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Modelling and dynamic response of turbocharger

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1 Introduction

Turbochargers are nowadays widely spread in many applications, especially in engines. Few decades back the “TURBO” label on a passenger car symbolized dynamic, power, performance and speed. Today it can be better described by the words efficiency, effectivity and it is closely connected with the downsizing trend in modern engines.

Turbocharger basically consists of a radial inflow turbine, a centrifugal compressor and a centre housing of rotating assembly. Therefore, it could be considered as a system which contains both static and rotating subsystems. Nowadays it is common that the turbine operates in very high speeds, for example 170 000 revolutions per minute. Because of latter, it is very important to provide stable connection between static and rotating parts of turbocharger. This connection is established with two radial and one axial bearings.

The aim of this paper is to derive a sufficient linearized computational model of turbocharger considering fluid-film bearings with floating rings and to study the changes in oil film stiffness according to radial and angular velocity of the rotor and to examine dynamic response of this system.

2 Mathematical and computational model

To obtain a computational model of the rotor, a finite element method was used, see Byrtus et al. (2010). Thimoshenko’s beam-shaft elements were used for discretization of rotating parts with mounted rigid discs. Main challenge was the modelling of oil film force characteristics in the radial fluid-film bearings with floating rings (Fig. 1). In particular cases, the nonlinear force terms can be derived based on the Reynolds equation which describes transmitting hydrodynamic pressures by the fluid film. The derived nonlinear force terms can be advantageously linearized around the static equilibrium and further employed in a lumped spring model Zeman et al. (2013), see Fig. 1.

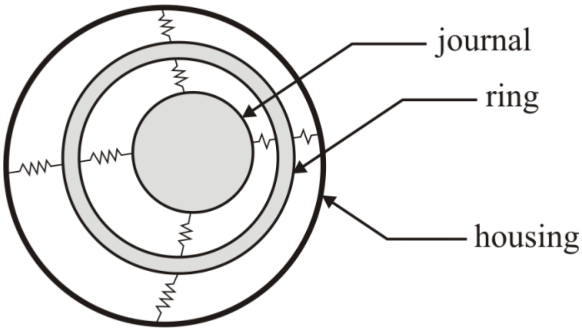

Figure 1: Linearized model of fluid-film bearing with floating ring

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The mathematical model of the complex rotor-bearing system was assembled by using Lagrange's equations and its final form states as follows

$$\boldsymbol{M}\ddot{\boldsymbol{q}} + \boldsymbol{\omega}\boldsymbol{G}\dot{\boldsymbol{q}} + (\boldsymbol{K}_s + \omega^2\boldsymbol{K}_\omega)\boldsymbol{q} = \boldsymbol{0} \quad , \quad \boldsymbol{M}, \boldsymbol{G}, \boldsymbol{K}_s, \boldsymbol{K}_\omega \in \mathbb{R}^{n,n} \quad , \quad \boldsymbol{q} \in \mathbb{R}^n, \qquad (1)$$

where $\boldsymbol{M}$ symbolizes the mass matrix of system, $\boldsymbol{G}$ is matrix of gyroscopic influences, $\boldsymbol{K}_s$ is matrix of static stiffness and finally $\boldsymbol{K}_\omega$ is matrix of softening during rotation. Vector $\boldsymbol{q}$ describes general coordinates and n corresponds to DOF number. External or material damping was not considered in this model.

The computational model was created, implemented and tested in MATLAB (R2013a). The model parameters correspond to real data of a turbocharger manufactured by ČZ a.s.

3 Results and conclusion

At first, modal analysis of conservative and non-conservative model was performed to verify the computational model, significant modal shapes were examined along with critical speeds using Campbell diagrams (Fig. 2). Further, the static equilibria depending on angular speed of the rotor was investigated. And finally, the dynamic response of unbalanced turbocharger rotor was studied. As a result, frequency spectrum and phase diagram was obtained.

Further, another analyses will follow - for example time dependent dynamic response using numeric integration or numerical implementation of nonlinear model.

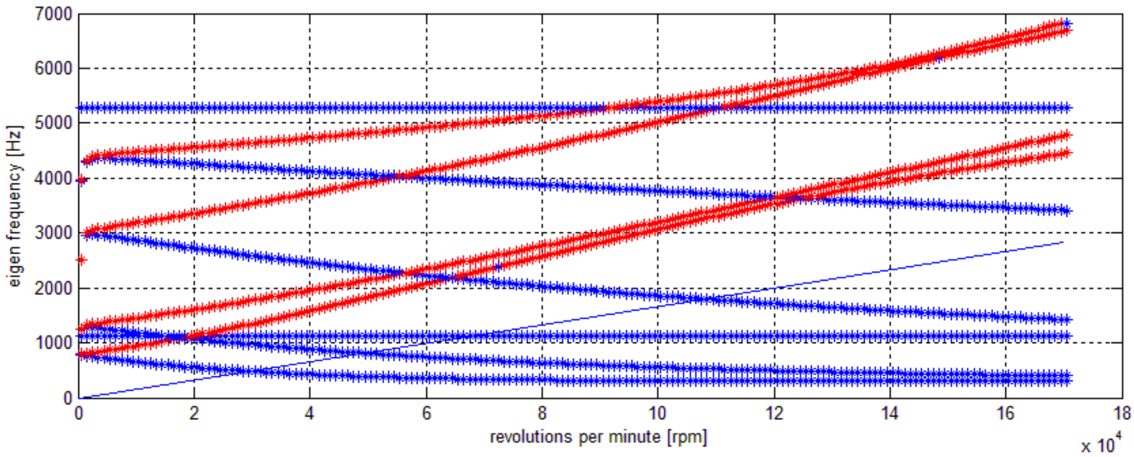

Figure 2: Campbell diagram of turbocharger rotor-bearing system

Acknowledgement

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