Experimental modal analysis of a gyroscopic rotor in active magnetic bearings

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***Abstract*—For a gyroscopic rotor in active magnetic bearings, the rigid body natural frequencies and mode shapes are investigated. An impulsive excitation force is generated by two rectangular pulses in the current control reference signal. With vibration measurements available in two perpendicular directions for each bearing, experimental modal analyses are performed at rotational speeds up to 100000 rpm. Identification results demonstrate the gyroscopic split of natural frequencies.**

I. INTRODUCTION

The dynamic behavior of rotors differs from the behavior of non-rotating structures. The multi-degrees-of-freedom description of a rotor is characterized by non-symmetric system matrices arising from gyroscopic and circulatory effects. An axially symmetric rotor exhibits double natural frequencies at rest, which are separated by the gyroscopic split from the onset of rotation. Such effects may be modeled and predicted in theory, but the results will depend on the accuracy of material parameters and the underlying properties of the bearings. As an alternative, natural frequencies and mode shapes can be identified by experimental modal analysis, which has been applied to rotors since first experiments described in [1]. The identification requires measured frequency response functions between excitation force and vibration response. Compared to non-rotating structures, rotors are difficult to excite; the hammer blow [1], shaker excitation with an auxiliary bearing [2], rotating unbalance [3], and electromagnetic exciters [4] have all been used with different effort, influence on the dynamic behavior of the rotor, and accuracy of the measured excitation force.

Active magnetic bearings offer another way to apply controlled excitation forces. There have been some attempts to use them for experimental modal analysis. In [5], sophisticated a test stand is described, and it is suggested to determine the magnetic force from measurements of the magnetic flux; however, the resulting modal parameters are missing. Lee, Ha, and Kim [6] described an experimental modal analysis of a shaft-type rotor in active magnetic bearings. The magnetic forces are measured by piezoelectric force transducers; together with voltage and displacement signals, they are used to identify the dynamic current to voltage relationship without directly measuring the control current. An experimental modal analysis of the closed loop system is performed with bandlimited random noise or sinusoidal excitation signals. At a rotational speed of 6800 rpm, four natural frequencies are

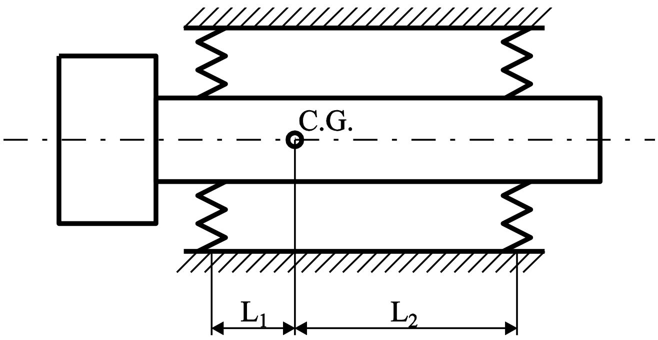

Figure 1. Schematic of rotor supported by active magnetic bearings.

identified together with the corresponding damping ratios. Backward and forward frequencies only differ by a few per cent and are extracted through complex modal testing [7].

In this paper, active magnetic bearings are used to determine the four rigid body modes of a gyroscopic rotor depending on the rotational speed. The magnetic bearing system includes an underlying current controller. An impulsive excitation force is generated by two rectangular pulses in the reference current; the resulting impulse in the actual current is measured in open loop and then applied to the closed loop system. The rotor is tested at rotational speeds up to 100000 rpm to obtain an experimental record of the gyroscopic effect.

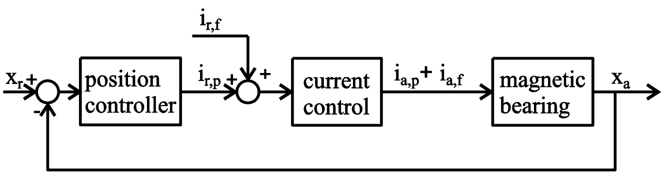

Figure 2. Block diagram of position control circuit.

