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# Effect of liner surface properties on wear and friction in a non-firing engine simulator

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## Abstract

The performance of a combustion engine is closely related to the friction force and wear between cylinder liner and piston rings. It is believed that this friction force can be significantly reduced by optimizing the surface topography of cylinder liners. Therefore, it is necessary to understand how liner surface topography affects wear, friction and lubricating oil consumption. Several experimental studies have been carried out for evaluating wear and friction in simulated engine conditions using Cameron–Plint wear testers, Pin-on-disk testers, SRV testers, etc. However, these studies do not reflect the true behaviour of inside the engine because of stroke length limitations. In this paper, a non-firing engine simulator has been developed in order to simulate engine conditions to a closer extent compared to these machines. This simulator can operate at similar linear speed, stroke, and load as real engine and can simulate almost all engine operating conditions, except firing pressures. In the present study, a production grade cylinder liner has been used for the experiments conducted using a custom-made non-firing engine simulator. The wear and surface property behaviour were evaluated at several locations in the liner and found that after running-in an engine, surface of cylinder liner exhibits plateau-honed-like characteristic. Energy dispersive analysis (EDS) has been carried out of liner and top ring for evaluating materials transfer. Coefficient of friction between three different liner segments and ring was evaluated using an SRV wear tester. Coefficient of friction in the piston ring–liner interface increases with increasing average surface roughness for liner.


*Keywords:* Reciprocating wear tester; Liner; Plateau honing; Surface profile; Scanning electron microscopy

## 1. Introduction

To meet competitive durability goals in the automobile industry, it is necessary to have improved understanding of the effect of wear in the cylinder liner piston assembly. 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 can lead to exceptionally higher wear and scuffing. Rough liners have very high coefficient of friction and high rates of wear in spite of their good oil retention capacity. Hence a balance between the two is required in order to achieve lower friction, wear and longer engine life.

The power cylinder is a major contributor to the overall mechanical friction of the engine. Mechanical power loss amounts to approximately 10–15% of the total fuel energy. Approximately half of this mechanical loss is because of friction at the cylinder liner–piston ring interface [1,2]. The Piston ring assembly plays a very important role, providing a dynamic seal between combustion chamber and crankcase. This seal minimises the expansion stage power loss due to pressure loss from the combustion chamber to the crankcase. Wear between the piston rings and cylinder

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liner has to be minimised in order to ensure lower power 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 condition. They contribute as much as twice the friction of each compression ring [3,4]. Wear near the top dead centre is often a limiting factor to the life span of an engine. 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 [5].

The study of the tribological properties of the cylinder liner and piston ring system in an internal combustion engine has attracted much attention in the last few years. Cylinder-liner wear is known to play a major role in internal combustion engine durability, performance, emissions, fuel economy and lube oil consumption.

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. These stresses are even higher 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 of improving the wear resistance of grey cast iron liner are: (a) adding special alloy elements, (b) using a surface treatment technique such as induction hardening, gas nitriding and (c) applying surface coatings [6]. 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.

### *1.1. Liner surface preparation*

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 10 μm deep). Second, a finish-honing step, also called plateau honing, gives a relatively smooth surface to the plateaux [7]. The cylinder bore honing quality is an essential factor for a good engine performance and durability. A bad surface finish may lead to excessive lubricating oil consumption, high piston ring wear and scuffing. 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 [8].

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 which get embedded in the liner surface. Plateau-honed surface is relatively stable in terms of wear. Therefore, initial wear can be controlled by building plateau-honed surface of the liner during the manufacturing process and thereby relieving the engine of the burden of large initial wear and associated debris. During the later part of life of the liner, the plateau surface would continue to posses relatively large, smooth plateau, which provide a large bearing area, and also deep valleys to retain oil for lubrication between the surfaces and provide a relief area for wear particles. Tim and Mike [9] examined the effect of plateau honing on both rough and smooth liners and found that smooth plateau liner ($R_a \approx 0.78$ μm) offers better finish with regard to oil consumption, ring wear, liner wear between the ring turnarounds and volume of liner material lost due to wear.

