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INFLUENCE OF THE LUBRICANT'S TEMPERATURE AT NON-BALANCED FLEXIBLE ROTOR BEHAVIOUR

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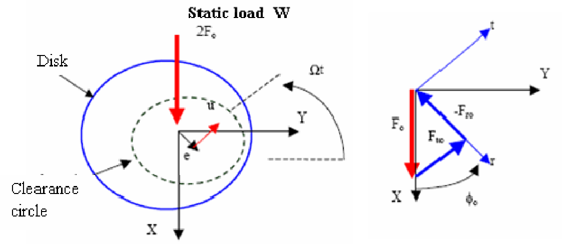
ABSTRACT
In this paper the influence of the lubricant's temperature through viscosity coefficient $\boldsymbol{\mu}$ *at the non-balanced synchronous flexible rotor behaviour has been analysed. The disk of rotor (X;Y) and displacement of the journal (x;y) are synchronous under non-balanced excitation.*
The equations of rotor/bearing system are at a position where coordinates of the rotor and the disk during the motion are at the origin of static equilibrium, while damping at the middle of rotor and the bearing's mass hasn't been taking into the consideration.
The system rotor/bearing was analysed for flexible rotor supported at hydrodynamic bearing with infinite short length for laminar flow of lubricant in bearing – without turbulence and without cavitations and as well as without error at mounting of the bearing.
Results are presented graphically and in tables through which the conclusions have been made.
The Simulations has been performed in Mathcad software.
Keywords: Lubricant's Temperature, Flexible Rotor, Rotor/Bearing System, Laminar Flow

1. INTRODUCTION
In this paper the influence of lubricant temperature in the bearings at the adopted rotor/bearing system is analyzed. The influence was described through viscosity coefficient $\boldsymbol{\mu}$ at the non-balanced synchronous flexible rotor.
The disk of rotor (X;Y) and the journal (x;y) are synchronous and under non-balanced excitation.
The geometry of bearing (L/D report) which is of great importance has been taken into consideration.
The rotor is supported at hydrodynamic bearings with laminar flow of lubricant without turbulence and without cavitations. The mounting/assembling error of bearings has not been taken into account.
All these effects were considered for the bearing with infinite short length.
The stiffness and damping coefficients for bearings are considered having in mind that their components 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 analyses of response of disk and bearings for three different working temperatures
The graphically presented results were a good base to build conclusions on system response and carried analyses.

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 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 and neglecting the error of bearing mounting.



Symbols: 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 – angular velocity

Figure 1. Rotor-Bearing system

The disk of rotor (X;Y) and the journal (x;y) are synchronous and under non-balanced excitation. The equations of motion of rotor and the bearings are at such position where rotor and disk coordinates have the same origin at static equilibrium. Damping at the middle of rotor and bearing mass are not taken into consideration.

a) Rotor

$$m\left(\frac{d^2X}{dt^2}\right)+k(X-x)=ma\omega^2\cos(\omega t)$$
$$m\left(\frac{d^2Y}{dt^2}\right)+k(Y-y)=ma\omega^2\sin(\omega t) \qquad (1)$$

b) Bearing (mass neglected)

$$(C_{I,J})\begin{pmatrix}\dfrac{dx}{dt}\\ \dfrac{dy}{dt}\end{pmatrix}+(K_{I,J})\begin{pmatrix}x\\ y\end{pmatrix}=-k\begin{pmatrix}x-X\\ y-Y\end{pmatrix} \qquad (2)$$

Substituting $a=0.5\,C$, $k=Ks$, $m=M$, $\omega=\Omega$, the solution is:

$$\begin{pmatrix}X_n\\ Y_n\\ x_n\\ y_n\end{pmatrix}:=\begin{bmatrix}-(\Omega_n)^2\cdot M+K_s & 0 & -K_s & 0\\ 0 & -(\Omega_n)^2\cdot M+K_s & 0 & -K_s\\ -K_s & 0 & K_s+Kxx_n+i\cdot\Omega_n\cdot Cxx_n & Kxy_n+i\cdot\Omega_n\cdot Cxy_n\\ 0 & -K_s & Kyx_n+i\cdot\Omega_n\cdot Cyx_n & K_s+Kyy_n+i\cdot\Omega_n\cdot Cyy_n\end{bmatrix}^{-1}\cdot\begin{bmatrix}M\cdot a\cdot(\Omega_n)^2\\ -i\cdot\left[M\cdot a\cdot(\Omega_n)^2\right]\\ 0\\ 0\end{bmatrix} \qquad (3)$$

3. RESULTS OF THE ANALYSIS

The results of the analysis based in adopted model expressed with equations from (1)-(3) for rotor and journal bearing response are graphically presented in Figure 2, 3 and 4 respectively for three working temperatures 60 ºC, 75 ºC and 90ºC for the constructive dimensions and data for a rotor of Siemens AG turbogenerator [6, 7]. The analysis has been carried for an optimal value of relation *L/D=0,45* [6]. Results and graphs are processed through MathCad software.

