Friction Between Piston Ring And Liner In IC Engine

Jenishkumar P. Modi1, D.C. Gosai1, Dr. K.N. Mistry1, Principal2, Associate Professor3
1,2 SVMIT Bharuch, 3GDEC, Navsari

Abstract--- In IC engine piston ring friction losses account for approximately 20% of total mechanical losses as reported in the literature. A reduction in piston ring friction would therefore result in higher efficiency, lower fuel consumption and reduced emissions. To reduce these losses, various parametric approaches are made particularly at design stage and experimental level. The goal of this study was to develop lubricants to improve engine efficiency, without adversely affecting oil consumption, blow by, wear and cost. The variable parameters are piston velocity, engine speed, oil viscosity, gas pressure, crank angle film thickness and coefficient of friction. Non variable parameter are system constant, bore diameter, ring tension, ring width, compression ratio, reciprocating mass, piston ring area and piston ring profile. The major assumptions for developing models are either hydrodynamic lubrication theory of Reynolds equation or boundary lubrication theory. Paper shows the effect of different lubricants i.e. SAE 2T, SAE 4T, SAE 20W40 on friction force acting between the piston rings and cylinder liner of IC engine.

I. INTRODUCTION
Friction is not desirable in IC Engines as it directly affects the performance of the engine. Frictional losses are estimated in the tune of about 17-18% in various engine dynamic systems i.e., crank, bearings, piston cylinder valves, pumps, connecting rod etc. Piston-cylinder contributes frictional losses about 40- 45% of the total frictional losses. So it is important to understand the mechanism of friction of Piston Ring Assembly in IC Engine Compression rings, oil control ring, piston skirt and piston pin are the main contributing components to Piston ring assembly friction. Static ring tension, gas pressure forces and inertia force are acting on Piston ring assembly. Ring width, ring face profile, surface roughness, ring tension, ring gap, ring land width, clearance liner temperature, skirt geometry and skirt bore clearance are the major design factors over and above material property of the “pair” in Piston ring assembly.

II. LITERATURE REVIEW
A. Y. Wakuri, T. Hamatake, M. Soejima t and T. Kitahara has conducted experimental work on friction force for a piston ring pack is conducted based on hydrodynamic lubrication theory. The friction characteristics of piston rings are evaluated with frictional mean effective pressure. The instantaneous friction force of a piston assembly under firing engine conditions is measured by an improved floating liner method in which the cylinder liner is supported by means of hydrostatic bearings. The oil-film thickness of the piston ring can be calculated by applying the hydrodynamic lubrication theory based on the Reynolds equation.

B. Sung-Woo Cho, Sang-Min Choi, Choong-Sik Bae has carried out experiments with a barrel shaped piston ring in flooded oil supply in an unpressurized cylinder. A friction force measurement system using the floating liner method was developed to study the frictional behavior of piston rings. The measurement system was designed to control the effect of the secondary piston motion and to control temperatures of the cylinder wall and oil. The friction force between the barrel shaped piston ring and the cylinder liner was measured under flooded oil supply conditions. The measured friction forces were classified into five frictional modes with regard to the combination of predominant lubrication regimes (boundary, mixed and hydrodynamic lubrication) and stroke regions (mid-stroke and dead centers).

C. Edward H. Smith did the study for design of experiments can be used with a ring pack simulation program to optimize the design of a piston-ring assembly. Ten factors are varied—six describing the ring profile, three ring tensions, and the lubricant viscosity. It is shown that an improved design can be achieved that reduces ring losses by 57% whilst reducing upward oil flow by 39%. This could lead to a 7% improvement in fuel economy provided there are no deleterious effects in other parts of the engine. It has been shown that a DOE approach can be used to predict response surfaces for the performance of a piston-ring pack. Significant reductions in frictional power loss can be achieved, without increasing oil consumption by adjusting the tensions, offset ratios and curvature so for the three ring sand the viscosity of the lubricating oil.

III. MATHEMATICAL SIMULATION
Lubrication characteristics are also complex in Piston ring assembly means boundary to fully hydrodynamic lubrication and with rapidly changing characteristics during an engine cycle. Friction processes in IC Engines are complex ranging from boundary to fully hydrodynamic lubrication conditions and with rapidly changing characteristics during an engine Cycle.

