中國機械工程學刊第三十九卷第四期第 365~374 頁(民國一百零七年)
Journal of the Chinese Society of Mechanical Engineers, Vol.39, No.4, pp 365~374 (2018)

An Improved TEHL Model for Main Bearings of IC Engine with Flexible Crankshaft and Block

Qing-hui Zhang*, Xu Zheng*, Huan-yu Zhang*, Jie Mao*, Ping Song* and Zhi-yong Hao*

Keywords: tribology, crankshaft, main bearing, TEHD, asperity contact, mass conserving.

ABSTRACT

Considering the elastic deformation and thermal distortion of bearing surface, an improved thermo-elasto-hydrodynamic (TEHD) lubrication model for crankshaft bearings is proposed in this study. Oil film viscosity and density variation caused by temperature and pressure are considered. In addition, the mass conserving cavitation algorithm is applied in the analysis. Based on the compliance matrix method, elastic deformation and thermal distortion for the bearing surface are calculated. In order to get the temperature distribution of oil film, the energy equation which considers heat convection in three directions and heat conduction in radial direction is applied and solved by finite difference method. Meanwhile, the lubrication status of crankshaft bearing is solved by Reynolds equation, and the motion of crankshaft is solved by dynamic equations. To investigate the effects of thermal transfer of oil film on the state of bearing lubrication quantitatively, the peak oil film pressure, flow rates, minimum oil film thickness and asperity contact pressure are calculated with the elasto-hydrodynamic(EHD) lubrication model and the improved TEHD lubrication model respectively. The research results verify the accuracy of the improved TEHD lubrication model, which can effectively predict the lubrication status of crankshaft bearing.

INTRODUCTION

The crankshaft bearing is a typical dynamical loaded thermo-elastic lubrication bearing (Kim et al., 2004). The lubrication state of main bearing oil film is of great influence to the stress distribution of crankshaft. Paranjpe et al. (2000) made comparisons between calculations and experimental measurements for minimum bearing oil film thickness in main and connecting rod bearings of a typical automotive v6 engine and found out that the theoretical calculations had a reasonable accuracy. Adatepe et al. (2011) compared the frictional behavior of the concentric circular cylindrical non-grooved (plain) and micro-grooved journal bearings by using the purpose-built journal bearing test rig. Considering the limitations of the experimental methods and accuracy of simulation model, the dynamic calculation for crankshaft and oil film has become a very important part of preliminary design for crank train system.

By far, the numerical simulation method for the lubrication model of bearing has made some remarkable progress. Wang et al. (2004, 2004) investigated the effects on the connecting rod bearing shape of the inertia force and the pre-load on the bolts, and found that a thinner minimum oil film and a larger peak hydrodynamic pressure appeared in a deformed connecting rod bearing than in a rigid connecting rod bearing. Li et al. (2011) studied the performance of rotor/bearing/seal system considering the coupled effects of the nonlinear oil film force, the nonlinear seal force, and the mass eccentricity of the disk.

Sun et al. (2007) studied the effect of lubrication status of bearing on crankshaft strength using the beam method and finite element method considering misalignment of crankshaft. Sun et al. (2009) used whole crankshaft beam-element finite-element method to calculate crankshaft bearing load and studied crankshaft deformation of a multi-cylinder engine. The results showed that this method was simple, convenient, time-saving for calculating crankshaft deformation with acceptable calculating precision. Sun et al. (2010) analyzed the EHL of crankshaft bearing under crankshaft deformation and whole block deformation. In these calculation models, only the crankshaft misalignment and the elastic deformation of bearing surface were considered. Gui et al. (2011) introduced a dynamic solution method for crankshaft bearing to get the journal center trajectory considering the lubrication mechanics of oil film. Li et al. (2012) developed the quasi-coupling calculation method of the transient fluid dynamics of the oil film in a journal bearing and rotor dynamics to

Paper Received December, 2015. Revised August, 2016. Accepted October, 2016. Author for Correspondence: Xu Zheng. E-mail: zhengxu@zju.edu.cn

* *College of Energy Engineering, Zhejiang University, Hangzhou 310027, China*

analyze the effects of misalignment on the transient flow field and performance characteristics of a journal bearing. However, in those papers, the mass conserving cavitation algorithm and thermal distortion of bearing surface were not included which, however, can't be ignored in engine bearing lubrication analysis.

Kim et al. (2001) used the TEHL model to analysis the lubrication performance of oil film considering mass conserving cavitation algorithm and thermal distortion of bearing surface. Fatu et al. (2006) presented a new model for TEHL applied to dynamically loaded bearings with time-dependent three-dimensional variable temperature a fast new heat flux conservation algorithm. By this article, he proved that thermal affects in the connecting rod bearing analyses took a very important part. Habchi et al. (2010) studied the effects of pressure and temperature dependence of a conventional lubricant’s thermal properties on the behavior of heavily loaded thero-elasto-hydrodynamic lubrication contacts, and found that the variations in thermal properties have negligible effect on film thickness under pure rolling regime. By far, most of the researches for bearing lubrication status focus on connecting rod bearing analysis. The study of thermal effects in the crankshaft bearing analysis has not been taken seriously.

To investigate the thermal effects in crankshaft bearing analysis, this paper takes advantage of TEHL model to apply it for lubrication status assessment of crankshaft main bearing. Firstly, a comprehensive simulation model with mass conserving cavitation algorithm is presented which includes crankshaft misalignment, thermal distortion and elastic deformation of bearing surface. Then, the oil film temperature is solved by finite difference method considering the heat convection in three directions and heat conduction in radial direction. Finally, the simulation results are compared with that of the EHL model, and the thermal effects to the lubrication status of crankshaft bearings has been studied.

