Effect of piston ring geometries on lubrication and friction in diesel engine

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Abstract. Lubrication and friction in piston rings are of a greater importance which has been receiving significant attention from tribologists for a long time. The main subsystems which contribute to friction include the ring-pack/liner, piston-skirt/liner, piston-pin/connecting-rod and connecting rod-crankshaft bearings. Piston rings are used in engines mainly to reduce leakage of gas in the combustion chamber and simultaneously provide a lubrication film in order to reduce friction during its motion. Wear between piston rings and liner is inevitable and therefore in order to reduce it, different types of ring geometries are being used which has its own advantages and disadvantages. Another major problem being faced during operation is that oil from the liner gets transported by pumping, reverse blow by, inertia and squeezing of oil due to ring dynamics into the combustion chamber. The present work focuses on to investigate different ring geometries and their effects on friction and lubrication oil consumption. It was found that Tangential force has an inverse relationship with Oil Film Thickness and Oil filling ratio. Top ring friction is high at Top dead centres due to boundary lubrication. Ring Barrel height has a direct relationship with asperity friction and an inverse relationship with Oil Film Thickness. Axial width is directly proportional to the wear of rings, which is due to the area exposed to back pressure on the rings exerted by gases from combustion chamber. Second ring closed gap has a direct relationship with Inter-ring pressure thus increasing reverse blowby. Rings with a Positive twist are found to be stable during operation. For this analysis, AVL Excite software is being used and results are validated with an experimental model.

1. Introduction
Modern IC Engines are expected to strictly adhere to efficiency standards and to be eco-friendly. Among the several factors, oil consumption and frictional power losses play vital roles in the performance of IC engines. Hence a good understanding of lubrication condition at piston ring-cylinder liner interface is vital in determining the source of frictional losses and consumption of lubricating oil. The wear behaviour in the top ring of a piston found that lubricant starvation in piston rings leads to surface damage and wear [1]. An analytical was developed a method for determining the bore wear pattern of the engine, which included ring geometries, film thickness, Archard's wear and

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thrust load [2]. A model of ring considering axisymmetric conditions was developed which helps in analysis of various parameters such as ring profile and oil flow [3]. The effect of wear due to piston secondary motion was studied and also due to ring twist by considering the influence of pressure distribution [4]. The study concluded that wear of corner of the ring/groove causes a difference in pressure above and below the rings. Study on the out-of-roundness error of the cylinder bores of the top compression ring was also carried out. The influencing parameters contact, lubrication, friction, power loss due to the variation of bore asymmetry is considered [5]. Further, the study also includes new and run-in ring profiles and topography. A rheological model of Newtonian lubricant, which is used to find the oil film thickness between elastohydro-dynamically lubricated smooth or rough surface. The asperity behaviour of the model with respect to the oil film is also investigated [6]. The effect of surface roughness in reducing friction forces of piston and cylinder contact was carried out [7]. The study reveals that brake means the effective pressure is directly dependent upon piston friction losses. The effect of surface texturing on transition in the regime of lubrication was also analysed. The study has been done by pin-on-disk apparatus, with results indicating transitions on Stribeck curve [8]. The lubrication regime transitions of piston ring-cylinder liner contact through a numerical and experimental model of Reynolds equation and film thickness equation subjected to suitable boundary condition. A comparison of this study between experimental and analytical results was also carried out which shows good agreement [9]. The influence of ring twist which affects the blowby of the engine, inter-ring pressures and also the study suggested that a special attention has to be paid to the second ring motion at the middle of intake and at the end of compression and exhaust stroke [10]. Over the majority of the stroke, the piston rings operate in conditions of starved lubrication. The thickness of lubricating oil film on the piston is related to the oil consumption and piston friction loss [11]. The present work focuses on the study of piston ring design by considering blow by, lubrication oil consumption and wear of rings. The present work focuses on the study of piston ring design by considering blow by, lubrication oil consumption and wear of rings.

2. Experimental Validation

As shown in Figure 1 b), the experimental setup consists of a 1.9L 4-stroke Diesel engine of bore 80 mm and stroke 92.8 mm. The ring surface has a surface roughness of $R_a = 0.2$ nm and the liner has a surface roughness of $R_a = 0.35$ nm. The oil used in both setup and simulation is SAE 10W30. Table 1 shows the properties of the ring used in the experiment. Figure 1 a) shows the simulated model with rings and liner. The main characteristics of AVL Piston Ring Dynamics Module are [12]

1. Each ring is modelled as a single mass. The interaction between the thrust and anti-thrust sides is given by a beam model and a model for pressure compensation.

