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APPLICATION OF MULTIBODY DYNAMICS TOOLS FOR DETERMINATION OF ENGINE EXCITED TURBOCHARGER VIBRATIONS

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Abstract: *The problem of turbocharger rotor vibration is widely published in the world literature. However, areas requiring more detailed investigation are the vibration caused by the kinematic excitation of the turbocharger housing including the housing stiffness consideration. The aim of this paper is to analyse the vibration caused by the excitation from the internal combustion engine (ICE) and to present vibration on a key component of the turbocharger, which is a radial hydrodynamic bearing. Due to the nature of the problem, a multibody dynamics based tool is used, supplemented with sub-models of bearings. The non-linear model allows an effective solution of the vibration induced by the existence of rotor resonance states, rotor instabilities of the bearing or excitation from the ICE. The results presented on the radial bearing show that additional side frequency components and their orders appear in the frequency response spectrum. In the state of strong excitation, these frequency components can cause an increase in the acoustic emissions of the turbocharger as well as the entire powertrain.*

Keywords: Rotor dynamics, Engine excitation, Turbocharger, Vibroacoustic.

1. Introduction

Modern means of transport, including cars, trucks or ships, are currently mainly based on powertrains incorporating turbocharged internal combustion engines (ICE). Turbochargers are therefore an important part of these powertrains and often have a great influence on the vibro-acoustics of the means of transport. Turbocharger vibrations are often induced by excitations due to rotor unbalance or rotor instabilities in hydrodynamic bearings. Increased vibrations are often associated with unstable rotor behaviour and can lead to damage to the turbochargers during operation. The main causes of damage may be the contact between rotor and housing or bearing damage due to the contact between shaft and bearing shell. These powertrains then require an increased level of operational maintenance costs, as reported by Stodola et al. (2019).

Increased vibrations are caused mainly by the existence of resonant states of the rotor and instabilities of the rotor mounted in radial hydrodynamic bearings. Maximum deflections during excitation are achieved in resonance, and these depend on the characteristics of the rotor, such as mass, mass moments of inertia, or rotor and bearing stiffness and damping. Unlike resonance, rotor deformations in unstable states increase extremely with time and is independent of rotor speed. Once the rotor speed exceeds the instability threshold, the rotor becomes unstable and the rotor deformation increases with time without limitation. The rotor deformation continues to increase as the rotor speed changes. Rotor instability is caused mainly by oil whirling in radial bearings. When the rotational speed of the rotor reaches approximately two times of the rotor natural bending frequency, the oil whirl changes to the oil whip. With oil whip, the rotor becomes unstable and the rotor deflection increases steadily until the bearing eccentricity reaches the bearing clearance, resulting in damage to the turbocharger radial bearings.

The excitation of the rotor due to ICE vibrations is mostly in the sub-synchronous region rather than in the super-synchronous region in frequency point of view with respect to rotor speed. In the case of ICE, the

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dominant excitation can be derived from the engine speed and the corresponding orders of the engine speed. The specific spreading rate of individual orders of the ICE speed depends mainly on its design, i.e. mainly on the cranktrain design. Each design is excited by the firing frequency and its corresponding orders. Practically, it is accurate enough to consider only orders of approximately up to three.

2. Methods

2.1. Review of State of the Art in Rotor Dynamics Modelling

Rotor dynamics is being extensively solved in the world literature, especially in cases of stationary stator. In this context, the literature includes a considerable amount of sources that study a wide range of phenomena associated with the dynamic behaviour of rigid or flexible, linear or non-linear rotor systems. Numerous studies have utilized the Finite Element Method (FEM) and applied it to modelling rotor dynamics. Nelson and McVaugh (1976) introduced one of the first rotor models using FEM, while considering influences of gyroscopic effects. Other approaches gradually expanded the computational models and enabled modelling of various processes occurring in rotor dynamics.

An important criterion for evaluation of a rotor is its stability. Various criteria are used to analyse rotor stability, for example, Nguyen-Schäfer (2015) shows the use of the Routh-Hurwitz criteria on the Jeffcott rotor model for linear rotor dynamics or Hopf bifurcation theory for evaluating the stability of nonlinear rotor dynamics. Briend et al. (2020) shows the use of Floquet theory to assess the stability of nonlinear rotor dynamics so he is able to describe stable and unstable areas.

