ELASTOHYDRODYNAMIC ANALYSIS OF A CONNECTING ROD BEARING FOR HIGH PERFORMANCE ENGINES

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SUMMARY
This paper describes an elastohydrodynamic analysis of the connecting rod big end bearing of a high performance Fiat internal combustion engine. Some of the problems arising in bearing lubrication are investigated in this work, which presents the first part of a more comprehensive effort. The bearing housing, loaded by considerable dynamic force, is regarded as a flexible structure. An enhanced mathematical model, based on a well established computational methodology able to solve the related mathematical problem, is briefly described and some of the results obtained are discussed, focusing on the effects of particular modelling issues such as bolt tightening.

Keywords: elastohydrodynamic, lubrication, bearing, connecting-rod, finite-element

1 INTRODUCTION
The importance of considering connecting rod flexibility in the elastohydrodynamic (EHD) analysis of big end bearings for production-level engines has been widely discussed and demonstrated by several authors [1-4]. This aspect becomes much more significant when studying the lubrication of a high performance engine, where the connecting rod structural elastic behaviour is enhanced by a relatively low stiffness. The continuous progress achieved in mechanical technology and in machine design calculations have allowed a progressive structural downsizing and lightening of the rototranslating members, and particularly of the connecting rods, which became more and more compliant.

The maximum rotation speed of a high performance engine is always considerable (over 15000 r.p.m.) and therefore the inertia forces acting on the journal may become much more important than the peak firing loads due to the combustion process. All of these factors, together with the high values of specific pressure that are commonly reached in these engine types, cause bearing deformations much larger than in conventional engines, making lubrication of connecting rod big end bearings very difficult.

The rotation speed considered in this study (8500 r.p.m.) is, however, relatively low and avoids some of the structural problems caused by very high bearing loads, like sliding or detachments at the interface between the cap and the rod.

2 MATHEMATICAL MODEL
The EHD mathematical problem has been solved with a computational algorithm implemented in a computer code build from a C++ source program. This algorithm is well documented in the literature and allows the solution of non-linear complementarity problems (see [3] for details). Thanks to its robustness it is particularly suited, unlike other algorithms based on direct iterative methods, to solve EHD problems characterised by heavy loads and very small film thickness.

2.1 Basic equations
Assuming linear structural behaviour of the bearing model, the bearing housing radial displacement he is calculated superimposing the deformation ho (initial displacement) due to bolt preload and the elastic deformation h1 induced by fluid film pressure p, obtained through a linear compliance operator L.

\[ h_e = h_0 + h_1 = h_0 + L_p \] (1)

The total lubricant film thickness h is then evaluated by adding to h_e the film thickness contribution due to the journal centre radial displacement relative to the undeformed bearing configuration centre, expressed as a function of the eccentricity components

\[ h = c - e_x \cos \theta - e_y \sin \theta + h_e \] (2)

The journal is submitted to periodic loads changing in direction and magnitude during the engine thermodynamic cycle. The finite element method and a fast Newton-Raphson scheme [4] are used to solve a non-linear integro-differential system, obtained by assembling the elasticity equations with Reynolds (3) and equilibrium (4) equations, to determine the evolution of the lubricant film: p = p(\theta, z) and h = h(\theta, z).

\[ \nabla \cdot h^3 e^{-\omega} \nabla p - 6 \mu_0 \left( \frac{\partial h}{\partial \theta} + 2 \frac{\partial h}{\partial t} \right) = 0 \] (3)

\[ \begin{cases} \int_{\Omega} p \cos \theta \partial \zeta dz - F_x = 0 \\ \int_{\Omega} p \sin \theta \partial \zeta dz - F_y = 0 \end{cases} \] (4)

A further non-linearity, due to the free boundary hydrodynamic conditions, is approached with Murty’s iterative cavitation algorithm [5].
2.2 Structural model

A detailed 3-D finite element model of the connecting rod big end bearing of a high performance Fiat internal combustion automotive engine has been meshed from a CAD model, neglecting the structural discontinuity at the junction plane between the cap and the rod. The bolt preload effect was taken into account by adding into the blind holes of the model, corresponding to the threaded seats of the bolts used to join the cap and the rod, some simple 3-D beam elements approximately simulating the bolt head and shank. If a suitable initial strain is applied to the beam elements of the bolt shank, a compressive stress will result in the solid elements all around the holes. The assumption of a continuous model is thus justified only when the compressive stress at the junction plane is much greater than the tensile stress induced by the dynamic inertia loads (which are particularly high at the beginning of the induction phase).