2. The friction between rings and liner is total friction including the viscous friction due to shearing lubricant given by Reynolds equation and Greenwood–Tripp asperity contact gives friction at contact points.

3. The modelling of piston ring includes moments and forces for each ring. The hydrodynamic pressure distribution between the ring running surface and liner is found by solving Reynolds’ equation in each time step.

4. The volumes are connected due to the actual clearances of ring end gaps and the actual position of the rings in the grooves. The possible gas flow behind the rings and between ring and groove flanks is considered.

5. The oil film is taken into account between the ring running surface and liner by calculating the pressure distribution in the clearance according to the liner and ring contours.

| Table 1. Ring Properties |
|--------------------------|
| **Properties of ring**   | **Values** |
| Liner height             | 188 mm    |
| Liner thickness          | 8 mm      |
| Mass of top ring         | 8.5 g     |
Mass of middle ring 7 g
Mass of oil ring 5 g
Hardness of piston rings 1200 MPa
Average temperature of top ring 220 °C
Average temperature of middle ring 180 °C
Average temperature of oil ring 170 °C

Some of the common methods used for determination of lubricating oil consumption are Drain plug discharging, Flow-metering using sensors, Weighing of oil source and Scaling of oil consumption [13]. The drain plug is the least approximate method because of oil being sticking to the surfaces increasing the error. The setup consists of oil source placed externally with scale and by taking the difference in values after running for required time, oil consumed is determined. Table 2 shows the difference between experimental and calculated values of Lubricating oil consumption. The errors associated are less than 10% which shows the model is simulated correctly.

Figure 1. (a) Simulated Model; (b) Experimental Setup

| Running Rpm (r.min⁻¹) | Calculated values (g.kW⁻¹.h⁻¹) | Experimental values (g.kW⁻¹.h⁻¹) | Relative error between values in % |
|-----------------------|---------------------------------|-----------------------------------|----------------------------------|
| 500                   | 0.776                           | 0.856                             | 9.34                             |
| 1000                  | 0.744                           | 0.819                             | 9.15                             |
| 2000                  | 0.573                           | 0.624                             | 8.17                             |
| 3000                  | 0.657                           | 0.713                             | 7.85                             |
| 4000                  | 0.684                           | 0.745                             | 8.18                             |

3. Results and Discussion
3.1. Influence of the tangential force of top ring
Tangential force occurs when the ring is in a compressed state inside a liner and due to this the ring exerts force all over the circumference of the liner. The created force generates the contact pressure between the rings and liner surface which leads to increase in friction and wear. In order to determine the effect, the study considers three trials of different tangential force 9 N, 12 N and 15 N for simulation of top ring. From Figure 2, it is inferred that due to change in tangential force, the oil film thickness layer also changes which is seen at crank angles 0°, 180° and 540°.
Oil filling ratio or Oil filling percentage is the amount of oil being collected downstream in the next groove during its motion. Figure 3 shows that oil filling ratio has an inverse relationship with tangential force. The excess oil film increases the hydrodynamic friction and also contribute to oil throw-off. Hence the tangential forces have to be varied according to the oil filling ratio because excess high oil film makes oil being carried to the next ring and increases the chances of oil throw off [14]. When considering the frictional forces, it is evident that increase in tangential force has the direct effect of increasing friction between ring faces and walls [10] such as at crank angles 270° and 450° which is shown in Figure 4.

3.2. Study of Asymmetric and Symmetric barrel faced Top Ring

Further, the study also investigates the friction power loss specific for the three rings. From figure 5 top ring friction losses are high only at the top dead centre because of boundary lubrication being present there and temperature is also high which decreases the oil viscosity [12, 15, 16]. In boundary lubrication, there is metal-metal contact between ring and liner compared to metal-oil film contact at other parts. Due to this, friction is highest at top dead centre (0° crank angle). Hence possible methods to reduce top ring friction were also identified.
The OFT (Oil Film Thickness) of the flatter ring profile is high at the dead centers and increases slowly during the piston acceleration. The OFT is dependent on the viscosity of the oil, higher the oil viscosity higher will be the OFT. Here viscosity of oil is maintained constant everywhere. But in the case of curved rings, the OFT is comparatively less at the dead centers and increases rapidly as the stroke progresses. The average friction produced by the flatter ring is higher than that of the curved ring for the entire engine cycles. Figure 6 shows the profile of symmetric and asymmetric (skewed) barrel ring. Also the asperity contact is predominantly influenced by the gas pressure acting on the back side of the rings as the top is exposed to high pressure gases [17].

