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ANALYSIS OF THE INFLUENCE OF BEARINGS' LUBRICANT TEMPERATURE AT MAIN PARAMETRES OF A ROTOR-BEARING SYSTEM

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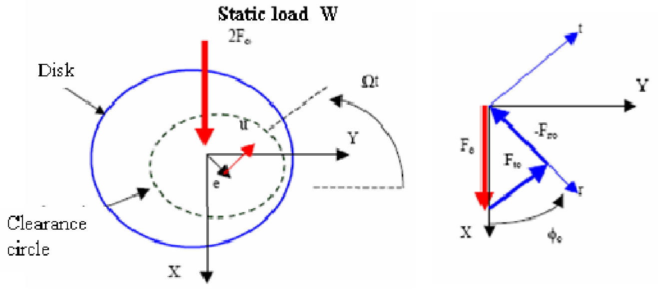
ABSTRACT
The analysis of the temperature of lubricant at bearings influence at adopted rotor-bearing system has been given in this paper.
The models of dynamic rotor-bearing system for both rigid and flexible rotor supported at bearing with infinite short length were adopted and analysed.
The system was analysed for laminar flow of lubricant in bearing – without turbulence and without cavitations.
The mathematical model for adopted dynamic rotor-bearing model was solved through MathCad software.
The conclusions have been made based on the results shown graphically and through diagrams.
Key words: Lubricant Temperature, Bearing, Rotor-Bearing System, Laminar Flow

1. INTRODUCTION

In this paper the rotor-bearing system is analyzed. The influence of lubricant temperature at bearings has been taken into consideration. Also, it was elaborated the behavior of the system for rigid rotor and for the flexible rotor, comparing the achieved results.
The geometry of bearing is of great importance, therefore the influence of the L/D report is considered as well.
The distribution of the pressure and forces in oil film of bearing were determined for laminar flow of lubricant in bearing, without cavitations and without turbulence and as well as without influence of thermal effects. All these effects were considered for the bearing with infinite short length.
Under these adopted considerations the stiffness and damping coefficients for bearing were calculated as the most important components that emanates from the influence of hydrodynamics processes in oil film bearing and have a decisive influence in rotor oscillations.
The mathematical model built based on adopted mechanical model enables calculation and analysis of the stiffness and damping coefficients, critic masses, threshold speed, natural frequencies and angular velocity for rigid and flexible rotor for the three working temperatures of lubricant.
Results were graphically presented showing a good start point to come to the needed conclusions.

2. MECHANICAL AND MATHEMATICAL MODELS FOR A ROTOR-BEARING SYSTEM

To analyze the behavior of rotor-bearing system the mechanical model was adopted considering the rotor as rigid and flexible supported at bearings with infinite short length *(L/D<0,5)* and the flow of lubricant in bearing is laminar – without turbulence and without cavitations.



Simbols: C - radial clearance of oil film;

C_{ij} - dumping coefficients;
K_{ij} - stiffness coefficients;
ε - journal eccentricity;
ϕ - positional angle of bearing;
L - axial length of bearing;
Ω –angular velocity of journal
ω – angulara velocity

Figure 1. Rotor-Bearing system

For such mechanical model can be formulated mathematical model.
Modified Sommerfeld Number is calculated by:

$$\frac{\mu\,\Omega\,L\,R}{4F_0}\left(\frac{L}{R}\right)^2 = \sigma = \frac{\left(1-\varepsilon^2\right)^2}{\varepsilon\sqrt{\left\{16\varepsilon^2+\pi^2\left(1-\varepsilon^2\right)\right\}}} \tag{1}$$

Total load, *Figure 1*, *W0=2F0* for two bearings supporting symmetric rotor is given by:

$$F_0 = \frac{\mu\,\Omega\,(L/C)^2\,L\,R}{4\sigma}$$

$$f_{r0} = -\frac{F_{r0}}{F_0} = \cos\phi_0 = \frac{4\sigma\varepsilon^2}{\left(1-\varepsilon^2\right)^2}\,; \quad f_{t0} = +\frac{F_{t0}}{F_0} = \sin\phi_0 = \frac{\pi\sigma\varepsilon}{\left(1-\varepsilon^2\right)^{3/2}} \tag{2}$$

