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ROMANIA

GERMANY

"GENERAL M.R. STEFANIK"
ARMED FORCES ACADEMY
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ANTIFRICTION MATERIALS TECHNOLOGIES TO ACHIEVE BY ELECTROMAGNETIC SUBMITTING

Elena Irimie (*)

THE MILITARY TECHNICAL ACADEMY

Abstract: *Sliding bearings are important machinery elements which sustained other members (spindles or cranked shafts) to send motions. By sliding surfaces, the bearings are taking over radial, axial and combined forces and in the same time, they allow the spindle to have rotary motions or oscillations. The relative motion between bearing and spindle is faced with a resistance due to friction, which the overcome necessitate energy input. Accordingly sliding bearings frictions, thermal effects is very important for practical applications. The paper presents a simulation procedure by finite elements for hydrodynamic lubrication of sliding bearings. We are also studied, using numerical analysis the influences of geometric discrepancy and pin-bearing misalignment. This research was realizing bases on the contract no. 2/2005, in CEEX Program*.

Keywords: *hydrodynamic lubrication, sliding bearings, MEF, geometric discrepancy, misalignment*

1. INTRODUCTION

Sliding bearings are machinery elements which sustained other members, like spindles or cranked shafts, to send motions. By sliding surfaces, the bearings are taking over radial, axial and combined forces and in the same time, they allow the spindle to have rotary motions or oscillations. The relative motion between bearing and spindle is faced with a resistance due to friction, which the overcome necessitate energy input. Accordingly sliding bearings frictions, thermal effects is very important for practical applications.

We have distinguished two kinds of friction: sliding friction, when the surfaces glide one over the other, and rolling friction, when the surfaces effectuate a rolling around an axis, contained by the contact instantaneous plane. Due to these friction, in the couple bearing - spindle develop heat and wear, dignified by substance loss. When the thermal effect and the wear, exceed the calculate values, the sliding bearing is take out of service. The knowledge of friction processes, the couple materials selection, the contact surfaces qualities and form design, correct lubrication with appropriate lubricant, are the main and efficient solutions to disprove and diminish the friction and his destroyer results.

The elaboration of a new realization technology for sliding bearings, with superior performances, suppose to know theirs roles, materials, types of existing sliding bearings and working conditions. In the paper, we are studied, using numerical analysis the influences of geometric discrepancy and pin-bearing misalignment.

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 films are very slim comparing the global dimensions of sliding bearing;

-the fluid inertia is 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]:

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

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 whole 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(θ,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 that 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, J0 = 0,2 mm.

-working conditions: stationary, isothermal, steady load space fixed, F = 25 kN,

- lubricant viscosity: η = 0.1 Ns/m2.

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.

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 .

3. NUMERICAL ANALYSIS OF GEOMETRIC DISCREPANCY AND MISALIGNMENT OF SLIDING BEARINGS

From OUTPUT DATA we are extracted the fictitious nodal temperatures and we have determined the sliding bearing elements real pressures. Then, by numerical integrating, we are determined the bearing capacity components and the deviation angle to vertical axis (Fig.4).

4. CONCLUSIONS & ACKNOWLEDGMENT

The numerical simulation of slide bearings with hydrodynamic lubrication allows obtaining useful solutions in the research and design of mechanical systems. 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.

The position, form and dimensional discrepancy influences on hydrodynamics sliding bearings work can be easy dignified by described numerical simulation procedure. In the paper we are considered three cases of discrepancy – parallel misalignment axis of pin and bearing, "oval" form of bearing and rippled form of bearing interior surface. The sources of these discrepancies are the technological processing, assemblage inaccuracy, working load deformation and wearing out.

Misalignment axis of pin and bearing produced a non-uniform wearing out at the ends of these elements. The "oval" bearing modified the pin hydrodynamic pressure distribution without consequences on sliding bearing work. The rippled form of sliding bearing interior surface produced some distortions of hydrodynamic pressure distribution with consequence of vibrations apparition.

