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A COMPLETE 3-D DESCRIPTION OF THE ELASTIC BEHAVIOR OF A PISTON RING AND ITS INFLUENCE ON THE TRIBOLOGICAL BEHAVIOR OF THE PISTON RING-CYLINDER LINER INTERFACE

TRACK OR CATEGORY

Engine & Drivetrain V

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INTRODUCTION

Advances in modern engine development are becoming more and more challenging. The intense increase of thermal and mechanical loads as a result of higher power density requires perfecting the function of all engine components especially with regard to emission and friction reduction.

In particular, piston rings pack represent one of the most important cause of mechanical friction loss in internal combustion engine [1], and inadequate ring-liner lubrication leads to high fuel/oil consumption and increased engine emissions with dramatic impact over the entire system efficiency.

A desirable piston ring-pack set has to provide efficient sealing performance with both the cylinder wall in a radial direction and the top or bottom sides of the piston ring housing in an axial direction, leading to minimal gas blow-by, oil consumption and friction loss. Moreover piston ring other requirement are low friction, low wear and good resistance against mechanical/thermal fatigue. This is a challenging task due to the nature of the phenomena and interactions associated with piston rings. For example, increasing the installed ring tension, which is a method to control oil consumption, also tends to increase ring friction. Hence, any attempt to optimize ring-pack performance output parameters requires a good understanding of the dynamics of all the involved components.

Different key elements have to be considered: the ring shape in its free state (namely free shape), the ring crossing-section geometry and the contact surface profile, which play important roles in determining the ring behavior.

In order to achieve sealing, the piston rings are first held against the cylinder liner in its front face by their tension after being installed into the cylinder bore. The contact pressure on the cylinder wall is achieved by the inherent spring force of the ring in conjunction with the gas pressure behind the ring. During different engine operating conditions, the piston rings experience dynamically changing forces and axial as well as radial movements of the rings can occur. The contact on the side of the piston housing is achieved by the axial forces acting on the ring. The axial forces are composed of the gas pressure above and under the ring, the mass forces (inertia), and the friction forces. These forces change their direction during the cycle, and, as a result, the piston ring moves from one side of the groove to the other during the engine cycle. This is known as ring fluttering when the axial movement becomes excessive. This behavior open an additional gas flow path: gas can flow around the inner diameter of the ring which results in very high engine blow-by loss. In addition the ring pack design should also consider other factors, including gas blowback and, as said before, ring pack friction. The blowback is the reverse process of blowby and is highly related to engine emission and oil consumption while ring friction can cause severe ring and cylinder wall wear, which results in the ring losing its sealing capability. These factors are related to the ring circumferential pressure distribution, which is defined by the ring free shape and the ring cross section geometry.

The understanding of the piston ring behavior is an hard challenge for automotive engineer. Firstly, in 1936 Castleman [2] investigated and proposed the concept of hydrodynamic lubrication for the piston ring. Thereafter, more and more research has been done in this field. Dowson et al. [3] predicted the behavior of a piston ring using the EHL theory. Sun [4] conducted his study for ring-bore conformability, in which the ring was modeled as a curved beam under in-plane loads. Liu and Tian [5, 6] developed an FEA tool for piston ring design. Ejakov et al. [7] modeled ring twist behavior predicting ring axial, radial displacements, bending and twisting angles along the ring periphery over an engine cycle.
In this contributions a complete 3D model of the piston ring is proposed, and the influence of an important parameter like the twist angle of the piston ring section on both ring-liner and ring-piston groove interaction is investigated.

INFLUENCE OF GEOMETRIC PARAMETERS

In the following the influence of the twist behavior of a compression ring will be examined on the ring performance. Two different approaches have been employed for the ring analysis: non-linear Finite Element analysis and Elasto-Hydrodynamic Lubrication analysis.

Non-linear Finite Element Analysis

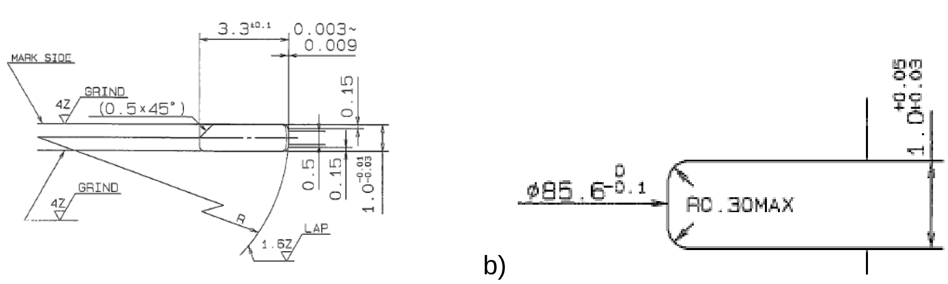
Both static and dynamic analysis have been performed. In particular static analysis have been used in order to investigate the influence of the twist behavior of the ring on the free shape of the ring and on the ring-liner interaction after ring mounting. Dynamic analysis have been employed in order to investigate the ring-housing interaction as a function of the ring twist and piston housing geometry.

