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Mittal, Sachin
Department of Mechanical Engineering, NIT

Dr. Vinod Kumar Mittal
Department of Mechanical Engineering, NIT

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Experimental Investigation for Tribological Properties in Mixed Regime of Lubrication with VG-46 and VG-32.

Sachin Mittal¹, Dr. Vinod Kumar Mittal^{2,*}

¹Department of Mechanical Engineering, NIT, Kurukshetra, India

²Department of Mechanical Engineering, NIT, Kurukshetra, India

*Author to whom correspondence should be addressed:

E-mail: sachin.mittal0529@gmail.com

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Abstract: One of the utmost important liquid lubrication regime is mixed lubrication regime as it occurs in almost all of the industrial machines & equipments. The most of the major impairments such as wear, micro pitting etc. happen in this lubrication regime. These days, mixed lubrication regime is the least tacit and so a lot of scope for the research is available in the same. Therefore, authors have focused themselves to investigate the effect on various tribological parameters i.e., friction & wear for sliding pair of EN31 & Brass with VG-46 & VG-32 hydraulic oils in mixed lubrication regime. This tribological pair is the most important and standard pair used in various industrial applications. VG-46 & VG-32 are also the most effective lubricants used in various industrial setups. This study will be helpful for various practical applications further because of its widely industrial applicability.

Keywords: mixed lubrication; frictional force, wear, EN31, Brass, VG-46, VG-32

1. Introduction

One of the extremely important, liquid lubrication regime is mixed lubrication regime as it occurs in almost all of the industrial machines & equipments. At present, mixed lubrication regime is the least tacit from other regimes and a lot of scope for the research is available in the same. One of the major reason behind the same is that, the study of this regime requires the acquaintance of both the boundary & fluid film lubrication, and understanding of the boundary lubrication is very complex & currently very limited knowledge is available for the same.

H.A. Spikes¹) recognized the major areas of progress & further improvements after reviewing the work in mixed lubrication regime. In this study, he has discussed about the film thickness and surface roughness and concluded that there is limitation of earlier models below certain values of λ . Electrical contact resistance and friction measurement techniques are also used for understanding the mixed lubrication behaviour. H.A. Spikes et al.²) have concentrated on two of the important practical features i.e. friction and fluid entrainment in mixed lubrication regime. Spike et al.³) reviewed the state of the knowledge (and lack of knowledge) of mixed lubrication. In this paper the authors have found it appropriate to distinguish between two different rough surface lubrication regimes. One is micro elastohydrodynamic lubrication and the second is mixed

film lubrication. Author has concluded that very rapid progress is currently taking place, both in experimental and modeling work in mixed film lubrication, but still there is a lot more to do. Hugh Spikes⁴) has reviewed and discussed the various upcoming challenges in the ongoing fundamental research in coming years. On the basis of the study of the current research activities trend & opportunities, five areas have been suggested for study in next few years. Simulation & modelling of mixed lubrication is one of the areas arose from them.

As per Koji Kato⁵), modern lubricated tribo-pair such as automobiles, airplanes and disk drives strongly require their wear rate predictions and control. Their much better tribological performances are being required under severer operating conditions which have never been experienced. Boundary and mixed lubrication are the major regimes where wear of these lubricated tribo-elements is concerned. For understanding the wear in boundary & mixed lubrication regimes, oil lubrication of steel is considered by author to see the representative wear properties and wear mechanisms. Nine wear modes in mechanical or tribochemical wear, lubricated wear of steels is characterized in relation to lubrication. H S Cheng⁶) updated on the previous reviews with a coverage of some recent analytical modeling developments in the past decade on mixed lubrication.

