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Mutual influence of crosshatch angle and superficial roughness of honed surfaces on friction in ring-pack tribo-system

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ABSTRACT

The cylinder bore surface texture, widely produced by the honing technique, is an essential factor for a good engine performance (friction, oil consumption, running-in, wear etc.). This explains the improvement and development of various new honing techniques. These different honing processes generate surfaces with various texture features characteristics (roughness, valleys depth, crosshatch angle, etc.).

This paper addresses a comparison of ring-pack friction for cylinder surfaces produced by plateau honing and helical slide honing. It takes in consideration the mutual effect of superficial plateau roughness amplitude and honing angle. A numerical model is developed to predict friction within the cylinder ring-pack system in mixed lubrication regime. The results show the effectiveness of helical slide honed surface texture in comparison to plateau honed bore surfaces.

1. Introduction

With a stringent legislation for the limitation of harmful polluting emissions, the improvement of environment efficiency of vehicle engine becomes a fundamental objective [1]. It is evident actually, based on several experimental and numerical studies, that cylinder engine surface texture considerably influences functional performances (running-in, friction, oil consumption, wear, etc.) of the ring-pack tribo-system [2–4].

Honing is a widely used finishing process for cylinder surfaces after drilling or turning operation to guarantee reproducibility with efficient productivity in mass production of cylinder liners [5]. In honing, abrasive stones are loaded against the bore and simultaneously rotated and oscillated. The industrial honing involves graduated stages of abrasive finishing processes, using at first coarse abrasive stones, and then progressively finer grades, so that a very structured surface of liner is produced. Characteristically, the resulting structured surface consists of smooth plateau textures separated by two or more bands of parallel deep valleys with stochastic angular position [6].

Hence, effort has been made in last decade for the improvement of honing process for cylinder liners finishing. Moreover, various new surface textures were proposed by the development of innovative honing techniques (slide honing, glide honing, brush honing, etc.) [7–11] and the surface texturing methods to meet both actual and future functional requirements [12,13].

Unfortunately, the whole effect of different cylinder liner finishes on ring-pack performance is not well understood [11]. Furthermore, the engineered optimized functional surface is still not well defined. Then, it is important to define the optimum texture characteristics to obtain the best functional properties.

Some topographical and lateral geometric properties of the cylinder bores affect considerably the functional performances of the ring-pack contact, particularly, the crosshatch angle [14–18], superficial plateau roughness [19,20] and the valley and surface patterns depth [14] and density [14]. It is believed that liner surfaces that have smaller valleys and smoother plateaus would reduce friction and oil consumption [9]. Moreover, the orientation of micro-grooves has a strong effect on the friction performance of sliding surfaces and optimal orientation is closely dependent on the contact conditions [17]. In hydrodynamic lubrication condition, two optimal ranges of crosshatch angle were determined ([40–60°] and [120–140°]) [14]. However, [7] shows that the steep honing angle did not present any improvement in ring-pack friction and wear at the top dead center of the cylinder but it yields to less oil consumption over the entire engine course in comparison to 40° honing angle [8,10].

In this paper, the mutual influence of honing angle and superficial plateau roughness was investigated based on comparison of surface textures generated by two kind of honing processes: the widely used Plateau honing with honing angle in the range of 40–60° and helical slide honing with grooves arranged in the direction of the piston in an elongated spiral (120–140° honing

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Nomenclature

a_h	Hertzian contact radius, m
b	translation coefficient, m
F_N	external applied load, N
H	dimensionless film thickness
H_0	dimensionless rigid body displacement
M, L	Moes dimensionless parameters
p_h	Hertzian pressure, Pa
p_r	constant, $p_r = 1.96.10^8$
P	dimensionless pressure
R_x, R_y	radii of curvature in x and y direction respectively, m
R	equivalent radius of curvature, m
T	time, s
T	dimensionless time: $T = tu_m/a_h$
u_i	velocity of surface i
u_m	mean velocity, $(u_1+u_2)/2$
x,y,z	space coordinates
X,Y,Z	dimensionless space coordinates
z_r	pressure viscosity index (Roelands), $z_r = p_r\alpha/(\ln(\eta_0)+9.67)$
α	pressure–viscosity coefficient Pa^{-1}
$\bar{\delta}$	dimensionless elastic deflection of the contacting bodies
$\bar{\eta}$	effective viscosity, dimensionless
θ	angle, rad
μ	viscosity, Pa s
$\bar{\mu}$	dimensionless viscosity
μ_0	ambient temperature zero-pressure viscosity, Pa s
$\bar{\rho}$	dimensionless lubricant density
ρ_0	ambient temperature and pressure density, kg m^{-3}
τ_e	equivalent shear stress, Pa
τ_0	characteristic shear stress of Eyring fluid, Pa
$\overline{\tau_m}$	dimensionless mean shear stress
Sa	arithmetical mean height of the surface, μm
Sz	maximum height of the surface, μm
Sk	core reduced height of the surface, μm

angle). Thus, simulation model in mixed lubrication regime was developed to predict friction between the ring pack and the cylinder liner. Then, based on real 3D surface measurements, an algorithm involving the generation of 3D surface topographies with controlled superficial plateau roughness and total surface height is established. As a consequence, the different effects (plateauing, smoothing and lateral surface texture) on friction of surfaces could be isolated. Finally, this method allows a comparison between the two honing processes and possible friction reduction strategy in ring-pack system is analyzed and discussed.

