Dealing with Frequency- and Speed-Dependent Bearing Data (Foil Bearings) in MADYN 2000

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1. Frequency and Speed Dependent Bearings

Many bearings have speed- and load-dependent characteristic, among them are fluid film bearings and rolling element bearings. The speed dependence of fluid film bearings is caused by the hydrodynamic effect, which increases with speed. In case of rolling element bearings, it is due to centrifugal and gyroscopic forces, which among others influence the contact angles. In both cases the linear dynamic behaviour is described by linearisation about the static equilibrium from the static load.

Some bearings such as magnetic bearings have frequency-dependent characteristics. It is caused by the controller as well as the characteristics of amplifier, sensor, actuator, and digitisation effects.

Some bearings have speed and frequency dependent characteristics. Among them are certain geometries of tilting pad fluid film bearings. The frequency dependence is caused by the tilting of the pad, which also causes a squeezing of the fluid. Assuming, that the pad tilting immediately follows the rotor movement to take its equilibrium position considering all static and dynamic forces, the tilting angle can be eliminated from the equations. The characteristics then is frequency independent. However, strictly speaking this is not the case. The movement of the pad is inhibited by squeezing and inertia. For some geometries (long, centrally supported pads) this is not negligible.

All the above-mentioned bearings are commonly used and are fully integrated into MADYN 2000, i.e., they can be modelled with all details and their characteristics are calculated and considered in all static and dynamic analyses.

Gas bearings and foil bearings also have frequency- and speed-dependent characteristics (see for example /1/). This also applies for the most widely used foil bearing type, the bump foil bearing. For this bearing type a big part of the frequency dependence is caused by the friction between bump foil and stator as well as top foil. A lot of research is going on to reliably determine the properties of bump foils. DELTA JS has contributed to this by initiating a research project (see /2/). However, currently predictions of rotors supported on bump foils are still unreliable. This is one of the reasons, why foil bearings are not yet integrated in MADYN 2000.

Nevertheless, rotors on bump foil bearings can be analysed with MADYN 2000 with imported speed and frequency dependent characteristics, which may stem from some of the existing programs developed at universities. This is demonstrated in the following.

Seals such as honeycomb and hole pattern seals also have frequency and speed dependent characteristics, mainly due to acoustic effects /3/. They can be treated in a similar way by means of imported data. It is planned to fully integrate them into MADYN 2000 soon.

2. Foil Bearing Example

2.1 Description of Foil Bearings

An example of a foil bearing is shown in figure 1. The bearing has 3 bump foils. The arrangement has a profiled clearance between top foil and rotor (preload) to improve the stability.

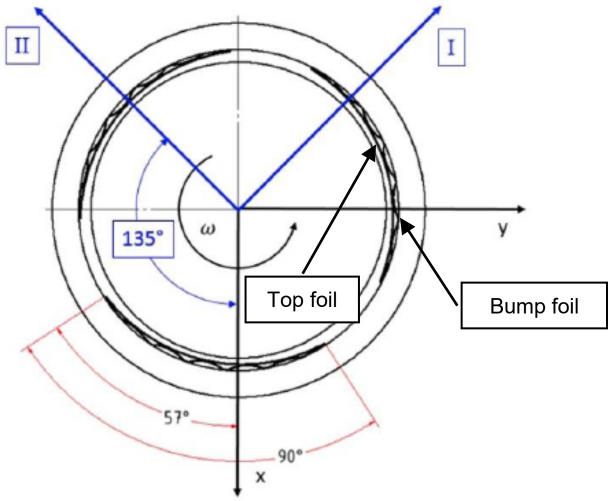

Fig.1: Example of a foil bearing (from /2/).

The bearing consists of an aerodynamic gas film, the top foil, and the bump foil making the top foil compliant. The friction between bump foil and stator as well as top foil provides the main damping to which the gas film and the gas between top foil and stator also contribute.

Due to the flexible top foil these bearings are rather robust. Bump foil bearings have a high load capacity compared to other types of foil bearings, e.g., the leave type. These properties are the reason, why they are used more and more as a simple oil free bearing solution.

