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SIMULATION PROCEDURE BY FINITE ELEMENTS FOR HYDRODYNAMIC LUBRICATION OF SLIDING BEARINGS

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ABSTRACT
The paper presents a simulation procedure by finite elements for hydrodynamic lubrication of sliding bearings. This procedure are based on an finite elements analysis module of thermal distribution assimilated by a pressure distribution in sliding bearing lubricant film and include calculus subroutines for conversion, preparing input data and automatic analysis of output data. This research was realizing bases on the contract no. 2/2005, in CEEX Program.
Keywords: hydrodynamic lubrication**,** sliding bearings**,** MEF**.**

1. INTRODUCTION
We present a simulation procedure by finite elements for hydrodynamic lubrication of sliding bearings. This procedure are based on an finite elements analysis module of thermal distribution assimilated by a pressure distribution in sliding bearing lubricant film and include calculus subroutines for conversion, preparing input data and automatic analysis of output data. These procedures are iterative applied for high precision.
The lubricant film particularities permitted to reduce the specific mathematical model for Newtonian fluids flow in solid space. Reynolds's equation represent the reduced of automaton for lubricant flow in sliding bearing and contains a few specifics simplifying hypothesis [1,2]:
- the lubricant film are very slim comparing the global dimensions of sliding bearing;
- the fluid inertia are negligible;
- the rounding of sliding bearing elements introduces negligible second order mechanical effects.

For stationary work conditions and incompressible lubricant, the Reynolds's equation is [1,2]:

$$\frac{\partial}{R\partial\theta}\left(\frac{h^3}{\eta}\frac{\partial p}{R\partial\theta}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial z}\right)=6R\omega\frac{\partial h}{R\partial\theta}, \tag{1}$$

where:

p = lubricant film pressure;
R = average lubricant film radius;
η = lubricant viscosity;
ω = relative angular speed pin-bearing;
h = lubricant film thickness.

The boundaries limits conditions associated to equation (1) are expressed as pressures on feed hole supply and the bearing ends, generally.
Under isothermal conditions viscosity is constant throughout the lubricant film. This condition presupposes a balance between the quantity of heat generated throughout the lubricant film and the heat absorbed and dissipated by the bearing elements.

2. NUMERICAL SIMULATION WITH FINITE ELEMENTS FOR HYDRODYNAMIC LUBRICATION OF SLIDING BEARINGS

The Reynolds's equation can be seeing like a thermal distribution in 2D space. The similarity of equation (1), including the boundary condition with thermal distribution permitted the numerical simulation with finite elements. We are considered the pressure distribution $p(\theta,z)$ like temperature. Similar, the thermal conductivity coefficients depend on lubricant film thickness and viscosity:

$$k_\theta = \frac{h^3}{\eta} \quad \text{and} \quad k_z = \frac{h^3}{\eta}\,. \qquad (2)$$

The right member of equation (1) is a heat source equivalent. The simulation with 2D finite elements was realized on lubricant film median surface using deltoid element (SHELL) with 6 nodes (suitable for curved surfaces). We are obtained 3245 elements and 6652 nodes which constitute the lubricant film finite elements model.

The finite element model considered only one supply nozzle through which the lubricant is introduced into the bearing at absolute pressure.

For exemplification we present the solution of hydrodynamic lubrication of a sliding bearing with follow dimensions:

- bearing sizes: diameter, D = 100 mm; length, L = 80 mm; radial play, J_0 = 0,2 mm.
- working conditions: stationary, isothermal, steady load space fixed, F = 25 kN,
- lubricant viscosity: η = 0.1 Ns/m^2.

The analysis of slide bearings has been carried out for a series of angular velocity steps of the shaft, ranging between 10 s^{-1} and 500 s^{-1} (Fig. 1), with developments for the rating with angular velocity of 200 s^{-1}. This was considered proper operation.