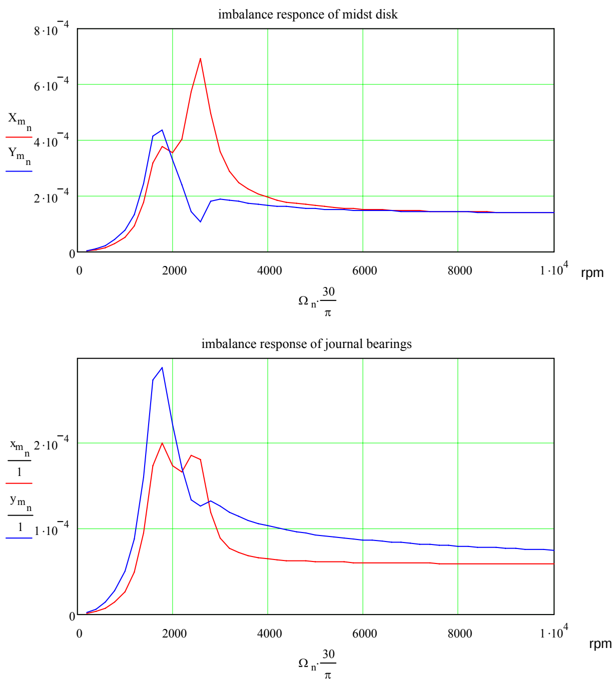


Figure 2. Disk and journal bearing response at T=60°

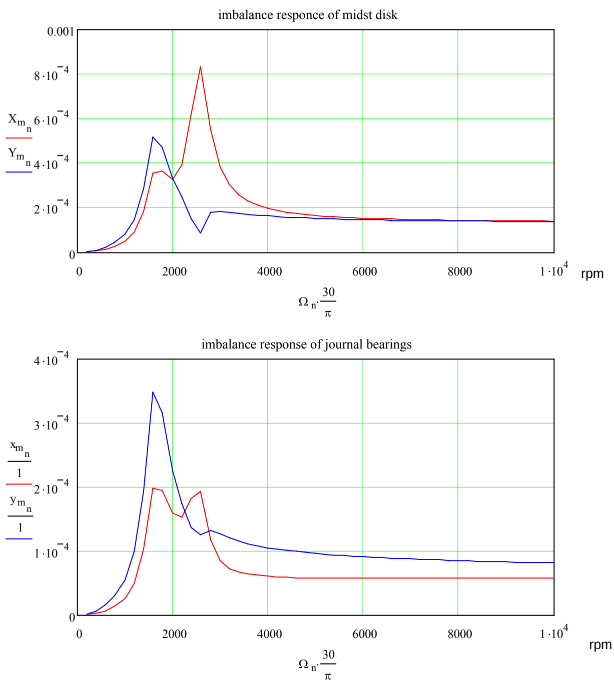


Figure 3. Disk and journal bearing response at T=75°

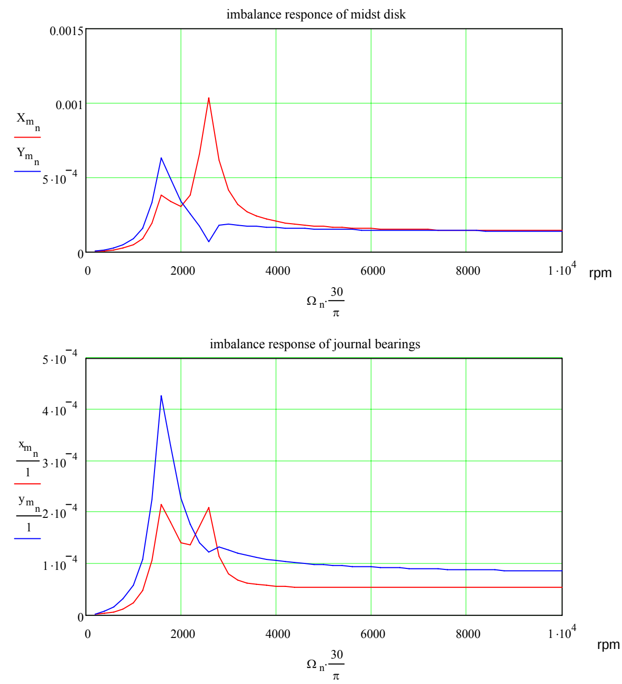


Figure 4. Disk and journal bearing response at T=90°

4. CONCLUSIONS

Based in analysis can be concluded that:

- Maximal amplitudes of deviation at the middle of disk and journal bearing increase with increase of the working temperature (*Figure 2,3 and 4)*;
- Maximal amplitudes at the middle of disk are higher for direction ***X***, while for journal bearing are higher for direction ***y***. are higher.
- The lubricant's temperature has no important influence at "critical speed" in relation to non-balance excitation at amplitudes' deviations.

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