II. EXPERIMENTAL SETUP

A schematic of the experimental setup is shown in Fig. 1. The rotor consists of a shaft that is supported by two active magnetic bearings and carries a cantilevered disc at one of its ends. Position sensors are situated immediately next to the bearings and measure the position of the rotor in two perpendicular directions. The rotor's weight is supported by permanent magnets. The position of the rotor is stabilized by the active magnetic bearing position control in two perpendicular directions for each bearing. It is assumed that in the immediate vicinity of the equilibrium position, the magnetic force from the bearing coils can be approximated by a linear function of coil current and position.

The system under investigation is constituted by the rotor in its position controlled active magnetic bearings. The same active magnetic bearings are used to generate the excitation force, which is possible by adding a defined impulse to the position control current. Figure 2 shows the respective block diagram. A suitable reference signal $i_{\mathrm{r,f}}$ is added to the reference current $i_{\mathrm{r,p}}$ from the position controller; assuming linear behavior of the current control, the resulting impulse $i_{\mathrm{a,f}}$ will be added to the actual position control current $i_{\mathrm{a,p}}$. To determine the required reference signal $i_{\mathrm{r,f}}$, the dynamic behavior of the current control is tested separately with the position control switched off for a few milliseconds.

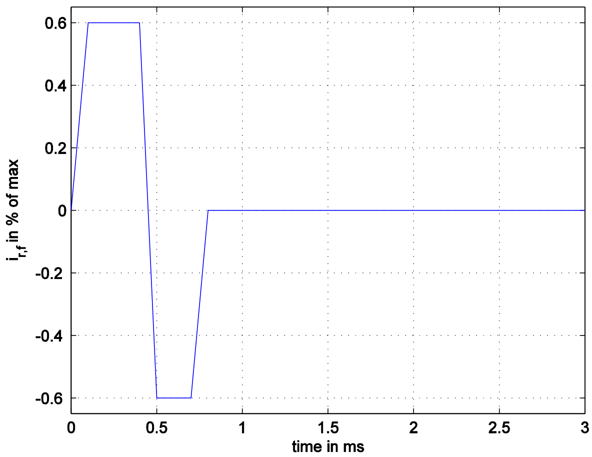

Figure 3. Reference current for excitation.

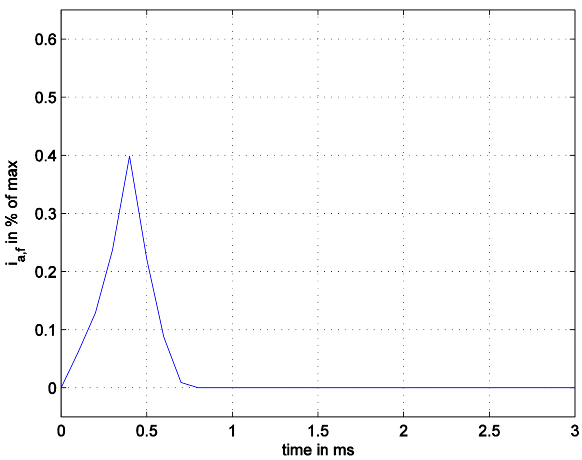

Figure 4. Actual current for excitation.

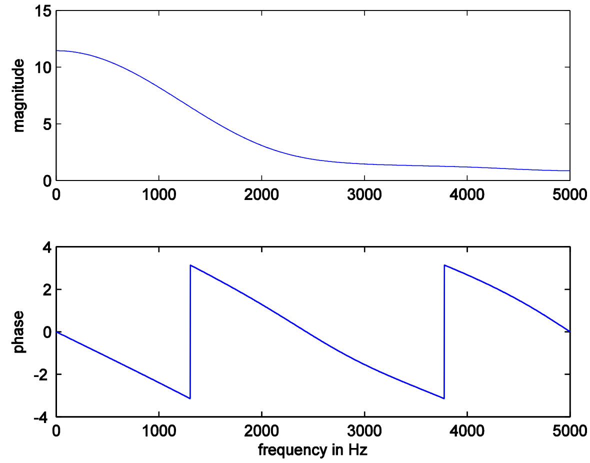

Figure 5. Frequency spectrum of actual current for excitation.

