THE INFLUENCE OF ROTOR UNBALANCE ON TURBOCHARGER ROTOR DYNAMICS

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SHRNUTÍ

Článek analyzuje vliv nevývahy na dynamiku rotoru turbodmychadla. Je zde prezentován 3D výpočtový model rotoru turbodmychadla a hydrodynamický model kluzné ložiska. Sestavení výsledného výpočtového modelu na úrovni virtuálního prototypu je provedeno v prostředí software pro řešení dynamiky těles. Článek prezentuje vybrané výsledky, na kterých je podrobně rozebrán vliv velikosti nevývahy na dynamiku rotoru turbodmychadla.

KLÍČOVÁ SLOVA: TURBODMYCHADLO, DYNAMIKA ROTORU, NEVÝVAHA, LOŽISKO S PLOVoucím POUZDREM.

ABSTRACT

This paper describes the influence of an unbalance on turbocharger rotor dynamics. The structural model of the turbocharger rotor and the hydrodynamic model of the journal floating ring bearing are described and assembled in multibody dynamics software. Moreover, the paper presents various results describing rotor dynamics where the influence of an unbalance is discussed.

KEYWORDS: TURBOCHARGER, ROTOR DYNAMICS, UNBALANCE, FLOATING RING BEARING.

1. INTRODUCTION

One of the modern methods for reducing emissions and increasing the power of combustion engines is by introducing a turbocharger into the power unit design. Since the demands being placed on combustion engines are continuing to rise, turbocharger manufacturers have to keep up. Turbochargers are currently made in large numbers and even the slightest simplification in the manufacturing process can lead to significant cost reductions.

One of the processes that have to be carried out during the turbocharger manufacturing process is balancing. Almost all rotating machinery is carefully balanced, and a turbocharger rotor is no exception. However, even the best balancing process cannot flawlessly balance the rotor, and there will always remain some residual unbalance. Moreover, the degree to which the residual unbalance influences the turbocharger rotor dynamics is discussed in this paper.

2. STATE OF THE ART

A turbocharger is quite a unique type of rotor-based machinery. One of its greatest challenges is its very high speed range. Modern turbochargers operate at speeds of up to $300\,000\text{ min}^{-1}$. Turbocharger rotor dynamics and its modeling were based on simple linear models, but this solution was not sufficient. A major breakthrough was introduced by Schweizer, who used

a multibody dynamics software (MBS) based tool for turbocharger rotor dynamics prediction. He published his model in [1] and [2]. This type of model is the most commonly used model worldwide for turbocharger rotor dynamics simulation. For example, Tian used this type of model to create a model to study the influence of rotor unbalance and engine vibration on turbocharger rotor dynamics, and published the results in [3] and [4].

3. TURBOCHARGER MODEL

The turbocharger model was assembled using an MBS software tool (namely MSC Adams). This enables a rotor model to be prepared and assembled with commercially available software tools, and only the rest of the models have to be prepared (e.g. floating ring bearing model).

3.2 ASSUMPTIONS OF THE JOURNAL BEARING MODEL

Since the journal bearing is represented in the MBS by a set of pre-calculated databases, several assumptions have to be considered to enable creation of the databases in advance. These assumptions are [5]:

1. the journal and bearing shell shapes are ideal cylindrical parts,
2. the journal and the bearing shell are rigid bodies without any deformations, an oil gap between the journal and the

shell is filled with the oil and the gap proportions are small in comparison with journal or bearing shell proportions,

3. only hydrodynamic friction can occur,
4. dynamic viscosity is constant for the whole gap volume and it is independent of local temperature and pressure,
5. inertia forces in the oil film are neglected,
6. lubricating oil is incompressible, and
7. oil flow is laminar.

3.2 BACKGROUND TO THE HYDRODYNAMIC JOURNAL BEARING MODEL

Although there are efforts to introduce ball bearings into the design of modern turbochargers, most modern designs use a journal bearing, i.e. semi or fully floating ring bearings.

Since the semi floating ring bearing can be represented as a fully floating ring bearing with disabled ring rotation, the rational conclusion is to model the fully floating ring bearing first. The fully floating ring bearing is shown in Figure 1. The bearing consists of three parts – housing bore (sleeve), shaft and floating ring. The floating ring separates the shaft from the housing bore; therefore, two oil films are created – inner and outer oil film. A floating ring bearing has three different eccentricities which describe the position of the floating ring and shaft – overall eccentricity e (between shaft and housing bore), inner eccentricity e_i (between shaft and floating ring) and outer eccentricity e_o (between floating ring and housing bore).

For precise rotor dynamics simulation all bearing parameters also have to be known – diameter, width and clearance for both inner and outer oil film, oil parameters – dynamic viscosity, oil input pressure and oil temperature.

