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Crankshaft journals bearings behavior under dilation and strain effects

B. Mansouri*, A. Belarbi, B. Imine***, N. Boualem******

**Laboratory of Aeronautics and Propulsive Systems (LAPS) - University of Sciences and Technology – Oran-Algeria, E-mail: Smail_Mansouri@yahoo.fr*

***Laboratory of Aeronautics and Propulsive Systems (LAPS) - University of Sciences and Technology – Oran-Algeria, E-mail: Belarbi_abd@yahoo.fr*

****Laboratory of Aeronautics and Propulsive Systems (LAPS) - University of Sciences and Technology – Oran-Algeria, E-mail:Imine_b@yahoo.fr*

*****Composite Structures and Innovative Materials Laboratory (LSCMI), University of Sciences and Technology – Oran-Algeria, E-mail:nour_boualem@yahoo.fr*

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1. Introduction

In tribology, the mechanisms impose their constraints and their operating conditions, cinematic, dynamic and thermal to their contacts; in return they can or not support these requests and thus can deteriorate or not the correct operation of these same mechanisms. The interface which is not other than fluid film interposed between surfaces in contact must support these constraints and ensure relative a correct operation between them. The increasingly severe technological requirements of the mechanisms and engines lead to an increase in dissipated energy; the generated temperature can be very high, then the viscosity decreases and thus the journal lift, as well as the materials mechanical characteristics. The thermal and mechanical deformations generated, can be considerable and very consequent at such point as they can compromise the working clearance and consequently the minimal film thickness.

After the analysis of the various bibliographies treating the thermo-elastohydrodynamic phenomena in the bearings, we present the theory used for the resolution of the thermo-electrohydrodynamic (T.E.H.D.) problem.

The effect of elastic deformation of the bearings, on the performance of connecting rod bearings has been studied by many research workers, this shows that is a key factor in the analysis of these bearings; to quote only some, [1]. Note that the effect of dilation of the elements of the journal, compromises the radial clearance under operation. [2] have shown that the variations found between the experimentation and theory were due to the thermoelastic effects, in 1991 [3]. Presented a study including the effect of dilation thermal of the bearing and/or the crankpin, and the elastic strain of the bearing due to the pressure field; In 2000, [4]. Presented numerical study for thermo-elastohydrodynamic comportment of the rod bearing, subject at a dynamical loading, they show that for the studies cases, the T.E.H.D. modeling does not bring much more precision than isothermal electro-hydrodymamic (E.H.D.) modeling, like the study realized one year ago by [5], in automotive engine with four cylinders in line.

Reference [6] presented a thermo-elastohydrodynamic study, where they analysed the influences of the conditions boundary. During the same time [7]. undertake an experimental study for the heating effects on the connecting rod bearings, in the same context and in the same year [8] make a (T.E.H.D.) study and show 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 [9] and thereafter by [10].

The taking into account of T.E.H.D. analysis. Is very recommended in the engines working under severe conditions, this to predict the performances of bearings in internal combustion engines.

2. Governing equations

The schematic diagram of connecting rod bearing is shown in Fig. 1, both of the value and direction of the load applied to the crankshaft vary with time, so the center of the crankpin has periodic motion relative to the connecting rod bearing center. The $X_0Y_0Z_0$ coordinate system, whose origine is fixed at the undeformed bearing center, is used as a reference coordinate system of the analysis. The problem is to find the pressure and temperature fields in the lubricant film, in order to find the most important characteristic in fonctionnement, which is the minimum film thickness.

