The Refinement of the contact compression ring chamfer for race engine conditions

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Abstract

In this paper we present a design modification to the piston ring coating chamfer, for stress reduction of this feature during operation. A common problem that piston ring have during operation is that they are designed to create a seal mean they will suffer a large levels of material loss due to wear. In this paper a method of reducing the active stress on the ring is discussed.

Keywords

Piston ring, coating, design and shear

1. Introduction

Combustion engines have been extensively researched since Ramsbottom's pioneering work in 1854 [1] on steam engines using a piston ring on piston. In 1962 Miller [2] made enhancements on this design which allowed for the pressure to also act on the backside of the ring resulting in an improved performance. Numerous improvements have been made in the materials and design of cylinder, piston and piston ring. Piston was redesigned to enable the piston ring to sit closer to the crown of the piston resulting in improved fuel consumption. Developments have also been rapid in engine oils, for example synthetic oils have been produced allowing for detergents to be incorporated allowing for carbon deposits from the burnt fuel to be maintained in a non-solidified form. Another advancement in piston ring technology was the torsional twist piston ring. The concept of creating a greater seal due to the material elasticity was first reported in work by A.E.H. Love in 1882 who suggested that the addition of geometrical features, such as a chamfer placed on the top back edge of the ring resulting in further improvement.

As the formulation and development of lubricants has evolved, so have piston ring designs. A common problem with the earlier oil would burn large levels of carbon which attachs to the piston ring causing corking. The piston ring has the highest wear ratio of the entire combustion engine. Early piston rings were run without wear

protection during operation. In the 1940's it was suggested electroplating Cr was employed to coat the piston ring increasing its wear resistance [3].

Even though the design of the piston ring has been investigated extensively for many years; this work focus on one of the most new features added to the piston ring; the contact face chamfer. The piston is a reciprocating mass system. As the piston moves from the bottom dead centre (BDC) to the top dead centre (TDC) the piston ring performs best when the contact is perpendicular, shown in Figure 1 - The top compression ring, oil ring with piston running perpendicular.
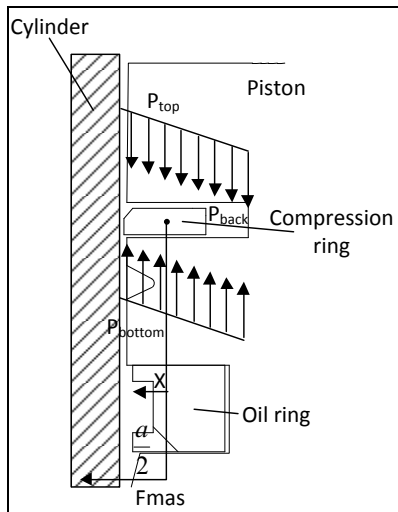

Figure 1 - The top compression ring, oil ring with piston running perpendicular

Due to piston to cylinder clearance and thermodynamic changes occurring in the piston during operation the piston can tilt, therefore at the ring top edge of the contact face is subject to large stresses. Companies are therefore adding a small chamfer on the top of the contact face to reduce stress. The current work will examine the chamfer design and suggest an optimal geometrical size.

In this paper the Finite Element Analysis (FEA) package COMSOL was used to examine the tilted ring. Three sets of worn BS-Grade 400 grey cast iron top compression rings where examined from KTM 525, Ford Cross Flow 1.6 and the Ford Duratec 1.6 engines. The rings were coated with MoS_2, Cr and MoS_2 respectively .

One of the components most prone to wear is the piston ring. The geometry of the ring has remained rectangular shape for almost 200 years. During the last 50 years piston ring have been altered [4,5] to incorporate a new torsional chamfer. By incorporating this design feature behavioural torsional twists could be incorporated. The new twisting motion offers simple piston ring twisting features during the combustion process resulting in superior performance as the piston ring can twist to reduce the possibilities of fatigue failure. As the engine load is increased to a point in which the engine is at bottom dead centre moving towards top dead centre the thrust coming from the system encourages the piston tilt which introduces an angular offset from the contact face leading to the geometric incorporation of a contact chamfer on the piston ring face.

