# A fast thermoelastohydrodynamic model for dynamically loaded journal bearings behaviour analysis

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ABSTRACT. *An efficient method for thermoelastohydrodynamic (TEHD) journal bearings lubrication analysis is presented. Lubrication film temperature is treated as a time-dependent two-dimensional variable and is averaged over the film thickness. In order to compute the film and solids temperature, a new heat flux conservation algorithm, based on both Finite Element (FEM) – Finite Volume (FVM) methods, is proposed. A key point in this analysis is the consideration of different heat transfer coefficients at film – solids and solids – surroundings boundaries. The Reynolds equation in the film is solved using the FEM discretization. A mass-conserving cavitation algorithm is applied and the effect of viscosity variation with the temperature is taken into account.*

RÉSUMÉ. *Une méthode efficace pour la lubrification thermoélastohydrodynamique des paliers est présentée. La température du film lubrifiant est considérée comme une variable bidimensionnelle, dépendante du temps et moyennée suivant l'épaisseur du film. Un nouvel algorithme de conservation du flux de chaleur basé sur les méthodes des éléments finis (FEM) et des volumes finis (FVM) est proposé pour le calcul de la température dans le film et les solides. Un point important pour cette analyse est le choix des différents coefficients de transfert de flux au niveau des interfaces film/solides et solides/milieu ambiant. La résolution de l'équation de Reynolds dans le film s'appuie sur une discrétisation par éléments finis. Un algorithme de conservation de débit massique est utilisé et la variation de la viscosité avec la température est prise en compte.*

KEYWORDS: *thermoelastohydrodynamic lubrication, Reynolds equation, finite element, bearing, connecting rod.*

MOTS-CLÉS : *lubrification thermoélastohydrodynamique, équation de Reynolds, éléments finis, palier, bielle.*

## 1. Introduction

Thermal effects play an important role in the engine bearing analysis and design. In order to assist designers in the effort to limit the maximum temperature in bearings, computer codes should include accurate thermal analysis. First, knowledge of oil temperature field is required to determine the oil viscosity which is strongly dependent of temperature. Knowledge of the temperature distributions in the bearings (housing and shaft) is also required to calculate thermal deformations. These deformations may significantly affect the film thickness profile and also the film temperature contours. Moreover, the deformation of the bearing due to hydrodynamic pressure has to be considered in conjunction with the thermal deformation. All these deformations result in a non-uniform effect on the film thickness and affect film thermal field as well as other bearing characteristics (maximum pressure, maximum temperature, ...).

Several thermohydrodynamic (THD) or thermoelastohydrodynamic (TEHD) analyses have been previously developed (Rohde *et al.*1975; Khonsari *et al.*, 1991; Fillon *et al.*, 1990, 1992; Paranjpe *et al.* 1994). Most of them are steady-state analyses. However, engine bearings cannot be considered as working under static load conditions. Few transient THD and TEHD problems were investigated. The first analysis were proposed by Ezzat *et al.*, 1974, Khonsari *et al.* 1992, but they do not include mass conserving cavitation. Paranjpe *et al.* 1995 proposed a transient THD problem of an engine crankshaft bearing, including mass conserving cavitation algorithm. Piffeteau and Souchet, 1999, 2001 studied the TEHD behaviour for connecting-rod bearing in which Reynolds equation is one-dimensional (1D) and the energy equation for the fluid and the heat equation for the solids are two-dimensional (2D). Kim *et al.* 2001 presented also a 2D TEHD study for a con-rod bearing. In this study the film temperature is averaged over the film thickness and the solid temperature does not have a time dependency.

In the present paper an efficient method for thermoelastohydrodynamic (TEHD) lubrication journal bearings analysis is presented. The model is based on FEM and FVM methods. The film temperature is averaged over the film thickness and the solid temperature is three-dimensional. The model is first checked by comparison with a steady state analysis. Then the results are presented for a dynamically loaded, connecting rod big end bearing.

## 2. Elastohydrodynamic governing equations

In this study the usual assumptions of lubrication theory are used. The flow is supposed to be laminar and the inertial effects are neglected in the film. These hypotheses allow the writing of standard Reynolds equation, in the incompressible case:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{6\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{h^3}{6\mu}\frac{\partial p}{\partial y}\right)=U\frac{\partial h}{\partial x}+2\frac{\partial h}{\partial t} \quad [1]$$

