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RESEARCH

# Wear Analysis of Top Piston Ring to Reduce Top Ring Reversal Bore Wear

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*Top ring*
*Wear depth*
*Boundary lubrication*
*Skewed barrel ring*
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A B S T R A C T

*The piston rings are the most important part in engine which controls the lubricating oil consumption and blowby of the gases. The lubricating film of oil is provided to seal of gases towards crankcase and also to give smooth friction free translatory motion between rings and liner. Of the three rings present top ring is more crucial as it does the main work of restricting gases downwards the crankcase. Boundary lubrication is present at the Top dead centre (TDC) and Bottom dead centre (BDC) of the liner surface. In addition to this, top ring is exposed to high temperature gases which makes the oil present near the top ring to get evaporated and decreasing its viscosity, making metal-metal contact most of the time. Due to this at TDC, excess wear happens on the liner which is termed as Top ring reversal bore wear. The wear rate depends upon many parameters such as lubrication condition, viscosity index, contact type, normal forces acting on ring, geometry of ring face, surface roughness, material property. The present work explores the wear depth for different geometries of barrel ring using Finite Element model with the help of Archard wear law and the same is validated through experimentation. The study reveals that Asymmetric barrel rings have less contact pressure which in turn reduces the wear at Top dead centre.*



## 1. INTRODUCTION

The friction between piston rings and liner causes wear of both surfaces over a span of time. Friction has its own applications like in Clutches and Brakes. Amount of friction depends upon the Tangential force of ring and gases acting on back side of ring. Oil ring has highest amount of friction due to expander ring present. Wear is defined as 'the progressive loss of material from the surface of a solid body due to mechanical action, i.e. the contact and relative motion against a solid, liquid or gaseous counter body' [1]. The first theory on lubrication between piston and liner was given by Castleman [2]. Another important study on lubricant film thickness was given by Elion and Saunders [3]. Ting and Mayer [4,5] developed an analytical model for determining the ring-bore wear mechanism for a reciprocating piston engine over a complete running cycle. The model used hydrodynamic lubrication theory to analyze the flow between ring and cylinder bore. The model also includes geometry of the ring, blow-by through the piston ring pack, minimum film thickness and wear using Archard law. Rohde et al. [6,7] used averaged Reynolds equation with the half-Sommerfeld boundary condition obtained by Patir and Cheng [8,9] and Asperity interactions by Greenwood-Tripp model [10]. The study reveals that piston ring friction is dependent on surface topography during its contact when it is in the mixed lubrication regime. Hu et al. [11] developed a non-axisymmetrical analysis technique with Greenwood-Tripp model for asperity contact including the effects of surface roughness and surface flow factors. The study concludes that that the contact pattern and film thickness between piston ring and cylinder bore are not exactly symmetric about the cylinder axis. Further the study also predicted that the frictional force between piston ring-cylinder by using viscous shearing force of the hydrodynamic film. Dowson et al. [12] used a hydrodynamic lubrication model to obtain film thickness measurements, study lubricant transport, and viscous friction at the piston ring-cylinder liner interface. Sanda et al. [13] developed a mixed lubrication model employing inlet boundary conditions based on laser-induced-fluorescence (LIF) oil film thickness measurement. The study compared the predicted and experimental lubricant film thickness and friction force. The study developed a numerical method to predict frictional performance of piston ring and cylinder bore contact operating in the mixed lubrication regime, considering lubricant rupture location, surface flow factors and metal-metal contact load. In this study, Patir and Cheng's [8,9] modified Reynolds equation, including surface roughness and surface flow factors and Greenwood and Tripp's [10] statistical surface asperity approach has been used to model piston ring and cylinder bore contact. Ahmed et al. [14] modelled to evaluate the effect of piston ring dynamics on friction force, frictional losses, and oil film thickness. Dynamic analysis software Excite Designer by AVL was used. The software considers variations in lubricant viscosity with respect to temperature using Vogel's equation and oil consumption.

