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Experimental Investigations on the Effect of Liner Surface Properties on Wear in Non-Firing Engine Simulator

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ABSTRACT

Several experimental studies have been conducted for evaluating coefficient of friction and wear in simulated engine conditions using a piston ring segment and a liner piece rubbing against each other in reciprocating mode under load and lubricated conditions. In the present experimental investigation, a non-firing engine simulator has been developed in order to simulate engine conditions to a much closer extent. This machine can operate at similar linear speed, stroke, and load and can simulate almost similar engine operating conditions except firing pressures. This machine can also be used for comparing liners with different surface properties and the effects of surface texture on wear and oil consumption.

One cylinder liner has been used for experimentation and the wear and surface properties behaviour were evaluated at several locations in the liner. Surface profile, roughness parameters are evaluated at several locations in the liner and at the top compression ring. Scanning electron micrographs are also prepared at these locations for comparative wear studies.

INTRODUCTION

Ever increasing pressure on emission reduction calls for reduction of oil consumption, lower coefficient of friction and lower wear in internal combustion engines. Cylinder liners require some of the most critical surface properties in terms of functionality. Piston ring and cylinder liner wear is a very important factor in determining effective engine life. A polished liner will not be able to retain oil and will have poor tolerance for wear debris. Poor lubrication gives rise to metal-to-metal contact between cylinder liner and piston rings, and lead to exceptionally higher wear and scuffing. Rough liners have very high coefficient of friction and high rates of wear in spite of good oil retention capacity hence a balance between the two is required in order to achieve lower friction, wear and higher engine life. In an engine, particulate matter (PM) generally results from incomplete evaporation and burning of the fuel droplets and lubricating oil. Researchers continue to explore means of reducing the consumption of lubricating oil, which would result in a lower lube oil contribution to PM emissions. There is a trade-off between reducing oil consumption and engine durability. Since durability is a very high priority in automotive engines, lube oil consumption can't be reduced at the expense of engine durability.

Only 25-35% of the input fuel energy is transferred to the transmission system as useful work. One important factor is the mechanical power loss, which contributes to approximately 15% of the total energy losses. Approximately half of this loss is because of friction at the liner - piston ring interface. Piston ring assembly plays a very important role to provide a dynamic seal between combustion chamber and crankcase. This sealing minimizes the expansion stage power loss due to pressure loss from the combustion chamber to the crankcase. Wear between piston rings and cylinder liner has to be minimized in order to ensure lower pressure loss from assembly over longer duration.

The components that are exposed to continuous friction are compression rings, oil control rings, piston skirt, and piston pin. Oil rings, due to their substantially higher ring tension, operate under boundary-lubricated conditions. They contribute as much as twice the friction of each compression ring [1]. The major design factors, which influence piston assembly friction, are; ring width, ring face profile, ring tension, ring gap, liner temperature, skirt geometry, and skirt- bore clearance [2].

There are a host of forces applied on a piston ring in an engine. These forces include several axial and radial forces. Axial forces acting on a ring include Mass force, Frictional force acting between liner and ring, Axial gas force, Hydrodynamic damping force due to oil filling of the groove, bending force due to thrust side and anti-thrust side etc. Radial forces include force due to residual stresses in the ring, gas forces acting on the rear side of the ring, frictional force between ring and ring groove, forces due to hydrodynamic pressure including radial damping force etc. All of these forces are shown in a schematic diagram in figure 1.

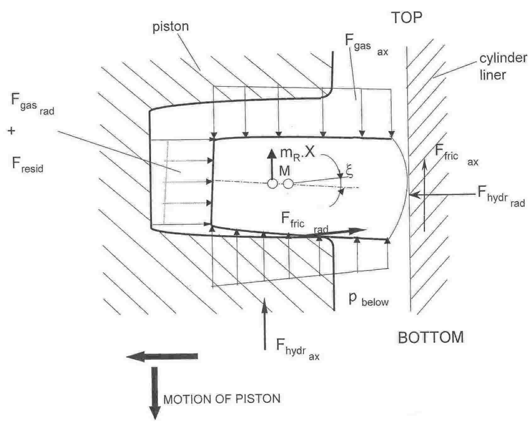

Figure 1: Forces Acting Upon a Ring

The study of the tribological properties of the cylinder liner and piston ring system in an internal combustion engine has attracted much attention for several years. Since most of the frictional losses in engines occur at the liner-ring interface and wear near the top dead centre is often a limiting factor to the life span of an engine.

