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# A study on the effect of oil pocket density on the effectiveness of boundary lubrication in the liner-ring interface

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## Abstract

Lubricant retention during sliding of mating surfaces is one of the major problems encountered in the industrial components. The efficiency of the engine is affected by the frictional losses taking place at the engine liner-piston ring interface. The coefficient of friction and the resulting wear is directly related to the effectiveness of the lubrication. One of the methods to achieve good lubrication is by making oil pockets on the surface of the component. This leads to a mixed lubrication regime.
The objective of this study focuses on the influence of oil pocket distribution on the effectiveness of lubrication. Oil pockets were made on the liner-ring samples. To simulate the wear behavior, reciprocating wear tests were performed on liner-ring samples at different experimental conditions. Three basic parameters such as load, temperature and frequency were considered for the analysis. The result of this experiment will be analysed.

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*Keywords:* *Liner; Piston ring; Oil pockets; Boundary lubrication*

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## 1. Introduction

The friction loss in an internal combustion engine is a major factor in determining the fuel economy and performance of the vehicle utilizing the power of the engine. Some 20–30% of the friction losses in a compression-ignition engine are due to the piston/cylinder system, of which a large part is attributed to the piston ring. Therefore, reductions in frictional losses will lead to an improvement in fuel economy as it will reduce both the exhaust and cooling losses also at the same ratio. The surface texturing of piston rings and cylinder liners have

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attracted attention of many researchers. Improving fuel efficiency with laser surface textured piston rings were studied by I. Etsion and E.Sher. The authors studied the potential use of piston rings with a surface micro- structure in the form of spherical micro-dimples to reduce the friction between rings and cylinder liner, where the entire ring face in contact with the cylinder liner was textured [1]. An optimum value of the ratio of micro-dimple depth over diameter was found, which yields a friction reduction of 30% and even more compared with an un-textured ring.Waldemar Koszela, Pawel Pawlus and Lidia Galda studied the effect of oil pockets size and distribution on wear in lubricated sliding [2]. The results of theoretical work showed a potential reduction of friction force of about 30% by ring surface texturing in comparison to non-textured rings under full lubrication conditions. These results were confirmed experimentally. Surface texturing is a means for enhancing tribological properties of mechanical components. Various techniques can be employed for surface texturing including machining, ion beam texturing, etching techniques and laser texturing. Perhaps the most familiar and earliest commercial application of surface texturing in engines is that of cylinder liner honing. The honing technology is still the leading one in cylinder texture preparation. But more recently laser surface treatment (LST) has also been applied to cylinder liners and rings, showing up to 4.5% improvement in the torque compared with a standard liner or ring. The dimples of mainly spherical shape are usually formed on stationary surface of smaller hardness. Three dimensions characterize surface texturing: diameter, depth and area density. Usually dimple depth over dimple diameter ratio range of 0.01–0.3 and area density to 30% exists for assemblies operated in lubricated sliding conditions. The laser texturing is the most popular technique in forming micro pits. However other methods may be used. The oil pockets (also known as micro pits holes, dimples or cavities) may reduce friction in two ways: by providing lift themselves as a micro-hydrodynamic bearing, and also by acting as a reservoir of lubricant. Holes can also serve as a micro-trap for wear debris in lubricated or dry sliding. Surface texturing resulted in minimizing the surface ability to seizure. The dimples existence from area density of 10% improved seizure resistance of sliding pair: steel– spheroidal cast iron. Textured surfaces can provide traps for wear debris in dry contacts subjected to fretting. The dimple existence could improve the fretting wear resistance and almost doubled the fretting fatigue life. The dimple size and distribution were selected initially in order to obtain the area density (ratio) in the range of 10–90%. Usually smaller dimple area ratio is used. But we would like also to analyze the effect of surface layer hardening (not only surface topography) on wear. The oil pockets depth to diameter ratios were between 0.03 and 0.11. This range was recommended in the literature. Specimen surface had dimples with depths ranging from 45 to 115 µm. Dimples depths were comparatively big since oil pockets should exist on the surfaces during test; the wear conditions were very severe (assumed wear amounts of specimens were about100µm).

## 2. Materials

The cylinder sleeve segments used for study were cut into 9 equal parts from 115mm diameter of cast iron cylinder sleeve with each part of 41.1×30×4.4 mm. Similarly the piston ring segments were cut into 9 equal parts from a 115mm diameter ring with each sample of 40 degrees profile angle. Two sets of coated rings were used for the experiment. The first set consisted of three ring segments without oil pockets and the second set consisted of three segments with oil pockets. Both the sleeve and ring were cut by wire cut process. The surface of the sleeve segments was as honed with 120° honing angle and cleaned with acetone. The lubricant employed for conducting the experimental runs of ring on liner was base oil. The test was conducted on a reciprocating wear testing machine for duration of four hours.

## 3. Experimental equipment and specimen preparation

All the experiments were conducted according to the standards of ASTM G181-11. Proper care and necessary precautions were taken during the experiment.