In this paper the examination of incorporating the work that was done by [6] on chamfers is studied. The examination of orientation of the chamfer is discussed. As noted by [6] incorporation of this chamfer relative to this chamfer orientation.

2. System

In combustion engine, the inertia forces act on the piston during the combustion process. Each inertial force acting upon the system is represented as $m\omega^2 R\cos\theta$ and $m\omega^2 R\frac{1}{n}\cos\theta$ which are the primary inertial force and the secondary inertial force respectively shown in figure 1.

During operation the calculation assumes that the axis from the centre of the crank has zero offset but the standard four stroke internal combustion engine piston has an analytical profile from the oil ring to the bottom of the skirt. The assumption is made up almost perfect cylindrical body is assumed after the thermodynamic changes. In this paper we assume full axial motion with the ring having torque acting upon it at all times. Assuming that the piston has thermodynamically changed to cylindrical geometry. The Pythagoras's theorem in Fig 2 (b) to calculate the angle of trajectory for the piston ring during contact is done where P_L is the piston length, C_L is the piston to cylinder clearance and B_L is the cylinder area covered while in operation.
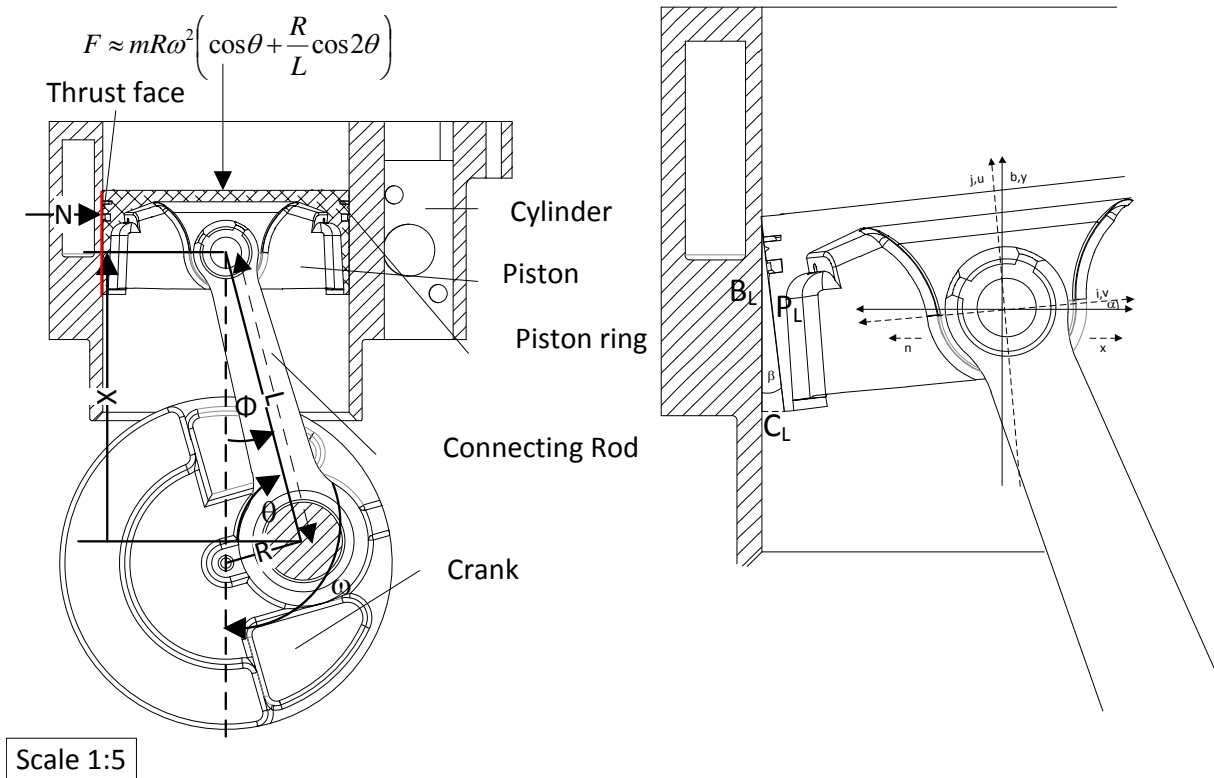
$$F \approx mR\omega^2\left(\cos\theta + \frac{R}{L}\cos 2\theta\right)$$

