tests are carried out for the same sliding distance which eventually yields a steady state coefficient of friction. White light interferometric study of the disc samples show the presence of a run-in wear track, Figure 3, on the surface of disc samples, corresponding to the steady state coefficient of friction. Of course there are a host of such characteristic curves for different applied pressures and surface temperatures. Therefore, a comprehensive characteristic carpet plot can be obtained. The procedure, including the appropriate international standards is highlighted in [19].

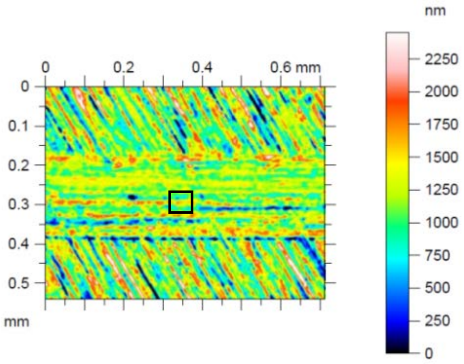

Figure 3 - An optical microscope image (×20 optic) of the wear track caused during testing.

A SEM image of the wear track is shown in Figure 4.

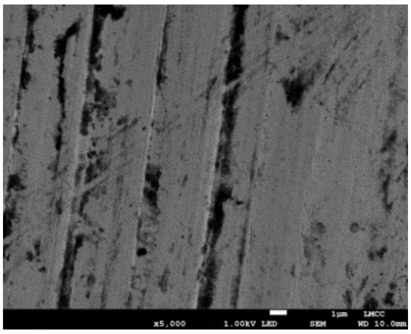

Figure 4 - An SEM image showing the surface of the disc within the wear scar

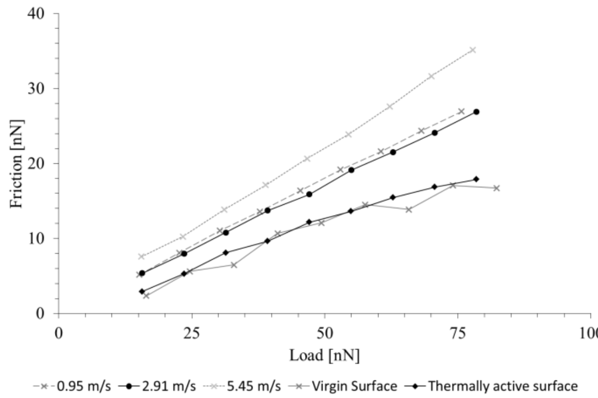

Figure 5 – The boundary friction coefficient for different test conditions

Table 1

Condition	Boundary coefficient of friction
0.95 m/s	0.36
2.91 m/s	0.34
5.45 m/s	0.45
Virgin Surface	0.22
Thermally active surface	0.24

A $1\mu m^2$ section of the wear track shown in Figure 3 was scanned by the AFM in lateral force mode. It was shown that there was significant variation in the coefficient of friction, Figure 5. This can be due to the deposition of anti-wear tribo-films from the lubricant, as this is known to increase the coefficient of friction, as seen in the results of Table 1. As further evidence for this, the AFM shows regions of significantly different friction within a sampled area. This sudden change in friction indicate changes in the surface contact with the AFM tip due to the formation of a tribo-film.

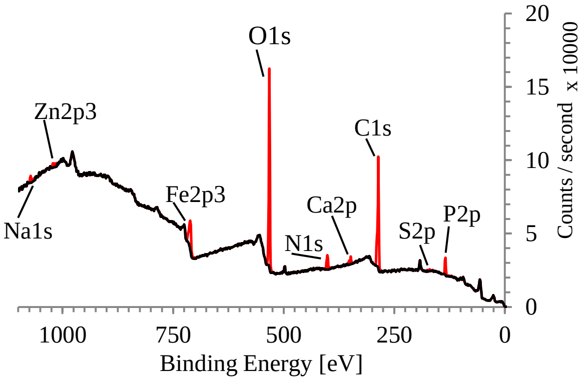

Figure 6 - A XPS spectrum for 0.95m/s test conditions

The XPS results show that the original dry sample surface contains Iron, Nitrogen, Calcium and Carbon, which are expected from the sample composition and the carburising hardening process. The XPS analysis for the post lubricated tested disc surface indicates the presence of additional elements such as Zinc, Sodium, Sulphur and Phosphorus, Figure 6. Elements of this type are present in the transmission fluid [9]. These elements offer performance enhancement as anti-wear, anti-oxidation agents and friction modifiers [9]. Therefore, the results show the presence of a tribo-film generated through combined shear, pressure and temperature activation of lubricant additives with the Pin-on-disc experiment. Furthermore, AFM-LFM shows that the boundary friction of the bulk disc specimen surface is affected by the formation of this tribo-film in the form of a change in the measured coefficient of friction.