Non-dimensional stiffness, respectively dumping coefficients for bearings are calculated through:

$$\begin{aligned}
k_{XX} &= K_{XX}\frac{C}{F_0} = \frac{f_{r0}}{\varepsilon\left(1-\varepsilon^2\right)}\left\{f_{r0}^2+1+2\varepsilon^2\right\}\\
k_{YY} &= K_{YY}\frac{C}{F_0} = \frac{f_{r0}}{\varepsilon\left(1-\varepsilon^2\right)}\left\{f_{t0}^2+1-\varepsilon^2\right\}\\
k_{YX} &= K_{YX}\frac{C}{F_0} = \frac{f_{t0}}{\varepsilon\left(1-\varepsilon^2\right)}\left\{f_{r0}^2-1+\varepsilon^2\right\}\\
k_{XY} &= K_{XY}\frac{C}{F_0} = \frac{f_{t0}}{\varepsilon\left(1-\varepsilon^2\right)}\left\{f_{r0}^2+1+2\varepsilon^2\right\}
\end{aligned}
\;,\quad
\begin{aligned}
c_{XX} &= C_{XX}\frac{C\Omega}{F_0} = \frac{2f_{t0}}{\varepsilon\left(1-\varepsilon^2\right)}\left\{\left(2+\varepsilon^2\right)f_{r0}^2+1-\varepsilon^2\right\}\\
c_{YY} &= C_{YY}\frac{C\Omega}{F_0} = \frac{2f_{t0}}{\varepsilon\left(1-\varepsilon^2\right)}\left\{\left(2+\varepsilon^2\right)f_{t0}^2-1+\varepsilon^2\right\}\\
c_{XY} &= C_{XY}\frac{C\Omega}{F_0} = \frac{2f_{r0}}{\varepsilon\left(1-\varepsilon^2\right)}\left\{\left(2+\varepsilon^2\right)f_{t0}^2-1+\varepsilon^2\right\} = c_{YX}
\end{aligned} \tag{3}$$

For small amplitudes of journal centre movement around equilibrium position, the rotational speed is given as follows:

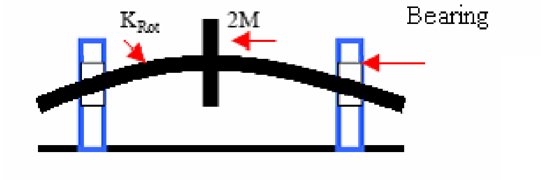
$$p^2\begin{bmatrix}\Delta x''\\ \Delta y''\end{bmatrix} + \begin{bmatrix}c_{XX} & c_{XY}\\ c_{YX} & c_{YY}\end{bmatrix}\begin{bmatrix}\Delta x'\\ \Delta y'\end{bmatrix} + \begin{bmatrix}k_{XX} & k_{XY}\\ k_{YX} & k_{YY}\end{bmatrix}\begin{bmatrix}\Delta x\\ \Delta y\end{bmatrix} = \begin{bmatrix}0\\ 0\end{bmatrix},\quad p^2 = \frac{C\,M\,\Omega^2}{F_0}$$

$$p_S^2\overline{\omega}_S^2 = k_{eq} = \frac{k_{XX}c_{YY}+k_{YY}c_{XY}-c_{YX}k_{XY}-c_{XY}k_{YX}}{c_{XX}+c_{YY}},\quad p_S^2 = M\Omega_S^2\left(C/F_0\right) \tag{4}$$

$$M\omega_S^2 = k_{eq}\left(\frac{F_0}{C} = K_{eq}\right),\ \overline{\omega}_S = \omega_S/\Omega_S,\ \omega_S = \sqrt{\frac{K_{eq}}{M}} = \omega_n$$

For the flexible rotor supported in bearings, *Figure 2*, non-dimensional coefficient from (4) has different value:

$$p_{Sf}^2 = \frac{p_S^2}{1 + k_{eq}\left(\frac{T}{C}\right)} \quad (5)$$



T=F₀/Krot

Figure 2.Flexible rotor supported on bearings

3. RESULTS OF THE ANALYSIS

The results of the analysis based in adopted model expressed with equations from (1)-(5) are graphically presented in Figure 3, for stiffness and dumping coefficients and critic mass and in Figure 4 for threshold speed and natural frequencies. Graphically presented results are for three working temperatures 60 ºC,
75 ºC and 90ºC for the constructive dimensions and data for a rotor of Siemens AG turbogenerator [6].
Results and graphs are processed through MathCad software.

