Numerical run-up simulation of a turbocharger with full floating ring bearings

E. Woschke, C. Daniel, S. Nitzschke, J. Strackeljan

Institut für Mechanik, Otto-von-Guericke Universität Magdeburg, Germany
e-mail: elmar.woschke@ovgu.de
e-mail: christian.daniel@ovgu.de
e-mail: steffen.nitzschke@ovgu.de
e-mail: jens.strackeljan@ovgu.de

Abstract The paper deals with the numerical simulation of stability problems of the bearings of a turbocharger model. Therefore a full-dynamical run-up of the turbocharger shaft up to operating speed is simulated. The usage of multi-body-systems (MBS) for the numerical simulation of complex dynamically loaded systems is well established. The formulation of the acting nonlinear force law, given by the Reynolds' differential equation, can be achieved by usage of a user defined force element. The full floating ring bearings implicate special problems of coupling of inner and outer lubricating film. The communication drill-holes in the full floating ring ensure the oil supply of the inner lubricating film and lead to a coupling of pressure. The integration of these boundary conditions is discussed by usage of finite differences as well as finite elements for the solution of Reynolds’ differential equation.

Key words: Multibody dynamics, numerical run-up, turbocharger, full floating ring bearing

1 Introduction

The permanent demand on reduction of fuel by simultaneous increase of engine power leads to downsizing in combination with high-performance concepts. Turbochargers can be applied to enhance the effective mean pressure. Thereby the turbine transfers the energy of the exhaust emission into rotational energy, which is used to increase the charging pressure by a compressor. From the thermodynamical point of view high rotational speed is the aim of technical engineering. Whereas from the mechanical point of view moderate rotational speed shall be aimed for, if floating bearings are used. The reason can be found in critical instabilities at higher rotational speed [3]. Because of these contrary aims full floating ring bearings are applied, which primary lead to an increasing allowable rotational speed range. Nev-
et however there are still instabilities in the bearing leading to high levels of vibration and massive acoustic problems.

2 Model setup

The turbocharger consists of a turbine and a compressor, whose blade wheels are mounted on a common shaft. The shaft is supported by two identical full floating ring bearings against the housing. Housing and bushing form the outer floating bearing, whereas bushing and shaft form the inner one. The outer lubrication film is supplied with fresh oil (90°C, 3.5bar) by an oil delivery drill-hole. The oil supply of the inner lubrication film is ensured by four communication drill-holes.

The MBS-model is assembled by six bodies: the housing, the turbine and compressor blade wheels, the shaft and two bushings. For further simulations the shaft can be modelled by usage of beam elements as an elastic body. The blade wheels are fixed on the shaft, what leads to a rotating assembly with six degrees of freedom, whereas the bushings have two translational and one rotational degree of freedom each (no inclination). For a run up simulation of four seconds real time the rotational speed is preset linear $n_S = 0 \ldots 100000\text{min}^{-1}$. The discription of a full floating ring bearing in MBS can be achieved under usage of a user written force element. This

1 commercial MBS-program SIMPACK: uforce - User manuel SIMPACK 8.901
Numerical run-up simulation of a turbocharger with full floating ring bearings

$u_{force}$ defines the interface between rotor- and hydrodynamics, Figure 1. This leads to the following equation of motion Eq. (1)

$$M(q) \cdot \ddot{q} + b(q, \dot{q}) = h(t, q, \dot{q}),$$  \hspace{1cm} (1)

with the mass matrix $M(q)$, the vector of gyroscopic forces and torques $b(q, \dot{q})$ and the vector $h(t, q, \dot{q})$, where the implementation of the nonlinear lubrication film forces and torques is performed in.

3 Hydrodynamics

The hydrodynamic pressure generation in a floating bearing is described by the Reynolds’ partial differential equation (R-PDE).

$$\frac{\partial}{\partial x} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial z} \right) = 6(U_1 + U_2) \frac{\partial h}{\partial x} + 12 \frac{\partial h}{\partial t},$$  \hspace{1cm} (2)

The solution of Eq. (2) can be accomplished within the hydrodynamic module in different qualities, varying from analytic solution of the short bearing theory over grid interpolation via the impedance method up to the numerical solution of the R-PDE in every time step. The advantages of the short bearing theory and the impedance method are determined by short simulation times. The usage is restricted to parallel lubrication gaps. Caused by the high clearance of the outer bearing the rotating assembly is able to take a skew position. Hence Eq. (2) has to be discretized under usage of the finite difference method (FDM) or finite element method (FEM) and solved numerically [1]. The implementation of the boundary conditions is of importance. Beside the Dirichlet- and the periodical boundary conditions the coupling of both lubrication films via the communication drill-holes has to be taken into account. To avoid interpolations, this leads to a description of both lubrication films with the bushing as the reference system. Further the fresh oil supply at the outer lubrication film has to be modelled in an adequate way. Another point is the implementation of cavitation effects, microhydrodynamics and turbulence, which are neglected in this paper.