This equation can be solved only for the full film zones. For the non-active (cavitation) film zone a second equation must be defined (Bonneau *et al.*, 2001):

$$U\frac{\partial \rho h}{\partial x}+2\frac{\partial \rho h}{\partial t}=0 \quad [2]$$

where $\rho$ is the density of the lubricant - gas mixture. In order to solve simultaneously equation [1] and [2], a universal variable $D$ and a replenishment variable $r$ are defined. If $\rho_0$ is the oil density the latter is given by $r=\rho h/\rho_0$. A generalised Reynolds equation can thus be written (Bonneau *et al.*, 2001):

$$\begin{aligned} &F\frac{\partial}{\partial x}\left(\frac{h^3}{6\mu}\frac{\partial D}{\partial x}\right)+F\frac{\partial}{\partial y}\left(\frac{h^3}{6\mu}\frac{\partial D}{\partial y}\right)= \\ &\qquad =U\frac{\partial h}{\partial x}+2\frac{\partial h}{\partial t}+(1-F)\left(U\frac{\partial D}{\partial x}+2\frac{\partial D}{\partial t}\right) \end{aligned} \quad [3]$$

The universal variable $D$ and the cavitation index $F$ are defined as follows:

- in full (active) zone : $\begin{cases} D=p-p_{cav}, & D\geq 0 \\ F=1 \end{cases}$

[4]

- in cavitated (non-active) zone: $\begin{cases} D=r-h, & D<0 \\ F=0 \end{cases}$

Without misalignment the film thickness equation is given by:

$$h(\theta,y)=h_0(\theta)+h_e(\theta,y)+h_t(\theta,y) \quad [5]$$

where:

$\theta=x/R$ is the angular coordinate for a housing of R radius,

$h_0(\theta)$ is the nominal film thickness, for a rigid bearing $h_0(\theta)=C(1-\varepsilon_x\cos(\theta)-\varepsilon_y\sin(\theta))$,

$h_e(\theta,y)$ is the elastic deformation of the bearing housing and shaft, due to the hydrodynamic pressure,

$h_t(\theta,y)$ is the deformation due to thermal expansion of the bearing housing and shaft.

The lubricant viscosity is assumed to vary only with temperature:

$$\mu(T) = \mu_0 e^{-\beta(T - T_0)} + \mu_a \quad [6]$$

where $\mu_a$ is an asymptotic viscosity, $\mu_0$ is the oil viscosity at $T_0$ and $\beta$ is the thermoviscosity coefficient.

The balance of the applied loads with the hydrodynamic pressures leads to:

$$\begin{cases} \int_S p \cos\theta \, dS - F_x = 0 \\ \int_S p \sin\theta \, dS - F_y = 0 \end{cases} \quad [7]$$

where $F_x$ and $F_y$ are the applied loads acting on the bearing.

The boundary conditions used to solve the modified Reynolds equation are based on the active/non-active film zone separation and have been already detailed by the authors in a previous study (Bonneau *et al.*, 2001).

## 3. Film thermal governing equation

In order to compute the film temperature, a heat flux conservation algorithm, based on FVM is proposed. Lubrication film temperature is treated as a time-dependent, two-dimensional variable, and is supposed to be constant over the film thickness:

For a *i* element of the film mesh (figure 1) with a $T_i$ mean temperature we can write:

$$\sum_{j=1}^{4} \rho C_p q_{ji} T^* + \iiint_e \mu \left[ \left( \frac{\partial u}{\partial y} \right)^2 + \left( \frac{\partial w}{\partial y} \right)^2 \right] dx dy dz - \rho C_p \frac{\partial T_i}{\partial t} - H_h (T_i - T_h) - H_s (T_i - T_s) = 0 \quad [8]$$

with:

– $\rho$, $\mu$ and $C_p$ – density, viscosity and specific heat of the lubricant, respectively,

– $q_{ji}$ – incoming/outcoming oil flow on element boundaries,

– $T^*$ – average film temperature of the flow across the boundary, such as: $T^* = T_j$ if $q_{ji} > 0$; $T^* = T_i$ if $q_{ji} < 0$,

– $T_h$ – housing temperature; $T_s$ – shaft temperature,

– $H_s$ and $H_c$ – exchange coefficients between the film/shaft and film/housing, respectively.

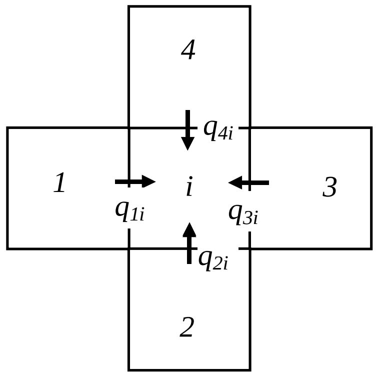


**Figure 1.** *Boundary conditions for one film mesh element*

Equation 8 is not convenient for the supply elements. The lubricant in supply elements is in fact a mixture between the supply oil and film oil and its temperature depends on the supply flow ($q_s$) and on the flow around the supply groove ($q_e$).

$$\text{if } q_s > 0 \text{ then } T_i = \frac{T_{supply} q_s + T^{**} q_e}{q_s + q_e} \tag{9}$$

$$\text{if } q_s < 0 \text{ then } T_i = \frac{T^{**} q_e}{q_e} \tag{10}$$

where $T^{**}$ is the temperature around the supply element and $T_{supply}$ is the oil supply temperature.

