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DYNAMIC ANALYSIS OF TWO-STAGE TURBINE EXPERIMENTAL STAND

J. Čečrdle*, O. Vích**

Abstract: *The presented paper describes the dynamic analyses of a newly designed experimental stand for testing of turbine wheels. The experimental device includes two overhung rotor discs attached to two coaxial shafts. The main focus is paid on the rotor dynamic analysis, i.e. the complex modes analysis including shafts rotation. The evaluated parameters include rotor revolutions and stiffness parameters of bearings. The evaluated results include characteristics of complex modes and identification of whirl resonances and critical speeds of the system.*

Keywords: Rotor dynamics, Coaxial rotors, Complex modes, Campbell diagram.

1. Introduction

The presented paper deals with the dynamic analysis of the newly designed experimental stand for turbine wheels testing. The device includes two coaxial shafts with two overhung discs. Both shafts are single-side connected one another. The inner shaft is attached to the outer shaft using one angular-contact bearing and one ball bearing. The outer shaft is attached to the bearing support using two angular-contact bearings. Bearing supports, which are used to measure the torque decay, are attached to the stator structure by means of two ball bearings. The circumferential force and the axial force is measured by means of dynamometers. Maximal estimated revolutions of rotor are 16 000 rpm.

Performed analyses included normal modes and mainly the complex eigenvalue analysis including rotor revolutions. The main reason is to predict the whirl resonances, critical speeds and potential instabilities of the system. Analyses performed using FEM included isolated (cantilevered) rotor and the rotor supported with two variants of the stator structure. The evaluated parameters included rotor revolutions and the stiffness parameters of bearings.

2. Theoretical Background

The support of the rotating parts (bearings) is considered as symmetrical; however the rotating structure itself is slightly unsymmetrical. Therefore, the analytical approach using a rotating coordinate system was employed. The equation of motion for a damped structure without external forces is written as:

$$[M]\{\ddot{q}\} + (2\Omega\,[C] + [D_I + D_A])\{\dot{q}\} + \big([K] + \Omega\,[D_B] - \Omega^2[Z] + \Omega^2[K_G]\big)\,\{q\} = \{0\} \qquad (1)$$

where [M] is mass matrix, [C] is gyroscopic matrix, [D_I] is internal viscous damping matrix, [D_A] is external viscous damping matrix, [K] is elastic stiffness matrix, [D_B] is antisymmetric external damping matrix, [Z] is centrifugal softening matrix and [K_G] is geometric differential stiffness matrix. In the modal solutions, we transform the equation using modal matrix [Φ] into the generalized form.

As the first step, real eigenvalue analysis is performed:

$$(-\omega_0^2[M] + [K])\{\varphi\} = \{0\} \qquad (2)$$

* Ing. Jiří Čečrdle, Ph.D.: Czech Aerospace Research Centre (VZLU), Beranových 130; 199 05, Prague; CZ, cecrdle@vzlu.cz

** Ing. Ondřej Vích: Czech Aerospace Research Centre (VZLU), Beranových 130; 199 05, Prague; CZ, ondrej.vich@vzlu.cz

The solution eigenvectors {φ} are collected into the modal matrix [Φ]. The subsequent rotor dynamic analysis has the character of the approximate solution as the limited, but sufficient and reasonable selection of modes is considered. For harmonic motion $\{q(t)\} = \{q\}\, e^{\lambda t}$, the eigenvalue problem in the rotating system is:

$$\left(\lambda^2[\bar{M}] + \lambda(2\Omega\,[\bar{C}] + [\bar{D}_I + \bar{D}_A]) + \left([\bar{K}] + \Omega\,[\bar{D}_B] - \Omega^2[\bar{Z}] + \Omega^2[\bar{K}_G]\right)\right)\{q\} = \{0\} \tag{3}$$

A loop is made over the selected rotor speed. Solutions are then sorted in frequencies and the mode tracking algorithm is applied to obtain the data for the Campbell diagram. The forward whirl resonance points are then found for the intersection with the 0P-line:

$$\left([\bar{K}] - \Omega^2([\bar{Z}] - [\bar{K}_G])\right)\{q\} = \{0\} \tag{4}$$

while the backward whirl resonance points are found as the intersection with the 2P-line:

$$\left([\bar{K}] - \Omega^2(j\,4[\bar{C}] - 4\,[\bar{M}] + [\bar{Z}] - [\bar{K}_G])\right)\{q\} = \{0\} \tag{5}$$

3. Analyses and Results for Isolated Rotor

The analyses of the isolated rotor were aimed at obtaining the basic data for the rotor itself. The frequency range of interest was set as 0 to 1000 Hz. For the isolated rotor, four pairs of modes representing transverse vibrations in two orthogonal planes were found. Due to the rotor unsymmetry, the frequencies were slightly diverse. In addition, axial vibration mode was found as well. Figures 1 to 4 show the mode shapes of the rotor bending modes (in the vertical plane).

