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Cross hatched texture influence on the load carrying capacity of oil control rings

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ABSTRACT

The optimization of the cylinder-liner/piston-ring contact performance (oil consumption, friction and wear) is the final objective of this work. The micro-geometry impacts directly the contact behavior. Cross-hatched textures are usually manufactured on cylinder liner surfaces. Grooves are often described as lubricant reservoirs and pipes redistributing oil. Grooves also change the pressure distribution and load carrying capacity.

The top ring generally has a parabolic shape. Oil control rings on the contrary are usually flat. Thus, the contact between an oil control ring and a smooth cylinder liner has no load carrying capacity. Cross hatched textures enable a certain load carrying capacity which depends on the operating conditions and the texture micro-geometry. This paper presents a numerical and an analytical analysis of this phenomenon. A set of functional parameters is proposed to model this contact and predict its load carrying capacity.

1. Introduction

1.1. Context

The automotive industry is focused on reducing power losses and pollution. The cylinder-liner/piston-ring contact is usually considered as the main contribution to engine friction. Moreover, excessive oil consumption is associated with polluting emissions and early deterioration of particle filters. Optimizing the cylinder-liner/piston-ring contact consists in finding a compromise between fuel consumption, lubricant consumption and cylinder liner life (wear or scuffing, see Willn [1]) as a function of the operating conditions (load, speed, viscosity…). Dynamic effects are also studied, see Tian [2].

Along with macroscopic design parameters, the micro-geometry is a possible optimization path. Most of cylinder-liner surface finishes are cross-hatched geometries which are obtained by plateau-honing. In the current paper, the cross-hatched texture influence on the cylinder-liner/piston-ring contact performance is studied. Fig. 1 shows an example of such a surface finish. The coupled influence of operating conditions and groove geometry on the contact performance is complex and is not accounted for in commercial softwares, see Fox [3] and Carden et al. [4]. Willis [5] reported that very smooth surfaces surprisingly do not perform best. Organisciak et al. [6] studied the aspect of oil redistribution generated by a cross-hatched texture.

Bouassida et al. [7] studied the cross-hatched texture influence on the top ring load carrying capacity. The current paper focuses on the oil control ring(s). The main difference is the ring radius of curvature: typically, the order of magnitude of one centimeter for a top ring and several meters for an oil control ring. Bouassida et al. [7] showed that the top ring load carrying capacity is deteriorated by the cross-hatched texture. This load carrying capacity reduction was quantified as a function of the operating conditions and the texture geometry (groove depth, width, angle and density).

However, a flat ring has no load carrying capacity. The cross-hatched texture provides a certain load carrying capacity and this load carrying capacity will be studied as a function of the groove geometry and the operating conditions. Chen [8] shows that a second source of load carrying capacity stems from the plateau roughness. This point is not considered in the current work. Moes [9] shows that the iso-viscous-rigid (IVR) load carrying capacity is supposed to increase with the radius of curvature. Biboulet et al. [10] and Ankouni et al. [11] demonstrated that the finite width of the ring causes geometric ‘starvation’ and leads to a quick decrease of the load carrying capacity when increasing the radius of curvature. Thus, the current work studies the case of asymptotic large radius of curvature and small ring width.

Wakuri et al. [12] showed the major influence of contact lubricant starvation on the ring/liner contact performance. The oil control ring purpose is to limit the amount of available oil for

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Nomenclature

A	dimensionless groove depth
h	film thickness
H	dimensionless film thickness, $H = h/h_0$
h_0	minimum film thickness
L_x, L_y	domain size
L_X, L_Y	dimensionless domain size
N	dimensionless pattern size (perpendicular to the groove direction)
p	pressure
P	dimensionless pressure $P = ph_0^2/(12\eta_0 u_m \lambda)$
r	cross-hatched texture geometry
$\mathcal{R}$	dimensionless cross-hatched texture geometry
S	dimensionless slope $S = 2A$
u_m	mean surface velocity $u_m = (u_1 + u_2)/2$
x	coordinate along the sliding direction (axial)
X	dimensionless coordinate $X = x/\lambda$
y	coordinate perpendicular to sliding (circumferential)
Y	dimensionless coordinate $Y = y/\lambda$
η	viscosity, assumed constant η_0
γ	curvature exponent
λ	groove width
θ	groove angle
$\varnothing_M$	maximum value
$\varnothing_\infty$	very distant groove asymptote
$\varnothing_{\frac{\pi}{2}}$	extrapolated behavior of a groove perpendicular to the sliding direction
$\overline{\varnothing}$	mean value

the other rings. Thus, its behavior has a direct consequence on the performance of other rings. The final goal is to compare computed friction to floating liner experiments as presented by Wakuri et al. [12], Jocsak et al. [13] or Liao et al. [14].

