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To cite this article: H M Checo et al 2017 IOP Conf. Ser.: Mater. Sci. Eng. 174 012035

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Down to the roughness scale assessment of piston-ring/liner contacts

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Abstract. The effects of surface roughness in hydrodynamic bearings been accounted for through several approaches, the most widely used being averaging or stochastic techniques. With these the surface is not treated "as it is", but by means of an assumed probability distribution for the roughness. The so called *direct*, *deterministic* or *measured-surface* simulation) solve the lubrication problem with realistic surfaces down to the roughness scale. This leads to expensive computational problems. Most researchers have tackled this problem considering non-moving surfaces and neglecting the ring dynamics to reduce the computational burden. What is proposed here is to solve the fully-deterministic simulation both in space and in time, so that the actual movement of the surfaces and the rings dynamics are taken into account. This simulation is much more complex than previous ones, as it is intrinsically transient. The feasibility of these fully-deterministic simulations is illustrated two cases: fully deterministic simulation of liner surfaces with diverse finishings (honed and coated bores) with constant piston velocity and load on the ring and also in real engine conditions.

1. Introduction

Fluid-dynamic bearings are present in many key high-load, high-speed precision applications where other mechanisms like ball bearings would not perform as well due to noise and vibrations, or because it is even impossible to apply another solution. In car engines they are present in the piston assembly, crankshaft bearings, valves, valve seats, camshaft contacts and seals in general. The piston assembly accounts for 50% of the engine friction losses, which is approximately 10% of the total fuel consumption. Half of that takes place in the piston ring/liner contact [1], thus being one of the major contributors to the mechanical power losses. Thus, it is important to understand the key factors influencing film formation and friction in fluid film bearings.

All the mentioned engine components work under severe loading conditions. A complete separation of both bearing surfaces (hydrodynamic lubrication regime) is not assured, and partial solidsolid lubrication (mixed lubrication regime) or mainly solid-solid contact (boundary lubrication regime) takes place. Under these conditions the real topography of the surfaces is a determining factor in the bearing performance.

Honing is the most common machining process used by the industry in the finishing of engine liners. These micropatterned surfaces consist of grooves whose width and depth is in the tens of microns, and a plateau with even smaller features (the surface roughness), in the order of the micrometer or even lower. With a domain in the centimeters (size of the ring pack, journal bearing,

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seal, etc.), this enforces the use of very fine meshes which is traduced in a greater computational cost. In this work an efficient numerical method to solve thin film (lubrication) problems down to the roughness scale is presented. As an application, these methods will help with the assessment of surfaces actually being produced by the industry, to analyze their performance in the piston ring/liner contact.

This paper is organized as follows: a general model for fluid lubricated bearings is presented in Section 2, along with a brief overview of the numerical methods employed. Results of numerical simulations with several measured rough surfaces are given in Sections 3 and 4. Finally, conclusions are drawn in Section 5.

2. Modeling and numerical method

The arrangement of Figure 1 shows a ring sliding over a textured surface (the liner) with velocity -u(t). This corresponds to a section in the piston's movement direction, however, the mathematical formulation of the problem remains general enough to represent other lubricated contacts (e.g., journal-bearing-like devices). The domain is a strip $\Omega = (x_l, x_r) \times (0, w)$. Notice that the domain is two-dimensional, in the directions x_1 and x_2 . Figure 1 shows just a section along x_1 . The geometry is determined by the functions h_U and h_L describing the upper (fixed) surface and lower (moving) surfaces respectively, and Z(t), which parametrizes the ring movement.



Figure 1. Section of the domain along the ring's direction of motion, along with the forces acting on it.

The ring dynamics is governed by Newton's Second Law. The forces W(t) acting on it are the applied ones (force of the gases of the combustion chamber $W^{gp}(t)$, elastic forces W^{ps}) and the reactive forces. The latter ones come from the interaction between the ring and the lubricating oil (the hydrodynamic forces) and between the ring and the liner (contact forces), here named $W^h(t)$ and $W^c(t)$ respectively. $W^c(t)$, usually given by the Greenwood-Tripp model [14], is a function of the separation Z(t) between the surfaces. To be able to compute W^h , the hydrodynamics must be solved first. Due to the geometry of the contacts and the high velocities involved *cavitation* takes place. It has been shown by Ausas et al [3] that a mass-conserving model is of foremost importance, which is confirmed by several authors, among them [4] who performed a careful comparison with experiments. Here we employ the Elrod- Adams model [5], which includes into the Reynolds equation conservative boundary conditions for cavitation.