Soren Andersson et al.⁷) has provided various friction models depending upon the surface running conditions, material behaviour & type of contacts. This study was

performed on various sliding contacts running with boundary and mixed lubrication regimes. R Larsson⁸⁾ in one of his contribution mentioned that the lubricant film thickness, friction, wear rate & lubricant flows are the key factors which are to be predicted by modelling of various lubricants under boundary & mixed lubrication regimes. He has also clarified that the area of lubrication modeling is having lot of scope for further research & improvement. It is almost impossible at this stage to predict in this area with the same reliability & correctness as in structural and fluid mechanics. Illner¹⁰⁾ evaluated the performance of the sliding contacts for the application of diesel injection pump in mixed lubrication regime & confirmed that the result for the experiments are coming in agreement with the simulation results. Issakson et al. and Nilsson et al.¹¹⁻¹³⁾, worked on the simulation study of the hydraulic oil lubricated partial journal bearing having steel roller & cast iron bushing. The frictional force values resulted from this simulation study are agreeable & in line with the measured values for most of the surface roughness conditions.

Sahlin et al.^{14,15)} accomplished the lubricant flow measurement simulation study between two flat plates & the results were very much encouraging and in agreement with measured values. Priestner et al.¹⁶⁾, Allmaier et al.¹⁷⁻¹⁹⁾, and Sander et al.²⁰⁻²²⁾ conducted various study on the behaviour of frictional force for journal bearings & provided broad comparisons of the same in weak mixed lubrication regime. Greenwood and Tripp model⁹⁾ was used in all of the above works for the purpose of the asperity contact pressure. In summary, we can infer from the above publications that frictional force for the different lubricant & material combinations of journal bearing in mixed lubrication regime can be easily & reliably predicted by micro- and macroscale simulation approach with load sharing. Albers and Lorentz et al.²³⁾, presented the three-dimensional microscopic study of mixed lubrication. The effect on the coefficient of friction and thermal dissipations because of variations in surface topography was also analyzed in detail in this study. Akchurin et al.²⁴⁾ computed the friction force of line contacts using simulation methods. S.Fricke et al.²⁵⁾ analyzed the change in behaviour on frictional parameters by varying surface form deviations in mixed lubrication regime. Anthony et al.²⁶⁾ reviewed and presented the mechanisms to reduce the friction & wear using nano particle additives in the lubricant. In addition to that Norfazillah et al.²⁷⁾ has analyzed the effect of activated carbon nanoparticle additive in modified jatropha oil on various tribological parameters and confirmed the suitability of this oil for various metal working fluid applications. The effect of flat surface micro pitting on frictional characteristics during sliding contact with lubrication was investigated by Aiman et al.²⁸⁾ They confirmed that micro pitting on flat surface is helpful to reduce friction between sliding surfaces. Changru et al.²⁹⁾ studied & illustrated the role of

lubricants in heat pump systems & its effect on heat pump performances.

W.K Shafi et al.³⁰⁾ & Love Kerni et al.³¹⁾ have investigated the effect on the various tribological parameters i.e. Friction & Wear for various vegetables oils in mixed lubrication regime. Researchers have majorly touched upon the areas of numerical modelling and simulation studies for mixed lubrication regime. Tribological parameters i.e. friction & wear study for various industrial pairs considering industrial lubricants in mixed lubrication regime are the necessity of the hour. This is because almost all of the failures for major industrial moving pairs happen in this regime only. Our recent study will surely help to bridge the gap in this area of research. As this tribological pair & VG-46 & VG-32 lubricants are used in various industrial applications. This study on the investigation of various tribological parameters for these industrial lubricants will be useful in further improvements in the design & selection of the materials for a large segment of machineries & equipment's.

2. Materials & methods

2.1 Base oils

ISO VG 46 & VG-32 hydraulic oils are high-quality oils designed to use in a variety of industrial and automobile applications. It is normally applied as a means to reduce the impact of unfavorable surface-to-surface contact in various machines & equipments.

These oils are having excellent anti wear, anti-corrosion, thermal stability along with low air entrainment, low foaming properties & resistant to oxidation. These oils are also having high shear stability because of their high viscosity index. Above properties make them suitable for use in most of the industrial applications. Table 1 gives the rheological properties of VG-46 & VG-32 oils. It can be inferred from the table that the viscosity of VG-46 oil is more as compared to the viscosity of VG-32 oil.

