

Effect of Lubricating Oil on Tribological behaviour in Pin on Disc Test Rig

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ABSTRACT

A test method has been developed to evaluate the friction and wear behaviour of cylinder liner and piston ring materials for four stroke engine system. Realistic engine oils are used to describe the behaviour of this test method. The friction and wear experiments were performed using pin-on-disc tribo tester. The effect of lubricants and load conditions are important aspects of this test method and are focus of this work. The test uses actual piston ring segments sliding on the disc of grey cast iron used in cylinder liners. A wide range of commercial lubricants including SAE10W30, SAE20W40 and SAE20W50 were used to analyze frictional and wear behaviour. Tests were conducted for constant load at 140 N for 105 min and increment load with the range from 20 N to 140 N for 105 min to evaluate the behaviour of frictional force and wear for cylinder liner and piston ring. Relative amount of wear is directly correlate with the effectiveness of the lubricant due to this wear was measured by weight loss before and after testing. Result shows that viscosity and variation of load plays a vital role to characterize the behaviour of frictional force and wear.

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1. INTRODUCTION

The automotive industry is facing tough international competition, government regulations and rapid technological changes. Every increasing government regulations require improved fuel economy and lower emissions from the automotive fuel and lubricant systems. Higher energy conserving engine oils and better fuel-efficient vehicles will become increasingly important in the face of the saving of natural resources and the lowering of engine friction. There are many hundreds of

tribological components, from bearings, pistons, transmissions and clutches to gears and drive-train components. The application of tribological principles is essential for the reliability of the motor vehicle and the energy conservation of our environment.

Engine lubricants or engine oils, are designed for use in internal combustion engines. Modern engines operate on a wide variety of fuels and in environments that involve temperature extremes; hence their lubrication is quite complex. A combustion engine lubricant must

possess attribute to help it perform the following functions effectively like permit easy sliding, maintain adequate viscosity at high temperature, lubricate and prevent wear, reduce friction, protect against rust and corrosion, keep engine parts clean, cool engine parts, seal combustion pressure, control foam. The quality of the lubricants used in the engine plays an important role in prolong the life and improve the performance of an engine [1].

Friction and wear are the two major factors which affect the life of engine. The energy derived from combustion of the fuel is distributed in an engine and a power-train system. Only 12 % of the available energy in the fuel is available to drive the wheels for a medium size passenger car during an urban cycle [2]. From this, 15 % being dissipated as mechanical, mainly frictional losses. Based on the fuel consumption data, a 10 % reduction in mechanical losses would lead to a 1.5 % reduction in fuel consumption. Friction losses the major portion (48 %) of the energy consumption developed in an engine [3]. From this friction loss, piston skirt friction, piston rings and bearings includes 66 % of the total friction loss and the valve train, crankshaft, transmission and gears are approximately 34 % [4]. From this analysis, we conclude that the piston rings and piston skirt against the cylinder wall is undoubtedly the largest contribution to friction in a power train system. From these frictional losses, we can analyze that the piston ring/engine cylinder bore system are important in achieving desired engine efficiency and durability in terms of power loss, fuel consumption, oil consumption and even harmful exhaust emissions. Due to the high frictional loss between the cylinder liner and piston ring, numbers of researchers have worked.

A Field test of any materials or lubricants in actual system is not preferable in the initial stage of the testing. High costing, complexity in operation and more accurate and precision instruments are required for the testing. More trial and error is required to determine the operating parameter, best suitable lubricants or material. The use of laboratory testing to simulate the engine environment offers many benefits like simplicity of operation, lowest testing cost, real time presentation of data to facilitate recognition of changing condition,

accurate and precise indication of wear rates, inexpensive and uniform consumable test pieces, small volumes of test fluid, controlled test environments and ambient condition, test pieces from a wide range of material and conditions [5]. H. Kaleli has developed the design of computer controlled tribometer for friction characterization of various friction sliding pairs at a laboratory scale [6].

It is extremely difficult even with the most well designed laboratory test to actually predict the performance in the engine since the operating environment is so complex chemically and mechanically. However, a well-designed laboratory test should be able to quickly rank various candidate materials or processes as to their relative performance in an operating engine. Material which have a long history of used can be used for validation purpose.

