Influence of Boundary Conditions on Starvation of Piston Ring Conjunction

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Abstract

It is important to determine realistic inlet boundary conditions to correctly predict lubricant film thickness and generated frictional power losses in all tribological conjunctions. This is also true of piston compression ring as well. A 2D hydrodynamic solver using Reynolds equation is to analyse the differences between predicted conditions with a flooded inlet and that arising from a more realistic determined zero-reverse boundary condition for lubricant flow post inlet wedge stagnation point. The case of a cylinder of a 4-cylinder 4 stroke gasoline engine, running at the engine speed of 1500rpm is considered. The results show that with a more detailed and realistic inlet boundary a significant reduction in the minimum film thickness is predicted which leads to increased friction throughout the engine cycle.

Keywords: Piston compression ring; inlet boundary condition, Starvation

Nomenclature

b	Ring face-width
c	Ring crown height
d	Ring radial width (thickness)
g	Ring end gap
k	Contact speed ratio
l	Connecting rod length
P_c	Combustion pressure
r	Crank-pin radius
r_0	Nominal bore radius

Greek Symbols

ϑ_i	Ratio of the film thickness at the stagnation point to the central contact film thickness
ϑ_e	Ratio of the film thickness at the rupture point to the central contact film thickness
K	Conformability coefficient (factor)
ρ_0	Lubricant density at atmospheric pressure
σ	RMS composite surface roughness
ς	Pressure coefficient of boundary shear strength of asperities
τ_0	Eyring shear stress

1. Introduction

Improved fuel economy and reduction of emissions are key drivers for increased efficiency of internal combustion (IC) engines. In an IC engine 15-20% of the overall losses can be attributed to friction, 40-50% of which are as a result of the piston-cylinder system [1, 2]. Therefore, accurate numerical prediction of frictional losses is quite important, which is critically dependent on all assumptions made, particularly the employed boundary conditions.

A fully-flooded inlet is usually assumed in most hydrodynamic analyses, including for piston ring to cylinder liner contact [3, 4, 5]. However, experimental investigations with instrumented floating liners have shown that for much of the engine cycle in situ generated friction points to the prevalence of mixed and boundary regimes of lubrication with observed wear, particularly at piston dead centre reversals. One reason for this is lack of a sufficient lubricant meniscus the inlet wedge of the contact for parts of the piston cycle [6, 7, 8]. Even with a sufficient volume of lubricant at the inlet to the conjunction, there are swirl and reverse flows which reduce the inbound volume of lubricant into the conjunction as shown in [9, 10]. The same is true of many other contacts, including under elastohydrodynamic conditions in rolling element bearings, observed as well as predicted [3, 10, 11, 12].

This paper demonstrates the necessity for determination and use of realistic inlet boundary conditions to correctly predict lubricant film thickness and generated frictional power losses.

2. Methodology

Tipei [9] showed that although the two-dimensional solution of Reynolds equation is applicable for practical problems, the justification for a two-dimensional flow continuity condition is only valid for special cases. It was shown experimentally that only a fraction of the inlet lubricant flow was drawn between the two surfaces for a given surface, load and speed, forming swirl on the upstream with reverse flow from the contact [12, 13].

By applying the Swift-Steiber boundary condition at the contact inlet ($dp/dx = 0$ and $p = 0$), Tipei [9] derived an expression between the film thickness at the inlet and outlet exit positions maintaining the continuity of flow condition as:

$$\frac{\cosh \vartheta_e}{\cosh \vartheta_i} = 1 - \frac{f(k)}{6(1+k)} \tag{1}$$

and:

$$\tanh\vartheta_e - \left[1 - \frac{f(k)}{6(1+k)}\right]\tanh\vartheta_i$$

$$-\left[1 - \frac{f(k)}{6(1+k)}\right]\cosh\vartheta_i \qquad (2)$$

$$\times\left[\arcsin(\tanh\vartheta_e) - \arcsin(\tanh\vartheta_i)\right] = 0.$$

ϑ_e is the ratio of the lubricant film thickness at the exit position to that at the centre of the contact, whilst ϑ_i is the same for the inlet position. k is the ratio of the velocities of the contiguous surfaces and $f(k)$ is a characteristic determined by the value of k. Calculation of the parameters from Equations (1) and (2), obtaining the minimum film thickness and distribution across the conjunction allows for calculation of the inlet stagnation and outlet separation points from the centre of the ring. The work of Mohammadpour et al. [11] and Birkhoff and Hays [12] agree with this assumption.

Using the proposed method of Tipei [9], it is possible to determine the position where an inlet boundary condition should be applied, as opposed to assuming a fully flooded inlet and in order to maintain flow continuity.

3. Results and Discussion

The proposed method is used to simulate an engine cycle of a C-segment vehicle with a 4-cylinder 4-stroke gasoline engine at a running speed 1500rpm with full throttle. The chosen speed corresponds to the vehicle running at 35km/h in 3rd gear in the New European Drive Cycle (NEDC), which is important for measurement of emission, particularly in cold operating condition. The engine torque is 52.03Nm.