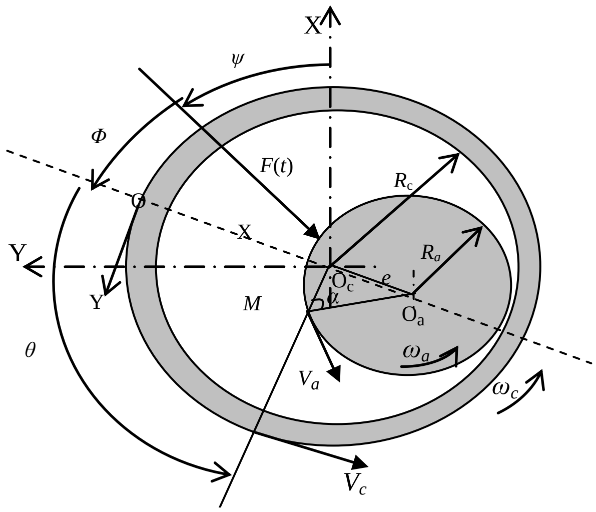

Fig. 1 Schematic diagram

2.1. Reynolds equation in transient state

The Reynolds equation is obtained from the Navier-Stokes equations, in the case of the dynamic mode, the additional data is the variation of the load in module and direction. (Fig. 1).

The dimensionless Reynolds modified equation is written as [1, 4, 5]:

$$\frac{\partial}{\partial\theta}\left(H^3\frac{\partial P}{\partial\theta}\right)+\left(\frac{R}{L}\right)^2\frac{\partial}{\partial Z}\left(H^3\frac{\partial P}{\partial Z}\right)=12\mu\frac{LD}{F\left(\dfrac{C}{R}\right)^2}\left[\left(\dot{\phi}-\overline{\omega}\right)\varepsilon\sin\theta+\dot{\varepsilon}\cos\theta\right], \tag{1}$$

where $\varpi=\dfrac{\omega_a+\omega_c}{2}-\dot{\psi}$ *is the average velocity of the crank-pin, rapported to the load;* $\dot{\psi}$ *is load angular velocity;* $\dot{\phi}$ *is angular Velocity of the centers line relatively at load;* $\dot{\varepsilon}$ *is the crushing velocity* $\left(\dot{\varepsilon}=\dot{e}/C\right)$*;* ω_a*,* ω_c *are angular velocities of the crankshaft and bearing.*

2.2. Mobility method

The second member of the Eq. (1) fact of appearing the two unknown factors of the problem $\dot{\varepsilon}$ and $\dot{\phi}$. The traditional solution is to give two values arbitrary to $\dot{\varepsilon}$ and to $\dot{\phi}$ and to use an iterative method on these two speeds until the hydrodynamic load *W* calculated equal and is opposed to the load applied *F*.

Speeds of the crankshaft centre inside the bearing are determined by writing the equality between the hydrodynamic force in film and the whole of the forces applied to the journal:

$$\vec{F}+\int_s\vec{p}\,dS=\vec{0}\,. \tag{2}$$

Relatively for the axis system lied to the canters line, the Eq. (3) became:

$$\left.\begin{aligned}F_\varepsilon&=F\cos\phi=-\int_{-L/2}^{L/2}\int_{\theta_e}^{\theta_s}P(\theta,Z)\cos\theta\,Rd\theta\,dZ=-W_\varepsilon\\F_\varphi&=F\sin\phi=+\int_{-L/2}^{L/2}\int_{\theta_e}^{\theta_s}P(\theta,Z)\cos\theta\,Rd\theta\,dZ=+W_\phi\end{aligned}\right\} \tag{3}$$

where W_ε and W_ϕ are the components of the hydrodynamic load according to the direction of eccentricity and its perpendicular.

Numerical calculations being very significant, we prefer to use the mobility method [11]. Which allows a fast and precises resolution of the problem.

$$\dot{\varepsilon}=\frac{F(C/R)^2}{\mu LD}M_\varepsilon\ \text{ and }\ \varepsilon\left(\dot{\phi}-\overline{\omega}\right)=\frac{F(C/R)^2}{\mu LD}M_\phi\,. \tag{4}$$

$\vec{M}$ Vector of mobility has as components:

$$\left.\begin{aligned}M_\varepsilon&=+M\cos\alpha\,;\\M_\varphi&=-M\sin\alpha\,,\end{aligned}\right\} \tag{5}$$

where α is the angle between the mobility vector and the eccentricity direction.