Both of the oil films are treated separately as a plain journal bearing. The hydrodynamic pressure and consequential forces and moments are solved numerically using the Reynolds equation. The equation is based on the modification of the Navier-Stokes equation and continuity equation transformed for cylindrical shapes of the bearing oil gap. The full form of the equation has been simplified and modified into the following two equations [5]:

$$\frac{\partial}{\partial \varphi} \left(H^3 \frac{\partial \Pi_D}{\partial \varphi} \right) + \frac{D^2}{B^2} \frac{\partial}{\partial Z} \left(H^3 \frac{\partial \Pi_D}{\partial Z} \right) = 6 \frac{\partial H}{\partial \varphi}, \quad (1)$$

$$\frac{\partial}{\partial \varphi} \left(H^3 \frac{\partial \Pi_V}{\partial \varphi} \right) + \frac{D^2}{B^2} \frac{\partial}{\partial Z} \left(H^3 \frac{\partial \Pi_V}{\partial Z} \right) = 12 \cos \varphi, \quad (2)$$

where: H is the dimensionless height of the oil film, φ is the dimensionless angle around pin axis, Z is the dimensionless coordinate along bearing width, D is bearing diameter (inner D_i or outer D_o diameter depending on which oil film is modelled), B is bearing width (inner B_i or outer B_o width depending on which oil film is modelled) and Π_D and Π_V are dimensionless parameters (further described in [5]).

Equation (1) describes the hydrodynamic pressure formed by tangential movement of the pin, and equation (2) describes the hydrodynamic pressure formed by radial movements of the pin. The transition into the dimensionless form enables the equation to be solved effectively and also the solving of the equation for groups of bearing (depending on D/B ratio).

The Reynolds differential equation is discretized by the finite difference method and iteratively solved by the Gauss-Seidel method employing successive over-relaxation strategy (SOR). The main parameter that influences the distribution of the hydrodynamic pressure is the non-dimensional height of the oil film H . The non-dimensional height of the oil film, considering eccentricity and both tilt angles, is shown in equation [6]:

$$H = 1 + \left(\varepsilon - Z \cdot \gamma \cdot (1 - \varepsilon) \right) \cdot \cos \varphi + \delta \cdot Z \cdot \sin \varphi \cdot \sqrt{1 - \left(\varepsilon + (1 - \varepsilon) \cdot |\gamma|^2 \right)}, \quad (3)$$

FIGURE 1: General arrangement of the journal bearing model used: a) Floating ring bearing model, b) Floating ring bearing general dimensions

OBRAZEK 1: Situační schéma použitého modelu ložiska: a) Schéma ložiska s plovoucím pouzdrem, b) Obecné rozměry ložiska s plovoucím pouzdrem

where: ε is dimensionless eccentricity (inner or outer eccentricity depending on which oil film is modelled), φ is dimensionless angle around the pin axis, Z is a dimensionless coordinate along bearing width and γ and δ are dimensionless journal tilt angles. The hydrodynamic pressure, which resulted from the solution of the Reynolds equation, has two parts – a positive and a negative part. However, the oil film can effectively support only positive pressure and not the negative pressure – tension. Therefore, the cavitation condition has to be introduced into the bearing model. The cavitation condition ensures that the pressure in the oil film is always higher or equal to the cavitation pressure. Besides cavitation, the solution also considers the tilt of the journal (described by two dimensionless angles – δ , γ). This enables turbocharger rotor dynamics to be studied in detail and also provides a better understanding of the relationship between oil film instability and shapes of individual eigen-modes. The journal bearing model is introduced to the MBS as a set of dimensionless result tables and equations. The resultant table carries the values of dimensionless forces and torques. This approach enables solving of the distribution of the hydrodynamic pressure in advance; and therefore, secure a faster solution. To create the hydrodynamic databases, the solver needs the diameter to width ratio; every other parameter (e.g. bearing width, diameter and clearance, oil viscosity etc.) is then entered afterwards into the conversion to the real values of forces and torques.

3.3 ROTOR MODEL ASSEMBLY

The creation of the turbocharger rotor model is carried out in two steps. First, the rotor FE model, or its parts, is created in the FE software. The specific material characteristics have to be set in this step. The second step requires transferring the FE model from FE software into the MBS. This step is achieved via the Craig-Bampton reduction, which reduces the number of DOF. The last step is to finish the model in the MBS. There are two basic ways to create the turbocharger rotor model in the MBS. The first is a fully flexible model. The model is created by modal reduction of the whole rotor (shaft, compressor and turbine wheel and thrust collar). This requires setting the material characteristics of the whole rotor in FE software and this makes it more difficult to modify the design or material characteristics of the rotor.