### 1.1. Top Ring Reversal Bore wear

P.S. Dellis [15] found that the piston rings near the dead centres the friction is of maximum due to mixed lubrication. Also due to high temperature present at the TDC, oil viscosity decreases making oil squeezing force to decrease and hence asperity contact becomes the dominant one determining friction. Another similar modelling was done by Mishra et al. [16] found that at TDC and BDC has the least oil film thickness and highest friction was found at TDC. The model was validated with experimental results obtained from Furuhama and Sasaki [17]. Bolander et al. [18] in his study of transition of lubrication regime in the piston ring-Liner surface found that at Dead centres, mostly boundary lubrication is present and asperity-asperity contact is there at TDC. Study was experimentally validated with analytical results showed good correlation. Mishra [19] found that in the engine cycle at the dead centres, motion reversals take place resulting in a mixed regime of lubrication. In the vicinity of the top dead centre a higher combustion force can contribute to an increased contact force and reduction in lubricant viscosity. Due to this the back pressure by oil also decreases. Tribodynamic analysis including the effect of generated temperature for the high pressure region round the TDC reversal from compression to power stroke was provided by Mishra [20]. Morris et al. [21] presented an analytical thermal model and numerical solutions for an average flow model. From the study, it was concluded that friction losses are mainly contributed by viscous shear under low temperature conditions, but boundary friction losses dominate at high temperature operating conditions such as at TDC of engine. The model is based on lubrication theory considering variation of lubricant viscosity with temperature. Rahmani et al. [30] included the out-of-roundness of cylinder bores profile. A multi lobed rough conjunction is used and lubricants flow is predicted for full engine cycle. The effect of wear on tribological characteristics were also included in the study. Zavos et al. [31] used Navier Stokes equation and experimental data from 2 Stroke engine. Surface irregularities are measured by using Coordinate Measuring machine (CMM). From their study it was found that for worn out profile, friction reduces but the mechanical stresses in the ring were increased.

### 1.2. Wear Model

Wear classification for steels over the wide range of loads and sliding velocities was given by Lim and Ashby [22]. They worked on simplified wear equations and based on the data from a large number of dry pin-on-disc experiments, a wear map is plotted giving the contours of wear regimes and the dimensionless wear rate as a function of dimensionless normalised pressure and dimensionless normalised velocity.

The widely used wear model for mild wear is $Q=Kp$, where the volume wear rate is proportional to the normal load. This model is often referred to as the Archard's wear law. But the basic form of wear model was introduced by Holm [23].

$$\frac{V}{S} = K\,\frac{F}{H} \tag{1}$$

The Eq. (1) is called Archard wear model [24] where $V$ is the wear volume (m$^3$), S is sliding distance (m), $K$ is dimensionless wear coefficient (less than unity), $F$ is normal force acting (N) and $H$ is the hardness of softer contact surface (Pa). In engineering applications the wear depth is of more interest, hence Archard proposed to divide both sides of Eq. (1) by the apparent contact area $A$ giving:

$$\frac{V}{As} = \frac{h}{s} = kp \tag{2}$$

Where $h$ is the wear depth (m), $k$ is dimensional wear coefficient ( Pa$^{-1}$ or m$^2$/N ) where $k = K/H$ and $p$ is the normal contact pressure (Pa). The wear process at any localized point can be formulated as:

$$\frac{dh}{ds} = kp \tag{3}$$

The simulation routine is implemented in ANSYS and looped through a series of static structural analyses of the FE model, each with an updated geometry of the liner surface. Each step starts with a static structural analysis that computes the contact pressure at each of the surface nodes. With the nodal contact pressure known, the wear during the step is computed by numerical integration of Eq. (3) according to the explicit Euler integration scheme [25]. The wear depth at node number $i$ after the $j^{th}$ simulation step is then:

$$h_{i,j} = h_{i,j-1} + \Delta h_{i,j} \tag{4}$$

where $h_{i,j-1}$ is the wear depth after the $(j-1)^{th}$ step, $\Delta h_{i,j}$ is the incremental nodal wear depth during the $j^{th}$ step, computed based on the nodal pressure $p_{i,j}$ and the incremental nodal sliding distance $\Delta s_{i,j}$ according to:

$$\Delta h_{i,j} = k p_{i,j} \Delta s_{i,j} \tag{5}$$

Due to large pressures at TDC, higher temperature exposed over the rings, change in viscosity with respect to temperature and boundary lubrication causes high wear rates between ring and liner. As shown in Fig. 1, bore wears only at TDC making it prone to gas blowby, oil blowback into combustion chamber and ring breakage.