J.J. Truhan et al. [7] have studied on the effect of oil condition and its effect on the friction and wear of piston ring and cylinder materials in a reciprocating bench test. To investigate the friction and wear behaviour of ring and liner, realistic lubricants are used. Wear can be expressed in mass loss, volume loss or depth of wear. Wear depth is good for wear measurement with compare to other method. M. Zheng et al. [8] have studied on the model for wear and friction in cylinder liners and piston rings. The model can predict the effects of surface roughness, asperity contact and temperature-pressure-velocity on wear, lubrication and friction of the piston rings and cylinder liner. The major contribution for the cylinder liner wear is abrasion, in top portion during the break-in period. There was the engine speed and load increased step by step from a low level to the full load, full speed condition in 14 break-in-steps. It is observed that there is good agreement between the predicted cylinder bore wear and measurement bore wear. J.J. Truhan et al. [9] have studied on the laboratory test to evaluate piston ring and cylinder liner materials for their friction and wear behavior in realistic engine oils. Here wear test were carried out at 240 N for 6 h at 1000 C with new ring segments. A ring segment was tested against a flat specimen of gray cast iron typical of cylinder liner. Different lubricants like Jet A aviation fuel, mineral oil and a new and engine-aged, fully formulated 15W40 heavy duty oil were used to

evaluate the sensitivity of lubricant condition. Wear was measured by weight loss wear volume and wear depth using a geometric model. The result shows that Jet A has higher wear & used 15W40 oil showed least wear. S.S. Venetia et al. [10] has performed an experiment on pin and vee block test machine (Falex) to measure coefficient of friction. Three engine oils and three gear oils were tested on the same loading – time procedure to analyzed coefficient of friction according to SAE class of viscosity, Degree of use (Fresh or aged) and manufacturer. The variation of coefficient of friction with load is almost similar for fresh oil. Used engine oil has increased coefficient of friction at higher loads due to aging and contaminants. M.A. Chowdhury et al. [11] have investigated and compared friction coefficient and wear rate of different steel material and observed that wear rate increases with increase of load and sliding velocity and mild steel offers highest wear rate. V. Laxshminarayana et al. [12] have study the influence of varying load on EN6alloy steel when it is sliding against EN31 alloy steel by using pin on disc apparatus. They have investigated about the friction coefficient and the wear rate at different normal loads and high load. The result shows that the wear rate increases with respect to load. When the load slowly increases, gradually the friction coefficient gets decreases accordingly. After certain load, it dramatic increase and then after, it decreases very gradually. This indicates that at high load condition, the phase change takes place and the material becomes ductile. By compare the brass steel and copper steel pair, the friction coefficient of copper steel pairing is higher and the wear rate is lower in the investigated parameter range [13]. Friction and wear behaviour of bronze and brass which have not large difference in micro hardness, the severe plastic deformation is also observed on the microstructure of the brass [14].

M. Laad et al. [15] investigate the tribological behavior of titanium oxide (TiO₂) nanoparticles as additives in mineral based multigrade engine oil by using Pin-on-disc tribo tester. All tests were performed under variable load and varying concentration of nanoparticles in lubricating oil Servo 4T synth 10W30. The friction and wear behavior were analyzed through the experiment. The result shows that the tribological properties of lubricating oil were enhanced due to the

addition of TiO₂ nanoparticles. The reduction in COF was the maximum for the smallest load 4 kg and minimum for the largest load 6 kg. With the use of CuO nanoparticles added with mineral oil, there was a significant reduction in both coefficient frictions (28.5 % approx.) and specific wear rate (70 % approx.) [16]. A.N. Farhanah et al. [17] performed and experiment to investigate the performance of lubricants for an IC engine. Engine oil from three different manufacturers with the same SAE viscosity grade (SAE10W30) is used to compare the performance of lubricants. Experiments were performed by four ball wear tester for different temperatures (40 °C, 70 °C and 100 °C) and varied speed from 1000 rpm to 2500 rpm.

From the graph, it is observed that oil B and oil C were more viscous than oil A, so at the same speed and load, they were able to increase the fluid film thickness and create high separation distance result is decrease coefficient of friction.

The pin-on-disc tribometer used to conduct the experiment. The tribological parameters like friction, wear and load carrying capacity of the lubricants were experimentally evaluated on tribological laboratory test rig. This paper presents the results on experimental evaluation of tribological performance of the commercial engine oils. The overall engine performance has been linked with quality and selection of the lubricant.

2. EXPERIMENTAL SETUP

2.1 Experimental Apparatus

The friction and wear tests were carried out by using laboratory equipment named Pin-on-disc type Tribometer TR20LE. The Tribometer consisted of a driven spindle and chuck for holding the revolving disc, a lever-arm device to hold the pin, and attachments to allow the pin specimen to be forced against the revolving disc specimen with a controlled load. The wear track on the disc was a circle; the tribometer also had a friction force measuring system (a load cell) to determine the friction coefficient. In this study, wide range of lubricants was used for the purpose of determining the sensitivity of the test methods to lubricant condition. They were (1)

SAE 10W30 (2) SAE 20W40 and (3) SAE 20W50. All the lubricants are commercial lubricants and recommended by a manufacturer for two wheeler engine. Table 1 shows the oil properties of various lubricants.

