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Modal damping identification of a gyroscopic rotor in active magnetic bearings

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Abstract

Modal testing is done for a rotor in active magnetic bearings whose first backward gyroscopic mode is prone to instability. In the absence of process forces, the rotor is excited from one of its active magnetic bearings, and modal parameters are identified at rotational speeds up to 60000 rpm. Experiments are done with excitation directions x and y and for different versions of the active magnetic bearing position controller. The excitation current consists of a series of triangular impulses; the frequency spectrum of the windowed excitation signal contains a narrow band that covers the first backward and forward gyroscopic modes of the rotor. Within this frequency band, natural frequencies, damping ratios, and mode shapes are identified. The results demonstrate the suitability of the chosen excitation and enable a comparison of active magnetic bearing position controllers that should stabilize the first backward gyroscopic mode.

1 Introduction

Self-excited vibrations of rotors may be caused by various effects. Some examples are given by internal damping, the skew-symmetric component of the fluid film bearing stiffness matrix, or circulatory forces originating from seals [3]. In a linearized model, such effects can give rise to the instability of individual vibration modes if the modal damping of the remaining system is too low. Instead of instability, nonlinearities often lead to limit cycles, which may still be unwanted in a real system.

If a rotor is supported by active magnetic bearings, there are some special reasons to keep its limit cycles as small as possible. First of all, the rotor must not touch the backup bearings. Moreover, the energy consumption of the active magnetic bearings depends on the vibration amplitudes of the rotor. If an active magnetic bearing position controller has to deal with a limit cycle, some control current will always be necessary, and the respective bearing will require more energy. On the other hand, the damping of individual vibration modes can be influenced by the active magnetic bearing position control, and the active magnetic bearing system includes excitation and measurement devices that should enable the identification of modal properties.

However, the experimental modal analysis of rotors in active magnetic bearings poses several difficulties. Unbalance excitation is always present and unbalance response adds to the response from the modal testing excitation. This requires a sufficient excitation level. On the other hand, responses should stay within the linear range of the active magnetic bearings and are in any case limited by the clearance of the backup bearings.

In [5], speed dependent modal models of a gyroscopic rotor in active magnetic bearings were obtained with impulsive excitation from one of the active magnetic bearings, and the gyroscopic split of natural frequencies was demonstrated for the four rigid body modes. Due to the application of an exponential window, modal damping values were much exaggerated; apart from that, a short single impulse leads to a poor frequency resolution and would not yield reliable damping values. However, impulsive excitation guarantees both a sufficient excitation level and a limited response.

In this paper, a similar rotor is investigated. The rotor consists of a shaft that carries a cantilevered disc and is run at rotational speeds up to 60000 rpm. Besides unbalance response, the rotor exhibits a tendency to vibrate in its first backward gyroscopic mode. This results in an unwanted limit cycle under certain operating conditions where self-excitation is present. Such vibrations shall be prevented by appropriate measures, which can be evaluated if the respective modal damping is known. For different rotational speeds, the modal damping of the first backward and forward gyroscopic modes is therefore identified from narrow-band measurements with a suitable arrangement of several impulses. Identification results are used to compare different versions of the active magnetic bearing position controller. In addition, the first backward and forward gyroscopic mode shapes are extracted.

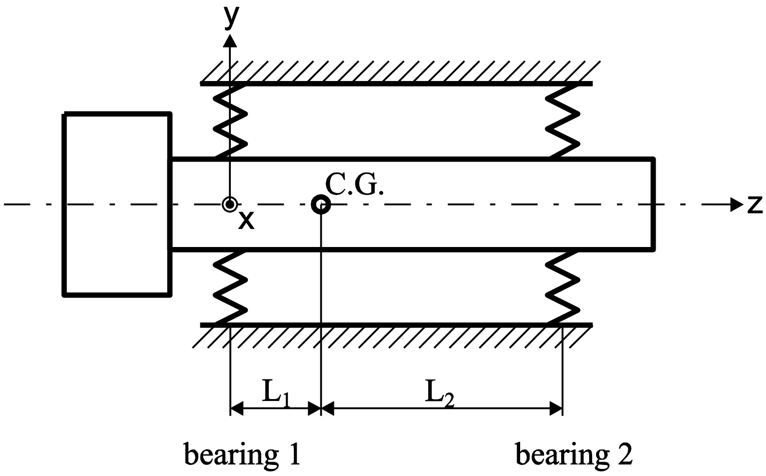

Figure 1: Schematic of rotor supported by active magnetic bearings.

