

Transient experimental and modelling studies of laser-textured micro-grooved surfaces with a focus on piston-ring cylinder liner contacts

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ABSTRACT

This paper presents a comparison between the results from numerical modelling and experiments to shed light on the mechanisms by which surface texture can reduce friction when applied to an automotive cylinder liner. In this configuration, textured features move relative to the piston-liner conjunction and to account for this our approach is to focus on the transient friction response to individual pockets as they pass through, and then leave, the sliding contact. The numerical approach is based on the averaged Reynolds' equation with the Patir & Cheng's flow factors and the p - θ Elrod-Adams mass-conserving cavitation model. The contact pressures that arise from the asperity interactions are solved simultaneously to the fluid flow solution using the Greenwood and Tripp method. The experimental data is produced using a pin-on-disc set up, in which laser textured pockets have been applied to the disc specimen. Under certain conditions in the mixed and boundary lubrication regimes, both model and experimental results show *i*) an increase in friction as the pocket enters the contact, followed by *ii*) a sharp decrease as the pocket leaves the contact, and then *iii*) a gradual decay back to the pre-entrainment value. From the evidence obtained for the first time from the proposed combined modelling and experimental investigation conducted under carefully controlled conditions, we suggest that these three stages occur due to the following mechanisms: *i*) a reduction in fluid pressure due to the increased inlet gap, *ii*) inlet suction as the cavitating fluid within the pocket draws lubricant into the contact, and *iii*) film thickness decay as oil is squeezed out of the contact. The interplay of these three mechanisms is shown to control the response of micro-textured surfaces under all lubrication regimes.

1. Introduction

Recent experiments [1–3] have shown that automotive piston-liner friction may be reduced by up to 50% if the surface of the liner is laser textured with certain configurations of micro-pockets. It is important to model this behaviour to understand and optimise the friction reduction mechanisms that are occurring. However, until now, very few models that predict the lubrication performance of textured surfaces under mixed and boundary lubrication regimes have been successfully compared to experimental data. As shown in very recent literature reviews [4,5], studies comprising both experimental and theoretical investigations on the effects of laser surface texturing are scarce. The vast majority of these studies focus on hydrodynamic lubrication regime, with just a few of them discussing the impact of textured features under mixed-lubrication regime [6,7]. This is due to the requirement for them to (i) reproduce experimental configurations with a certain degree of fidelity, (ii) conserve mass properly, and (iii) account for transient, boundary lubrication conditions. To address this,

the current paper presents a comparison between the results from a newly developed numerical model, which fulfils these criteria, and an experimental test rig operating under closely matched conditions.

A general finite volume method (FVM) has recently been proposed for solving Reynolds equation with a mass-conserving cavitation model on irregular grids [8]. This has been applied here to solve textured lubrication problems more effectively than alternative formulations recently proposed in the pertinent literature [9–11].

A challenge with modelling surface texture is that numerous experiments have shown texture reduces friction in the mixed and boundary regime [12–16] – but often not in the hydrodynamic regime [17–19]. However, the majority of texture modelling have included only hydrodynamic behaviour and have largely ignored mixed lubrication and asperity contact. In order to address these issues and provide an effective means of predicting the behaviour of surface texture, we put forward a new modelling approach for textured surface lubrication that is based on the solution of the averaged Reynolds equation using a mass-conserving formulation with the inclusion of flow factors that capture flow through

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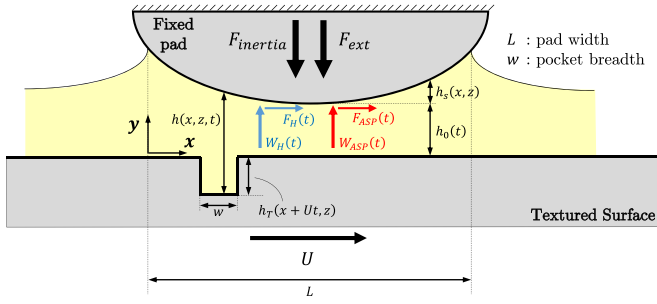


Fig. 1. Schematic illustration of the moving texture contact interface considered in this work. Notice that the fluid flow problem is described with respect to the reference coordinate system attached to the fixed pad (upper surface).

asperities and the contact pressures that arises from the asperity interactions. The focus of this contribution is to specifically look at the behaviour of the fluid and fluid/solid interactions and the local transient effects (cavitation) in the neighbourhood of individual pockets. Both simulation and experimental results obtained from a pin-on-disc test are used to support such investigations.

2. Mathematical modelling

In this section, the mathematical formulations adopted in this contribution for modelling the mixed lubrication regime of piston-ring cylinder liner contacts are briefly presented. The system configuration showing the main geometric and kinematic features used in the mathematical models is illustrated in Fig. 1. The reader is referred to [20] for more details of the modelling and solution framework here adopted.