Table 1. Oil Properties.

	Viscosity (cSt)		Flash Point	Viscosity Index
	40 °C	100 °C	°C min	
SAE10W30	119.75	11.51	228	102
SAE20W40	124	14.3	240	114
SAE20W50	156.3	17.87	256	126

2.2 Test specimen materials for cylinder liner and piston ring segment

The disc was prepared from cast iron, which is suitable material for the automotive engine parts such as liners. Table 2 shows an elemental analysis of the cast iron material used in this study.

Table 2. Material composition of prepared cylinder liner on X-met 5000-portable Analyzer.

Elements	Fe	C	Si	Mn
	Rest	3.16	2.05	0.67
Weight (%)	Cr	Cu	P	S
	0.2	0.27	0.21	0.062

After analyzing the composition and microstructure of liner material, cast iron discs are prepared by dry sand mould casting, and subsequently turned and grounded. Disc was of 79.8HRC hardness and Ra roughness of disc was

0.575±0.2 µm. The discs are made of 165 mm diameter and 8 mm thick.

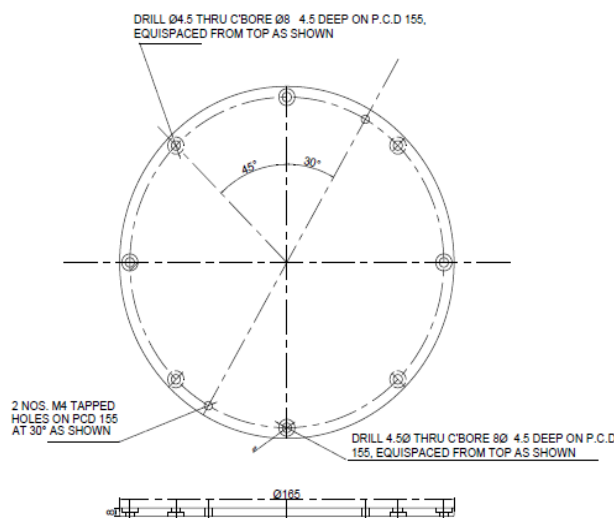


Fig. 1. Specification and photographic view of disc.

Table 3. Operating parameters for experiment on pin-on-disc test rig.

Test rig	Lubricant	Speed (rpm)	Track (mm)	velocity (m/s)	Time (min)	Cycles	Distance (m)	Load condition
Pin on disc	SAE 20W50 SAE 20W40 SAE 10W30	1800	60	5.652	15	27000	5086.8	Load: 20 N to 140 N Increment of load: +20 N
		1350	80	5.652	105	20250	5086.8	Load:140 N (constant)



Fig. 2. Piston ring, top pin and its mounting.

The piston ring segments used in this study were from an actual Honda engine with capacity of 100CC. Pin specimen were of Ra roughness of test surface was $2.75 \pm 0.05 \mu\text{m}$. The segment of ring is made to fix on the top of the standard pin with 10mm diameter. The ring segment with the pin is inserted in a holder, which is stationary. To fix position of ring segment properly in a pin, a taper groove is machined on the pin.

2.3 Experimental Procedure

Prior to each test, the pins and disks were cleaned by acetone to remove metal fragments and oil from the surface. The disc was inserted carefully in the holding device so that it remained perpendicular to the axis of the resolution. The pin specimen was inserted in the holder and adjusted to make it perpendicular to the disc surface when in contact to maintain the necessary contact conditions. Proper mass was added to the system lever to the selected force pressing the pin against the disc. The electric motor was started and the speed of the disc was adjusted to the desired value.

Lubricant was applied between the pin and disc to satisfy lubrication conditions. Frictional force was measured from the controller and weight loss of the pin was measured using electronic weighing balance (accuracy of 0.0001 mg). The above procedure was repeated for all the tests. The Win Ducom software was used for data acquisition and display of results. The Win Ducom instrumentation and data acquisition were used to measure RPM, wear, and frictional force.

The entire tests have been performed under wet condition by using various lubricants. Increment load tests as well as constant load tests have

been performed, with the objective to establish a relation between them for a given cylinder liner-piston ring combination. Increment load tests were performed with an initial load of 20 N which increased with 20 N after every 15 min until a load of 140 N was reached (at 105 min). Constant load tests were performed with a load of 140 N for fixed 105 min. Both friction and wear were studied for all the test lubricants.

3. RESULTS AND DISCUSSIONS

3.1 Frictional Behaviour

A series of experiments were conducted to evaluate the friction characteristics of sliding elements using pin-on-disc tribometer applying the lubricant at the interface for a sliding distance 5067 m with various lubricating oil.