The Reynolds equation modified is put in the form, while posing $P=\overline{P}\,M$.

$$\frac{\partial}{\partial\theta}\left(H^3\frac{\partial\overline{P}}{\partial\theta}\right)+\left(\frac{R}{L}\right)^2\frac{\partial}{\partial Z}\left(H^3\frac{\partial\overline{P}}{\partial Z}\right)=12\cos(\theta+\alpha)\,. \tag{6}$$

The second member of Eq. (6) utilizes only one unknown factor, the direction of mobility α; the solution of this equation became simplified. However the determination of α, which for a given eccentricity, is only function of the angle of chock ϕ requires to use a numerical method of interpolation (the relation enters α and ϕ is a priori unknown). Moreover the module of the vector mobility *M* equal to 1 and is re-actualized with the computed value with the step of previous time. The boundary conditions associated with Eq. (6) are those of Reynolds, by taking atmospheric pressure like reference, they are written:

$$\begin{cases}\overline{P}\left(Z=\pm1/2\right)=0 & \forall\ \theta\,;\\\overline{P}\left(\theta=0\right)=\overline{P}\left(\theta=2\pi\right) & \forall\ Z\,;\\\overline{P}=0\ et\ \dfrac{\partial\overline{P}}{\partial\theta}=0 & \text{In film boundary};\\\overline{P}=0 & \text{In cavitation Zone}.\end{cases} \tag{7}$$

The components of the load without dimension are written:

$$\begin{cases}\overline{W}_\varepsilon=\displaystyle\int_{-L/2}^{+L/2}\int_{\theta_e}^{\theta_s}\overline{P}(\theta,Z)\cos\theta\,d\theta\,dZ\\\overline{W}_\phi=\displaystyle\int_{-L/2}^{+L/2}\int_{\theta_e}^{\theta_s}\overline{P}(\theta,Z)\sin\theta\,d\theta\,dZ\end{cases} \tag{8}$$

The module of the load is given by:

$$\overline{W}=\sqrt{\overline{W_\varepsilon}^2+\overline{W_\phi}^2}\,. \tag{9}$$

What makes it possible to calculate the module of the vector mobility and the angle of shock.

$$M=\frac{2}{\overline{\overline{W}}}\quad\text{and}\quad\phi=\tan^{-1}\left[-\frac{\overline{W}_\phi}{\overline{\overline{W}}_\varepsilon}\right]. \tag{10}$$

The dimensioned hydrodynamic load is equal to the load applied *F* and is written:

$$\begin{cases}W_\varepsilon=\dfrac{F}{2}M\overline{W}_\varepsilon\\W_\phi=\dfrac{F}{2}M\overline{W}_\phi\end{cases} \tag{11}$$

Since the eccentricity varies in time, therefore it is necessary to choose an interval of time between two successive points $\Delta t = \left| \dfrac{\Delta\theta_2\left(o\right)}{6\omega_v\left(Tr/mn\right)} \right|$.

2.3. Minimal film thickness

The minimal film thickness of lubricating, without thermal and elastic deformation, is expressed by:

$$h_{min} = C\left(1+\varepsilon\cos\theta\right). \tag{12}$$

To which we must add the deformations due to the fields of pressures and the thermal deformations or dilations.

$$h_{min\,f} = h_{min} + \delta h_p + \delta h_d \tag{13}$$

where δh_p is the elastic strain; $\delta h_d = \delta h_c - \delta h_a$ is the differential dilatation between the crankpin and the bearing.

2.4. Energy equation

The energy equation permitted to calculate the temperature field in the fluid, it translates the energy conservation and permitted to study the thermal transfers in the journal. According to [12] the energy equation is written:

$$\rho C_p\left(\frac{u}{R}\frac{\partial T}{\partial\theta} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z}\right) = K\frac{\partial^2 T}{\partial\theta^2} + \mu\left(\left(\frac{\partial u}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial y}\right)^2\right) \tag{14}$$

Where the first term of the first member represents the convection, the first term of the second member conduction and the second term of the second member, viscous dissipation.

