TRIBOLOGY OF PISTON SKIRT CONJUNCTION

B. Littlefair[1], S. Howell-Smith[2], S. Theodossiades[1], P.D. King[1], H. Rahnejat[1]

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1 INTRODCTION

Frictional losses in the piston skirt to cylinder liner conjunction account for approximately 2.5% of the energy supplied to the modern car [1]. These losses are contributed by viscous shear of the lubricant and asperity interactions on the contiguous surfaces. However, for most of the piston cycle the regime of lubrication is dominated by elastohydrodynamics or hydrodynamics. Hence, friction due to viscous shear is dominant.

Most idealistic analyses employ a "cold" piston skirt shape and use either a measured profile or by approximated polynomials as the input shape [2-4]. In reality, however, pistons are subject, not only to contact forces, but also thermo-mechanical distortion. These are as the result of thermal expansion of the piston as well as its global mechanical deformation *in situ*. They alter the piston-liner conjunctional gap. The piston structure is designed in such a way as to prevent gross localised wear in service by means of skirt profile and structural stiffness modification [5]. Considering the combined effect of global as well as local deformation of the skirt under the influence of contact force, it is vital to take into account the effect of shape and rigidity of both the piston and liner structures in an integrated thermoelastic and elastohydrodynamic analysis. This approach is more representative of the *in situ* "hot" skirt condition as noted by McClure [6].

This paper shows the significant differences observed in the generated pressures, film thickness and friction by comparing "cold" piston profiles; disregarding large scale global deformation and "hot" thermo-elastically deformed skirt conjunctions with representative skirt stiffness.

2 SIMULATION CONDITIONS AND THEORETICAL FORMULATION

A piston without a gudgeon pin offset and a centre of gravity coincident with the axial piston centroid is used here in order to limit the dynamic instability. Specifically, an aluminium alloy high performance engine piston with a partial circumferential skirt is employed, as described by Balakrishnan *et al* [3]. The skirt has a complex shape, and regarded as *optimised* in order to reduce friction. The skirt profile is furnished with relief radii at its top and bottom edges in order to encourage lubricant entrainment into its contact with the cylinder liner through a hydrodynamic wedge. These relief radii also reduce edge stress discontinuity, which would otherwise result in abrupt end profiles in all tribological conjunctions. The reduced radii towards the top of the skirt also allows for the weighted thermal expansion caused by the large thermal gradient there. Clearly to compute the resulting piston conformability and the gap in the conjunction, all these factors should be considered. To calculate the conjunctional pressures the following steps are undertaken:

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[1] *Wolfson school of Mechanical & Manufacturing Engineeering, Loughborough University, Loughborough, UK, mmbrl@lboro.ac.uk*

[2] *Capricorn Automotive, Basingstoke, Hampshire, UK,*

1. Calculation of correct liner and piston geometry due to thermal expansion (thermal analysis).
2. Calculation of contact load between the piston skirt and cylinder bore (for quasi static load convergence)
3. Calculation of piston and liner structural deformation based on the thermally deformed geometry (from part 1) and material properties, resulting in an initial conjunctional gap.
4. Calculation of lubricant reaction based on the piston sliding velocity, lubricant rheology and the representative initial geometry.

Using Finite Element (FE) method, the "hot" piston skirt profile due to thermal distortion is calculated from the predicted temperature distribution reported by Bosch [7]. The cylinder liner is also subjected to a thermal gradient. The liner temperature gradient has been characterised by Capricorn Automotive by a series of positional temperatures and has been used as input into an FE model to derive the liner thermal distortion.

The distortion caused by the mechanical gas pressure loading is applied, post thermal distortion. The condition of maximum combustion pressure was applied to the piston crown with the inertial force, derived from the primary acceleration of the piston, whilst constraining the piston small end. The resulting geometry is therefore due to the combined effects of system forces, as well as thermal loading and the differential thermal effects.

The full formulation of the lubricated film thickness requires the inclusion of the above thermo-mechanical distortions and the simultaneous solution with Reynolds equation.