Wear of liner is different at thrust side than anti-thrust side and also it varies over the stroke length. This can be seen in an exaggerated view of liner shown in Fig. 1. Higher

Table 1
Effect of honing angle on oil consumption [9]

| Honing angle (°) | Oil consumption (g/kWh) |
|---|---|
| 23 | 0.58 |
| 70 | 0.59 |
| 120 | 0.37 |

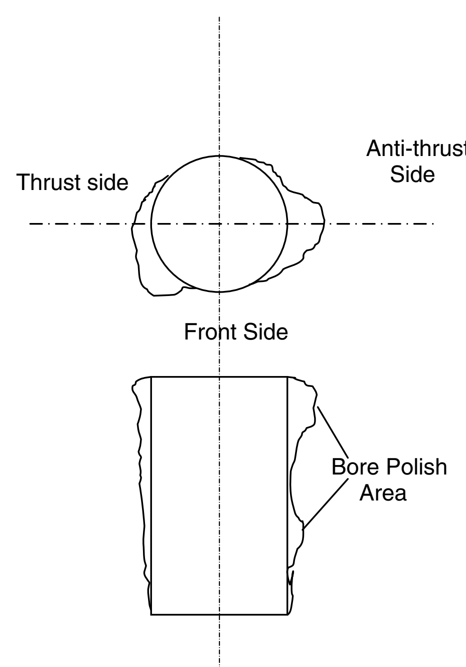


Fig. 1. Exaggerated diagram of typical cylinder liner wear [14].

wear takes place at TDC in a cylinder liner and there is measurable wear all over the stroke length [10,11]. As can be seen from Fig. 4, higher wear takes place on the anti-thrust side [12].

The piston rings are main cause of liner wear rather than piston skirt. At TDC location, mainly boundary lubrication conditions are responsible for wear. There are other factors also such as: highest combustion chamber pressure, thinnest lubricating films, high temperatures and greatest concentration of acids in the localised region. The film thickness at the TDC location is normally very small, in the range of 0–10 μm, because of the low piston sliding velocity in this region. The maximum oil film thickness is reported at the mid-stroke region, where the piston speed is maximum and hydrodynamic lubrication conditions are achieved. Thus piston rings move in both hydrodynamic and boundary lubrication regimes. Oil cleanliness has a significant effect on ring and liner wear. Used oil 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. These third bodies 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 an important technique to characterise 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 undeformed surface, depending on their proximity [13]. Surface profiles change as the surfaces are used and friction, deformation and wear occur. A useful measure for describing how surface profiles change is the bearing area curve. The curve has a physical meaning and represents the material ratio when slicing the surface at a certain height.

Bearing length ratio estimates radial wear of engine cylinder bore. The bearing length ratio parameter provides a more meaningful wear estimate than the arithmetic average parameter. 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, 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.

Ting [14] 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 piston velocity. As the piston ring begins to slide with increasing velocity, friction forces fall quickly and become rather small in the mid-stroke region. At the end of the stroke, friction rises to another large friction force value. 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 force developed near the ring reversal regions is high, it does not contribute significantly towards the total ring friction power loss because of low sliding velocities in the corresponding region.

The objective of the paper is to fabricate a machine that closely approximates the wear process occurring between cylinder liner and piston ring inside an internal combustion engine. This machine is expected to provide assistance in having an improved understanding of the wear process and will also enable us to make a proper choice of materials and surface properties for engine liners.

### 1.2. Reciprocating wear simulator

Reciprocating wear simulator was designed and developed for experimental simulation of the effect of various surface properties on the frictional forces, wear and lubricating oil consumption in an engine.

A single cylinder diesel engine (Make: Cooper, UK; Model: CVR-5) was converted into this simulator. The main advantage of this simulator is that it takes full-length stroke as in the case of actual engine rather than a short stroke as in the case of reciprocating wear tester. Fig. 2 shows the reciprocating wear simulator and Fig. 3 shows the schematic of the loading mechanism.

