10th International Research/Expert Conference
"Trends in the Development of Machinery and Associated Technology"
TMT 2006, Barcelona-Lloret de Mar, Spain, 11-15 September, 2006

NUMERICAL ANALYSIS PREDICTION OF GEOMETRIC DISCREPANCY AND MISALIGNMENT OF SLIDING BEARINGS PERFORMANCE

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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. In the paper we are studied, using numerical analysis the influences of geometric discrepancy and pin-bearing misalignment. This research was realize bases on the contract no. 2/2005, in CEEX Program.
Keywords: sliding bearing, 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. Special emphasis is also laid on bearing dynamics, stability and operation under variable loads and speeds.
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.

2. NUMERICAL ANALYSIS OF GEOMETRIC DISCREPANCY AND MISALIGNMENT OF SLIDING BEARINGS
We will 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. This procedure 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 introduce 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 boundary limits conditions associated to equation (1) are pressures to | - feed hole;
| - sliding bearing ends.

The Reynolds's equation can be see 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:

$$k_\theta=\frac{h^3}{\eta}\quad \text{and}\quad k_z=\frac{h^3}{\eta}\ . \tag{2}$$

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) and 6 nodes (suitable for curved surfaces). We are obtained 3245 elements and 6652 nodes which constitute the lubricant film finite elements model. For exemplification we are considered a sliding bearing with:

- 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/m^2.

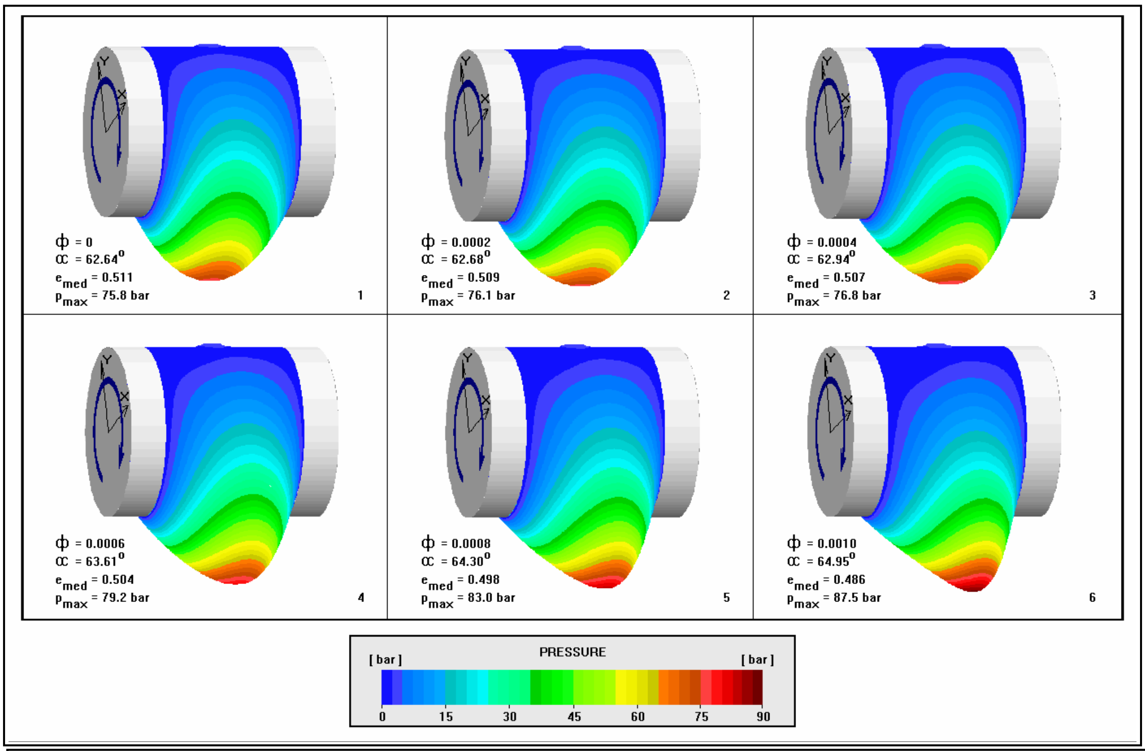
In these conditions, the pin position in sliding bearing are described by:

- mean eccentricity measured in pin median plane, (e_{med});
- positioning angle of pin median section center, (α);
- relative discrepancy of parallel alignment, (Φ).

The solved procedure are incorporated in a iteration loop, including 3 steps:

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

In Figures 1, 2 and 3 we are presented the results of numerical simulation.
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.
For numerical simulation we used COSMOS M .