The general form of Reynolds equation is given as:

$$\frac{\partial}{\partial x}\left[\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial y}\left[\frac{\rho h^3}{\eta}\frac{\partial p}{\partial y}\right]=12\left\{u_{av}\frac{\partial}{\partial x}(\rho h)+v_{av}\frac{\partial}{\partial y}(\rho h)+\frac{d}{dt}(\rho h)\right\} \tag{1}$$

The ultimate term on the RHS of the equation is the squeeze film action. In the quasi static analysis performed here this term is neglected. It can play a significant load at piston reversals (TDC and BDC). At maximum combustion load the velocity of the piston is 9.14 m/s and therefore the load carrying capacity is dominated by lubricant entrainment– the Couette flow terms on the RHS. The penultimate term due to side leakage is also neglected. The speed of entraining motion is obtained as:

$$u_{av}=\frac{1}{2}(u_b+u_p)=\frac{1}{2}\frac{dx}{dt}=\frac{\dot{x}}{2} \tag{2}$$

The Reynolds solver only considers the lubrication of the skirt, in this case represented as a rectangular footprint area discretised in the x and y directions (x is the direction of sliding). To consider elastohydrodynamic lubrication it is necessary to recompute the skirt deflection at each of the skirt nodal positions. This is achieved by taking into consideration the effective skirt stiffness.

The *in situ* skirt profile as previously described is incorporated as the reference initial shape prior to any localised deflection due to generated conjunctional pressures. To find this reference shape, the body was meshed with a suitably scaled tetrahedral mesh, utilising quadratic tetrahedral elements. These were forced over a pattern matching the Reynolds solver discretisation, as the *in situ* profile at all conjunctional locations are required for the elastic film shape, *h* in Reynolds equation:

$$h = lin - s + \sigma + c \tag{3}$$

Lin is the liner profile, *c* is an initial nominal gap and *s* is the thermo-elastically deformed profile obtained through FE analysis. Each of physical nodes is subjected to an equal unit force and the resulting deflection is determined. Using the recorded deflection and applied force an approximate nodal stiffness is calculated using f=kσ, where f is the equal unit force applied, σ deflection recorded and k is the localised nodal stiffness. Therefore, using each nodal stiffness in isolation with the summated localised pressure from Reynolds, the deflection at each skirt node can be approximated.

Lubricant density and viscosity variations with pressure play an important role in all lubricated contacts. For the solution provided here (isothermal conditions) the input viscosity at ambient pressure is adjusted for the working temperature of the cylinder. The viscosity-pressure dependence is taken from Roelands [8]:

$$\bar{\eta} = \frac{\eta}{\eta_o} = \exp(\ln \eta_o + 9.67)(-1 + (1 + 5.1\text{x}10^{-9}\, p)) \tag{4}$$

The density variation with pressure is due to Dowson and Higginson [9]:

$$\rho = \rho_0 \left(1 + \frac{0.6 \times 10^{-9}\, p}{1 + 1.7 \times 10^{-9}\, p} \right) \tag{5}$$

There are two convergence criteria to be satisfied before the solution is obtained. The first one is pressure convergence within the low relaxation Newton-Raphson scheme with Gauss-Seidel iterations. The following criterion is employed:

$$\frac{\sum_{i=1}^{nx} \sum_{j=1}^{ny} \left| p_{(i,j)}^{n} - p_{(i,j)}^{n-1} \right|}{\sum_{i=1}^{nx} \sum_{j=1}^{ny} p_{(i,j)}^{n}} \le \varepsilon_{rp} \tag{6}$$

The error tolerance ε_{rp} is set to 10^{-4}. If the convergence criterion is not achieved, then each of the nodal pressures is updated using under-relaxation as:

$$p_{(i,j)}^{n} = p_{(i,j)}^{n-1} + \vartheta_p \Delta p_{(i,j)}^{i} \tag{7}$$

The next convergence criterion is the achievement of contact load, *F* by the integrated pressure distribution:

$$W = \iint p(x,y)\,dxdy\,. \tag{8}$$

Where the load convergence is calculated as;

$$\frac{|W - F|}{F} \le \varepsilon_L \tag{9}$$

Where ε_L is 5.10^{-3}. If this criterion is not met, then the nominal gap size c is adjusted and the entire computational procedure repeated. The gap adjustment is as follows:

$$c = c_{n-1}\left(\left(\frac{F}{W}\right)\Omega\right) \tag{10}$$

where Ω is a damping or gap relaxation factor, typically in the order of 10^{-5}.

Figure 1 shows the resultant skirt pressure distribution, also superimposed in an isobaric form on the piston skirt in the inset to the figure.