2.2 Rotor Model

The rotor on foil bearings is shown in figure 2. It has been tested in the FVV research project /2/ up to a speed of about 90’000rpm. The short grey shaft with a very small diameter at the right represents the lateral stiffness of a metal below coupling, which was used in the test set up. The right end of the coupling is clamped and laterally rigid supported. Vibrations were measured at the marked sensor locations.

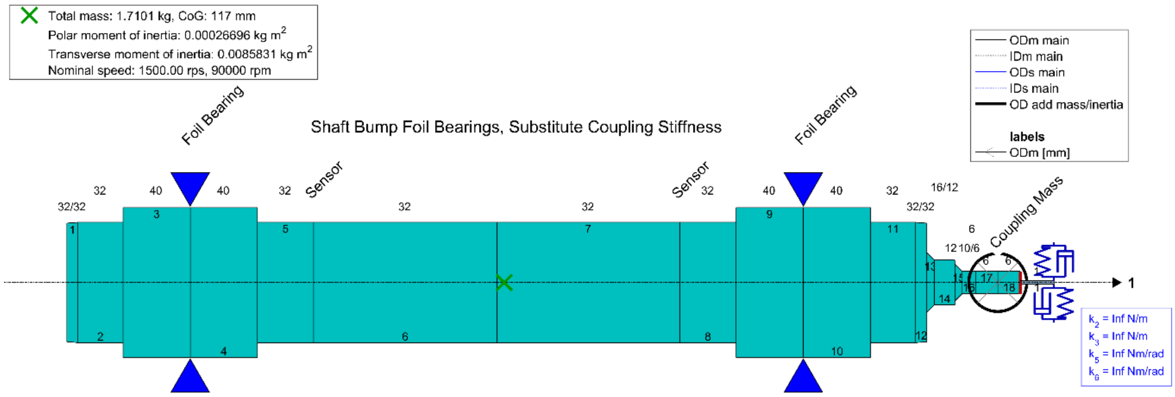

Fig.2: Rotor model on foil bearings

2.3 Bearing Characteristics

Most programs developed in research projects for foil bearings calculate the usual 2x2 damping and stiffness matrix for the two lateral directions for different speeds. The frequency dependence is considered by coefficients for different frequency ratios (frequency/speed).

For each of these coefficients transfer functions in polynomial form can be adapted to capture the frequency dependence. This can be done in MATLAB using the function “invfreqs”, which is part of the signal processing toolbox. In figures 3 and 4 the coefficients for different frequencies at a speed of 100’000rpm are shown together with the adapted functions. The coefficients were calculated with a program developed in the above-mentioned research project (see /2/). For following orders for the numerator and denominator polynomials were used in the current example for the polynomials:

Stiffness coefficients k_{ij}: zero, i.e. it is constant as can be seen in figure 3.
Daming coefficients d_{ij}: 2

Once polynomial transfer functions of the damping and stiffnesses have been created a state space system with matrices A,B,C,D for each can be built with 4 inputs (displacements and velocities in 2- and 3- direction) and 2 outputs (forces in 2- and 3-direction). This can be done using the MATLAB function “ssdata”, which is part of the control toolbox. To capture the speed dependence, the matrices must be created for several speeds. A load-dependence can also be defined by creating the matrices for several loads. In the present case, there is only one load caused by the weight, which is practically the same for the two bearings. The speed- and load-dependent state space matrices can be loaded as RFB (radial fluid film bearing characteristic) of the type “Import with dimension” via a MATLAB mat-file. The format of the content is described in chapter II.6.8 of the documentation.