## 4. FEM formulation for the EHD problem

The FEM formulation for the EHD problem has been previously described by the authors (Bonneau *et al.*, 2001). However, for a better understanding of the present paper, a short review is given:

– Two different problems must be formulated:

- *Problem 1*: the film thickness is known and the active zone/non-active film zone separation coordinates are sought;

- *Problem 2*: the active and the non-active film zone are known; the film pressure and thickness, which satisfy the Reynolds equation, the equilibrium equations and the elastic equations, are sought.

– Both problems must be solved at each time step. To solve problem 1, the modified Reynolds equation [3] is considered. Problem 2 is solved using the classic

Reynolds equation [1], the equilibrium equations and the relation between the film thickness and the hydrodynamic pressure thanks to compliance matrix.

How the computing of the two problems is similar, only the modified Reynolds equation development is presented here. A linear system of algebraic equations in **D** is obtained:

$$\mathbf{R} = [\mathrm{M}]\mathbf{D} + \mathbf{S} \tag{11}$$

where **R** is the residual vector, **D** is the universal variable and [M] is a *n* rank matrix; *n* is the number of nodes defined for the film domain.

One term of the [M] matrix can be written as:

$$\begin{aligned} M_{jk} = \sum_{n=1}^{ne}\sum_{m=1}^{npg}\Bigg( & \frac{h_m^3}{6\mu}\sum_{k=1}^{nne}\left(\frac{\partial N_{mj}}{\partial x}\frac{\partial N_{mk}}{\partial x}+\frac{\partial N_{mj}}{\partial y}\frac{\partial N_{mk}}{\partial y}\right)F_k - \sum_{k=1}^{nne}\frac{\partial N_{mj}}{\partial x}N_{mk}(1-F_k) \\ & -2\frac{1}{\Delta t}\sum_{k=1}^{nne}N_{mj}N_{mk}\left(1-F_k(t)\right)\Bigg)\Delta\Omega_m \end{aligned} \tag{12}$$

where *m* represents one of the *ngp* Gauss points on *n* element and *nne* the number of nodes per element. $N_{mj}$ is the weight function relative to *j* node while $N_{mk}$ is the interpolation function relative to *k* node. $F_k$ represents the status of *k* node and takes the value 1 if it is in the active zone and 0 in the opposite case.

The vector **S** is the RHS member of modified Reynolds equation. The *j* term of **S** be written as:

can

$$\begin{aligned} S_j = \sum_{n=1}^{ne}\sum_{m=1}^{npg}N_{mj}\Bigg( & U\frac{\partial h_m}{\partial x}+2\frac{h_m(t)-h_m(t-\Delta t)}{\Delta t} \\ & +2\frac{1}{\Delta t}\sum_{k=1}^{nne}N_{mj}N_{mk}\left((1-F_k(t-\Delta t))D_k(t-\Delta t)\right)\Bigg)\Delta\Omega_m \end{aligned} \tag{13}$$

The elastic film thickness at node *j* under the hydrodynamic pressure field is:

$$h_e(j) = \sum_{k=1}^{n}C(j,k)f_k\left(p\right) \tag{14}$$

where [C] is the radial displacement at node *j* due to a unit load *f* at node *k*; *f* is determined by the integration of the pressure field (Annexe 1 - Bonneau *et al.*, 2001).

The system of discrete Reynolds equation and equilibrium equations, non-linear in *p*, $\varepsilon_x$ and $\varepsilon_y$, is solved through the Newton-Raphson method.

## 5. Numerical formulation for the thermal problem

The bearing and shaft temperature fields are three-dimensional. Using the FEM discretization two thermo ([$\mathbf{C_T}$]) and thermo-elastic ([$\mathbf{C_{eT}}$]) compliance matrices are precomputed. Successively, at each element of the housing and shaft surface mesh, a unit heat flux load is applied while the flux remains null on the others and the heat transfer equation, without the transient term, is solved. Thermal field under this unit load gives an elementary thermal solution. Using this thermal field the solid deformations are computed, so an elementary thermo-elastic solution is also obtained.

For a *i* element of the solid surface mesh located on the housing, being at a mean temperature $T_i$, we can write:

$$-k\phi_i = H(T - T_0 - T_i) \qquad [15]$$

where $k$ is the solid thermal conductivity, $\phi_i$ the mean flux passing through the element surface, $H$ a heat transfer coefficient, $T_0$ the reference temperature and $T$ the external temperature.