Figure 2 shows the corresponding oil film thickness contours.

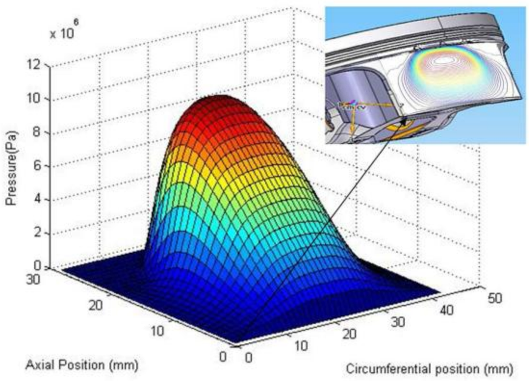

Figure 1: Conjunctional pressure distribution

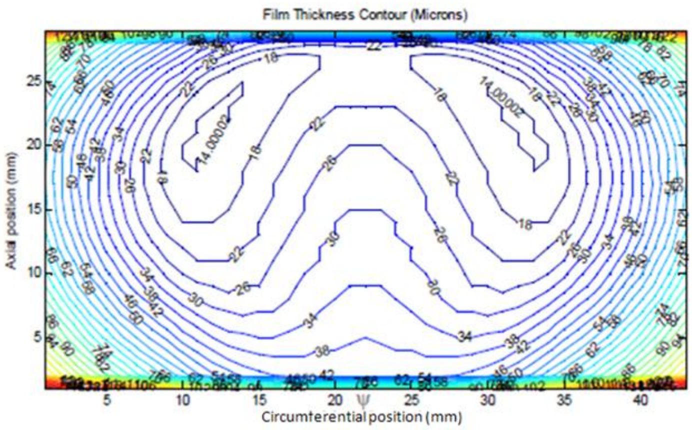

Figure 2: Oil film thickness contours corresponding to pressure distribution in fig. 1

A comparison between the results obtained here for a “hot” skirt to those by Balakrishnan *et al* [3] for the same piston, but with assume “cold” skirt shows a larger and wider contact area with reduced pressures. The “cold” skirt profile with no global thermo-elastic distortion is less conforming. Figure 3 is the isobaric pressure distribution for the “cold” skirt analysis by Balakrishnan *et al* [3].

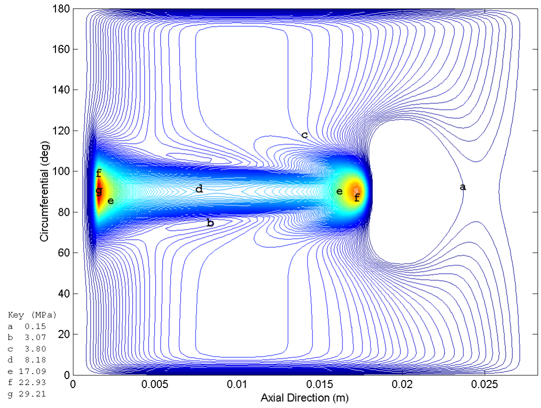

Figure 3: Isobaric pressure distribution for a “cold” skirt (after Balakrishnan *et al* [3])

3 CONCLUSION

It is clear that thermo-elastic deformation of piston whilst *in situ* in a fired engine alters the piston skirt profile. This depends on the structure of the skirt and its localised stiffening. The global deformation alters the gap shape, which in turn changes the contact footprint, generally making it more conforming. The resulting lager area of the contact compared with the idealised “cold” skirt analysis yields lower pressures as shown by comparison of figures 1 and 3 in the current paper. The analysis shows that relief radii play an important role in the entrainment of the lubricant into the contact, but for a representative “hot” skirt analysis remain outside the region of significant pressure, hence the edge stress discontinuities noted with “cold” skirt analyses do not actually occur in most cases. The maximum pressure calculated in the current analysis is 10MPa, whilst that for the same conditions, not taking into account the global thermo-elastic distortion by Balakrishnan *et al* [3] is 29 MPa due to abrupt change prior to the relief radius at the leading edge of contact. Another important point to note is that the generated pressures are actually quite low and insufficient to cause any significant piezo-viscous action of lubricant. Thus, the prevailing condition is actually hydrodynamic, with all deformation being of a global, rather than localised nature.

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