Figure 3 summarizes the effect of 140N constant load for various lubricants on the friction behaviour of sliding pair of cylinder liner and piston ring with respect to time. It is observed from the figure that the nature of the curve is similar for all lubricants. During the initial run for constant load, there is significant variation in frictional force up 35 min and then follow steady state up to the end of the experiment. The reason is initial roughness of the sliding pair hence high asperity contact generates which results high frictional force.

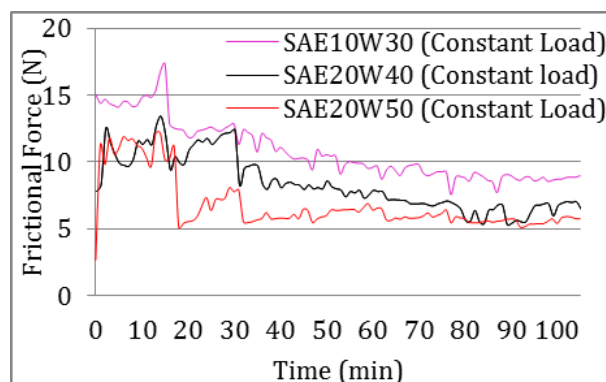


Fig. 3. Frictional force variations with time for constant load.

During the experiment, the gradual degradation of the surface and oil added are the only source for the increase in friction force. The oil film formation may not develop sufficient between the components during the initial run which results high frictional force. Further increase of

time, constant fluid film formed and mixed lubrication regime developed between the pair hence partial load carrying capacity increase up to certain extent which results frictional force is nearly constant.

By comparing the three lubricants, SAE10W30 offer the lowest viscosity produced the highest frictional force. This is due to insufficient lubricant film formed between the pair due to thinner oil. SAE10W30 offered 27.81 % and 35.81 % high frictional force as compare to the SAE20W40 and SAE20W50 respectively.

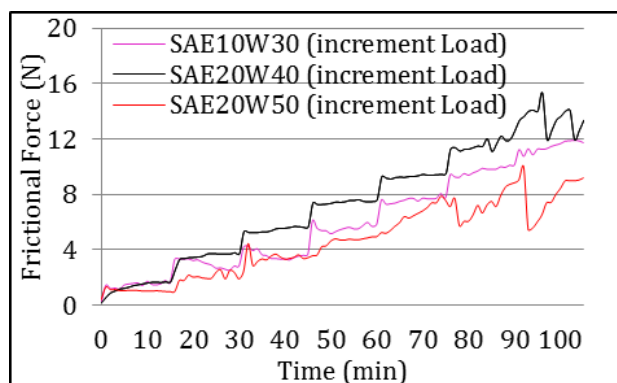


Fig. 4. Frictional force variation with time for increment load.

Figure 4 shows the variation in frictional force with time for different lubricants with increment load. With the increment of 20 N load step wise after every 15 min, the trend of the frictional force is positive and it increases throughout the experiments. It is observed from the graph that after every 15 min frictional force is suddenly increases and be a stable for a certain time. When the oil is added between the specimens and load is applied between the specimen after every 15 min for lubrication purpose, result is increased frictional force. This may be due to the rupture of the fluid film formation between the components.

Further increase of time from 60min to 70min, the rate of increment of frictional force is observed low. This may be indicate that the asperity contact between the pin and disc are reduced after some time due to the effect of wear and surface be a smoother so it offers minimum frictional force.

SAE10W30 offered 14.14 % and 21.63 % high frictional force as compare to SAE20W40 and SAE20W50 respectively.

SAE20W50 is more viscous, so at the same speed and load, lubricants are able to make a fluid film and create high separation distance between the two surfaces. Hence, it prevents the surface contact and able to reduce the frictional force.

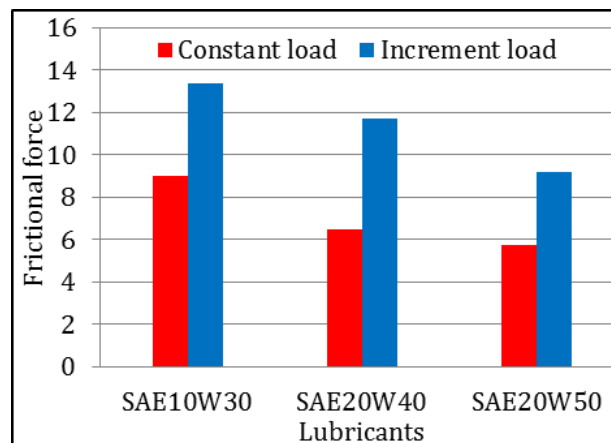


Fig. 5. Variation in frictional force for constant load.