Finite Element static analysis

The radial pressure distribution between the ring and the cylinder liner is a crucial aspect that determines the sealing performance of the contact between the ring periphery and liner inner wall. Different approaches exist for defining the free shape of the ring in order to obtain a particular ring-cylinder interaction, see [8]. In this contribution the following equation for the free ring free profile have been investigated:

$$R = R_b + \frac{pR_b^{\,4}}{EJ}\left(1 - cos(\alpha) + \frac{1}{2}\alpha sin(\alpha)\right) + \frac{R_b}{2}\left(\frac{pR_b^{\,3}}{EJ}\right)^2\left(\alpha - \frac{1}{2}\alpha cos(\alpha) - \frac{1}{2}sin(\alpha)\right)\left(3sen(\alpha) - \alpha cos(\alpha)\right) \qquad (1)$$

where R is the radius for the center of gravity of the ring section, R_b is the nominal radius for the center of gravity of the ring section, J is the moment of inertia of the ring section referred to the axis for the center of gravity of the ring section parallel to the ring axis, E is the young modulus of the material and p is the constant pressure between the ring and the liner. Equation (1), that returns good results for non-twist rings [4] has been here employed to define the external profile of the twist ring of Figure 1. Then a Finite Element analysis has been performed to simulate the ring mounting inside the liner. As a consequence of the twist behavior of the ring, a non-uniform contact has been observed at the interface between the ring and the liner. Figure 2a depicts the gap between the liner and the ring after ring mounting. A maximum gap of about half a micron has been registered. Figure 2b shows the axial displacement of the upper surface of ring after ring mounting. The area exhibiting the maximum axial displacement perfectly matches with the area where the maximum gap has been detected, thus confirming that the twist behavior of the ring is the responsible of the non-uniform contact.



a) b)

Figure 1: a) piston ring section dimensions; b) piston groove dimensions.

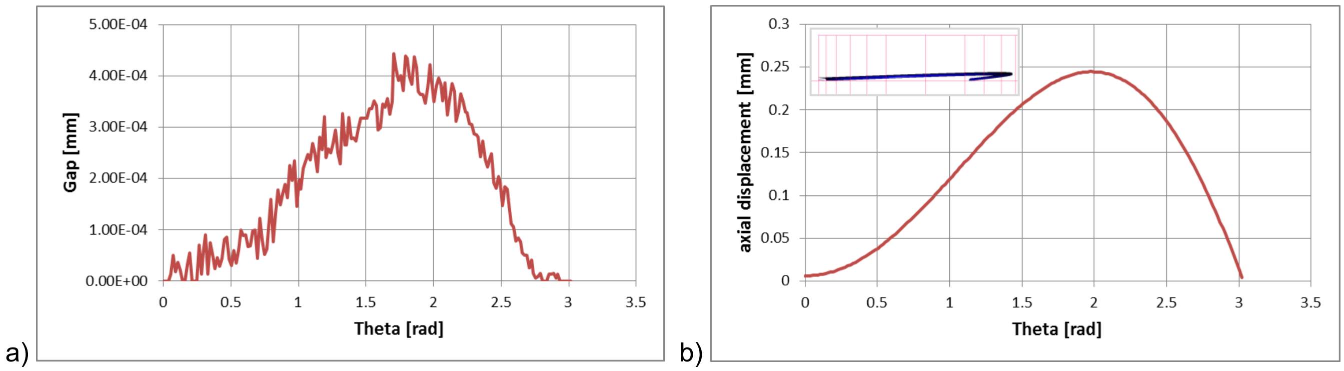


Figure 2: a) gap between the ring and the liner after mounting; b) axial displacement of the ring upper surface.

Finite Element dynamic analysis

A dynamic model has been prepared in order to analyze the dynamic behavior of the ring when inserted into the piston groove. The influence of twist and grove dimensions have been investigated.

Two different couplings have been analyzed exhibiting minimum and maximum axial gap between the ring and the groove, namely 0.04mm and 0.08mm, see Figure 1. Moreover, a no-twist ring has been analyzed in order to highlight the twist influence. Figure 3a shows the inertial force acting on the ring as a function of the time. Oscillations of the force have been registered at the instants when the inertial forces change in sign. Figure 3b highlights the ring fluttering. It can be noticed that the fluttering is dramatically reduced when the twist ring is considered. Moreover, for the twist ring, the amplitude of fluttering is higher when the axial gap is maximum.