Hydrodynamic bearings play a key role in the behaviour of rotor systems. The dynamic behaviour of the bearing and accurate prediction of rotor vibration is conditioned by a suitable computational model of the bearing. The properties of hydrodynamic bearings represented by the force reactions in the lubrication layer are strongly non-linear in regard to the pin displacement and velocity.

The issue of a rotor excitation due to rigid support movements is reported much less frequently. Dakel et al. (2012) described a dynamic rotor analysis with a kinematic excitation of a rigid support movement using a harmonic function, and in a further work by Dakel et al. (2014), there is described nonlinear dynamics of a turbocharger rotor with the influence of various type of excitations. An interesting multi-frequency parametric excitation of the rotor mount is also presented by Briend et al. (2020).

Contemporary computational approaches in rotor dynamics increasingly use commercial systems based on the multibody dynamics (MBD). These approaches allow complex non-linear transient models to be defined, while still allowing the implementation of custom sub-models. One of the first computational models using MBD is presented by Schweizer (2008). The MBD model of the engine turbocharger rotor of the commercial vehicle engine is presented by Novotný et al. (2018) and shows an innovated model of a radial hydrodynamic bearing with a floating ring and includes the influence of variable lubricant properties and the influence of lubricant temperature and lubricant grooves and bores. In another publication, Novotný and Škara (2019) present the application of the MBD model to the turbocharger rotor dynamics.

A very specific issue is the amplification of vibration due to the vibration of the turbocharger housing. This issue is not yet sufficiently described in the literature. It is obvious from the usual geometric shape of the housing that the computational model should use its three-dimensional (3D) discretized formulation, for example FEM.

2.2. Modelling of Turbocharger Dynamics

A computational model of a turbocharger in this paper is assembled in the software for multibody dynamics solution. This model includes body models of various levels, such as the mass point model used for unbalance, the rigid body model used for non-deformable bodies such as compressor and turbine wheels, or the 3D model of the flexible body to describe the rotor shaft and turbocharger housing. 3D elastic models are based on reduced FEM models, using Component Mode Synthesis (CMS) according to Craig (1981). The interaction between the rotor and the housing is solved by specialized sub-models of hydrodynamic bearings. The effective radial bearing model used for transient simulations is described by Novotný et al. (2018) and a similar thrust bearing model can be found in Novotný et al. (2019). The rotor axial load is calculated based on the measured pressures and temperatures of the turbine and the compressor inlets and outlets according to the procedure described by Nguyen-Schäfer (2015). The graphical representation of the computational model is shown in Fig. 1.

