Thermoelastohydrodynamics of a rough piston compression ring-to-cylinder bore conjunction

This item was submitted to Loughborough University's Institutional Repository by the/ an author.

Citation: BAKER, C.E. ... et al, 2011. Thermoelastohydrodynamics of a rough piston compression ring-to-cylinder bore conjunction. STLE 66th Annual Meeting & Exhibition, 15th-19th May 2011, Atlanta, Georgia, USA, pp.182-188.

Additional Information:

- This conference paper is closed access.

Metadata Record: [https://dspace.lboro.ac.uk/2134/14004](https://dspace.lboro.ac.uk/2134/14004)

Version: Accepted for publication

Publisher: STLE

Please cite the published version.
Abstract:
A thermo-elastohydrodynamic analysis of piston compression ring conjunction is presented. Regime of lubrication alters with contact load and piston kinematics. These affect the parasitic losses, improve fuel efficiency and emissions. These are significant, particularly in high performance engines. Lubricant film thickness determines the mechanisms of friction; viscous shear and boundary friction. The friction generated heat reduces the effective lubricant viscosity. Salient features include in-situ ring geometry within an out-of-round bore and ring in-plane modal behaviour. The results show thin films, promoting some asperity interactions. The analysis is a guide for surface modification (coating and/or textured features) to improve lubrication.

Introduction:
Parasitic losses account for 20% of all losses in an IC engines, thus reducing them would improve fuel efficiency. This is an important objective because of fuel costs and diminishing fossil fuels. Frictional losses are a significant proportion of all parasitic losses. They occur in load bearing conjunctions in IC engines, particularly in the piston system, which account for nearly half of all these losses, 60-70% of which are due to ring-pack. Thus, much work is devoted to lubricant rheology and surface topography. The plethora of interacting phenomena, such as lubricant rheology, surface topography, contact geometry, combustion strategy, contact kinematics and system dynamics makes for a complex multi-scale problem. Progressively, predictive methods are used as a cost saving tool in design and development instead of physical testing. The initial approaches were based on isothermal analysis of ring-bore hydrodynamic and later elastohydrodynamic conjunctions [1-3]. However, like many thin film conjunctions, the generated temperatures affect lubricant viscosity, thus the load carrying capacity and viscous friction. Hence, a thermo-elastohydrodynamic (TEHD) analysis is
required. For example, Ghosh and Gupta [4] have shown that a significant difference in load-carrying capacity, film thickness and rolling traction at high speeds results, when considering thermal effects. For TEHD combined solution of Reynolds and energy equations is required. The approach is through numerical analysis, such as that reported by Almqvist and Larsson [5]. It is also necessary to account for the temperature of the bounding surfaces, such as the compression ring and the bore. This together with numerical solution of TEHD yields long computation times. It is contrary to industrial time-scales. Hence, closed form analytical solutions are desired, which are viewed as rather elegant. A closed form solution is provided by Dunaevsky and Vick [6].

This paper presents TEHD of piston compression ring-bore conjunction, taking into account conduction through the bounding surfaces and convection through lubricant flow. Furthermore, it is a fast analytic solution.

1- The Analytical Approach:

Heat is generated in the contact through viscous shear of the lubricant as well as compressive heating. Some of this heat is carried away by lubricant entrainment (x direction) and through side leakage. The three dimensional general form of the energy equation is:

$$\rho c_p \left( u \frac{\partial \theta}{\partial x} + v \frac{\partial \theta}{\partial y} \right) - \alpha \theta \left( u \frac{\partial p}{\partial x} + v \frac{\partial p}{\partial y} \right) = k \frac{\partial^2 \theta}{\partial z^2} + \eta \left( \frac{\partial u}{\partial z} \right)^2 + \left( \frac{\partial v}{\partial z} \right)^2 $$

(1)

One can neglect side leakage of the lubricant in the piston ring-bore conjunction. Therefore [7]:

$$\rho c_p u \frac{\partial \theta}{\partial x} - \nu \theta u \frac{\partial p}{\partial x} = k \frac{\partial^2 \theta}{\partial z^2} + \eta \left( \frac{\partial u}{\partial z} \right)^2 $$

(2)

An order of magnitude analysis by Gohar and Rahnejat [7] shows little convection for thin films, thus:

$$-k \frac{\partial^2 \theta}{\partial z^2} = \nu \theta u \frac{\partial p}{\partial x} + \eta \left( \frac{\partial u}{\partial z} \right)^2 $$

(3)