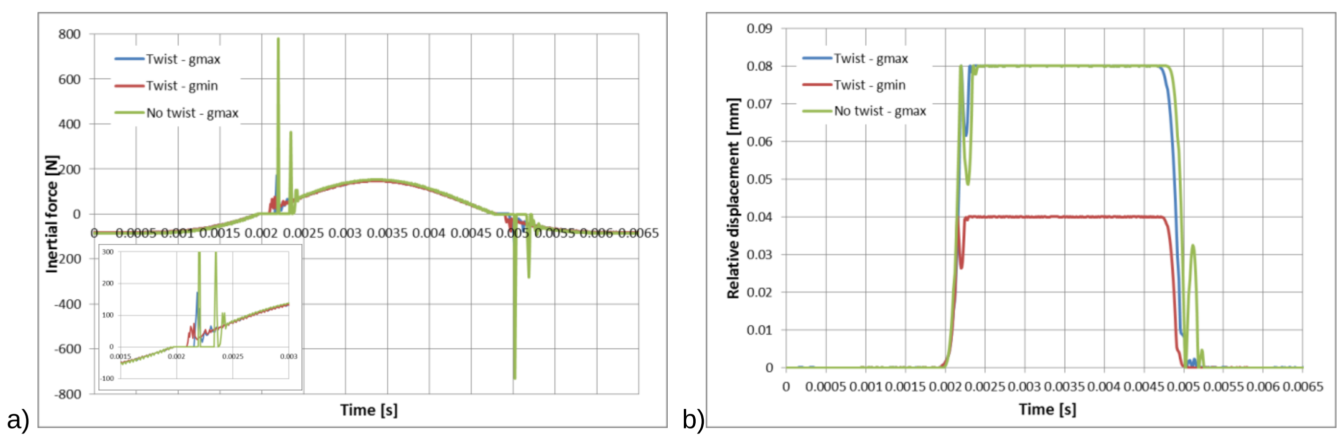


Figure 3: a) inertial force acting on the ring; b) relative displacement between the ring and the groove.

Elasto-Hydrodinamic Lubrication Analysis

The mass conserving lubrication algorithm developed in [9-11] has been employed to perform preliminary 3D non-axisymmetric elasto-hydrodynamic lubrication analysis of the twist ring. In particular, stationary analysis have been done at the instants when the relative velocity between the ring and the liner is maximum. Results in terms of minimum film thickness, hydrodynamic pressure distribution and direct contact pressure distribution have been analyzed. A different behavior of the ring has been detected when results of the downward stroke are compared to those of the upward stroke. In particular, Figure 4 depicts the hydrodynamic pressure. As a consequence of the twist of the ring, the external profile of the ring is non-symmetrical with respect to the axial direction thus generating different pressure profiles when the relative velocity between the ring and the liner changes in sign.

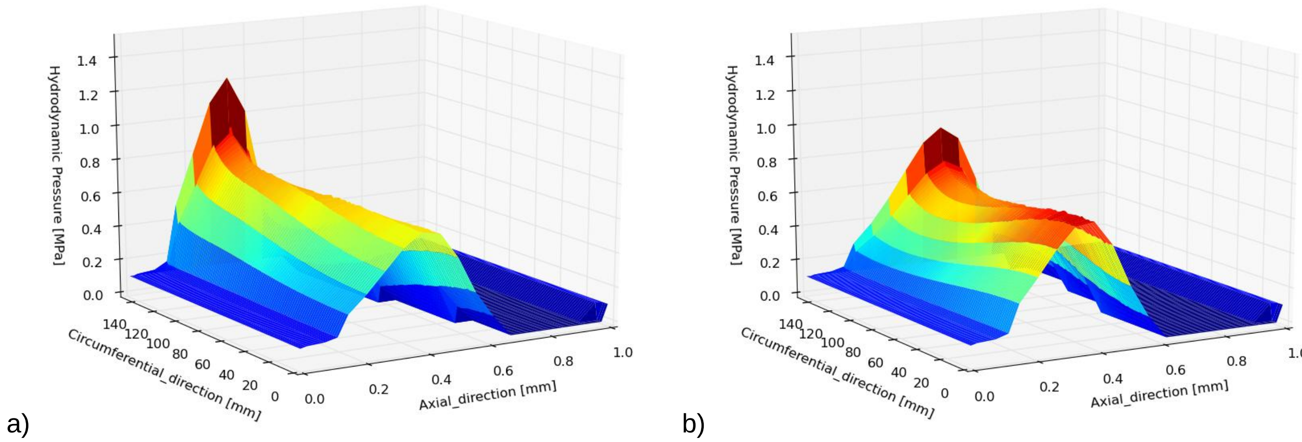


Figure 4: hydrodynamic pressure a) downward stroke; b) upward stroke.

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KEYWORDS

Piston ring profile, twist, lubrication, cavitation, complementarity.