Figure 5 summarizes the effect of constant load and increment load on the frictional force for various lubricants. Increment load shows the high frictional force for all the lubricants. Due to the increment load, fluid film is rupture which results the high frictional force. Frictional force is increased by about 32.91 %, 44.72 % and 37.28 % for SAE10W30, SAE20W40 and SAE20W50 respectively. SAE10W30 offer the maximum frictional force for constant load and increment load condition both. It shows that the viscosity plays a vital role for lubrication purpose. SAE20W40 and SAE20W50 showing high viscosity due to this it can be capable to form the fluid film between the liner and piston ring. That fluid film can resist the load and the result is reduced frictional force.

3.2 Wear Behaviour

Wear tests were conducted with the same lubricants which are used for the friction testing and the procedure is also same. Figure 6 shows the variation in wear with the applied load.

For constant load, wear increases with time and be stable or follow slightly negative trend with respect to time. During the initial run condition, surface irregularities of the specimen plays a major role to increase the wear. Surface irregularities increase the real contact area between the sliding surfaces which cause maximum wear. After travelling some distance,

surface of the specimen becomes smooth and the stable fluid film can be generate, result is reduced wear. When lubricant is applied between the surfaces, some pressure will generate during running condition which will help the surface apart and prevent surface contact. Hence, for a constant load, fluid film will resist the load and reduce the wear.

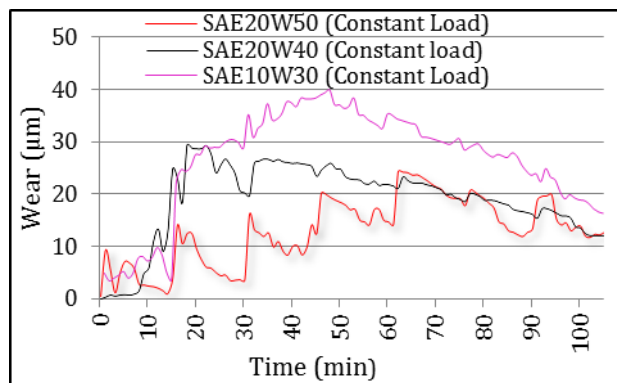


Fig. 6. Wear variation with time for constant load.

Wear of various lubricants increases at the initial stage but SAE10W30 offers the maximum wear. After surface become uniform, it will follow negative trend for all lubricants. SAE10W30 has a lower viscosity as compare to other lubricants, so viscosity of the oil will influence the wear to increase because it might be under boundary condition. SAE10W30 oil may contain les antioxidant additives where antioxidant additive will help to form a surface film to reduce metal to metal contact and hence reduce wear. SAE20W40 and SAE20W50 lubricants offers minimum wear because of mixed lubrication regime development due to its high viscosity. It has good anti-wear ability compare to other lubricants. It keeps the oil film uniform in order to avoid surface contact. Hence viscosity plays a vital role to form a film and reduce the wear between the sliding surfaces.

Figure 7 shows the increment load condition and here we have observed that with the increment of load; wear exactly follow the same pattern. With the increase of load, wear increases and try to be stable within that condition. In the initial phase, load is carried by the surface asperities rather than by the lubricant film. Lubricant film is not formed properly indicate the boundary film lubrication. The properties of the two mating surfaces play a far more significant role than the lubricant which result is increased wear. During running condition, film will formed

between the specimens but when 20N load applied suddenly, it will rupture and increase the rate of wear. Initially the rate of increment of wear is more with load but after 60min to 70min, it gradually decreases. This may be due to the surface roughness of the mating surfaces. Initially more asperity contact are caused between the surfaces but during running condition, surface becomes smooth and both surfaces makes close contact with each other result is increment of wear is gradually decreases.

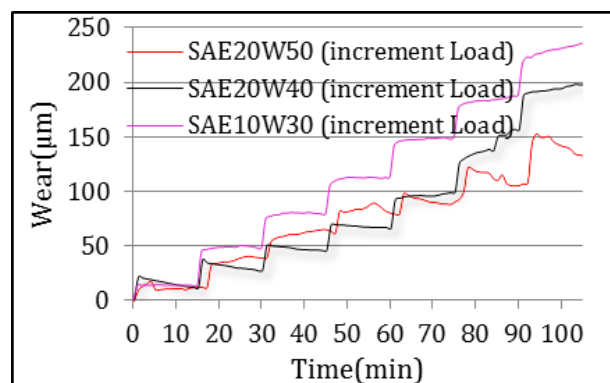


Fig. 7. Wear variations with time for increment load.

All lubricants follow the same pattern but SAE10W30 offers the maximum wear as compare to SAE20W40 and SAE20W50. SAE20W50 offers minimum wear because of mixed lubrication regime development due to its high viscosity. It has good anti-wear ability compare to other lubricants. It keeps the oil film uniform in order to avoid surface contact.

