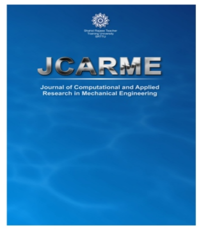
SRTTU

JCARME

ISSN: 2228-7922

Journal of Computational and Applied Research in Mechanical Engineering
Vol. 5, No. 1, pp. 13- 24, Autumn-Winter 2015-16
jcarme.srttu.edu



Thermo-elasto-hydrodynamic (TEHD) analysis of oil film lubrication in big end bearing of a diesel engine

H. Chamani[a,*], H. Karimaei[a], M. Bahrami[b] and S. M. Agha Mirsalim[b]

[a]*Iran University of Science and Technology, Tehran, Iran*
[b]*Amirkabir University of Technology, Tehran, Iran*

Article info:

Received: 07/01/2015
Accepted: 11/05/2015
Online: 11/09/2015

Keywords:

Thermo-elasto-hydrodynamic (TEHD) analysis,
EHD,
BE bearing,
Diesel engine.

Abstract

Nowadays, due to the increasing power of diesel engines, especially heavy duty diesel engines, and increasing gas pressure inside the combustion chamber, the forces acting on the engine bearings have dramatically raised. On the other hand, becaus eof the competition in the market, it is necessary to increase the engine bearing life and reduce its failure as much as possible. The engine bearings analysis is a vital issue in engine design process as well as other related engineering tasks such as engine power upgrading, reverse designing, and bearing failure analysis. So, many attempts have been made to simulate accurate engine bearings. In this paper, results of a thermo-elasto-hydrodynamic (TEHD) analysis of a connecting rod big end (BE) bearing of a heavy duty diesel engine are presented. Here, the oil film viscosity is considered a function of oil's local temperature and pressure. Effects of flexibility of bearing shell and connecting rod structure are also considered. Therefore, the computed oil film pressure and temperature distributions are relatively precise. In the proposed analytical procedure, at first, elasto-hydrodynamic (EHD) analysis is carried out and the averaged fluid velocity in the bearing is obtained. Then, the averaged heat transfer coefficient between oil film and crank pin is calculated, which is used as an input in TEHD analysis. Results of EHD and TEHD analyses are compared with each other and the main characteristic parameters in bearing design are reported and interpreted.

Nomenclature

$\partial/\partial n$: Normal differential
A, B, C, D, E and F: Rodermund parameters
B_i: Biot number (heat transfer coefficient)
C_R: Bearing radial clearance
$\bar{D}$: Condensed damping matrix of a body
e_x and e_y: Eccentricity in x and y directions
f_1, f_2 and f_3: Constants of Dowson-Higginson's equation
$\bar{f}$: Condensed vector of forces and moments
f^* : Vector of connecting forces and moments
f^a : Vector of external loads
$\bar{f}^{gyros}$: Vector of condensed gyroscopic terms
$\bar{f}^{rbAcc}$: Vector of condensed rigid body acceleration
h: Clearance
$\bar{K}$: Condensed stiffness matrix

*Corresponding author
Email address: h.chamani@gmail.com

$\bar{M}$: Condensed mass matrix of a body
q_a : Vector of condensed generalized displacements
T_0: Temperature in reference condition
T_a: Ambient temperature
T_B: Surface temperature
u: Oil film velocity in x direction
u_{Shell}: Circumferential velocity of bearing shell
$u_{Journal}$: Circumferential velocity of journal shaft
x: x direction
z: Axial position at each point of bearing
α_x and α_y: Miss aliment in x and y direction
β: Angular position at each point of bearing
δ: Radial deformation of bearing surface
η: Dynamic viscosity
θ: Oil fill ratio
$\bar{\tau}_{Asperity}$: Shear stress in surface roughness
ρ_0: Density in reference condition

1. Introduction

The market competition between engine manufacturers and component suppliers has led to demand for higher bearing lifetime and lower bearing failure. Therefore, an accurate and exact analysis method of lube oil film performance of the engine bearings seems to be vital.

Effect of elastic deformation of bearings on the performance of crankshaft bearings has been studied by many researchers ([1]-[17]), which shows it was a key factor in the analysis of these bearings. Elasto-hydro-dynamic (EHD) analysis method is a powerful tool for studying oil film lubrication performance in the engine bearings. In this analysis, the stiffness of bearing shell and housing as well as the special geometrical properties of the bearing shell and journal, such as bearing profile, special pressure boundary condition, and oil supply can be considered in simulation. Furthermore, the effect of bearing shell profile due to bearing assembly and/or manufacturing issues can be accounted in an EHD analysis ([1] -[3]). In this type of analysis, beside the orbital path of journal relative to bearing shell, the oil film pressure variations, which is emanated by the applied bearing forces, lead to bearing deformation and change its clearance. On the other hand, the clearance variations lead to change in the oil film pressure distribution and this iterative analysis extends the oil film pressure distribution over a larger surface of bearing.

In thermo-elasto-hydrodynamic (TEHD) analysis, effects of the local oil film temperature variations are also taken into account. This method has been used by some researchers ([4][8]) in the oil film performance analysis of BE bearings in spark ignition (SI) engines. A numerical method for the transient TEHD analysis of oil film in BE bearing of an SI engine was presented in [6]. In [5], the results of TEHD analysis of a connecting rod BE bearing were compared and validated with the experimental data for the conditions of low working temperature as well as the bearing radial clearance of higher than 50 μ.