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

Currently, Testing can be done at room temperature at variable load. In a fired engine, peak combustion pressures that push on the back of the top ring can reach

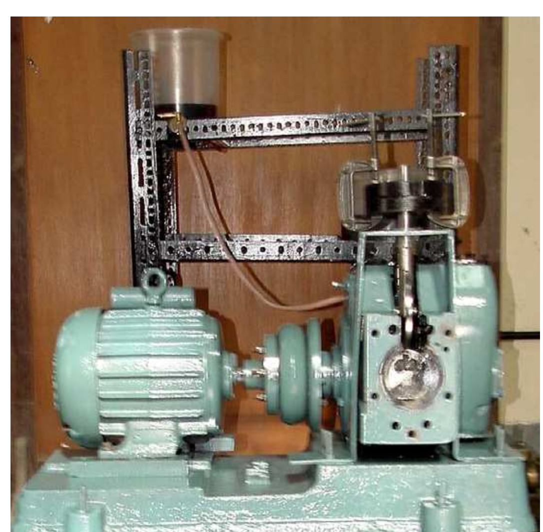

Fig. 2. Reciprocating wear simulator.

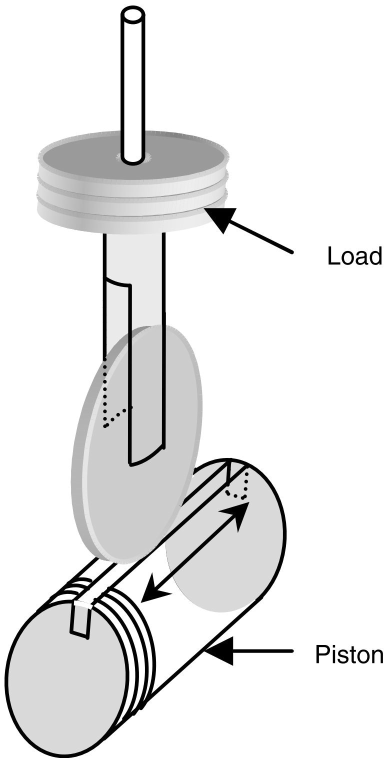


Fig. 3. Loading mechanism on the piston.

up to 9.4 MPa. This pressure, when applied to the ring piece used in these tests, would be equivalent of a 415 N load. It should be noted that the peak pressure in the four stroke engine do not last over the entire cycle of the engine. Only the ring tension exerts a force on the liner. This is approximately equivalent to a 14 N load on the ring piece [15]. The forces that piston rings experience are a combination of forces such as several axial and radial forces including mass force, friction 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. The liner diameter is 82 mm. The stroke length of the simulator is 82 mm, at an oscillating frequency of 25 Hz. However, provisions can be made in the simulator to heat the liner and conduct the tests at an elevated temperature close to actual engine conditions. Provisions are made for lubrication of the liner using SAE 20W40 engine lubricating oil.

Since, this reciprocating machine is a non-firing machine hence additional loading to compensate for gas forces is done by employing a special loading mechanism to provide additional load in radial direction. Load is given on the piston, as shown in Fig. 3. 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. Provisions are made to stop the circumferential movement of the rings at the piston grooves by providing pins in suitable locations in the ring grooves of the piston.

## 2. Experimental matrix

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

The liner was machined to remove a 40 mm wide segment along its length and then installed in the simulator. A new set of piston rings was installed. Top ring was a chrome plated cast iron ring. The simulator was assembled and lubricated before starting of the test. The simulator was operated at 1500 rpm for 30 h at a constant load (60 N). The rings were weighed initially and after every 10 h for evaluating loss of ring materials. Scanning electron microscopy (SEM) was conducted for liner surfaces after a predefined intervals. The surface profile was done for liner surfaces before commencing the test and after completion the test. All these results are shown in tables and figures in the following section.

## 3. Results and discussion

Wear factor is used to estimate the change in shape as a result of wear. Wear factor is calculated for top, second and third ring after 10, 20, and 30 h running duration. The wear factors are calculated by the formula given by Wear factor

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

The results of wear factor for various rings shown in Fig. 4. Wear factor for first 10 h decrease for all rings. After 20 h, the wear factor become fairly stable and subsequently, the material removal rate from the rings reduce and remain almost constant. This is possibly because after 20 h, liner surface becomes similar to plateau-honed surface, which is a stable form in terms of wear.

