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Influence of thermally induced changes in lubricating gap clearance and oil viscosity on nonlinear oscillations of hydrodynamically supported rotors

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Despite many advantages, hydrodynamic bearings tend to induce nonlinear vibrations on rotor systems at low external loads. The occurrence and also the amplitudes of these vibrations are very sensible to the operating conditions of the system such as occurrence of gaseous cavitation, oil viscosity (strongly dependent on oil temperature), bearing clearance (dependent on metal temperature and thermal expansion coefficients) or rotor imbalance. This contribution shows the calculation of nonlinear rotor oscillations with the help of a detailed transient 3D-thermo-hydrodynamic bearing model. Referring to experimental data of a real exhaust turbocharger with semi-floating ring bearings, this holistic approach is used for a separate investigation of a) the influences of the temperature-dependent oil viscosity in the lubricating films and b) the influences of temperature-related changes in the lubricating gap on the nonlinear oscillations of the rotor.

1 Theoretical background

The hydrodynamic pressure in the lubricating films is calculated using the Reynolds-equation and a mass-conserving cavitation algorithm, which was described in [1]. The resulting pressure field is used to compute the temperatures in the oil film by solving the three-dimensional energy equation and the heat conduction equations for the shaft and the bushing as stated in [2]. On the one hand, the Reynolds-equation and the energy equation are coupled through the hydrodynamic pressure and the lubricant fraction obtained from the cavitation algorithm. On the other hand, they are coupled through changes in oil viscosity and bearing clearance due to changes in operating temperature. The equations are incorporated in force elements for the software EMD (Enhanced Multibody Dynamics[1]) and are then applicable to simulate systems of flexible rotors supported by hydrodynamic bearings. All equations are solved iteratively in the time domain to include relevant nonlinear effects and to model the coupling between thermal and mechanical regime correctly.

2 Case study: Turbocharger rotor supported by semi-floating ring bearings

2.1 Simulations with full thermo-hydrodynamic bearing model

Measurement data of the shaft movement of a real exhaust turbocharger rotor was available for two different oil inlet temperatures and showed significant differences as depicted in Fig. 1(a) and Fig. 1(c), respectively. While all other operating conditions remained very similar, the measurement with higher oil supply temperature showed the occurrence of subsynchronous vibrations as opposed to the measurement with low oil supply temperature, which only showed speed-synchronous oscillations as a result of the static imbalance at the impellers of the rotor. The difference between $T_{\text{oil supply,cold}}$ and $T_{\text{oil supply,hot}}$ translates into a viscosity ratio of the oil at the inlet of $\eta_{\text{cold}}/\eta_{\text{hot}} = 3.14$.

Simulations of this rotor were carried out with the detailed thermo-hydrodynamic bearing models outlined above. Therein, a run-up of the rotor was calculated within a time integration scheme starting from centred position at a low rotational speed $n_{\text{Rotor, start}}$ and ending at the maximal rotational speed of the turbocharger $n_{\text{Rotor, max}}$. In this model, the equation of motion of the rotor is solved including an online solution of the describing partial differential equations (PDE) of the bearings (Reynolds PDE, energy equation and heat conduction equation). This leads to an accurate representation of the bearing characteristics at every possible position of the rotor in the gap clearance, which manifests in very good accordance between measurement and simulation for both oil supply temperatures (cf. Fig. 1).

2.2 Investigation of the individual influences of temperature-dependent bearing properties

The measurements presented in the previous section only differ by the oil supply temperature. Therefore, temperature dependent bearing characteristics, especially lubricant viscosity η and gap clearance c, must have an influence on the dynamic behaviour of the investigated rotor. This can be examined using the enhanced thermo-hydrodynamic calculation routines for the bearings.