3.1 Solution by usage of FDM

The application on the lubrication film of a floating bearing leads after implementation of the boundary conditions to a linear system of equations Eq. (3)

$$K \cdot p = r,$$  \hspace{1cm} (3)
with the symmetric positive definite conductivity matrix $K$, the vector of unknown pressures $p$ and the inhomogeneous part $r$. The boundary conditions are realised by implementation of penalty factors. To model the communication drill-holes the discretization of the inner and outer lubrication film has to be equidistant, Figure 2.

![Fig. 2 Equidistant discretization on a communication drill-hole](image)

Due to the different inner and outer radius of the bushing different numbers of states occur in the inner and outer lubrication mesh.

### 3.2 Solution by usage of FEM

Finite elements represent an alternative strategy to discretize the R-PDE. Based on the variation functional Eq. (4), which is equivalent to Eq. (2)

$$
\Pi = \int_{G} \frac{h^3}{\eta} \left( \frac{\partial p}{\partial x} \right)^2 + \frac{h^3}{\eta} \left( \frac{\partial p}{\partial z} \right)^2 + 12(U_1 + U_2) \frac{\partial h}{\partial x} p + 24 \frac{\partial h}{\partial t} \rho dG,
$$

the solution can be achieved by minimization of Eq. (4).

$$
\frac{\partial \Pi}{\partial p} = 0 \quad (5)
$$

$$
u = \sum_{i=1}^{n} N_i \cdot u_i \quad \text{mit} \quad u = h, \dot{h}, p \quad (6)
$$

The resulting isoparametric elements are capable to mesh almost arbitrary areas. The evaluation of Eq. (5) with the ansatz Eq. (6) leads to a linear system of equations according to Eq. (3). The advantage of FEM against FDM is determined by an advanced modeling of the boundary of the communication drill-holes. The only limitation is given by the necessity to mesh the circumference of the drill-holes equidistant on the inner and outer lubrication film to get dedicated pairs of nodes.

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2 The penalty factor $\alpha$ is chosen depending on the major entry on the principal diagonal of $K$, whereby a compromise between computation time and accuracy of the boundary conditions has to be found.
4 Influence of coupling boundaries

To compare the simulation results between FDM and FEM the pressure distribution at a given time step is examined exemplarily.

![Pressure distribution](image)

Fig. 3 Pressure distribution (top: inner gap, bottom: outer gap, left: FDM, right: FEM)

Figure 3 shows nearly identical pressure distributions. Minor changes appear in the area of the communication drill-holes. As a result of the FEM formalism they are not part of the solution space, whereas due to the FDM formalism they are discretized. These different approaches lead, caused by different total areas, to minor changes concerning the friction torque

\[
T_F = \sum_{i=1}^{n} r_i \left( \int_{A_i} h_i \frac{\partial p_i}{\partial x} + (U_2 - U_1) \frac{\eta}{h_i} \right) dA
\]

(7)

and result in slightly different rotational speeds of the bushing, Figure 4. The FDM and the FEM approach determine respectively the lower and upper boundary of the real behaviour, which is only predictable by the solution of the three dimensional Navier-Stokes equation. This can not be realized due to computational time restrictions.
5 Conclusion

The general influence of these coupling boundaries can be clarified e.g. by integral quantities like the bushing rotational speed.

![Fig. 4 bushing rotational speed of a run-up simulation](image)

The plotted curves in Figure 4 show in principle a similar behaviour concerning the run-up. There are three changes of the slope, which characterize changes of performance of the full floating ring bearing. The last one indicates a total instability of the system [4]. The figure points out, that the coupling boundaries lead to a general decrease of the characteristic frequencies and dominate the prediction of the total instability.

The verification of the simulation results can only be achieved by comparison to measurements. First measurements [2] of shaft displacement show good correlations to the simulation results. There are unconsiderations like temperature rise and elasticity of the shaft, which explain different results. For a complete verification additional datas like rotational speed of the bushing, temperature rise and the distribution of unbalance have to be measured.

References