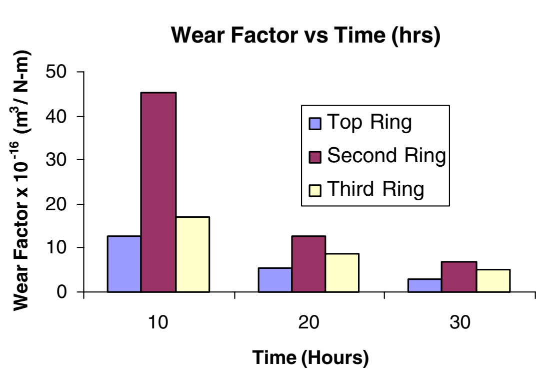


Fig. 4. Wear factor vs. time.

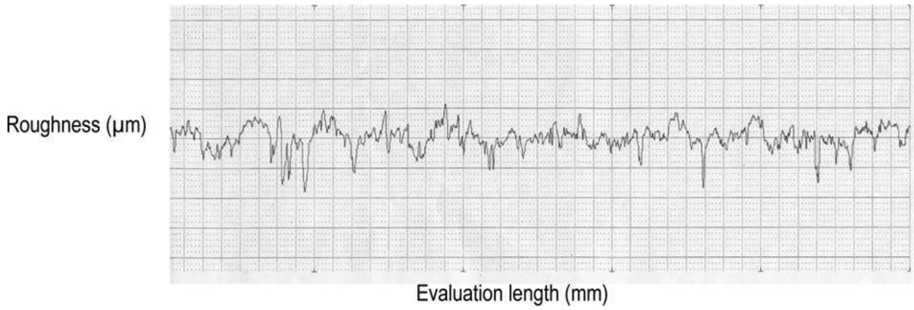


Fig. 5. Profile of fresh liner.

### 3.1. Surface profile measurements

The surface profile of the liner surface was done using two-dimensional stylus-based surface profiling instrument (Make: Mitutoyo, Japan; Model: SJ 301). The surface profile of the liner was done before starting the test (Fig. 5) and then after the commencement of the test at three locations (close to TDC, mid-stroke position and close to BDC) at anti-thrust side (Figs. 6–8). The profiles of the liner clearly show that higher wear takes place at TDC location and

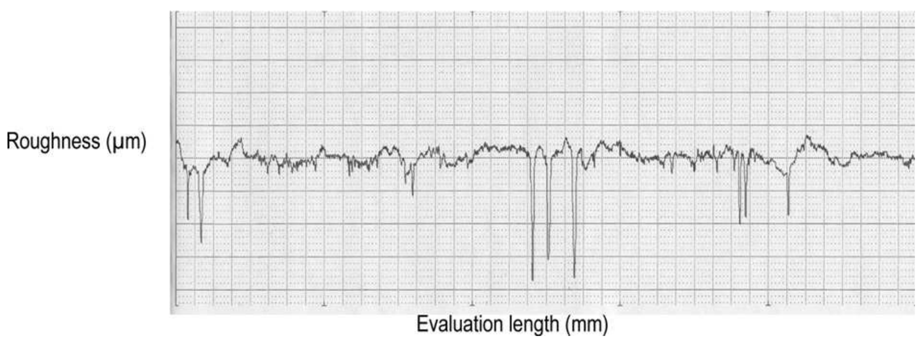


Fig. 6. Profile of liner after experiment (TDC).

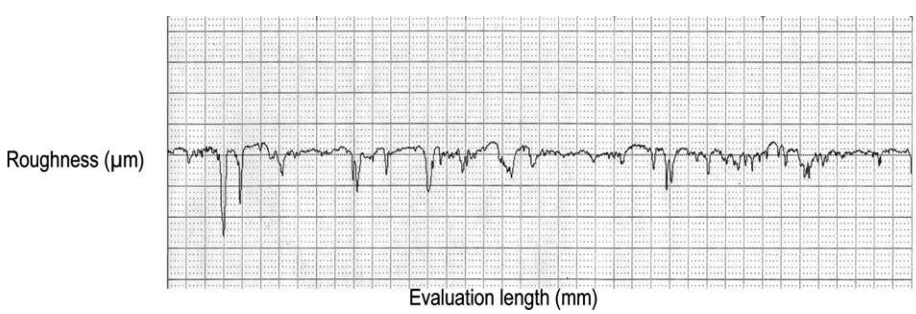


Fig. 7. Profile of liner after experiment (mid stroke).

