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THE OIL PRESSURE DISTRIBUTION MODIFICATIONS DUE TO THE EFFECT OF UNALIGNED SHAFT

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Abstract: *In order to improve the calculation of the critical rotor speed by including the flexibility of the bearing oil film due to the angular displacements of the shaft, the oil pressure distribution law was obtained first. There are considered both angular and squeeze effects induced by the angular velocities of the moving shaft. An expression for the film thickness in an unaligned journal bearing is developed. This paper presents numerical results for dimensionless pressure versus angular and axial co-ordinate of the journal bearing.*
Key words: *pressure distribution, journal bearing, shaft conical motion*

1. INTRODUCTION

The Reynolds equation was analytically solved and pressure distribution is presented as a function of principal parameters for the case of unaligned journal bearings.

A study of the local minimum dimensionless film thickness expression is made. (Baskharone et al., 1991)

In the end of the article, some drawings, representative for the pressure distribution modifications, due to the effect of unaligned shaft, in stationary case, are presented. (Kanemory & Iwatsubo, 1992)

2. NOMENCLATURE

B - width of bearing;
$\overline{B}$ - dimensionless width of bearing, B/D;
d - diameter of journal;
D - diameter of bearing;
e - journal lateral eccentricity;
h - film thickness;
$\overline{h}$ - dimensionless film thickness, $2h/J$;
$\overline{h}_m(\overline{z})$ - function of the minimum dimensionless film thickness;
$\left(\overline{h}_m\right)_{aligned}$ - minimum dimensionless film thickness for aligned case;
$\left(\overline{h}_m\right)_{\max}$ - maximum value of function $\overline{h}_m(\overline{z})$;
$\left(\overline{h}_m\right)_{\min}$ - minimum value of function $\overline{h}_m(\overline{z})$;
J - diametral clearance, $D - d$;
n - shaft speed;
p_m - average pressure;
$\overline{p}$ - dimensionless pressure, $p \cdot \Psi^2 / (3\pi \cdot n\eta)$;
x, y, z- cartesian co-ordinates (fixed $xOyz$ – frame);
x_b, y_b- co-ordinates of point M_b on the bearing contour;
x_s, y_s- co-ordinates of point M_s on the shaft contour;
$\overline{z}$ - dimensionless axial co-ordinate, $2z/B$;
$\tilde{z}$ - dimensionless axial co-ordinate corresponding to $\left(\overline{h}_m\right)_{\max}$
X,Y,Z- cartesian co-ordinates (rotated $XOYZ$–frame);
α, β - angular displacements;
$\dot{\alpha}, \dot{\beta}$ - angular velocities;
$\dot{\alpha}^*, \dot{\beta}^*$ - dimensionless angular velocities, $\left(\dot{\alpha}, \dot{\beta}\right)/(\pi \cdot n)$;
ε - eccentricity ratio, $2e/J$;
Ψ - dimensionless clearance, J/D;
η - dynamic viscosity;
θ - angular co-ordinate. (Someya, 1989)

3. THE PRESSURE DISTRIBUTION LAW

Using a new mathematical model of a short length bearing where the shaft has a conical motion, the oil film thickness was calculated: (Suciu & Parausanu, 1996)

$$h = \sqrt{\left(x_b - x_s\right)^2 + \left(y_b - y_s\right)^2} \qquad (1)$$

and then:

$$h \cong \frac{J}{2} + e\cos\theta - z\alpha\cos\theta + z\beta\sin\theta \qquad (2)$$

or in dimensionless form:

$$\overline{h} = \frac{2h}{J} = 1 + \varepsilon\cos\theta - \frac{\overline{B}}{\Psi}\overline{z}\alpha\cos\theta + \frac{\overline{B}}{\Psi}\overline{z}\beta\sin\theta \qquad (3)$$

where: $\overline{z} = 2 \cdot z/B$; $\overline{B} = B/D$ and $\Psi = J/D$. For each fixed $\overline{z}$, the local minimum dimensionless film thickness expression is:

$$\overline{h}_m(\overline{z}) = 1 - \frac{\overline{B}}{\Psi}\sqrt{\left(\frac{\varepsilon\Psi}{\overline{B}} - \overline{z}\alpha\right)^2 + \left(\overline{z}\beta\right)^2} \qquad (4)$$

Considering the relation of $\overline{h}_m(\overline{z})$, it is obvious that at the central plane the same minimum film thickness is obtained as in the aligned journal bearing case:

$$\overline{h}_m(\overline{z} = 0) = 1 - \varepsilon = \left(\overline{h}_m\right)_{aligned} \qquad (5)$$

Additionally, it can be demonstrated that in the point

$$\tilde{z} = \frac{\alpha \cdot \varepsilon \cdot \Psi}{\overline{B} \cdot (\alpha^2 + \beta^2)} \qquad (6)$$

the functions have a maximum value higher than the aligned journal bearing minimum thickness:

$$\left(\overline{h}_m\right)_{\max} = 1 - \frac{\varepsilon \cdot \beta}{\sqrt{\alpha^2 + \beta^2}} \geq 1 - \varepsilon \qquad (7)$$

Finally, the minimum value of the function $\overline{h}_m(\overline{z})$ is reached at the left bearing end, namely:

$$\left(\overline{h}_m\right)_{\min} = \overline{h}_m(\overline{z} = -1) = 1 - \sqrt{\left(\varepsilon + \frac{\overline{B}}{\Psi}\alpha\right)^2 + \beta} \qquad (8)$$

this is lower than the minimum film thickness of the aligned case. (San Andres, 1993)

When the Reynolds equation is integrated twice along the axial co-ordinate z under boundary conditions, the pressure distribution has the form:

$$p=\frac{6\pi\cdot n\eta\overline{B}^2}{\Psi^2}\mathrm{E}_1+3\pi\cdot n\eta\,\mathrm{E}_2\left(\mathrm{E}_3-\mathrm{E}_4\right) \qquad (9)$$

where $\mathrm{E}_1=\mathrm{E}_1(\theta,\bar{z})$, $\mathrm{E}_2=\mathrm{E}_2(\theta)$, $\mathrm{E}_3=\mathrm{E}_3(\theta,\bar{z})$, $\mathrm{E}_4=\mathrm{E}_4(\theta,\bar{z})$.