In some of the previous works ([3], [4] and [7]), an EHD simulation model has been performed to predict the frictional losses in journal bearings. In a recent work [9], this simulation model was extended to a TEHD simulation to consider the effect of local temperatures on the bearing. Detailed comparisons of the results between these models showed that, to have full film lubricated and even slightly mixed lubricated operating conditions, the inclusion of local temperature results in rather small corrections to the properties should be studied.

A study including the thermal effect on the bearing and/or crankpin, and the elastic strain of the bearing due to the pressure field was presented in [4]. A numerical study was presented for thermo-elasto-hydrodynamic comportment of the rod bearing, subject to dynamical loading. It was found that, for the studied cases, the TEHD modeling did not bring much higher precision than isothermal EHD modeling, like mentioned in the study one year ago in [10], which was in automotive engine with four cylinders in line. In [11], a thermo-elasto-hydrodynamic study was presented, in which the influences of the boundary conditions were analyzed. During the same time, in [12], an experimental study was performed for the heating effects on the connecting rod bearings. In the same context and in the same year, a TEHD study was performed [13], which showed the influences of the heating and mechanical effects on the behavior of a big end journal of the diesel engine Ruston-Hornsby 6

VEB Mk-III, whose study was undertaken before by [14] and thereafter by [15]. Considering TEHD analysis is more recommended in the engines working under severe conditions for predicting the performances of bearings in internal combustion engines.

In this paper, at first, the lube oil film analysis of connecting rod BE bearing of a heavy duty diesel engine is performed by EHD method. Then, the convective heat transfer coefficient between lubricant and bearing shell and also journal is calculated by EHD analysis results and used as an input boundary condition in TEHD analysis. Here, the oil film viscosity is considered a function of oil's local temperature and pressure. The effects of flexibility of bearing shell and connecting rod structure are also considered. The results of EHD and TEHD analyses are compared and the main characteristic parameters in bearing design are reported and interpreted thoroughly. More details about the proposed method are presented below. AVL\Excite software [16] is a powerful tool in bearing analysis that is employed for EHD and TEHD analyses of connecting rod BE bearing in the current work.

2. Theory of TEHD analysis of oil film

In EHD analysis, the effect of elastic displacements of the bearing shells and its housing has to be included. The oil film thickness (h) is expressed by Eq. (2) in the consideration of the initial geometrical clearance, misalignment of journal, and elastic deformation of bearing structure.

$$h(\beta,z) = C_R - \left(e_x + \alpha_y Z\right)\cos\beta + \\ -\left(e_y + \alpha_x Z\right)\sin\beta + \delta(\beta,z) \tag{1}$$

It should be noted that parameter δ is the radial deformation of the bearing surface obtained from the nodal displacements of the bearing surface along the radial and circumferential axes. The nodal displacements of the bearing surface are determined by solving the equations of motion (Eq. (2)) for the condensed FE model of bearing structure as follows:

$$\bar{M}.\ddot{q}_a + \bar{D}.\dot{q}_a + \bar{K}.q_a = \bar{f}^{*} + \bar{f}^{a} + \\ + \bar{f}^{gyros} - \bar{f}^{rbAcc} \tag{2}$$

EHD method is based on solving the Reynolds equation (3) [16] in bearing surface. Equation (3) includes a mass conserving cavitation model, which is reflected by the additional variable called clearance fill ratio (θ). For θ=1, the equation will become the ordinary Reynolds equation. 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. It is defined as the fraction of volume filled with oil to the total volume. Fill factor equal to one indicates that the gap is fully filled with oil and zero indicates a completely empty gap.

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

In TEHD analysis, the oil viscosity depends on oil's local temperature and pressure. In this method, the energy equation is also solved as well as Reynolds equation. The energy equation of the oil film considers heat convection in all three directions and heat conduction in the radial direction [16]. In addition to oil film compression, viscous heating and friction power loss, due to surface roughness contact, are considered in the calculations. With considering all of the above terms, the oil film equation is formulated as follows [16]:

$$\bar{\rho}.c_p.\left\{\begin{aligned}&\frac{\partial\bar{T}}{\partial t} + \bar{u}.\frac{\partial\bar{T}}{\partial\bar{x}} + \bar{w}.\frac{\partial\bar{T}}{\partial z} + \frac{1}{\bar{\bar{h}}}.\\ &.\left[\bar{v} - \bar{y}.\left(\frac{\partial\bar{h}}{\partial t} + \bar{u}.\frac{\partial\bar{h}}{\partial\bar{x}} + \bar{w}.\frac{\partial\bar{h}}{\partial z}\right)\right].\frac{\partial\bar{T}}{\partial\bar{y}}\end{aligned}\right\} + \\ + \frac{\bar{T}}{\rho}.\frac{\partial\bar{\rho}}{\partial\bar{T}}\Big|_P.\left(\frac{\partial\bar{p}}{\partial t} + \bar{u}.\frac{\partial\bar{p}}{\partial\bar{x}} + \bar{w}.\frac{\partial\bar{p}}{\partial z}\right) - \frac{\kappa}{\bar{\bar{h}}^2}.\frac{\partial^2\bar{T}}{\partial\bar{y}^2} = \\ = \frac{\bar{\eta}}{\bar{\bar{h}}^2}.\left[\left(\frac{\partial\bar{u}}{\partial\bar{y}}\right)^2 + \left(\frac{\partial\bar{w}}{\partial\bar{y}}\right)^2\right] + \frac{\bar{\tau}_{Asperity}}{\bar{\bar{h}}^3}.\left(u_{Shell} - u_{Journal}\right) \tag{4}$$

To do thermal analysis of the bearing housing, temperature continuity, heat flow, and Biot boundary conditions in internal surface are used according to Eq. (5). Journal temperature is assumed to be constant as an input to the analysis.

$$(\frac{\partial T_B}{\partial n}) + B_i(T_B - T_a) = 0 \tag{5}$$

3. Boundary conditions

3. 1. Thermal boundary conditions

Depending on the thermal analysis type of bearing structure, which has to be chosen in software, there are two choices for thermal boundary conditions in TEHD analysis as follows:

1. Temperature boundary condition when bearing housing is omitted in thermal analysis (see Fig. **1**).
2. Temperature boundary condition when bearing housing is considered in thermal analysis. In this condition, the temperature variations in bearing shell and its housing are calculated (see Fig. 2).