### 3.1.Description of Equipment

Wear tests on cylinder sleeve were performed using high frequency reciprocating wear testing machine. The machine is equipped with data acquisition and control systems. The test apparatus consists of an upper specimen reciprocating over the lower specimen with a constant stroke length. The drive consists of a motor of power 1 kW which drives a cam for executing the reciprocating motion. The frequency of reciprocation can be varied between 10Hz and 15Hz. The average sliding speed for each stroke, s, in metres per second, is calculated as follows:

$$s = 2fL$$

In the above equation ‘f’ is the frequency of reciprocation in cycle per second and ‘L’ is the stroke length in meters. The machine is provided with facility for setting the load, temperature, frequency and duration of the test. It is provided with a provision for wear depth measurement using LVRT. It is unnecessary to set the stroke length equal to the full stroke of the piston in the engine because the greatest frictional influence of the materials is experienced at the ends of the ring travel where operation in the boundary lubrication regime increases the likelihood that contact will occur between the surfaces of the ring and cylinder materials. The stroke length should typically range between 5 and 10 times the width of the worn-in contact face of the piston ring specimen.

*Fig.1.(a)the experimental setup (b)close view of the upper and lower specimen holders*

### 3.2. Ring Specimen

The ring specimen is prepared by wire cut from a production piston ring of 115 mm diameter. Each ring specimen is cut at 40 degrees central angle. The ring is coated cast iron production piece. Proper care was taken in order to clamp the ring in its specific holder in order to avoid loosening during sliding.

### 3.3. Cylinder Bore Specimen

The specimen of cylinder bore is of cut section of production finished 115 mm diameter cylinder bore. The cylinder bore specimen is a cut section. Proper holders are provided in order to mount the bore specimen in a lubricant container.

### 3.4. Specimen Fixturing

The ring specimen was fixed in a specimen holder which is screwed on to the reciprocating arm provided in the setup. The bottom specimen was fixed in the specimen holder which is placed in the container. The container can be filled with the lubricant. Proper alignment and centering between sliding surfaces is a critical factor for ensuring repeatable friction test results. Alignment affects the distribution of normal forces on the contact surface as well as the lubrication regimes that change as the ring specimen moves back and forth.

### *3.5. Lubricant*

The lubricant used to analyze the wear behavior of the bore and ring specimens was base oil (SAE 30) with viscosity index of 97 and with flash point of 252°C.

## 4. Procedure

There are several parameters that can affect the wear. Load, temperature and frequency are the three main parameters which are considered for the wear tests and the following parameters were varied accordingly depending on the runs. Here the gray cast iron liner was made to wear against two sets of rings one with oil pockets and another without oil pockets under lubricated conditions. Each set were further subdivided into two groups i.e. with and without oil pockets. The process parameters and their working ranges are given as in Table 1. Initial mass of the specimens were measured using an electronic weighing machine. Ring and liner specimens were mounted in respective holders and verified for the proper alignment. Load, temperature of lubricant and speed were adjusted according to the experimental design. The actual tests of 9 runs were conducted for 4 hours with base oil as lubricant.The lubricant bath is filled to cover the contact surface with at least 2mm of the selected lubricant. After completing the test, the specimens were allowed to cool with the load removed. Specimens were removed, cleaned thoroughly in acetone and were dried. Final mass of the specimen was measured and the wear volume of the specimen was calculated by mass loss method. The coefficient of friction and wear volume were the responses whose behavior is mainly dependent on operating parameters.

Table 1. Parameters of the experiment

| Parameter | Range |
|---|---|
| Stroke | 15mm |
| Frequency | 10Hz |
| Loads | 100,150,200N |
| Temperature | 100 Degree |
| Make | DUCOM -Bangalore |

## 5. Results and discussion

In the first phase of the experiment, ring segments without oil pockets were tested and results were obtained. As shown in table 2 and 3 the mass loss in both liner as well as ring increases as load increases. Wear keeps steadily increasing with respect to time. It is the highest at 200 N, followed by 150N and then 100 N with the least wear observed.

Table 2. Mass loss observed in liner specimen at various loads when tested with coated rings without oil pockets

| Sn no | Load (N) | Temperature (ºC) | Frequency(Hz) | Initial mass (gm) | Final mass (gm) | Mass loss (gm) |
|---|---|---|---|---|---|---|
| **1** | 100 | 100 | 10 | 13.3200 | 13.3193 | 0.0007 |
| **2** | 150 | 100 | 10 | 12.5915 | 12.5903 | 0.0012 |
| **3** | 200 | 100 | 10 | 12.0303 | 12.0287 | 0.0016 |

Table 3. Mass loss observed in coated ring specimen without oil pockets at various loads

| Sn no | Load (N) | Temperature(ºC) | Frequency (Hz) | Initial mass (gm) | Final mass (gm) | Mass loss (gm) |
|---|---|---|---|---|---|---|
| **1** | 100 | 100 | 10 | 2.1330 | 2.1325 | 0.0005 |
| **2** | 150 | 100 | 10 | 2.1711 | 2.1705 | 0.0006 |
| **3** | 200 | 100 | 10 | 2.1666 | 2.1658 | 0.0008 |

The Fig 2 is the graph showing the variation coefficient of friction in rings without oil pockets at various loads. The graph does not show much change in trend and all the three individual graphs follow almost the same path and fall in the same region (0.110 - 0.090) thus suggesting boundary lubrication regime.