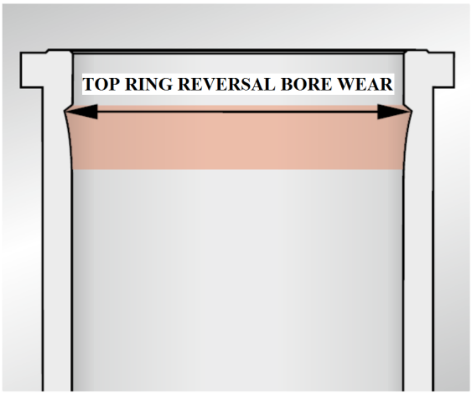



**Fig. 1.**Top ring reversal bore wear.

Main focus is to reduce Top Dead reversal bore wear and effect of different geometries of asymmetric barrel rings on wear rate.

## 2. EXPERIMENT AND MODEL VALIDATION

The piston ring-liner is modelled using CATIA and imported into ANSYS Workbench. In Dead centres, metal-metal contact is present, hence the ring-liner is modelled as dry sliding contact. Axisymmetric analysis is used as both ring and liner are symmetrical about its axis (Fig. 2). As in Fig. 3, Augmented Lagrange contact algorithm and Friction contact type ($\mu$ = 0.5) is used between ring-liner. Pin-on-Disc experiment was done for the same dimensions of the ANSYS model to find the total volume of wear. Figure 6 shows the mass loss due to wear after the running. The Total volume of wear is found by the Eq. (2) for ANSYS model. The wear rate [26] was found by Eq. (6):

$$K\ (mm^3/Nm) = \frac{\text{Mass Loss}\ (mg)}{\rho(mg/mm^3)\ F(\text{N})\ d(m)} \tag{6}$$

**Table 1.** Properties of Ring and Liner.

| Property | Piston Ring | Liner |
|---|---|---|
| Material Type | Steel C15 | Steel C30 |
| Poisson's Ratio | 0.3 | 0.3 |
| Hardness in BHN | 130 | 180 |

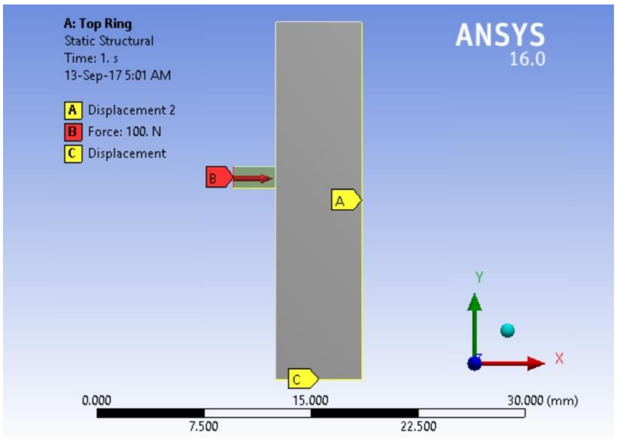



**Fig. 2.** Ring-Liner Axisymmetric model.

The boundary conditions applied on the axisymmetric model are force on back side of piston ring, X-axis displacement constrained on outer diameter of Liner and Y-axis displacement constrained on bottom side of both ring and Liner. The height and width of liner is 0.15 m and 0.1 m. Similarly the height and width of piston ring is 0.034 m and 0.05 m. The element used in model is PLANE 182 which is 4 Node structural solid element.