Figure 2. **(a) Cylinder, Piston, Piston ring, Connecting rod and Crank configuration, (b) showing the piston tilt**

The distance of B_L can be represented as

$$B_L{}^2 = \sqrt{P_L{}^2 - C_L{}^2}$$

By using this equation the angle of the ring is represented as:

$$\emptyset = \sin\left(\frac{C_L}{P_L}\right)$$

Using Love principles [6], the torsional stress found in the piston ring can be calculated. The principles of rotation noted by [4] denotes coordinates of (i, j) in the Fig 2 (b) show the principles of the axis inertia elastic energy in the twist of the piston ring. The coordinates (u, v) are rotated the axis angle of α shown in the diagram. Hence the equations for this response are as follows:

$$\Delta k_i = -\beta k_j - \frac{d^2 v}{ds^2}; \quad \Delta k_j = +\beta k_i + \frac{d^2 v}{ds^2}; \quad \tau = \frac{d(-\beta + k_j u + k_i v)}{ds}; \quad ds = r_s d\varphi$$

In the equations above β, Δκ, τ and ds are the twist angle, change of the curvatures, torsion in the ring and the arc length respectively. The last boundary condition to be considered when simulating this type of system is the piston ring pressures. These are calculated by using the following equations. The first principle to be calculated in the pressures is the negative direction pressure which is as follows.

$$p(x) = \frac{1}{2}(P_{bottom} + P_{back})\frac{x}{a} + (P_{bottom} - P_{back})$$

The resulting gas pressure acting on the ring contact face gap is calculated using

$$p(x) = -\frac{1}{2}\left(P_{top} - P_{bottom}\right)\frac{x}{a} - \left(P_{top} - P_{bottom}\right)$$

The gas force in the system can be calculated from the pressure from the top and bottom, based on the circumference of the ring thickness a.

$$Gas_F = -\frac{1}{2}a\left(P_{top} - P_{bottom}\right)$$

In this paper the lubrication is assumed to be fully flooded. Also following methods given in reference [7] which studied the effects of the geometry of the piston in relation to piston ring blow by. The lubrication distribution of the system is calculated using the method shown in reference [8] method for the trigometric oil film thickness distribution. This method is represented by

$$h_m(\emptyset) = A + B|\sin\emptyset|$$

When applying the forces of the ring into the simulation the simple equation of force is employed $F = \frac{p}{a}$ where F, p and a are the force, pressure and area respectively. The material deformation the material of the ring is an important factor to take into consideration. This is done by calculating the shear stress acting at that point.

$$\tau = \frac{\sigma_1 - \sigma_2}{2} \qquad \text{where } \sigma_1 \text{ and } \sigma_2 \text{ are the major and minor principle stresses respectively.}$$

To construct an accurate simulation the of the ring the dimensions of the ring the British standards have been studied to meet ISO standards ISO 6622-1 Internal combustion engines - Piston rings - Part 1: Rectangular rings made of cast iron.
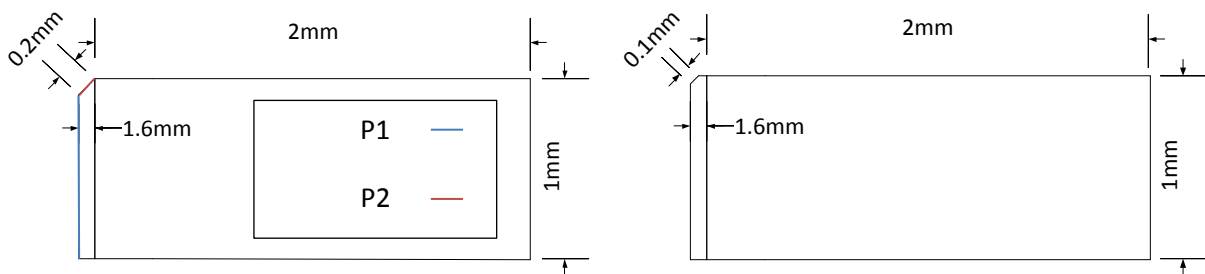

