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THE DYNAMIC BEHAVIOR OF THE FAST ROTATING SYSTEM IN DEPENDENCE ON OPERATING PARAMETERS OF BEARING SUPPORTS

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Abstract: Typical components of many industrial devices are rotating parts. In case of automotive engines, turbochargers are used in order to increase the engine efficiency. These turbochargers are understood as relatively small rotating parts working in aggressive conditions. Although they are usually mounted in bearing housing by two journal bearings and one thrust bearing, in case of the rotor lateral vibration analysis the thrust bearing influence is neglected due to its minimal influence. Each journal bearing is of two lubricating layers in series that are separated by a free floating ring. The interconnection of two lubricating layers causes a significant increase in computational time in case of numerical solution of equations of motion to analyse real time motion of turbocharger rotor. For this reason, this paper is focused on turbocharger rotor-bearing system description in time domain.

Keywords: turbocharger rotor; floating ring bearings; dynamics

1 Introduction

In technical practice, there is often an effort to increase machine performance while maintaining safety and economy requirements [1]. For this reason, there is a need to produce lightweight structures while maintaining machine service life. The change in material properties is accompanied by changes in dynamic behaviour, even under both normal and non-standard operating conditions. In order to detect operational changes, the computational models describing complex systems are built not only to affect real system properties but also to describe its behavior during operation. According to [2], the concept of the mathematical model construction is based on system decomposition on individual components (subsystems) that are usually considered as flexible bodies. Constraints between subsystems are usually of strongly non-linear nature. Then, the mathematical description is given by a larger number of nonlinear equations. For this reason, the goal of this paper is to present a full computational model of turbocharger rotor-bearing system as one component of automotive engine.

2 Mathematical model of turbocharger rotor-bearing system

In order to analyse operation of the turbocharger, the mathematical description of rotor-bearing system is needed. The rotor-bearing system of a turbocharger is performed in sequential main steps. In the first step, the turbocharger rotor is derived using the finite element method (FEM). With regard to the geometry of the rotor, the mathematical description of the floating ring bearing is derived in the second step. The third step is to link the partial models mentioned above to obtain a sufficiently precise description of a rotor-bearing system. This procedure is described in detail in [2]. Thus, the flexible rotor supported by two fully floating ring bearings that is affected by non-linear forces can be mathematically described in the form

$$\mathbf{M}_{\Sigma}\ddot{\mathbf{q}}_{L}\left(t\right)+\mathbf{B}_{\Sigma}\dot{\mathbf{q}}_{L}\left(t\right)+\mathbf{K}_{\Sigma}\mathbf{q}_{L}\left(t\right)=\mathbf{f}\left(t\right) \tag{1}$$

where the matrices $\mathbf{X}_{\Sigma}$ are of block structure including blocks describing rotor $\mathbf{X}_{R}$ and blocks connected with bearings $\mathbf{X}_{B}$. The position of the blocks is given by generalized coordinates vector $\mathbf{q}_{L}\left(t\right)$ including the subvector $\mathbf{q}_{R}\left(t\right)$ of nodal displacements and the subvector $\mathbf{q}_{B}\left(t\right)$ of lateral displacements of floating

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rings centres. The excitation forces vector $\mathbf{f}(t)$ contains the vectors of nonlinear bearing forces, the rotating unbalance vector, and the static load vector.

3 Application

For the simulation purpose, a real turbocharger rotor produced by the company ČZ a.s. was used. The turbocharger rotor is of total weight of $0.25\,\mathrm{kg}$. It is simplified to be decomposed on a mild steel shaft of total length $0.103\,\mathrm{m}$ and two rigid discs of total weight $198\,\mathrm{g}$ representing the turbine wheel (T) and the compressor wheel (C). The rotor configuration is shown in Fig. 1. The shaft of the rotor is discretized using 11 nodes to 10 finite elements, each of them is of circular cross-section area. In the firmly mounted bearing housing, the turbocharger rotor is supported by two oil lubricated floating ring bearings marked $\mathrm{B_T}$, $\mathrm{B_C}$. The positions of bearing supports correspond with nodes of discretization 3 and 6, respectively.