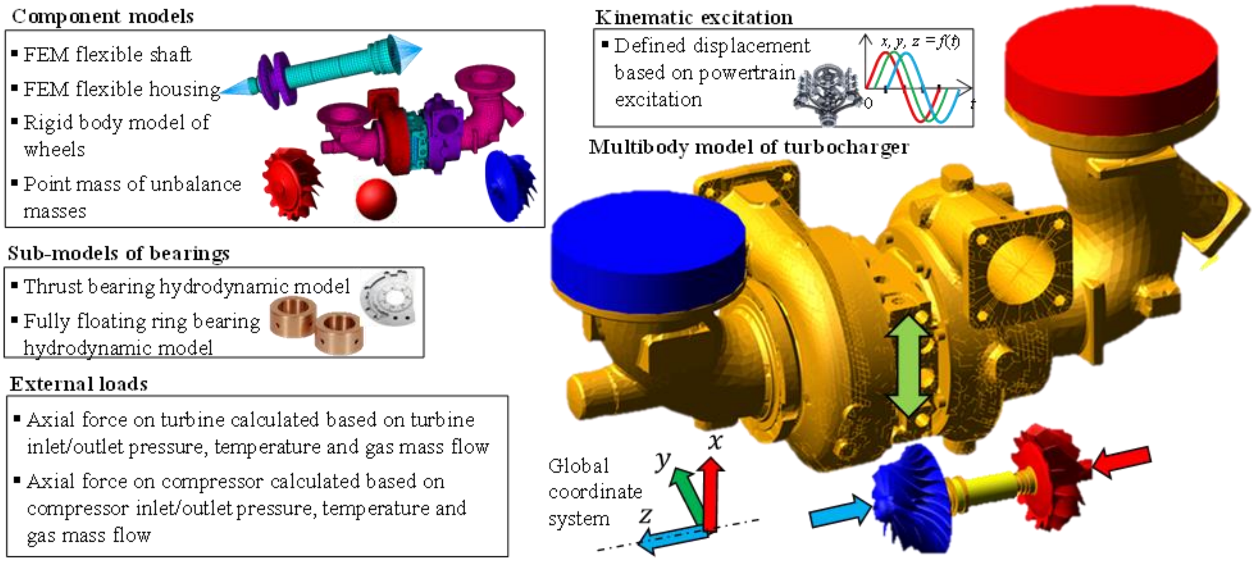

Fig. 1: Graphical representation of the multibody computational model. The model includes bodies, forces constrains, functions, variables and data structures.

The kinematic excitation is realized by the prescribed movement of the turbocharger housing in the form of velocity with a sine shape with amplitude $v_{\mathrm{a}} = 0.067\ \mathrm{m/s}$. The frequency of this excitation is increased in the range of $f = (0 \div 300)$ Hz over the time range $\Delta t = 10$ s, except for the initial time interval corresponding to 10 rotor revolutions necessary for movement stabilisation.

3. Result Discussions

In most publications it is assumed that the given kinematic excitation is directly transferred to the rotor mount. However, this is not exactly true. The turbocharger housing is flexible and thus vibrates depending on the excitation frequency. Therefore, the rotor deflection at bearing location can be increased or decreased.

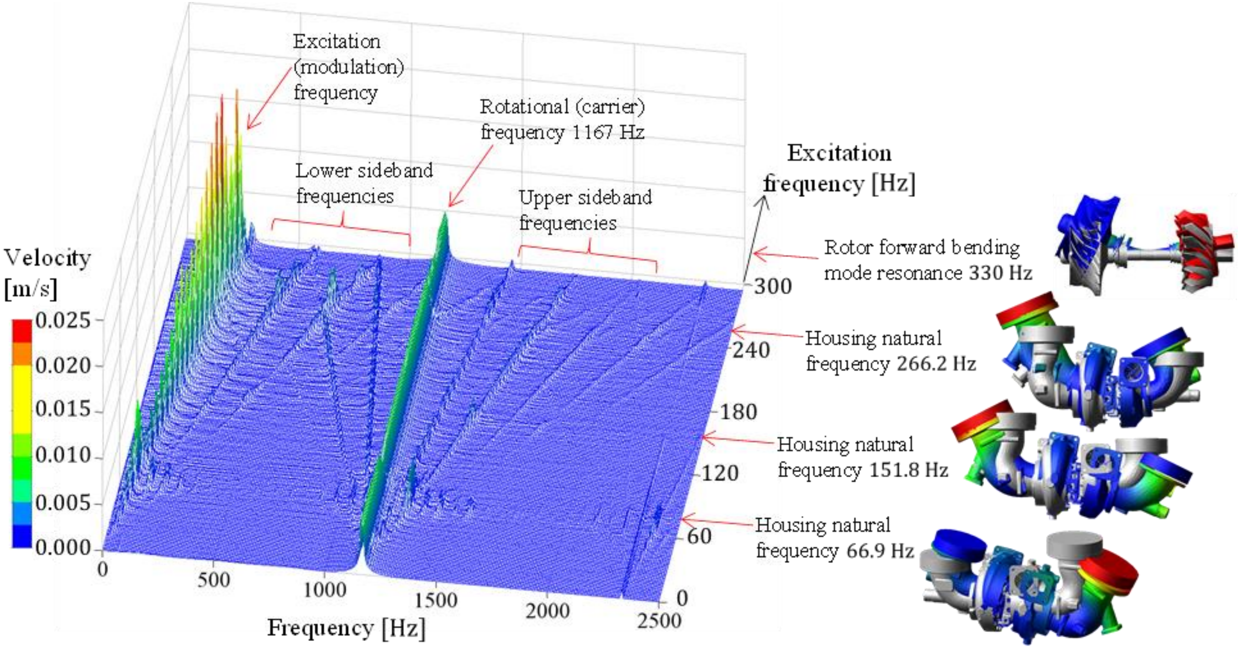

Fig. 2: Frequency spectrum of the floating ring velocity in the x-axis relative to the bearing shell on the compressor side. Three natural frequencies of the turbocharger housing and the first natural bending frequency of the rotor are indicated on the excitation frequency axis.