Equation 15 written for each solid surface element leads to a linear system of algebraic equations in $\phi_i$ and $T_0$:

– if *i* is a ambient/solid surface element

$$\frac{H_{amb}}{k}\left( T_{amb} - T_0 - \sum_{j=1}^{nsurf} C_T(i,j)\phi_j \right) = \phi_i \qquad [16]$$

– if *i* is a supply/solid surface element

$$\frac{H_{supply}}{k}\left( T_{sup\,l} - T_0 - \sum_{j=1}^{nsurf} C_T(i,j)\phi_j \right) = \phi_i \qquad [17]$$

– if *i* is a film/solid surface element

$$\frac{H_{film}}{k}\left( T(i) - T_0 - \sum_{j=1}^{nsurf} C_T(i,j)\phi_j \right) = \phi_i \qquad [18]$$

where $H_{film}$ $H_{amb}$, and $H_{supply}$ are the heat transfer coefficients between solid and the film, ambient medium and the supply oil, respectively; $T_{amb}$, $T(i)$, $T_{supply}$ are the ambient, film and supply oil temperature, respectively.

In order to take into account the heat flux conservation in the solids, a last equation must by written:

$$\sum_{j=1}^{nsurf} \phi_j S_j = 0 \qquad [19]$$

where $S_j$ is the surface of a *j* solid-mesh element.

Finally, the linear system is: [A] **Φ** = **B** [20]

where:

$$[A] = \begin{bmatrix} \frac{H_{amb}}{k}C_t(i,j)-1 & \frac{H_{amb}}{k}C_t(i,j) & \frac{H_{amb}}{k}C_t(i,j) & \frac{H_{amb}}{k} \\ \frac{H_{film}}{k}C_t(i,j) & \frac{H_{film}}{k}C_t(i,j)-1 & \frac{H_{film}}{k}C_t(i,j) & \frac{H_{film}}{k} \\ \frac{H_{supply}}{k}C_t(i,j) & \frac{H_{supply}}{k}C_t(i,j) & \frac{H_{supply}}{k}C_t(i,j)-1 & \frac{H_{supply}}{k} \\ S_j & S_j & S_j & 0 \end{bmatrix}$$

$$\mathbf{B} = \begin{bmatrix} \frac{H_{amb}}{k}T_{amb} \\ \frac{H_{film}}{k}T(i)_{film} \\ \frac{H_{supply}}{k}T_{supply} \\ 0 \end{bmatrix} \qquad \mathbf{\Phi} = \begin{bmatrix} \phi_i \\ \phi_i \\ \phi_i \\ T_0 \end{bmatrix}$$

Using the $\varphi_i$ coefficients, the thermal field and the deformation due to thermal expansion of the bearing housing and shaft can be computed.

The temperature of a *j* solid surface mesh element is given by:

$$T(j) = T_0 + \sum_{k=1}^{nsurf} C_t(j,k)\phi_k \qquad [21]$$

By linear combination of thermo-elastic elementary solution, we get the film thickness term due to the thermal field at *j* film surface mesh node:

$$h_t(j) = \sum_{k=1}^{nsurf} C_{eT}(j,k)\phi_k \qquad [22]$$

In order to compute the film temperature, the equation 8 is developed. For a *e* film-mesh element:

$$
\begin{aligned}
&\sum_{j=1}^{4}\sum_{pg=1}^{2}\rho_0 C_p q_{ji} T^* dS + \sum_{m=1}^{npg}\left(\frac{h_m^3}{12\mu_e}\sum_{k=1}^{nne}\left(\frac{\partial N_{mk}}{\partial x}p_k\frac{\partial N_{mk}}{\partial x}p_k + \frac{\partial N_{mk}}{\partial y}p_k\frac{\partial N_{mk}}{\partial y}p_k\right)\right.\\
&\quad\left. + \frac{U^2\mu_e}{h_m}\right)\Delta S - \sum_{m=1}^{npg}\rho_0 C_p \frac{T_e^t h_m^t - T_e^{t-\Delta t} h_m^{t-\Delta t}}{\Delta t}\Delta S \\
&\quad - \sum_{m=1}^{npg}\left(H_h\left(T_e - T_h\right) + H_s\left(T_e - T_s\right)\right)\Delta S = 0
\end{aligned}
\qquad [23]
$$

where $m$ represents one of the $npg$ Gauss points on $e$ element and $nne$ the number of nodes per element. $N_{mk}$ is the interpolation function relative to $k$ node and evaluated at point $m$.