The second way (shown in Figure 2) is to discretize only the shaft in the FE software. This leads to a flexible shaft, but the rest of the parts – compressor and turbine wheel, thrust wheel etc. – are considered rigid. The rigid parts are represented in the MBS by mass, tensor of inertia, centre of gravity location and optionally geometry. This allows quick incorporation of the change into the turbocharger rotor design (e.g. design of the compressor and turbine wheel).

FIGURE 2: MBS turbocharger rotor modelling approach
OBRÁZEK 2: Přístup k vytvoření MBS modelu rotoru

In this paper the second approach was chosen (shown in Figure 2). Since both wheels are represented in MBS as rigid bodies, their deformation is neglected. However, since their stiffness is much greater than the stiffness of the shaft, it is a feasible approach. Rigid wheels also do not allow analysis of blade vibration, but since their vibrations are mostly excited and affected by air/exhaust gases flow, the results would be misleading anyway. When the turbocharger rotor model has been completed, other parts of the turbocharger are added. If the floating ring bearing is used in the design, floating rings have to be considered in the model. Also, the model of the turbocharger housing has to be considered – either flexible or rigid. The rigid housing model can be used for general rotor dynamics analysis. The flexible housing is essential for noise and vibration analysis. The influence of engine vibrations on turbocharger rotor dynamics can also be examined by introducing excitation into the housing mounting points.

4. SIMULATION RESULTS

4.1 SIMULATION INITIAL CONDITIONS

Before the results are presented, it is necessary to set out the initial conditions. Initial conditions introduced into the turbocharger rotor dynamics simulation can be divided into several groups:

1. Rotor assembly conditions:
 - a. shaft is positioned in the middle of the floating ring ($e_i=0$),
 - b. floating rings are positioned in the middle of the housing bore ($e_o=0$), and
 - c. rotor speed is zero.

2. Loads acting on the rotor and bearing assembly (all of them are acting from the beginning of the simulation):
 - a. gravity acting in the vertical direction (rotor assembly axis of rotation lies in the horizontal plane),
 - b. oil pressure is 3.5 bar (the oil input pressure does not influence the rotor dynamics due to bearing model simplification, but it influences the bearing oil flow and oil warm up), and
 - c. rotor assembly speed increases linearly. The rotor assembly is driven and the speed is controlled through the auxiliary body and torsional spring (the rotor speed can also be increased by the torque, but from the point of view of speed control, it is better to define the rotor speed).
3. Housing conditions:
 - a. the housing is defined as a rigid body, and
 - b. housing bore misalignment is neglected.

4.2 INFLUENCE OF INDIVIDUAL PARAMETERS ON ROTOR DYNAMICS

Since most modern turbochargers are created by redesigning older ones, for turbocharger manufacturers it is important to have knowledge of how the change of an individual parameter will influence rotor dynamics of the turbocharger as a whole. Moreover, the design of a turbocharger, especially of the turbine and compressor wheel, is often adjusted to perfectly fit a specific engine. Since the turbocharger development relies on measurement, this results in a requirement to perform many necessary experiments, which is expensive and time consuming.

4.3 UNBALANCE

One of the closely observed parameters is the unbalance of the turbocharger rotor. Even though the manufacturers try to keep the unbalance as low as possible, there will always be some residual unbalance left. It is impossible to create a perfectly balanced rotor; therefore, the balance must be set as a compromise between the residual unbalance and costs.

The unbalance of the turbocharger rotor is modelled by two additional test masses in two planes on the rotor – the compressor and turbine back-plate, as shown in Figure 3. The unbalance is then defined in unit $g \cdot mm$.

4.4 INFLUENCE OF THE UNBALANCE ON ROTOR DYNAMICS

For proper discussion of the influence of the unbalance change on turbocharger rotor dynamics, first the behaviour of the basic state has to be known. Furthermore, suitable parameters describing rotor dynamics have to be chosen. For this purpose, the overall eccentricity and displacement on the turbine and compressor wheel nose were selected. The displacement on the wheel nose is then represented by a waterfall diagram.

FIGURE 3: The location of planes for addition of the test masses on the turbocharger rotor

OBRAZEK 3: Poloha rovin pro přidání testovacích hmot na rotoru turbodmychadla

The eccentricity of the floating ring bearing on the compressor side (the relative overall eccentricity) is shown in Figure 4. It can clearly be seen that the eccentricity is fairly low until the rotor speed reaches around 40 000 min^{-1} . Above this rotor speed, the relative overall eccentricity rises to 0.6.