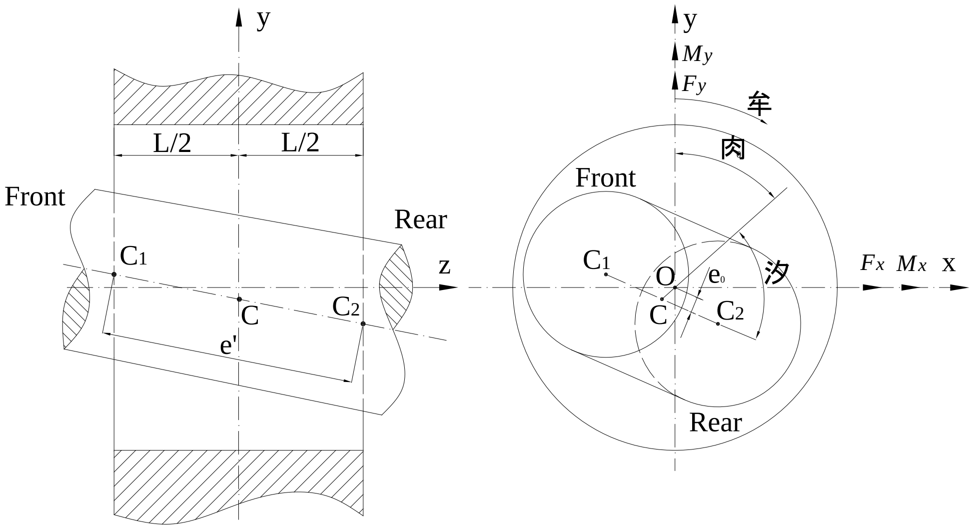


Fig. 1. Configuration of misaligned journal bearing

GOVERNING EQUATIONS

As an elastic system, the crankshaft misalignment takes place because of the periodic changes of magnitude and direction of bearing load during the engine operation cycles. Figure 1 shows the diagram for the misalignment of crankshaft journal (Guha et al., 2000). In the diagram, a Cartesian coordinate system is employed as the reference coordinate with its origin fixed at the bearing center.

Reynolds Equation

The Reynolds equation for the radial sliding bearing can be described as follows:

$$\nabla\left(\frac{\rho h^3}{12\mu}\nabla p\right)=\frac{1}{2}\nabla\left(\rho h U\right)+\frac{\partial \rho h}{\partial t}. \tag{1}$$

Where $\nabla = i\frac{\partial}{\partial x}+j\frac{\partial}{\partial z}$, U is velocity at circumferential direction of journal. p is oil film pressure, ρ is the oil film density, μ is the viscosity of oil film, h is thickness of oil film.

Film Thickness

Considering the journal misalignment, elastic deformation and thermal distortion of bearing surface, the oil film thickness can be expressed by

$$h=c+e_0\cos\left(\theta-\psi_0\right)+\frac{e'}{L}y\cos\left(\theta-\alpha-\psi_0\right)+ h_e+h_t \tag{2}$$

where c is radial clearance, e_0 and ψ_0 represent the eccentricity vector of the journal at the axial mid-plane of bearing, L is length of bearing, e' is the magnitude of the projection of journal center line on the mid-plane, h_e is the elastic deformation, h_t is the heat distortion.

Assuming the radial sliding bearing deformation as a linear process, the elastic deformation and heat distortion of bearing surface can be calculated by compliance matrix method.

$$h_e = [K][P_o] \tag{3}$$

$$h_t = [K]\int_v B^T D\varepsilon_T dv. \tag{4}$$

Where [K] is compliance matrix, [P_o] is oil film load matrix, B is geometry matrix, D is the elastic matrix, ε_T is thermal strain. The compliance matrix can be solved by finite element method (Sun et al., 2007).

Energy Equation

As oil film temperature is an important item for the oil property and oil film thickness, it must be calculated precisely. The energy equation proposed by Bukovnik et al. (2006) considering the heat convection in three directions and heat conduction in radial direction.

$$\rho c_p\left\{\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+w\frac{\partial T}{\partial z}+\frac{1}{h}\left[v-y\left(\frac{\partial h}{\partial t}+u\frac{\partial h}{\partial x}+w\frac{\partial h}{\partial z}\right)\right]\frac{\partial T}{\partial y}\right\}+$$
$$+\frac{T}{\rho}\frac{\partial\rho}{\partial T}\bigg|_p\left(\frac{\partial p}{\partial t}+u\frac{\partial p}{\partial x}+w\frac{\partial p}{\partial z}\right)-\frac{k}{h^2}\frac{\partial^2 T}{\partial^2 y}=$$
$$\frac{\mu}{h^2}\left[\left(\frac{\partial u}{\partial y}\right)^2+\left(\frac{\partial w}{\partial y}\right)^2\right]+\frac{\tau_a}{h^3}\left(u_{shell}-u_{journal}\right). \tag{5}$$

Where v, u, w is velocity of oil film, c_p is Specific heat, T is the temperature of oil film, k is heat conductivity, τ_a is shear stress, u_{shell} and $u_{journal}$ is the velocity of contacting surface.

The heat transfer between the oil film and the bearings surface and journal is assigned as boundary conditions for the energy equation. With the assumption that journal temperature is constant, the heat generated within oil film flows into bearing shell continuously. The general form of the solid heat conduction equation can be expressed by eq. (6); the outer surface boundary condition can be expressed by eq. (7).

$$\frac{\rho_s c_{ps}}{k_s}\frac{\partial T}{\partial t} = \nabla\nabla T. \tag{6}$$

$$\frac{\partial T}{\partial n} + B_i \Delta T = 0. \tag{7}$$

where $\nabla = i\frac{\partial}{\partial x} + j\frac{\partial}{\partial z}$, ρ_s is density of bearing material, c_{sp} is specific heat, k_s is heat conductivity, B_i is biot number.

Equations of Density and Viscosity

The μ and ρ are function of temperature and pressure, μ can be described by Barus-Reynolds eq. (8), ρ can be described by eq. (9) (Bair et al., 2001; Stachowiak et al., 2001)

$$\mu(p,T) = \mu_0 \exp\left[\alpha p - \beta(T - T_0)\right]. \tag{8}$$

$$\rho(p,T) = \rho_0\left(1+\frac{D_1 p}{1+D_2 P}\right)\left[1-\xi_0 e^{\vartheta p}(T-T_0)\right]. \tag{9}$$

Where α is Pressure-viscosity coefficient, β is temperature-viscosity coefficient, D_1 and D_2 are the density-pressure constant, ξ_0 is thermal expansion coefficient.