Two different element types have been used to mesh the bearing structural model: 8-node and 20-node isoparametric solid elements (the second mesh is shown in Figure 1). The bearing longitudinal midplane (x-y plane) is a symmetry plane. The x axis is the connecting rod axis. Full constraint boundary conditions (fixed nodes) are assumed at the model upper end. Over this limiting surface the connecting rod flexibility does not influence, except by rigid body motion, the bearing radial compliance.

![Figure 1: Big end bearing FEM structural model](image)

A regular brick element mesh was mapped all around the bearing housing cylindrical surface, to permit the simplest coupling between the solid 3-D structural element mesh with a 2-D fluid element mesh and to obtain accurate displacement results.

The finite element modelling of the lubricant fluid film turns the continuous pressure \( p \) and film thickness \( h \) into \( n \) discrete nodal unknowns \( p_i \) and \( h_i \). The compliance linear operator \( L \) is then converted into a compliance matrix \( L_{ij} \), linking a unit nodal pressure at node \( j \) to a radial displacement at node \( i \).

2.3 Linking structural and hydrodynamic models

Two regular rectangular fluid element meshes have been used to solve the Reynolds equation: a 100 x 6 first order element (4-node) mesh has been coupled to the 8–node structural model mesh, a 60 x 4 second order element (8-node) mesh has been coupled to the 20–node structural model mesh. In both cases two different hypotheses have been alternatively assumed, as explained below, in calculating the \( L_{ij} \) matrix. The linking between nodal pressures and nodal radial displacements (i.e. nodal variations of the film thickness due to the bearing flexibility) may be conveniently expressed by introducing a flexibility matrix \( H_{ij} \), relating nodal forces \( f_j \) to nodal displacements \( h_i \). Therefore, if \( A_{ij} \) is a proper area matrix, the \( h_i \) contribution is

\[
   h_i = H_{ij} f_j = H_{ij} A_{ik} p_k = L_{ik} p_k
\]

Resorting to the virtual work principle, the area matrix \( A_{ij} \) may be determined by imposing the equivalence between the virtual energy due to nodal forces and the virtual energy due to nodal pressures, obtaining a banded (non-diagonal) area matrix. As an alternative a more straightforward and physically intuitive approach may be used when the regular fluid element mesh is made of first order 4-node rectangular elements. If the nodal pressures are converted into nodal forces through a simple product by the respective nodal influence areas, a lumped (diagonal) \( A_{ij} \) matrix is easily calculated. The \( i \)-th diagonal term of this lumped matrix is then the sum of all the terms in row \( i \) of the banded matrix. This second way of calculating \( L_{ik} \) is expected to balance the higher stiffness of the structural first order 8-node elements.

The \( H_{ij} \) flexibility matrix is easily calculated by loading all of the nodes on the bearing housing cylindrical surface \( S_b \) with unit nodal forces in the radial direction and retrieving the radial component of the calculated nodal displacement. The initial displacement vector \( h_0 \) is simply evaluated from the deformed shape of the surface \( S_b \) produced by the bolt preload.

Both displacements are influenced by the distance between the bearing centre and the model upper end, where full constraint boundary condition (fixed nodes) are assumed. The \( S_b \) mean displacement should then be deducted from all displacements to exclude rigid body motion components from the above calculations.

3 DATA

The basic engine, bearing and lubricant data are all shown in Table 1. In the same table the force caused by the tightening torque applied to the bolts is included.