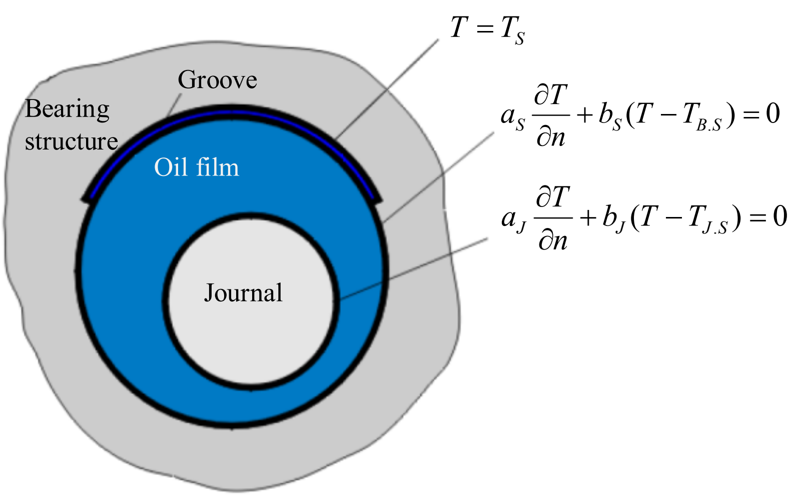


Fig. 1. Thermal boundary condition number 1 [16].

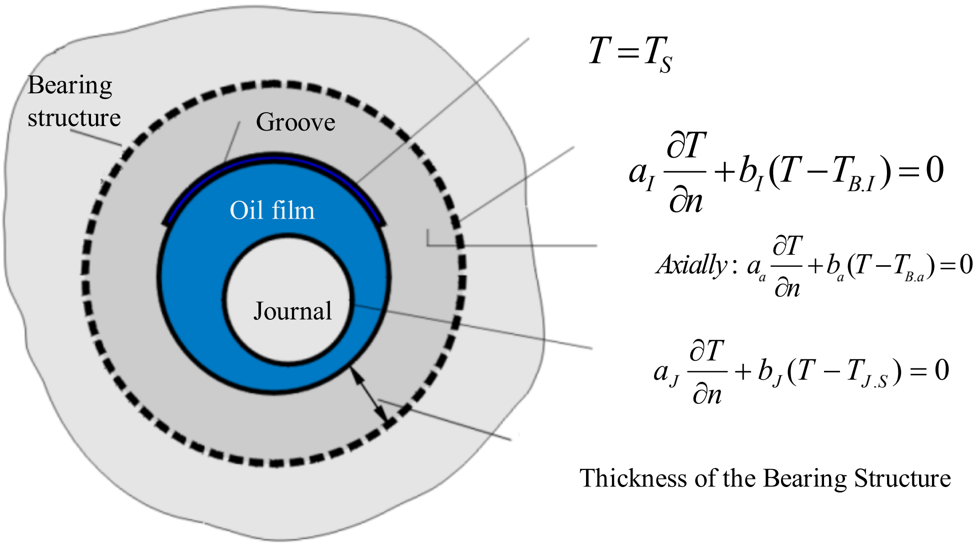


Fig. 2. Thermal boundary condition number 2 [16].

3. 2. Initial boundary conditions

Initial boundary conditions are as follows:

1. Oil pressure at bearing inlet =4 bar
2. Oil pressure at bearing leakage (ambient pressure)=1 bar
3. Inlet oil temperature= 90°C
4. Temperature of solid housing= 90°C
5. No slip boundary condition is considered at the interface of fluid and structures (both of the shaft and bearing shell)

4. Oil film viscosity

In TEHD joint, oil viscosity is considered as a function of oil film temperature. In order to improve the accuracy of results, the oil viscosity can be defined as a function of oil film pressure and shear stress rate. In terms of fluid density, it is also better to be considered as a function of oil film temperature and pressure in the calculations.

In this calculation, the oil viscosity is considered as a function of oil local temperature and pressure, based on Broderbund's equation [16]:

$$\eta(\rho,T) = A.\exp\left(\frac{B}{T+C}\left(1+\frac{p}{F}\right)^{D+E.\frac{B}{T+C}}\right) \quad (6)$$

The dependence of oil density on both temperature and pressure is regarded using Dowson-Higginson's equation [17]:

$$\rho(p,T) = \rho_0.\left(1+\frac{f_1.p}{1+f_2.p}\right).[1-f_3.(T-T_0)] \quad (7)$$

5. EHD analysis of the oil film

5. 1. Simulation

In order to perform the EHD analysis of BE bearing, the transient finite element analysis of connecting rod has to be done. The dynamic analysis has to be carried out to calculate the applied forces to BE bearing. Also, the deformation measure of the bearing shell can be obtained from finite element analysis of the connecting rod.

Transient dynamic analysis of complete finite element model of the connecting rod is too time-consuming. To decrease the computational time of this analysis, one method known as sub-structuring method is used, in which the mass and stiffness matrices of structure are reduced to a condensed form. In this way, degree of freedom (DOF) of the structure is reduced significantly. When DOF of structure is decreased, then the computational time of analysis is significantly decreased.