Asperity Contact Equation

The asperity contact between crankshaft and main bearing usually happens because of the poor lubrication status. To solve this problem, the Greenwood-tripp theory is introduced in the mixed lubrication scheme, which can be expressed by

$$p_a = \frac{16\sqrt{2}\pi}{15}\left(\zeta R_s \chi_s\right)^2\sqrt{\frac{\zeta}{R_s}}E^* F'\frac{h-\delta_s}{\chi_s}, \tag{10}$$

$$F' = \begin{cases} 4.4086e^{-5}\left(4-\dfrac{h-\delta_s}{\chi_s}\right)^{6.804} & ,\dfrac{h-\delta_s}{\chi_s} < 4 \\ 0 & ,\dfrac{h-\delta_s}{\chi_s} \geq 4 \end{cases}. \tag{11}$$

Where p_a is nominal pressure, ζ is Root Mean Square of the summit roughness, R_s is radius of asperity summit, χ_s is numbers of summits of each surface, δ_s is composite value of summit heights. E^* is composite elastic modulus for rough surfaces, and can be expressed as

$$E^* = 1\bigg/\left(\frac{1-v_1^{\,2}}{E_1}+\frac{1-v_2^{\,2}}{E_2}\right). \tag{12}$$

Where E_1, E_2 are elastic modulus, v_1, v_2 are Poisson ratios of materials near the contacted surfaces.

Mass Conserving Cavitation Algorithm

The mass conserving cavitation algorithm is based on the idea of Kumar and Booker (Kumar et al., 1991). This algorithm assumes that the oil film region contains liquid region and gas/vapor region. The liquid is assumed to be incompressible, and the gas/vapor is assumed to be compressible with zero bulk modulus. The density and viscosity of the biphase lubricate mixture can be expressed as

$$0 \leq \rho \leq \rho_{liq}. \tag{13}$$

$$0 \leq \mu \leq \mu_{liq}. \tag{14}$$

The relation between the density and viscosity in the biphase region is assumed

$$\mu / \mu_{liq} = \rho / \rho_{liq}. \tag{15}$$

According to the oil density, the oil film can be divided into complete film region and incomplete film region. The complete film region is filled with liquid oil

($\rho = \rho_{liq}$), and the incomplete film region is filled with liquid oil and oil gas ($\rho < \rho_{liq}$). The complete oil film region can be further divided into region A ($\partial\rho / \partial t = 0$ and $p \geq p_{cav}$) and region B($\partial\rho / \partial t < 0$ and $p = p_{cav}$) by iterative procedure. After this, the density and the viscosity of oil film can be solved by the Reynolds eq. (1).

Force Equilibrium Equation

When ignoring the inertia of oil film, the motion of crankshaft can be expressed by

$$m\frac{du_j}{dt} = f_j + f_p. \tag{16}$$

Where u_j is the velocity of journal center, f_i is external force acting on journal center, f_p is the hydrodynamic force acting on journal.

APPLICATION MODELS

Taking a 6-cylinder diesel engine as an example, the lubrication status of the crankshaft main bearing is analyzed at the rated condition of the engine. The parameters of the engine configuration and the boundary conditions for lubrication analysis are listed in table 1.

The crank train model includes crankshaft, flywheel, pulley and pulley damper. As shown in Figure 2, the crankshaft is divided by hexahedron elements which contain 236,480 elements, 254,887 nodes. The engine block model includes engine block, main bearing cap and bearing shell. As shown in Figure 3, engine block is divided by tetrahedron elements which contain 758,972 elements, 210,795 nodes.

Table 1**.** Parameters of the TEHL model

Engine configuration		Type	Diesel engine
		Configuration	6 cylinders, L type
		Bore	0.128 m
		Stroke	0.155 m
		Firing order	1-5-3-6-2-4
		Power	338 kw
		Speed	1900 r/min
Bearing Configuration	Width of bearing	L	0.032 m
	Radius	R	0.052 m
	Radius clearance	c	52 μm
Lubricant	Density	ρ	831.9 kg/m^3
	Viscosity	μ	11.4 MPa·s
	Temperature	T	373 K
	Viscosity-temperature constant	β	0.03 K^{-1}
	Viscosity-pressure constant	α	2.2e-8 m^2/K
	Specific heat	c_p	1950 J/kg·K
	Heat conductivity	k	0.14 W/m·K
Bearing structure	Elastic modulus	E_1	210 GPa
	Poisson ratio	ν_1	0.3
	Specific heat	c_{sp}	400 J/kg·K
	Heat conductivity	k_s	50 W/m·K
	density	ρ_s	7800 kg/m^3
Journal structure	Elastic modulus	E_2	210 GPa
	Poisson ratio	ν_2	0.3
Operating conditions	Rotational speed	ω	1900 r/min
	Supply pressure	p	5 bar
	Supply temperature	T	373 K
	Ambient temperature	T	373 K

Fig. 2. Finite element model of crankshaft

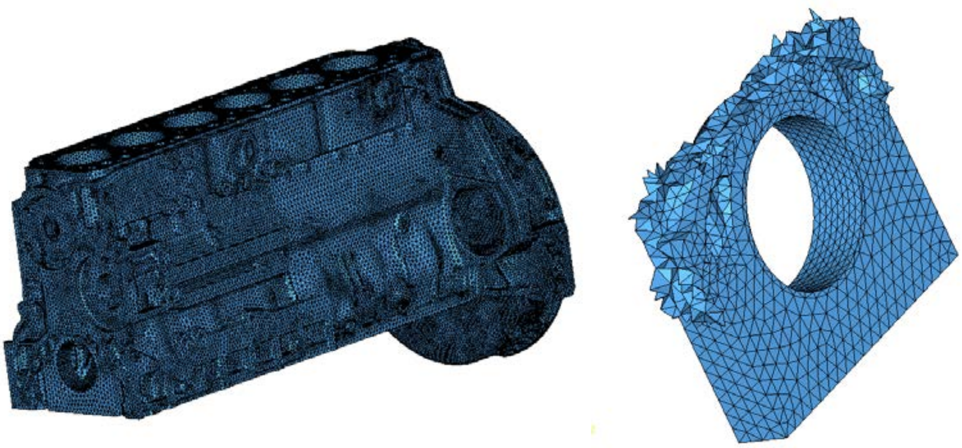
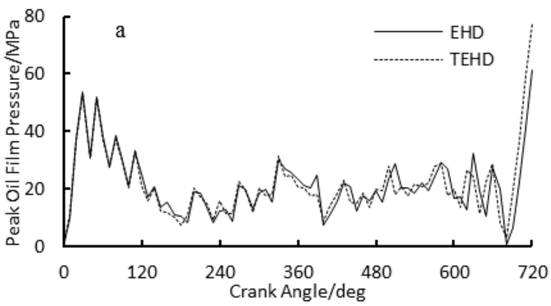


