Tribological study of hydrodynamic behavior piston ring / cylinder

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Abstract

The hydrodynamic behaviour of piston ring-cylinder liner is complex and it is difficult to predict its efficiency at the design stage. The piston ring has been the subject of many theoretical and experimental investigations from many viewpoints. In all these investigations, a known profile is assumed for the piston ring face and the minimum oil film thickness, hdrodynamic pressure distribution is computed under a multitude of simplifying assumptions. This project aims to develop a numerical model of piston ring dynamics and lubrication in internal combustion engines, it is currently estimated that the piston ring - cylinder bore friction accounts for up to 25% of the power loss in a typical engine, while oil transported to the combustion chamber by the piston and ring-pack contributes significantly to engine emissions. A profile of piston ring model was first developed to allow fast calculation of approximate piston ring dynamics.
The finite difference is applied to solve the governing equations of lubrication of the piston rings and to calculate the hydrodynamic pressure. The ring is assumed to have a circular profile in the direction of motion. This profile changes with time because tilting of the ring with the engine cycle is taken into account.

Keywords: *piston ring, hydrodynamic lubrication, friction coefficient, oil viscosity*

1. Introduction

The piston top compression ring plays a vital role in an efficient engine operation as it prevents the combustion gas leakage and allows heat dissipation, but contributes towards mechanical friction. Under severe operating conditions, the ring-block interface contributes about 20% of the total engine mechanical frictional loss [1-2]. Hence, the piston rings are lubricated by oil, the film thickness of which results in low friction and reduced wear. Major factors affecting the oil film thickness are bore distortion, piston speed, lubricant viscosity, top ring face profile, ring flexibility and boundary conditions.

The Parabolic face profile has the advantage that it tends to be self perpetuating under wear since ring tends to rock inside its groove during reciprocating movement and causes preferential wear of its edges [3]. This model generates hydrodynamic pressure fields and minimum hydrodynamic film thickness profiles as functions of engine crankshaft rotation of 720 degree, apart from calculating the friction coefficient. Influence of load on the tribological condition in piston ring and cylinder liner contacts in a medium-speed diesel engine[4]. The experimental method for measuring the oil-film thickness between the piston-ring and cylinder-wall of internal combustion engines and compressor friction influence the minimum film theckness [5-6].

2. Reynolds equation

When hydrodynamic or mixed lubrication occurs, an averaged flow-factor Reynolds analysis is used to model the lubricant pressure and flows, and the interaction between the lubricant and surface asperities. Hydrodynamic support of the ring load depends on a “wedge” effect in which relative motion between sliding surfaces and changing flow area combine to increase pressure in the lubricant. The fluid pressure is then able to support an external load.

Because of this effect, a positive pressure increase will occur in the oil in the converging section of the ring/liner interface, and pressure will decrease in the diverging section.

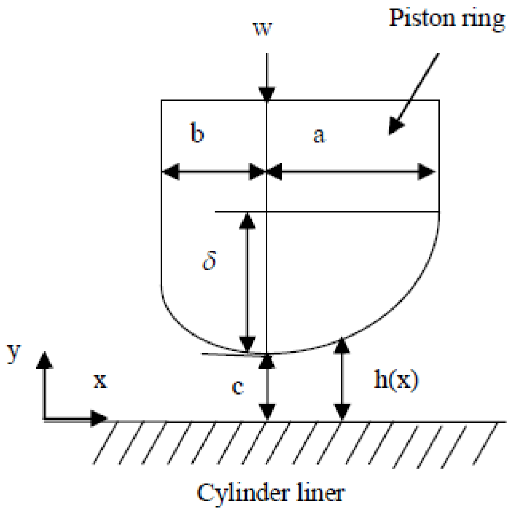

Fig. 1 Surface shape of the compression ring

Analysis of the lubricant pressure and flow between ring and liner is based on Reynolds' equation, which is applicable for thin film flows where viscous phenomena dominate fluid inertia. The Reynolds relationship is derived from conservation of momentum for the fluid (Navier-Stokes relations) and conservation of fluid mass (Continuity), and (for a one-dimensional system) is given by [7]:

$$\frac{\partial}{\partial x}\left(\frac{h^2}{m}\frac{\partial P}{\partial x}\right) = -6U\frac{\partial h}{\partial x} + 12\frac{dc}{dt} \qquad (1)$$

x: the coordinate distance reckoned from the leading edge of the ring ;h :oil film thickness;P: hydrodynamic pressure;U: piston velocity in axial direction; C: minimum film thickness; t :time; m: oil mass