The cylinder bore honing quality is an essential factor for a good engine performance and durability. A bad surface finish can result in an excessive lubricating oil consumption, high piston ring wear, and scuffing occurrence. Honing angle, which is determined by the vertical and rotational movement of honing head, is directly related to oil consumption. Lubricating oil consumption decreases with increasing honing angle as shown in table 1 [3].

Table 1: Effect of Honing Angle on Oil Consumption [3]

Honing Angle	Oil Consumption (g/kWh)
23^0	0.58
70^0	0.59
120^0	0.37

HISTORICAL PERSPECTIVE

The cylinder walls are stressed mechanically by high gas pressure and side thrust of the piston, as well as thermally due the high gas temperatures. Since all these stress-induced factors are cyclic in nature, the cylinder liner materials must have good mechanical and fatigue strength, otherwise cylinder bore distortion or early material fatigue failure may take place. Liner assembly stresses are also very high and should not be ignored. They are greater than the firing stresses and the stress due to piston slap.

In addition, the tribological properties such as wear and scuff resistance must also be satisfactory because metal-to metal contact between the piston rings and the cylinder liner do occur. However, all these desirable properties cannot be found in a single material. “Trade-off” between the mechanical and tribological properties must be considered during the selection of the appropriate liner material based on the application requirements. Grey cast iron is widely used as liner material for heavy-duty diesel engines. Three methods to improve the wear resistance of grey cast iron liner are: adding special alloy elements, using surface treatment technique such as induction hardening, gas nitriding, and applying surface coatings [4]. Induction hardened liner and special alloy grey cast-iron liners offer better wear resistance with new and used oil as compared to grey cast iron liners.

The preparation of the surface of cylinder bore is a multi-stage process. Surfaces are typically machined in two steps. First, a rough honing gives the right cylindericity, and engraves deep valleys on the surface (up to 10µm deep). Second, a finish-honing step, also called plateau honing, gives a relatively smooth surface to the plateaux [5]. After running in an engine for a relatively short time, a normally honed liner will exhibit a surface profile similar to a plateau honed liner. However, the large number of wear debris generated during the running-in period may damage the engine severely as they act as abrasive particles and get embedded in the liner surface. Plateau honed surface is relatively stable in terms of wear. Therefore, it is thought that initial wear can be controlled by building it into the liner during the manufacturing process thereby relieving the engine of the burden of large initial wear and its associated debris. During the later part of life of the liner, the plateau surface would continue to possess relatively large, smooth plateau, which provide a large bearing area, deep valleys to retain oil for lubrication between the surfaces and provide a relief area for wear particles. Tim Hegemir et al. [6] examined the effect of plateau honing on both rough and smooth liners and found that smooth plateau liner ($R_a \approx .78\mu m$) offers the best finish with regard to oil consumption, ring wear, liner wear between the ring turnarounds, and volume of liner material lost due to wear.

The wear of liner is different at thrust side and anti thrust side and also it is different over the stroke length. This can be seen in an exaggerated view of liner shown in figure 2. Higher wear takes place at TDC in a cylinder liner and there is measurable wear all over the stroke length [7,8]. As can be seen from figure 2, higher wear takes place on the anti-thrust side compared to the thrust side [9].

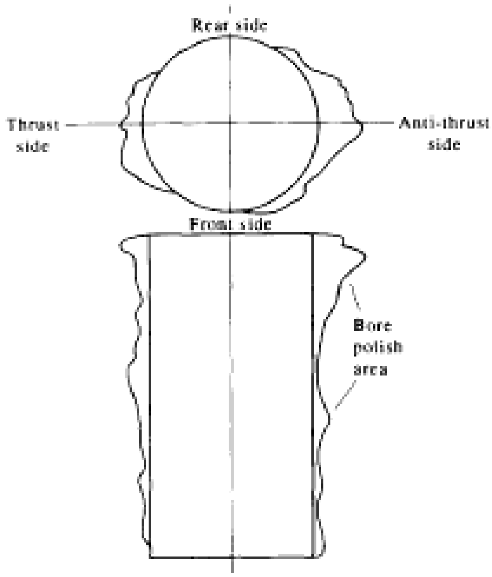

Figure 2: Exaggerated Diagram of Typical Cylinder Liner Wear [9]