III. EXCITATION SIGNAL

To obtain an excitation that is easy to implement in the existing software, an impulsive force signal is sought, which should be broadband enough to cover the frequency range of interest. Figure 3 shows the selected reference signal $i_{\mathrm{r,f}}$; it consists of two rectangular pulses of different signs and lengths. The resulting impulse $i_{\mathrm{a,f}}$ after the current control is depicted in Fig. 4. The shape of the impulse is nearly triangular, and its frequency spectrum is shown in Fig. 5. The impulse $i_{\mathrm{a,f}}$ in the bearing coil current is assumed to generate a proportional magnetic force.

The excitation can be applied in two perpendicular directions per bearing, which is done in separate tests. Since measurements are made for a number of rotational speeds, stepped sine testing would be a tedious procedure. With random noise, there would be no simple control over the maximum amplitude of the response. Compared to chirp, impulsive excitation is much easier to implement and is therefore used although moderate accuracy must be expected.

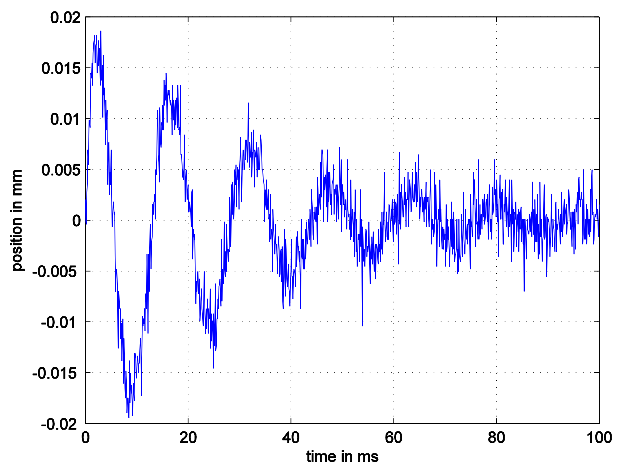

Figure 6. Vibration response at bearing 1, direction x, rotor at rest.

IV. VIBRATION RESPONSES

To determine the natural frequencies and mode shapes of the rotor, the impulsive force is applied in one degree of freedom, while vibration responses are measured in all degrees of freedom. The active magnetic bearings provide four position sensors so that a rigid body model with four degrees of freedom can be studied. Figure 6 shows the vibration response at bearing 1, direction x, of the rotor at rest with the impulsive force applied in same degree of freedom; to extract the relevant transient process, an exponential window is applied to the signal, the result of which is displayed in Fig. 7. The application of an exponential window introduces additional damping into the system; as natural frequencies and mode shapes are hardly changed, this is common practice to avoid leakage errors with impulsive excitation [2].

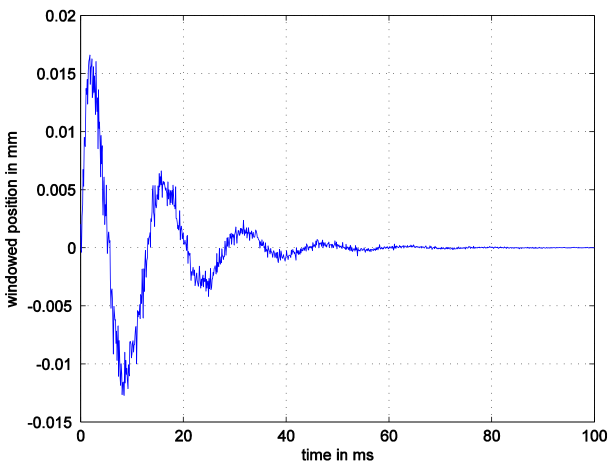

Figure 7. Vibration response at bearing 1, direction x, rotor at rest, after application of an exponential window.