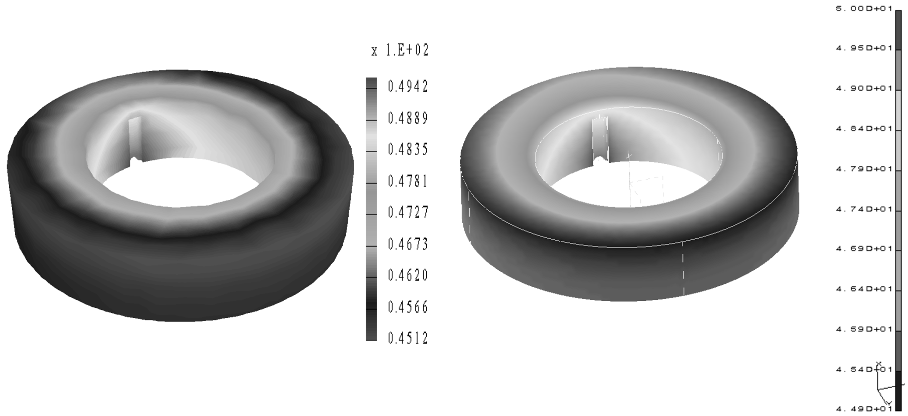


**Figure 2.** *Validation of the solid thermal 3D model (comparison with a commercial software)*

## 6. Numerical algorithm

The numerical algorithm proposed by Bonneau *et al.*, 2001, is adapted for the TEHD problem. Choosing the time step for the transient problems must be directly link with the functioning cycle of the studying mechanism. In this case, for a connecting rod bearing at 6 500 rev/min, the time step is around 0.022 s. Nevertheless, convergence problems can induce a reduction of the initial time step. The convergence criterion is the equality between the applied load and the hydrodynamic pressure at 0.01%.

*input data*
*initialisation*
*do for each time step*
    *do until stability of domain, film thickness and pressure*
        *do until stability of the cavitated area: Problem 1*
            *Compute D (modified Reynolds equation)*

```
            Modify  the cavitated boundaries
        end
        do until R(h,p) > ε (Newton – Raphson method): Problem 2
            Compute R
            Correct the pressure, the eccentricity
            Compute the elastic deformations
            Modify the film thickness
        end
        Compute a new thermal field
    end
    write pressure, film thickness, temperature, etc.
end
```

## 7. Code validation

Mitsui (1987) presented a thermal experimental study for a steady state journal bearing. Comparisons are made with some of his experimental measures. The test bearing has 100 mm in nominal diameter and 70 mm in length. No valid information is given for the solid thermal properties. However, considering that the bush is made in steel, 50 W/(m°C), 7 500 kg/m$^3$ and 400 J/(kg°C) values have been chosen for the housing conductivity, density and specific heat, respectively. The journal temperature is fixed at 49°C. One housing axial groove 60mm in length and 10° in arc angle is used.

### 7.1. *Solid thermal 3D model validation*

In order to validate the thermal 3D model for solids, a comparison is made between results obtained with the presented model and a commercial FEM software (*SDRC – I-DEAS*). The Mitsui bush geometry is chosen. The heat transfer coefficients between the solid and the ambient, film and supply groove oil are 100 W/(m$^2$°C). The ambient temperature is 30°C; the supply oil and film temperature is 100°C.

Figure 2 show a very good correspondence between the two computation results for the minimal and maximal temperature and for the thermal field.

### 7.2. *Complete model validation*

A first comparison with Mitsui bearing is made for 2 500 rev/min rotational speed, 0.78 radial clearance, 98 kPa supply pressure and 5.68 kN applied load. The initial supply temperature is 40.3 °C and the ambient temperature is 27.3 °C. The lubricant characteristics are presented in Table 1.

**Table 1.** *Oil properties – transformer oil*

| | |
|---|---|
| Viscosity [Pa.s] | 0.00736 |
| Specific heat [J/(kg°C)] | 1970 |
| Density [kg/m$^3$] | 862 |

Figure 3 shows the bearing bush thermal field obtained for 10 W/(m$^2$°C) heat transfer coefficient between the housing and the surroundings. The maximum bush temperature is 48.4 °C is in good corresponding with Mitsui experimental measures (48°C – Figure 4). In the same time the isothermal lines are quasi identical, excepting the colder point. In the present case, the colder point is located on the supply groove surface, which seems to be natural. For the Mitsui case the colder point is displaced in the journal rotating direction, which cannot be really explained.