The piston rings are main cause of wear rather than piston skirt. At TDC, presence of boundary lubrication regime is a major contributor to wear along with other factors, which include: highest contact pressures, thinnest lubricating films, high temperatures, and greatest concentration of acids. The film thickness at the TDC was found to be very small, in the range of 0-30µm, because of the low sliding velocity and high temperatures at this location. The maximum oil film thickness was found at the mid-stroke region because of hydro-dynamic lubrication regime and maximum piston speed. Thus piston rings move in both hydrodynamic and boundary lubrication regimes. Oil cleanliness has a significant effect on ring and liner wear. Used oils contain wear debris and oil degradation products, which may act as third body wear particles between the ring-liner interface, thus accelerating the wear of both ring and liner surfaces. Third body wear particles and external contaminants, if any, can adhere to, or embed themselves into one material, and may cause grooves in the counter surface that can eventually result in progressive loss of material.

The bearing length curve (Abbot's curve) is a very important parameter of the roughness profile. It is an important element in the evaluation of the actual contact area, because it defines an approximate value of the actual area between the specific rough surface and the ideal un-deformed surface, depending on their proximity [10]. Cylinder bore surfaces, which show proper shape of Abbot's curve and optimum surface roughness amplitude parameters ensure rapid running-in and significantly lower wear of the engine, as well as improved engine performance and lower oil consumption. Some roughness parameters can also be found out from Abbot's curve. The points of maximum and minimum curvature are the limits of the characteristic regions of the curve, the region of individual peaks, the rough core region, and the region of deep scratches. At the flex point, the minimum slope of the curve over the whole region can be determined, which is very important with respect to friction and wear.

L.L. Ting [11] measured coefficient of friction, friction force, and power loss in the piston ring-liner assembly by designing a slider-crank mechanism test rig. Friction forces and friction coefficient rise sharply and instantly to a larger value corresponding to zero ring velocity. As the piston ring begins to slide with increasing velocity, friction forces falls quickly and becomes rather small in the mid stroke region. At the end of the stroke, friction rises until another large friction force value is reached. Power loss is the product of friction force and piston velocity. It is interesting to note that the downward stroke power loss is always greater than that of the upward stroke. Although friction forces developed near the ring reversal regions are high, they do not contribute significantly towards the total ring friction power losses because of low sliding velocities in that region.

The objective of the paper is to fabricate a machine that closely approximates the wear process that occurs between the cylinder liner and piston ring inside an engine. This machine will provide us assistance in having a better understanding of the wear process and will also enable us to make a proper choice of the materials for different components of the engine.

RECIPROCATING WEAR SIMULATOR

A reciprocating wear simulator was designed and developed at IIT Kanpur. The main objective of developing this simulator is to investigate the effect of various surface properties of liner and ring on the frictional forces and lubricating oil consumption in an engine. A single cylinder engine was converted into a machine, which can simulate exact engine operating conditions except firing pressures. Actual production grade piston, rings and liner can be used in this simulator in order to approximate the engine conditions to a much closer extent compared to reciprocating wear tester. The advantage of this simulator was also that it was exposed to full stroke length rather than a short stroke as in the case of reciprocating wear tester. Figure 3 represents the reciprocating wear simulator.

The simulator was driven by 1 horsepower electric motor. The piston used in the simulator has five rings (two compression rings, one scrapper ring and two oil rings).

The loading on the rings can be changed by a suitable loading arrangement provided in the simulator as shown in figure 4. Currently, testing can be done at room temperature at variable load. However provision can be made for heating the liner to the temperatures close to actual firing engine. The simulator is lubricated at several locations using gear-pump based lubrication system. The oil used for this purpose is SAE 20 W 40. The stroke length of the simulator is 82 mm, and an oscillating frequency is 25 Hz. The forces that piston rings experience are a combination several axial and radial forces including mass force, frictional force at ring-liner interface, hydrodynamic damping force due to oil filling of the grooves, bending forces due to thrust side and anti-thrust side, forces due to ring residual stresses, friction force at ring-groove interface, and force due to hydrodynamic pressure including radial damping force.