REFERENCES

1. Constantinescu, V.N., Nica, A., Pascovici, M.D., Sliding Bearings, Ed. Tehnica, Buc,, 1980.
2. Frêne, Jean: Lubrification hydrodinamique. Paliers et Butée - Collection de la Direction des Études et Recherches d'Électricité de France, Editions Eyrolles, Paris, 1990.
3. Hamrock, Bernard J. Fundamentals of Fluid Film Lubrication McGraw-Hill, Inc. 1994,(December,2009).

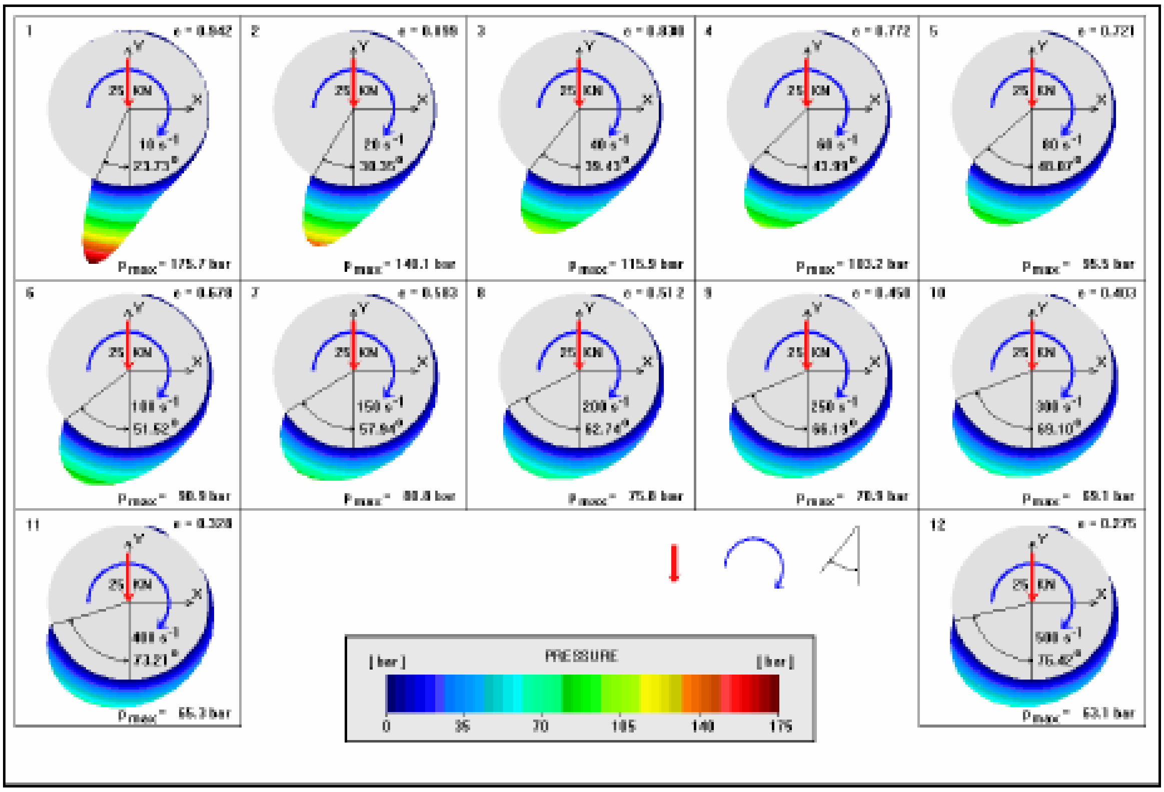

Fig. 1 Pin hydrodynamics pressure distribution function of angular velocity of the shaft