Figure 3 (a) ISO 6622-1 Internal combustion engines cast iron standard ring dimensions based on 90mm diameter. (b) new Dickinson design

Figure 3 (a) ISO 6622-1 Internal combustion engines cast iron standard ring dimensions based on 90mm diameter(a) shows the KTM dimensions of the compression ring which abide by the British ISO standards based on a compression ring with a 90mm diameter and Figure 3 (b) shows the new proposed design of the ring coating chamfer. By using methods shown in Figure 2. (a) Cylinder, Piston, Piston ring, Connecting rod and Crank configurationthe inertia forces where calculated. The range of 6000-10000RPM was used from the recommended operation range of the KTM 520 engine. The forces calculated were part of the boundary conditions of the model. These forces allow for the inertia to be considered on the ring along with all the pressures. It should be noted that in this paper the description of radial forces and hydrodynamic lubrication of the ring face are not discussed.

3. Results

Figure 4. The ring stress concentration using Comsol shows the ring stress concentration areas under stress during the rise from BDC to TDC confirming the behaviour previously observed in reference [7]. This problem being investigated refers to the piston ring contact chamfer and the thrust face, (plus also the boundary behaviour between them both) shown in Figure 3 (a) ISO 6622-1 Internal combustion engines cast iron standard ring dimensions based on 90mm diameter. (b) new Dickinson design P1 and P2 respectively. At P2 it is typical to note a greater deformation of the ring that what can been seen on the opposite side.
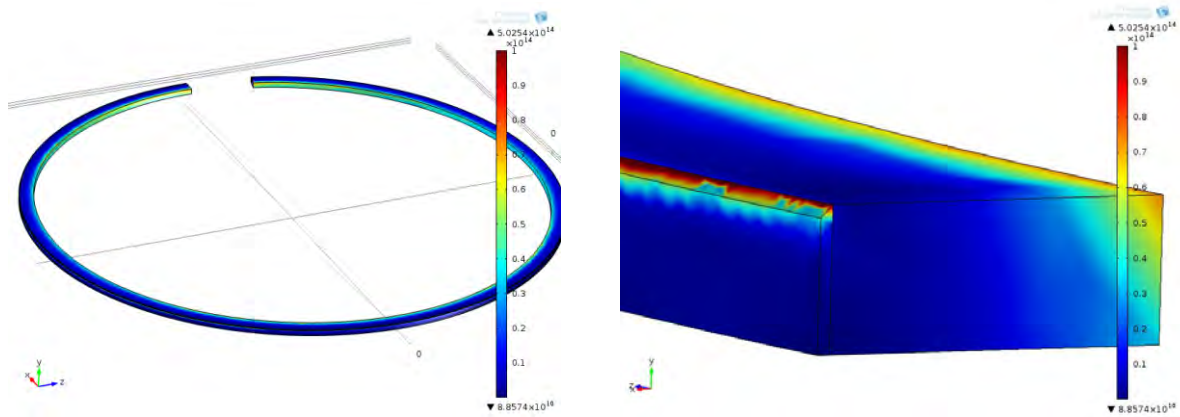

Figure 4. The ring stress concentration using Comsol.