2 Experimental setup and previous measurements

The setup of the system identified in this paper is similar to the experimental setup used in [5]; slight differences between individual mechanical or electrical components as well as active magnetic bearing controller hardware or software may have occurred. Figure 1 shows a schematic of the rotor. It consists of a shaft that is supported by two radial active magnetic bearings and carries a cantilevered disc at one of its ends. Currently, the position of the rotor is stabilized by four independent PID position controllers. The system under investigation is constituted by the rotor in its active magnetic bearings. Two perpendicular position sensors are situated immediately next to each bearing so that a four-degrees-of-freedom model can be studied.

Under practical operating conditions, self-excited vibrations in the first backward gyroscopic mode of the rotor have been experienced. In the experimental setup, the rotor is taken out of its intended application and is thus running without any self-excitation from process forces.

Broadband measurements aiming at the identification of all four rigid body modes were performed in [5]. An impulsive excitation force was generated by bearing 1 in direction x, and the first pair of modes was identified. The experiment was repeated with an impulsive excitation force from bearing 1, direction y, yielding another result for the first pair of modes. Similar experiments with the excitation from bearing 2 gave two results for the second pair of modes. These measurements were conducted from 0 to 100000 rpm and should give an overview of natural frequencies and mode shapes for the entire rigid body frequency range. An exponential window had to be applied to all vibration responses, which made modal damping values come out much larger than in the physical system. Furthermore, the shortness of the windowed vibration responses reduced the true frequency resolution to an extent that led to considerable inaccuracies in all modal parameters. The first forward gyroscopic mode never came out clearly, and natural frequencies of the rotating system depended on the excitation direction. Still, identification results demonstrated the gyroscopic split of natural frequencies and suggested that a greater length of the relevant vibration response time signal should lead to better results.

3 Narrow-band excitation

To improve the accuracy of modal parameters and obtain reliable damping values, it would be an obvious idea to repeat the experiments with random noise or sinusoidal excitation as described by Lee, Ha, and Kim [4]. However, a number of arguments remain in favour of impulsive excitation. The maximum value of the impulse can be chosen just as high as allowed by the linear operating range of the active magnetic bearings, which should make sure that the modal testing response can be distinguished from unbalance and natural vibrations that are not caused by the impulse. Furthermore, an impulsive excitation current is easy to implement in the existing active magnetic bearing software. It is generated by adding two rectangular pulses of different signs an lengths to the reference current from the position controller as explained in [5]. Due to the dynamic behaviour of the current control, an impulse results in the actual current. The shortness of the impulse allows the measurement of its shape by switching off the position control for a few milliseconds. This avoids uncertainties in the actual excitation current arising from any assumed or identified behaviour of the current control.

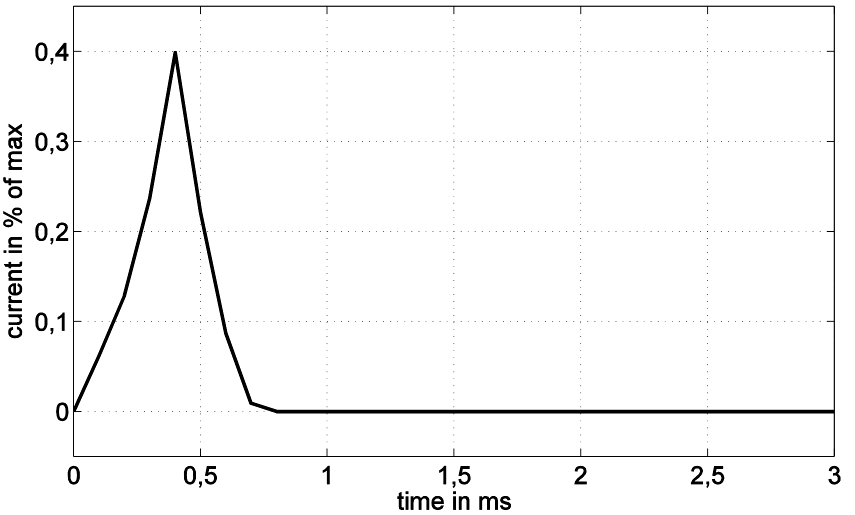

Figure 2: Single excitation current impulse.