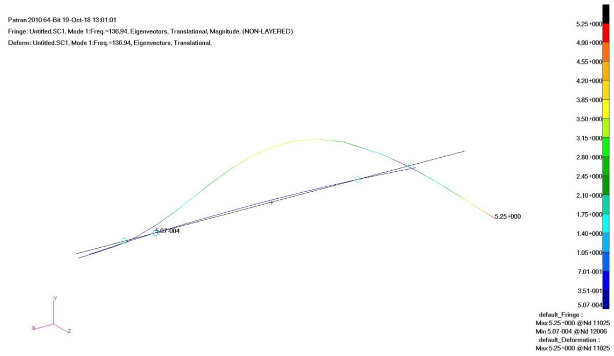

Fig. 1: 1st bending mode shape, f = 136.9 Hz.

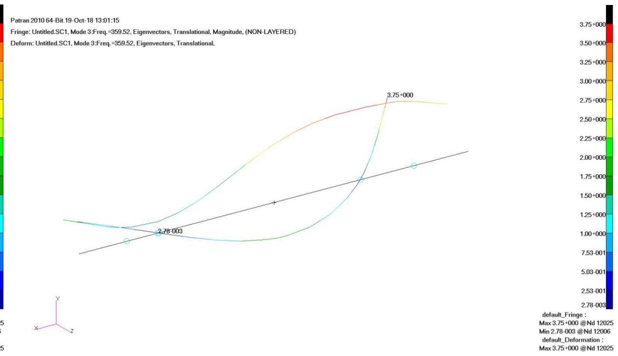

Fig. 2: 2nd bending mode shape, f = 297.2 Hz.

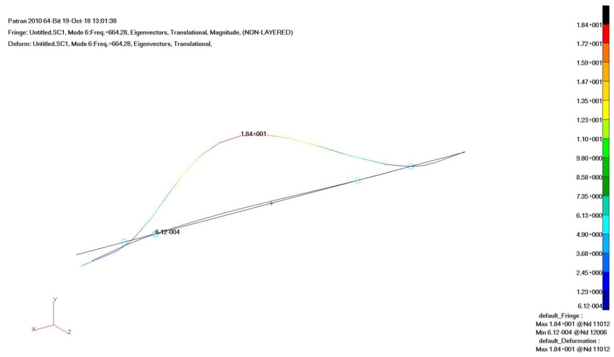

Fig. 3: 3rd bending mode shape, f = 664.2 Hz.

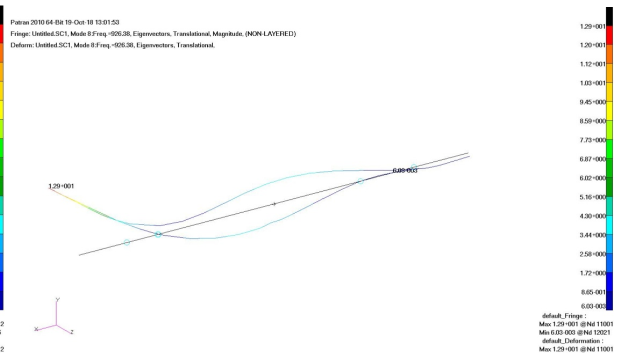

Fig. 4: 4th bending mode shape, f = 926.4 Hz.

The frequencies, listed in the captions of figures 1 to 4, represent the figures for the rotor speed of zero. Dependence of frequencies on the rotor speed is demonstrated in the Campbell diagram shown in figure 5. This plot represents the analysis considering rigid bearing. Apart from this, the flexible bearings (considering two stiffness values) were analyzed as well. The reason was to obtain the applicable range of the resonance frequencies as the bearings stiffness data were not available. The resonance frequencies are represented by crossings of modes nr.1, 3, 6, 8, which are the backward whirl modes, with the 2P-line (dashed line). The maximal influence of bearing stiffness on resonance frequency was found for the 2nd bending mode (mode nr.3), for which the influence of bearing stiffness reaches -15 %.

