THERMOELASTOHYDRODYNAMIC ANALYSIS OF DYNAMICALLY LOADED JOURNAL BEARINGS

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Summary. Thermo-elastic effects are important in the evaluation of oil film thickness, maximum pressure and temperature of oil film and they determine reliable operation of single bearing and the bearing system. For the correct operation of bearing, the information regarding thermo-elastic deflections should be available at the early design stage of the bearing.

Manufacturers of engines are trying to receive more efficient, powerful, and operating at increased speed, durable engines with minimum power loss what causes the necessity to consider more design factors of journal bearings as, e.g. mixed friction operation during start up and slowing down of engine, thermoelastohydrodynamic (TEHD) operation of journal bearing, an effect of surface roughness and the engine housing deflections on the bearing performances. The design of reliable, dynamically loaded journal bearing can be achieved by applying modern methods of theoretical and experimental investigation. It consists in determination of the static characteristics of bearing for the TEHD lubrication including the oil film pressure and temperature distribution, the journal centre trajectory, the transient TEHD conditions of bearing operation and the dynamic characteristics.

The paper introduces the possibility of TEHD calculation of dynamically loaded journal bearings on the basis of developed programs of numerical simulation of bearing operation.

1. Introduction

The engine’s tribosystem comprises the dynamically loaded journal bearings of crankshaft, connecting rod and piston pin. These bearings are running out of performance margins under increasingly more severe operating conditions (higher speeds, loads), which enhances the danger of seizure or fatigue. Dynamically loaded journal bearings of internal combustion engines should be designed like very reliable and durable bearings operating for very long time. Manufacturers of engines are trying to receive more efficient, powerful, and operating at increased speed, durable engines with minimum power loss what causes the necessity of considering more design factors of journal bearings, e.g. mixed friction operation during start up and slowing down of engine, elastohydrodynamic (EHD) or thermoelastohydrodynamic (TEHD) operation of journal bearing, effect of surface roughness and the engine housing deflections on the bearing performances [1-9]

The extension of classic theory of hydrodynamic lubrication describes the problem of thermoelastohydrodynamic (TEHD) by the backward reaction of bearing deflection as result of oil film pressure and temperature, on the lubricating gap geometry and the oil film pressure and temperature. This procedure considers the problem of TEHD quasistatic, i.e. the bearing deflection follows the external loads. More exact solution and varied operation conditions (an increase in the number of crankshaft revolutions) need to include the dynamic effects when the internal forces introduce additional deformations caused by local inertia actions. It leads to the fully dynamic thermoelastohydrodynamic analysis. In this analysis, the static
characteristics [1-3] of bearing for the TEHD lubrication of bearing including the oil film pressure and temperature distribution, the journal centre trajectory [3], the transient TEHD conditions of bearing operation [4, 5] and the dynamic characteristics [5] should be determined.

The paper introduces some problems of TEHD of dynamically loaded journal bearings with special attention kept on the calculation of static characteristics that are based on developed programs of numerical simulation of bearing operation.

2. Static characteristics

The static characteristics of journal bearing can be received for stationary or transient conditions including adiabatic laminar or adiabatic turbulent oil film [6, 8] as well as for TEHD lubrication by coupled solution of Reynolds, energy, viscosity, geometry and elasticity equations. These characteristics allow to precise design of dynamically loaded journal bearings. Below, the Reynolds equation allowing the calculation of oil film pressure distribution, the calculation of the journal centre trajectory and remarks on EHD and TEHD lubrication are described [10-12].

2.1. The journal centre trajectory

The solution of Reynolds equation (1) together with the energy and viscosity [10] equations allows the calculation of bearing in the conditions of nonisothermal oil film.

\[
\frac{\partial}{\partial \phi} \left( \frac{H^3}{\eta} \frac{\partial \bar{p}}{\partial \phi} \right) + \frac{(D/L)^2}{\eta} \frac{\partial}{\partial z} \left( \frac{H^3}{\eta} \frac{\partial \bar{p}}{\partial z} \right) = \frac{\partial H}{\partial \phi} + \frac{12}{\omega} \frac{\partial H}{\partial t}
\]

(1)

where: \( H = \frac{h}{(R-r)} \) - dimensionless oil film thickness, \( h \) - oil film thickness (\( \mu m \)), \( \bar{p} = \frac{p\psi^2}{\eta \omega} \) - dimensionless oil film pressure, \( p \) - oil film pressure (MPa), \( r \) - journal radius (m), \( L \) - bearing length (m), \( D \) - bush diameter (m), \( R \) - bush radius (m), \( \phi \), \( \omega \) - peripheral and axial co-ordinates, \( \psi = \omega t \) - dimensionless time, \( \omega \) - angular velocity, \( \eta \) - dimensionless viscosity.

It was assumed that on the bearing edges the oil film pressure \( p(\phi, z) = 0 \) and in the regions of negative pressure \( p(\phi, z) = 0 \). The oil film pressure distribution determined from Eqn. (3) has been substituted for energy equation [10] to receive the oil film temperature and viscosity distributions. The values of temperature \( T(\phi, z) \) on the bearing edges \( (z = \pm L/2) \) were determined by parabolic approximation [10]. The exponential equation of viscosity has been applied [10].