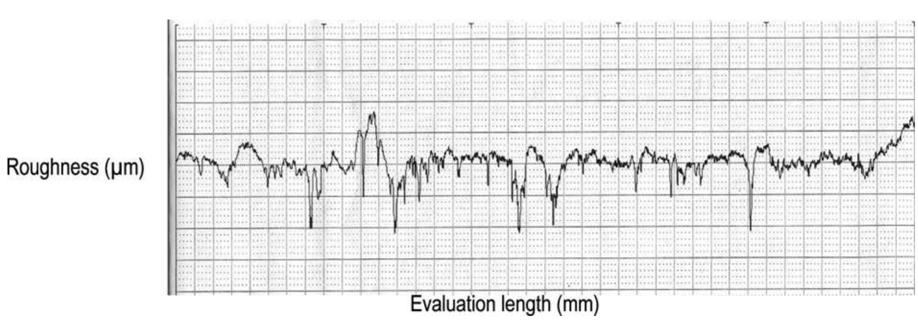


Fig. 8. Profile of liner after experiment (BDC).

finer plateau-honing takes place after 30 h in reciprocating mode. After running in an engine, surface profile of liner also exhibits plateau-honing like characteristics. This surface of the liner from reciprocating wear simulator is relatively stable in terms of wear (Fig. 6). Therefore, building plateau-honed liner at the time of engine production can control initial wear of liner and thus reduce the wear debris generation during initial running-in period. The results of the profile are also supported by the roughness parameters (Table 2). It is clearly evident from Table 2 that higher wear takes place at TDC location and relatively lower at mid-stroke position. BDC position showed slightly higher wear than that of mid-stroke position. This is because at the extreme positions (TDC and BDC), linear speed of the piston is close the zero and it is maximum at mid-stroke position. There is possibly a breakdown in the oil film between rings and liner at extreme positions due to very low piston speeds.

Roughness values of the cylinder liner, after experiment, become different at different locations because of different lubrication regimes in these regions.

Fig. 9 shows bearing area curve of unworn liner. Initially, percentage of surface area available for bearing is less. It can be also seen in Fig. 9 that slope of the curve is sharp. After experiment, percentage of surface area available for bearing is higher at lower percentage of depth as shown in Fig. 10. The $R_k$ parameters are derived from bearing area curve. The $R_k$ family of parameters was used to characterise 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 [13]. Value of $R_k$ parameters of liner is given in Table 3. 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.

Table 2
Roughness data of liner

| Roughness parameter | Before experiment (μm) | After experiment | | |
|---|---|---|---|---|
| | | TDC (μm) | Mid (μm) | BDC (μm) |
| $R_a$ | 0.76 | 0.24 | 0.46 | 0.32 |
| $R_q$ | 1.00 | 0.43 | 0.73 | 0.46 |
| $R_z$ | 5.68 | 2.80 | 4.17 | 2.80 |
| $R_p$ | 2.87 | 0.69 | 1.07 | 1.68 |
| $R_t$ | 7.34 | 4.42 | 7.64 | 3.86 |
| $R_v$ | 4.48 | 3.73 | 6.57 | 2.18 |
| $R_{sk}$ | −0.74 | −4.14 | −3.50 | −0.35 |

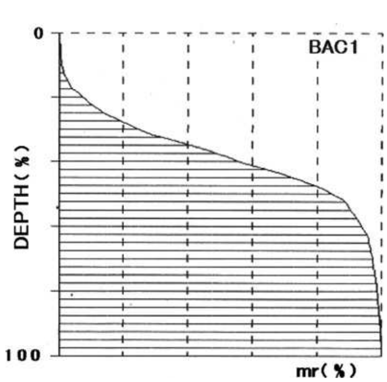


Fig. 9. Bearing area curve of fresh liner.