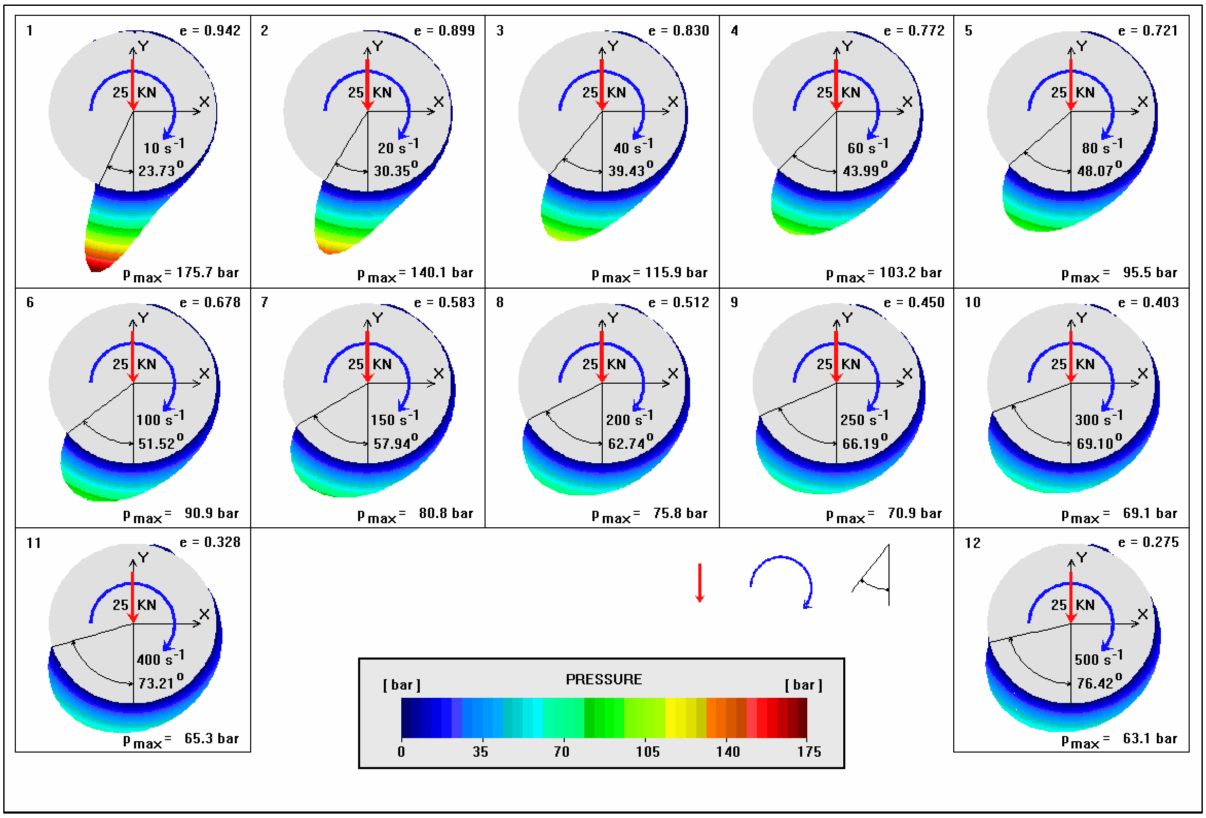


Figure 1. Pin hydrodynamics pressure distribution function of angular velocity of the shaft

Under stationary and isothermal operation, under the prescribed load, the pin occupies within the bearing position determined by eccentricity ε and the positioning angle α.

For each operation mode the solution is obtained iteratively, controlling the two parameters ε and α to reach the foisting bearing capacity with a tolerance f±0,5%.
The solved procedure is incorporated in an iteration loop, including 3 steps:

- INPUT DATA preparing;
- Running thermal analysis module;
- OUTPUT DATA interpretation.

The input data preparations was done using a calculus subroutine, which starting from the values reached by the parameters ε and *α* determine the average thickness of the lubricant film on the finite elements and than the material coefficients (2).
After the modification of the material properties the standard thermal analyses program is run, then the out put data are interpreted also using a calculus subroutine.
A part of the results using MEF in lubrication of sliding bearings are presented in figure 2 and figure 3.
For numerical simulation we used COSMOS M .