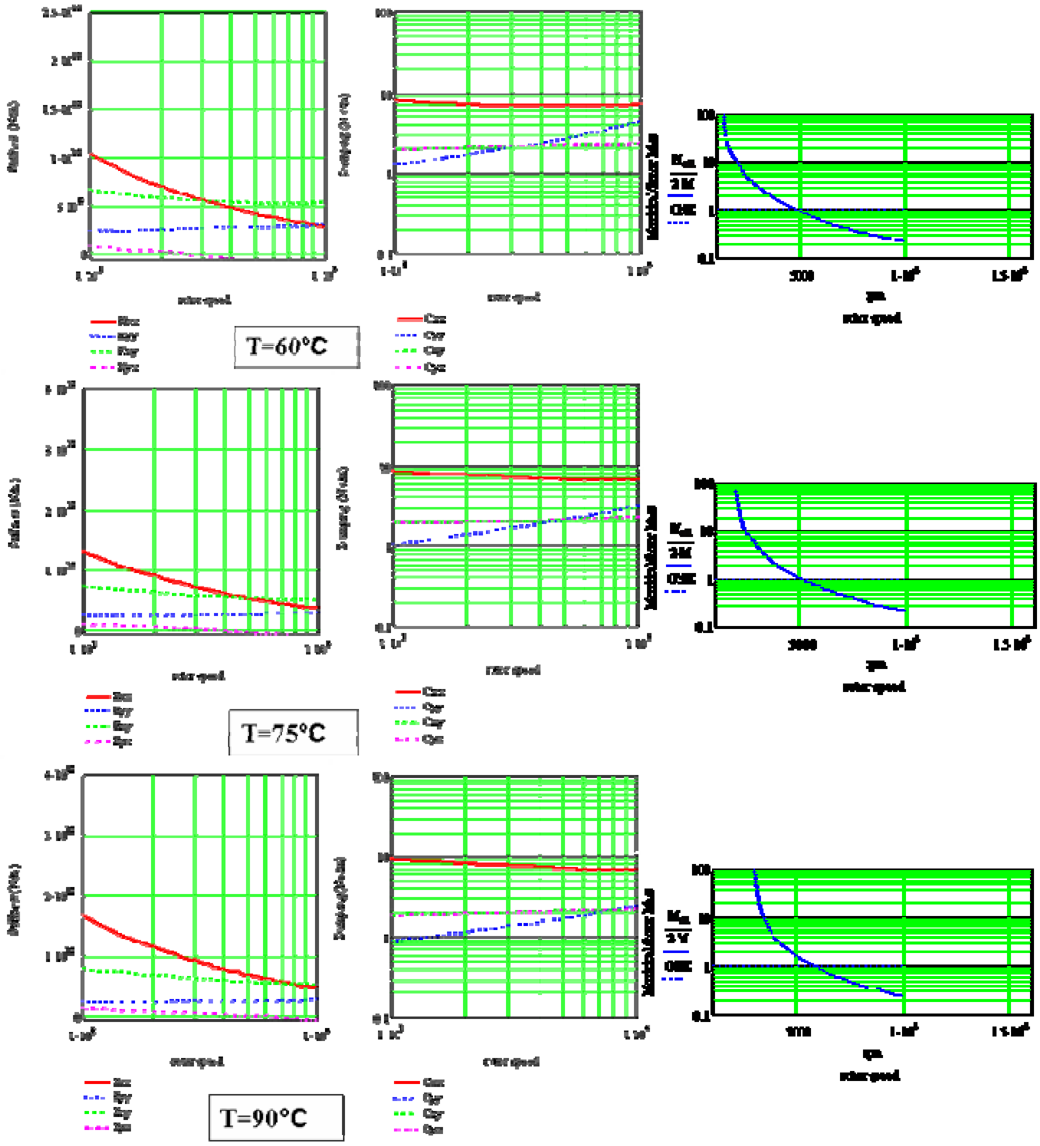


Figure 3.Graphs for stiffness, dumping coefficients and critic mass for three temperatures