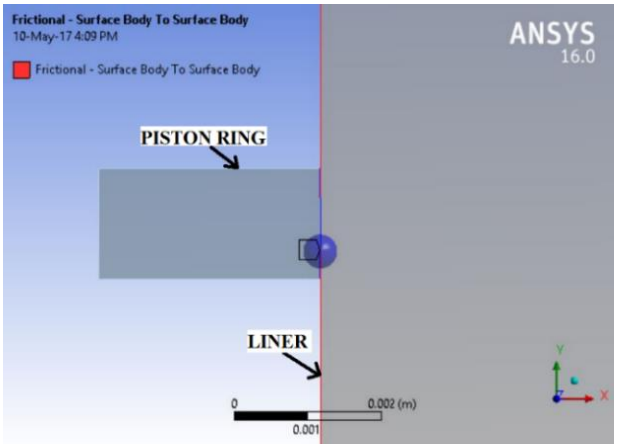



**Fig. 3.** Contact in ring-liner model.

The contact type used is Frictional contact with μ = 0.5 and the piston ring is modelled as target and liner is modelled as contact. The number of nodes and elements in Piston rings is 1458 and 593. Likewise number of nodes and elements in Liner is 27650 and 7921.

Figure 4 shows the cylindrical pin that is used for determining the wear rate. Figure 5 shows the experimental setup of Pin on disc from which volume of material reduced for particular length of running is calculated. The surface conditions, rpm of disc, duration of running, Load on pin and lubrication determines the amount of wear. Dry conditions without oil is carried out as the model is metal-metal contact. Different iterations are done to determine average wear rate of pin.


**Fig. 4.** Mild Steel Pin.

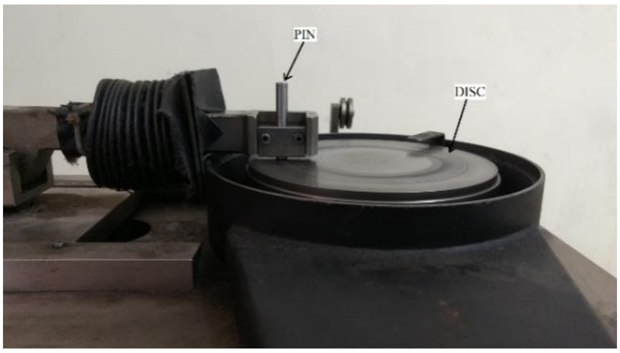



**Fig. 5.** Pin-on-disc experiment.


a)


b)

**Fig. 6.** a) Mass of Pin before running and b) Mass of Pin after running.

**Table 2.** Experimental values from Pin on Disc.

| Test | Mas Loss (mg) | Force (N) | Rpm | Time (s) | Distance (m) | Wear Rate of pin |
|---|---|---|---|---|---|---|
| 1 | 0.0033 | 200 | 200 | 220 | 230.1 | 0.9137E-5 |
| 2 | 0.0041 | 250 | 200 | 220 | 230.1 | 0.9082E-5 |
| 3 | 0.0064 | 250 | 300 | 220 | 345.4 | 0.9347E-5 |
| 4 | 0.0075 | 300 | 300 | 220 | 345.4 | 0.9220E-5 |

Wear rate of pin is found by Eq. (6):

$$K = \frac{0.0033}{7850\ X\ 10^{-6}\ X\ 200\ X\ 230.03}$$

$$K = 0.91 \times 10^{-5}\ mm^3/N\ m$$

Wear rate of pin is taken as $K_s$ = 0.92 X $10^{-5}$ $mm^3/N$ m by averaging tests. Total volume of wear from Pin on disc experiment is 0.42038 $mm^3$ and for Finite Elemental model 0.41659 $mm^3$. The error is 0.9 % which is acceptable. The average wear rate of pin is 0.9192 $mm^3/N$ m. Wear on disc is neglected as the amount of wear on disc is negligible compared with wear on pin.

$$\%\ of\ error = \frac{0.42038-0.41659}{0.42038}\ X\ 100 = 0.9\ \%$$

## 3. SKEWED BARREL RINGS

In Skewed barrel rings, the curvature of the ring is not symmetrical but it is towards the bottom side of the ring. Due to this, the high pressure difference acting on the area reduces [27] and friction between Ring-Liner also decreases. Hence the net normal force that will be acting on rings reduces [28]. Due to this less amount of force on back side of ring reduces friction between ring and liner. Hence the wear on pin also reduces because of less amount of normal force.