The model uses finite differences for the solution of a 2D Reynolds equation to obtain the generated hydrodynamic pressure distribution as shown by Bewsher et al. [14]. The input parameters for the engine, lubricant and material surfaces are provided in Tables 1, 2 and 3. Figure 1 shows the combustion pressure, P_c, throughout the engine cycle.

Table 1: Engine data

Parameters	Values	Units
Torque	52.03	Nm
No. of cylinders	4	-
Engine type	Gasoline SI	-
Crank-pin radius, r	39.75	mm
Connecting rod length, l	138.1	mm
Bore nominal radius, r_0	44.52	mm
Ring crown height, c	10	μm
Ring axial face width, b	1.15	mm
Ring radial width, d	3.5	mm
Ring free end gap, g	10.5	mm

Table 2: Lubricant properties in atmospheric pressure and 40°C

Parameters	Values	Units
Lubricant viscosity, η_0	0.05	kg/ms
Lubricant density, ρ_0	833	kg/m³
α_0	1 x 10⁻⁸	m²/N
Eyring shear stress, τ_0	2	MPa

Table 3: Material properties and surface topographical parameters

| Parameters	Values	Units
Liner material	Grey cast iron	-
Modulus elasticity of liner material	92.3	GPa
Poisson ratio for liner material	0.211	-
Density for liner material	7200	kg/m³
Ring material	Steel SAE 9254	-
Modulus elasticity of ring material	203	GPa
Poisson ratio for ring material	0.3	-
Roughness parameter ($\zeta K\sigma$)	0.04	-
Measure of asperity gradient (σ/K)	0.001	-
Shear strength of asperities (ς)	0.17	-
Density for ring material	7700	kg/m³
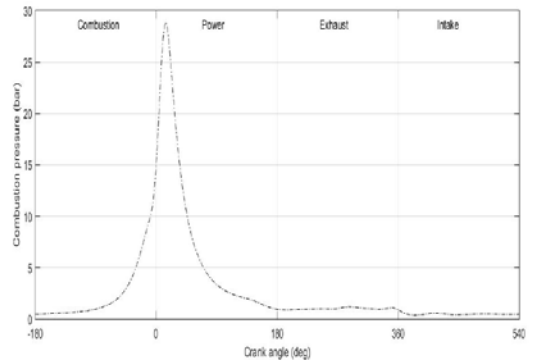
Figure 1   Combustion Pressure

The results obtained are for the predicted minimum film thickness and total friction within the piston ring conjunction. The position of top dead centre (TDC) occurs when the crank angle is at $0°$ at the onset of the power stroke, with combustion occurring at $20°$.

Figure 2 shows the minimum film thickness variation within the piston ring conjunction. It can be seen that by applying Tipei's inlet condition, placing the inlet boundary at the stagnation point, where no reverse flow takes place, there is a reduction in the lubricant film as a result of inlet starvation. There is maximum

film thickness reduction of 4 µm at the midpoint of each engine stroke. This reduction in minimum film thickness leads to increased conjunctional friction as shown in Figure 3.
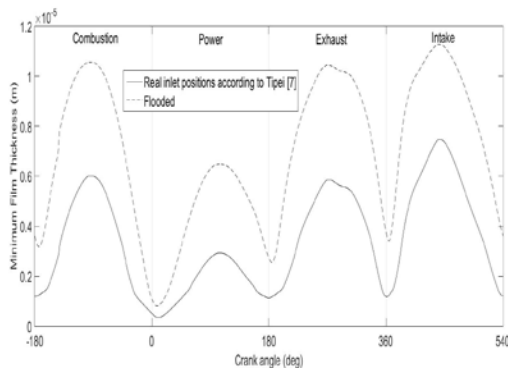

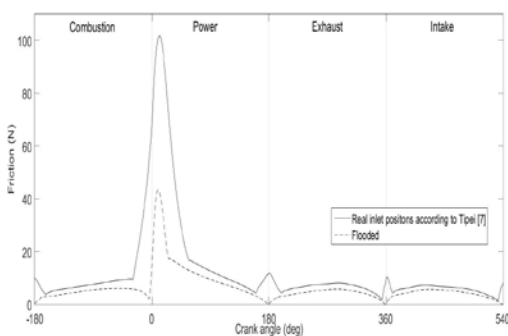*Figure 2   Cyclic variation of minimum film thickness*

Figure 3   Cyclic total (asperity and hydrodynamic) friction

4. Concluding Remarks

The paper presents the effect of boundary conditions on starvation of the piston ring conjunction. The analysis shows that by correct positioning of the inlet boundary there is a reduction in the thickness of entrained lubricant to the conjunction, and thus increased frictional power losses. The results show that by accounting for starvation within the piston ring conjunction and not assuming a fully flooded condition it is possible to bridge the gap between theoretical predictions and experimental measurements.

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6. References

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