In sub-structuring method, all nodes which contact with other components and also the ones to which the external loads have to be retained in the FE model and the others can be removed. Thus, for connecting rod analysis using sub-structuring method, the BE and SE (small end) nodes have to be retained in the FE model, because they are in contact with crank pin and piston pin, respectively.

Finite element analysis of the connecting rod using sub-structuring method is carried out in ABAQUS software. Due to high oil pressure gradient in bearing edges, finer mesh is used around these areas.

The single crank train including a connecting rod, crank pin, and piston pin is modeled in order to carry out EHD analysis of oil film in BE bearing.

5. 2. Results of EHD analysis of oil film

Effect of the variations of local oil temperature is neglected in EHD method. Maximum oil film hydrodynamic pressure of the BE bearing of connecting rod versus crank angle is shown in Fig. 3. The maximum oil film pressure occurs at 371 degrees of crank angle equal to 11 degrees after firing TDC (top dead center of piston) and is about 115 MPa. This point is close to the crank angle, at which the maximum gas pressure takes place inside the combustion chamber. In this situation, the connecting rod is placed under compression and, consequently, the maximum pressure in the bearing occurs in the upper shell.

The simulation is run for many engine working cycles and it is observed that the steady state conditions are achieved in the second cycle. So, the presented results are extracted from the second cycle.

As shown in Fig. 4, the minimum oil film thickness occurs at 17 degrees after firing TDC and it is about two µ. The situation, in which the minimum oil film thickness takes place, is very close to the place where the maximum oil film hydrodynamic pressure happens, i.e. at the moment of connecting rod compression.

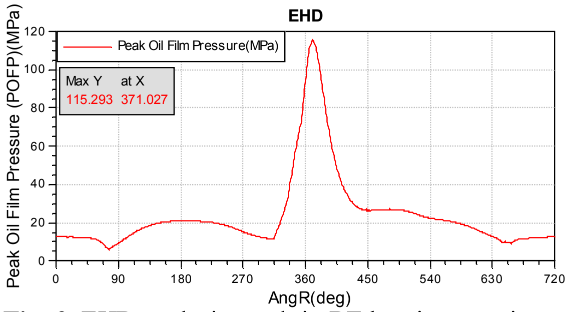


Fig. 3. EHD analysis result in BE bearing, maximum oil film hydrodynamic pressure verses crank angle at 1500 rpm.

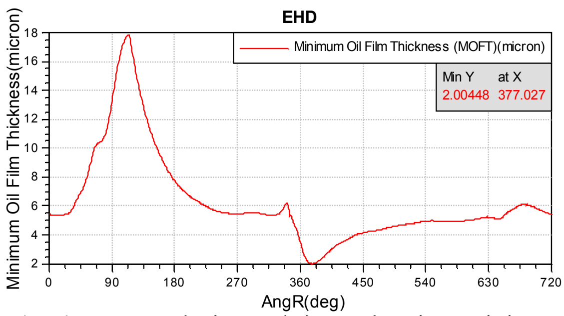


Fig. 4. EHD analysis result in BE bearing, minimum oil film thickness versus crank angle at 1500 rpm.

The bearing shell deformation due to firing loading is shown in Fig. 5. In this figure, the distribution of oil film clearance height due to peak firing loading (connecting rod compression) at 1500 rpm is illustrated. Thus, the clearance between the bearing and shaft can be seen in all situations at the moment of peak firing loading in the range of 2 and 205 µ. The bearing shell deformation due to inertia loading, at 360 degrees after firing TDC, is shown in Fig. 6, when the connecting rod is in the tension. So, clearance between the bearing and shaft is least in the lower shell. Minimum clearance height is in the range of 5 to 235µ. Clearance height distribution in the bearing is affected by bearing shell deformation and connecting rod orbital motion around the crankpin. On the other hand, clearance height distribution affects the axial and circumferential oil film flow velocities. The lubricant flow velocity influences the convective heat transfer rate between oil flow and shaft and also between oil flow and bearing surface.

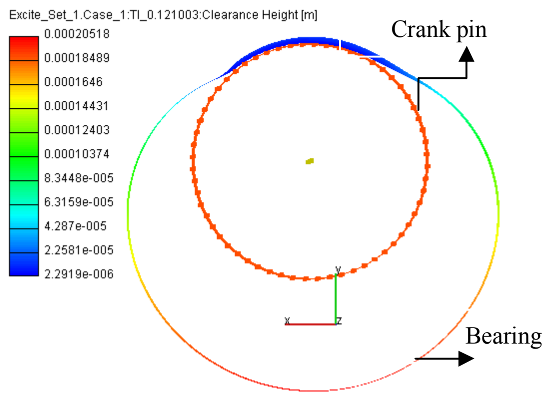


Fig. 5. EHD analysis result in BE bearing, the distribution of oil film clearance height due to peak firing loading at 1500 rpm (10 degrees after TDC).

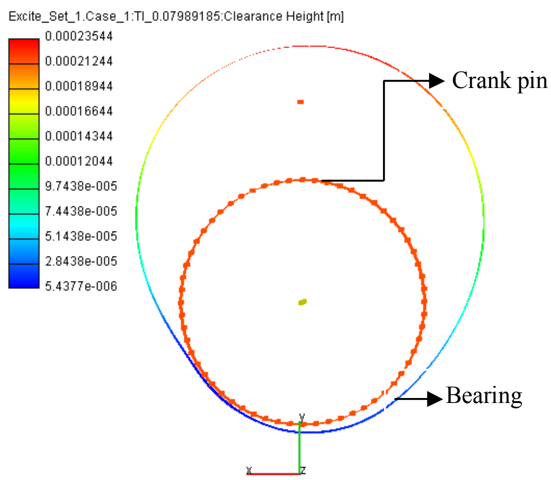


Fig. 6. EHD analysis result in BE bearing, the distribution of oil film clearance height due to inertia loading at 1500 rpm (at the time of exhaust TDC).