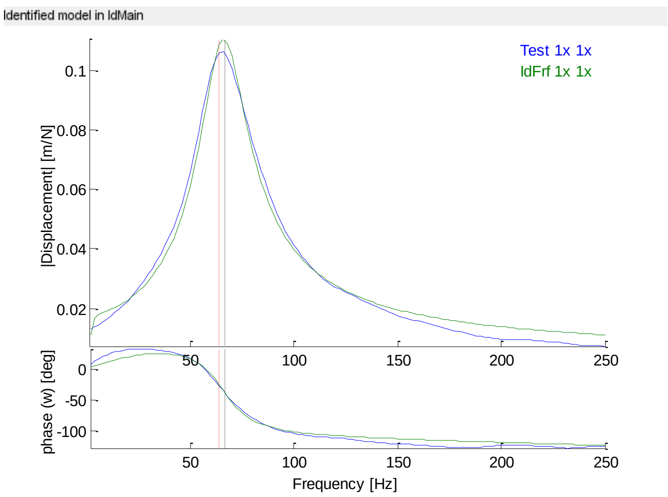

Figure 8. Measured and estimated point frequency response function, rotor at rest.

V. MODAL PARAMETERS

Natural frequencies and mode shapes of the rotor are determined at rest and at rotational speeds of 20000, 40000, 60000, 80000, and 100000 rpm. In each case, a single test provides four position signals which contain the vibration responses. With these signals windowed, the frequency response functions with respect to the impulsive force signal are calculated by Discrete Fourier transform, averaged over four measurements, and passed on to the Matlab Structural Dynamics Toolbox for modal parameter extraction [8].

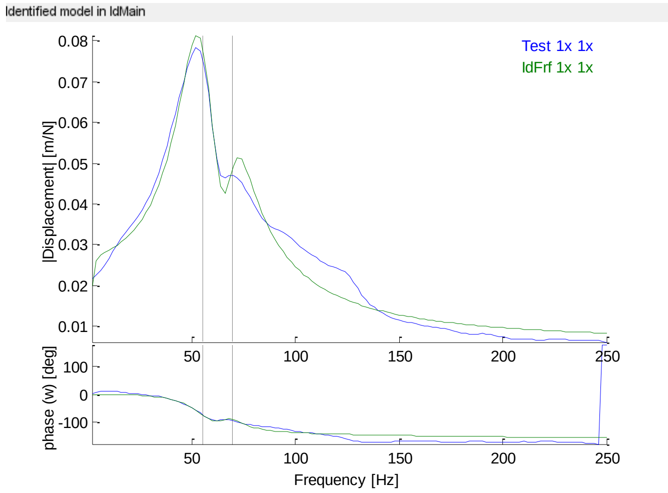

Figure 9. Measured and estimated point frequency response function at 60000 rpm.

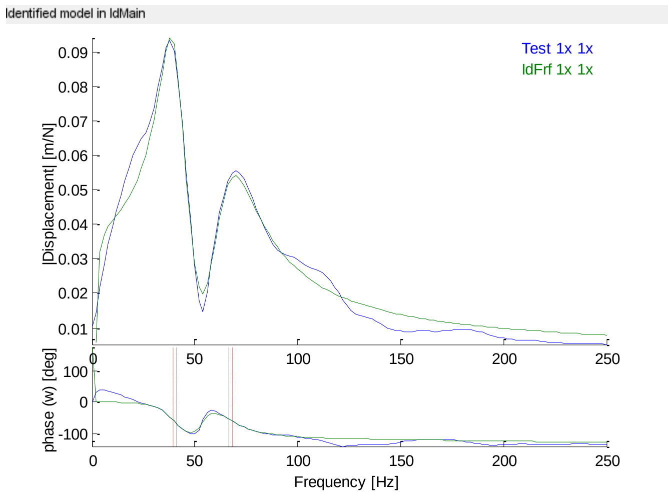

Figure 10. Measured and estimated point frequency response function at 100000 rpm.

First tests are done with the excitation applied at bearing 1, direction x. Figure 8 shows a comparison between the measured and estimated point frequency response function for the rotor at rest. The first resonance at 66 Hz is clearly reproduced by the estimate. In Fig. 9, measured and estimated point frequency response functions are depicted for a rotational speed of 60000 rpm. Due to the gyroscopic effect, the first resonance has split into one at 55 Hz and another at

71 Hz. This goes further at 100000 rpm, where these two resonances appear at 40 Hz and 70 Hz as can be seen in Fig. 10. For both 60000 and 100000 rpm, the mode shape animation shows that the lowest natural frequency belongs to a backward mode; the next mode shape, which should be forward, comes out less clearly.