Fig. 3. Finite element model of engine block

SIMULATION RESULTS

To investigate the effect of thermal performance of oil film on the lubrication status of bearing, the EHL model is used as contrast model which ignores the temperature items in eq. (2), (7), (8) and the energy eq. (5). Firstly, the rationality of the simulation results is verified through comparing the minimum oil film thicknesses from the EHL and the TEHL models with the experimental data in Paranjpe et al.'s work [2]. The minimum oil film thickness of the main bearings from the EHL and the TEHL models is in the range of 1.2~1.9μm and 0.6~1.7μm, respectively. Moreover, that of the third main bearing is 1.8μm and 1.3μm, respectively. The minimum oil film thickness of the third main bearing from experiment for a V6 engine under 1500r/min, 192N·m condition using 5W-30 oil is 1.16μm [2]. The simulation results are very similar to the experimental result and they are all in the same order of magnitude. This can verify the rationality of the simulation results to some extent.

The comparison between TEHD simulation results and EHD simulation results are shown in Figure 4, Figure 5 and Figure 6. The maximum oil film pressure, the minimum film thickness and the maximum asperity contact pressure are displayed respectively from 0°CA and 720°CA. As shown in the figures, the highest pressure and the minimum oil film thickness of crankshaft main bearing has changed dramatically when the bearing surface thermal distortion is considered. For example, the highest pressure of forth bearing decreases from 104.4 MPa to 87.6 MPa at 610°CA while it decreases from 109.6 MPa to 82.3 MPa at 380°CA of the seventh bearing. The minimum thickness of the fourth bearing decreases from 1.54 μm to 0.85 μm at 610°CA while it decreases from 1.18 μm to 0.54 μm at 380°CA of the seventh bearing. Moreover, the maximum asperity contact pressure of the fourth bearing increases from 68.7 MPa to 128.6 MPa at 610°CA while it increases from 87.3 MPa to 146.6 MPa at 380°CA of the seventh bearing.



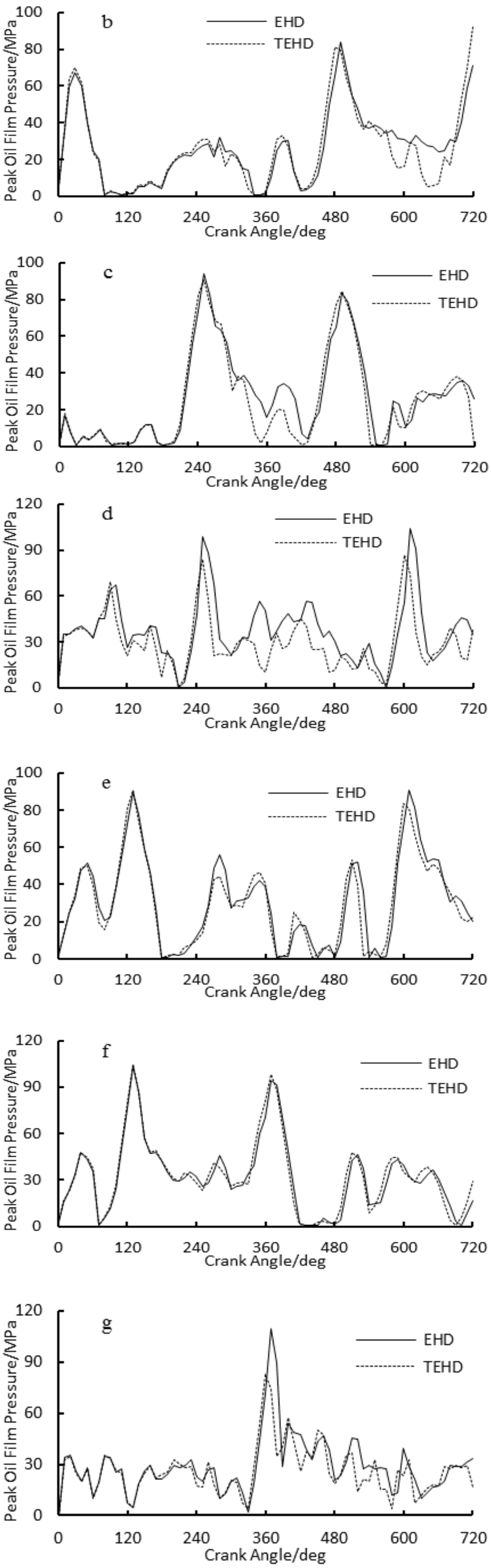
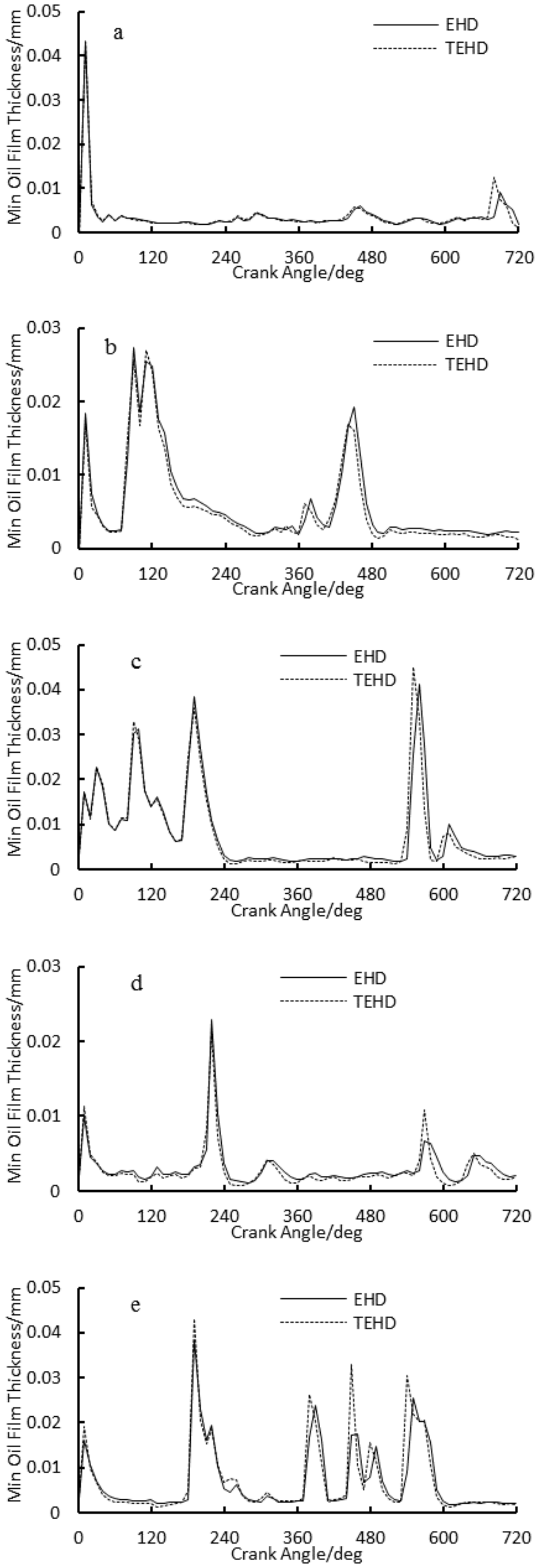