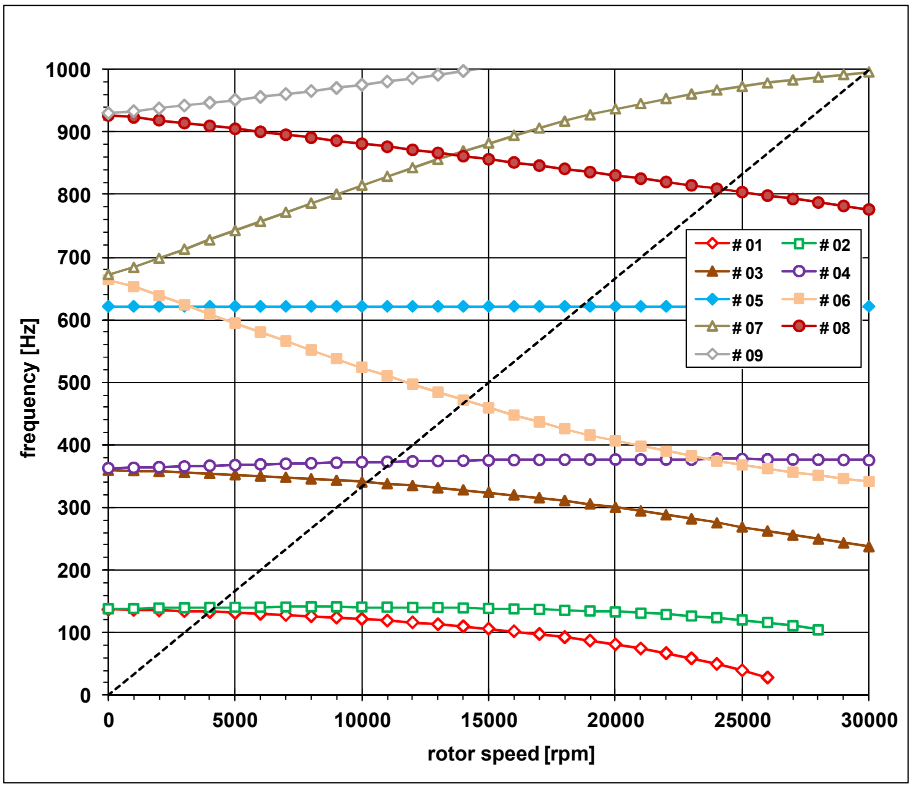

Fig. 5: Campbell diagram, isolated rotor, rigid bearings

4. Analyses and Results for Complete Device

Final calculations were performed on the complete device, i.e. rotor supported with the stator structure. The stator structure was considered in two variants – without and with the supporting arm. These analyses aimed to assess the influence of the flexible stator structure on the dynamic characteristics of the rotor. Another issue was to evaluate the lowest modes of the stator structure. The first mode represented the vibrations of the stator structure in the vertical plane as shown in figures 6 and 7. The effect of stator supporting arm (figure 7) on the frequency of the first mode was +10.4 %.

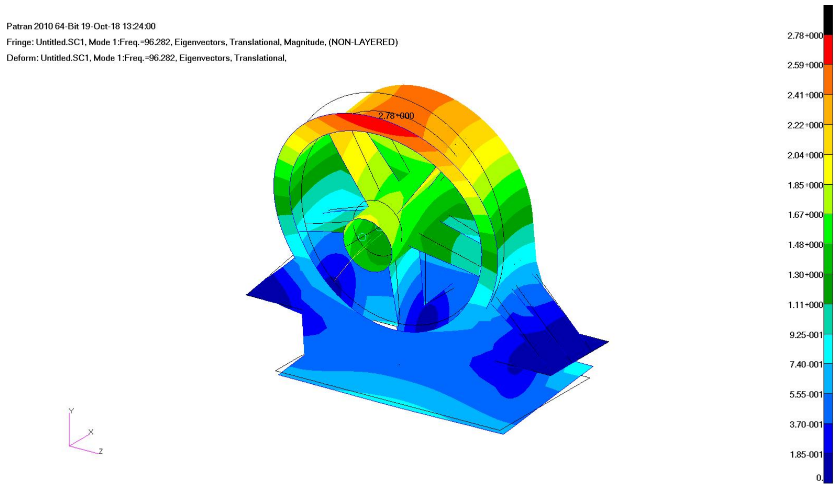

Fig. 6: 1st stator mode, without supporting arm, f = 96.3 Hz

Finally, the example of Campbell diagram for the complete structure is shown in figure 8. The highest effect of the stator structure was found for the 2nd bending rotor mode, the effect on frequency was rather significant - it was as high as -36.7 %. In the Campbell diagram, the large influence of the rotor speed on frequency can be found for the rotor modes. However, another higher-order modes show the whirl vibrations as well. Therefore, many resonance points were found. Note that the mode tracking algorithm was not able to recognize modes properly in some areas of the frequency crossing. Nevertheless the manual correction was not performed as the diagram is understandable enough.