An analytic solution to the above equation is obtained by Karthikeyan et al [8], superimposing the effects of compressive and viscous shear heating for grease lubricated bearings. Note that greases do not readily convect heat. Their analytic solution can be extended, by not resorting to superposition:

$$-k \frac{\partial \theta}{\partial z} = \nu h \frac{\partial p}{\partial x} \int_0^h \theta u dz + \eta h \left( \frac{\partial u}{\partial z} \right)^2 $$

(4)

The left hand side of (4) represents the heat flow through conduction and can be stated in the form [8]: $-k(\partial \theta/\partial z)=k(\Delta \theta/h)$, where: $\Delta \theta=\theta_0-\theta$, this being the average temperature rise at any location x within the contact. If $\theta_0$ is the temperature at the inlet, then: $\theta=\theta_0+\Delta \theta$. Now replacing these into (4), yields:

$$\delta \theta(x) = \frac{\theta_0 \nu h \frac{\partial p}{\partial x} \int_0^h \theta u dz + \eta h \int_0^h \theta u^2 dz}{k - \nu h \frac{\partial p}{\partial x} \int_0^h \theta u dz} $$

(5)
Use of Peclet number shows that (5) is valid for thin elastohydrodynamic films [7]. However, in parts of the piston cycle, for example, at mid-span with high speed of entraining motion some heat is taken away through convection. Following the same approach and accounting for convection:

\[
\delta \theta(x) = \frac{\theta_0 \varphi \frac{\partial p}{\partial x} \int_0^h u_0 dz + \eta \int_0^h u_0^2 dz}{\rho c_p \int_0^h u_0 dz + \frac{1}{h} \varphi \frac{\partial p}{\partial x} \int_0^h u_0 dz}
\]  

(6)

Reynolds equation with no side leakage in circumferential direction of the ring is:

\[
\frac{\partial}{\partial x} \left[ \frac{\rho h^3}{\eta} \left( \frac{\partial p}{\partial x} \right) \right] + \frac{\partial}{\partial y} \left[ \frac{\rho h^3}{\eta} \left( \frac{\partial p}{\partial y} \right) \right] = 6 \mu \frac{\partial}{\partial x} (\rho h) + 12 \frac{\partial (\rho h)}{\partial t}
\]  

(7)

The conjunction geometry is due to a nominal clearance, \( h_0 \), the ring geometry, \( s(x) \) global deformation \textit{in situ} (Mishra et al [9]), \( \Delta(y) \) and any localised deflection due to generated pressures \( \delta(x,y) \) given by the general elasticity potential equation (9):

\[
h(x,y) = h_0 + s(x) + \Delta(y) + \delta(x,y)
\]  

(8)

\[
\delta(x,y) = \frac{2}{\pi E'} \int_0^{2\pi} \int_0^{\beta} \frac{dx_1dy_1}{\left[ (x-x_1)^2 + (y-y_1)^2 \right]^{1/2}}
\]  

(9)

The effective viscosity [10], \( \eta_e \), and density [11,12], \( \rho \), of the lubricant are:

\[
\eta_e = \eta_0 \exp \left[ \ln(\eta_0) + 9.67 \left( \frac{\theta - 138}{\theta_0 - 138} \right)^{\frac{1}{1.98 \times 10^8 \left( \ln(\eta_0) + 9.67 \right)^{1.7 \times 10^{-9} (\rho - \rho_{atm})}} \right]
\]  

(10)

\[
\rho = \rho_{atm} \exp \left[ \frac{\zeta (\theta - \theta_0)}{1 + 0.6 \times 10^{-9} (\rho - \rho_{atm}) \left( \rho - \rho_{atm} \right)} \right]
\]  

(11)

2- Results and Discussion:

Typical data for the engine used in this study is listed in Table 1. Figure 1 shows the predicted temperature rise during the 4 stroke cycle of the engine at different speeds. Convective cooling has a significant effect in preventing the temperature rise inside the contact. This is particularly true with thicker films away from the dead centres. In the vicinity of reversal points, the temperature rise is smaller due to a lower sliding speed, thus reduced viscous friction. In these localities both conduction and convection contribute. At the dead centres, themselves, cessation of motion constitutes no viscous friction, only small friction due to compressive heating. Note that the current analysis disregards heating due to any boundary friction. It is assumed that asperity interactions contribute to surface temperature rise in a localised manner. To note the effect of heat generation due to boundary interactions a mixed TEHD is required. In addition, it can be observed that the effect of convective cooling is much larger at higher engine speeds because of a thicker film.
Table 1. Data used in the analyses

