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ANALYSIS OF SLIDE BEARING COMPUTATIONAL MODELS CONSIDERING ELASTIC DEFORMATIONS AND ROUGH SURFACES

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Abstract: *The paper presents computational simulation strategies of the slide bearing lubrication as a fluid-structural problem. Finite difference method is proposed for a solution of the fluid problem described by Reynolds differential equation and Finite Element Method is used for a solution of the structural problem. Bearing loads are calculated by the Virtual Engine model assembled and solved in Multibody System. The proposed approaches include temperature and pressure dependent viscosity and density of bearing lubricant. All the computational approaches are applied on a main bearing of modern in-line three-cylinder engine.*

Keywords: Elastohydrodynamics, Rough surface, Stiffness matrix, Coupled solver, Viscosity.

1. Introduction

In the course of time, the hydrodynamic (HD) theory has been developed by a relative large number of authors. Subsequently, the HD has been enhanced by elastic deformation influences (EHD) and this theory has been extended for many mechanical components, for instance gears etc.

HD theory presumes that a bearing shell and a pin are without any deformations. Therefore, a relative eccentricity can reach a maximal value of 1. Slide bearings of present combustion engines are highly loaded and the relative eccentricity sometimes exceeds value of 1. This is caused by elastic deformations, mainly of the bearing shell nevertheless, in general, also by the pin deformation. These conditions can be found in some modern turbocharged diesel engines.

One important effect takes place when local bearing clearance values drop to extremely low levels, surface asperities on a pin and a bearing shell start interaction with each other and thereby create boundary lubrication conditions. This effect should also be incorporated into the computational model.

2. Theoretical Background

A slide bearing solution can be presumed as the coupled structural-fluid problem and it covers a solution of three basic equations (1, 2, 4). These equations have to be solved simultaneously.

In general, if the modified Navier-Stokes equation and continuity equation are transformed for cylindrical forms of bearing oil gap together with restrictive conditions, e.g. Novotny (2009), the behaviour of oil pressure can be now described by Reynolds differential equation. This frequently used equation is derivated for a bearing oil film gap and can be written in the form:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial z}\right)-\frac{\partial(u\rho h)}{\partial x}-\frac{\partial(\rho h)}{\partial t}=0 \ , \qquad (1)$$

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where p is a pressure, x and y are coordinates, t is time, h is oil film gap, η is dynamic viscosity of oil, ρ is density of oil and u is an effective velocity.

Generally, oil properties and oil viscosity respectively are dependent on the pressure, temperature or shear stress. High viscosity differences can be achieved especially when machinery parts include point or line contact (roller bearings or a cam/tappet contact). Roelands (1966) formula is one of the simplest descriptions of oil viscosity vs. pressure.

The influence of oil temperature on its viscosity is also significant. Therefore, it is necessary to incorporate the temperature dependence in a form of a computational model. In practice, the oil temperature is not constant. However, variable oil temperatures considerably increase a model complexity. Therefore, oil temperature is rated as constant for the model for every temperature cycle.

The oil film thickness including elastic deformations is calculated by introduction of stiffness matrices of a shell or a pin respectively as:

$$\mathbf{h}(x,z) = \mathbf{h}_{rigid} + \mathbf{K}_{shell}^{-1}(\mathbf{p}_{hydro} + \mathbf{p}_{rough}) + \mathbf{K}_{pin}^{-1}(\mathbf{p}_{hydro} + \mathbf{p}_{rough}), \tag{2}$$

where $\mathbf{K}_{shell}$ and $\mathbf{K}_{pin}$ are stiffness matrices of the shell and the pin respectively, $\mathbf{p}_{\mathrm{hydro}}$ is a matrix of hydrodynamics pressures and $\mathbf{p}_{\mathrm{rough}}$ is a matrix of pressures due to a contact of surface roughness. FE model of the pin is relative simply one (cylinder under boundary conditions) and it is generated by a user written macro. FE model used for a generation of the shell stiffness matrix is based on 3D CAD model of an engine block. Ideally, the oil film gap h_{rigid}, considering a rigid pin and shell, is defined as:

$$h_{rigid} = R - r + e\cos(\varphi-\delta) + B\sin(\varphi-\delta)\tan\alpha_x + B\cos(\varphi-\delta)\tan\alpha_y, \tag{3}$$

where B is bearing width, R is shell radius, r is pin radius, e is eccentricity, φ is circumferential angle, δ is angle of minimal oil film thickness and α_x and α_y are tilting angles.

The third fundamental equation is force equilibrium and it can be presented as:

$$\sum_{i=1}^{n} F_i = 0, \tag{4}$$

External load, hydrodynamic and rough contact forces are only forces considered for the force equilibrium of slide bearing solution. External loads include also inertial and other forces and they are completely obtained by a solution of Virtual Engine in Multibody software, presented by Novotny (2009).