As shown in Fig. 7, due to more deformation of bearing shells at its edges, the minimum oil film thickness usually takes place at bearing edges, whereas oil film thickness increases along bearing center. In Fig. 8, hydrodynamic pressure distribution of oil film at 10 and 360 degrees after firing TDC is shown.

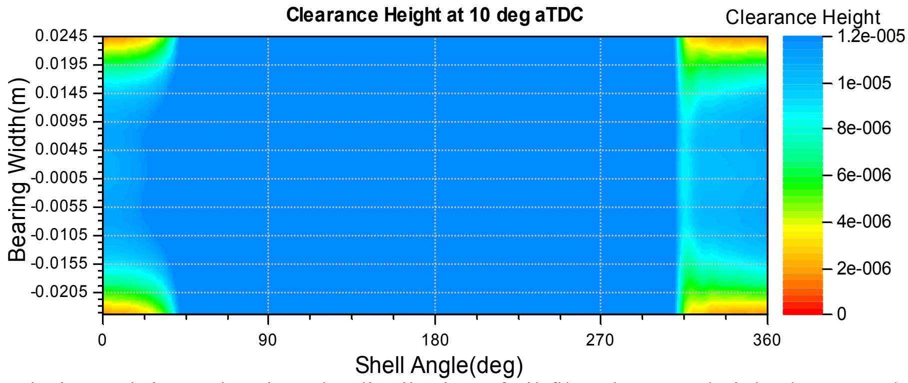


Fig. 7. EHD analysis result in BE bearing, the distribution of oil film clearance height due to peak firing loading at 1500 rpm (10 degrees after TDC).

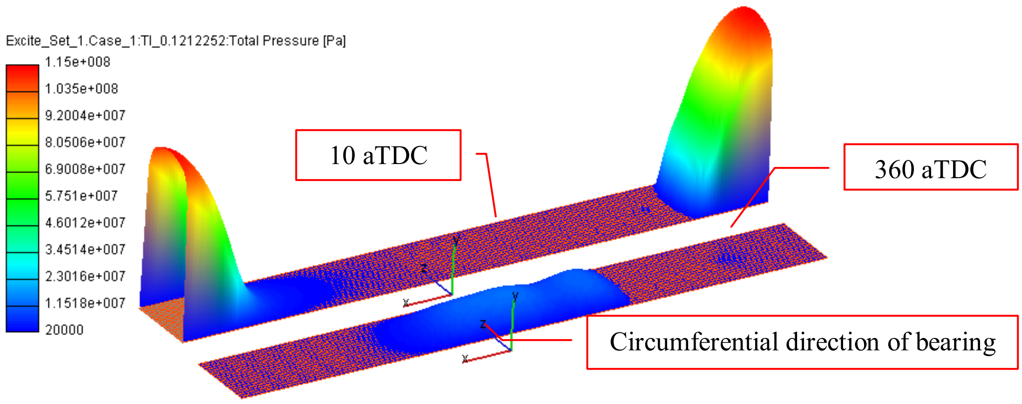


Fig. 8. EHD analysis result in BE bearing, the distribution of oil film hydrodynamic pressure at 10 and 360 degrees after firing TDC, at 1500 rpm.

6. TEHD analysis of the oil film

6. 1. Simulation

TEHD and EHD analyses have similar bases. The major difference between these analyses is heat transfer analysis of oil film and bearing housing. Therefore, the boundary condition of oil film heat transfer with bearing shell and shaft has to be defined.

As for the calculation of convective heat transfer coefficient of lubricant, it is better to utilize the average of axial and circumferential volume flow, as shown in Fig. 9 and Fig. 10, respectively. These 2D-contours are extracted from previous EHD analyses. It should be noted that the AVL-Excite software takes only one value as heat transfer coefficient; therefore, it is essential to consider the average value of both of the axial and circumferential flows in the calculation of heat transfer coefficient.

The convective heat transfer coefficient between oil film and bearing shaft is considered 25000 W/m^2K. Because of high lubricant velocity, the convective heat transfer coefficient of the critical regions is in the range of 20000 to 30000 (W/m^2 K) and its exact amount can be calculated using CFD analysis. The convective heat transfer coefficient in bearing housing is assumed to be 10000 W/m2K.

It should be noted that, in TEHD analysis, some engine working cycles have to be passed until the bearing structure temperature becomes stable. So, the analysis is performed in four working cycles:

- The first two cycles for temperature stability in the bearing structure
- The third cycle for removing transient effects and stabilization of fill ratio
- The forth cycle for getting final analysis and expected results.

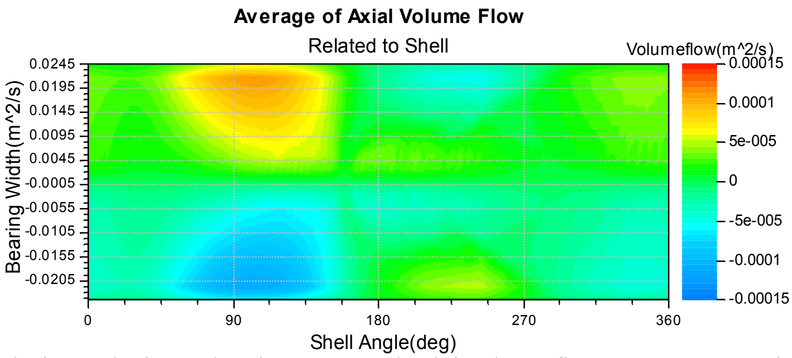


Fig. 9. EHD analysis results in BE bearing, averaged axial volume flow over one operating engine cycle.