The complete mathematical problem [6] to be solved reads:

"Find trajectory Z(t), and fields p(t), $\theta(t)$, defined on $\Omega \times [0,T] = \{(x_{1l}, x_{1r}) \times (0,w)\} \times [0,T]$ and periodic in x_2 , satisfying

13th International Conference on Tribology, ROTRIB'16

IOP Conf. Series: Materials Science and Engineering 174 (2017) 012035 doi:10.1088/1757-899X/174/1/012035

$$\begin{cases} Z(0) = Z_0, Z'(0) = V_0, \\ \theta(x_{1l}, x_2, t) = \frac{d_{oil}}{h(x_{1l}, x_2, t)} \,\forall x_2 \in (0, w), \forall t \\ \theta(x_{1r}, x_2, t) = \frac{d_{oil}}{h(x_{1r}, x_2, t)} \,\forall x_2 \in (0, w), \forall t \\ p > 0 \Rightarrow \theta = 1, \theta < 1 \Rightarrow p = 0, 0 \le \theta \le 1 \end{cases}$$
(1)

and

$$m\frac{d^2Z}{dt^2} = W(t) + W^h(t) + W^c(t), \ W(t) = W^{gp}(t) + W^{ps},$$
(2)

$$\nabla \cdot \left(\frac{h^3}{12\mu}\nabla p\right) = \frac{u(t)}{2}\frac{\partial h\theta}{\partial x_1} + \frac{\partial h\theta}{\partial t}$$
(3)

where

$$h(x_1, x_2, t) = \begin{cases} h_L(x_1 - ut, x_2) + h_U(x_1, x_2) + Z(t) & if a \le x_1 \le b \\ h_L(x_1 - ut, x_2) + e & otherwise \end{cases},$$
(4)

$$W^{h}(t) = \frac{1}{w} \int_{a}^{b} \int_{0}^{w} p(x_{1}, x_{2}, t) dx_{1} dx_{2}, W^{c}(t) = \frac{1}{w} \int_{a}^{b} \int_{0}^{w} p_{con}(x_{1}, x_{2}, t) dx_{1} dx_{2},$$
(5)

and the functions h_L , h_U , W(t), and u(t) are known explicitly, d_{oil} and e are positive constants, m is the ring's linear mass and the contact pressure p_{con} is given by the Greenwood-Tripp model".

The honed surface is set on the moving surface of the lubricated pair. Moving textures are more challenging from the numerical point of view since the problem becomes intrinsically transient. Then, the dynamics of the ring must be solved coupled with the hydrodynamics and the other forces involved.

The tribological quantities computed are the clearance and the hydrodynamic lift

$$C(t) = \min_{(x_1, x_2) \in \Omega} h(x_1, x_2, t), \ L(t) = \frac{1}{w} \int_a^b \int_0^w p(x_1, x_2, t) dx_1 dx_2,$$
(6)

and the total friction force

$$F(t) = \frac{-1}{w} \int_{a}^{b} \int_{0}^{w} \left(\mu \frac{ug(\theta)}{h} + \frac{h}{2} \frac{\partial p}{\partial x_{1}} - p \frac{\partial h_{L}}{\partial x_{1}} + \mu_{s} p_{con} \right) dx_{1} dx_{2}$$
(7)

It can be noticed that the last term of equation (7) is the contact friction force, and the rest of the terms represent the hydrodynamic friction force. The instantaneous power loss is also computed

$$P(t) = F(t)u(t).$$
(8)

Function g in equation (7) is defined by

$$g(\theta) = \begin{cases} 1 & if\theta > \theta_s \\ 0 & otherwise \end{cases}$$
(9)

The parameter θ_s is a threshold for the onset of friction, which can be interpreted as the minimum oil fraction for shear forces to be transmitted from one surface to the other. Here is taken as $\theta_s=0.95$ for all cases.