Fig. 4. Maximum pressure of Oil film of bearings: a~g represent bearing 1~7 respectively



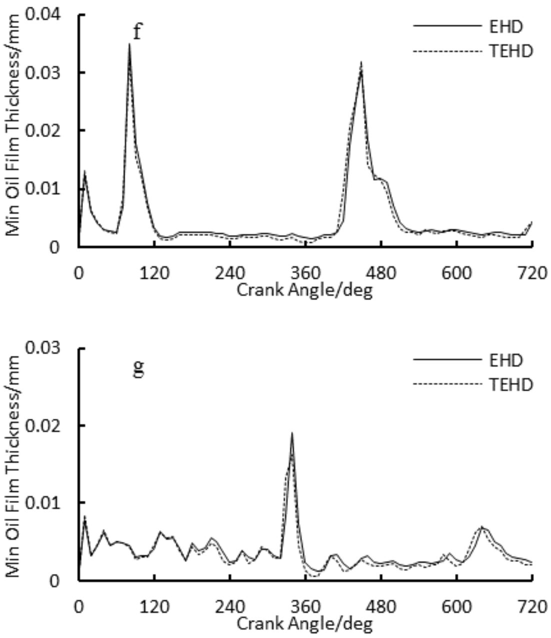
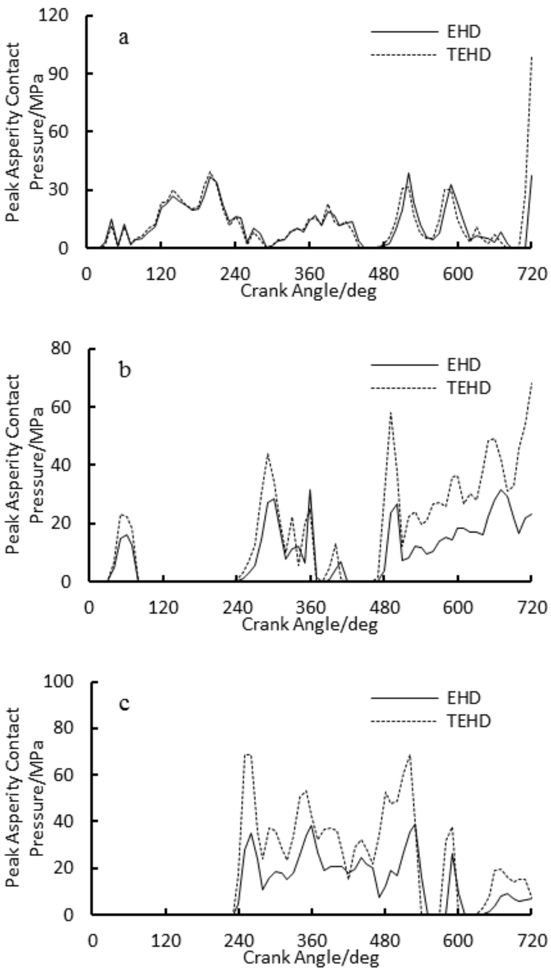


Fig. 5. Minimum Oil Film Thickness of bearings: a~g represent bearing 1~7 respectively



