Received: 9 June 2020 | Accepted: 25 November 2020

DOI: 10.1002/pamm.202000053

Influence of lubricant film cavitation on the rotor dynamic system behaviour of an exhaust gas turbocharger rotor

Christian Ziese[1,*], **Cornelius Irmscher**[1], and **Elmar Woschke**[1]

[1] Institute of Mechanics, Otto von Guericke University, 39106 Magdeburg, Germany

The vibration behaviour of fast rotating rotors is significantly influenced by the bearing properties. Lubricant film induced excitations can cause sub-synchronous rotor oscillations known as oil-whirl and oil-whip phenomena. The non-linear bearing properties depend primarily on the lubricant properties, kinematics of bearing partners and especially on the occurrence of cavitation. Outgassing processes lead to a two-phase flow consisting of gas and oil, which can influence the bearing stiffness and damping respectively and consequently the rotor response behaviour. In this contribution, the oscillations of a semi-floating ring supported turbocharger rotor are investigated under the influence of lubricant film cavitation. For this purpose, run-up simulations are carried out under the assumption of mass-conserving cavitation according to the two-phase model and compared with measurements. In order to illustrate the influence of outgassing processes, a comparison is made with non-mass-conserving cavitation theory according to the assumptions of Half-Sommerfeld.

1 Introduction

Hydrodynamic bearings are preferably used to support fast rotating rotors. With respect to their properties, they exhibit a highly non-linear behaviour, which can lead to additional rotor excitations in terms of sub-synchronous oscillations. The bearing properties depend on the kinematic states of bearing partners, thermo-hydrodynamic conditions and the occurrence of cavitation. Outgassing processes lead to simultaneous occurrence of gas and oil in the lubrication gap and thus have an influence on bearing stiffness and damping. In order to model outgassing processes, the two-phase model is applied as a mass-conserving cavitation theory. The model provides the opportunity to consider phase transitions influenced by hydrodynamic pressure and lubricant film temperature. In this contribution, the influence of cavitation on the rotor dynamic system behaviour of a turbocharger rotor is investigated and compared with run up measurements.

2 Theoretical basics

For modeling of an incompressible laminar flow in journal bearings, the Reynolds differential equation

$$0 = \underbrace{-\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right) - \frac{\partial}{\partial y}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial y}\right)}_{\text{Poiseuille-flow}} + \underbrace{\frac{\partial}{\partial x}\left(\rho h \frac{u_s + u_h}{2}\right)}_{\text{Couette-flow}} + \underbrace{\frac{\partial}{\partial t}(\rho h)}_{\text{Squeeze film flow}} \tag{1}$$

can be used, whereby the flow components consisting of a Poiseuille-, Couette- and squeeze film flow are considered. Accordingly, a fluid can flow if a pressure gradient (Poiseuille flow) occurs at a control volume. Furthermore, due to the relative movement between bearing partners, a Couette flow can appear and the oil is transported into the narrowing lubrication gap. Transient processes are taken into account via the squeeze film flow, which is essential for bearing damping. Eq. 1 is solved numerically by using the finite volume method and online in a time integration scheme.

In order to take outgassing processes into account, the two-phase model is applied, which considers the total gas mass m_B consisting of a separate gas phase $m_{B\ undis}$ and gas dissolved in the lubricant film $m_{B\ dis}$

$$m_B \quad = \quad m_{B\ dis} + m_{B\ undis} = \left(r + \alpha_B \frac{p}{p_0}\right)\frac{V_{oil}\, p}{R\, T} \quad . \tag{2}$$

Therefore, the ideal gas law and Bunsen solubility (Henry-Dalton law) are used to determine the total gas mass. For calculating the lubricant fraction at current operating condition, a comparison is made between reference m_0 and current m_B state. With knowledge of the reference temperature T_0, reference pressure p_0 and reference bubble content r_0, the bubble content r and accordingly the lubricant fraction F can be calculated as a function of the hydrodynamic pressure p and lubricant film temperature T

$$F = \frac{1}{1+r} = \frac{p}{(r_0 + \alpha_B)\, p_0 \dfrac{T}{T_0} + (1-\alpha_B)\, p} \quad . \tag{3}$$

Subsequently, the lubricant properties in Eq. 1 can be scaled via the lubricant fraction. The occurrence of cavitation thus has an influence on hydrodynamic pressure distribution.