By looking at the standard geometry of the square piston ring the elevated stress concentration of the ring on the Figure 3 (a) ISO 6622-1 Internal combustion engines cast iron standard ring dimensions based on 90mm diameter. (b) new Dickinson design P2 are observed. The reason for the incorporation (or not) of the chamfer on the contact face becomes apparent when the behaviour of the system is simulated under running conditions. This depends on the piston inertia forces acting on the engine, this can be seen in Figure 6. Comparing 200um against 100um chamfers. zone 1, where the material can with stand the stress that is being applied upon it. Studies into the elastic properties of materials [4,6] show that if a material has a chamfer placed on it with forces acting in the directions of the chamfer, the two components should not meet between face to face allow some clearance for stress distribution. The piston speeds reach speed in excess of 9000 rpm and the ring top contact edge can easily break away from the substrate.
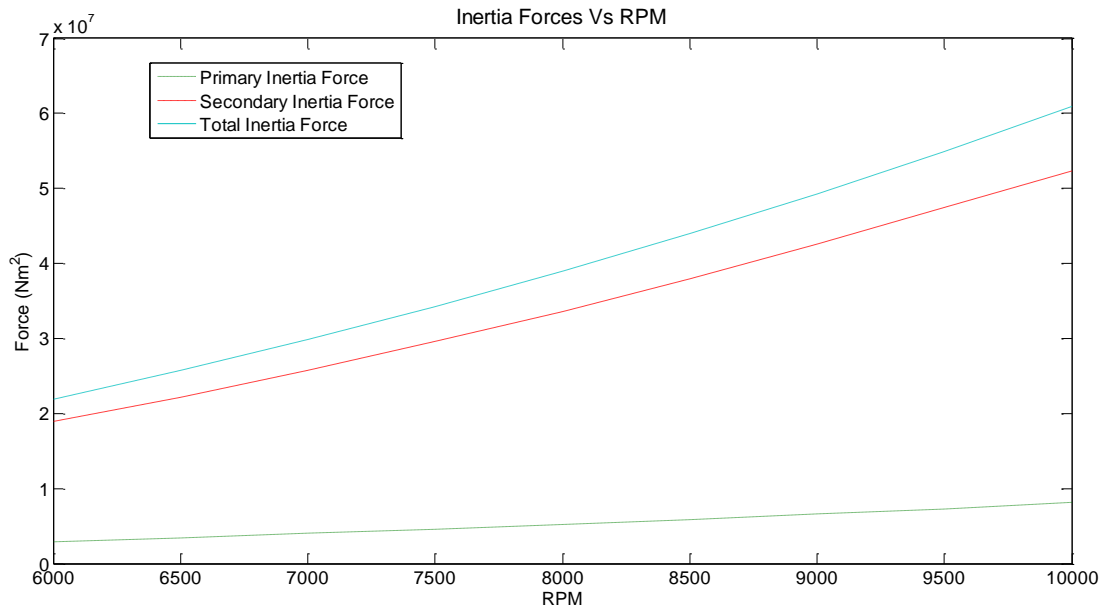
Figure 5. The inertia force and RPM rise.

Figure 6. Comparing 200um against 100um chamfers. shows the response of both ring configurations at 10,000 rpm the last elastic properties of the grey ductile cast iron reaches natural frequency resulting in a large elastic response, the area noted in the graph is zone 2. The piston ring material at these speeds shows a signs of shear stress. On examination of the ring the shearing process takes effect between 10,000 and 11,000 RPM suggesting that the ring should not operate in such conditions. The profile of the ring has a stereotypical rectangular ring with contact chamfer attached. The standard design of the piston ring chamfer can be modified to half of its original size Figure 3 (a) ISO 6622-1 Internal combustion engines cast iron standard ring dimensions based on 90mm diameter. (b) new Dickinson design, this in turn conforming to the work that has been seen in [4,6].

The ring with modified geometry was run again using the same boundary conditions as the original simulation the former geometry with the new geometry.