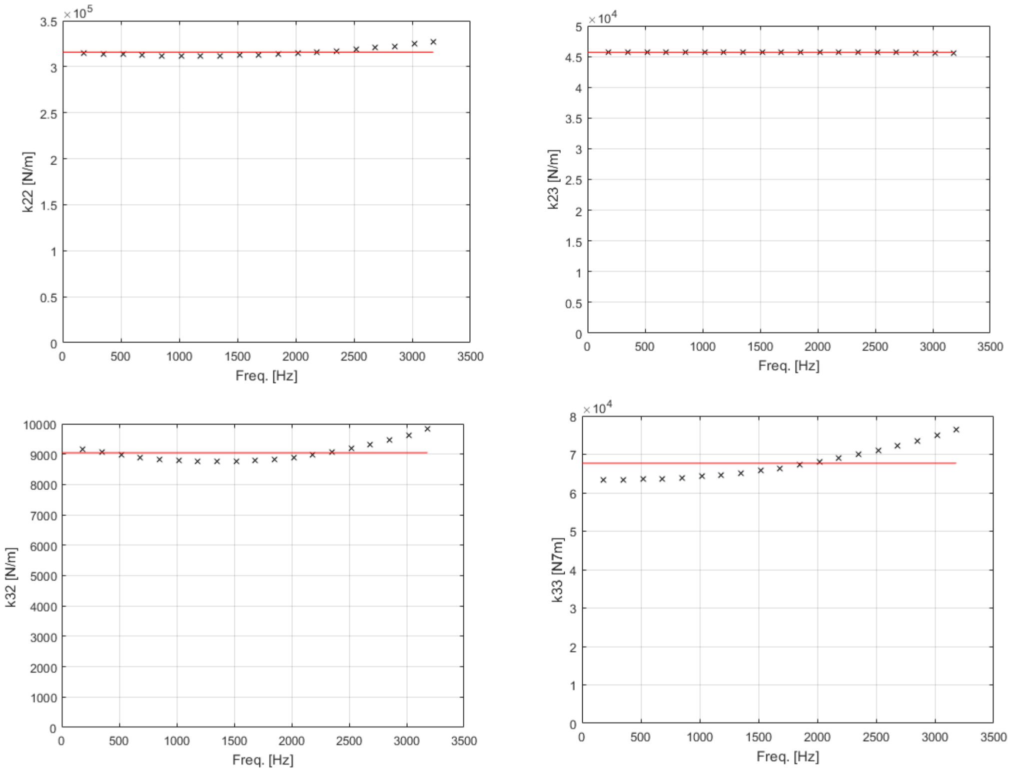

Fig. 3: Calculated (black cross) and adapted (red line) frequency dependent stiffness coefficients

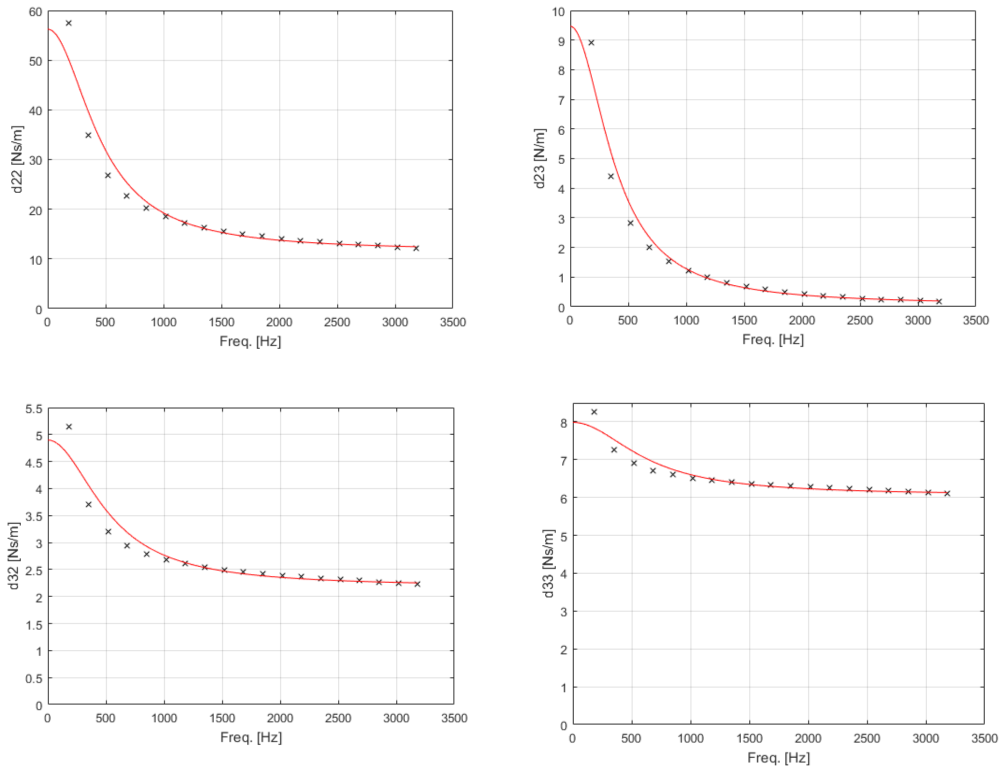

Fig. 4: Calculated (black cross) and adapted (red line) frequency dependent damping coefficients