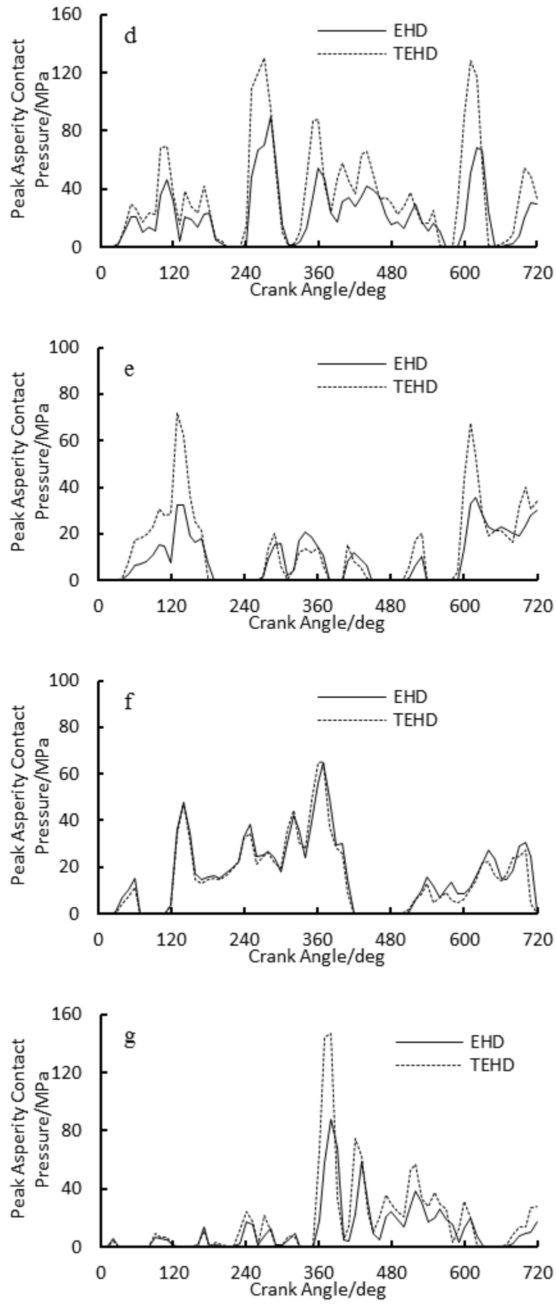


Fig.6. Maximum Asperity contact pressure of bearings: a~g represent bearing 1~7 respectively

The distribution of hydrodynamic pressure and asperity pressure of each bearing is shown in Figure 7~ Figure 13. In these figures, the left vertical axis is bearing width; the right vertical axis is bearing pressure; the horizontal axis is circumference of bearing in angle mode. From the distribution diagram we can find that, the hydrodynamic pressure of each bearing is at the safe level, and the hydrodynamic pressure area is extensive at each bearing. The asperity contact pressure of the fourth bearing and the seventh bearing is especially high. Besides, all the asperity contacts occur at the edge of the bearings, which is the main reason of eccentric wear for crankshaft Journal.

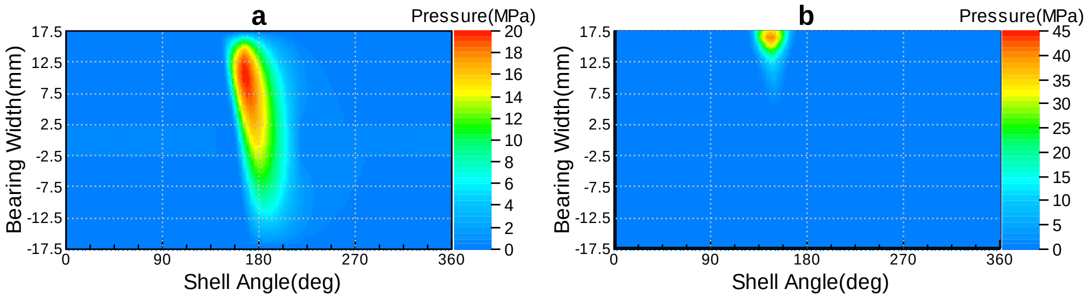


Fig. 7. bearing 1 at the 520 °CA:(a) hydrodynamic pressure; (b) asperity contact pressure

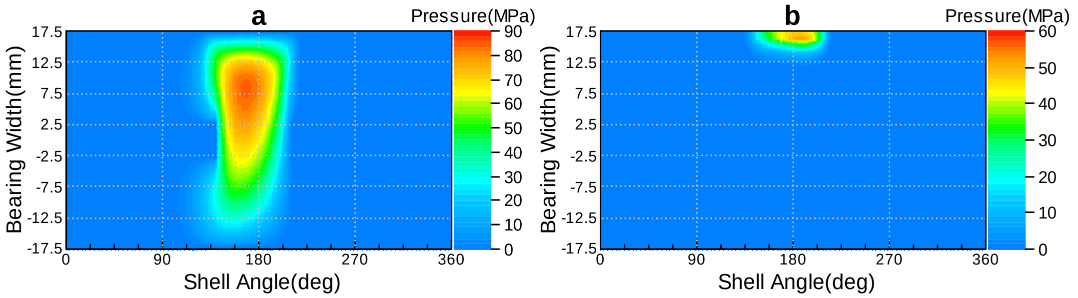


Fig. 8. Bearing 2 at the 490 °CA:(a) hydrodynamic pressure; (b) asperity contact pressure

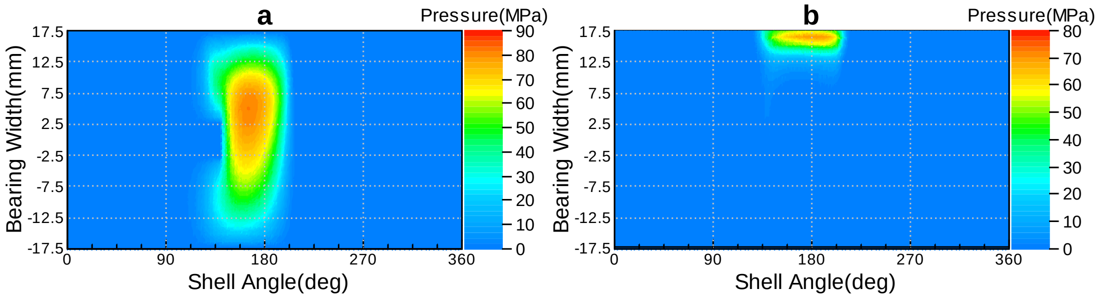


Fig. 9. Bearing 3 at the 260 °CA:(a) hydrodynamic pressure; (b) asperity contact pressure