2. Numerical model for hydrodynamic friction simulation in piston ring-pack system

A numerical model was developed to estimate the friction generated between rough surfaces. The model considered the real topography of the cylinder bore surfaces. The objective of this model is to compare the predicted friction coefficients between two kinds of cylinder bore textures in accordance with their superficial plateau roughness amplitude.

2.1. Elastohydrodynamic equations

The generalized Reynolds' equation introduced by Najji [21] has been used to estimate the pressure distribution, film thickness, and the friction coefficient. This equation has the advantage of not being restricted to a particular non-Newtonian law. The steady state equation in the dimensionless form is given by

$$\frac{\partial}{\partial X}\left\{\frac{\bar{\rho}H^3}{\lambda'}\left(\frac{1}{\overline{\eta''_e}}-\frac{\overline{\eta_e}}{\overline{\eta'_e}^2}\right)\frac{\partial P}{\partial X}\right\}+\frac{\partial}{\partial Y}\left\{\frac{\bar{\rho}H^3}{\lambda'}\left(\frac{1}{\overline{\eta''_e}}-\frac{\overline{\eta_e}}{\overline{\eta'_e}^2}\right)\frac{\partial P}{\partial Y}\right\}$$
$$=\frac{\partial\bar{\rho}H(u_2-(\overline{\eta_e}/\overline{\eta'_e}^2)(u_2-u_1))}{\partial X}+\frac{\partial\bar{\rho}H}{\partial T} \quad (1)$$

where $\lambda' = \frac{R^2\mu_0}{p_h a_h^3}$, $\frac{1}{\overline{\eta_e}} = \int_0^1 \frac{dZ}{\bar{\eta}}$, $\frac{1}{\overline{\eta'_e}} = \int_0^1 \frac{ZdZ}{\bar{\eta}}$ and $\frac{1}{\overline{\eta''_e}} = \int_0^1 \frac{Z^2dZ}{\bar{\eta}}$

The effective viscosities could be calculated considering the shear-thinning effect as (Eyring fluid)

$$\frac{1}{\bar{\eta}} = \frac{1}{\bar{\mu}}\frac{1}{\overline{\tau_m}}\sinh(\overline{\tau_m}) \quad (2)$$

with $\overline{\tau_m} = \tau_e/\tau_0$, where, τ_0 is a reference shear stress and $\tau_e = \sqrt{\tau_x^2+\tau_y^2}$ is the equivalent shear stress inside the lubricant film.

The lubricant's viscosity and density are assumed to depend on pressure. The Dowson and Higginson formula [22] (Eq. (3)) and Roelands law [23] (Eq. (4)) were used

$$\bar{\rho}(P) = \left[1+\frac{0.6\times10^{-9}P.p_h}{1+1.7\times10^{-9}P.p_h}\right] \quad (3)$$

where, ρ_0 is the density at ambient pressure.

$$\bar{\mu}(P) = \exp\left((\ln(\mu_0)+9.67)\left(-1+\frac{Pp_h}{p_r}\right)^{z_r}\right) \quad (4)$$

where, μ_0 is the viscosity at ambient pressure, p_r is a constant equal to 1.96×10^8, and z_r is the pressure viscosity index expressed by $z_r = \alpha p_r/(\ln(\mu_0+9.67))$ where α is a pressure–viscosity coefficient (varying between 1.10^{-8} and 2.210^{-8}).

Finally, the boundary condition $P=0$ and the cavitation $P(X,Y)\geq 0$ $\forall X,Y$ must be satisfied during the simulation.

The film thickness equation is given in dimensionless form by the following equation:

$$H(X,Y,T) = H_0(T)+\frac{X^2}{2}+\frac{Y^2}{2}+\bar{\delta}(X,Y,T)-\overline{Z_h}(X,Y,T) \quad (5)$$

where, Z_h is the height surface topography at each position (X,Y) and $\bar{\delta}(X,Y,T)$ is the dimensionless surface deformation calculated by

$$\bar{\delta}(X,Y,T) = \frac{2}{\pi^2}\iint_\Omega \frac{P(X',Y',T)dX'dY'}{\sqrt{(X-X')^2+(Y-Y')^2}} \quad (6)$$

The global force balance condition is given by

$$\frac{2\pi}{3} = \iint_{\Omega_c} P(X,Y,T)dXdY \quad \forall T \quad (7)$$

2.2. Numerical procedure

The Reynolds equation was solved by the finite difference method in order to obtain the film pressure distribution. A second order accuracy with respect to both space and time [24], was used. The discredited equation was solved by the Jacobi line relaxation [25]. The full-scale mixed EHL approach developed by Hu and Zhu