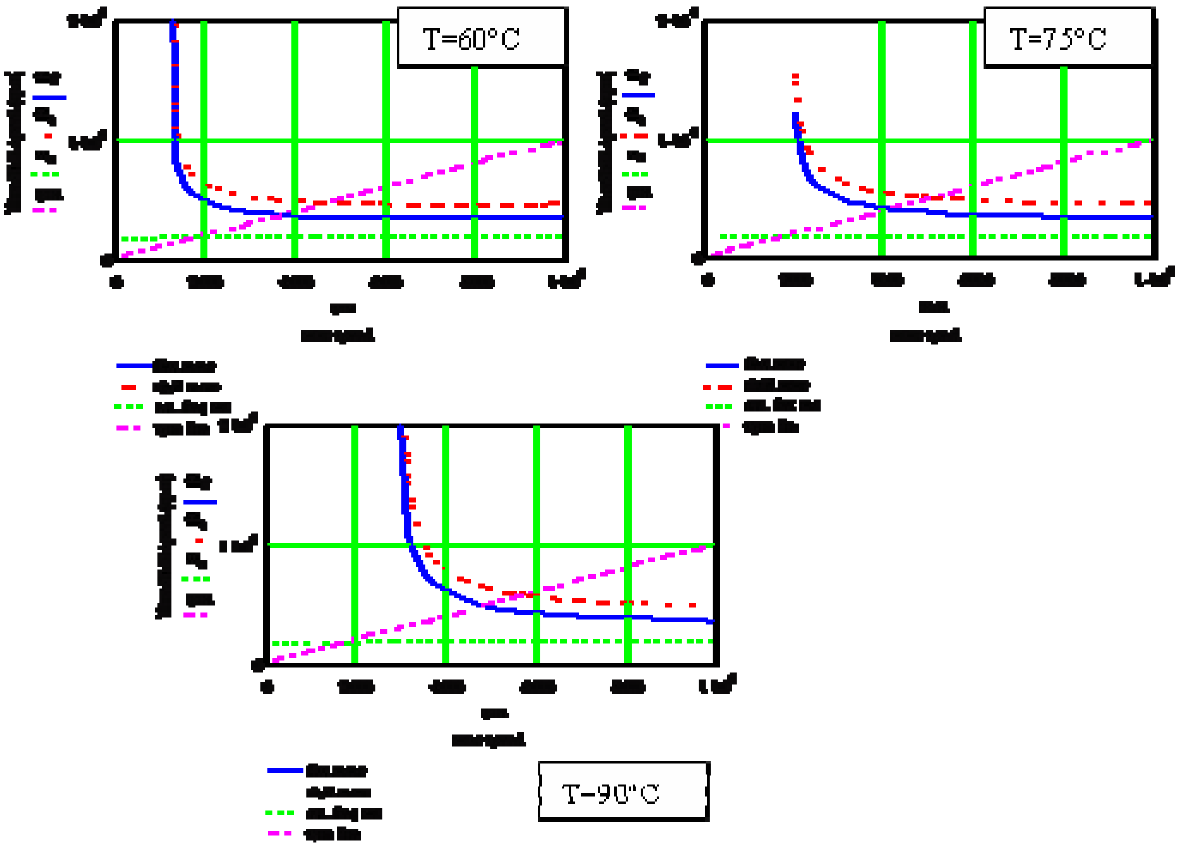


Figure 4. Graphs for threshold speed, natural frequencies for three working temperatures

4. CONCLUSIONS

Based in analysis can be concluded that:

- For working oil temperature increase the stiffness coefficients (*Kxx;Kyy;Kxy;Kyx*) also slightly increases, but the damping coefficients (*Cxx;Cyy;Cxy;Cyx*) have slight decrease *(Figure 3)*;
- The limit of critical speed at rigid rotor connected to critic mass MCR increses due to working temperature of lubricant *(Figure 3)*;
- Due to increase of working temperature of oil *T*, the natural frequency for rigid rotor decreases non-noticeably *(Figure 4)*;
- Threshold speed instability due to working temperature of oil *T* increases for rigid rotor, while decreases for flexible rotor *(Figure 4)*.

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