2.5. Heat conduction equation

Our study takes into account the transfer of heat by conduction in the bearing, In order to determine the thermal deformations of the elements of the journal, the temperatures in the solid elements must be known, the equation of heat (15) is:

$$\frac{\partial^2 T_c}{\partial R_c^{\,2}} + \frac{1}{R_c}\frac{\partial T_c}{\partial R_c} + \frac{1}{R_c^{\,2}}\frac{\partial^2 T_c}{\partial\theta^2} + \frac{\partial^2 T_c}{\partial z^2} = 0 \tag{15}$$

where T_c is the temperature in any point of the bearing according to R_c, θ, and z.

2.6. Calculation of final film thickness

Owing to the fact that the crankpin is massive and in rotation, owing to the fact that metals are of good thermal drivers relative to other materials, owing to the fact that the temperature does not have an effect localized like the pressure, but cumulative until thermal stability, we estimate that a calculation of simple dilation at an average temperature of the crankpin is sufficient.

Contrary to the non revolving bearings, the crank pins are constantly in rotation, it is not the same surface which is always under constraint, therefore heat is not localized in the same portion of surface, but distributed according to its structure.

We estimate that the ideal and the finality are that after the established mode, and thus thermal stability, the radial clearance under operation is not compromised and is assured.

2.7. Global heating effect

The determination of the distribution of the temperature in lubricating film like in the solids in contact is done by the resolution of the equation of energy, the equation of Reynolds generalized and also the equation of conduction of heat in the crankpin and the bearing. The resolution of this three-dimensional problem was considered only very recently.

The energy dissipated in the contact is significant and the temperatures in the fluid and contiguous materials with film are raised, it results from it a fall from viscosity and thus a reduction from the bearing pressure of the journal. A simple analysis consists in carrying out a total heat balance and thus determining an average temperature value and viscosity of the lubricant. The average temperature value will be calculated starting from the power dissipated during the cycle, average viscosity being obtained starting from a law of variation of viscosity according to the temperature; generally we neglect the effect of the pressure on viscosity in journal bearings.

The formula of Mac Coull and Walther was retained by the A.S.T.M. It is expressed in the form:

$$Log\left(Log\left(\nu+\alpha\right)\right) = ALog\left(T\right) + B\,, \tag{16}$$

where v is the cinematic viscosity in centistokes, mm²/s; T is the absolute temperature and A, B are the specific constants of the lubricant; the parameter α depends on viscosity v.

$$P = P_1 + P_2 + P_3\,, \tag{17}$$

where

$$P_1 = e\dot{\phi}F\sin\phi \tag{18}$$

represents the power dissipated by load rotation;

$$P_2 = \dot{e}F\cos\phi \tag{19}$$

represents the power dissipated by the film crushing;

$$P_3 = C_a\omega_a + C_c\omega_c \tag{20}$$

represents the power dissipated by the fluid shear.

2.8. Temperature elevation

If it is admitted that the power dissipated in film is evacuated by the lubricant, the rise in the temperature can be written:

$$\Delta T = \frac{P_{moy}}{Q_{moy}\,\rho C_p} \tag{21}$$

with C_p specific heat of the lubricant, ρ density of the lubricant at the supply temperature and Q_{moy} the medium flow.

References [13, 14] propose to approach the maximum value of the temperature by the following empirical relation:

$$T_{max} = T_a + 2\,\Delta T\,. \tag{22}$$

The minimal film thickness will be finally the minimal thickness of the fluid film at which we add displacements due to the pressures (mechanics) and displacements due to the dilatation.

2.9. Mechanical deformation

The connecting rod material is assumed to be isotropic, the stress-strain relationship can be written as:

$$\{\sigma\} = [D]\{\varepsilon\}\,; \tag{23}$$

$$[D] = \frac{E}{(1+\nu)(1-2\nu)} \times \begin{bmatrix} 1 & \nu & \nu & 0 & 0 & 0 \\ \nu & 1 & \nu & 0 & 0 & 0 \\ \nu & \nu & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{1-2\nu}{2} & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1-2\nu}{2} & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{1-2\nu}{2} \end{bmatrix} \tag{24}$$

Stress analysis for the bearing structure, is accomplished by the finite elements software CASTEM 2000, with a pressure of 1Mpa, the meshing used is hexaedric of 8 nodes.