* Corresponding author: e-mail: christian.ziese@ovgu.de, phone: +49 391 67-52885

3 Results of run up simulation

The investigated turbocharger rotor is supported by floating ring bearings, whereby the inner gap has a multi-lobe geometry and the outer lubricating film is designed as a squeeze film damper[1]. The results of run-up simulations with respect to the cavitation theory of two-phase model and Half-Sommerfeld-Solution respectively and the shaft motion measurement are shown in Fig. 1. The run-up simulations were executed by using EMD[2].

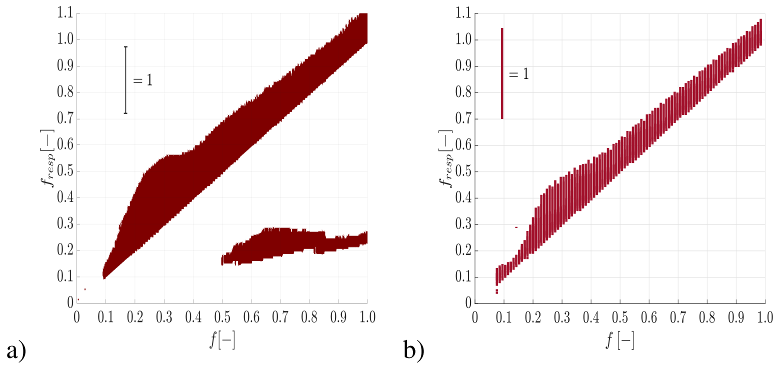

Fig. 1: Frequency spectrum of shaft motion: a) measurement, b) simulation with Half-Sommerfeld solution, c) Two-Phase Cavitation

Concerning the shaft motion measurement, one sub-synchronous rotor response was detected from $f = 0.5$. In contrast, the run-up simulation under the assumption of Half-Sommerfeld-Solution does not show any oil-whip. Due to the fully filled lubrication gap regardless of bearing operating condition, the bearing stiffness and damping are overestimated. On the other hand, the run-up simulation taking into account outgassing processes shows sub-synchronous vibrations from $f = 0.6$. Outgassing can thus lead to a softer and more realsitic bearing behaviour, since the lubrication gap is only partially filled with oil, and consequently sub-synchronous oscillations can occur.

In addition to the evaluation of the frequency spectrum, the pressure distribution and lubricant fraction at compressor bearing is shown at $f = 0.6$, see Fig. 2. There, three pressure areas are established at inner lubricating gap due to the multi-lobe geometry. The lubricant distribution shows that the minimum lubricant fraction occurs at the bearing edge, since the oil can flow out freely. In contrast, the outer lubrication gap is sufficiently supplied with oil.

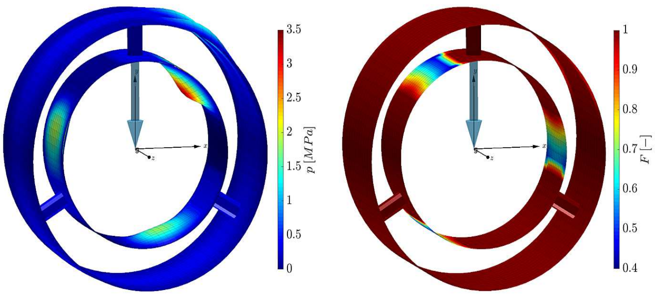

Fig. 2: Pressure and lubricant fraction distribution at compressor sided bearing at the beginning of sub-synchronous oscillations $f = 0.6$

Acknowledgements Open access funding enabled and organized by Projekt DEAL.

References

[1] Woschke, E.; Daniel, Chr. ; Nitzschke, St.; Excitation mechanisms of non-linear rotor systems with floating ring bearings - simulation and validation, International Journal of Mechanical Sciences, 2017, https://doi.org/10.1016/j.ijmecsci.2017.09.038

[2] Nguyen-Schäfer H.; Nonlinear Rotordynamic Computations of Automotive Turbochargers Using Rotating Floating Ring Bearings at High Rotor Speeds, International Conference on Vibrations in Rotating Machines, 2013

[1] There is no Couette-flow at the outer lubrication gap, because the rotational movements of the bushing are prevented. (semi-floating ring bearing)

[2] Extended multi-body dynamics: Program system, developed by RDU-GmbH and Otto von Guericke University