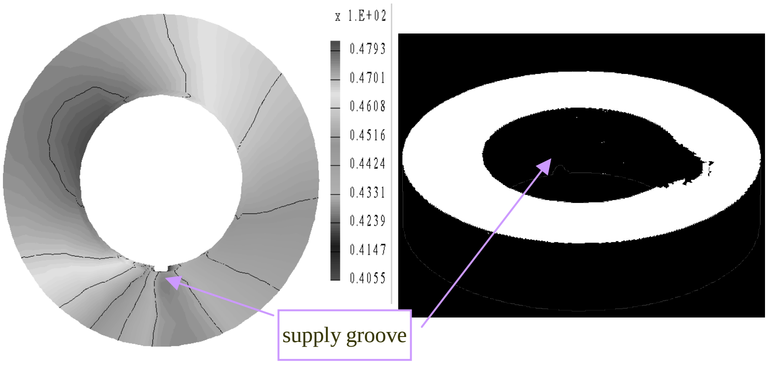


**Figure 3.** *Validation of the solid thermal 3D model (comparison with experimental data)*

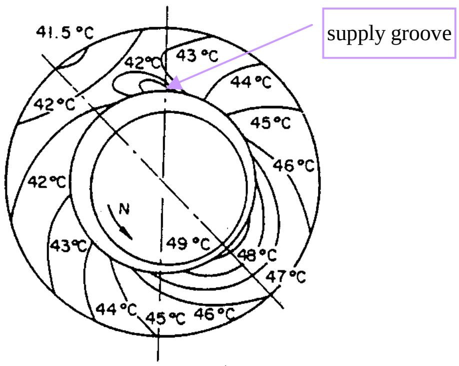


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**Figure 4.** *Mitsui experimental data*

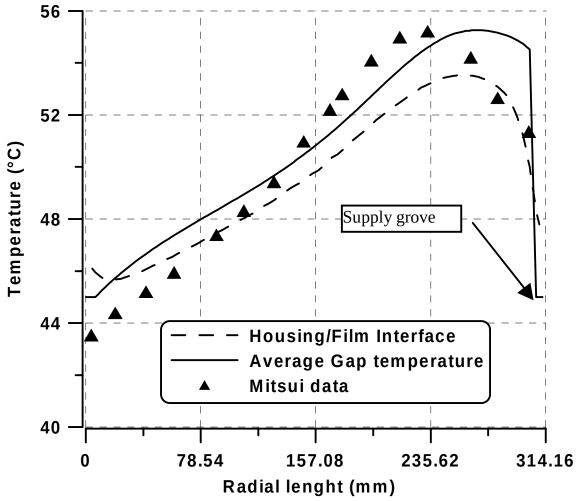


**Figure 5.** *Film temperature field – Comparison with Mitsui's*

**Table 2.** *Oil properties – turbine oil*

| | |
|---|---|
| Viscosity [Pa.s] | 0.0192 |
| Specific heat [J/(kg°C)] | 1950 |
| Density [kg/m$^3$] | 859 |

A second comparison is made for 2 250 rev/min rotational speed, 0.78 radial clearance, 98 kPa supply pressure and 3.92 kN applied load. The initial supply temperature is 40.3°C and the ambient temperature is 22.5°C. The heat transfer coefficient between the housing and the surroundings is 10 W/(m$^2$°C). The journal temperature is fixed at 49°C and 5 000 W/(m$^2$°C) heat transfer coefficient between the film and the solids is chosen.

The lubricant characteristics are presented in Table 2.

Figure 5 shows the average film and the bush film interface temperature variation on the middle-bearing plane. The Mitsui experimental data are also presented. Similar results are noticed. The small difference between the theoretical study and the experimental measures can be explained by the fact that the real solid characteristics (conductivity, density, transfer coefficients, etc.) are not given by Mitsui.

**Table 3.** *Operating conditions and bearing characteristics for a gasoline engine con-rod bearing*

| | |
|---|---|
| Bearing Radius | 24 mm |
| Bearing Axial Length | 19 mm |
| Radial Clearance | 0.015 mm |
| Crank-Shaft Arm-Length | 41.75 mm |
| Connecting-Rod Length | 144 mm |
| Oil Supply Pressure | 0.5 MPa |
| Coefficient of thermal expansion | $12.10^{-6}$ °$C^{-1}$ |
| Thermal conductivity | 50 $W.m^{-1}$°$C^{-1}$ |
| Specific heat | 500 $J.kg^{-1}$.°$C^{-1}$ |
| Density | 7900 $kg.m^{-1}$ |
| Young’s modulus | 200 GPa |
| Poisson’s ratio | 0.3 |