Table 1. Rheological properties of VG-46 & VG-32 oils

S. No.	Properties	VG-46	VG-32
1	Viscosity (mm ² /s) at 40°C	46.0	32.0
2	Density (Kg/m ³) at 40°C	0.861 *10 ³	0.857 *10 ³
3	Flash point (°C)	232	220
4	Pour point (°C)	-27	-27
5	Viscosity Index	98	98

2.2 Tribopair

EN 31 steel discs with hardness of 60 HRC were selected for the experimentation because of their excellent hardness & abrasion resistance properties.

Spherical balls made up of brass material were used for the testing because of its wide application in bearings & mobile applications. The diameter of the disc is 40.0 mm \pm 0.1 mm & thickness is 4 mm. Surface finish of the discs are maintained up to the level of N3-N4. The diameter of the ball is 12.7 mm & having smooth surface finish. This tribopair of EN 31 steel & brass are being used in most of the industrial components & machineries. This certainly makes our study very useful & pertinent in most of the industrial applications. Mechanical properties of both the material selected for testing are given in Table 2.

Table 2. Mechanical properties of ball & disc materials

S. No.	Properties	EN 31 steel	Brass
1	Tensile strength (N/mm ²)	750	360
2	Yield strength (N/mm ²)	450	140
3	Mod. of elasticity (N/mm ²)	215000	117000
4	Density (g/cm ³)	7.81	8.39
5	Hardness (HRC/HRB for Brass)	60, 40	93
6	Poisson's ratio	0.27- 0.30	0.34

2.3 Experimentation

Tribological parameters testing for the selected tribopair of EN-31 & brass using VG 46 & VG-32 lubricants were accomplished on a Ball/Pin-on-disc tribometer which is operated through computer using ASTM G-99 standard. Photographic view of the experimental setup is shown in Fig 1.

Load cell was available in the tribometer for obtaining the frictional forces during experiments. Speed, wear, time & frictional forces readings were displayed on the digital instrument. This instrument was in synchronization with the computer through DUCOM software where all the relevant graphs were plotted & recorded for future references.

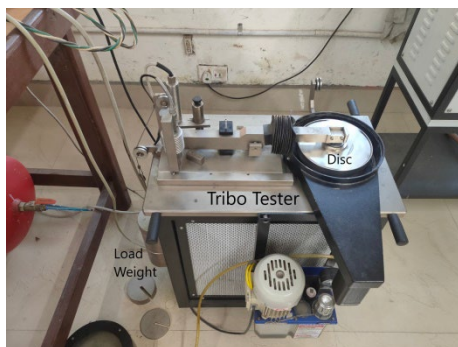
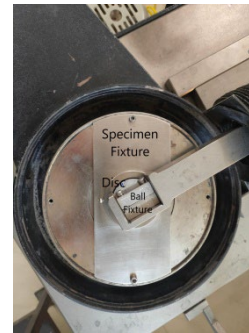


Fig. 1: Ball/Pin on disc tribometer.

Disc was placed & tightened in the fixture provided as an attachment to this setup. PCD of 6mm was set using the linear scale adjustment mechanism. Lubricant was applied on the disc in drop wise manner. Disc & balls were cleaned using acetone before & after the experiments. Stribeck curve for the tribopair taking both the lubricants (VG-32 & VG-46) were plotted one by one after conducting experiments & mixed lubrication regime range for the same were achieved. Tribological tests in this mixed lubrication regime were conducted at different speeds & with constant load.

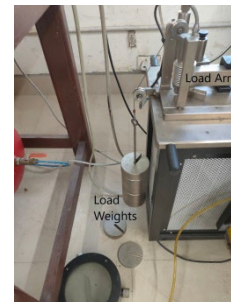
Fig 2 shows the photographic view of the PCD linear adjustment scale, disc adjustment, load mechanism, digital instrument & computer system.