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2.4 Campbell Diagram, Stability Assessment

The Campbell diagram with the speed and frequency dependent bearing characteristics as descried in the precious chapter together with the mode shapes in compact form including the orbits at the location of maximum deflection and the whirling direction is shown in figure 5. Note, that positive whirling is from 2 to 3.

The damping ratios of the forward whirling parallel and tilting modes with an elliptical orbit mainly in the vertical direction considerably decrease with increasing speed but remain stable. The bending modes have poor damping because the foil bearings are too weak to provide a relevant damping to this mode.

In the test of the rotor an onset of instability could be observed at about 90’000rpm in the form of a sub-synchronous with a dominating frequency of 180Hz as can be seen in figure 6. The shown results from /2/ were made in balanced condition. Further measurements with set unbalances (see next chapter) did not change the stability behaviour significantly. The still stable forward tilting mode (mode 4) in figure 5 has a frequency of 150Hz at 90’000rpm.

Thus, neither the instability nor the frequency of the unstable mode could be predicted satisfactorily.

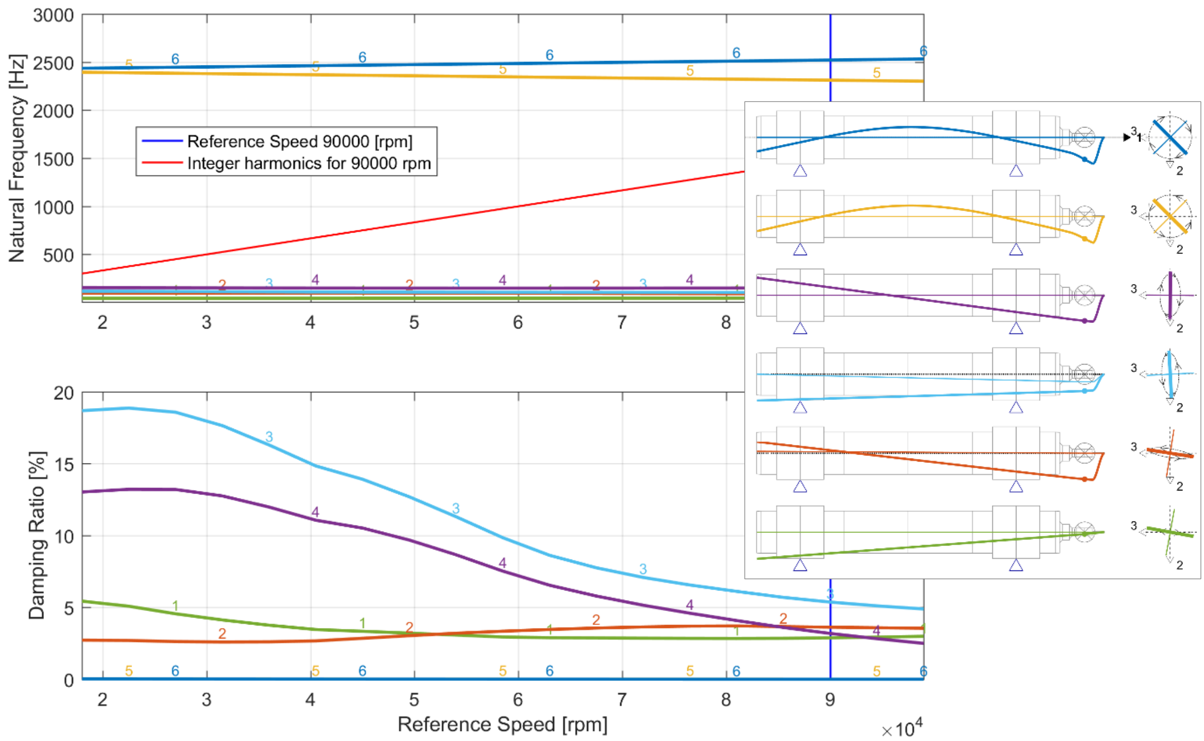

Fig. 5: Campbell diagram with mode shapes at 90’000rpm

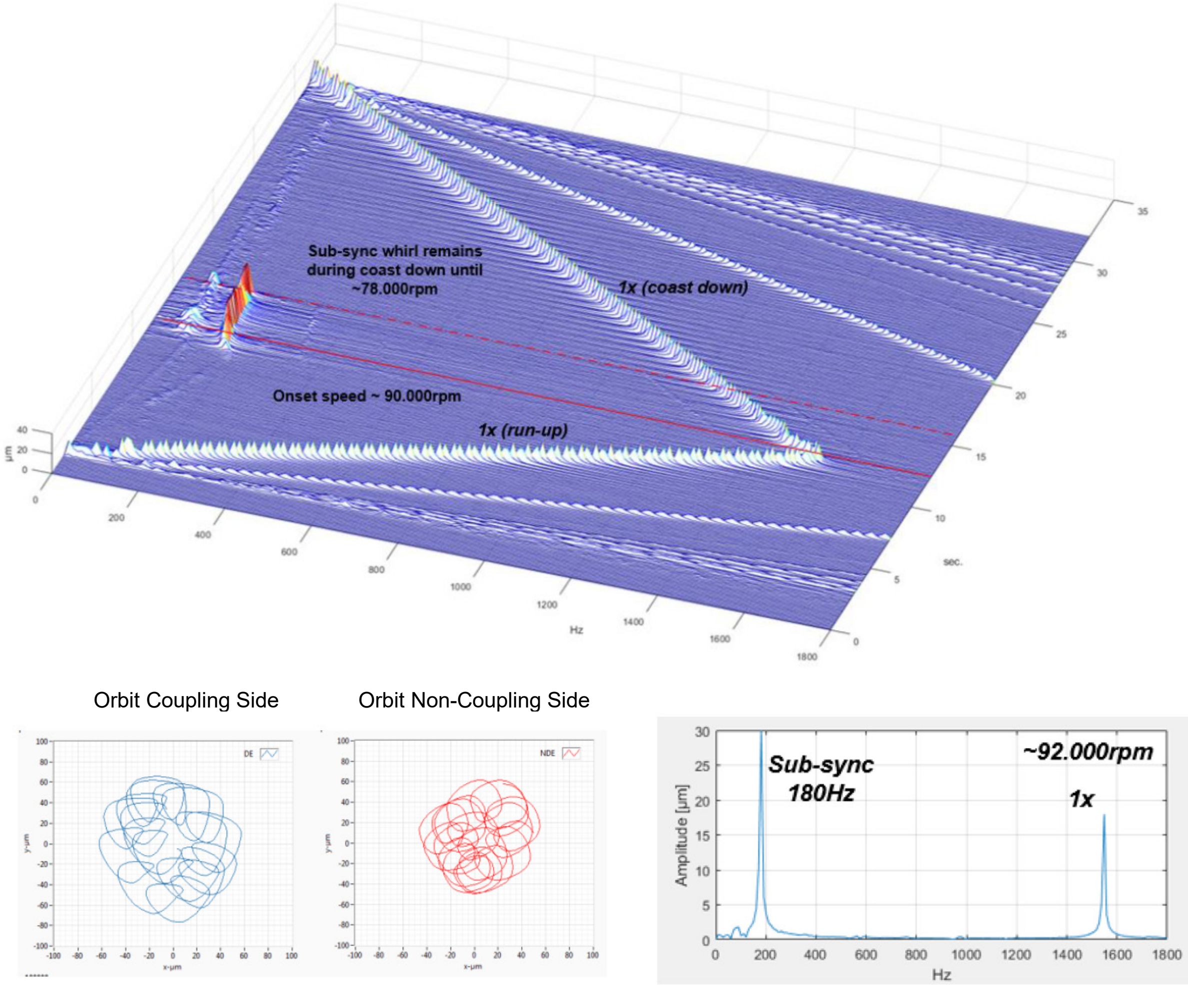

Fig. 6: Measured Waterfall plot of the test, spectrum and orbits at 92’000rpm (from /2/)

2.5 Unbalance Response

The unbalance load as well as the calculated and measured synchronous response are shown in figure 7 and 8. The shape of the response is shown at 2 speeds, 40’000rpm and 90’000rpm.