In dimensionless form the pressure distribution is given by equation (10).

$$\bar{p}=\frac{p\Psi^2}{3\pi\cdot n\eta}=2\overline{B}^2\,\mathrm{E}_1+\Psi^2\,\mathrm{E}_2\left(\mathrm{E}_3-\mathrm{E}_4\right) \qquad (10)$$

4. NUMERICAL RESULTS

In Figure 1 four sketches of dimensionless pressure (p/p_m) versus angular co-ordinate $(\theta\in[0;2\pi])$ and axial co-ordinate $(z\in[-B/2;B/2])$, are considered, the main parameters having values presented in Table 1.

These drawings are representative for the pressure distribution modifications due to the effect of unaligned shaft, in stationary case (when there are important angular displacements α and β).

Thus, Figure 1a is a plot of the reference case, namely the aligned narrow journal bearing case. Pressure distribution occurs as a single symmetric “mountain” regarding the axial co-ordinate, being placed approximately in $[0;\pi]$ θ domain.

Figures 1b, 1c and 1d are dedicated to the stationary case when angular displacement α is with an order greater then β . The differences between these cases and the reference case are very significant. For the lowest eccentricity $(\varepsilon=0)$ two “pressure mountains” occur which have approximately the same peak. The “mountain” placed in $[\pi;2\pi]$ θ domain decreases with the increasing of eccentricity until it disappears at $\varepsilon=0.2$, the pressure distribution becoming strongly non-symmetric, with a higher peak of pressure. So, it is obvious that this pressure distribution produces not only a load capacity, but a moment, too.

5. CONCLUSIONS

This work occurs as a hydrodynamic analysis regarding the problem of an unaligned narrow journal bearing.

The stationary unaligned narrow journal bearing case appears as a particular case of the non-steady case, when $\dot{\alpha}=\dot{\beta}=0$.

The pressure distribution is much different in comparison with the aligned case, becoming strongly non–symmetric, so it is obvious that this pressure produces not only a load capacity, but a moment too.

Although, in the central plane the same minimum film thickness as in the aligned case is obtained, the minimum oil film thickness is inferior to that corresponding in the aligned case.

The relative difference of the attitude angle is between 25…35% in normal working conditions, and this important variation can badly affect the oil supplying of the journal bearing.

Figure	$\overline{B}$	Ψ ‰	ε	α [rad]	β [rad]	$\left(\frac{p}{p_m}\right)_{max}$
1a	0.5	1	0.1	0	0	0.4924
1b	0.5	1	0	10^{-3}	10^{-4}	0.5282
1c	0.5	1	0.05	10^{-3}	10^{-4}	0.8048
1d	0.5	1	0.2	10^{-3}	10^{-4}	2.2479

Tab. 1. The main parameters values

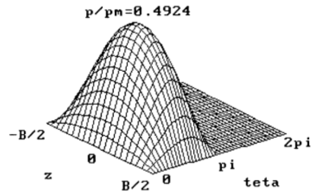

Fig. 1a. Dimensionless pressure distribution – case a

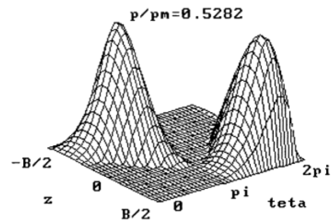

Fig. 1b. Dimensionless pressure distribution – case b

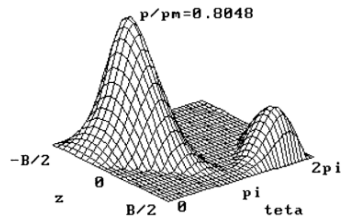

Fig. 1c. Dimensionless pressure distribution – case c

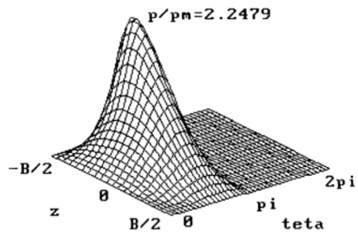

Fig. 1d. Dimensionless pressure distribution – case d

It was observed the angular displacements and velocities produce strong effects on the moment attitude angle.

As a consequence of the non–symmetric pressure distribution, a torque occurs. This must be taken into account when calculating the stiffness and the damping coefficients of unaligned journal bearings. This work is just a preliminary announcement of some results regarding these calculations.

6. REFERENCES

Baskharone, E.A. & Hensel, S.J. (1991). "*Interrelated Rotordynamic Effects of Cylindrical and Conical Whirl of Annular Seal Rotor"s*, ASME, Journal of Tribology, vol. 113, pp. 470-480

Kanemory, Y., Iwatsubo, T. (1992). "*Experimental Study of Dynamic Fluid Forces and Moments of a Long Annular Seal"*, Journal of Tribology, vol. 114, 2, pp. 773-778.

San Andres, L.A. (1993). "*Dynamic Force and Moments for Short Length Annular Seals", ASME,* Journal of Tribology, vol. 115, pp. 61-70

Someya, T. (1989). "*Journal Bearing Databook"*, Springer Verlag, Berlin

Suciu, C.V. & Parausanu, I. (1996). "*A Complet Investigation of unaligned narrow"*, Journal Bearing, Mecanique appliquee, Vol. 41, pp. 99-120