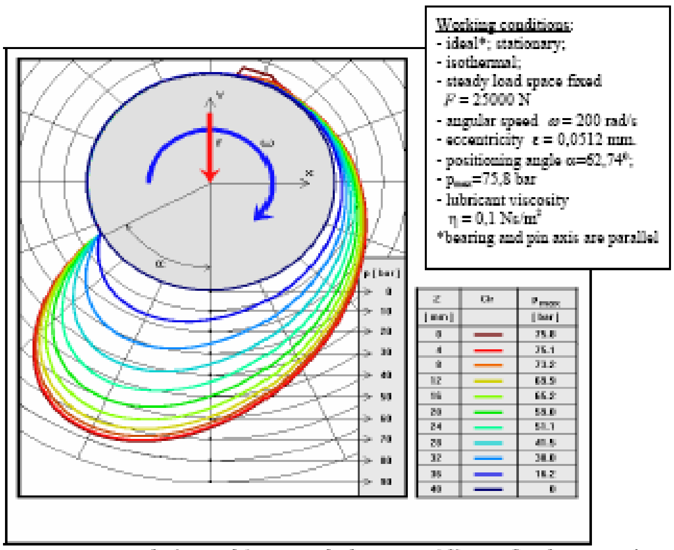

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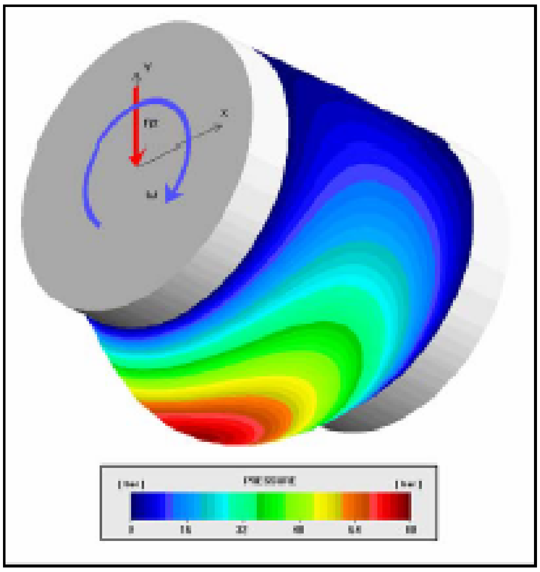
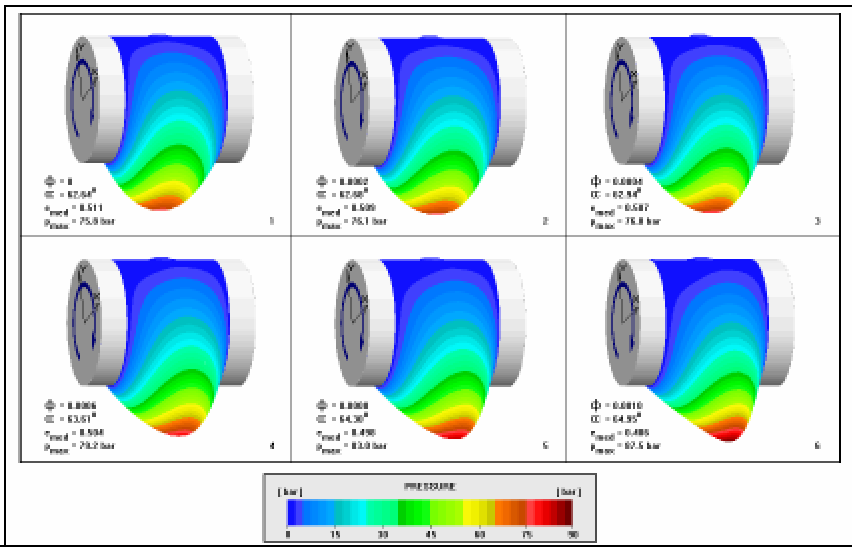
Fig. 2 The pressure curves on axle journal in normal planes equidistant for the operation with the eccentricity ε =0,0512 mm. and the angular speed ω = 200 s-1.