In pure hydrodynamic lubrication, a sufficient amount of oil separates the two surfaces such that there is no asperity contact between them. In hydrodynamic lubrication Oil film thickness can be finding out by Reynolds theory [1]. The analysis of the hydrodynamic lubrication is treated as if it were one dimensional. The Reynolds equation which determines the oil-film pressure is given by Equation [1] as follows

$$\frac{\partial}{\partial x} \left( h^3 \frac{dp}{dx} \right) = 6 \mu \frac{dh}{dx}$$

In above given equation we can put the value of different surface profile (h) and can find out the pressure distribution along the x-direction for that profile. In this study surface profile for parabolic shape is taken as [9]

$$h = h_0 + \frac{x^2}{2R}$$
Friction Between Piston Ring and Liner in IC Engine

For hydrodynamic lubrication we can find the friction force and load carrying capacity of piston ring using below Equations. \[ F = \pi D \int_{0}^{B} \tau \, dx \]

For hydrodynamic lubrication we can find the friction force and load carrying capacity of piston ring using below Equations [6].

Boundary lubrication occurs only when \( \frac{h}{\sigma} < 4 \) in boundary lubrication friction force occurs due to asperity contact of surface. For boundary lubrication Greenwood-Tripp's model is used [7].

Friction force per unit area for Boundary Lubrication between piston ring and cylinder liner is given by the following equation [7].

\[ F = A_{asp} \times \tau \times \sigma \cdot \rho \cdot \tau \]

According to Greenwood- Tripp's model contact pressure is

\[ \rho = \left[ K_c \left( 4 - \frac{h}{\sigma} \right)^2 \right] \]

\( K_c \) = parameter dependent on material properties of ring and liner
\( h \) = oil film thickness (µm)
\( \sigma \) = combined surface roughness (µm)
\( z \) = correlation constant

Combined surface roughness given by [7].

\[ \sigma = (\sigma_{ring}^2 + \sigma_{liner}^2)^{1/2} \]

IV. ENGINE SPECIFICATIONS

<table>
<thead>
<tr>
<th>Engine Type</th>
<th>Single Cylinder, Two Stroke Petrol Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement</td>
<td>100 cm(^3) (6.10 Inch(^3))</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>10:1:1</td>
</tr>
<tr>
<td>Maximum Power</td>
<td>11.50 Hp@7500 RPM</td>
</tr>
<tr>
<td>Maximum Torque</td>
<td>8.60 N.M@6500 RPM</td>
</tr>
<tr>
<td>Cylinder Bore (D)</td>
<td>0.050 m</td>
</tr>
<tr>
<td>Stroke (L)</td>
<td>0.050 m</td>
</tr>
<tr>
<td>Connecting Rod Length</td>
<td>0.080 m</td>
</tr>
<tr>
<td>Crank Radius</td>
<td>0.026 m</td>
</tr>
<tr>
<td>Gas pressure (p(_1))</td>
<td>30 bar</td>
</tr>
<tr>
<td>Gas pressure (p(_2))</td>
<td>15 bar</td>
</tr>
<tr>
<td>Surface Roughness ((\sigma_{ring}))</td>
<td>0.182 µm</td>
</tr>
<tr>
<td>Ring Tension</td>
<td>30 N</td>
</tr>
<tr>
<td>Surface Roughness ((\sigma_{liner}))</td>
<td>3.082 µm</td>
</tr>
</tbody>
</table>

V. VELOCITY OF TEST ENGINE

Instantaneous velocity of piston is carried out by following equation [6].

\[ U = r\omega \left( \sin \theta + \frac{\lambda}{2} \sin 2\theta \right) \]

\( U \) = Instantaneous velocity of piston (m/s)
\( r \) = crank radius (m) = 0.026 m
\( \omega \) = (2Nπ)/60= angular speed
\( \Theta \) = crank angle
\( \lambda \) = con rod ratio = \( r/l \) = 0.325

Instantaneous Velocity for test engine at different speed is as follows:

Graph 1: Velocity V/S Crankangle Oil Film Thickness Of Test Engine With 2T, 4T And 20w40 Oil

Oil film thickness between piston ring and cylinder liner is varying with the velocity, load, and viscosity. As discussed above oil film thickness \( h_0 \) can be calculated by Equation [9].

\[ h_0 = \frac{2\mu R}{L} \]

From above equation we can find out the oil film thickness for different RPM of engine at different crank angle as per below graphs for different lubricants.