(a)



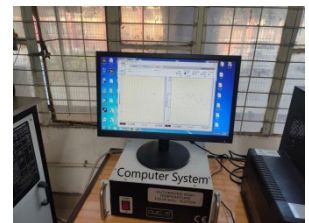
(b)



(c)



(d)



(e)

Fig. 2: Images of tribometer (a) PCD adjustment scale (b) Disc adjustment (c) Load mechanism (d) Digital instrument (e) Computer system

3. Results and discussions

3.1 Stribeck curve

Stribeck curve shows the relationship between, a dimensionless parameter of lubrication (λ) and the coefficient of friction (COF). This dimensionless

parameter is also referred as Hersey number which is defined as below.

$$\text{Hersey number}(\lambda) = (\mu \cdot N)/P$$

The experiments were performed for the selected tribopair of EN-31 & brass to generate the stribeck curve for both the lubricants. Fig 3 shows the stribeck curve for VG 32 & VG 46 oils. It can be evident from the plot that COF value increases to some extent initially and then decreases towards the minimum value for both the oils. This conversion is marked as the change of boundary lubrication regime to mixed lubrication regime. The maximum value of COF in case of VG-46 oil is 0.189 as compared to 0.236 in case of VG-32 oil for the specific film thickness of 0.75. This point indicates the origination point of mixed lubrication.

COF value decreases further from this point & attains the minimum value at specific film thickness of 1.02. Minimum value of COF for VG-46 oil is recorded as 0.161 as compared to 0.176 in case of VG-32 oil. Further increase in COF from this minimum value indicates the conversion of mixed lubrication regime to hydrodynamic lubrication regime. So this point at specific film thickness of 1.02 is the end point of mixed lubrication regime.

These ranges of mixed lubrication were used for further experimentation. Various tribological parameters (Frictional force, COF & Wear) were investigated for the above tribo pair in this mixed lubrication regime.

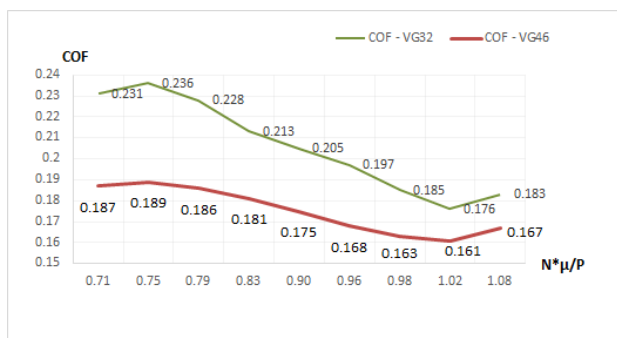


Fig. 3: Stribeck curve for VG-32 & VG-46 oils.

3.2 Frictional characteristics

The frictional characteristics comparison for the selected tribopair of EN-31 (HRC 60) & brass by using VG 46 & VG-32 lubricant is shown in Fig 4. The characteristics have been plotted for load of 120N and speed of 95 RPM & 108 RPM respectively.

It is evident from the figure that for both the lubricants, coefficient of friction increases abruptly for the first 10-20 sec. But the value of COF is 0.2 in case of VG-32 lubricant & 0.19 in case of VG-46 lubricant. It can also be perceived from the plot that continuous fluctuations in coefficient of friction are observed after 20s because of breakdown of lubricant leading to surface

to surface contact. It can also be inferred from the figure that the fluctuations are more in case of VG-32 lubricant as compare to VG-46 lubricant. This clearly shows that there would be high COF & its fluctuations for VG-32 lubricant compare to VG-46 lubricant during sliding of brass on EN31 (HRC 60) material.

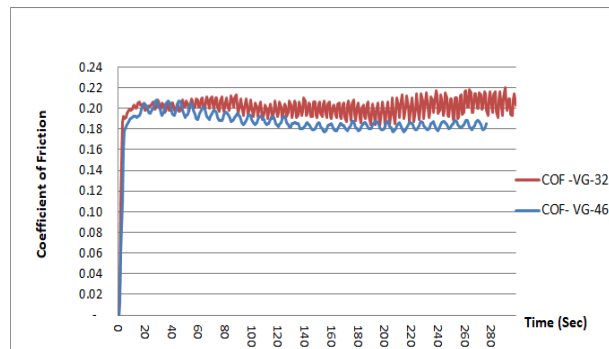


Fig. 4: Friction characteristics (COF) at load of 120N.