From Figure 7 it is found that the friction reduces for asymmetric ring compared to symmetric rings. Also on considering only the asperity friction which is the dominant factor in the top ring friction, it is inferred that the friction reduces by about 50% from 150 N to 100 N. This large reduction in asperity friction is due to the fact that point of contact of asymmetric barrel rings is away from the center of the axis of the ring due to which the pressure difference acting on top and bottom side of the ring reduces [18]. As the asperity friction reduces, the wear of the rings also reduces making the ring to last longer [19].

![Figure 7. The friction of a) Symmetric ring; b) Asymmetric ring](image)

Also from Figure 8, it can be seen that the OFT of asymmetric is less than symmetric. It is due to the fact that the asymmetric barrel point is placed at the lower edge facing downwards so that naturally the oil scraping property of this ring higher and oil does not flow again into the groove through the bottom. Therefore the lubrication oil consumption (LOC) is greatly reduced [20].

![Figure 8. Oil Film Thickness of a) Symmetric ring; b) Asymmetric ring](image)

### 3.3. Effect of top ring barrel height

Barrel height is the total length of the ring curvature which a prime design parameter in the piston ring which is shown in Figure 9. Increasing the ring height will have an impact on the OFT. In order to study the effect of ring height, three trials are simulated for 0.01 mm, 0.03 mm and 0.045 mm of ring barrel height.

![Figure 9. Top ring showing barrel height and axial width](image)

From Figure 10 it is found that the large barrel height has high average friction and it is mainly due to increase in asperity friction. In addition to that, it was also found that the asperity friction is
dominant at crank angle 0° due to lubricant starved regime at dead centres called Ring reversal points [19]. At that points, high wear happens due to metal-metal contact between piston ring and Liner.

Figure 10. Total friction for barrel height of a) 0.01 mm; b) 0.03 mm; c) 0.045 mm

From Figure 11, it is seen that the oil film thickness gets reduced as barrel height increases. Hence less OFT makes ring-liner direct contact at dead centres contributing to increasing asperity friction. So barrel height of every ring is dependent on the oil viscosity and liner properties of the engine [21].

Figure 11. Oil Film thickness for barrel height of a) 0.01 mm; b) 0.03 mm; c) 0.045 mm

3.4. Influence of axial width of top ring
Axial width is the total height of ring measured in the y-direction which is already shown in Figure 10. The axial width has a direct impact on the ring weight as its main dimension is reduced. The area of contact is the root of friction and its effect is being studied by three samples of the different axial width of top rings such as 1 mm, 1.3 mm and 1.5 mm. From Figure 12, there is a reduction in friction as axial width gets reduced which is due to the fact as the area of contact gets reduced [11]. Obviously, as the friction gets reduced, wear of the ring also gets reduced.
Also, the reason which restricts the use of very thin rings is that as axial width reduces, the mass of the ring also gets reduced hence the ring gets lifted for more duration during operation making it vulnerable to oil transport through the gaps. Due to this the number of hydrocarbon in effluent increases and LOC also increases which are not recommended.

3.5. Effect of Second Ring Closed Gap

Closed Gap (CG) is the amount of gap present between two ends of the ring when it is in a compressed state inside the cylinder. The top ring CG has a very much direct effect on blowby. The top ring CG is made as minimum as possible but not too close so that the ring end gets joined due to thermal expansion. The study investigates the effects of a second closed gap on LOC for three trials (0.55 mm, 0.35 mm, 0.25 mm) with top ring CG kept constant as 0.35 mm.

Inter-ring pressure is influenced by the gases flowing from the top ring through end gap, running face, grooves and also by sealing efficiency of both the rings. Inter-ring pressure is not allowed to reach high, because during exhaust stroke due to high pressure it tries to lift the top ring from its lower groove position, which is noted as ring flutter. It is not desirable because due to lifting, gases flow from crevices towards the combustion chamber and it makes the oil to flow with them towards the chamber consuming the lubrication oil. This condition is termed as reverse blowby. Also due to continuous lift and drop of top ring, they are prone to breaking at high speeds which is undesirable. Hence in order to eliminate this effect, 2nd ring gap is always made higher than a top ring to create an easy flow path.