3. Numerical Methods

To simplify the writing and to improve the numerical solution of equation (1), the following variables can be used

$$H = \frac{h}{R-r} = \frac{h}{c}, \quad \varphi = \frac{x}{R}, \quad Z = \frac{z}{B}, \quad P = \frac{pc^2}{R^2\eta_0}, \quad \xi = \frac{\bar{\rho}H^3}{12\bar{\eta}}.$$
$$k = \frac{B}{R} = \frac{2B}{D}, \quad \bar{\eta} = \frac{\eta}{\eta_0}, \quad \bar{\rho} = \frac{\rho}{\rho_0}, \quad \omega_e = \omega - 2\dot{\delta}. \tag{5}$$

The symbol H denotes a dimensionless thickness of the oil gap, D is bearing diameter, η_0 is dynamic viscosity at atmospheric pressure and room temperature and ρ_0 is the density at atmospheric pressure. By using equations (5), equation (1) can be written in the form:

$$\frac{\partial}{\partial\varphi}\left(\xi\frac{\partial P}{\partial\varphi}\right) + \frac{1}{k^2}\frac{\partial}{\partial Z}\left(\xi\frac{\partial P}{\partial Z}\right) - \omega_e\frac{\partial(\bar{\rho}H)}{\partial\varphi} - \frac{\partial(\bar{\rho}H)}{\partial t} = 0. \tag{6}$$

The transient term (time derivative of $\bar{\rho}H$) of equation (6) is now neglected if only a comparison of different approaches is required. Otherwise, this term is also important because slide bearing operates under highly time variable conditions.

For the discretization of equation (6), the finite difference method (FDM) is used and for numerical solution, Gauss-Seidel method supplemented by strategies (e.g. SOR – Successive over relaxation method) to accelerate the calculation is used. Equation (6) is discretized and solved for each point of the grid according to the relationship:

$$\bar{P}_{i,j} = \tilde{P}_{i,j}(1-\omega_{op_i}) + \omega_{op_i}\delta_{i,j} \text{ and } \bar{P}_{i,j} = 0 \text{ if } \bar{P}_{i,j} < 0 . \qquad (7, 8)$$

The $\delta_{i,j}$ is defined as

$$\delta_{i,j} = \frac{a_1(\xi_{i+1/2,j}P_{i+1,j} + \xi_{i-1/2,j}P_{i-1,j}) + a_2(\xi_{i,j+1/2}P_{i,j+1/2} + \xi_{i,j-1/2}P_{i,j-1/2}) - a_3(\bar{\rho}H_{i+1,j} - \bar{\rho}H_{i-1,j})}{a_1(\xi_{i+1/2,j} + \xi_{i-1/2,j}) + a_2(\xi_{i,j+1/2} + \xi_{i,j-1/2})} , \qquad (8)$$

where

$$a_1 = \frac{1}{h_\varphi} \, , \; a_2 = \frac{1}{(kh_Z)^2} \, , \; a_3 = \frac{1}{2h_\varphi} . \qquad (9)$$

Coefficients h_φ and h_Z are the integration steps, the wavy line indicates the value of the previous iteration, the overline denotes the value of the current iteration. ω_{opi} is over-relaxation parameter.

The nature of the coupled fluid-structural problem is non-linear, therefore the Newton-Raphson algorithm (NRA) is used for a solution of two dimensional problem.

The pure boundary lubrication according to Greenwood and Tripp (1970) is used when oil supply is insufficient. The nominal pressure can be calculated as

$$p_{rough} = K_{GT} E' F_{h/\sigma} , \qquad (10)$$

where

$$K_{GT} = \left(8\pi\frac{\sqrt{2}}{15}\right)(N\beta\sigma)^2\left(\sqrt{\frac{\sigma}{\beta}}\right) \text{ and } E' = \frac{E_1E_2}{E_2(1-\upsilon_1^2)+E_1(1-\upsilon_2^2)} . \qquad (11)$$

E_1 and E_2 denotes Young's modulus of the pin and shell respectively, υ_1 and υ_2 are Poisson numbers, σ is composite summit height standard deviation, β is radius at asperity summit and N is number of asperities per unit area.

4. Results

The relative eccentricity, total friction moment, maximal hydrodynamic pressure, minimal oil film thickness and oil flow are the results selected to present different computational approaches.