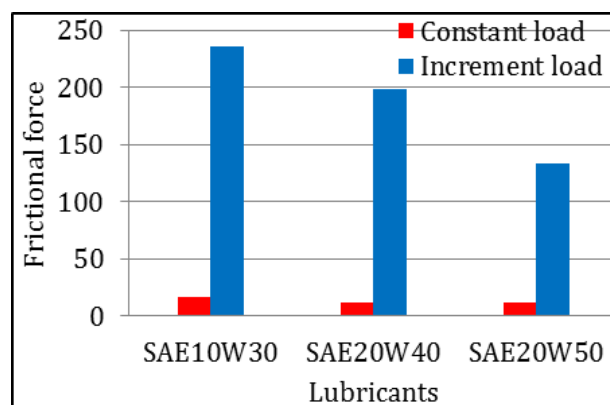


Fig. 8. Variation in wear for constant load and increment load.

Figure 8 summarizes the effect of constant load and increment load on the wear. The figure shows high value of wear with the increment load condition. In the increment load, fluid film cannot

resist the suddenly applied load, the result is that the fluid film rupture and increase the wear. It was observed that the increment load offers 93.09 %, 93.91 % and 90.77 % more wear as compare to constant load for SAE10W30, SAE20W40 and SAE20W50 lubricants.

3.3 Rate of Increment of wear with increment load for various lubricants

Figure 9 shows the increment load effects on wear for various lubricants. Increment load tests were performed with an initial load of 20 N which increased with 20 N after every 15 min until a load of 140 N was reached (at 105 min).

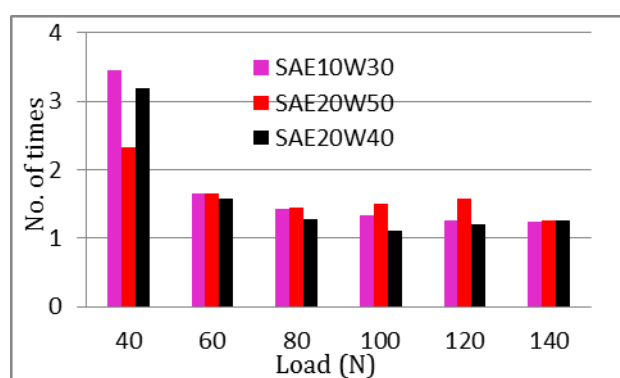


Fig. 9. Increment of wear with increment load for various lubricants.

From the graph, it is observed that after 20 N load, Lubricants offers 27.78, 16.76 and 41.43 times more wear as compare to the initial load (0 N). But after 4 N, the rate of increment of wear is significantly low and the range is about 2.1 to 3.5 times more. During the starting period of the experiment, instrument requires some time to be stable after load applied and it also takes some time to form the fluid film between the sliding surfaces. Hence, it indicates boundary lubrication during the initial period which shows significantly high wear. After increment of 4 kg load, the range of wear increment from about 1.1 to 1.6 times more as compare to the previous load condition.

During the starting period, the liner and piston ring have some initial roughness hence surface irregularities cause between the contact areas. Due to the roughness, asperity contact occurs and large actual contact area would be generated. When the rubbing process start, topography geometry changes to a great extent. Mainly in the severe loading tests, it was noticed that the

surface roughness had completely change the orientation of the grooves after sliding test. It became smooth and gradually decreases the wear increment with increase of load.

3.4 Weight loss of cylinder liner and piston ring

Wear can be expressed as mass loss in gram. To measure the weight of the specimen, new ring and disc were used for each test in order to measure the loss of weight after every experiment. The relative amount of the wear correlates with the effectiveness of the lubricant.

Figure 10 shows the comparison of the weight loss of cylinder liner and piston ring for constant load and increment load for various lubricants. Increment load offers more weight loss as compare to constant load for all lubricants. For a piston ring, it offers 32.97 %, 21.54 % and 23.73 % and for cylinder liner, it offers 33.45 %, 24.32 % and 16.13 % higher weight loss as compare to constant load for SAE10W30, SAE20W40 and SAE20W50 respectively.

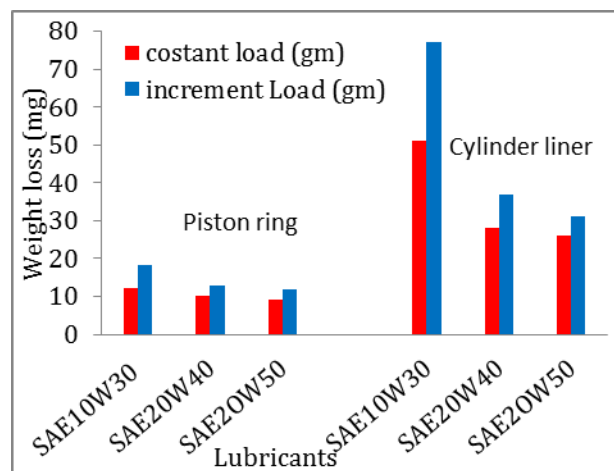


Fig. 10. Comparison of the piston Ring and cylinder liner weight loss for increment load and constant load.