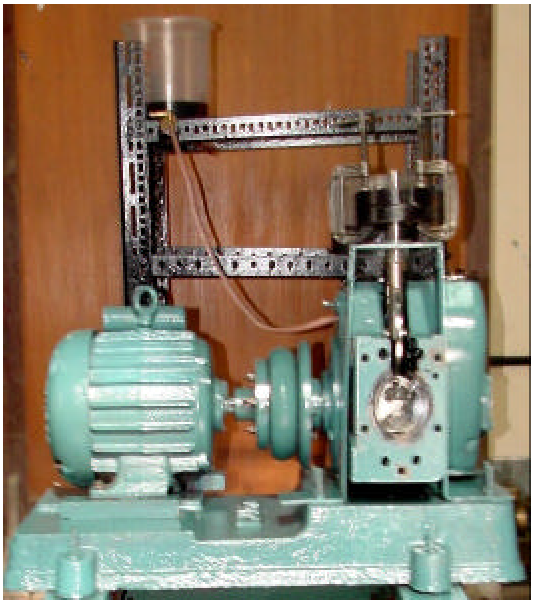

Figure 3: Reciprocating Wear Simulator

Since, this reciprocating machine is a non-firing machine hence additional loading to compensate for gas forces is done using a special loading mechanism to provide additional load in radial direction. Load is given on the piston, as shown in figure 4. The liner diameter is 82 mm. The liners need to be cut in order to make provisions for loading of the rings. Hence a 40 mm wide cut is made on the liner along its length so that load can be transferred directly on the piston and in turn on to the rings. A suitable groove is also machined on the piston and a wheel is provided so that the load can be transferred even when the piston is reciprocating. The liner used in the simulator is shown in figure 5. Provisions are made to stop the circumferential movement of the rings in the piston grooves by providing pins in suitable locations.

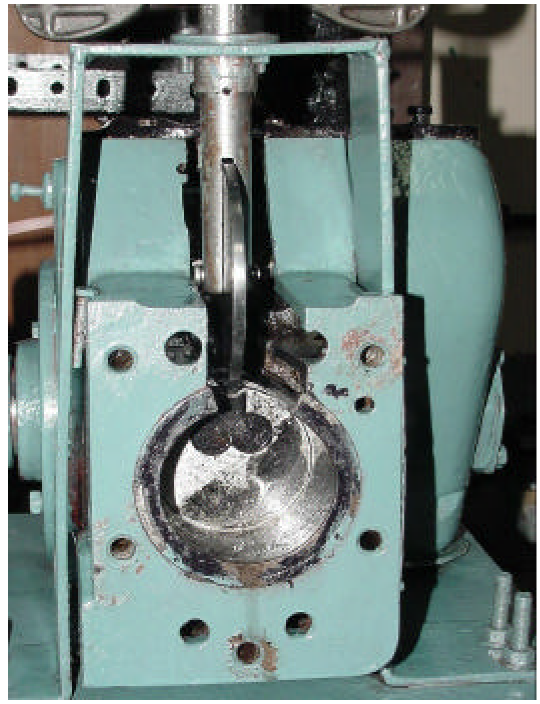

Figure 4: Loading Mechanism on the Piston

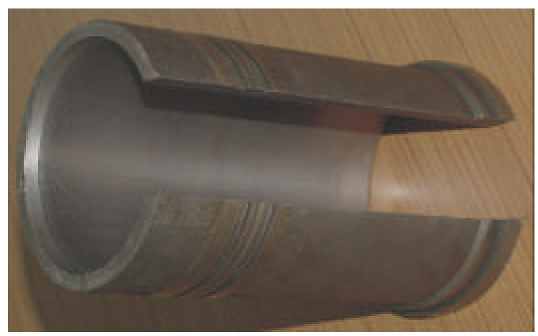

Figure 5: Design of Liner

Experimental Matrix

The grey cast iron cylinder liner surface was prepared by honing. Honing was done using honing sticks of grain size 150 and the corresponding material removal was about 0.06mm from the diameter. The finish honing of the liner was done using honing rods of grain size 280, which corresponds to a material removal of approx 0.02mm from the diameter.

The liner was then suitable machined for removing a 40 mm wide segment along its length and then it was installed in the simulator. A new set of piston rings was installed before starting this experimental investigation. The simulator was then assembled and lubricated before executing the test. The simulator was run at 1500 RPM for

30 Hours with a constant load 60N. The rings were weighed initially and they were weighed after every 10 hours for the loss of material. Scanning electron microscopy (SEM) was also conducted for top ring and liner surfaces after a definite interval and whenever possible. The surface profile was done for liner and top ring surfaces initially and at the end of the experiments. All these results are shown in tables, micrographs and profiles in the following section.