The conservative finite volume method described in [7] is used to discretize the Elrod-Adams equation (eq. (3)), coupled with the dynamics through a Newmark scheme. Results are accelerated through a multigrid implementation [8] and shared-memory parallelization.

3. Measured surface simulations

The usual treatment given to surface roughness by means of stochastic models [9, 10], averaged models [11, 12] or homogenization methods [13, 14, 15, 16] is avoided. The measured topography of the surface is employed, and its impact on tribological quantities of interest is studied.

To describe each of these surfaces, we resort to their Abbott-Firestone parameters [17]. Its related roughness parameters R_k (plateau height), R_{pk} (reduced peak height), R_{vk} (reduced valley height), M_{r1} (percentage of R_{pk} peaks) and M_{r2} (1- M_{r2} can be understood as the percentage of R_{vk} valleys) are shown in table 3.

To compare performance in terms of friction and clearance, a fixed load W(t) = W = -0.6N was imposed on the ring. This load was chosen large enough so that the small clearance enforces a deterministic simulation, but not as high as to originate contact between the ring and the liner surface. For the same reason, surface A was omitted from this group of simulations. It is a surface with large peaks as its parameter $R_{pk}=2.64 \ \mu m$ shows, and consequently contact takes place for the chosen load. Surface data pre-processing (peak removal and surface smoothing) will be treated in the next section.

The ring's linear mass was set to m = 0.01 kg/m, and its initial position for all cases is $Z_0=1.5 \ \mu m$. Enough oil is provided at the entrance $(d_{oil} = 10 \ \mu m)$ so as to achieve fully flooded conditions, and variable is set to the large value $e = 500 \ \mu m$. No contact takes place in the simulations shown in this section, so $p_{con} = 0$ and 6000 time steps are solved with

 $\Delta t = 0.1 \ \mu s$. For the simulations here performed the lowermost strip of width w = 0.2 mm ($(x_1, x_2) \in [0, L_s] \times [0, 0.2]$) is chosen from each of these surfaces. This choice of domain size is a good trade-off between computational time and representativity of the whole sample [8]. The domain is discretized in 2000×400 finite volumes. Details of these simulations can be seen in [8].

Minimum clearance C(t) and friction force F(t) are shown in Figure 2 for a periodic solution in each measured surface. It seems proper to recall at this point that each measured-surface is extended periodically so as to fulfill the complete distance travelled by the ring.

Results were obtained for rings with circular profiles with curvature radius R = 8 and 128 mm. The surfaces with lowest minimum clearance, listed from smaller to larger values are: D, B, C, F and E. This can be understood taking a look at its respective R_{pk} values: 0.85, 0.84, 0.71, 0.14 and 0.18 μ m. Surfaces with higher asperities have a lower minimum clearance C.

C and the averaged friction force F(t) over each period $2L_s$ ($L_s \times L_s$ is the sample size) are given in Table 2. In spite of the minimum clearance being closely related to wear, it might not always correlate well with hydrodynamic friction. Surface D presents the largest friction force for R=8 but the second largest for R=128. This makes surface F particularly interesting, given that it shows a higher C with higher friction. This can be explained by looking at Z(t) for each surface, as in Figure 3(b). It is much lower for F, leading to a lower average h which means higher shear stresses. The high peaks of surface D ($R_{pk}=0.85$) induce its small minimum clearance.

Surface	L_s (mm)	Resolution (µm)
A	1.241	0.982
В	1.241	0.982
С	0.84	0.822
D	0.84	0.822
E	1.8	2.0
F	1.0	1.0

Table 1. Dimensions and resolution of the surfaces of measured topography.



Figure 2. Minimum clearance C(t) and friction force F(t) for five of the six measured surfaces of Table 3. Only the last period of the periodic solution is shown for each surface. Figures (a) and (b) display results for R = 8 mm, while Figures (b) and (d) do the same for R=128 mm.



Figure 3. Z(t) for (a) R=8 mm and (b) R=128 mm for the five of the six measured surfaces of Table 3.