The turbocharger can be considered as a non-linear vibration system, and in the case of larger excitation amplitudes, these non-linearities become more evident. In the case where the system is driven by the carrier excitation frequency, due to rotor unbalance, and at the same time excited kinematically at the housing attachment, additional response components will appear in the frequency spectrum. These components can be observed, for example, on hydrodynamic bearings. Fig. 2 shows a frequency spectrum of the floating bearing ring in the x-axis relative to a bearing shell. In the frequency spectrum, the response is not affected only by the rotational frequency of the rotor, but also by the side frequency bands and their orders. At the same time, all frequency bands are amplified locally in places where the turbocharger housing resonates or where the rotor natural frequency is present. Fig. 2 also shows responses to up to 300 Hz excitation frequency, and it is in this region that the deflections begin to increase again because the excitation frequency is close to the rotor's bending frequency. This result is chosen because it demonstrates the frequency distribution caused by excitation of the turbocharger housing by ICE and it determines a turbocharger vibroacoustic performance.

4. Conclusions

The turbocharger as a non-linear system from a mechanical point of view causes a generation of side frequency components of vibrations in the acoustic spectrum. In conventional cases where the turbocharger housing excitation is not so dominant, these side frequency components may be insignificant and will not be reflected in the overall vibroacoustic spectrum of the turbocharger. On the other hand, in the case of strong housing excitation amplitudes, these components may already cause significant noise, and in some cases such vibration may lead to damage to the turbocharger rotor bearings.

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References

Briend, Y., Dakel, M., Chatelet, E., Andrianoely, M. A., Dufour, R. and Baudin, S. (2020) Effect of multi-frequency parametric excitations on the dynamics of on-board rotor-bearing systems. Mechanism and Machine Theory 145.

Craig, R. R. (1981) *Structural Dynamics*. First edition. John Wiley & Sons.

Dakel, M., Baguet, S. and Dufour, R. (2014) Nonlinear dynamics of a support – excited flexible rotor with hydrodynamic journal bearings. Journal of sound and vibrations, 333, pp. 2774-2799.

Dakel, M., Baguet, S. and Dufour R. (2012) Dynamic analysis of a harmonically excited on-board rotor-bearing system, in: 10th International Conference on Vibrations in Rotating Machinery, London, pp. 57-68.

Furch, J. (2016) A model for predicting motor vehicle life cycle cost and its verification. Transactions of FAMENA, 40, 1, pp. 15-26.

Nguyen-Schäfer, H. (2015) Rotordynamics of Automotive Turbochargers. Second edition. Springer, Heidelberg.

Nelson, H. D. and McVaugh, J.M. (2010) The dynamics of rotor bearing system using finite elements. ASME Journal of Engineering for industry, 98, pp. 593-600.

Novotný P., Hrabovský J., Juračka J., Klíma J. and Hort, V. (2019) Effective thrust bearing model for simulations of transient rotor dynamics. International Journal of Mechanical Sciences, 148, pp. 374-383.

Novotný, P. and Škara, P. (2019) Multibody Simulation of Turbocharger Transient Rotordynamics. in: Engineering Mechanics 2019. pp. 259-262.

Novotný, P., Škara, P. and Hliník, J. (2018) The effective computational model of the hydrodynamics journal floating ring bearing for simulations of long transient regimes of turbocharger rotor dynamics. International Journal of Mechanical Sciences, 148, pp. 611-619.

Schweizer, B., Rieger, N. F., Black, H. F. and Thomas Ch. B. (2010). Dynamics and stability of turbocharger rotors. Archive of Applied Mechanics, 80.

Stodola, P. and Stodola, J. (2019) Model of Predictive Maintenance of Machines and Equipment. MDPI Applied Sciences, 10, pp. 1-16.