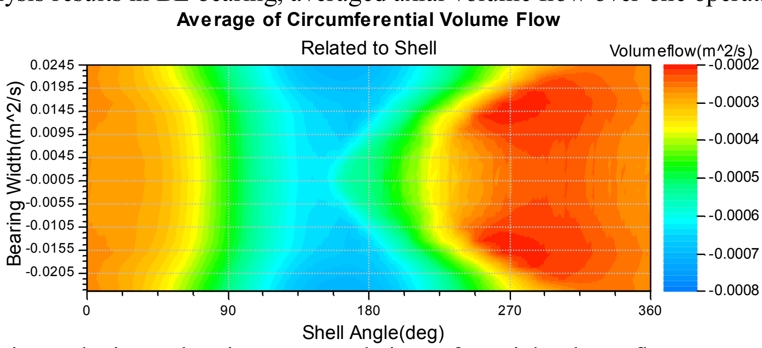


Fig. 10. EHD analysis results in BE bearing, averaged circumferential volume flow over one operating engine.

Therefore, the presented results are related to the forth engine's working cycle.

6. 2. Results of EHD Analysis of oil film

Generally, the results show that there is no considerable difference between the results of EHD and TEHD analyses; but, the required time of TEHD analysis is much more than that of EHD analysis.

Variations of averaged and maximum oil film temperature in terms of crank angle are shown in Fig. 11. As can be seen in this figure, the mean temperature variation is low and negligible in one engine loading cycle. In TEHD analysis, the effect of temperature rise is local and just has an effect on the minimum oil film thickness and maximum oil film pressure, both of which are negligible.

The results of EHD and TEHD analyses such as variations of maximum oil film pressure and minimum oil film thickness are compared in Fig. 12 and Fig. 13, respectively. Comparison shows that TEHD analysis estimates slightly higher maximum oil film pressure and lower minimum oil film thicknesses, because there is a relation between minimum oil film thickness and maximum oil film pressure. The higher the oil film pressure, that the thinner the oil film thickness would be.

The minimum oil film thickness takes place at the crank angle near maximum in-cylinder gas pressure position. The applied loads proliferate in this region and cause high thermal load. So, oil viscosity is reduced in the position of minimum oil film thickness. If oil film viscosity is decreased, the minimum oil film thickness will decrease. Also, position of maximum oil film temperature is in accordance with the position of minimum oil film thickness. The minimum oil film thickness and asperity contact often take place in the same position; so, the asperity contact causes heat generation and, therefore, oil film temperature to increase.

In Fig. 14, friction power loss variation of BE bearing versus crank angle in EHD and TEHD analyses is shown. Friction power loss at 0 and 720 degrees obtained from EHD and TEHD is 1.77 and 1.4, respectively. Total power loss is the sum of hydrodynamic power loss and asperity friction power loss.

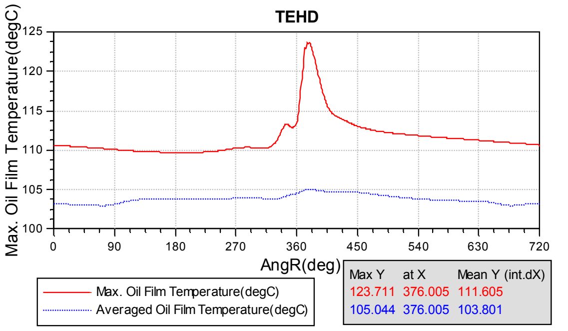


Fig. 11. TEHD analysis results in BE bearing, averaged and maximum oil film temperature variations versus crank angle.

In Eq. (4), the last term is related to asperity friction power loss.

Due to the assumed Newton fluid properties, the hydrodynamic friction within a radial slider bearing can be computed via Newton's shear stress equation:

$$\tau_{Stress} = \bar{\eta}.\frac{\partial \bar{u}}{\partial \bar{y}} \tag{8}$$

The above equation considers a product of the oil film viscosity and the derivative of the circumferential oil film velocity field with respect to the clearance gap height direction. The hydrodynamic power loss can be calculated by the following equation.

$$Hydrodynamic\_power\_loss = \omega.\int \tau_{stress} r dA \tag{9}$$

Local oil film viscosity in TEHD analysis is less than EHD analysis (Fig. 15); so, friction power loss is less than the latter. With regard to the mentioned issues, it can be said that the temperature distribution curves of bearing shell are similar to the wear distribution pattern of bearing shell surface. Due to long-time temperature response of bearing structure, bearing shell temperature does not change at different crank angles. In other words, the temperature distribution curve of bearing shell is steady state during one engine operation cycle.

In Fig. 16 and Fig. 17, 3D temperature distribution of the oil film and the bearing shell at 16 degrees after firing TDC (position of maximum oil film temperature) is respectively shown. The maximum oil temperature occurs in the regions with low clearance height, which corresponds to bearing edges and leads to high heat load in this region.

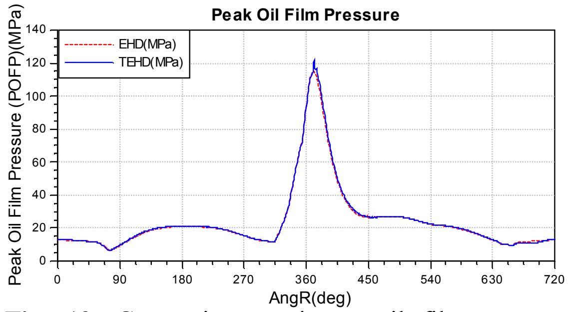


Fig. 12. Comparing maximum oil film pressure variations versus crank angle in EHD and TEHD analyses.