The chamfer is reduced from 200um to 100um, conforming to Love's law [6]. The shearing process noted in Figure 3 (a) ISO 6622-1 Internal combustion engines cast iron standard ring dimensions based on 90mm diameter. (b) new Dickinson design (b) has been reduced offering a greater material stability.
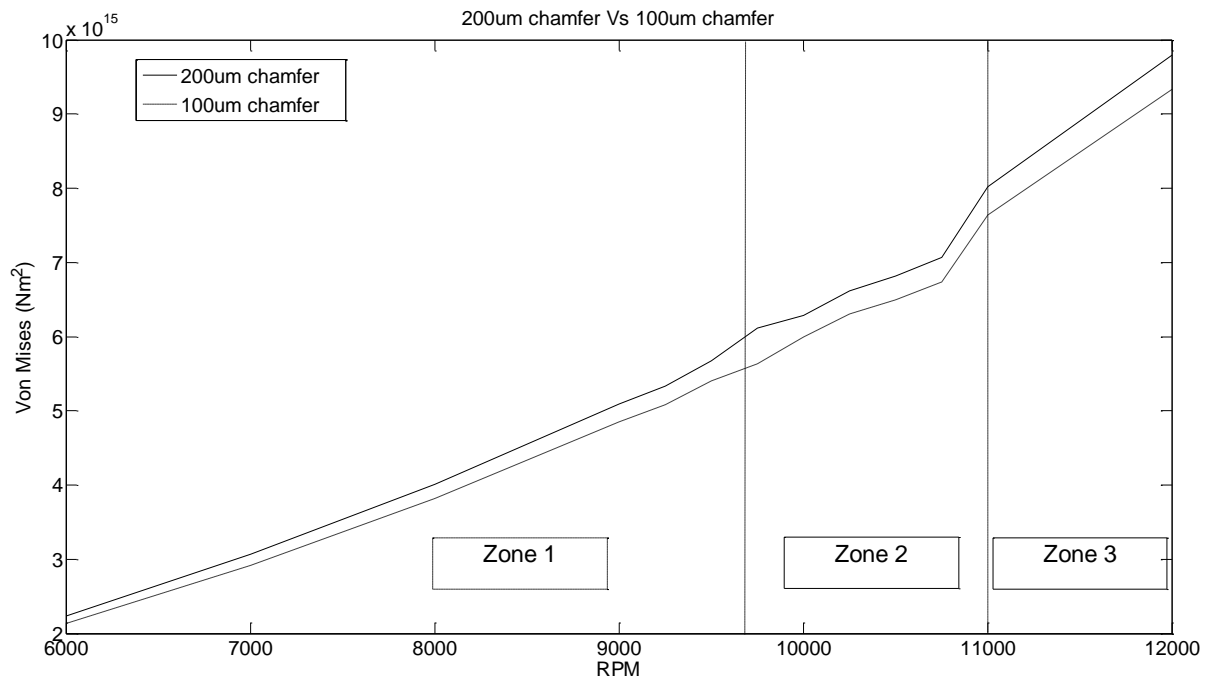
Figure 6. Comparing 200um against 100um chamfers.

The results show that mathematical work completed in reference [6] can affect the performance of the internal combustion engine through geometric refinement of this chamfer. By examining the geometric design of this chamfer on the contact face there are improvements in performance.

The inertia forces confirm the increase in the RPM can lead to massive loading on the piston. By incorporating all the equations mentioned earlier into the simulated model an accurate model of the piston ring loading can be made.

4. Discussion

By examining the initial results in Figure 6. Comparing 200um against 100um chamfers. an intense deformation of shear stress is defined. In zone 2 shearing motion essentially appears around the 10,000 to 11,000 RPM range, initially indicating the wear resistant coating of MoS_2. As the piston leaves from bottom dead centre to top dead centre and initial tilting take effect the contact face of the ring is placed into a point where shearing could may occur if this angle and force continuous. Since a continuous axial motion in the Z plane linear type is assumed cracking on the MoS_2 face may be visible under an examination using the scanning electron microscope.

As this paper assumes race car conditions the combustion pressure is extremely high compared to the normal performance output. From the suggestions made by [5-7], the chamfer placed upon contact face of the piston ring should allow for a material deformation by allowing the material to freely deform under stress during operation. If we simply take the material thickness and divide the material thickness by two stress reduction will become apparent [4,6]. If these geometrical changes are made a reduction in the shear force acting upon the ring at 10,000 to 11,000 RPM which are clearly seen in Figure 6 in Zone 2. However when the operation of the engine moves into Zone 3 the response becomes an almost perfect projection from the performance seen in Zone 1. Hence the need to ensure that during operation the ring can withstand the point in which Zone 2 is crossed.

Further research could consider if the ring is rotating along the Z axis continuously as suggested in [4,9], as these results show that the material loss would be considerably greater as shear stress would have an elevated impact of the piston ring under observation.

5. Conclusion

In this paper we have presented the design modifications to the common ISO 6622 piston ring, which present a possible method to reducing the piston chamfer stress. The engine that all of these results where based upon is the KTM 525, the piston ring is a grey Fe substrate with a M_oS_2 coating which is then machined to have a chamfer on the upper corner of the ring. Thus the modification that is suggested has no major changes to the manufacturing method of the piston ring. By making this change to the ring, one possible effect is that the coating will have a larger life cycle than seen in the past. The results do however suggest that a common attribute of material fatigue should be visible on piston rings after use. Hence further publications could be aimed at the investigation of this structural weakness.

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