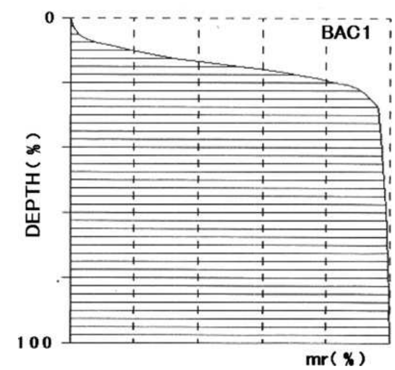


Fig. 10. Bearing area curve of liner after experiment (TDC).

Table 3
$R_k$ parameters of liner surface

| Roughness parameter | Before experiment | After experiment | | |
|---|---|---|---|---|
| | | TDC | Mid | BDC |
| $R_{pk}$ | 0.74 μm | 0.28 μm | 0.42 μm | 0.22 μm |
| $R_k$ | 2.49 μm | 0.75 μm | 0.98 μm | 0.91 μm |
| $R_{vk}$ | 2.05 μm | 0.85 μm | 1.25 μm | 1.31 μm |
| $Mr_1$ | 7.6% | 8.1% | 6.0% | 5.3% |
| $Mr_2$ | 89.8% | 86.4% | 82.2% | 87.3% |

### 3.2. Scanning electron microscopy (SEM)

Scanning electron microscopy for liner was done before starting the test as shown in Fig. 11. The magnification used is 400×. The SEM was also conducted for the liner segments after completing the test at the three locations similar to profile. One can see the typical cross-hatched honed structure in the liner initially. Figs. 12–14 show

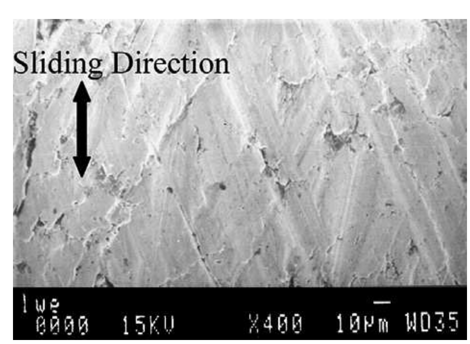


Fig. 11. SEM micrograph of the liner surface before starting the test.

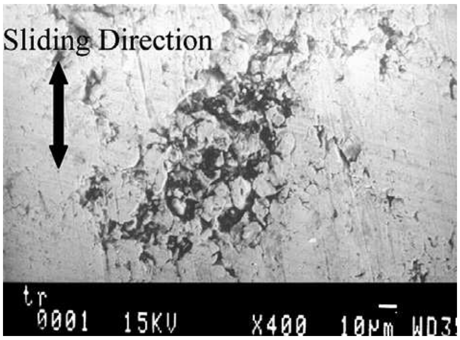


Fig. 12. SEM micrograph of the liner surface after 30 h at TDC location.

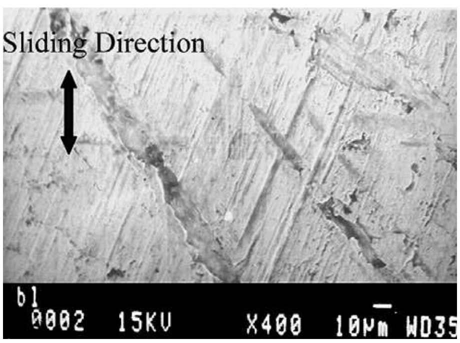


Fig. 13. SEM micrograph of the liner surface after 30 h at mid-stroke location.

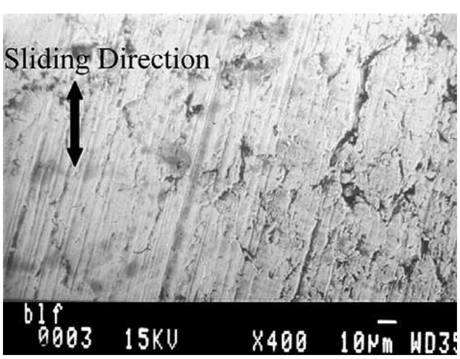


Fig. 14. SEM micrograph of the liner surface after 30 h at BDC location.

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 h of simulator operation.