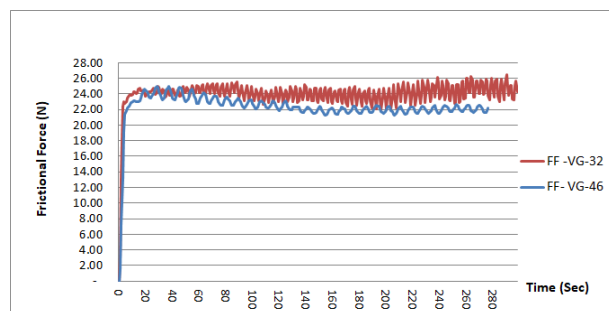


Fig. 5: Friction force behaviour at load of 120N.

Figure 5 illustrates the behavior of frictional force for the selected tribopair of EN-31 (HRC 60) & brass by using VG 46 & VG-32 lubricant. It can be inferred from the figure that frictional forces increase abruptly for the first 10s. Frictional force of 22N is observed for the VG-46 oil & 24N for VG 32 oil. Continuous fluctuations in the frictional force value are also observed for both the oils after 20-30s. This may be ascribed to the absence of the effective layer of the lubricant which has resulted to direct surface to surface contact. These fluctuations are more in case of VG-32 oil as compare to VG-46 oil.

The above behaviour is attributed to the enhanced adsorption ability of VG 46 oil on the said tribo pair in mixed lubrication regime as compare to VG 32 oil, which has resulted in better lubricating film formation between the sliding surfaces throughout the movement & hence lower value of coefficient of friction & frictional forces.

3.3 Effect of speed on coefficient of friction

Coefficient of friction was achieved for the EN-31 (HRC 60) base material with VG-32 & 46 lubricants for different speeds and constant load of 120N in mixed lubrication regime. (Fig 6). This is clearly evident from

the figure that coefficient of friction is maximum at sliding speed of 90 RPM & then it decreases with increase in sliding speed.

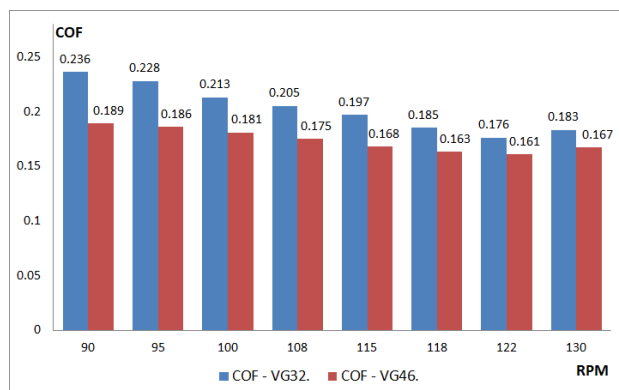


Fig. 6: Effect of speed on coefficient of friction.

This behaviour is because, at low speeds the thickness of the lubricant layer is comparatively less which has led to direct contact between surface asperities & hence abrupt increase in coefficient of friction. Increment in sliding speed will help in increase in thickness of lubricant between the surfaces and reduction in interactions between asperities & hence decrease in friction and hence COF.

Figure also shows that the COF for the VG-32 lubricant is higher as compare to VG-46 lubricant for a given speed & load. This confirms that VG-46 oil result in less friction & COF as compare to VG-32 oil in mixed lubrication regime. So use of VG-46 oil should be preferred for sliding contacts in mixed lubrication regime. This behaviour is accredited to higher viscosity of the VG-46 oil w.r.t. VG-32 oil, which will result in higher thickness of lubrication & hence lesser frictional & COF properties for VG-46 oil.

3.4 Wear behaviour

The wear behaviour comparison for the selected tribopair of EN-31 (HRC 60) and brass by using VG 46 & VG-32 lubricant is shown in Fig 7. The characteristics have been plotted for load of 120N and speed of 59.6 mm/s & 67.8 mm/s respectively.