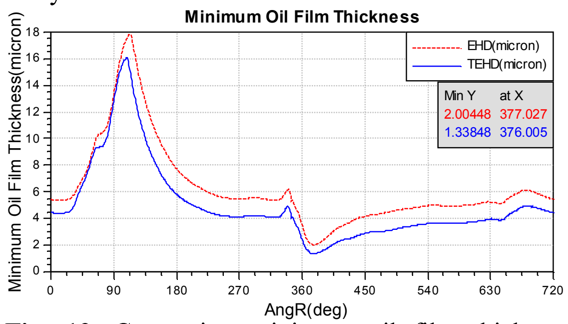


Fig. 13. Comparing minimum oil film thickness variations versus crank angle in EHD and TEHD analyses.

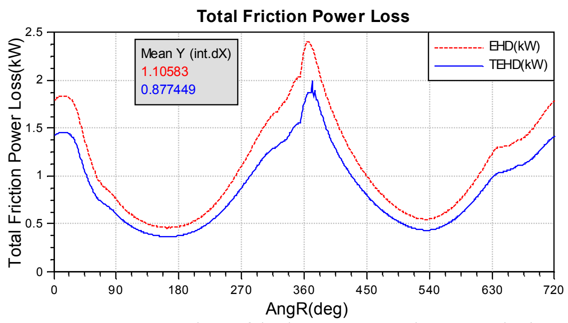


Fig. 14. Comparing friction power loss variations versus crank angle in EHD and TEHD analyses.

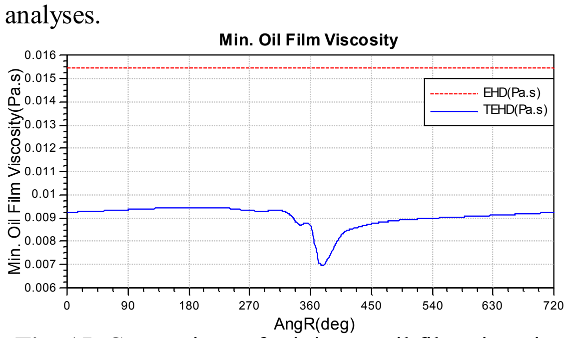


Fig. 15. Comparison of minimum oil film viscosity versus crank angle in EHD and TEHD analysis.

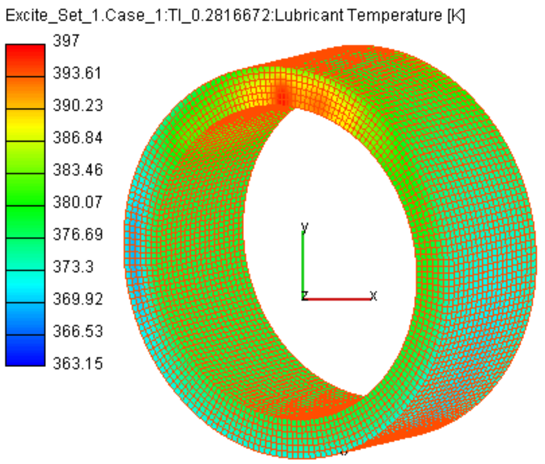


Fig. 16. TEHD analysis results, 3D temperature distribution of the oil film due to peak firing at 1500 rpm.

Temperature of bearing structure, due to high thermal inertia, does not significantly change in one engine operating cycle and remains approximately constant. So, at some crank angles, at which friction power loss is low, the temperature of bearing shell affects the oil film temperature. For example, in Fig. 18, 3D temperature distribution of the oil film and bearing shell is illustrated at 360 degrees after TDC (position of maximum inertia forces). As can be observed, the maximum oil film temperature occurs near the bearing edges.

Normally, it is difficult to measure the BE oil temperature. In this research, in order to measure the BE oil temperature, a special pan is designed and installed on the engine crankcase. The oil exiting from each of the BE bearings is collected by these special pans and its temperature is measured using thermocouples. Comparison between measured and calculation results shows that the error of the mean temperature of the oil film is about 5%. The average temperature of the bearing output oil, which is obtained from the simulation result, is in very good agreement with the results of the experimental measurements.

7. Conclusions

In the current paper, elasto-hydrodynamic and thermo-elasto-hydrodynamic analyses of oil film in connecting rod big end bearing of a

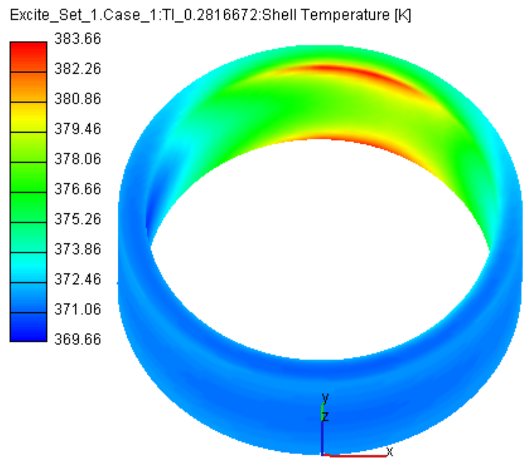


Fig. 17. TEHD analysis results, 3D temperature distribution of bearing shell due to peak firing at 1500 rpm.