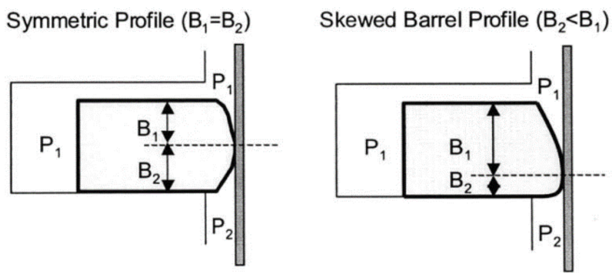



**Fig. 7.** Symmetric and Skewed Barrel rings [27].

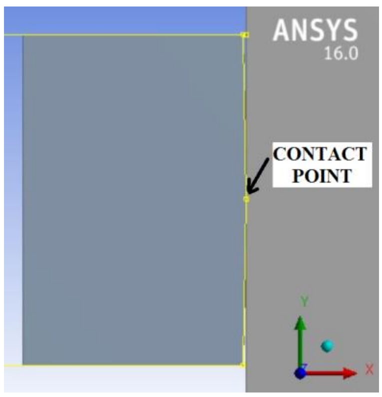
The friction force acting on the top ring is pure metal-metal contact due to high temperature of gases acting from combustion chamber and it is the product of the radial load acting on the back of the ring and the coefficient of friction given by [27]:

$$F_f = a_{asp}[\,(p_1 - p_2)B_2 + W] \qquad (7)$$

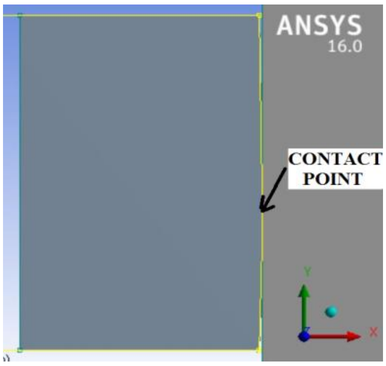
where $a_{asp}$ is the coefficient of friction between the ring-liner and W is the ring load due to tension. Three different models of Asymmetric barrel top rings are modelled and their contact pressures have been found. Based on the force acting on back side of ring, wear reduces.

## 4. ANALYSIS AND RESULTS

The ring and Liner contact point is a very important factor as it also facilitates easy removal of oil from liner surface. Static Structural analysis of ring-liner is carried out. Figure 8 shows the contact points of three barrel rings for $B_2/B_1$ ratios of 1, 0.6, and 0.3 in ANSYS workbench.




a)

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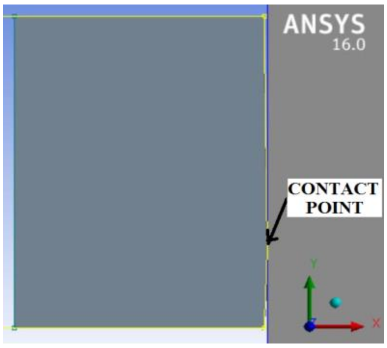
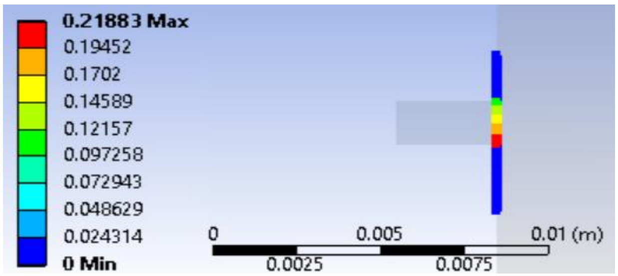



c)

**Fig. 8.** Ring-Liner contact for Barrel rings with $B_2/B_1$ ratio a) 1, b) 0.6 and c) 0.3.

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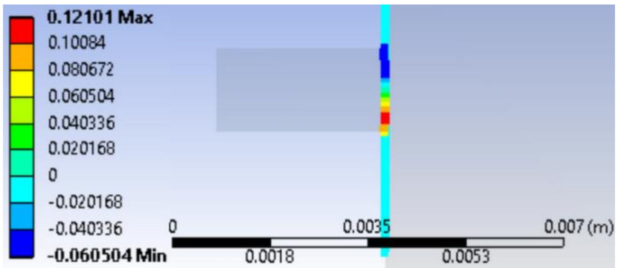