### 3.3. Energy dispersive spectrometer (EDS)

The elemental composition at a point, along a line, or in a defined area can be easily determined to a high degree of precision (~0.1 wt%) using EDS. The objective of EDS analysis applied to this study is to identify the materials transfer between piston rings and liner surfaces during experiment. The EDS analysis was conducted for liner and ring segments to evaluate the metal species present on the surface.

X-ray micro-analysis spectrum of liner surface (Figs. 15 and 16) showed a typical EDS spectrum of an outer surface of liner. The major elements are iron, manganese, chromium and silicon. After experiment, weight percentage of elements slightly decreases due to the materials transfer between piston rings and liner surface. Table 5 summarises the weight percentage of major elements in the outer layer of top ring surface. Initially, there is coating of chromium and manganese on the outer surface of top ring; hence EDS analysis (Fig. 17) shows only peaks of chromium and manganese. After experiment, coating on the top ring surface has been worn out during experiment (Fig. 18).

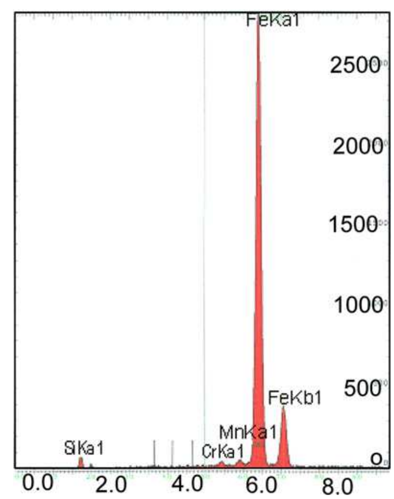


Fig. 15. X-ray microanalysis spectra of the surface of liner (before experiment).

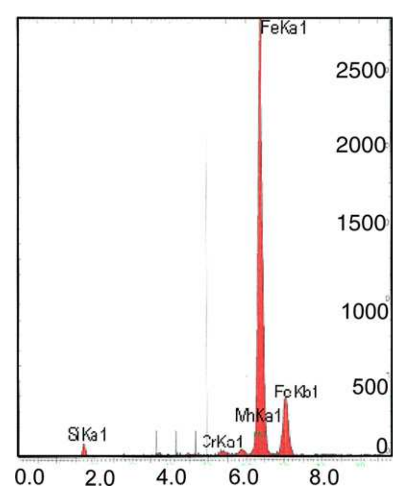


Fig. 16. X-ray microanalysis spectra of the surface of liner (after 30 h of experiment).

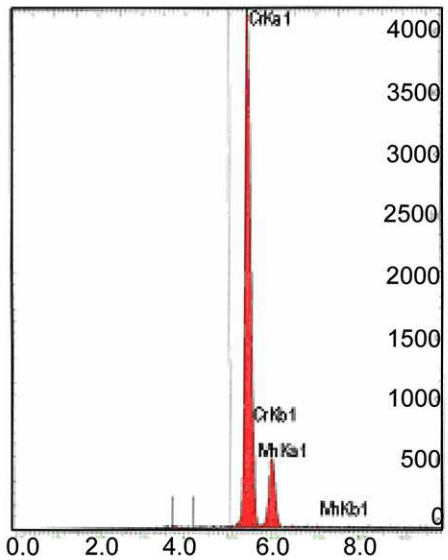


Fig. 17. X-ray microanalysis spectra of the surface of top ring (before experiment).

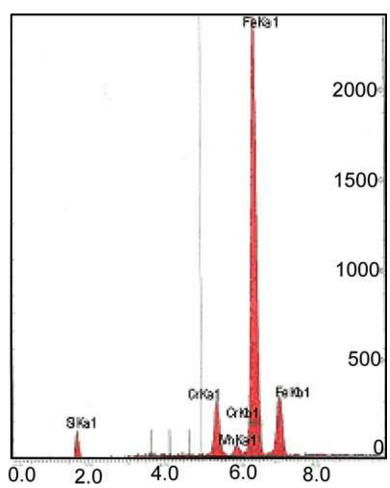


Fig. 18. X-ray microanalysis spectra of the surface of top ring (after 30 h of experiment).