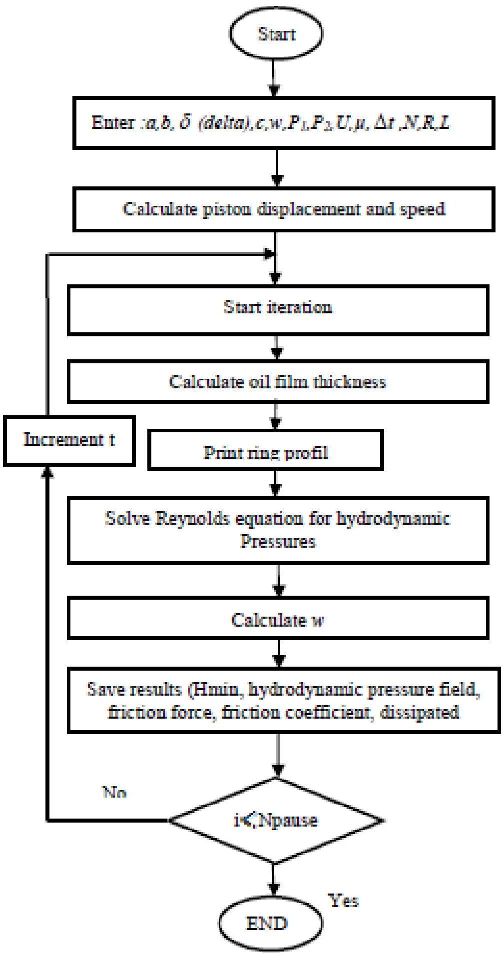

Fig. 2 Flowchart of computational scheme

3. Numerical results and discussion
3.2 Effect of oil viscosity

for δ (delta)=0.25

Fig. 3 show the hydrodynamic pressure between the cylinder and the ring. This pressure is maximum at the point C due to the influence of the profile of the ring and of the combustion gases on the oil film; the hydrodynamic pressure is proportional to the viscosity of the oil film.

Fig 4 and 5represents the variation of the minimum film thickness of the oil film and the force of friction, the force of friction is inversely proportional to the viscosity of the oil film and the minimum film thickness to the crank angle.Fig 6 and 7 represent the variation of the dissipated power and the friction coefficient proportional to the two parameters crank angle and inversely proportional to the viscosity of the oil film.