Fluid film is formed between the cylinder liner and piston ring during running condition. In increment load, load is applied after every 15min, so fluid film may be disturbed and increases the wear as compare to constant load condition. SAE10W30 oil shows highest weight loss as compare to SAE20W40 and SAE20W50.

Figure 11 shows the weight loss of cylinder liner and piston ring. The cylinder liner disc showed two to five times' greater weight loss as compare

to piston ring. In an operating engine, this relationship is reversed. [7,19].

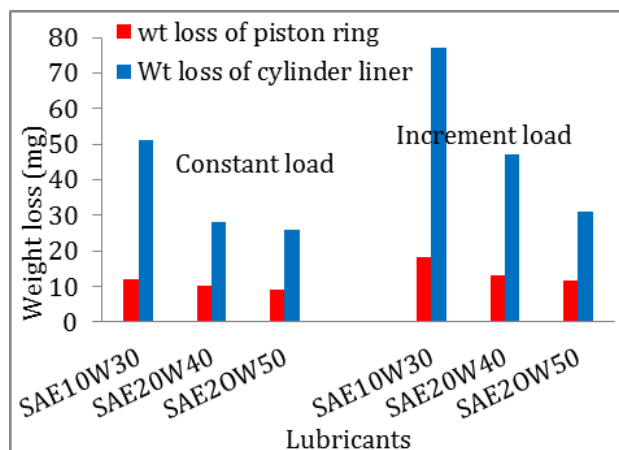


Fig. 11. Comparison of the weight loss due to constant load and incremental load for cylinder liner and piston ring.

For a constant load, cylinder liner offers 76 %, 65.57 % and 65.38 % while for incremental load it offers 76.36 %, 72.34 % and 61.93 % higher weight loss as compare to piston ring.

Table 4. Weight loss with SAE10W30 lubricants.

Lubricants	% Reduction for Constant load		% Reduction for Incremental load	
	Piston ring	Cylinder liner	Piston ring	Cylinder liner
SAE20W40	16.39 %	45.35 %	28.57 %	38.96 %
SAE20W50	26.23 %	49.26 %	35.16 %	59.74 %

SAE10W30 is the low viscous oil so it cannot be capable to form a lubricated film between the specimens. For incremental load, the weight loss is higher due to the rupture of fluid film after every 15 min when load is applied.

In an actual engine, the piston ring experiences the more wear as compare to cylinder liner but in test rig, it's vice versa. This is due to the dissimilarities in geometry between the rig test and actual engine.

From the four strokes, during the compression and combustion time of the fuel and air, the ring and liner experiences the highest loading near the TDC. As the piston is lowered means near the BDC, the load decreases and wear is reduced [20]. Only the small element of the liner near this top ring position will experience enhanced. In the test rig, constant contact found between the disc and the pin, which produce constant

stress exists between the two mating surfaces. So instead of cyclic loading, load is constant throughout the operation which results high wear generate in test rig.

5. CONCLUSION

By comparing the constant and incremental load for various lubricants, it is clear that increment load offers friction and wear. The cylinder liner and piston ring shows high weight loss with incremental load.

By comparing the tribological properties of these lubricants, it is cleared that SAE20W40 and SAE20W50 have better performance compare to SAE10W30.

Both the oils are more viscous and may contain more antioxidant additive. Antioxidant additive will help to form a surface film to reduce metal-to-metal contact and hence reduce wear and frictional force. Antioxidant additive used to slow down the rate of oxidation where it plays their role for stabilization purpose and enhance their performance of lubricant

It is observed that, SAE20W50 gives better performance than the SAE20W40. But manual of two wheeler recommends SAE20W40 because SAE20W50 have more viscosity improver, which will cause more sludge than the SAE20W40. In winter, oil flow problem may occur which effect some cold starting issues may occur due to their high viscosity. This is because today's engines are built with tighter bearing clearances to take advantage of the fuel economy benefits of lower viscosity oils. It is not really a good idea to use thicker oil in one of these engines because it will disrupt the oil flow characteristics of the engine and may create excessively high oil pressure.

REFERENCES

- [1] C.M. Taylor, "Automobile Engine Tribology- Design consideration for Efficiency & Durability", *Wear*, vol. 221, no. 1, pp. 1-8, 1998.
- [2] B.S. Andersson, 'Company perspectives in vehicle tribology—Volvo', In: *17th Leeds-Lyon Symposium on Tribology-Vehicle Tribology*, Tribology series, 18. Oxford, UK: Elsevier Ltd; 1991, pp. 503-6.