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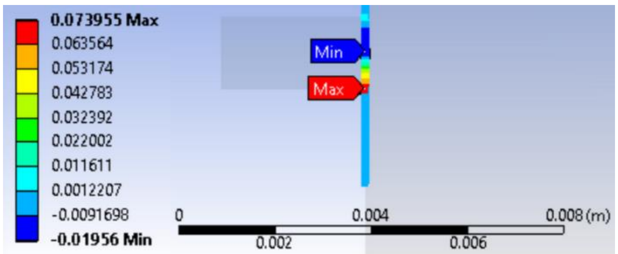


c)

**Fig. 9.** Contact pressures for Barrel ring with $B_2/B_1$ ratio a) 1, b) 0.6 and c) 0.3.

Figure 9 shows the contact pressures of three different geometries of ring and liner. This is due to the reduction in frictional force of the ring by Eq. (7). By Archard wear law, the wear depth $h$ is directly proportional to the contact pressure $p$.

**Table 3.** Wear depth and Contact pressure of three Barrel rings.

| Type of ring | $B_2/B_1$ ratio | Max contact pressure (Pa) | Wear depth per sliding distance (dh/ds) |
|---|---|---|---|
| Barrel | 1 | 0.21883 | 2.008E-3 |
| Asymmetric 1 | 0.6 | 0.12101 | 1.113E-3 |
| Asymmetric 2 | 0.3 | 0.073955 | 6.803E-4 |

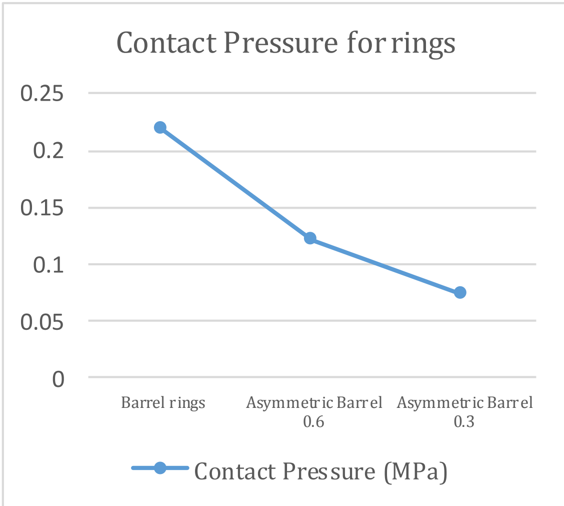


**Fig. 10.** Contact pressure for three rings.

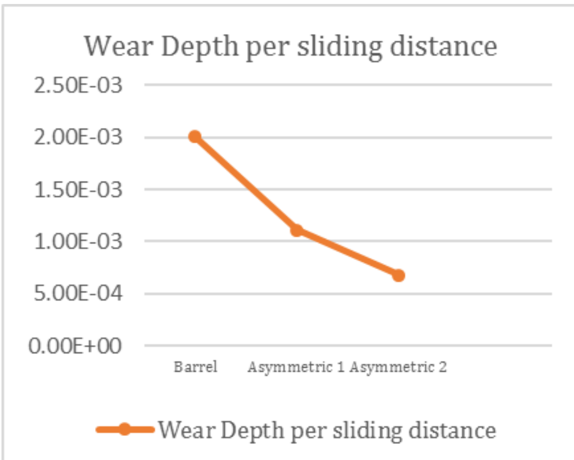


**Fig. 11.** Wear depth per sliding distance for three rings.

Figures 10 and 11 shows how Contact pressure and Wear depth are reduced as the point of contact for asymmetric barrel rings are shifted downwards. Also less $B_2/B_1$ ratio facilitates easy removal of oil from liner surface. Hence the Top ring reversal bore wear is being reduced.

## 5. CONCLUSION

Asymmetrical barrel rings is making less contact pressure with liner, due to this the major friction parameter which is asperity friction is being reduced. Because of this asperity friction, wear on liner becomes less and Top ring reversal bore wear reduces. Nowadays Top groove is given tilt, to shift the point of contact same as asymmetrical rings to reduce friction. Also new rings are made with reduced axial width. The geometry of asymmetrical ring is difficult to manufacture and there is scope in that part. In this study, Surface characteristics of ring and thermal effects are not considered. Study including above said effects with experimental analyses can be studied on rings in the future.

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