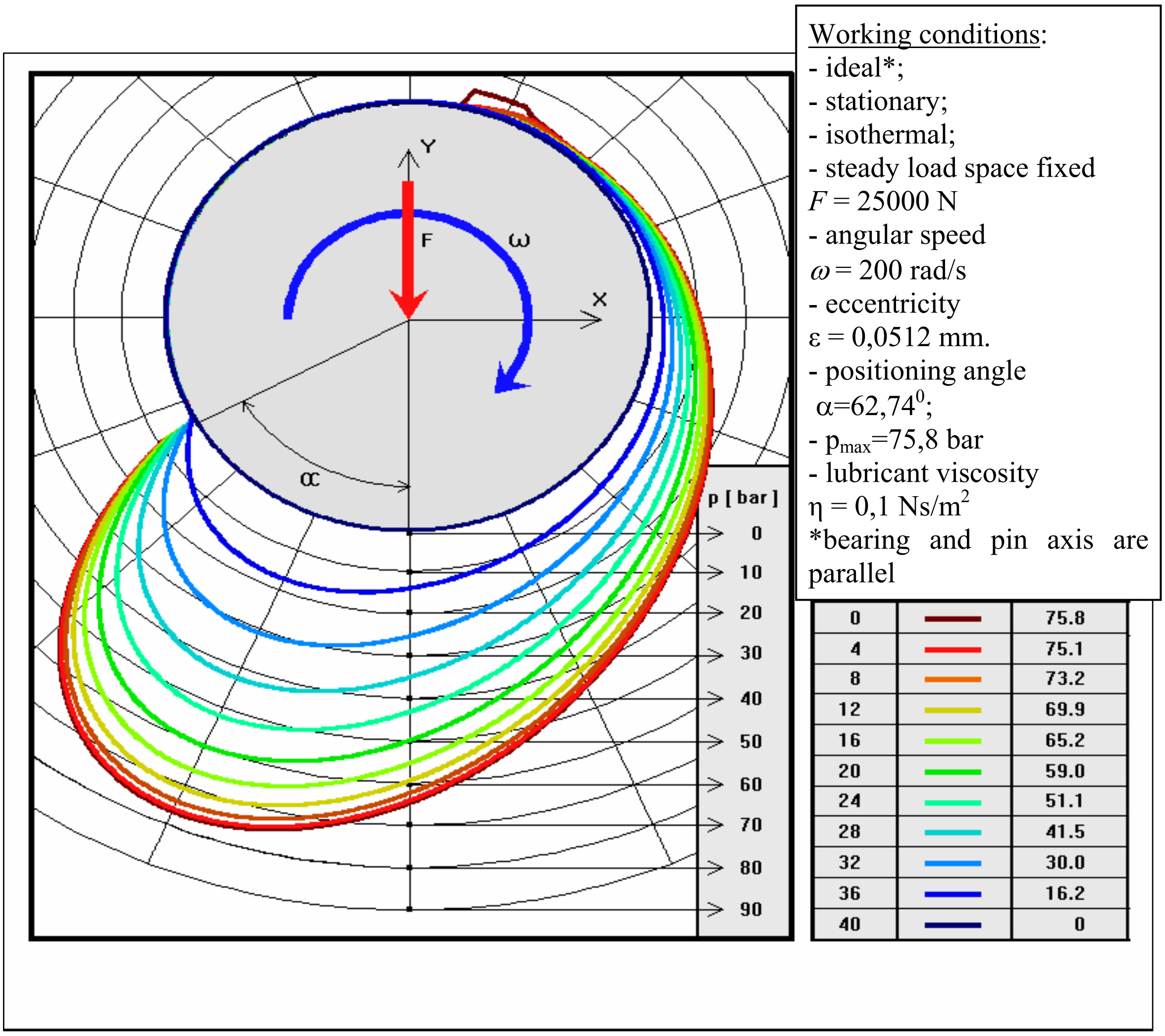
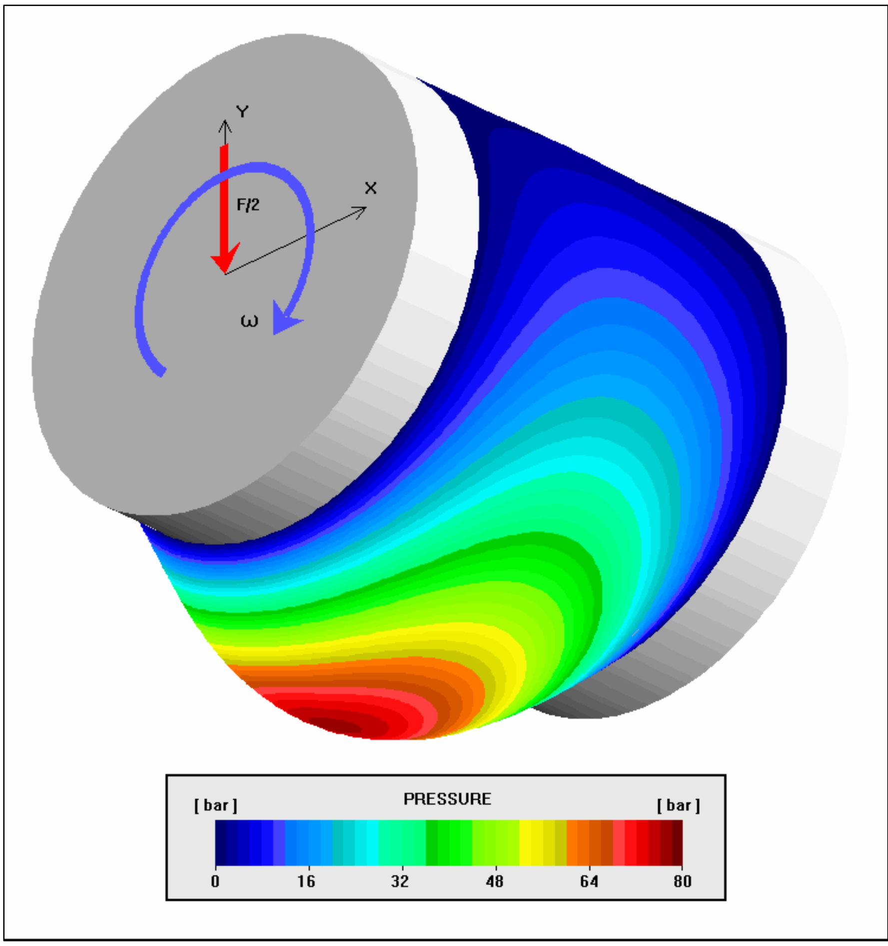