4. Engine cycle simulations

Let us inspect for a moment the surface *E*. Each measurement point corresponds to a $2\mu m \times 2\mu m$ cell. This gives for the measured surface a total of 810.000 measurement points. Computational domains are usually larger than the sample size, so the total number of cells required to solve the problem can reach several million. Convergence studies suggest a mesh spacing of one fourth of the measurement with a Courant number ($u_{max} \frac{\Delta t}{\Delta x}$) of 1.0 [18]. This further increases the computational burden.

	R = 8 mm		R = 128 mm	
Surface	$F(t) (10^{-3}N)$	С (µт)	$F(t) (10^{-3}N)$	С (µт)
В	5.21	0.13	9.45	0.11
С	4.92	0.35	8.50	0.32
D	5.45	0.02	9.74	0.04
Ε	4.95	0.84	8.34	0.87
F	5.2	0.81	11.47	0.56

 Table 2. Friction and minimum clearance for the measured surfaces of table 3.

Deterministic simulations of actual liner surfaces face a wide variety of challenges. Ring, piston and fluid dynamics, contact forces among others must be modeled. Once this has been dealt with, there remain difficulties associated with the numerics. Supposing a domain large enough to hold all three rings of a pack, of 15 mm in the movement direction and 1.8 mm in the radial direction (the complete width of the measured surface of figure 2), a structured mesh large enough to capture the level of detail of the surface would have approximately 32 million finite volumes. If a Courant number of 1.0 is to be kept with a constant time step Δt , more than a million iterations in time will be necessary to simulate just one engine cycle. This, in a single desktop machine will take **months** to compute. However, some pre-processing and approximations with acceptable error can be done to decrease the computational time.

5. Engine cycle simulations with a single compression ring

Surfaces A and B are used in the simulations in this section. As the samples sizes are 1.241mm in x_1 and 0.913 in the x_2 direction it must be repeated somehow all along the stroke, which for the provided engine data is 156mm. The election here was the most straightforward: they were repeated contiguously until the 156mm of the stroke were fulfilled.

The ring geometry employed is the one of a compression ring used, which is shown in Figure 4, while the engine data is given in Figure 5, corresponding to a rotational speed of 1900 rpm. The stroke's length is 156 mm and the bore diameter 132 mm. This constitutes a challenging simulation, as the piston velocity and pressure coming from the combustion chamber are really large.

Different liner configurations were considered. Each configuration is made of three regions as illustrated in Figure 8, for which the A and the B textures are alternatively used. The three cases actually considered are given in the table below.

Liner	Region I	Region II	Region III
1	Α	В	Α
2	В	Α	Α
3	В	Α	В

Table 3. Textures on each region of the liner.

The sizes of the different regions are also indicated in the figure as well as the position of the top dead center (TDC) and the bottom dead center (BDC). In behalf of this and Figure 7, the ring's transition between regions during the power stroke are at the 38.2 (region III to II) and 140.0 (region II to I) crank angle degrees. Details of the physical and numerical parameters adopted in the simulations are given in [8].



Figure 4. Section of the domain showing the ring and liner arrangement used in every simulation in this section.



Figure 5. Gas pressure (top) and piston velocity (bottom) as functions of the crank angle.

6. Comparison of results for the three liners

Results for the three liners are shown in Figure 6. These simulations have taken approximately 20 hours on a desktop computer with a six-cored Intel i7-4930K processor at 3.4GHz. The plotted variables are the minimum clearance, the contact friction force, the total friction force and the total friction power as functions of the crank angle. The region over which the ring is sliding at each angle is marked on the horizontal axis of the top figure. Jumps in the solutions are due to the differences in roughness between surfaces *A* and*B*.



Figure 6. Minimum clearance, total friction force and friction power loss as functions of the crank angle for the three different liner configurations.

The maxima in clearance are reached close to the mid-strokes, simultaneously with the highest piston velocities. Clearance reaches its minimum points at the TDC and BDC, being its global minimum at the TDC during the power stroke. Notice that some surface penetration occurs. This is due to the highest loads on the ring taking place during the explosion in the combustion chamber. As piston velocity is null close to the dead centers, friction force is mostly due to contact.