4. Conclusions

A detailed test methodology is presented with the aim of analysing the lubricant-surface combination as a system, representative of meshing of gear teeth pairs of high performance racing transmissions. This contact is simulated by a Pin-on-disc tribometer under similar conditions pertaining to the regime of lubrication and traction. Extensive experimental analysis through a stringent testing procedure and analysis process shows that the method is suitable for characterisation of lubricant-surface systems. The results of the analysis show good agreement with those reported in literature, thus imparting a good degree of confidence.

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References

[1] E. Fatourehchi, V. Elisaus, M. Mohammadpour, S. Theodossiades, and H. Rahnejat, “Efficiency and Durability Predictions of High Performance Racing Transmissions,” *SAE Int. J. Passeng. Cars - Mech. Syst.*, vol. 9, no. 3, pp. 2016-01–1852, 2016.

[2] D. Dowson, *History of Tribology*, Second. London: Professional Engineering Publishing Ltd, 1998.

[3] D. Dowson and G. R. Higginson, *Elasto-Hydrodynamic Lubrication*, Second. Oxford: Pergamon Press, 1977.

[4] M. Mohammadpour, S. Theodossiades, H. Rahnejat, and P. Kelly, “Transmission efficiency and noise, vibration and harshness refinement of differential hypoid gear pairs,” *Proc. Inst. Mech. Eng. Part K J. Multi-body Dyn.*, vol. 228, no. 1, pp. 19–33, 2013.

[5] S. V. Hainsworth and N. J. Uhure, “Diamond-Like Carbon Coatings for Tribology: Production Techniques, Characterization Methods and Applications S.V. Hainsworth & N.J. Uhure Department of Engineering, University of Leicester, University Road, Leicester, LE1 7RH,” *Methods*.

[6] G. E. Totten, L. Wedeven, M. Anderson, and J. Dickey, *Bench testing of industrial fluid lubrication and wear properties used in machinery applications*. 2001.

[7] S. Howell-Smith, H. Rahnejat, P. D. King, and D. Dowson, “Reducing in-cylinder parasitic losses through surface modification and coating,” *Proc. Inst. Mech. Eng. Part D J. Automot. Eng.*, vol. 228, no. 4, pp. 391–402, 2014.

[8] a V Olver, “Testing transmission lubricants: the importance of thermal response,” *Proc. Inst. Mech. Eng. Part G J. Aerosp. Eng. 1989-1996 (vols 203-210)*, vol. 205, pp. 35–44, 1991.

[9] L. R. Rudnick, Ed., *Lubricant Additives*, Second Edi. New York: CRC Press, 2009.

[10] P. J. Blau, “The significance and use of the friction coefficient,” *Tribol. Int.*, vol. 34, no. 9, pp. 585–591, 2001.

[11] H. Ronkainen *et al.*, “Coating evaluation methods: A round robin study,” *Tribol. Lett.*, vol. 17, pp. 453–463, 1990.

[12] M. Leighton, T. Nicholls, M. De la Cruz, R. Rahmani, and H. Rahnejat, “Combined lubricant-surface system perspective: Multi-scale numerical-experimental investigation,” *Proc. Inst. Mech. Eng. Part J J. Eng. Tribol.*, vol. 0, no. 0, pp. 1–15, 2016.

[13] B. K. L. Johnson, “Regimes of elastrohydrodynamic lubrication,” *Mech. Engeering Sci.*, vol. 12, no. 1, pp. 9–16, 1970.

[14] H. Rahnejat and R. Gohar, *Fundamentals of Tribology*, Second. London: Imperial Collage Press, 2012.

[15] R. Gohar, *Elastrohydrodynamics*, Second. London: Imperial Collage Press, 2001.

[16] M. Reiner, "The Deborah Number," *Phys. Today*, vol. 17, no. 1, p. 62, 1964.

[17] C. Roelands, "Correlational Aspects of the Viscosity-Temperature Pressure Relationship of Lubricating Oils," Delft Universtiy of Technilogy, Netherlands, 1966.

[18] G. Dalmaz, A. Lubrecht, D. Dowson, and M. Priest, *Transient Processes in Tribology*, First. Amsterdam, Netherlands: Elsevier B.V., 2004.

[19] E. Humphrey, T. Gkinis, N. Morris, M. Leighton, R. Rahmani, and H. Rahnejat, "Clutch Lining Frictional Characteristics under Thermal Tribodynamic Conditions," pp. 1–12, 2016.