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Performance evolution of a worn piston ring

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ABSTRACT

Reducing the fuel consumption of a combustion engine has been an important design issue. Engine friction has to be reduced and the piston ring - cylinder liner contact is a major source of friction. However, the piston ring geometry can evolve due to wear, and the ring can acquire a flat part. The current work analytically studies the hydrodynamic friction and load carrying capacity of a twisted parabolic - flat (worn) piston ring. The outcome is presented as a function of twist angle, total width and flat width.

With the chosen functional parameters of the piston ring, the evolution of stroke-averaged friction and film thickness with wear (increasing flat width) is studied. One finds that the hydrodynamic friction increases with initial piston ring wear, but the minimum thickness of the generated film increases as well. This ensures the ring can continue to operate safely and last. Imposing a constant film, the hydrodynamic friction can be reduced a little for wide rings by using a narrower parabolic-flat ring. No such reduction is possible for narrow rings.

1. Introduction

The piston ring cylinder liner (PRCL) contact in an internal combustion engine largely determines the overall engine performance and polluting emissions. It performs a mobile sealing action between the combustion chamber and the lower engine. However, this perfect sealing function has to be achieved with minimum frictional losses and negligible wear.

The PRCL contact has been modeled for over four decades. The main objective of these studies is the prediction of friction and oil consumption and its reduction. Three classical papers on this subject were written by Dowson et al. [1] and Jeng [2,3]. A more general study of the piston ring contact can be found in Tian et al. [4], Richardson [5], Gamble et al. [6] and Tomanik [7].

Recent work by the authors has focused specifically on friction generation and load carrying capacity (LCC) of a parabolic piston ring using analytical integration of the Reynolds equation [8]. For a parabolic ring, wear can cause a "flat" part. Substantial wear can occur during running in Refs. [9,10] or around top dead center (TDC) due to high temperature and inadequate oil supply [11,12].

Piston ring wear is relatively complicated to predict, as wear is the least understood of the three main processes in tribology: friction, lubrication and wear [13]. Priest et al. [10] developed a numerical model, including an assumed ring twist, that predicts the dynamics, lubrication and wear of piston rings. Tian [14] illustrated the dynamic effects of a piston and two rings on friction and wear for a heavy-duty diesel engine, using existing mixed lubrication models. More recent work by Zabala et al. [15] proposed an AVL Excite software-based wear simulation. The difference between the simulated and experimental results was less than 5%.

The wear of piston rings and cylinder walls has a significant effect on the performance of the piston assembly [13]. The Ph.D. thesis of M. Priest [16] attributes an observed reduction in fuel consumption of the engine in the early stages of running to the wear of the second ring. Francisco et al. [17] applied a full-scale mixed lubrication regime model to a Twin Land Oil Control Ring-honed cylinder liner contact. An increase of the hydrodynamic LCC was observed for the worn surfaces.

In order to understand how the hydrodynamic friction and LCC of a worn piston ring will evolve, the authors studied a parabolic-flat ring geometry in Ref. [18] and an approximate geometry including ring twist in Ref. [19]. This twist angle can either be generated during manufacturing or can occur during operation, due to piston secondary motion and ring dynamics [20–22].

Theoretical work by Ruddy et al. [20] has shown that gas pressure differences across a ring can cause a small twist. This twist has a significant impact on the ring LCC.

This work extends the work by Pettavino et al. [18] and by Fang et al. [19], using a more precise twisted parabolic-flat geometry. Whatever the origin of the angle, the proposed model can account for an arbitrary angle variation.

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Notation

h	film thickness
H	dimensionless film thickness, $H = h/h_0$
h_0	minimum film thickness
$\mathscr{J}$	
l	length of the flat part of a ring
L	dimensionless length of the flat part of a ring $L = l/\sqrt{h_0\,R_x}$
l_{tot}	total ring width
L_{tot}	dimensionless total ring width $L_{tot} = l_{tot}/\sqrt{h_0\,R_x}$
$\mathcal{M}$	$\mathcal{M} = (\mathscr{Q}^2 - \sqrt{1-2\alpha\mathscr{Q}N})/\mathscr{J}$
N	dimensionless start of pressurized domain $N < 0$
P	pressure
P	dimensionless pressure $P = ph_0\sqrt{h_0}/(12\eta u_m\sqrt{R_x})$
$\mathscr{Q}$	$\mathscr{Q} = \sqrt{\alpha^2+1}$
$\mathcal{R}$	$\mathscr{R} = \alpha L + 1$
R_x	reduced radius of curvature
SRR	slide-to-roll ratio $(u_2 - u_1)/u_m$
$\mathscr{S}$	$\mathscr{S} = 1/(\mathcal{M}^2 + 1)$
$\mathscr{T}$	$\mathscr{T} = \arctan(\alpha^2/\mathscr{J}) - \arctan(\mathcal{M})$
u_1	piston ring velocity
u_2	cylinder liner velocity
u_m	mean surface velocity $u_m = (u_1 + u_2)/2$
w	load per unit length
W	dimensionless load per unit length $W = wh_0/(12\eta u_m R_x)$
w_c	friction force due to Couette flow per unit length
W_c	dimensionless friction force due to Couette flow $W_c = w_c/(12\eta u_m\sqrt{R_x/h_0})$
w_p	friction force due to Poiseuille flow per unit length
W_p	dimensionless friction force due to Poiseuille flow $W_p = w_p/(12\eta u_m\sqrt{R_x/h_0})$
x	coordinate in the direction of sliding
X	dimensionless coordinate $X = x/\sqrt{R_x h_0}$
Z	dimensionless coordinate $Z = (\mathscr{Q}^2 - \sqrt{1-2\alpha\mathscr{Q}X})/\mathscr{J}$
α	dimensionless piston ring twist angle
η	lubricant viscosity
LCC	abbreviation for load carrying capacity