The behavior of the three liner arrangements is similar. In general the B texture exhibits less friction at each location. Averaging the friction power over the four-stroke cycle we obtained:

Liner	Average power loss
1:A - B - A	442 W/cylinder
2:B - A - A	519 W/cylinder
3:B - A - B	479 W/cylinder

Table 4. Average power loss per cylinder for each configuration.

The least friction losses correspond to Liner 1, which has texture B at Region II (middle) and A at Region III (top). The intermediate friction losses correspond to Liner 3, which has texture A at Region II and B at Region III. In the current configuration the region in which the smoother texture B is more beneficial is Region II (middle).

7. Conclusions

The modeling, numerical treatment and numerical simulations of surfaces with measured topography in hydrodynamic and mixed lubrication regime have been summarized in this work. These surfaces with measured topography move with respect to the ring surface (in a piston ring/liner contact) and thus are intrinsically transient. Cavitation effects are taken into account and also the ring dynamics, which are governed by the hydrodynamic forces and the solid-solid forces arising from the contact of both the ring and the liner.

Dynamic simulations are performed on several samples of measured surfaces, corresponding mostly to honed cylinder liners of car engines. The effects of surface roughness are discussed. Some strategies to decrease the computational time in highly demanding complete engine cycle simulations while keeping errors at a low level are presented, and the results of these simulations compared. These studies were made possible due a multigrid implementation and a shared memory parallelization.

Acknowledgements

The authors thank the financial support of this work provided by CAPES (grant PROEX-8434433/D), FAPESP (grants 2011/24147-5 and 2014/19249-1) and CNPq (grant 447904/2014-0).

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Texture	$x_1[mm] \times x_2[mm] \times h_L[\mu m]$	Abbott-Firestone parameters	Texture	$x_1[mm] \times x_2[mm] \times h_L[\mu m]$	Abbott-Firestone parameters
A	4 0.8 0.6 0.4 0.2 0 0 0 0 0 0 0 0 0 0 0 0 0	$R_{pk} = 2.64 \ \mu m$ $R_{k} = 0.91 \ \mu m$ $R_{vk} = 7.97 \ \mu m$ $M_{r1} = 5\%$ $M_{r2} = 74\%$	D	$\begin{array}{c} 0.8 \\ 0.6 \\ 0.4 \\ 0.2 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ $	$R_{pk} = 0.85 \ \mu m$ $R_{k} = 0.86 \ \mu m$ $R_{vk} = 1.57 \ \mu m$ $M_{r1} = 9\%$ $M_{r2} = 86\%$
В	$\begin{array}{c} 0.8 \\ 0.6 \\ 0.4 \\ 0.2 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ $	$R_{pk} = 0.84 \ \mu m$ $R_{k} = 0.62 \ \mu m$ $R_{vk} = 2.94 \ \mu m$ $M_{r1} = 7\%$ $M_{r2} = 79\%$	Ε	$ \begin{array}{cccccccccccccccccccccccccccccccccccc$	$R_{pk} = 0.18 \ \mu m$ $R_{k} = 0.21 \ \mu m$ $R_{vk} = 6.46 \ \mu m$ $M_{r1} = 3\%$ $M_{r2} = 77\%$
С	$\begin{array}{c} 0.8\\ 0.6\\ 0.4\\ 0.2\\ 0\\ 0\\ 0\\ 0\\ 0\\ 0\\ 0\\ 0\\ 0\\ 0\\ 0\\ 0\\ 0\\$	$R_{pk} = 0.71 \ \mu m$ $R_{k} = 0.27 \ \mu m$ $R_{vk} = 9.94 \ \mu m$ $M_{r1} = 8\%$ $M_{r2} = 85\%$	F	$\begin{array}{c} 1 \\ 0.8 \\ 0.6 \\ 0.4 \\ 0.2 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ $	$R_{pk} = 0.14 \ \mu m$ $R_{k} = 0.21 \ \mu m$ $R_{vk} = 10.6 \ \mu m$ $M_{r1} = 5\%$ $M_{r2} = 71\%$