$$\delta h_p = M_f(\theta,z)\,P(\theta,z)\,, \tag{25}$$

where $M_f(\theta,z)$ is the compliance matrix, obtained for a unit pressure of 1 MPa, displacement in each point is obtained by multiplication of the coefficient of the matrix by the corresponding pressure of the node considered, in this way we do not have recourse to each iteration on the pressure to calculate by finite elements displacement [15]. The purpose of this last work [15]. Is to develop as accurately as possible the thermoelastohydrodynamic effect for a bearing working in quasi-static conditions, taking into account the thermal effects as well as the thermal and mechanical deformations. Moreover, experimentations have validated this study regardless of whether it is boundary conditions or assumptions which are being considered. Misaligned and worn bearings have been considered. Experimental data have been obtained for various operating conditions including misalignment (large load, high speed, high misalignment torque).

The calculation of the mechanical deformations is applied only to the bearing, the crankpin relative with the bearing is bulkier and the deformations are negligible.

2.10. Dilation effect

Both of the crankpin and the bearing are dilated.

For the crankpin and the bearing:

$$\delta h_{a,c} = R_{a,c}\alpha_{a,c}\left(\overline{T_{a,c}} - T_0\right), \tag{26}$$

R_a, R_c are respectively the crankpin and the bearing radius; T_a, T_c are respectively the crankpin and bearing average temperatures.

The two effects combined bring to a differential expansion and must be added to the film thickness.

$$\delta h_d = \delta h_c - \delta h_a\,; \tag{27}$$

$$h_{min\,f} = h_{min} + \delta h_p + \delta h_d\,. \tag{28}$$

3. Resolution procedure

To determine the pressure field in lubricating film, finite difference method is applied for the resolution of the modified Reynolds Eq. (7). The associated linear system cannot be solved directly, because of use of the boundary condition of Reynolds (relations (8)), we thus applied the iterative method of Gauss Seidel with sur-relaxation coefficient, the calculation of deformations was done according to laws' of elasticity and is solved by the known software CASTEM 2000.

Displacements are given for a unit pressure of 1 MPa, to have the real displacement of a node it is necessary to multiply the matrix compliance obtained, by the pressure in this node [8, 15].

The reference pressure is the atmospheric pressure; the temperature is determinate by the heat flow in the journal, due to heat transfer in the bearing and the viscous dissipation in lubricant. The temperatures in film and bearing interface are equal (thin film), the thermal transfer in the crankpin is neglected, and we consider for this last, only dilatation between an average and reference temperatures.

The pressures in the sections of entry and of exit of film are equal to the supply pressure, that of the edges to the atmospheric pressure.

For displacements, the bearing being embedded in the big end, radial displacements on the level of the external radius of the bearing are taken as null.

3.1. Simulation results

We have made our calculation in the connecting rod bearing of Ruston and Hornsby 6veb-x MkIII4 Stroke Diesel engine, as the bearing has a full circumferential oil groove, the calculation is performed on a single land of the bearing. The load diagram at 600 rpm is in Fig. 2. Bearing dimensions, material properties and operating conditions are listed in Table. Boundary conditions are Reynolds conditions.