The load history (combustion process loads and inertia loads), is periodic and is shown in Figure 2, on a 720 degree crank angle basis, for a rotation speed of 8500 r.p.m.. It has been calculated, as a function of the crank angle, by summing the inertia forces (obtained via a lumped-mass procedure), to the gas pressure forces, evaluated by superimposing a suitable combustion function to a politropic compression-expansion law. In
the polar load diagram of Figure 2, 0 deg crank angle corresponds to the beginning of induction phase.

<table>
<thead>
<tr>
<th>DATUM TYPE</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crank length [m]</td>
<td>0.0395</td>
</tr>
<tr>
<td>Rod length [m]</td>
<td>0.137</td>
</tr>
<tr>
<td>Engine speed [r.p.m.]</td>
<td>8500</td>
</tr>
<tr>
<td>Bearing width [m]</td>
<td>0.02085</td>
</tr>
<tr>
<td>Bearing diameter [m]</td>
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</tr>
<tr>
<td>Radial clearance [m]</td>
<td>0.00004</td>
</tr>
<tr>
<td>Bolt diameter [m]</td>
<td>0.01</td>
</tr>
<tr>
<td>Bolt tightening torque [N m]</td>
<td>46</td>
</tr>
<tr>
<td>Bolt equivalent preload [N]</td>
<td>44800</td>
</tr>
<tr>
<td>Young modulus [GPa]</td>
<td>110</td>
</tr>
<tr>
<td>Poisson ratio</td>
<td>0.329</td>
</tr>
<tr>
<td>Lubricant dynamic viscosity [Pa s]</td>
<td>0.015</td>
</tr>
<tr>
<td>Piezo-viscous coefficient [Pa⁻¹]</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 1: Engine, bearing and lubricant data

4 RESULTS

Because of symmetry, only one half of the model shown in Figure 1 was analysed. The mesh subdivision in the circumferential direction \( \theta \) was chosen with great care, verifying that for a rigid bearing (zero compliance and zero initial displacement) asymptotic results are obtained (relative variation lower than 0.3%) for a mesh subdivision with more than 100 nodes along \( \theta \). The mesh subdivision in the axial direction is far less influential and was fixed to 7 nodes. For the same total number of nodal points in the circumferential direction, a mesh of first order structural and fluid elements was found to result, with respect to a second order element mesh, in a lower rate of convergence. In addition, the maximum value of the peak pressure in the cycle plotted vs. the number of nodes along \( \theta \), shows a curve asymptotically converging with many oscillations.

Nevertheless a 100 x 6 first order element mesh was used to calculate useful results for comparison purposes. They were obtained assuming both a lumped and a banded area matrix. The lumped area matrix was found not to allow convergence when applied to second order elements. These results will then be compared to the ones obtained for a 60 x 4 second order element mesh with a banded area matrix. The total number of nodes on \( S_b \) is a bit greater in the second order mesh (849 nodes with respect to 707 nodes). A coarser 20-node element mesh could not be generated, because of topological problems encountered by the CAD mesh generator.

All of the results shown in Figures 3-9 have been obtained for a second order element mesh. A comparison useful to appreciate the differences induced in the results by the two mesh types (first and second order) is shown in Figures 10 and 11. In Figure 3 and Figure 4 the journal centre orbits are presented. The dash-dot circles are reference orbits of constant relative eccentricity (1 and 2).
In both Figures 3 and 4, a journal centre orbit calculated neglecting the bolt preload is added for comparison.

Although this solution is not realistic (at the junction plane a strong tensile stress may occur) the comparison is helpful to distinguish in Figure 3 the influence of bearing flexibility (the difference between rigid solution and elastic bearing solution without bolt preload) from the influence of bolt preload (the difference between elastic bearing solution without bolt preload and elastic bearing solution with bolt preload), as plotted in Figure 4, where a significant reduction of eccentricity due to bolt preload is shown.

The minimum film thickness and the peak film pressure are plotted respectively in Figures 5 and 6, where three curves are superimposed: a) rigid bearing solution; b) flexible bearing solution without bolt preload; c) flexible bearing solution with bolt preload.