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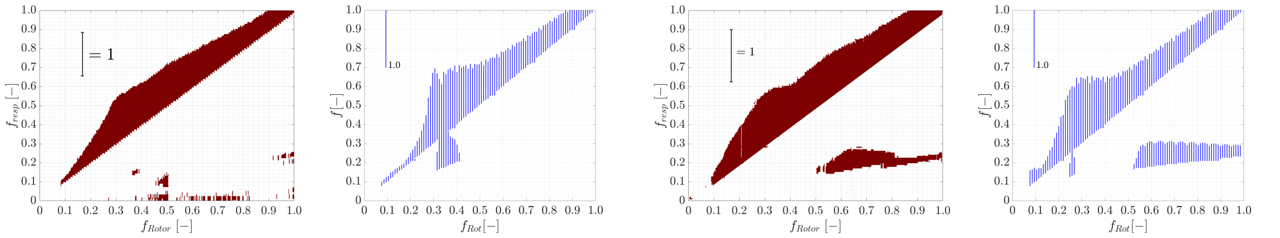
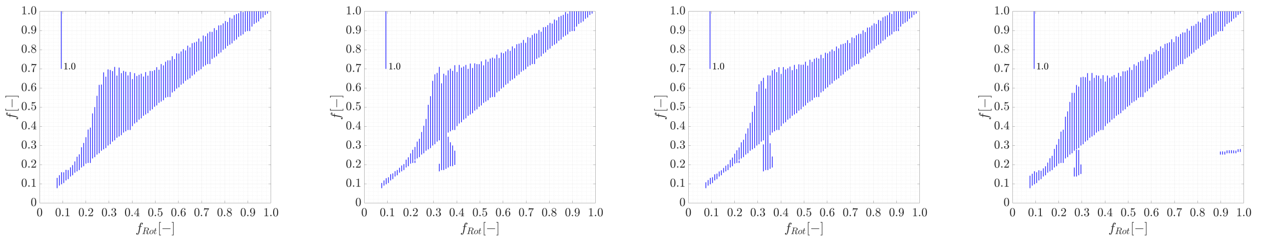


(a) Measurement, $T_{\text{oil supply,cold}}$ **(b)** Calc. 3D-THD bearing model **(c)** Measurement, $T_{\text{oil supply,hot}}$ **(d)** Calc. 3D-THD bearing model

Fig. 1: Comparison between measurements and simulations for the two oil supply temperatures $T_{\text{oil supply,cold}}$ and $T_{\text{oil supply,hot}}$



(a) Sim. 1b with η from $T_{\text{oil supply,hot}}$ **(b)** Sim. 1b with c from $T_{\text{oil supply,hot}}$ **(c)** Sim. 1d with η from $T_{\text{oil supply,cold}}$ **(d)** Sim. 1d with c from $T_{\text{oil supply,cold}}$

Fig. 2: Influences of thermally dependent viscosity η and bearing clearance c at $T_{\text{oil supply,cold}}$ (Figs. 2a, 2b) and $T_{\text{oil supply,hot}}$ (Figs. 2c, 2d)

Figs. 2(a) and 2(b) show investigations with respect to the simulation in Fig. 1(b), where the oil viscosity η (Fig. 2(a)) and the bearing clearance c, respectively (Fig. 2(b)), are exchanged by the values calculated at $T_{\text{oil supply,hot}}$. In the first case, the subharmonic vibration between $f_{Rot} = 0.3 - 0.4$ disappears, whereas in the second case, the changes in rotor movement are negligible compared to the original spectrogram shown in Fig. 1(b). Neither of the two simulations show an occurence of the prominent subharmonic vibration, which was recorded in the measurement at $T_{\text{oil supply,hot}}$.

In the reversed cases shown in Figs. 2(c) and 2(d), the major difference compared to the simulation depicted in Fig. 1(d) is the disapperance of the subharmonic vibration starting $f_{Rot} = 0.5$. The simulation result Fig. 2(c) shows a slight increase regarding the amplitudes and the intervall of occurrence of the first subharmonic vibration, whereas in the simulation result Fig. 2(d), a small fraction of the second subharmonic vibration is still present in an intervall of $f_{Rot} = 0.9 - 1.0$.

3 Conclusions

It can be concluded, that the nonlinear, subharmonic vibrations recorded in measurements of this rotor-bearing system can only be predicted correctly if both of temperature-dependent quantities (η, c) are specified for the correct temperature niveau. Compared to the influence of the global temperature niveau, the local changes of η and c due to heating during the operation of the turbocharger seem to be of minor influence in this example. Changing the bearing clearance as shown in Figs. 2(b) and 2(d) yields slightly smaller changes whith regard to the reference calculations in Fig. 1(b) and Fig. 1(d).

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