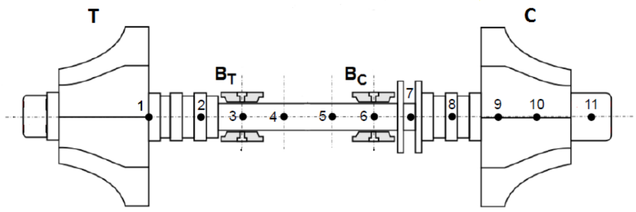

Figure 1: Turbocharger rotor-bearing system scheme.

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In order to determine the influence of bearing parameters, two states given by varying dimensions of bearing components due to the thermal expansion were considered. For the first state, the cold component dimensions were considered. The second state is given by dimensions changed due to high temperatures acting during the operation. For both cases, the lubricant temperatures in all oil films are assumed to be constant. The lubricating layers of B_T are of temperatures $\mathrm{T_I^T} = 140^{\circ}\mathrm{C}$, $\mathrm{T_O^T} = 120^{\circ}\mathrm{C}$ and lubricating layers of B_C are of temperatures $\mathrm{T_I^C} = 110^{\circ}\mathrm{C}$, $\mathrm{T_O^C} = 100^{\circ}\mathrm{C}$. Other bearing parameters can be found in [2]. The analysis was performed in operating speed range from 60 krpm to 110 krpm.

4 Results and discussion

Based on described procedure, a computational model was derived. The modelled turbocharger rotor-bearing system was investigated with respecting of the rotor unbalance. The two nodal mass unbalances Δ = 0.076 $\mathrm{g \cdot mm}$ are connected with the wheels and their relative positions are given by angle 180^o. Considering 11 nodes of discretization, the computational model given by Eq. 1 contained the generalizes coordinates vector $\mathbf{q}_L(t)$ of size 48x1. The numerical solution was implemented in the computational system MATLAB using the ODE15s library function. A simulation time shift corresponded to 500 times the length of one rotor rotation period for each chosen constant rotational speed from the defined speed range.

The dynamic behavior was assessed to consider on several aspects. Results of simulations provided time dependent values of lateral displacements of discretization nodes. Therefore, trajectories of bearing journal centers identical to nodes 3 and 6 and floating ring centers were evaluated. For the selected speed of rotation 100 krpm, the orbits are shown in Fig. 2 and Fig. 3, respectively.

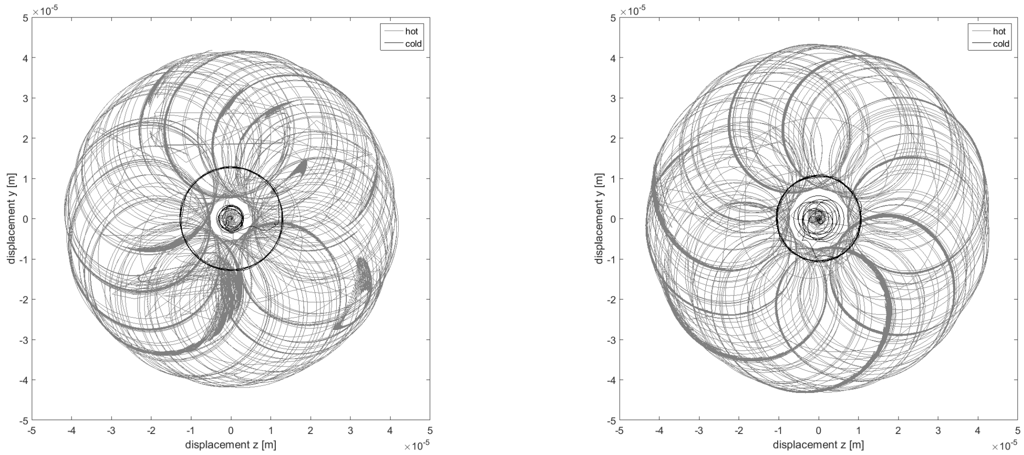

Figure 2: The orbits of journal centers of bearing $\mathrm{B_T}$ (on the left) and $\mathrm{B_C}$ (on the right) for rotational speed 100 krpm.