The bearing flexibility reduces the absolute minimum film thickness (from 2.37 µm to 0.76 µm) and shifts the crank angle corresponding to this minimum, from 634 degs (exhaust phase) to 2 degs (induction phase). This shift is due to the different structural support stiffness provided by the bearing, quite low at 2 degs (where the load is directed against the cap, relatively thin and flexible), much more considerable at 634 degs (where a significant component of the load is directed against the connecting rod shank, which has high axial stiffness).

The flexible bearing solution with bolt preload (curve c) shows an absolute minimum of 1.75 µm at 321 degs crank angle, where a slight minimum film thickness reduction may be observed with respect to curve b (influence of bolt preload).

Also in Figure 6 the influence of bearing flexibility appears to be significant at a crank angle of approximately 2 degs, where a strong peak pressure reduction, due to a widening of the active film region, induced by the considerable deformation of the cap, is observed. This pressure reduction is still consistent with a minimum film thickness reduction, as already stated in [6], since a bimodal pressure field and an oil film pocket develop.

Two main effects are put in evidence by a simultaneous analysis of Figures 5 and 6.

A bolt preload induced stiffening effect appears near 2 degs crank angle, where the peak pressure curve c approaches curve a and the minimum film thickness curve c strays from curve a less than curve b does.

A second bolt preload induced effect, referred to as a shape effect, is observed at 321 degs crank angle, where the minimum film thickness curve c reaches its absolute minimum.

Near this crank angle the peak pressure values of curve c are much larger than the values of both curves a and b, and the minimum film thickness of curve c is a bit lower than curve b. A similar effect may be observed near 667 degs, where the journal centre position and the load vector are almost the same as at 321 degs.

The shape effect may be explained by observing Figure 7, showing the initial displacement $h_0$ spatial distribution, which presents two waves. Near 321 degs and 667 degs crank angle this waviness modifies the film thickness in the active film region (see Figure 8 between $\theta=250^\circ$ and $\theta=300^\circ$ and compare with Figure 7).
One of these waves produces the above described pressure peaks of curve \( c \), as shown in the 3-D plot of Figure 9, and reduces the carrying capacity of the bearing.

In Figure 10 three curves of the peak film pressure history (curve type \( c \)) are compared:

- first order mesh with banded area matrix;
- first order mesh with lumped area matrix;
- second order mesh with banded area matrix.

Some differences in peak pressure values at 321 and 667 degress may be noticed. At these crank angles, due to the linear interpolation of first order elements, the quick circumferential variation in the pressure around \( \vartheta = 300^\circ \) causes higher values of the peak pressure. This behaviour is less evident using the lumped area matrix.

In any case the consequences on film thickness evaluation are not very significant, as Figure 11 shows.

The assumption of a continuous structure at the junction plane was checked and was found to be correct since normal stresses at the interface are always and everywhere compressive stresses.

5 CONCLUSIONS

The EHD analysis of a big end bearing for a high performance engine gives no realistic results without modelling the tightening of the bolts joining the cap and the body of the rod.

On minimum film thickness the bolt preload may produce opposite effects at different crank angles:

- the stiffening effect increases the minimum film thickness in a highly compliant carrying region like the cap, when the dynamic load is directed against it;
- the shape effect decreases the minimum film thickness when the initial displacement contribution to the film thickness field causes a high circumferential film thickness variation rate in a part of the carrying region.

Some authors [7, 8] have demonstrated that the bearing elastic deformations due to the inertia forces acting on
the connecting rod mass influence the EHD behaviour of production-level engine bearing. This should be more likely in the case studied here, and therefore this model will be enhanced in the near future.

A further modelling aspect neglected here is the structural non-linearity which may arise from slides or detachments between the contact surfaces of the rod and the cap. A more sophisticated non-linear structural model should then be used to analyse the bearing behaviour at journal rotation speeds higher than 8500 r.p.m..

6 ACKNOWLEDGEMENTS

The authors wish to acknowledge gratefully the MURST 1999 financial support and thank Prof. A. Garro, Prof. A. Manzi and Ing. G. Righes for the precious suggestions and advice arising from their long experience in lubrication and engine design.

7 REFERENCES