From these EDS experiments, it is evident that silica particles get embedded in the liner surface. This is clearly reflected by increase silica concentration in Table 4. In the rings, the hard chromium plating gets removed and the base material of ring (containing iron) gets exposed, as evident from Table 5.

Table 4
Concentration of the major elements in the outer layer of liner

| Elements | Weight percentage | |
|---|---|---|
| | Before experiment | After experiment |
| Fe | 97.55 | 97.43 |
| Mn | 1.34 | 1.27 |
| Cr | 0.74 | 0.73 |
| Si | 0.37 | 0.57 |

Table 5
Concentration of the major elements in the outer layer of the top ring

| Elements | Weight percentage | |
|---|---|---|
| | Before experiment | After experiment |
| Fe | – | 92.20 |
| Mn | 2.73 | 0.73 |
| Cr | 97.27 | 6.06 |
| Si | – | 1.01 |

### 3.4. Coefficient of friction

SRV optimal wear tester is used to simulate the relative motion of piston ring–liner interface inside an engine. Due to physical restriction, the ability to change the stroke on this tester is limited. However, fairly good result for the lubricity, coefficient of friction and specific wear rate of the surface properties can be achieved using this machine.

Ring segments were made from the top compressor ring of the engine. The top ring is the hardest piston ring with chrome plating. Three disks were machined from the engine liner, having different surface roughness. A pin segment was held tightly in an aluminium holder provided in the SRV tester. The pin performs a reciprocating motion against the disk made from the engine liner dipped in the lubricating fluid. Various operating parameters were fixed as follows:

Stroke length 1 mm
Oscillation frequency 25 Hz
Load 60 N
Duration of run 30 min.

In SRV, piston ring segment and liner disk are in hydrodynamic lubrication regime. It creates a condition, which is similar to mid stroke position. When coefficient of friction measured by SRV using different liner then it is found that friction coefficient increases with increasing surface roughness of liner as shown in Fig. 19. However, value of friction coefficient lies between 0.02 and 0.025.

From Fig. 19, it can be clearly seen that the average coefficient of friction is 0.02, 0.022 and 0.025 for average surface roughness of 0.60 μm, 0.74 μm and 1.34 μm, respectively. This reveals to coefficient of friction in the piston

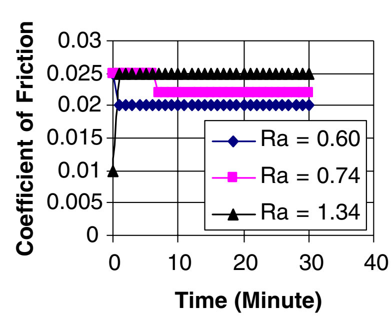


Fig. 19. Coefficient of friction vs. time for liner with different surface properties.

ring–liner interface increases with increasing average surface roughness for liner. This suggests that building plateau-honed surface at the time of production of the engine not only reduces the effective wear of the engine during its life time but also reduces the coefficient of friction.

## 4. Conclusions

A reciprocating wear simulator has been designed and developed for the purpose of simulating dynamic engine operating conditions and studying the wear characteristics of the liner and piston ring surfaces. Surface profile and SEM tests were conducted on liner and rings at different stroke position, which reveal that highest amount wear takes place at TDC location. Even at the BDC location, the wear was higher than the mid-stroke position because of failure of hydrodynamic lubrication regime. After running in an engine, surface profile of liner changes and becomes to a similar plateau-honed liner. Therefore, building plateau-honed liner at the time of production can control initial wear of the engine and thus reduce the amount of wear debris generation during the initial running-in period. Building-in plateau honed-liner surface at the production stage will reduce the initial wear due to running-in. Energy dispersive analysis reveals the composition of liner and piston ring surface material. The major elements of liner material are iron, manganese, chromium and silicon. During the experiment, concentration of silica goes up. In the ring, the hard chrome plating wears out with time and the base metals gets exposed towards the end of the experiment. Coefficient of friction measured by SRV reciprocating wear tester. Experimental data reveals that friction coefficient increases with increasing surface roughness of liner surface. However, friction coefficient varies in the range of 0.02–0.025 in the mid-stroke position.

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