RESULTS AND DISCUSSION

The simulator was run for 30 hours and then the experimental data was analysed for material removal rate or wear rate (Wear factor) for the three top rings (two compressor rings, one scrapper ring). The wear factors are calculated by the formulae given by equation 1.

Wear Factors

$$K = \frac{wear\_volume}{load \times Total\_sliding\_distance} \quad (1)$$

The data was analysed and plotted for the top three rings in figure 6.

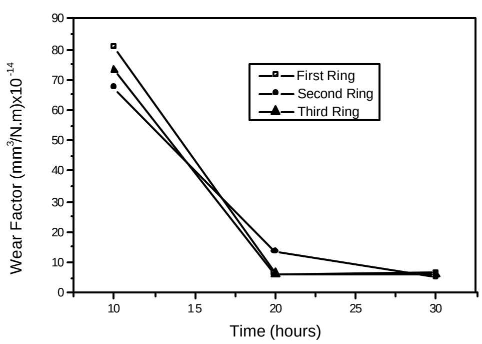

Figure 6: Wear Factors for the Top Three Rings

Wear rates for initial 10 hours decreases for all rings as shown in figure 6. After 20 hours, the wear rates become fairly stable and subsequently, the material removal rate from the rings reduces and remains fairly constant.

Surface Profile

The surface profile of the liner surface was done using Mitutoyo make surface roughness tester, model SJ-301. The profile of the liner was done before starting the test as shown in figure 7 and then the profiling was done after the conclusion of the test at three locations as shown in figure 8-10. The roughness data of the profile is given in table 2.

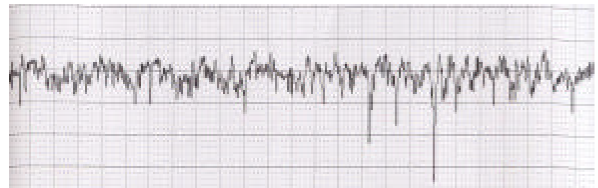

Figure 7: Profile of Fresh Honed Liner
(Vertical Scale: 10.0 µm/cm)

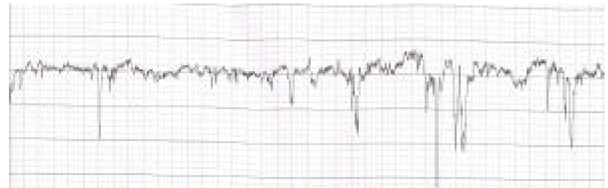

Figure 8: Profile of Liner after Experiment (TDC)
(Vertical Scale: 2.0 µm/ cm)

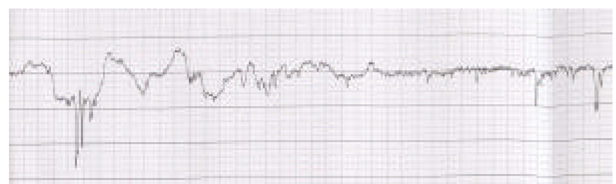

Figure 9: Profile of Liner after Experiment (Mid Stroke)
(Vertical Scale: 5.0 µm/ cm)

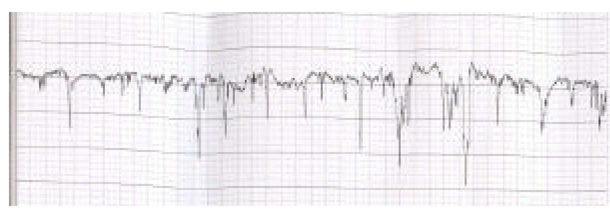

Figure 10: Profile of Liner after Experiment (BDC)
(Vertical Scale: 2.0 µm/ cm)

The profiles of the liner clearly show that there was higher amount of wear at the TDC location and finer plateau-honing takes place after 30 hours of operation in reciprocating mode. After running in an engine, surface profile of liner also shows similar plateau-honed liner. This surface is relatively stable in terms of wear. Therefore, building plateau-honed liner can control initial wear of an engine and thus reduce the amount of wear debris generation during the initial running-in period. The results of the profile are also supported by the roughness parameters shown in table 2 below. It is clearly evident from the data that the wear is higher at TDC location and is lower in mid-stroke position. BDC position shows higher wear than the mid-stroke position. This is because at the extreme positions of TDC and BDC, the linear velocity of the piston is close the zero and its maximum at mid-stroke position. There is breakdown in the oil film at extreme positions due to low piston velocities.