2.1. Fluid film lubrication

The hydrodynamic pressure built up over the contact interface under mixed lubrication conditions are mathematically modelled using the average Reynolds equation based on the Patir & Cheng's flow factors formulation [21,22]. The fluid film cavitation phenomenon is accounted for through the $p - \theta$ Elrod-Adams mass-conserving cavitation model [23,24], which automatically satisfies the complementary JFO conditions for mass-conservation throughout the lubricated domain [25]. Thus, by assuming the moving texture configuration illustrated in Fig. 1 for constant sliding speed, the modified average Reynolds equation with $p - \theta$ cavitation model can be expressed as:

$$\frac{\partial}{\partial x} \left(\phi_{p_x} \frac{\rho h^3}{12\mu} \frac{\partial p_H}{\partial x} \right) + \frac{\partial}{\partial z} \left(\phi_{p_z} \frac{\rho h^3}{12\mu} \frac{\partial p_H}{\partial z} \right) = \frac{\partial}{\partial x} \left[\rho \theta \frac{U}{2} (h \phi_c + \sigma_R \phi_{s_x}) \right] + \frac{\partial (\rho \theta h \phi_c)}{\partial t}$$

Complementary conditions for cavitation:

$$(p_H - p_{cav})(1 - \theta) = 0 \rightarrow \begin{cases} p_H > p_{cav} & \rightarrow \theta = 1 & (\text{pressured regions}) \\ p_H = p_{cav} & \rightarrow 0 \leq \theta < 1 & (\text{cavitation regions}) \end{cases} \quad (1)$$

where $p_H(x, z, t)$ is the hydrodynamic pressure, $h(x, z, t)$ the geometry of the oil film thickness, U the sliding velocity of the moving (bottom) surface, μ the lubricant dynamic viscosity and $\theta(x, z, t)$ the film fraction cavitation parameter [23,24]. The coefficients ϕ_{p_x} and ϕ_{p_z} are the pressure flow factors, ϕ_{s_x} the shear flow factor along the sliding motion, σ_R the combined standard deviation of the surface roughness and ϕ_c the contact factor [26,27]. The lubricant film thickness can be written in the coordinate system $Oxyz$ attached to fixed pad as follows (see Fig. 1):

$$h(x, z, t) = h_T(x + Ut, z) + h_s(x, z) + h_0(t) \quad (2)$$

where h_0 is the reference minimum oil film thickness, h_s the fixed pad geometry and h_T the instantaneous geometry of the textured slider moving through the pad.

The solution of Eq. (1) is carried out numerically using a recently proposed finite volume discretization scheme, namely Element-Based

Finite Volume Method (EbFVM), for solving Reynolds equation with mass-conserving cavitation model on unstructured meshes [8].

2.2. Asperity contact model

The asperity contact pressures that arise due to the surfaces interaction under boundary and mixed lubrication conditions are calculated through the statistical-based Greenwood-Tripp (GT) model for rough contacts [28]:

$$p_{ASP} = \begin{cases} \frac{16\pi\sqrt{2}}{15} E^* (\eta_s^2 \beta_s^{3/2} \sigma_s^{5/2}) F_{5/2}(\bar{h}), & p_{ASP} < H_V \\ H_V, & p_{ASP} \geq H_V \end{cases} \quad (3)$$

where $p_{ASP}(x, z, t)$ is the asperity contact pressure, $\bar{h}(x, z, t) = \left(\frac{h - Z_s}{\sigma_s} \right)$ the dimensionless separation distance of the surfaces, E^* the combined elastic modulus, defined as $E^* = \left(\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right)^{-1}$, and H_V the Vickers' hardness of the softer material. The magnitude of the contact pressures depend on the following parameters associated with the statistical distribution of the asperity heights: Z_s the asperity mean height, σ_s the standard deviation of the asperity heights, β_s the asperity mean curvature radius and η_s the asperity density. The values of such rough contact parameters are deterministically calculated from 3D surface roughness measurements based on the methodologies described in [29]. In this case, every asperity is identified as the local maximum points above the surface reference plane and thus calculations of the asperity mean height, standard deviation, mean curvature radius and density are performed numerically. The function $F_{5/2}(\bar{h})$, representing the Gaussian distribution of the asperity heights is approximated by a polynomial function whose coefficients can be found in [20]. Notice that the threshold limit value of H_V imposed to the contact pressures extend the use of elastic GT model to elastic-perfectly plastic contacts. Furthermore, despite the limitations of the GT model it is still widespread for rough contact calculations and has proven to be robust and efficient to provide satisfactory predictions for surfaces with Gaussian asperity heights distribution as those considered in the present contribution (see Section 3.1).

2.3. Equilibrium and total friction equations

The equilibrium equation in y-direction of the system illustrated in Fig. 1 can be written as:

$$W_H(t) + W_{ASP}(t) - F_{ext}(t) = m \frac{d^2 h_0}{dt^2} \quad (4)$$

where $F_{ext}(t)$ is the external force acting on the fixed pad, m the equivalent mass of the vertical moving parts, and $W_H(t)$ and $W_{ASP}(t)$ the load-carrying forces generated by the hydrodynamic and asperity contact pressures, which are obtained from integration over the pad domain as:

$$\int_{-\frac{B}{2}}^{\frac{B}{2}} \int_{-\frac{L}{2}}^{\frac{L}{2}} [p_H(x, z, t) + p_{ASP}(x, z, t)] dx dz - F_{ext}(t) = m \frac{d^2 h_0}{dt^2} \quad (5)$$

where L and B are the pad width and the pad length in the z-direction, respectively (see Fig. 1).

Similarly, the total friction force acting on the fixed pad can be calculated by combining the dissipative forces promoted by the hydrodynamic and rough contact effects. Accordingly:

$$F(t) = \int_{-\frac{B}{2}}^{\frac{B}{2}} \int_{-\frac{L}{2}}^{\frac{L}{2}} \left[\underbrace{\left(\frac{h}{2} \frac{\partial p_H}{\partial x} \phi_{p_x} - \frac{\mu \theta U}{h} (\phi_f + \phi_{f_{s_x}}) \right)}_{\tau_H(t)} + \underbrace{(\mu_{bc} p_{ASP})}_{\tau_{ASP}(t)} \right] dx dz, \quad (6)$$

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