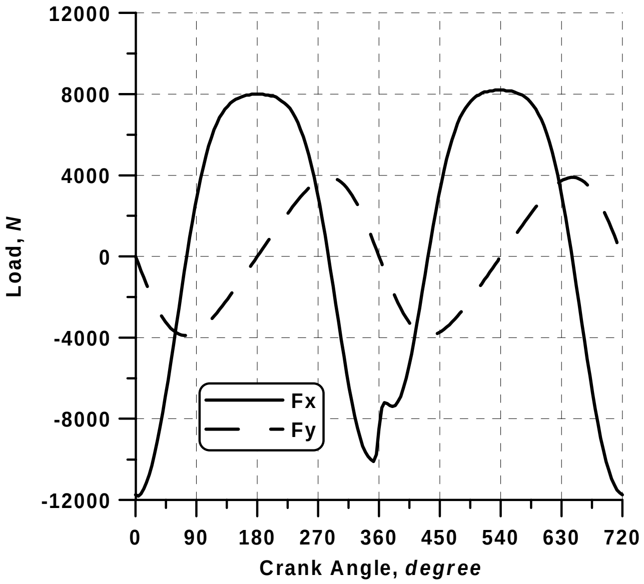


**Figure 6.** *Load diagram for a connecting rod bearing at 6 500 rev/min*

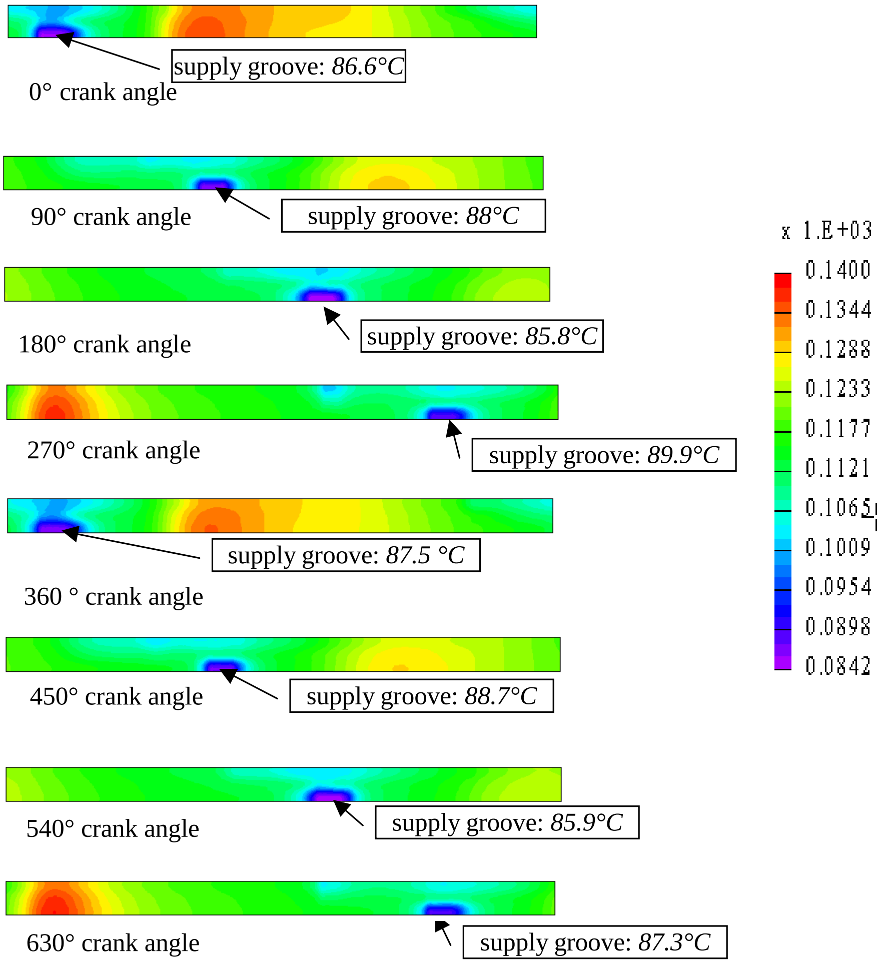


**Figure 7.** *Thermal fields for 0°, 90°, 180°, 270°, 360°,450°, 540° and 630° crank angles*

## 8. Dynamically loaded connecting – rod bearing

A TEHD study is presented for a typical big end con-rod bearing, used in spark ignition engines. The bearing characteristics and the operating conditions are reported in Table 3. Figure 6 represents the load diagram at 6 500 rev/min. Properties of the sample oils used in this study are listed in Table 4. The heat transfer coefficient between the solids and the surroundings is 100 W/($m^2$°C). The heat transfer coefficient between the solids and the oil film is 10000 W/($m^2$°C).