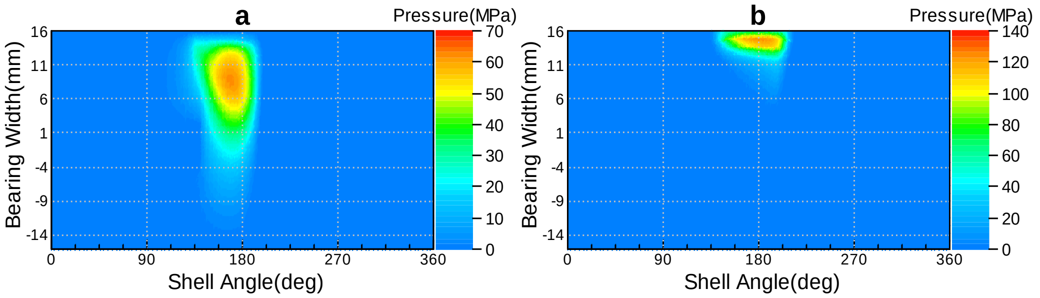


Fig. 10. Bearing 4 at the 620 °CA:(a) hydrodynamic pressure; (b) asperity contact pressure

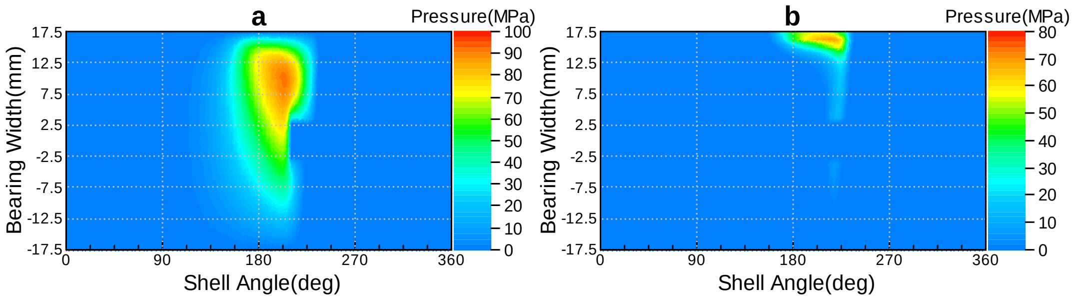


Fig. 11. Bearing 5 at the 130 °CA:(a) hydrodynamic pressure; (b) asperity contact pressure

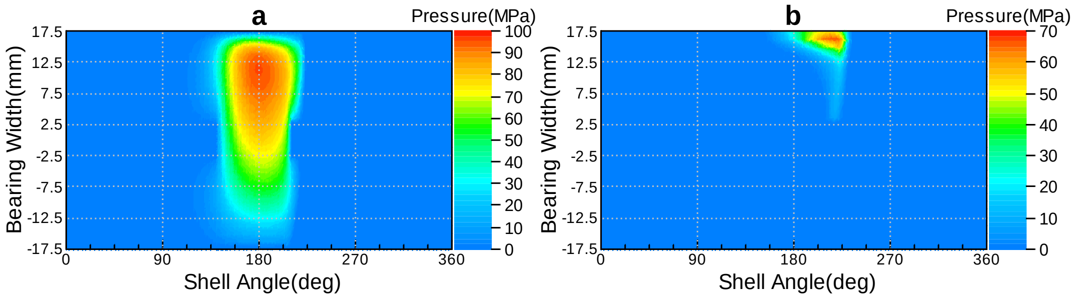


Fig. 12. Bearing 6 at the 380 °CA:(a) hydrodynamic pressure; (b) asperity contact pressure

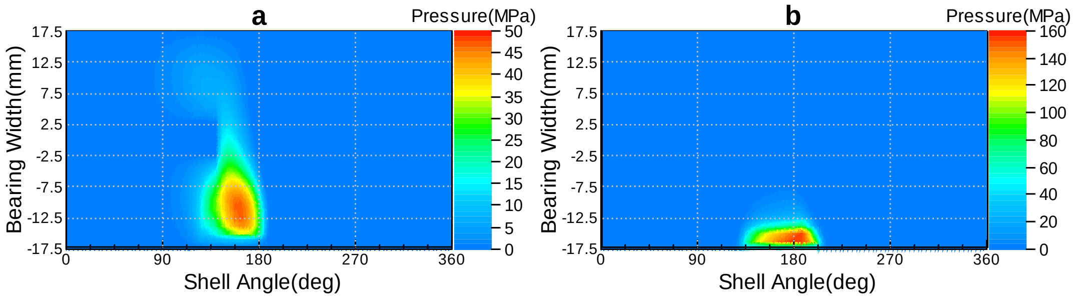


Fig. 13. Bearing 7 at the 380 °CA:(a) hydrodynamic pressure; (b) asperity contact pressure

The simulation results are shown in Table 2, only the first and second bearing minimum oil film thickness meets the design limits (1.5μm), the asperity contact pressure of the fourth and seventh bearing extends the design limits (80MPa), and this asperity contact leads to serious power losses. Among all the bearings, the fourth bearing oil film temperature is the highest. The TEHL simulation results mean that, the crankshaft main bearings are at poor working condition, and the fourth bearing is at very dangerous condition.

Table 2. Simulation results

Bearing No.	Minimum Oil Film Thickness/ μm	Peak Oil Film Pressure/ MPa	Max Hydrodynamic Friction Power Loss/ kW	Asperity Contact Pressure/ MPa	Max Asperity Friction Power Loss/ kW	Percentage of Asperity Contact/ %	Averaged Oil Film Temperature / □C
1	1.7	57.2	1.5	40.4	7.3	17.6	120.1
2	1.5	84.3	2.6	51.6	10.5	21.2	124.0
3	1.3	92.3	0.6	71.0	11.9	22.3	130.0
4	0.7	87.6	0.8	127.9	14.4	19.3	144.7
5	1.3	89.8	0.7	70.3	6.3	19.9	123.3
6	0.8	100.1	0.8	68.2	1.7	21.5	114.1
7	0.6	82.3	0.9	154.7	15.5	16.6	125.0

CONCLUSIONS

1. Considering the journal misalignment and bearing surface thermal distortion, an improved TEHL model with the asperity contact equation and mass-conserving algorithm is proposed for crankshaft main bearing analysis.