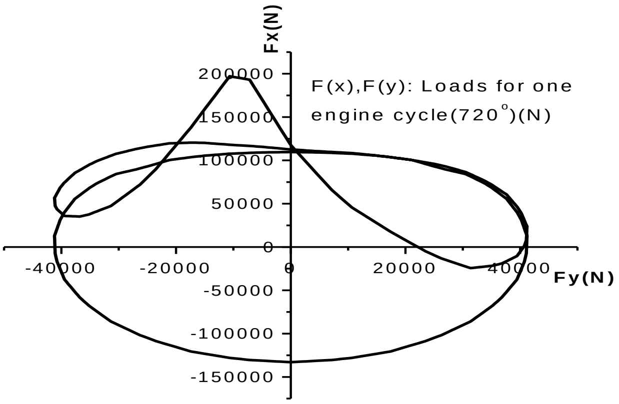

Fig. 2 Load diagram

Table

Journal bearing data and operating conditions

Bearing	Total length of the journal	0.1270 m
	Journal diameter	0.2030 m
	Radial clearance	82.55 µm
Lubricant	Density ρ	850 kg/m^3
	Viscosity at 40°C	0.095 Pa s
	Specific heat C_p	2000 J/kg°C
Bearing structure	Young modulus E	214 GPa
	Poisson's ratio ν	0.25
	Dilation coefficient α	$1.1{\times}10^{-5}$ / °C
	Heat conductivity K_C	50 W/m. °C
Working conditions	Rotational speed ω	600 rpm
	Supply pressure Pa	1.05×105 Pa
	Supply temperature	60°C
	Ambient temperature	60°C

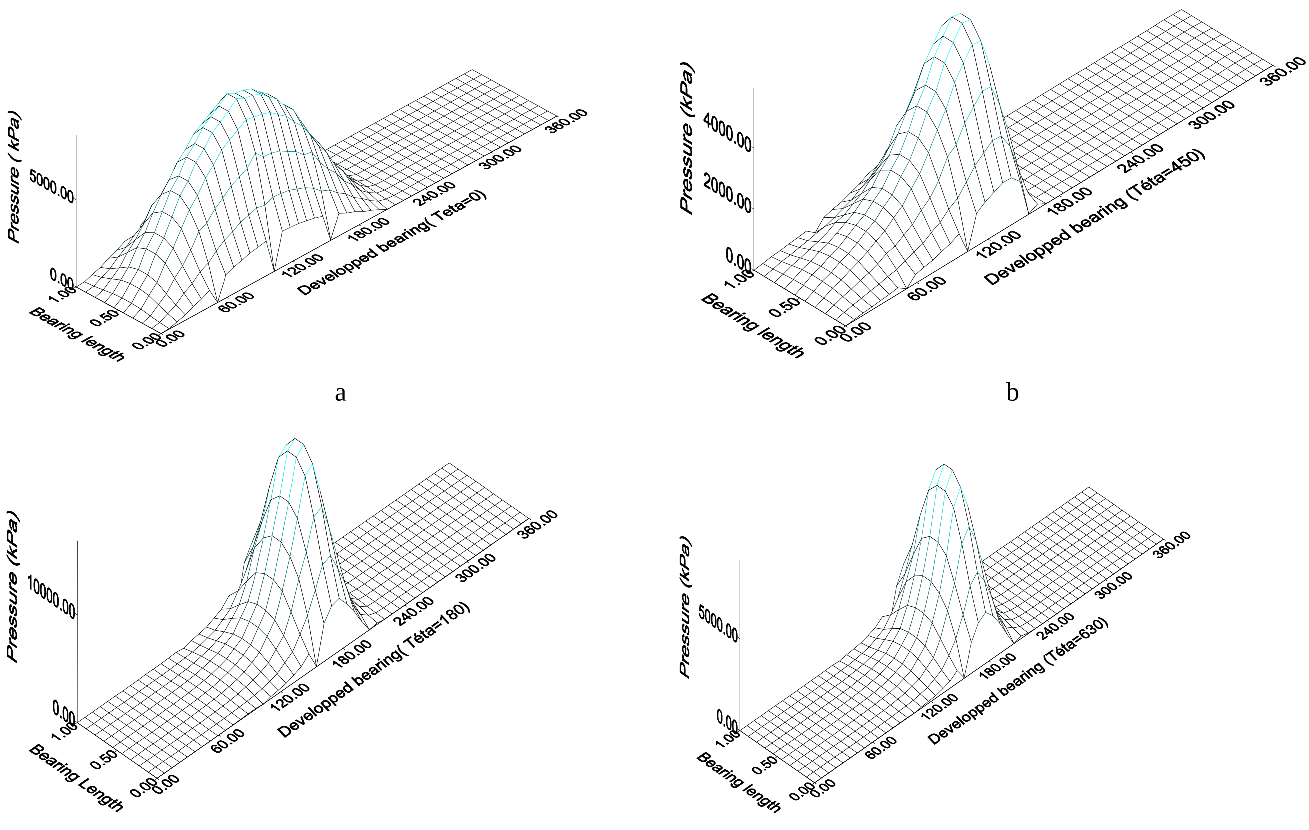

Fig. 3 Distribution pressure: a) crank-angle = 0°); b) crank-angle=450°); c) crank-angle = 180°; d) crank-angle = 630°