Working conditions:
- ideal*; stationary;
- isothermal;
- steady load space fixed $F = 25000$ N**
- angular speed $\omega = 200$ rad/s
- positioning angle α=62,74°;
- p_{max}=75,8 bar
- lubricant viscosity $\eta = 0,1$ Ns/m²

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

Fig. 3 Pin hydrodynamics pressure distribution.



Bearing sizes: diameter, D = 100 mm; length, L = 80 mm; radial play, J = 0,2 mm.
Working conditions: stationary, isothermal, steady load space fixed, F = 25 kN, constant angular speed ω = 200 rad/s;
Lubricant viscosity: 0.1 Ns/m².
Notations: Φ - relative discrepancy of parallel alignment; α - positioning angle of pin median section center; ε_{med} - mean eccentricity measured in pin median plane.

Fig. 4 Pin hydrodynamics pressure distribution in parallel misalignment conditions

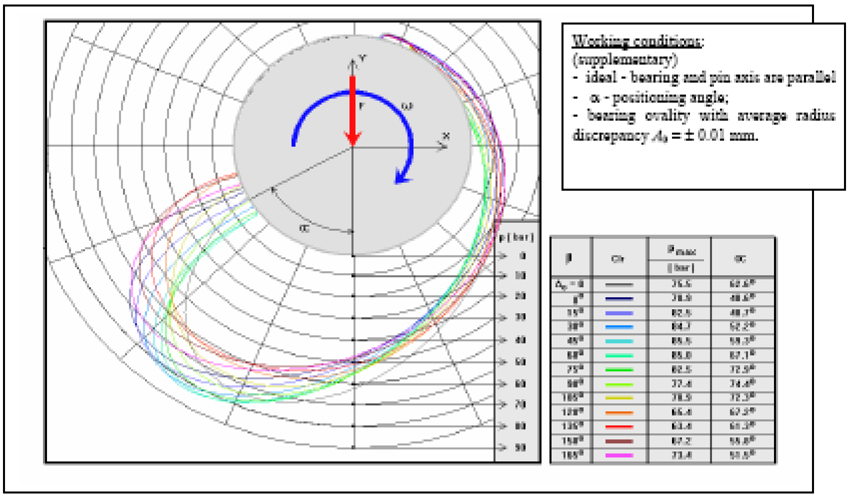


Fig. 5 Hydrodynamics pressure distribution in bearing median plane
("oval" form of interior surface)

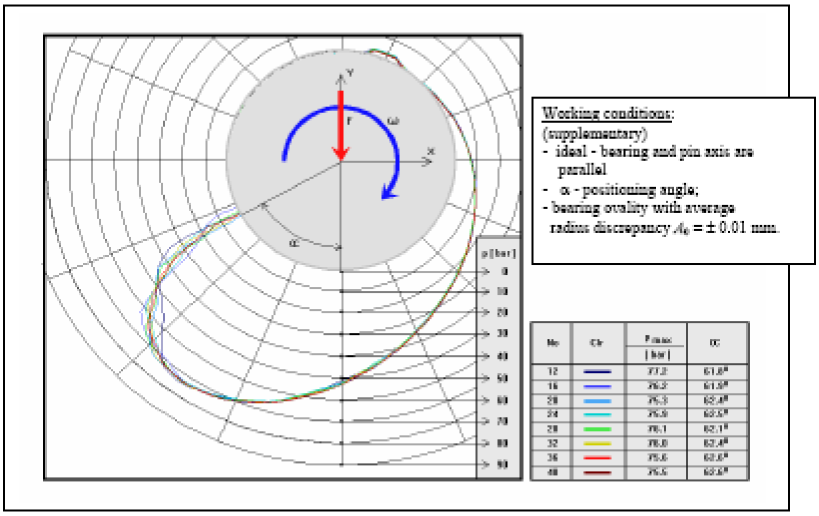


Fig. 6 Hydrodynamics pressure distribution in bearing median plane
(rippled form of interior surface)