2. The level of peak oil film pressure, minimum oil film thickness, asperity contact pressure and oil flow rate have changed dramatically compared with EHL model. The oil film thermal performance has obvious effects on the lubrication status of the bearing. The importance of thermal effects in crankshaft bearing lubrication analysis has been proved in this study.

3. The TEHL model proposed in this paper is a more precise model to evaluate the lubrication status of crankshaft bearings.

ACKNOWLEDGEMENT

This research was supported by the National Natural Science Foundation of China (Grant No. 51705454) and National Key Research and Development Program of China (Grant No. 2016YFB0101604).

REFERENCES

Kim T J., Han J S. “Comparison of the dynamic behavior and lubrication characteristics of a reciprocating compressor crankshaft in both finite and short bearing models,” *Tribology Transactions*, Vol. 47, No.1, pp. 61-69(2004).

Paranjpe R S., Tseregounis S I., “Viola M B. Comparison between theoretical calculations and oil film thickness measurements using the total capacitance method for crankshaft bearings in a firing engine," *Tribology Transactions*, Vol. 47, No.1, pp.345-356(2000).

Adatepe H., Biyiklioglu A., Sofuoglu H. "An experimental investigation on frictional behavior of statically loaded micro-grooved journal bearing," *Tribology International*, Vol. 44, pp.1942-1948(2011).

Wang D., Keith T G., Yang Q M., "Vaidyanathan K. Lubrication analysis of a connecting-rod bearing in a high-speed engine. Part ii: lubrication performance evaluation for non-circular bearings," *Tribology Transactions*, Vol. 47, pp.290-298(2004).

Wang D., Keith T G., Yang Q M., "Vaidyanathan K. Lubrication analysis of a connecting-rod bearing in a high-speed engine. Part i: rod and bearing deformation," *Tribology Transactions*, Vol. 47, pp.280-289(2004).

Li W., Yang Y., Sheng D., Chen J H., Che Y Q. "Nonlinear dynamic analysis of a rotor/ bearing/ seal system," *Journal of Zhejiang University-SCIENCE A (Applied Physics & Engineering)*, Vol. 12, No.1, pp.46-55(2011).

Sun J., Gui C. "Effect of lubrication status of bearing on crankshaft strength," *Journal of Tribology*, Vol. 129, No.4, pp.887-894(2007).

Sun J., Wang J., Gui C L. "Whole crankshaft beam-element finite-element method for calculating crankshaft deformation and bearing load of an engine," *Journal of Engineering Tribology*, Vol. 224, pp.299-303(2009).

Sun J., Cai X., Liu L. "Research on the effect of whole cylinder block on EHL performance of main bearings considering crankshaft deformation for internal combustion engine," *Journal of Tribology*, Vol. 132, No.4, pp.1-6(2010).

Gui C L., Sun J., He Z X., et al. "A dynamic solution method for dynamically loaded bearing," *Tribology Transactions*, Vol. 54, No.3, pp.384-393(2011).

Li Q., Liu S L., Pan X H., Zheng S Y. "A new method for studying the 3D transient flow of misaligned journal bearings in flexible rotor-bearing systems," *Journal of Zhejiang University-SCIENCE A (Applied Physics & Engineering)*, Vol. 13, No.4, pp.293-310(2012).

Kim B J., Kim K W. "Thermo-elastohydrodynamic analysis of connecting rod bearing in internal combustion engine," *Journal of Tribology*, Vol. 123, No.3, pp.444-454(2001).

Fatu A., Hajjam M., Bonneau D. "A new model of thermoelastohydrodynamic lubrication in dynamically loaded journal bearings," *Journal of Tribology*, Vol. 128, pp.85-95(2006).

Habchi W., Vergne P., Bair S., et al. "Influence of pressure and temperature dependence of thermal properties of a lubricant on the behaviour of circular TEHD contacts," *Tribology International*, Vol. 43, pp.1842–1850(2010).

Guha S K. "Analysis of steady-state characteristics of misaligned hydrodynamic journal bearings with isotropic roughness effect," *Tribology International*, Vol. 33, No.1, pp.1-12(2000).

Bukovnik S., Dorr N., Caika V., et al. "Analysis of diverse simulation models for combustion engine journal bearings and the influence of oil condition," *Tribology International*, Vol. 39, No.8, pp.820-826(2006).

Bair S., Jarzynski J., "Winer W O. The temperature, pressure and time dependence of lubricant viscosity," *Tribology International*, Vol. 34, No.7, pp.461-468(2001).

Stachowiak G W., Batchelor A W. *Engineering Tribology*, Boston: Butterworth-Heinemann (2001).

Kumar A., Booker J F. "Finite element cavitation algorithm," *Journal of Tribology*, Vol. 113, No.2 pp.276-286(1991).

基於柔性曲軸和缸體的內燃機主軸承改進 TEHL 模型研究

張慶輝 鄭旭 張煥宇 宋平 郝誌勇
浙江大學能源工程學院

摘 要

針對軸承表面的彈性變形和熱變形，本文提出了一種改進的曲軸軸承熱-彈性-流體動力學（TEHD）潤滑模型。該模型考慮了由溫度和壓力引起的油膜粘度和密度變化，並結合質量守恒空化算法應用於分析。基於柔度矩陣法，計算了軸承表面的彈性變形和熱變形。為了獲得油膜溫度分布，采用考慮三維分布的熱對流和徑向熱傳導的能量方程，並結合有限差分法進行求解。同時利用雷諾方程對曲軸軸承的潤滑狀態進行求解，並通過動力學方程求解曲軸運動特性。為了定量研究油膜熱傳導對軸承潤滑狀態的影響，分別采用彈性-流體動力學（EHD）潤滑模型和改進的 TEHD 潤滑模型，計算了油膜最高壓力、流量、最小油膜厚度和粗糙面接觸壓力。研究結果驗證了改進的 TEHD 潤滑模型的準確性，可以有效預測曲軸軸承潤滑狀態。