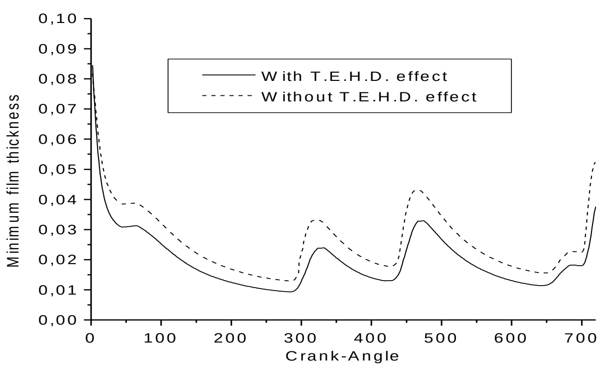

Fig. 4 Minimum film thickness

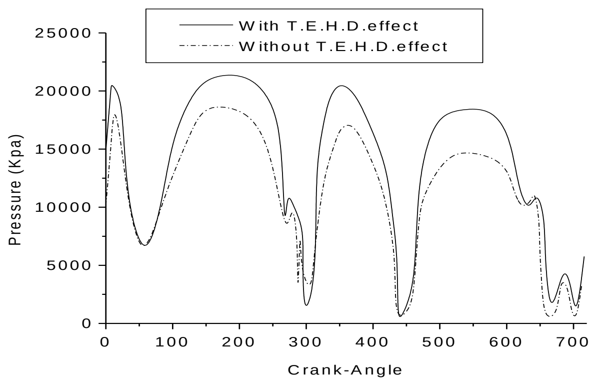

Fig. 5 Maximum film pressure

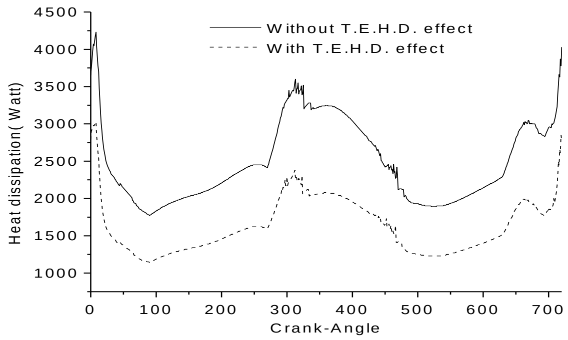

Fig. 6 Heat dissipation(in bearing)

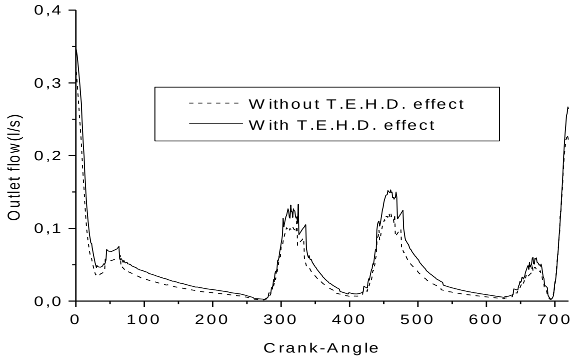

Fig. 7 Outlet flow rate

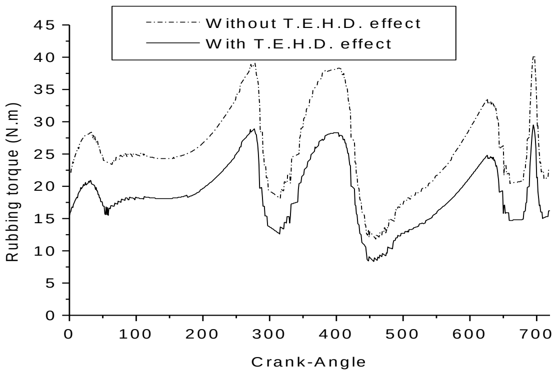

Fig. 8 Rubbing torque

4. Discussion

We remarked in résults, like shown in all figures, that to take the thermo-elastohydrodynamic analysis in consideration, all the bearing characteristics varies. This shows that the effect T.E.H. D. must be taken into account in this kind of contact.