2. Theory

The flow in the circumferential direction is neglected. As such the lubrication problem becomes one dimensional (1D). The finite ring width generates geometric starvation. The surfaces and counter surfaces are assumed perfectly smooth (no asperity contact) and steady state conditions are applied. The lubrication regime is assumed to be isoviscous rigid, as the pressure is assumed too low to generate piezo-viscous effects or significant elastic deformation.

The 1D dimensionless ring geometry is described by:

$$H(X) = \begin{cases} 1 - \dfrac{\alpha X + \sqrt{\alpha^2+1}\left(\sqrt{1-2\alpha X\sqrt{\alpha^2+1}}-1\right)}{\alpha^2}, & \text{if} \quad X \le 0; \\ 1 + \alpha X, & \text{if} \quad 0 < X \le L. \end{cases} \tag{1}$$

The term "width" refers to the 3D ring width in the sliding direction X (cf. Fig. 1). Classical lubrication assumes pressure build-up to start at $X = -\infty$. This paper uses $X = N$ as the start of the pressure domain (starved section). This causes a reduced LCC and a modified friction, termed geometrical starvation.

In Fig. 1, the black curve represents the precise geometry used in this work, with a dimensionless twist angle of -0.1. The red curve is an approximation, from the previous work [19]. The green curve indicates a parabolic-flat piston ring geometry with zero twist angle.

Knowing the geometry, the pressure can be obtained by integration of the dimensionless, incompressible, 1D Reynolds equation:

$$\frac{\partial}{\partial X}\left(H^3\frac{\partial P}{\partial X}\right) = \frac{\partial H}{\partial X} \tag{2}$$

Integration gives:

$$\frac{\partial P}{\partial X} = \frac{H - H^*}{H^3} \tag{3}$$

where H^* is the value of H such that:

$$H^* = H|_{\partial P/\partial X=0} \tag{4}$$

Imposing pressure and flow continuity between the two solutions at $X = 0$ and 0 pressure at $X = L$ and $X = N$, yields the pressure distribution. The H^* value is an indicator of the starvation level, but now it depends on the values of α, L and N.

3. Pressure distribution

For the starved case, the pressure build-up starts at $X = N$. For $N = -\infty$, the contact is fully flooded. The following definitions are used to simplify and shorten the expressions:

$$\mathcal{R} = \alpha L + 1,$$

$$\mathscr{Q} = \sqrt{\alpha^2 + 1},$$

$$\mathscr{J} = \sqrt{\alpha^2(2\sqrt{\alpha^2+1} - \alpha^2)},$$

$$\mathcal{M} = (\mathscr{Q}^2 - \sqrt{1 - 2\alpha\mathscr{Q}N})/\mathscr{J},$$

$$\mathscr{S} = 1/(\mathcal{M}^2 + 1),$$

$$\mathscr{T} = \arctan(\alpha^2/\mathscr{J}) - \arctan(\mathcal{M}),$$

$$Z = (\mathscr{Q}^2 - \sqrt{1 - 2\alpha\mathscr{Q}X})/\mathscr{J}.$$

The pressure distribution $P_1(Z)$ for $X \le 0$ is given by:

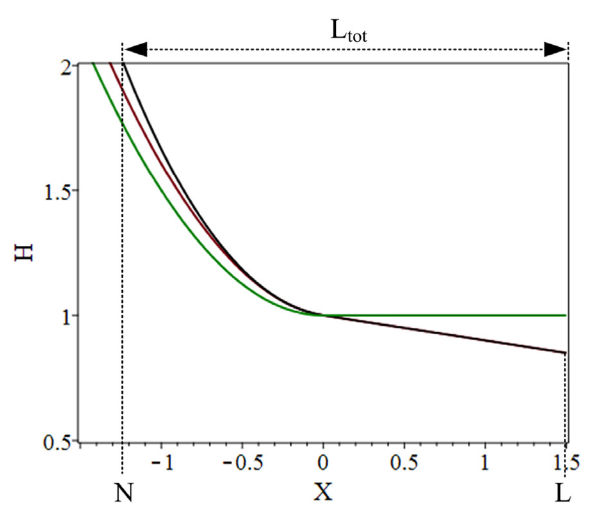

Fig. 1. Dimensionless 1D geometry for a parabolic-flat piston ring, Black: the precise twisted shape at $\alpha = -0.1$, Red: approximate shape, Green: no twist. (For interpretation of the references to colour in this figure legend, the reader is referred to the Web version of this article.)