From equation for SAE 2T oil

Graph 2: Oil Film Thickness V/S Crankangle

For SAE 4T oil

Graph 3: Oil Film Thickness V/S Crankangle
For SAE 20W40 Oil

Graph 4: Oil Film Thickness V/S Crankangle Friction Force For Boundary Lubrication With Sae 2T, 4T And 20w40 Oil

Friction force for Boundary Lubrication between piston ring and cylinder liner is given by the

\[ F = A \eta \frac{P}{C} \]

From above equation.

Theoretical Average Friction Force For Sae 2T, 4T And 20w40 Oil

Graph 5: Average Friction Force V/S Rpm

VI. EXPERIMENTAL FRICTION FORCE

Input Parameters for Experimental Work

- Load: 75 N
- Temperature: 100 °C
- Speed: 500 RPM, 1000 RPM, 1200 RPM, 1500 RPM
- Lubricant: SAE 2T, SAE 4T, SAE 20W40 OIL
- Time: 30 min

Experimental Average friction force with 2T, 4T and 20W40 OIL

Graph 6: Average Friction Force V/S Rpm

Comparison Of Theoretical And Experimental Friction Force

Comparison of Theoretical and Experimental Friction Force with SAE 2T Oil

Graph 7: Average Friction Force V/S Rpm

Comparison Of Theoretical AND Experimental Friction Force WITH Sae 4T Oil

Graph 8: Average Friction Force V/S Rpm

Comparison OF Theoretical AND Experimental Friction Force WITH Sae 20w40 Oil

Graph 9: Average Friction Force V/S Rpm

VII. RESULTS AND DISCUSSION

It has been observed from the graph no (2) of minimum oil film thickness v/s crank angle for SAE 2T OIL that as speed of the engine increases oil film thickness also increase and it is found that value of the oil film thickness is maximum for the 1500 RPM and it is of 0.3667 µm and minimum for the 500 RPM and it is of 0.1222 µm.

It has been observed from the graph no (3) of minimum oil film thickness v/s crank angle for SAE 4T OIL that as speed of the engine increases oil film thickness also increase and it is found that value of the oil film thickness is maximum for the 1500 RPM and it is of 0.84 µm and minimum for the 500 RPM and it is of 0.2799 µm.

It has been observed from the graph no (4) of minimum oil film thickness v/s crank angle for SAE 20W40 OIL that as speed of the engine increases oil film thickness also increase and it is found that value of the oil film thickness is maximum for the 1500 RPM and it is of 1.12 µm and minimum for the 500 RPM and it is of 0.437 µm.
thickness is maximum for the 1500 RPM and it is of 0.1445 µm and minimum for the 500 RPM and it is of 0.04817 µm.

It has been observed from the graph no (5) of Theoretical Average Friction force v/s RPM that as speed of the engine increases friction force also decrease and it is found that value of the maximum friction force for 500 RPM and minimum friction force for 1500 RPM and it’s value is 9.6 N and 9.23 N for 2T OIL, 9.56 N and 8.63 N for 4T OIL, 9.98 N and 9.68 N for 20W40 OIL respectively.

It has been observed from the graph no (6) of Experimental Average Friction force v/s RPM that as speed of the engine increases friction force also decrease and it is found that value of the maximum friction force for 500 RPM and minimum friction force for 1500 RPM and it’s value is 10.07 N and 8.5 N for 2T OIL, 11 N and 8.51 N for 4T OIL, 11.5 N and 8.99 N for 20W40 OIL respectively.

It has been observed from the graph no (7) (8) (9) of Average Friction force v/s RPM that value of theoretical friction force and experimental friction force for SAE 2T, 4T, 20W40 are about the same for different RPM.

VIII. CONCLUSION

Under the Mathematical model proposed and simulation along with result comparison is made with experimental work for different lubricants

Oil Film Thickness

Oil film thickness depends on the various parameters i.e gas pressure, ring tension etc. here from results we can conclude that as the speed increase oil film thickness also increase. Friction Force

The nature of the simulated result of ring friction force at different crank angle is in the form of the sinusoidal nature. As the piston in the downward motion, the ring friction force acting in the upwards direction during the half working cycle. The next half cycle, it shows negative direction. Friction force between piston ring and cylinder liner is linearly decreasing as speed increases as shown in graph of friction force v/s crank angle. And average friction force is also decreases as speed increases. By comparing the friction force with SAE 2T, 4T, AND 20W40 oil, Average friction force with SAE 4T is less compared to others with same engine.

REFERENCES