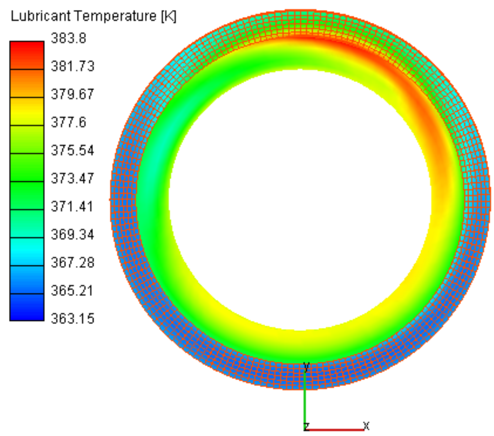


Fig. 18. TEHD analysis results in BE bearing, 3D temperature distribution of the oil film and bearing shell at 1500 rpm, TDC exhaust.

heavy duty diesel engine were carried out. Key parameters of oil film lubrication analysis such as maximum oil film pressure, minimum oil film thickness, and friction power loss were compared in both EHD and TEHD analyses and interpreted completely.

Using the results of EHD analysis, the averaged convective heat transfer coefficient between lubricant and bearing housing was calculated. Then, it was used as an input to TEHD analysis. The results showed no considerable difference between the results of EHD and TEHD analysis, but the required time of the TEHD analysis was much more than that of EHD analysis. This comparison demonstrated that the TEHD method estimated higher maximum oil film pressure and lower minimum oil film thicknesses. The maximum oil temperature occurred in the regions with low clearance height, which corresponded to bearing edges and led to high heat load in this region. Temperature of bearing structure, due to high thermal inertia, did not significantly change in one operation cycle of the engine. So, in some crank angles with low friction power loss, the bearing shell temperature affected the oil film temperature.

References

[1] Hamidreza Chamani, Hadiseh Karimaei, "*analysis of oil film lubrication in main bearings and connecting rod big end bearings of a diesel engine by using elasto-hydrodynamic method*", 6th ICICE, Iran, Tehran, (2009).

[2] Hadiseh Karimaei, Hamidreza Chamani, "*parametric study of oil film lubrication analysis in connecting rod big end bearing of a diesel engine by using elasto-hydrodynamic method*", 6th ICICE, Iran, Tehran, (2009).

[3] Hadiseh Karimaei, Hamidreza Chamani, "*modeling and failure analysis of wear in main bearings and connecting rod big end bearings of a heavy duty diesel engine*", 18th ISME, Iran, Tehran, (2009).

[4] S. Piffeteau, D. Souchet and D. Bonneau, "Influence of thermal and elastic deformation on connecting-rod big end bearing lubricated under dynamic loading", *Journal of Tribology*, Vol. 122, No. 1, pp. 181-191, (2000).

[5] D. Souchet, L. V. Hoang and D. Bonneau, "Thermo-elasto-hydrodynamic lubrication for the connecting rod big-end bearing under dynamic loading", *Proc. Instn Mech. Engrs*, Vol. 218, No. 5, Part J: J. Engineering Tribology, pp. 451-464, (2004).

[6] A. M. A. El-Butch, "Transient Thermo Elasto-Hydrodynamic Lubrication of Connecting Rod Big-End Bearings", *SAE*, No. 2002-01-1731, (2002).

[7] M. Hanahashi, T. Katagiri and Y. Okamoto, "Theoretical Analysis of Engine Bearing Considering Both Elastic Deformation and Oil Film Temperature Distribution", *SAE*, No. 2001-01-1076, (2001).

[8] Y. Okamoto, "Numerical analysis of lubrication in a journal bearing by a thermo-elasto-hydrodynamic lubrication (TEHL) model", *Int. J. Engine Res.,* Vol. 6, No. 2, pp. 95-105, (2005).

[9] H. Allmaier, C. Priestner, F. M. Reich, H. H. Priebsch and F. Novotny-Farkas, "Predicting friction reliably and accurately in journal bearings extending the EHD simulation model to TEHD", *Tribology International*, Vol. 28, No. 1, pp. 20-28, (2013).

[10] T. Garnier, D. Bonneau and C. Grente; "Three-dimensional Ehd behavior of the engine block/crankcrankpin assembly for a four cylinder inline automotive engine", *ASME Journal of Tribology*, Vol. 121, No.1, pp. 721-730, (1999).

[11] D. Souchet, P. Michaud and D. Bonneau, "Big end thermal study, Proc. 15th French congress of mechan-ics", Nancy, France (paper N◦376) :CD, 6 pages, (2001).

[12] L. V. Hoang, D. Bonneau, "Experimental approach of the lubrication of the big end journals under dynamic loading", *Proc. 15Th French congress of mechanics*, Nancy, France (article N◦382), CD, 6 pages, (2001).

[13] J. K. Byung and W. K. Kyung, "Thermo-elasto hydrodynamic analysis of connecting rod bearing in internal combustion engine", *Transaction of the ASME*, Vol. 123, No. 3, pp. 444-454. (2001).

[14] K. P. Oh and P. K. Goenka, "The elasto hydrodynamic solution of journal bearings, under Dynamic Loading", *ASME Journal of Tribology*, Vol. 107, No. 3, pp. 389-395, (1985).

[15] A. Kumar, P. K. Goenka and J. F. Booker, "Modal analysis of élasto hydrodynamic lubrication; a connecting rod application", *ASME Journal of Tribology*, Vol. 112, No. 3, pp. 524-534, (1990).

[16] AVL/Excite power unit theory manual, Version 7.0.3, (2007).

[17] D. Dowson and G. R. Higginson, "Elasto-hydrodynamic Lubrication", Pergamon Press, (1966).