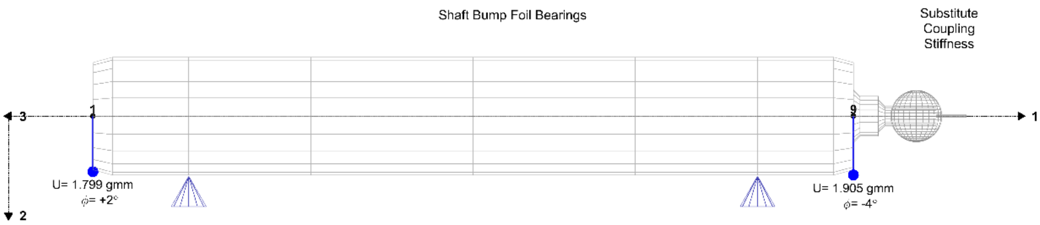

Fig. 7: Unbalance load case to compare the measured and calculated response

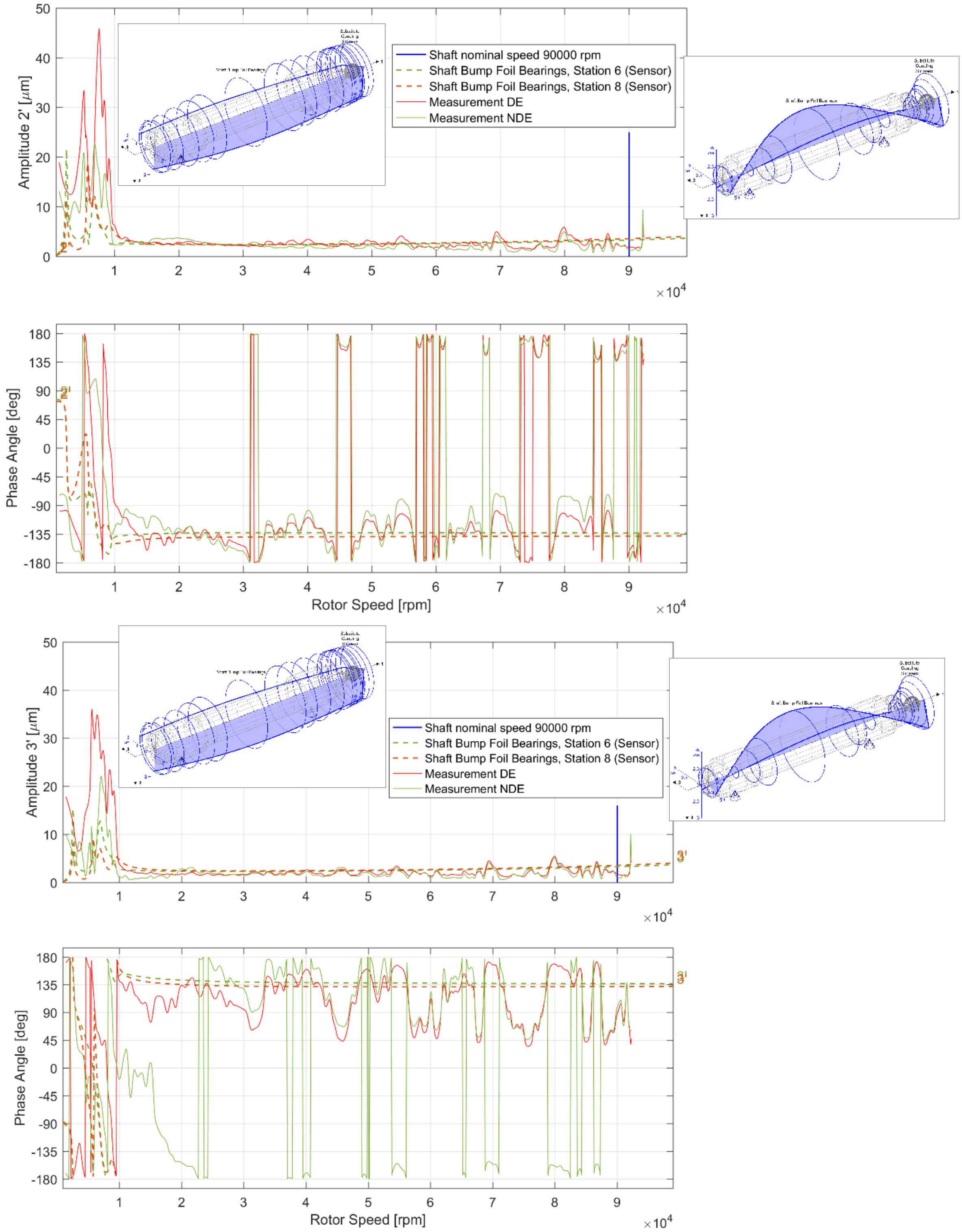

Fig. 8: Measured and calculated unbalance response in 2' and 3' direction (45° to vertical) with shapes at 40'000rpm and 90'000rpm.

The measured response in the low-speed range with the resonances of the rigid body modes is not very clear and deviates considerably from the calculated response, most probably due to the highly non-linear behaviour and the not fully air borne condition. At higher speeds the agreement is better, but the bearing then does not influence the behaviour to a high extent, due to its low stiffness and damping. At very high speed the rotor bends.

3. Summary

MADYN 2000 is well capable of handling load-, speed- and frequency-dependent bearing data. Among other bearings foil bearings have such characteristics.

In contrast to fluid film bearings, rolling element bearings and magnetic bearings rotordynamic predictions with foil bearings are not very reliable, especially with the popular bump foil bearing. Reasons are the friction, which always has a certain level of uncertainty, inaccuracies of the foil surface and insufficient models capturing the dynamic behaviour of the foil.

Foil bearings are therefore not yet integrated into MADYN 2000. Nevertheless, frequency-, load- and speed-dependant data can be imported into MADYN 2000. In this document this is demonstrated for a bump foil bearing. The characteristics of this bearing has been calculated with a special bump foil bearing program (see /2/) in the form of speed- load- and frequency-dependent rotordynamic stiffness and damping coefficients. This is the current practice. It is shown how to process these data in MATLAB and import them into MADYN 2000 in the form of state space matrices. Note, that more advanced models will allow direct creation of create state space matrices without the intermediate step of coefficients (see for example /4/).