- [3] M. Nakasa, 'Engine friction overview', in: *Proceedings of International Tribology Conference*, 29 Oct - 2 Nov 1995, vol. 3, Yokohama, Japan.
- [4] J.A. Spearot, 'Friction, wear, health, and environmental impacts—tribology in the new millennium', A key note lecture at the STLE Annual Meeting, Nashville, Tennessee, May, 2000.
- [5] M.J. Neale and M. Gee, 'Guide to Wear Problems and Testing for Industry', ISBN No 0-8155-1471-9, Williams Andrew Publishing, Norwich, NY, 2001.
- [6] H. Kaleli, 'New Universal Tribometer as Pin or Ball-on-Disc and Reciprocating Pin-on-Plate Types', *Tribology in Industry*, vol. 38, no. 2, pp. 235-240, 2016.
- [7] J.J. Truhan, J. Qu and P.J. Blau, 'The effect of lubricating oil condition on the friction and wear of piston ring and cylinder liner materials in a reciprocating bench test', *Wear*, vol. 259, no.7, pp. 1048-1055, 2005.
- [8] M. Zheng, A. Naeim, Henein and B. Walter, 'A model for wear and friction in cylinder liners and piston rings', *Tribology Transaction*, vol. 49, no. 3, pp.315-327, 2006.
- [9] J.J. Truhan, J. Qu and P.J. Blau, 'A rig test to measure friction and wear of heavy duty diesel engine piston rings and cylinder liners using realistic lubricants', *Tribology International*, vol. 38, no. 3, pp. 211-218, 2005.
- [10] S.S. Venetia, 'measurement of coefficients of friction of Automotive lubricants in pin and vee block test machine', in: *5th International Conference, "Computational Mechanics and Virtual Engineering"*, COMEC 2013, Brasov, Romania, 24- 25 October 2013.
- [11] M.A. Chowdhury, 'Experimental Investigation on Friction and Wear Properties of Different Steel Materials', *Tribology in Industry*, vol. 35, no. 1, pp. 42-50, 2013.
- [12] V. Laxminarayana and V.P. Balu, 'Influence of varying load on wear rate and frictional Resistance of en-8 steel sliding against en-31 steel', *ARPN Journal of Engineering and Applied Sciences*, vol. 10, no. 13, pp. 5655-5658, 2015.
- [13] V. Krasnik and J. Schlattmann, 'Experimental Investigation of the Friction and Wear Behaviour with an Adapted Ball-On-Prism Test Setup' *Tribology in Industry*, vol. 37, no. 3, pp. 291-298, 2015.
- [14] S. Senhadji, F. Belarifi and F. Robbe-Valloire, 'Experimental Investigation of Friction Coefficient and Wear Rate of Brass and Bronze under Lubrication Conditions', *Tribology in Industry*, vol. 38, no. 1, pp. 102-107, 2016.
- [15] M. Laad and V.K.S. Jatti, 'Titanium oxide nanoparticles as additives in engine oil', *Journal of King Saud University - Engineering Sciences*, <http://dx.doi.org/10.1016/j.jksues.2016.01.008>, 2016 (In Press).
- [16] S. Bhaumik and S.D. Pathak, 'Analysis of Anti-Wear Properties of CuO Nano-particles as Friction Modifiers in Mineral Oil (460cSt Viscosity) using Pin-On-Disk Tribometer', *Tribology in Industry*, vol. 37, no. 2, pp. 196-203, 2015.
- [17] A.N. Farhanah and M.Z. Bahak, 'Engine oil wear resistance', *Jurnal Tribology (MYTRIBOS)*, vol. 4, pp. 10-20, 2015.
- [18] G.D. Thakre and M.R. Tyagi, 'Study on variation or tribo performance of commercial engine oils', *Industrial lubrication and Tribology*, vol. 65, no.5, pp. 341-347, 2013.
- [19] B. Besergil, S.M. Ozer and M. Eskici, 'Study on lubricant quality and its effect on engine component performance', *Industrial Lubrication and Tribology*, vol. 60, no. 4, pp. 172-177, 2008.
- [20] R.A. Mufti and M. Priest, 'Effect of Engine Operating Conditions and Lubricant Rheology on the Distribution of Losses in an Internal Combustion Engine', *ASME Journal of Tribology*, vol. 127, no. 4, pp. 826-836, 2009.
- [21] E. Tomanik, 'Friction and Wear Bench Tests of Different Engine Liner Surface Finishes', *Tribology International Journal*, vol. 41, no. 11, pp. 1032-1038, 2008.