Figs. 3, a-d, shows the three dimensional pressure distributions for four crankshaft angles and show that the pressure in bearing differs according to the crankshaft position.

In Fig. 4, we see that the T.E.H.D. effect increases the minimum film thickness; this is due to the bearing elements expansion. In Fig. 5 we note that the T.E.H.D. effect increases the pressure in the film compared to the hydrodynamic effect only. The dissipated power is presented in Fig. 6, indeed the increase in the operating clearance increases the oil quantity and consequently reduces the friction. In Fig. 7, we notice that the leakage flow is not greatly affected by the T.E.H.D. effect, which is interesting since the oil entrant quantity contributes in the lubrication to ensure the minimum required film thickness; and finally in Fig. 8, we see that the T.E.H.D. effect reduces the friction torque, which is well agrees with the previous results i.e. an minimal film thickness, more fluid quantity and low power dissipation.

5. Conclusion

A thermo-elastohydrodynamic lubrication analysis of crankshaft and connecting rod bearings (Dynamic load), is proposed which includes thermal dilatation and elastic deformation of the bearing surface. Simulation results show that the thermal distortion has remarkable effects on the bearing performance such as the minimum film thickness, maximum film pressure and oil flow rate; and among all these characteristics the minimal film thickness is the most important and must be assured, because it characterizes the working clearance.

Mechanical and thermal deformations were found to be very significant in precisely determining the performance of a bearing subjected to severe operating conditions.

It is concluded that the thermo-elasto-hydrodynamic lubrication analysis is very recommanded to predict the performance of crankshaft bearings in internal combustion engines, also the misalignement and the wear will be studied, to predict the sever work conditions of engines actually.

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PLĖTIMOSI IR DEFORMACIJŲ VEIKIAMŲ ALKŪNINIO VELENO KAKLIUKŲ GUOLIŲ FUNKCIONAVIMAS

R e z i u m ė

Pastaruoju metu plačiai, bet atskirai vienas nuo kito buvo nagrinėjami temperatūros ir slėgio sukelti šilumos ir mechaniniai efektai. Tik keletas darbų buvo skirta abiem reiškiniams, kartu vykstantiems alkūniniame velene ir padėvėtuose guoliuose, kuriems keliami labai griežti reikalavimai. Siekiant priartėti prie tikrovės ir sugretinti su kitais darbais, buvo susidomėta tampriu termodinaminiu reiškiniu. Tiriamasis modelis įvertina deformacijas, atsirandančias dėl guolio įtempių, taip pat išsiplėtimą bei klampos sumažėjimą dėl šilumos mainų tarp tepalo ir guolio. Vidaus degimo variklio alkūninio veleno guolių modeliavimo rezultatai rodo, kad šiluminiai poveikiai turi didelę įtaką guolių darbui.

B. Mansouri, A. Belarbi, B. Imine and N. Boualem

CRANKSHAFT JOURNALS BEARINGS BEHAVIOR UNDER DILATION AND STRAIN EFFECTS

S u m m a r y

The heating and mechanics effects due to the temperatures and pressures in bearings were largely studied these last years but separately; few studies are devoted to the two phenomena acting together in dynamic mode, such as the crankshaft and rod bearings; where the requirements are very severe. In order to approach reality and confronting with other studies being interested in the thermo-elastohydrodynamic phenomena (TEHD). The studied model takes into account the deformations due to the pressures field for the bearing as well as the dilation effect and the reduction of the lubricant viscosity, due to the heat transfer effect, between oil and bearing. Simulation results of the crankshaft bearings of in internal combustion engine are presented and show that the thermal distortion has remarkable effects on the bearings performances.

Keywords: compliance matrix, elastic strain, dilation, thermo-elastohydrodynamic, T.E.H.D.

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