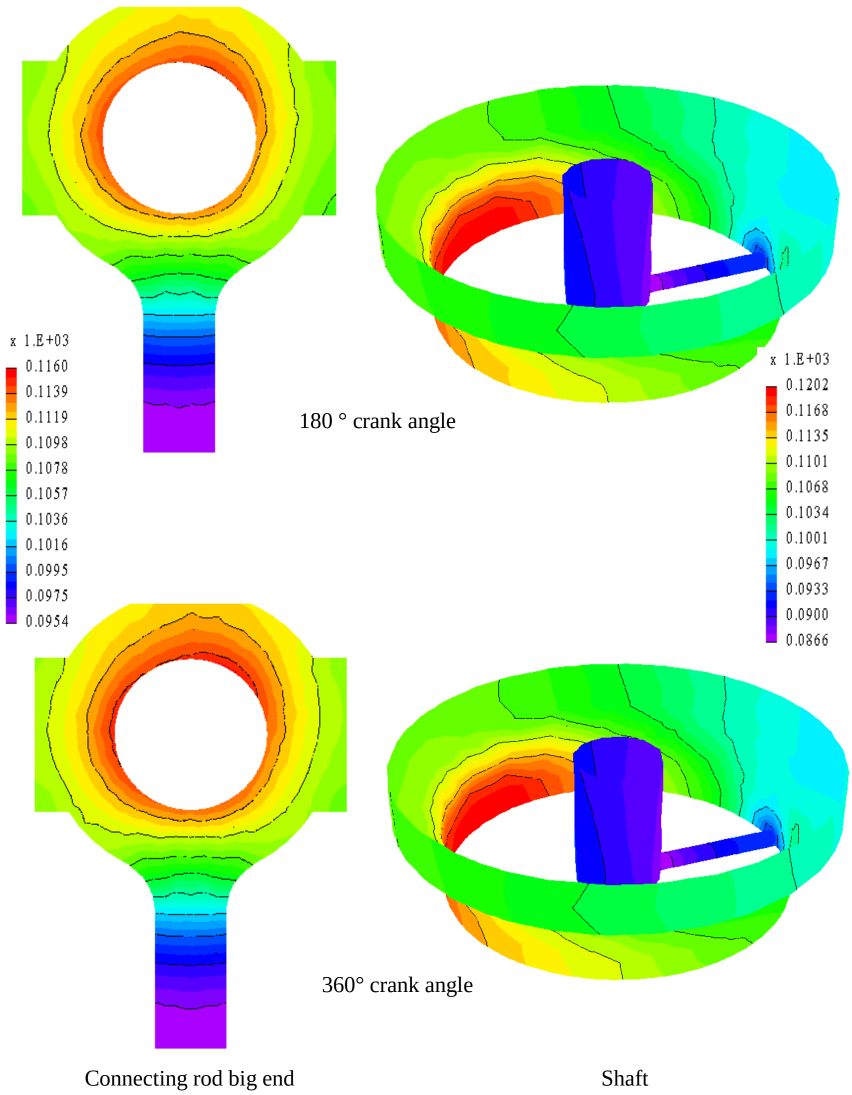


**Figure 8.** *Solid thermal field for 180° and 360° crank angles*

Thermal field for 0°, 90°, 180°, 270°, 360°, 450°, 540° and 630° crank angle are presented in figure 7. It can be observed that the average temperature of the supply groove is around 86°C, even if the initial supply temperature is 80°C. The maximum film temperature, for a complete cycle, is 140°C and the average temperature is 120°C.

**Table 4.** *Oil properties*

| | |
|---|---|
| Oil Viscosity at 20°C | 0.135 Pa.s |
| Asymptotic viscosity | 0.0035 Pa.s |
| Thermoviscosity coefficient | 0.03662 $°C^{-1}$ |
| Density | 860 $kg.m^{-3}$ |
| Thermal conductivity | 0.14 $W.m^{-1}.°C^{-1}$ |
| Specific heat | 2000 $J.kg^{-1}.°C^{-1}$ |

An important parameter is the bearing deformation generated by the thermal field. If the maximum solid deformation due to the pressure field is around 0.01 mm, the thermal field induces 0.026 mm shaft and 0.027 mm housing maximum deformation whereas the minimum film thickness is about 0.002 mm. In fact, if the housing deformations increase the gap, the shaft dilatation will decrease the film thickness. The combination of both induces a reduction of the minimum film thickness of about 0.001 mm which cannot be neglected.

Figure 8 shows the thermal fields for the housing and the shaft at 180° and 360° crank angle.

**Table 5.** *Comparison between EHD and TEHD model*

| | EHD case | TEHD case |
|---|---|---|
| Minimum film thickness | 4.18 µm | 2.1 µm |
| Maximum film pressure | 31.3 MPa | 41.7 MPa |
| Oil Flow | 0.06 l/min | 0.11 l/min |

Table 5 shows the main parameter for both EHD and TEHD cases. Major differences can be observed which underline the importance of the thermal effects in the con-rod bearing behaviour.

## 9. Conclusion

A fast and efficient method for the dynamically loaded journal bearing has been presented. The model has been validated through comparison with previous experimental data. A connecting rod bearing application is presented, in order to demonstrate the algorithm efficiency.

The major difficulty in using this algorithm is to choose the heat transfer coefficients between the film and the solids. These coefficients are supposed to be constant for the whole cycle. Actually they should be a function of the local film and solid temperature.

## 10. References

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