From Figure 13, it is found that the inter-ring pressure goes on increasing as decreasing the 2nd ring closed gap because of path restriction for gas flow. It is also found that the pressure builds up after 0° crank angle and between 360° crank angle (expansion process and exhaust process) due to constrained flow for gases. Due to this pressure build up, top ring flutter happens at exhaust process which lifts the top ring from the lower groove position. From Figure 14, it is found that Top ring does...
not lift at high (0.55 mm) 2nd ring closed the gap as pressure build is low and also lift duration increases as the closed gap decreases.

**Figure 14.** Axial Position – Lift of Top ring for 2nd Ring closed gaps of 0.55 mm, 0.35 mm and 0.25 mm

**Figure 15.** Oil throw off for 2nd ring closed gap of 0.55 mm, 0.35 mm and 0.25 mm

Due to this flutter, the axial contact between ring and groove do not occur. The force of contact between the top ring and groove increases as closed gap decreases, due to this the top rings are prone to early ring breakage because of cyclic loading (fatigue failure). From Figure 15 it is also found that the oil throw off is higher for low CG rings. Due to gas flow from the 2nd ring to the combustion chamber, so called reverse blowby occurs, which makes oil to flow from top ring groove to piston top land. Due to this accumulation of oil, they are thrown towards combustion chamber due to the inertia of movement [14].

### 3.6. Effect of Twisting in Second Ring

Rings are chamfered at inside on bottom or top side due to this change in area, forces act upon the ring causing them to twist inside the groove. As the bevel cut is made inside the ring they are also called as internal bevel rings. Based on the geometry of the ring, there are two types of twisting – positive and negative twisting, each has its own advantages. As in Figure 16, the rings are beveled at an angle of 45°, based on the position of bevel the twisting occurs. If the ring is beveled on Top inside edge, then it is positive twisting and if the ring is beveled on the bottom inside edge, then negative twisting occurs. Ring twist is obtained by the moment balance, and this moment is due to forces from ring’s axial and radial forces.

**Figure 16.** Ring profile of a) positive twisting ring; b) negative twisting ring

From Figure 17 it is very clear that negative ring is erratic compared with the positive ring. The reason is due to point of contact of rings and gas dynamics that happens during the motion. As the flow area changes due to the gas forces, it induces the twisting effect in the rings. With a positive twist ring, the ring at the top of the groove has less exposed surface area which creates less force for pushing the ring down. Hence the stronger inertia force makes the ring stay at the top of the groove [24]. When the ring is at the bottom of the groove, the high exposed surface area makes the ring stays close to the lower groove. Hence the movement is stable in positive twist rings [22, 25].
One major drawback that was with positive rings is the consequence of reverse flutter of the top ring [17, 25]. The lift duration of the top ring is high, when the 2nd ring used is positive twist ring. As a consequence, the oil consumption also increases due to flowing into the combustion chamber. But in case of negative twist rings, the bottom side of ring contacts at lower groove restricting the flow of oil into the groove as shown in Figure 18. Hence all scraped oil must flow downwards into the combustion chamber, hence lubrication oil combustion is improved due to no reverse flutter.

4. Conclusion
The piston ring design was simulated to study its influence on wear and lubrication. The design parameters which were being studied includes tangential force, asymmetric profile, barrel height, axial width, closed gap and twist of rings. The inferences were

- The tangential force has an inverse relationship with Oil Film Thickness and oil filling ratio. Top ring friction is high at dead centers due to boundary lubrication.
- By implementing asymmetric profile, the overall friction reduces due to asperity friction which is influencing parameter in dead centers and hence the wear also reduces. Also by using this profile, the Oil Film Thickness is also reduced towards crankcase.
- Increase in barrel height increases asperity friction and reduces hydrodynamic friction. Also, there is a reduction in Oil Film Thickness towards the crankcase.
- As axial width reduces, the exposed area to gases reduces and hence wear of the ring also reduces. Due to a reduction in axial width, the mass of the ring is decreased and it has increased effect on duration of ring lift making it vulnerable to reverse blowby.
- If second ring closed gap is lower or same as first ring closed gap, Inter-ring pressure above second ring builds up making the top ring to flutter and causes reverse blowby.
- It is found that Negative twisting rings are unstable compared to positive rings due to exposed area to combustion gases. Also when using positive rings gas flow downwards is restricted causing the pressure build up and have a reverse flutter.

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