It is evident from the figure that for both the pairs, wear rate increases abruptly initially for 10-15s & reached to the maximum value of the wear. This is because of very minimal film thickness is present between the sliding surfaces at the start of the experiment & sliding of the soft material brass on hardened surface with direct contact has resulted in immediate large volume wear on both the surfaces, majorly on the soft brass surface. Then the wear rate decreases gradually with time as most of the wear has already been happened & with increase in sliding speed, lubrication film thickness will also start increasing which has resulted in gradually decrease in wear rate.

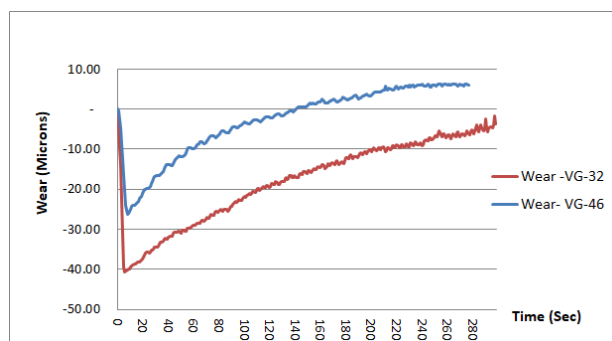


Fig. 7: Wear behaviour at the load of 120N.

It can also be seen from the plot that the wear rate & its fluctuations are more in case of VG-32 lubricant as compared to VG-46 lubricant because of its less viscosity & lower adsorption properties with the said tribo pair in mixed lubrication regime. Fig 8 shows the wear patterns on the worn out EN31 discs with VG-46 & VG 32 lubricants which was resulted from the experiments.

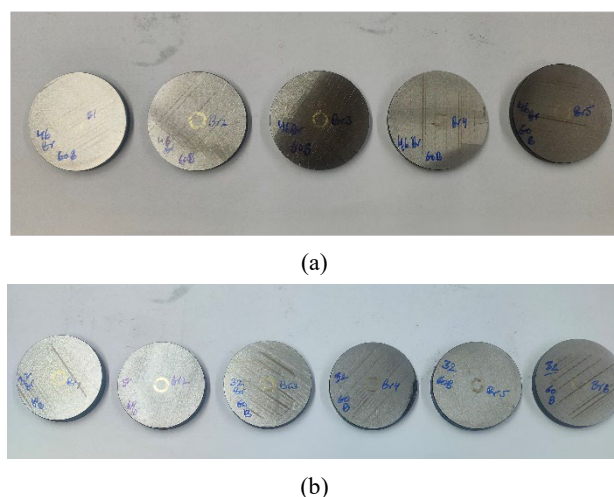


Fig. 8: Images of worn out EN31 specimens (a) VG-46 lubricant (b) VG-32 lubricant

4. Conclusions

The study was performed on the tribopair of EN 31 steel with varying lubricants i.e. VG-46 & VG-32 and brass as sliding material in mixed lubrication regime. This regime was attained by plotting the stribeck curve for the above lubricants. The major conclusions drawn from the study are given as below:

1. In mixed lubrication regime; frictional forces, coefficient of friction & their fluctuations with time are less in case of VG-46 lubricant as compared VG-32 lubricant. The improvement in these parameters is due to the higher adsorption ability of the VG-46 oil.
2. This is concluded from the study that the coefficient of friction decreases with increase in the sliding speed in mixed lubrication regime

for both the lubricants.

3. It is also determined from the study that the surface wear rate & its fluctuations w.r.t time in mixed lubrication regime are on higher side for VG-32 oil as compare to VG-46 oil.

Therefore, it can be concluded from the study that VG-46 lubricant will be having better surface adsorption & viscosity properties which has resulted in efficient film formation between the sliding surfaces which has resulted in less coefficient of friction, frictional force & wear rate as compared to VG-32 lubricant & are preferable over the same in mixed lubrication regimes.

Nomenclature

PCD	Pitch circle diameter
COF	Coefficient of friction
VG	Viscosity grade
N	Speed (RPM)
P	Load (kgf)

Greek symbols

μ	Dynamic viscosity of lubricant
λ	Hersey number or specific film thickness

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