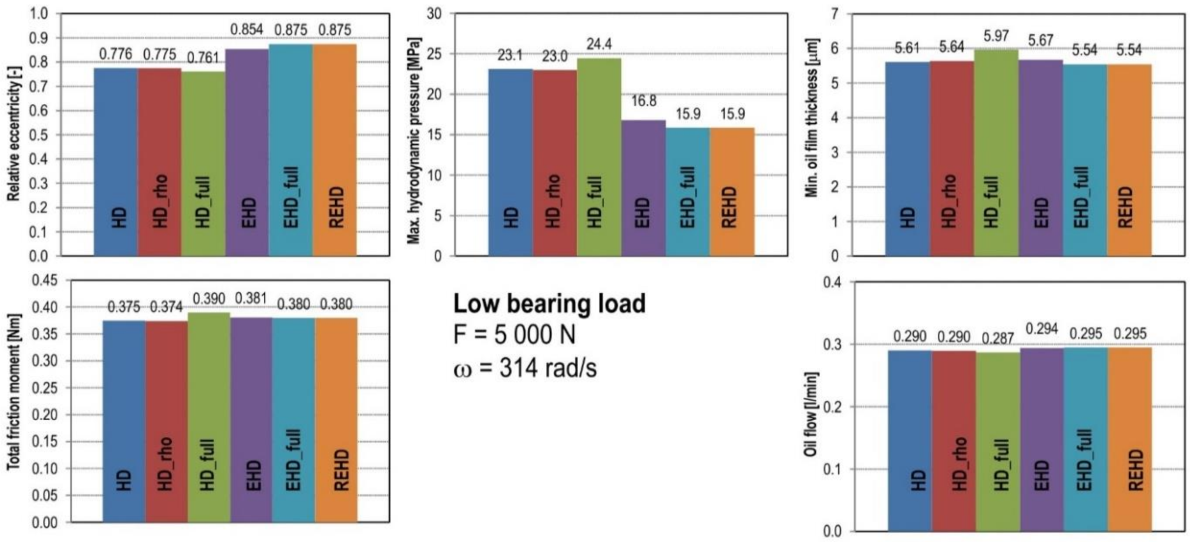

Fig. 1: Result comparisons for low bearing load.

For comparison, these computational approaches are considered: the hydrodynamic solution (HD); the hydrodynamic solution under variable oil density (HD_rho); the hydrodynamic solution under variable oil density and dynamic viscosity (HD_full); the elastohydrodynamic solution (variable viscosity and density) with rigid pin approach (EHD); the elastohydrodynamic solution considering also elastic pin (EHD_full); the elastohydrodynamics and contacts of rough surfaces solution (REHD). The REHD approach considers variable density, viscosity and elastic deformations of the pin and the shell, this approach is taken as a base for all comparisons.

The results are presented for the main bearing of 1.2 litre SI engine. The engine speed of 3000 rpm is always applied. Different loads are used to demonstrate the results: 5 kN as an example of relative low bearing loads (Fig. 1); and 15 kN as a peak bearing load for the target engine (Fig. 2).

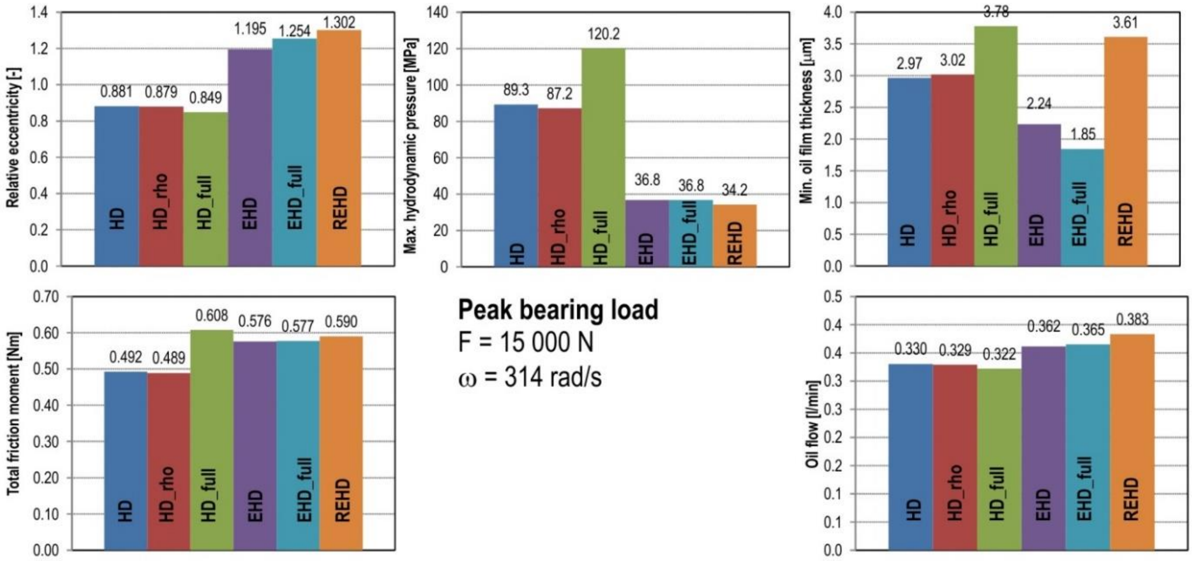

Fig. 2: Result comparisons for peak bearing load.

5. Conclusions

The comparison of the different computational strategies shows that for low bearing load the more complex models based on EHD with rough contacts do not introduce any decisive differences. On the other hand if the load increases, the importance of elastic deformations and contacts of roughness peaks (mixed lubrication conditions) requires more sophisticated computational models.

Of course, there is much more parameters describing the solved system (shear stress influences, design of the engine block, surface treatment etc.) that influences the complexity of appropriate computational models but future approaches for a solution of the slide bearing are evident: full thermoelastohydrodynamic solution incorporating contacts of surface roughness of anisotropic properties.

Discretisation using FDM, FEM, or FVM (Finite Volume Method) solved iteratively by Gauss-Seidel method with SOR are the ways how we have to solve the coupled fluid-structural problem.

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