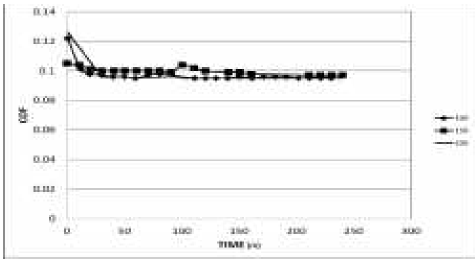

Fig.2. Graph showing the variation of coefficient of friction with respect to time.

In the second phase of the experiment the wear tests were carried out at three loads of 100,150,200N using the coated rings with micro sized oil pockets. The Table 3 shows the mass loss in liner material while Table 4 shows the mass loss in coated ring with oil pockets. From both the tables we can summarize that there is a small decrease in the mass loss in coated rings with pockets when compared with the mass loss observed in rings without oil pockets. Hence lesser wear was observed in the liner-ring interface.

Table 3. Mass loss observed in liner specimen when tested with coated rings (with oil pockets) at various loads

| Sn no | Load (N) | Temperature (ºC) | Frequency(Hz) | Initial mass (gm) | Final mass (gm) | Mass loss (gm) |
|---|---|---|---|---|---|---|
| **1** | 100 | 100 | 10 | 12.7136 | 12.7132 | 0.0005 |
| **2** | 150 | 100 | 10 | 12.8333 | 12.8327 | 0.0008 |
| **3** | 200 | 100 | 10 | 13.5442 | 13.5433 | 0.0009 |

Table 4.Mass loss observed in coated ring specimen (with oil pockets) at various loads.

| Sn no | Load (N) | Temperature (ºC) | Frequency(Hz) | Initial mass (gm) | Final mass (gm) | Mass loss (gm) |
|---|---|---|---|---|---|---|
| **1** | 100 | 100 | 10 | 2.1962 | 2.1960 | 0.0003 |
| **2** | 150 | 100 | 10 | 2.0889 | 2.0885 | 0.0004 |
| **3** | 200 | 100 | 10 | 2.1456 | 2.1451 | 0.0006 |

The wear data follows a steady path as it increases with respect to time. The wear is observed to be less in coated rings with oil pockets compared to rings without oil pockets. The coefficient of friction as shown in Fig 3 observed in all three loads follow almost the same trend, starting at a higher value and coming down to lower values towards the end of the experiment (0.110-0.090) there by suggesting boundary lubrication.

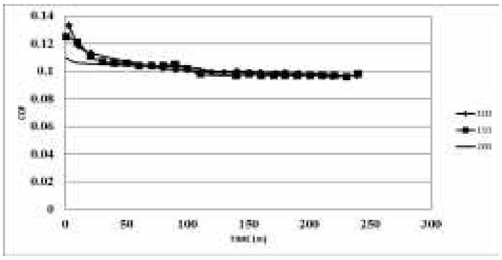

Fig.3. Graph showing the variation of COF with respect to time at various loads

The percentage decrease in mass loss observed in both liner and ring (without oil pockets and with oil pockets) have been summarized in Table 5 and Table 6.While in liner specimen mass loss is observed to be in a range between 28-38%, the decrease in mass loss in ring specimens are found to be 40% at 100N, 33.3% at 150N and 28.5% at 200N.This may be due to transfer of material that occurs at a certain level when two bodies slide at very high speeds and loads.

Table 5. Mass loss percentage observed in ring specimens at various loads

| Load (N) | Mass loss without dimples (gms) | Mass loss with dimples (gms) | Percentage of mass loss(%) |
|---|---|---|---|
| 100 | 0.0007 | 0.0005 | 28.27 |
| 150 | 0.0012 | 0.0008 | 33.33 |
| 200 | 0.0016 | 0.0010 | 37.50 |

Table 6. Mass loss percentage observed in ring specimens at various loads.

| **Load** (N) | Mass loss without dimples (gms) | Mass loss with dimples (gms) | Percentage of mass loss(%) |
|---|---|---|---|
| 100 | 0.0005 | 0.0003 | 40.00 |
| 150 | 0.0006 | 0.0004 | 33.33 |
| 200 | 0.0008 | 0.0006 | 25.00 |

## 6. Conclusion

Surface texturing of the ring surface by laser sintering technique resulted in significant improvement in wear resistance in comparison to a system with untextured samples. The following conclusions were made.

1. The effective percentage reduction in wear was found to be in the range of 25-40 in the case of ring and liner samples. Hence making oil pockets on the ring samples was found to be an effective method of reducing the wear.
2. The oil pockets area ratio should not be very large, because it could cause increase of unitary pressures and then increase of wear intensity. The smallest wear was obtained for the largest dimple depth.
3. When the test condition approaches the boundary regime, most of the finishes showed the usual boundary friction coefficient of about 0.11.

## 7. Acknowledgement

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