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Analysis of diverse simulation models for combustion engine journal bearings and the influence of oil condition

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Abstract

The paper deals with the comparison of different models, from simple to complex, for the simulation of non-stationary response of the journal bearings used in combustion engines. The variety of simulation models covers classical methods from Holland, Butenschoen and numerical methods based on the Hydrodynamic (HD), Elasto-hydrodynamic (EHD) and Thermo-elasto-hydrodynamic (TEHD) lubrication theory. Several crankshaft main bearings and connecting rod big end bearings are investigated. The comparison includes the following bearing parameters: Peak Oil Film Pressure (POFP), Minimum Oil Film Thickness (MOFT) and oil flow. Calculation time is compared, too. Estimation of oil viscosity is discussed over a typical temperature and pressure range found in combustion engines, including the influence of the fuel dilution as an example of the oil aging phenomena.

Keywords: Bearing; Simulation; Hydrodynamic; Viscosity; Fuel dilution

1. Introduction

Performance and reliability of journal bearings has always been of central importance for the crank train operation. In this paper, the results of different models for the simulation of non-stationary response of the journal bearings used in combustion engines are compared. These simulation models include classical methods from Holland and Butenschoen and numerical methods based on the Hydrodynamic (HD), Elasto-hydrodynamic (EHD) and Thermo-elasto-hydrodynamic (TEHD) lubrication theory. Holland and Butenschoen algorithms are semi-analytical and work only with fully filled bearing, rigid shell and constant viscosity. HD and EHD finite volume methods can account for cavitation (filling ratio) and pressure-dependent viscosity. The HD algorithm implies rigid and EHD elastic bearing shell. The TEHD method implies elastic shell and works with pressure and temperature dependent viscosity.

Generally, pressure and temperature have been considered as potentially important variables in numerical simulation but shear thinning and especially oil condition are often ignored. The latter can be dated back to the lack of cooperation between oil manufacturers, oil condition laboratories and simulation software developers. Research work on oil viscosity comprising temperature, pressure and shearing is rare.

Within this work, several crankshaft main bearings and connecting rod big end bearings are investigated. Bearing loads are obtained from the simulation of the complete engine. Selected simulation results are discussed and several conclusions are drawn. All simulations presented in this paper are performed using AVL software EXCITE and EXCITE Designer.

2. Overview of the simulation models

2.1. Holland and Butenschoen model

All methods described in this paper are based on the Reynolds equation. Butenschoen and Holland [1,2] consider Reynolds equation (1) in a specific coordinate system:

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Nomenclature

…‘	function in (x,y’,z,t)	**D**	damping matrix of the shell
…	function in (x,$\bar{y}$,z,t)	f	nodal force acting on oil film
δ	angle of smallest oil gap in spatial system	f_A	outer force acting on journal
ε	relative eccentricity	f_J	oil film force acting on journal
ϕ	angle starting at the widest gap	h	clearance
η	dynamic viscosity	**K**	stiffness matrix of the shell
θ	filling ratio	m	journal mass
κ	thermal conductivity of oil	**M**	mass matrix of the shell
κ_B	thermal conductivity of shell	p	pressure
ρ	oil density	t	time
ρ_B	shell density	T	oil temperature
ω*	effective angular velocity	T_a	ambient temperature
$\frac{\partial}{\partial n}$	normal derivative	T_B	shell temperature
B_i	Biot number (heat transfer coefficient)	u_B	shell deformations
BR	width of bearing shell	u,v,w	velocities of oil film
c_B	specific heat of shell	U	journal circumferential velocity
c_P	specific heat of oil	$x,y,\bar{y},z$	coordinates of oil film
D	diameter of bearing shell	x_J	journal position

$$\frac{\partial}{\partial\phi}\left((1+\varepsilon\cos\phi)^3\frac{\partial p}{\partial\phi}\right)+\left(\frac{D}{BR}\right)^2\frac{\partial}{\partial z}\left((1+\varepsilon\cos\phi)^3\frac{\partial p}{\partial z}\right)$$
$$=-6\cdot\left(\varepsilon\sin(\phi-\delta)-\frac{2\varepsilon}{\omega^*}\frac{\mathrm{d}\delta}{\mathrm{d}t}\sin(\phi-\delta)-\frac{2}{\omega^*}-\frac{\mathrm{d}\varepsilon}{\mathrm{d}t}\cos(\phi-\delta)\right) \tag{1}$$

To solve this differential equation, the journal motion is divided into a plain rotary motion and plain displacement. Boundary conditions for solving the equation are:

- Pressure at the bearing edges is equal to zero: $p(\phi,z=\pm\mathrm{BR}/2)=0$.
- For the rotation, pressure in the widest gap is zero: $p_D(\phi=0,z)=0$.
- By Reynolds, for the rotation, pressure is zero in the areas where the pressure gradient in the circumferential direction becomes zero. This will happen on a bended line $\phi=\phi_0(z)$: $p(\phi=\phi_0(z),z)=0$ with $\partial p/\partial\phi_{\phi=\phi_0}=0$.

This boundary condition is not considered by Holland. Bearing supporting power is calculated by the equations containing Sommerfeld numbers [3] for rotation and radial displacement. For their calculation Holland and Butenschoen have used different approximations.

2.2. Hydrodynamic and elasto-hydrodynamic methods (HD and EHD)

HD and EHD methods are based on the solution of the extended Reynolds equation given by:

$$\frac{\partial}{\partial x}\left(\frac{1}{12\eta}\theta h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{1}{12\eta}\theta h^3\frac{\partial p}{\partial z}\right)$$
$$=\frac{u_2+u_1}{2}\frac{\partial(\theta h)}{\partial x}+\frac{\partial(h\theta)}{\partial t} \tag{2}$$

Reynolds equation is solved for pressure p in the lubrication region and filling factor θ in the cavitation region. The filling factor serves to model the cavitation effects and is defined as the fraction of volume filled with oil to the total volume [4]. Numerous authors considered the solution of the Reynolds equation (2) with and without the shell deformation [5–10]. Shell deformation due to the oil film pressure is considered by the linear dynamic equation system (3) obtained from the FEM discretization of the shell. In our case this equation is solved by means of the implicit time integration method.

$$\mathbf{M}\ \ddot{u}_B+\mathbf{D}\ \dot{u}_B+\mathbf{K}\ u_B=f \tag{3}$$

The structural matrices are subsequently condensed to the radial degrees of freedom of the nodes lying at the inner shell surface. Motion of the journal is determined by the equation of the bearing pin subjected to external and inertia forces:

$$m\ddot{x}_J=f_J+f_A \tag{4}$$

2.3. Thermo-elasto-hydrodynamic method (TEHD)

TEHD method solves the Reynolds equation together with the energy equation of the oil film. Within the Reynolds equation both oil viscosity and density vary in three dimensions of space [11]. Reynolds equation is again solved for pressure p in the lubrication region and filling