Table 2: Roughness Parameter of the Liner

Roughness Parameter	Before Experiment (µm)	After Experiment		
		TDC (µm)	Mid (µm)	BDC (µm)
R_a	0.7	0.22	0.46	0.24
R_q	1.06	0.39	0.61	0.39
R_z	7.48	2.39	2.87	2.36
R_p	2.47	0.71	1.29	0.68
R_t	13.48	4.02	3.56	3.59
R_v	11.02	3.31	2.27	2.91
R_{sk}	-3.10	-3.21	-0.98	-2.68

The average roughness (R_a) value was 0.7 initially but after experiment, this value of R_a is different at different positions because of different lubrication regimes in these regions. The peak roughness R_p is the height of the highest peak in the roughness profile. Similarly, R_v is the depth of the deepest valley in the roughness profile. The total roughness, R_t, is the sum of these two, or the vertical distance from the deepest valley to the highest peak. Since, valley serves as a reservoir for lubrication oil and value of R_p,R_v, and R_t was lower at TDC compared to other position after experiment. This clearly indicates that higher wear takes place at TDC location. Skewness describes the shape of an engineering surface. A negative skewed surface is predominantly valley based while a positively skewed surface is predominantly peak based. When a cylinder bore is run against piston rings under normal operating conditions, it can be seen from table 2 that the scale of roughness is reduced because the upper surface asperities have been removed.

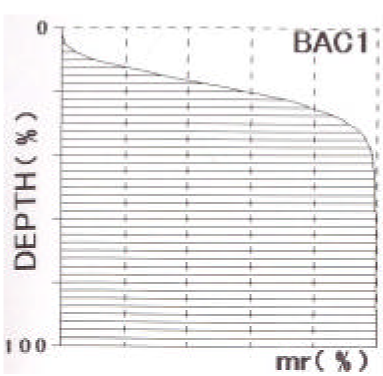

Figure 11: Bearing Area Curve of Honed Liner

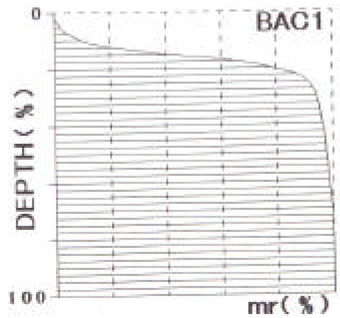

Figure 12: Bearing Area Curve of Liner after Experiment (TDC)

The R_k family of parameters was used to characterize the liner surface. These parameters were selected over more conventional parameters because they take into account not only the height of surface features, but also their width. The R_k parameters are derived from bearing area curve of the surface. Figure 11 and figure 12 shows bearing area curve of liner, before and after the experiment.

Table 3: R_k Parameters of Liner Surface

Roughness Parameter	Before Expt.	After Experiment		
		TDC	Mid Stroke	BDC
R_{pk}	0.55 µm	0.15 µm	0.88 µm	0.21 µm
R_k	2.0 µm	0.60 µm	1.03 µm	0.56 µm
R_{vk}	1.46 µm	1.00 µm	2.03 µm	1.34 µm
Mr_1	7.0%	5.4%	16.1%	3.7%
Mr_2	86.3%	82.0%	80.4%	79.1%

The ratio of R_{vk}/ R_k is called "Plateauness". As the part becomes more and more plateaued, the R_k becomes smaller and smaller, causing the Plateauness to increase as seen in table 3. Since more wear takes place at TDC and BDC so valve of R_k is lower compared to mid stroke after experiment.