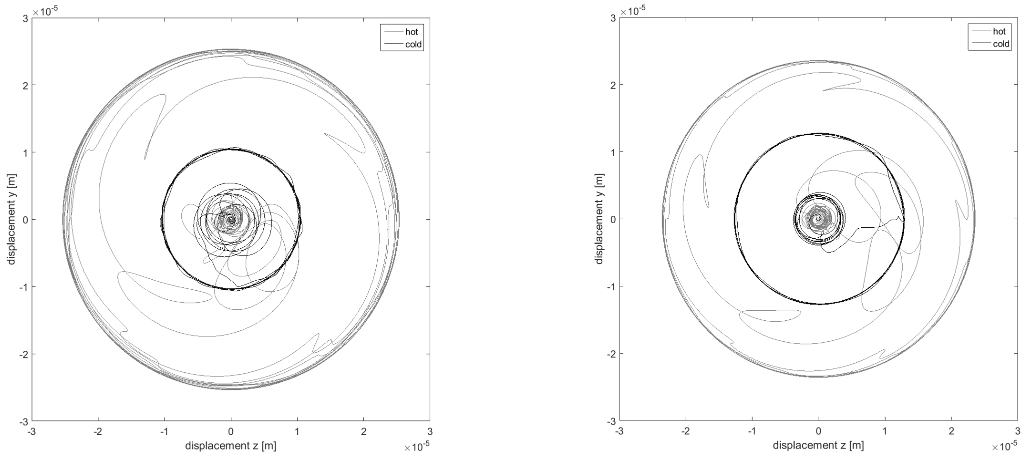

Figure 3: The orbits of floating ring centers of bearing $\mathrm{B_T}$ (on the left) and $\mathrm{B_C}$ (on the right) for rotational speed 100 krpm.

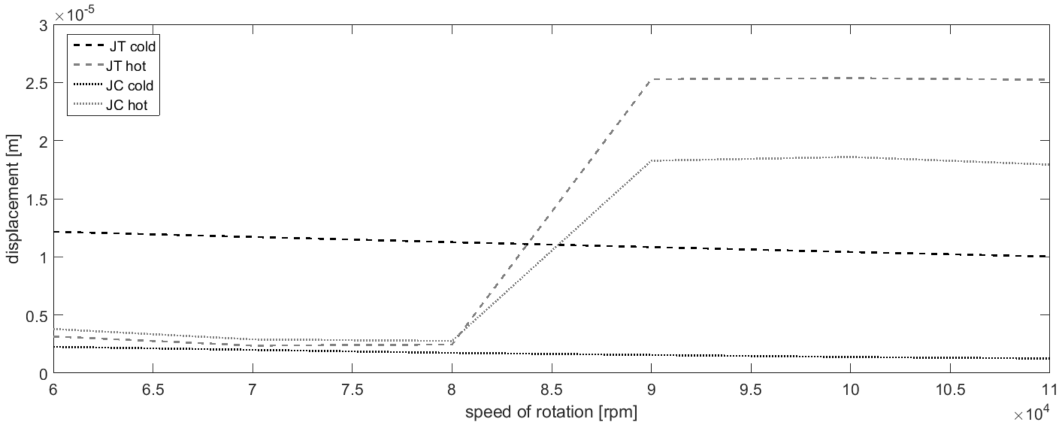

Figure 4: Dependence of maximal displacement magnitude of the journal centers.

Furthermore, the maximum displacement of journal centers at operating speeds for both bearing parameter variants was determined. Corresponding curves are shown in Fig.4.

5 Conclusion

On the basis of derived mathematical description of turbocharger rotor-bearing system the motion of turbocharger rotor was numerically investigated. In the operational speed range of automotive turbocharger, the several values of constant rotational speed was chosen to compare the amplitudes and trajectories of journal centers. The obtained computational results showed that the rotational speed affects the amplitudes of vibration. With the increasing speed of rotation the centers of the bearing journals for cold case are of lower displacement amplitudes. The hot case is more complicated. From 60krpm to 80 krpm and from 90 krpm to 110 krpm the journal center amplitudes decreased. Between 80 krm and 90.krpm the change in system characteristic occurred.Therefore, the purpose of further research is to identify the cause of the turbocharger rotor vibration amplitude change.

Acknowledgement

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