The behaviour of the rotor on bump foil bearings has been analysed (Campbell diagram and unbalance response) and the results are compared to measurements.

The measured onset of stability at 90’000rpm is not predicted by the model, although the calculated damping of the rigid body modes reduces considerably with speed. However, it does not become negative. The measured sub-synchronous frequency of 180Hz deviates to some extent from the forward whirling tilting mode of 150Hz, which causes the instability.

The measured unbalance response in the speed range of the resonance of the rigid body modes is not very clear. Calculate results in this speed range deviate considerably. At higher speeds the agreement is reasonable, however, the behaviour at these speeds does not depend anymore on the bearings, which are very soft.

4. References

/1/ Eliott Guenat, Jürg Schiffmann: Dynamic force coefficients identification on air-lubricated herringbone grooved journal bearing. Mechanical Systems and Signal Processing 136, 2020.

/2/ Folienlager II, Vorhaben Nr. 1267: Betriebssichere aerodynamische Folienlager für die Lagerung schnell laufender Rotoren. FVV Abschlussbericht, 2019.

/3/ George F. Kleynhans, Dara W. Childs: The Acoustic Influence of Cell Depth on the Rotordynamic Characteristics of Smooth Rotor / Honeycomb-Stator Annular Gas Seals. ASME paper No. 96-GT-122, presented at the ASME Turbo Expo, Birmingham, UK, 1996.

/4/ Philip Bonello: A new method for the calculation of the Campbell diagram of a foil-air bearing rotor model. SIRM 2019 -13th International Conference on Dynamics of Rotating Machines, Copenhagen, Denmark, 13th – 15th February 2019.