Figure 2. The pressure curves on axle journal in normal planes equidistant for the operation with the eccentricity ε =0,0512 mm. and the angular speed ω = 200 s⁻¹.



Working conditions:
- ideal*;
- stationary;
- isothermal;
- steady load space fixed $F = 25000$ N**
- angular speed $\omega = 200$ rad/s
- positioning angle $\alpha=62{,}74^0$;
- $p_{max}=75{,}8$ bar
- lubricant viscosity $\eta = 0{,}1$ Ns/m^2

*bearing and pin axis are parallel
** equal distribution on pin ends

Figure 3 Pin hydrodynamics pressure distribution.

3. CONCLUSIONS

The numerical simulation of slide bearings with hydrodynamic lubrication allows obtaining useful solutions in the research and design of mechanical systems. To make the presentation simpler we had in view stationary operation with fix load in space for which we used by similitude the simplified Reynolds equation (1).
The generalization of the simulation procedure through the generalized Reynolds equation, when time is a variable, allows the approach on stationary, dynamic phenomena taking place in the hydrodynamic slide bearings.
In the complete analysis of the insertion process is required where the real heat field developed in the bearing depends on time we have a complicated problem to solve which is associated with an effective thermal analysis problem. The proposed procedure is able of approaching the more general problem of hydrodynamic slide bearings irrespective of their shape, their operation on the lubricants properties.

4. REFERENCES

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