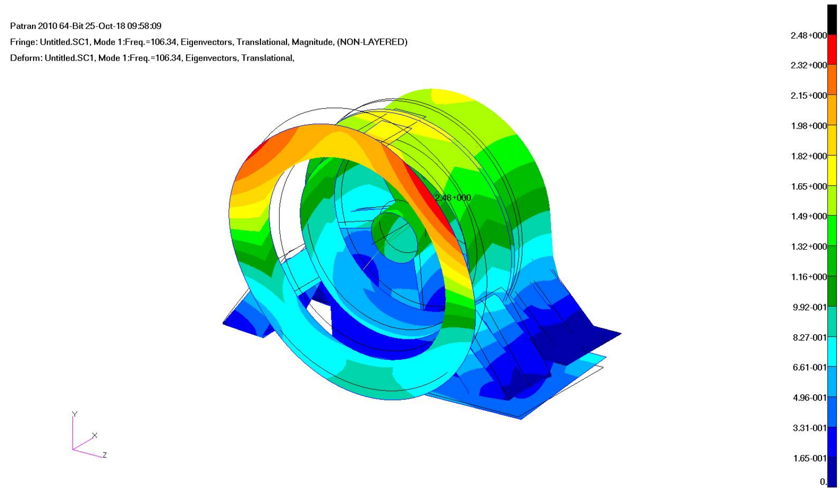

Fig. 7: 1st stator mode, with supporting arm, f = 106.3 Hz

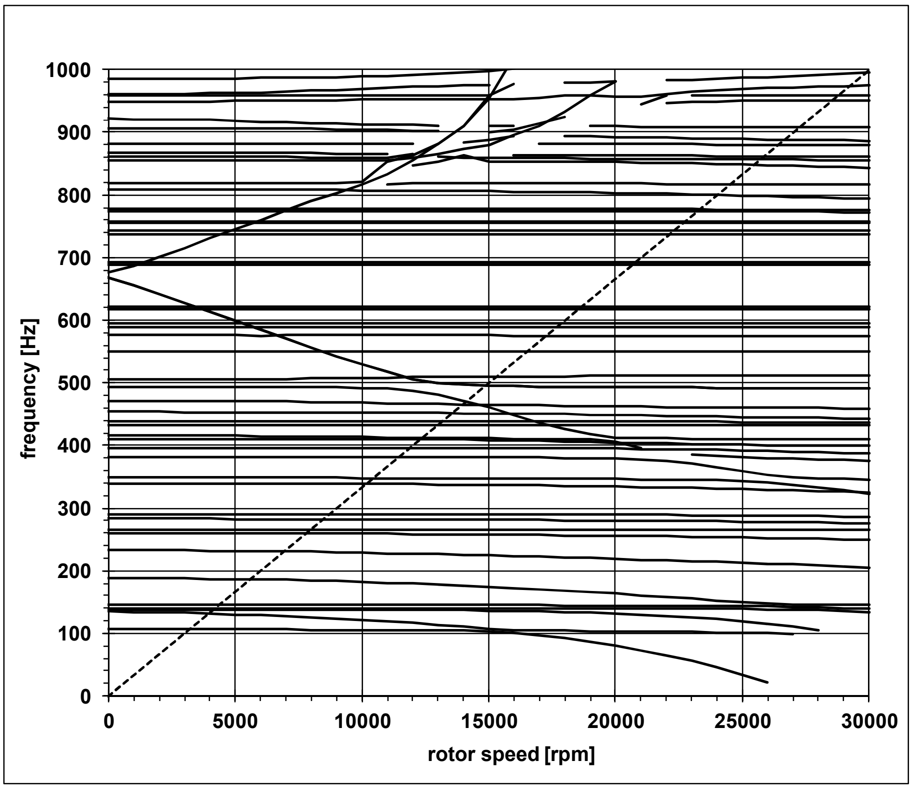

Fig. 8: Campbell diagram, complete device, rigid bearings

5. Conclusion

This paper presented dynamic analyses of the experimental stand for the turbine wheels testing. The purpose of analyses was to evaluate the possible critical areas concerning the dynamic behaviour of the complete device. The performed studies include mainly the complex eigenvalue analysis including the rotor rotation (rotor dynamics).

References

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