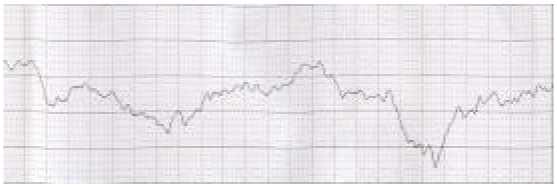

Figure 13: Surface Profile of Top Ring before Experiment
(Vertical Scale: 2.0 µm/ cm)

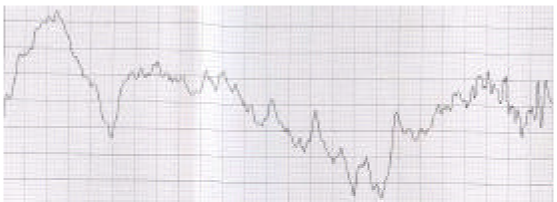

Figure 14: Surface Profile of Top Ring after Experiment
(Vertical Scale: 0.5 µm/ cm)

The profile of the top ring was done before and after experiment as shown in figure 13 and figure 14. Roughness parameters of ring profile are given in table 4.

Table 4: Roughness Parameters of the Top Ring

Roughness Parameter	Before Experiment (µm)	After Experiment (µm)
R_a	0.7 µm	0.28 µm
R_q	0.87 µm	0.36 µm
R_z	3.08 µm	1.74 µm
R_p	0.48 µm	0.85 µm
R_t	3.08 µm	1.74 µm
R_v	2.60 µm	0.89 µm
R_{sk}	-1.64	-0.11
R_{pk}	1.07 µm	0.58 µm
R_k	1.19 µm	0.68 µm
R_{vk}	0.45 µm	0.65 µm
Mr_1	21.7%	13.1%
Mr_2	85.3%	84.7%

The parameters indicate the extent of wear on the top ring during the experiment while using honed liner.

Scanning Electron Microscopy

In failure analysis, Scanning electron microscopy is a natural extension of optical microscopy. The use of electrons instead of a light source provides much higher magnification (up to >10,000X), unique imaging, and the opportunity to perform elemental analysis and phase identification. There are many advantages to using the SEM instead of a light microscope. The SEM has a large depth of field, which allows a large amount of the sample to be in focus at one time. The SEM also produces images of high resolution, which means that closely spaced features can be examined at a high magnification. Preparation of the samples is relatively easy since most SEMs only require the sample to be conductive. The combination of higher magnification, larger depth of focus, greater resolution, and ease of sample observation makes the SEM one of the most heavily used instruments in research areas today.

The SEM for liner was done before starting the test as shown in figure 15. The magnification used is 400X. The SEM was also conducted for the liner segments after completing the test at the three locations similar to profile. Figure 15 shows a scanning electron microscope (SEM) photograph of the fresh honed liner. One can see the typical crosshatched honed structure in the honed liner. Figure 16-18 show SEM micrographs of liner at TDC, mid position and BDC respectively. It is clearly visible that higher wear takes place at TDC location compared to mid position and BDC location. The crosshatched honing marks are completely removed from TDC location clearly showing the extent of wear that takes place in 30 hours of simulator operation.

The SEM for the top ring was also conducted before executing the test and the micrograph is shown in figure 19. The SEM of the top ring was also conducted after 20 hours and 30 hours respectively as shown in figure 20-21. These also reflect that there is significant amount of wear that takes place on hard surface of the chrome plated top ring.

The SEM micrographs shown in figure 15-21 clearly show the extent of wear on ring and liner surfaces at different locations and verify the earlier results of wear rates and profile.

Figure 15: SEM Micrograph of the Liner Surface before Starting the Test

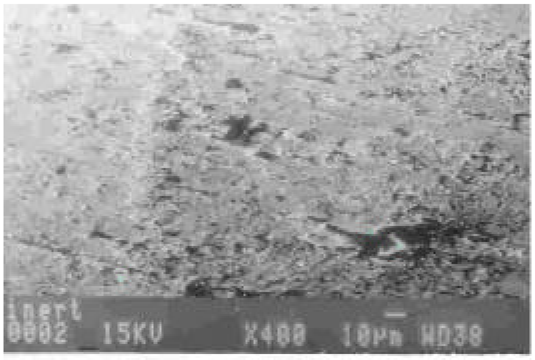

Figure 16: SEM Micrograph of the Liner Surface After 30 Hours at TDC Location

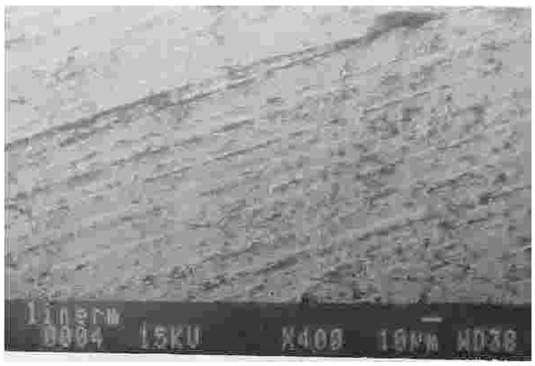

Figure 17: SEM Micrograph of the Liner Surface After 30 Hours at Mid-Stroke Location