The cause for the rise in the relative eccentricity above the speed of 40 000 min^{-1} can be seen in the waterfall diagram in

FIGURE 4: Original state relative overall eccentricity on the compressor side

OBRAZEK 4: Celková relativní excentricita ložiska na kompresorové straně výchozí konfigurace

FIGURE 5: Waterfall diagram of displacement on the compressor side of the original design

OBRAZEK 5: Kaskádový diagram výchylky na kompresorové straně výchozí konfigurace

TABLE 1: The unbalance considered in simulations
TABULKA 1: Hodnoty nevývahy uvažované v simulaci

	Unbalance [$g \cdot mm$]	
	On the compressor wheel	On the turbine wheel
Original state	1.2	1.85
20% higher unbalance	1.44	2.22
100% higher unbalance	2.4	3.7
20% lower unbalance	0.96	1.48
50% lower unbalance	0.6	0.925

Figure 5. The displacement is measured on the compressor (turbine) wheel nose relative to the housing in axis X shown in Figure 1. The reference value for transformation to a dB-value is $1 \cdot 10^{-6}m$. The cause of the higher overall eccentricity is the sub-synchronous vibration, which originates from instability of the outer oil film. Since our interest is the influence of the unbalance it is necessary to look at the synchronous vibration, which it is directly excited by it. For the purpose of this paper, several additional test mass values representing unbalance were selected as shown in Table 1.

FIGURE 6: Influence of unbalance change on the mean value of the overall relative eccentricity on the bearing on compressor side
OBŘÁZEK 6: Vliv změny nevývahy na střední hodnotu celkové relativní excentricity ložiska na kompresorové straně

The change of the unbalance and its influence on the rotor dynamics is shown in Figure 6 and Figure 7. From the figures it can be clearly seen that the difference between original state and the ones with alternated unbalance is not very high. However, when the waterfall diagram is considered the difference can be clearly seen. This difference is illustrated in Figure 8 and Figure 9. These figures show the difference between two extreme cases – one with 50% lower unbalance and one with 100% higher unbalance. The difference between these two cases clearly

FIGURE 8: Influence of the unbalance change on the waterfall diagram of displacement on the compressor side: a) 50% lower unbalance, b) 100% higher unbalance
OBŘÁZEK 8: Vliv změny nevývahy na kaskádový diagram výchylky na kompresorové straně: a) o 50% nižší nevývaha, b) o 100% vyšší nevývaha

FIGURE 9: Influence of the unbalance change on the waterfall diagram of displacement on the turbine side: a) 50% lower unbalance, b) 100% higher unbalance
OBŘÁZEK 9: Vliv změny nevývahy na kaskádový diagram výchylky na turbínové straně: a) o 50% nižší nevývaha, b) o 100% vyšší nevývaha

FIGURE 7: Influence of unbalance change on the mean value of the overall relative eccentricity on the bearing on turbine side
OBRAZEK 7: Vliv změny nevyváhy na střední hodnotu celkové relativní excentricity ložiska na turbinové straně

shows that size of the unbalance has the most influence on the synchronous component of vibration. The sub-synchronous components are not significantly affected by its influence.

5. CONCLUSION

The high demands placed on modern power units, and internal combustion engines in particular, has led to even higher demands on their components, with turbochargers being no exception. Therefore, turbocharger manufacturers strive to develop more advanced turbochargers for the lowest cost possible. This leads to the trend of simplification of the manufacturing process, with one of the important processes under consideration being balancing. In this paper the influence of rotor unbalance on rotor dynamics is examined using an advanced turbocharger model. Results indicate that higher unbalance increases the magnitude of the synchronous vibration component (see Figure 8 and Figure 9). However, Figure 6 and Figure 7 suggest that the overall magnitude of vibration remains almost the same. This could mean that with the amount of unbalance the amplitude of dominant frequency of turbocharger noise and vibration can be adjusted. Therefore, the noise and vibration can be tuned to be more tolerable. However, this statement needs to be verified experimentally.

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LIST OF NOTATIONS AND ABBREVIATIONS

B	bearing width
B_i	bearing width (inner oil film)
B_o	bearing width (outer oil film)
D	bearing diameter

D_i	bearing diameter – sleeve (inner oil film)
d_i	bearing diameter – journal (inner oil film)
D_o	bearing diameter – sleeve (inner oil film)
d_o	bearing diameter – journal (inner oil film)
e	overall eccentricity
e_i	inner eccentricity
e_o	outer eccentricity
h	oil film gap height
H	dimensionless oil film gap height
p	pressure
u	effective velocity
x, y, z	coordinates
Z	dimensionless coordinates along bearing width
γ, δ	dimensionless journal tilt angles
ε	relative eccentricity
η	dynamic viscosity
ρ	density
φ	dimensionless angle around pin axis
Π_D, Π_V	dimensionless parameters (further described in [5])
DOF	degree of freedom
FE	finite element
MBS	multi-body system

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