From Figs. 8, 9, and 10, it is not clear what becomes of the second rigid body resonance of the rotor at rest. With the excitation applied at bearing 1, this resonance was excited near a node. All tests are therefore repeated with the excitation applied at bearing 2, direction x. For the second pair of natural frequencies, backward and forward modes can be distinguished from 40000 rpm upwards.

Experimental modal analyses are also performed with the excitation applied at bearing 1, direction y, and at bearing 2, direction y.

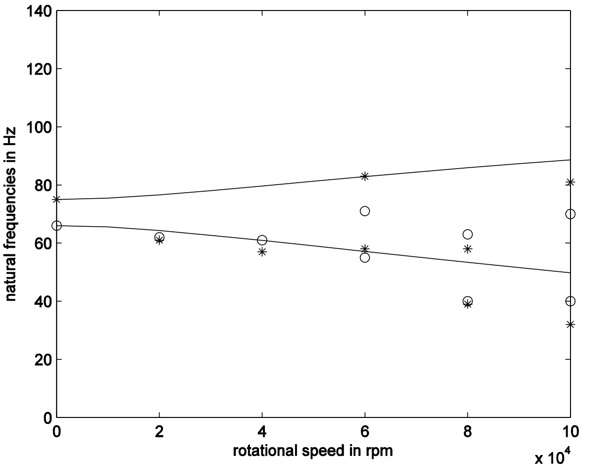

Figure 11. Calculated (-) and measured natural frequencies, excitation at bearing 1, directions x (o) and y(*), depending on the rotational speed.

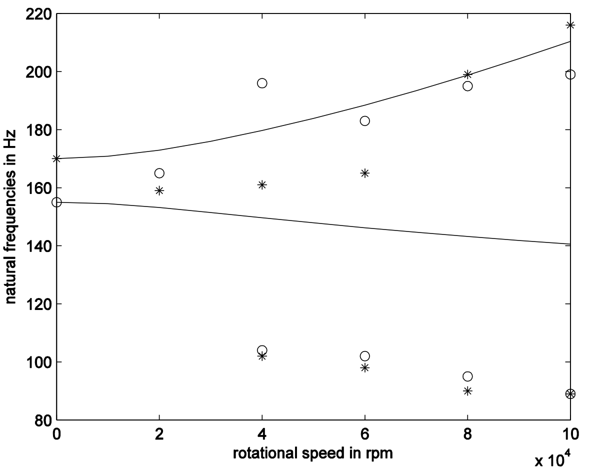

Figure 12. Calculated (-) and measured natural frequencies, excitation at bearing 2, directions x (o) and y(*), depending on the rotational speed.

VI. Stiffness Identification

All four bearing stiffness values are identified using the mass geometry of the rotor and the first and second natural frequencies in directions x and y at rest. With the identified stiffness values, the relationship between the rotational speed and the first pair of natural frequencies is depicted in Fig. 11. This figure also shows the measured natural frequencies with the excitation applied at bearing 1 in directions x and y. From 60000 rpm upwards, two natural frequencies can be distinguished for each excitation direction. In theory, these should be independent of the excitation direction and follow the calculated values. The remaining discrepancies may be explained by the poor frequency resolution resulting from the shortness of the windowed vibration responses.

Calculated and measured values for the second pair of natural frequencies are given in Fig. 12. In this case two branches are measured from 40000 rpm upwards. It is remarkable that for the lower branch of natural frequencies, measured results show a strong deviation from calculated ones although they agree fairly well among both excitation directions. This could mean that the rotor did not behave like a rigid body and may be explained by the type of connection between the disc and the shaft.

VII. Conclusions

The rigid body natural frequencies and mode shapes of a gyroscopic rotor in active magnetic bearings have been investigated. The active magnetic bearings have been used to generate an impulsive force signal, which offered a straightforward way to perform an experimental modal analysis. For both pairs of natural frequencies, identification results demonstrate the gyroscopic split. However, advanced excitation methods will be required to improve the accuracy of the results.

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