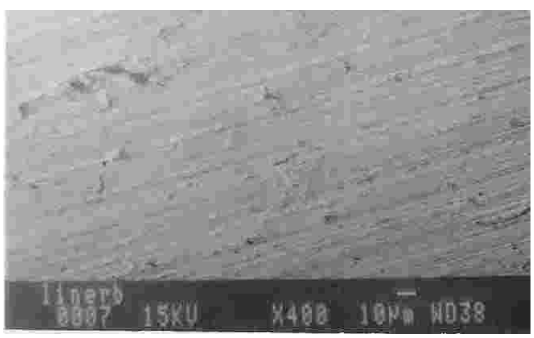

Figure 18: SEM Micrograph of the Liner Surface After 30 Hours at BDC Location

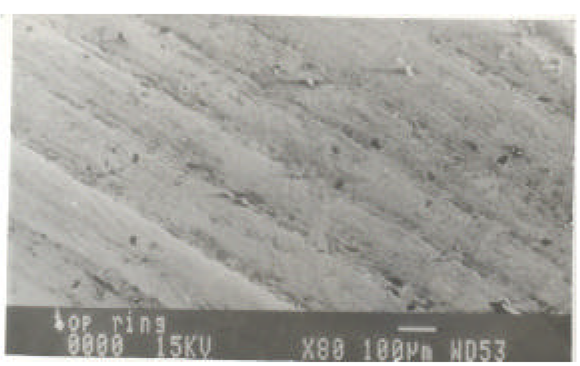

Figure 19: SEM of Unused Top ring

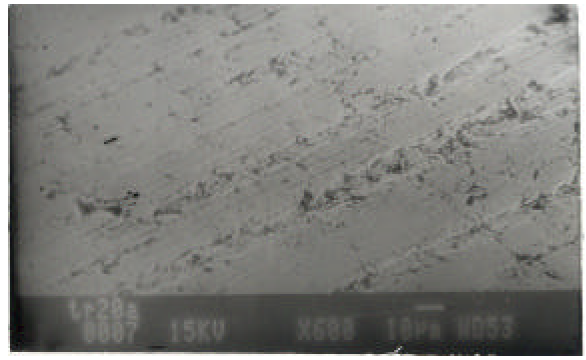

Figure 20: SEM of Top Ring after 20 Hours.

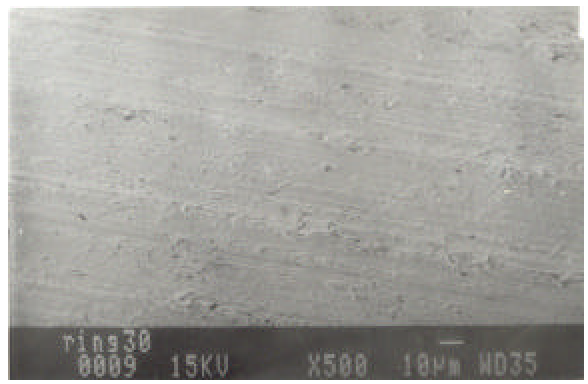

Figure 21: SEM of Top ring after 30 hours.

Conclusions

A reciprocating wear simulator driven by an electric motor has been developed for the purpose of simulating dynamic engine operating conditions for a purpose of studying the wear characteristics and profiles of the liner and piston surfaces. Surface profile and SEM tests were conducted on liner and rings at different locations, which reveal that highest amount wear takes place at TDC location compared to other positions. Even at the BDC Location, the wear was higher than the mid-stroke position because of the failure of hydrodynamic lubrication regime in BDC area. The wear profile and scanning electron micrographs successfully represent the areas of higher wear in an actual engine. This study clearly reflect that building in plateau-honed surface during the engine manufacturing stage will reduce the running period, and improve the life of the engine liner.

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