Various texture patterns are studied in the literature and more specifically dimple textures. Both experimental [15–20], and numerical [21–23] approaches are encountered. However, significant contradictions exist between the findings of these papers. Tala-Ighil et al. [24] and Dobrica et al. [25] model the dimple texture influence.

The current paper has three sections: a few numerical examples to illustrate some remarkable trends of the pressure distribution, then a simplified analytical model to enable a physical understanding and finally an analysis of a numerical parametric study. The main results of the current paper are:

- The cross-hatched texture provides a load carrying capacity to a flat oil control ring.
- This load carrying capacity is a function of the operating conditions and the cross-hatched texture geometry (groove depth, width…).
- A simplified 1D analytical model predicts the existence of an optimum groove geometry (regarding the dimensionless load carrying capacity).
- This analytical model also allows one to predict the influence of the groove density.
- A set of 2D numerical results is presented; the groove angle and depth influence are included in the curve-fitting equations.

2. Contact modeling and groove geometry

The dimensional Reynolds equation, assuming incompressible and isoviscous lubricant reads:

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(h^3 \frac{\partial p}{\partial y}\right) - 12\eta_0 u_m \frac{\partial h}{\partial x} = 0 \tag{1}$$

$$p \geq 0 \tag{2}$$

A steady-state Reynolds equation is used. Considering a finite width ring, the problem is time-dependent in the inlet and outlet region. Here, the ring is assumed to be sufficiently thick compared with the pattern size along the sliding direction. Under these conditions, numerical calculations did not show cavitation inside the calculation domain even with the deepest grooves. Low to moderate pressures are encountered in such contacts which justify these IVR assumptions. The following dimensionless parameters are used:

$$x = \lambda X$$
$$y = \lambda Y$$
$$h = h_0 H$$
$$p = \frac{12\eta_0 u_m \lambda}{h_0^2} P \tag{3}$$

The dimensionless Reynolds equation reads:

$$\frac{\partial}{\partial X}\left(H^3 \frac{\partial P}{\partial X}\right) + \frac{\partial}{\partial Y}\left(H^3 \frac{\partial P}{\partial Y}\right) - \frac{\partial H}{\partial X} = 0 \tag{4}$$

$$P \geq 0 \tag{5}$$

A zero pressure is imposed on both boundaries X_a and X_b whereas the contact is periodic in the Y direction (circumferential). The mean pressure reads:

$$\overline{p} = \frac{1}{L_x L_y} \iint_\omega p(x,y)\, dx\, dy \tag{6}$$

$$\overline{P} = \frac{1}{L_X L_Y} \iint_\Omega P(X,Y)\, dX\, dY \tag{7}$$

The film thickness including the cross-hatched texture geometry is

$$h = h_0 + r \tag{8}$$

$$H = 1 + \mathcal{R} \tag{9}$$

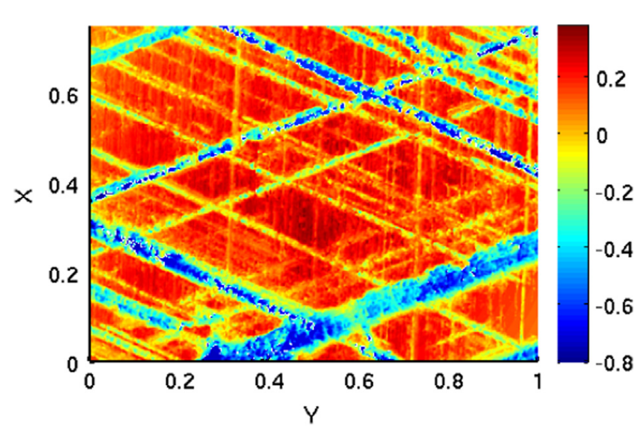

Fig. 1. Typical measured cross-hatched pattern (patch 250 × 190 μm, maximum depth ≈ 3 μm).