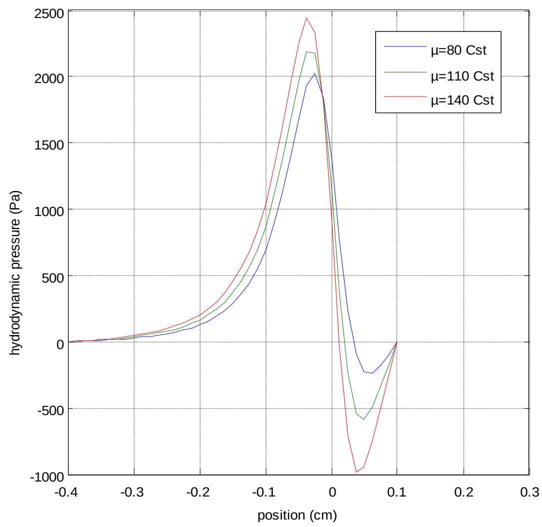

Fig.3 The hydrodynamic pressure with different oil viscosity

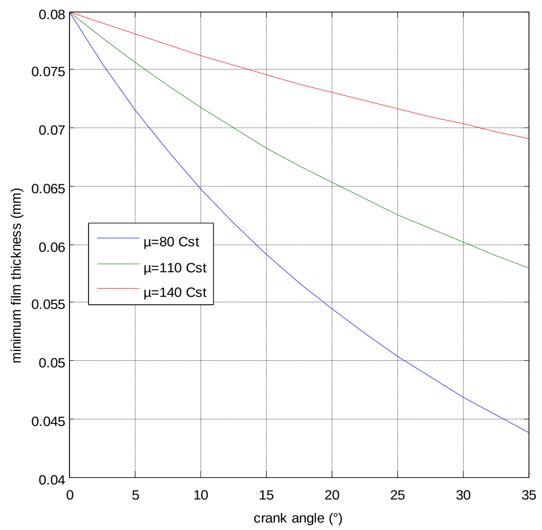

Fig. 4 Minimum film thickness Vs crank angle

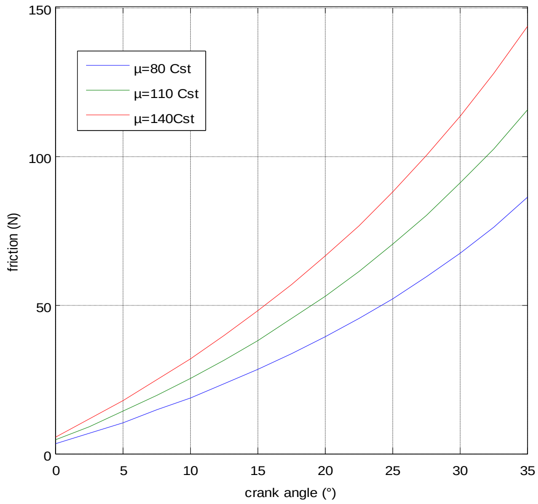

Fig. 5 Friction force Vs crank angle

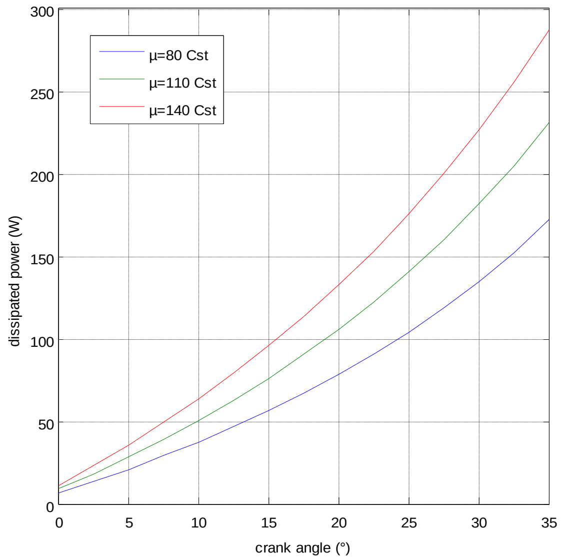

Fig. 6 Dissipated power Vs crank angle

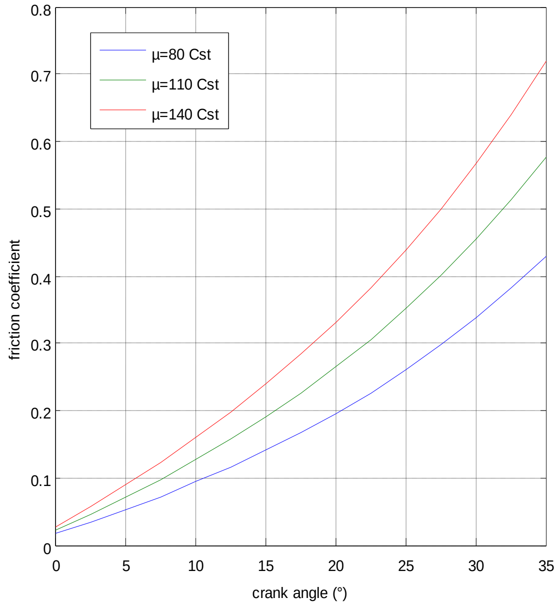

Fig. 7 Coefficient of friction Vs crank angle

4. Conclusion

The results of tribological characteristics such as the movement of the piston, hydrodynamic pressure the minimum film thickness, the friction force, dissipated power and the coefficient of friction were studied in relation to the viscosity lubricant, ring profile and load *w*. Oil viscosity directly affects friction in the hydrodynamic regime. Indeed, the hydrodynamic friction increases with viscosity. The viscosity also indirectly affects the contact friction by determining the oil film thickness. The reduction in viscosity can reduce the hydrodynamic friction, but also leads to a reduction in the oil film thickness.

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