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine stroke</td>
<td>79.5</td>
<td>mm</td>
</tr>
<tr>
<td>Engine speed</td>
<td>2000 and 4000</td>
<td>rpm</td>
</tr>
<tr>
<td>Bore nominal radius</td>
<td>44.50</td>
<td>mm</td>
</tr>
<tr>
<td>Ring axial height</td>
<td>1.2</td>
<td>mm</td>
</tr>
<tr>
<td>Liner surface roughness (RMS)</td>
<td>0.26</td>
<td>µm</td>
</tr>
<tr>
<td>Ring profile</td>
<td>Parabolic</td>
<td></td>
</tr>
<tr>
<td>Lubricant inlet temperature</td>
<td>313</td>
<td>K</td>
</tr>
<tr>
<td>Lubricant density</td>
<td>849.7@15[°C], 833.8@40[°C]</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Lubricant kinematic viscosity</td>
<td>59.99@40[°C], 9.590@100[°C]</td>
<td>cSt</td>
</tr>
</tbody>
</table>

Figure 1: Average rise in contact temperature

Peclet number, obtained as $Pe = \left( \frac{\rho c_p}{k} \right) \left( \frac{h}{B} \right) \int_0^h u dz$ is the ratio of convection to conduction. This is shown in figure 2 for cases studied in figure 1. It is clear that with thicker films at mid-span and in engine strokes other than the power stroke, convection cooling dominates. Only at reversals or in their immediate vicinity does conduction play a role, with the Peclet number being less than unity. It is noteworthy that the current analysis assumes a fully flooded inlet. In practice, this may not be the case as much thinner lubricant films may be formed, which would result in the diminution of the effect of convection cooling. Because of this reason, one function of the compression ring is to carry the heat away through conduction. The current analysis also assumes the inlet temperature to be the same as the bulk oil temperature of the lubricant. In practice the inlet temperature is as the result of thermal balance at the nib of the contact due to bulk oil temperature and those of the bounding solids. This makes the oil temperature higher than that assumed here. Figures 3 and 4 show the variations of minimum film thickness, as well as friction on the liner surface for the entire engine cycle. As can be seen in figure 3, film thickness reduces when conduction is considered as the main cooling mechanism. This is due to the higher temperature of the lubricant within the contact. However, when
the convection cooling effect is considered alongside with conduction, the minimum film thickness becomes close to the case of an isothermal condition. The difference between the friction produced in the case of isothermal flow and the one with convective cooling is significant in the power stroke. This can be seen in figure 4 for even a slight rise in the temperature.

<table>
<thead>
<tr>
<th>Compression</th>
<th>Power</th>
<th>Exhaust</th>
<th>Intake</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conduction &amp; Convection (2000rpm)</td>
<td>Conduction &amp; Convection (4000rpm)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 2: Variations of the average Peclet number

<table>
<thead>
<tr>
<th>Compression</th>
<th>Power</th>
<th>Exhaust</th>
<th>Intake</th>
</tr>
</thead>
</table>

Figure 3: Variation of minimum film thickness with crank angle
Conclusion
It is found that unlike thin elastohydrodynamic films in concentrated counterformal contacts under largely thermo-hydrodynamic condition in ring-bore conjunction, convection cooling through lubricant entrainment plays an important role. This reduces the temperature rise, thus higher effective viscosity would enhance load carrying capacity.

Acknowledgements:
The authors would like to express their gratitude to the Engineering and Physical Sciences Research Council (EPSRC) for the funding of the Encyclopaedic Program Grant, under whose auspices this work is carried out. Thanks are also due to the industrial partners of this research, particularly Aston Martin.

Nomenclature:
- \( B \): ring axial height
- \( c_p \): heat capacity at constant pressure
- \( h \): film thickness
- \( k \): thermal conductivity
- \( l \): length from the bearing leading edge
- \( p \): pressure
- \( u,v \): \( x,y \) velocity distribution components
- \( x,y,z \): Cartesian coordinates
- \( \alpha, \beta \): coefficients used in Houpert's equation
- \( \eta_e \): effective dynamic viscosity
- \( \theta \): temperature
- \( \xi \): density-temperature factor
- \( \rho \): density
- \( \nu \): thermal expansion coefficient

Abbreviations
- DP: Detonation Point
- Pe: Peclet number
- TDC: Top Dead Centre

Subscripts
- \( 0 \): inlet
- \( atm \): atmospheric condition
- \( s \): surface of ring
References:


Keywords: transient mixed thermo-elastohydrodynamics, piston ring