The method applied to the solution of the Reynolds, energy, viscosity and geometry equations as well as calculation of the journal centre trajectory [12] assumes the superposition of pressures during the calculation of the components of oil film resultant force, the aligned journal and bush axis, nondeformable journal and deformable bearing bush. The method assumes also: the non-isothermal model of oil film, the regions of negative pressures are neglected, finite model of bearing, the oil supply pressure and temperature. The pressure, temperature and viscosity [11, 12] fields computed from eqn. (1), by means of a method applied in [6] allow the determination of the resultant force of bearing. Equating the oil film force \( \bar{W} \) with the applied load \( \bar{F} \) yields at any instant, and at assumed external dynamic load

\[
\bar{F} = \bar{W} \left( \frac{L}{D}, \varepsilon, \alpha, \dot{\varepsilon}, \dot{\alpha} \right).
\]

(2)
where: \( L/D \) - relative length of bearing, \( \varepsilon, \alpha \) - relative eccentricity and attitude angle
\( \dot{\varepsilon}, \dot{\alpha} \) - time derivatives of relative eccentricity and attitude angle.

Eqn (2) determines the eccentricity, attitude angle, i.e. journal centre trajectory (Fig. 1.) as the functions of crankshaft rotation angle [10-12].

2.2. EHD and TEHD solutions

For the correct operation of bearing, the information regarding thermo-elastic deflections should be available at early design stage of the bearing [1,13]. These effects are important in the evaluation of oil film thickness, maximum oil film pressure and temperature, affecting reliable operation of single bearing and the bearing system. Classical hydrodynamic lubrication theory assumes rigid surfaces of the mating walls. Since engine bearing structures are becoming more and more flexible, at higher bearing loads and extreme thermal conditions the bearing performance should be analysed on the assumption that the walls are elastic. An example of bearing deflections in the axial cross-section, caused by separate action of oil film pressure or temperature distributions, is introduced in Fig. 2. [3]. There is a difference in the shape of bush cross-section deflected as result of both types of loads what certifies the need for simultaneous calculations with the oil film pressure and temperature.

![Diagram](image)

Fig. 1. Journal centre trajectory computed for models of oil film [8]

![Diagram](image)

Fig. 2. Deflections of bearing axial cross- two section caused by the oil film pressure and temperature [3].

Ozasa, et al. [14] carried-out an elastohydrodynamic (EHD) analysis of connecting rod big-end bearings theoretically and experimentally under motoring conditions. The measurements of the journal centre orbit agree well with predictions of the EHD model (Fig. 3); the dotted lines show the initial clearance circle of the bearing, and the journal centre orbit exceeds the limit on the cap side, showing the bearing deformation when the engine is in operation. Ushijima, et al. [9] calculated the EHD performance of an engine bearing with both elastic deformation of the bearing housing and the pressure-dependent viscosity increase in lubricant being considered.
Fig. 3. Journal centre orbits, experiment versus EHD lubrication theory (4000 rpm) [14].

Fig. 4 and Fig. 5 show that the maximum values of oil film pressure and the frictional torque during one cycle decrease, compared to the prediction of the conventional rigid film model. Their study on the individual effect of each deformation and the viscosity change on the bearing performance has found that the increase in viscosity leads to an increase in the temperature and in frictional torque, though the elastic deformation causes a decrease in these values.

Fig. 4. Effect of elastic deformation on the peak oil film pressure [9]

An effect of EHD and TEHD conditions on the 2-lobe bearing structure stresses and deflections distribution have been considered in [3]. As results of this investigation the maximum deflection of the bearing structure, caused by action of oil film pressure (EHD conditions) only was 9 μm (Fig. 4). Total deflections generated by the simultaneous action of oil film pressure and temperature are shown in Fig. 5; in this case the maximum deflection
was 46 μm. Due to the symmetry in the oil film pressure and temperature distributions (no misalignment of axis) Fig. 4 and Fig. 6 show the deflections for half of bush.

![Fig. 6. Deflections of the journal bearing structure caused by oil film pressure](image)

![Fig. 7. Deflections of the journal bearing structure caused by oil film pressure and temperature](image)

This investigation confirms that for exact calculation of the bearing design there is the need to consider an effect of both oil film pressure and temperature distributions.

5. Conclusions

The problems of modern dynamically loaded engine journal bearings can be solved by coupled application of theoretical and experimental investigations. The following conclusions can be drawn:

1. The Reynolds equation including the surface roughness allows the calculation of bearing in the fluid or mixed lubrication regimes.
2. Journal bearing centre trajectory determined on both classic and TEHD theory of lubrication guarantee exact values of maximum oil film pressure and temperature as well as minimum value of oil film thickness, i.e. the most important parameters of bearing operation.
3. The developed program of journal centre trajectory calculation coupled with commercial programs of finite element methods allows further investigation with dynamically loaded bearings under TEHD conditions.

For more exact calculation of high speed, dynamically loaded journal bearings the Reynolds equation including the turbulence correlation coefficients should be applied. This problem can be solved by joining the own developed program of dynamically loaded journal bearings [3] to the program of calculation of high speed bearings [6] including the turbulent oil film.
References