Bearing sizes: diameter, D = 100 mm; length, L = 80 mm; radial play, J = 0,2 mm.
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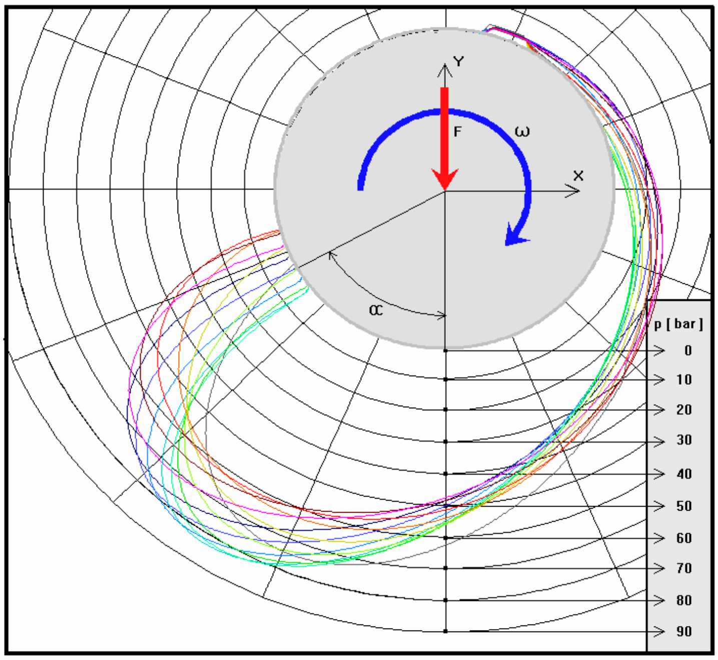
Lubricant viscosity: 0.1 Ns/m^2.
Notations: Φ - relative discrepancy of parallel alignment; α - positioning angle of pin median section center; e_{med} - mean eccentricity measured in pin median plane.

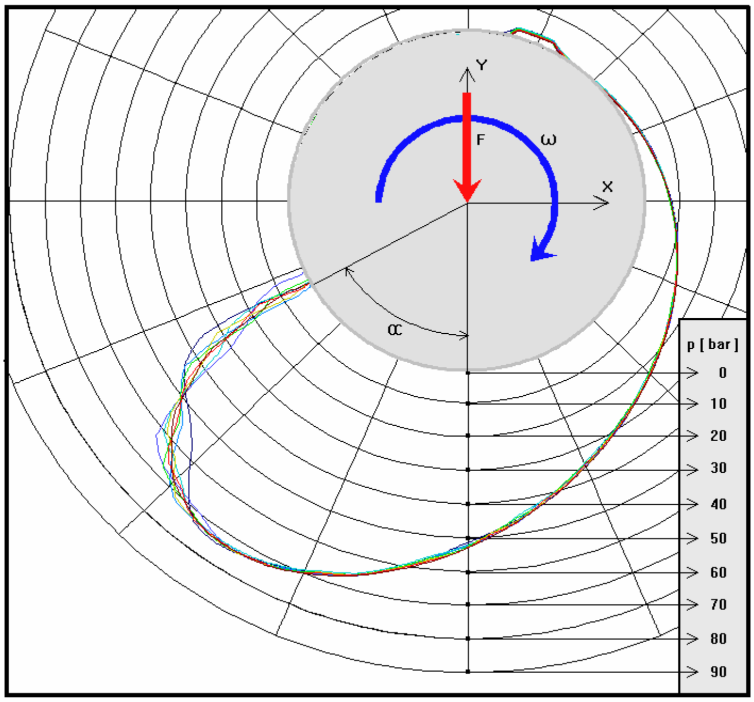
Figure 1. Pin hydrodynamics pressure distribution in parallel misalignment conditions



Working conditions:
(supplementary)
- ideal - bearing and pin axis are parallel
- α - positioning angle;
- bearing ovality with average radius discrepancy $A_0 = \pm$ 0.01 mm.

β	Clr	p_{max} [bar]	α
$A_0 = 0$		75.5	62.6°
0°		78.9	48.6°
15°		82.5	48.7°
30°		84.7	52.2°
45°		85.5	59.3°
60°		85.0	67.1°
75°		82.5	72.9°
90°		77.4	74.4°
105°		70.9	72.3°
120°		65.4	67.2°
135°		63.4	61.3°
150°		67.2	55.8°
165°		73.4	51.5°

Figure 2. Hydrodynamics pressure distribution in bearing median plane
("oval" form of interior surface)



<u>Working conditions</u>:
(supplementary)
- ideal - bearing and pin axis are parallel
- α - positioning angle;
- bearing ovality with average radius discrepancy $A_0 = \pm 0.01$ mm.

No	Clr	p_{max} [bar]	α
12		77.2	61.8°
16		76.2	61.9°
20		75.3	62.4°
24		75.9	62.5°
28		76.1	62.1°
32		76.0	62.4°
36		75.6	62.6°
40		75.5	62.6°

Figure 3. Hydrodynamics pressure distribution in bearing median plane (rippled form of interior surface)

3. CONCLUSIONS

The position, form and dimensional discrepancy influences on hydrodynamics sliding bearings work can be easy dignified by described numerical simulation procedure. Generally, for hydrodynamics sliding bearings problems, we have the solutions for ideal situation. 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.

4. REFERENCES

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