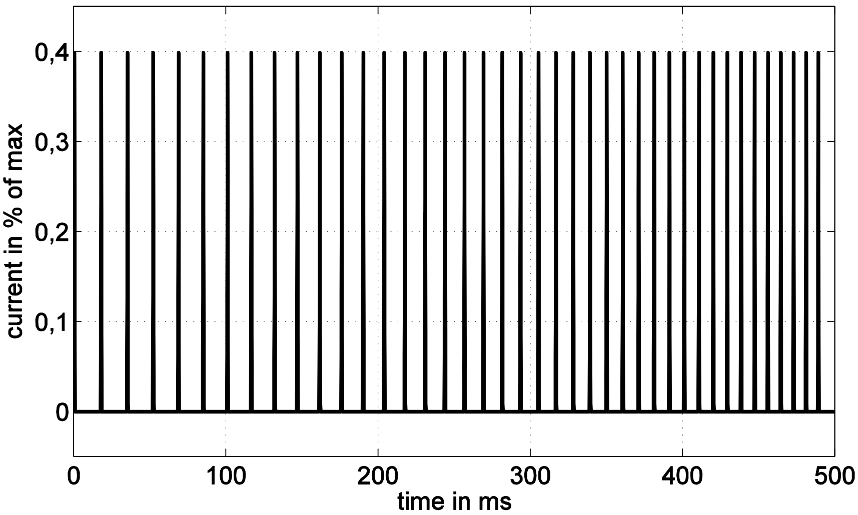

Figure 3: Series of excitation current impulses.

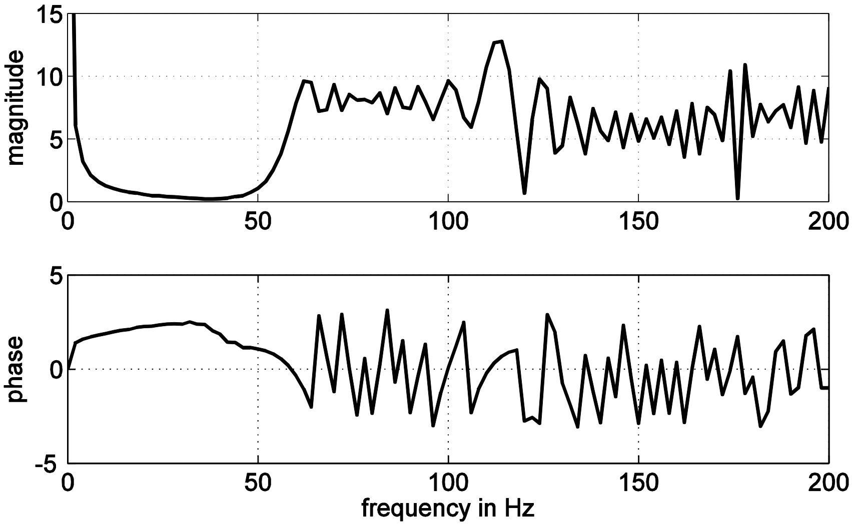

Figure 4: Frequency spectrum of excitation.

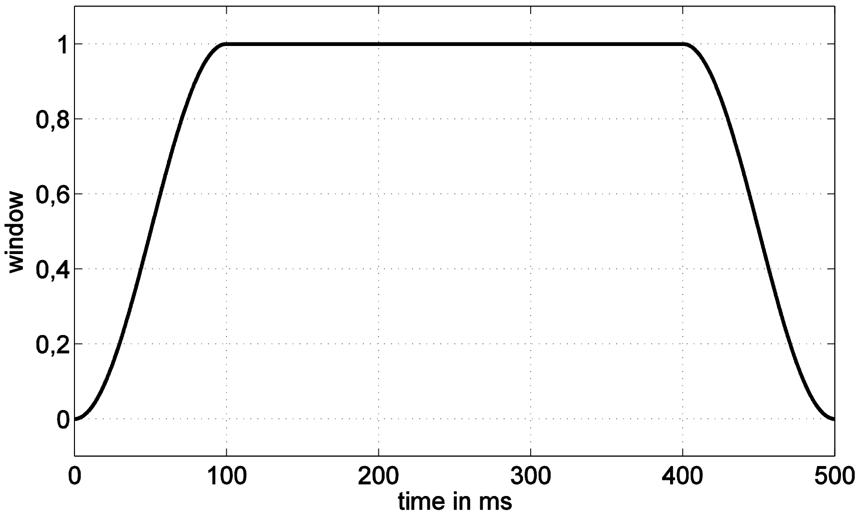

Figure 5: Window function.

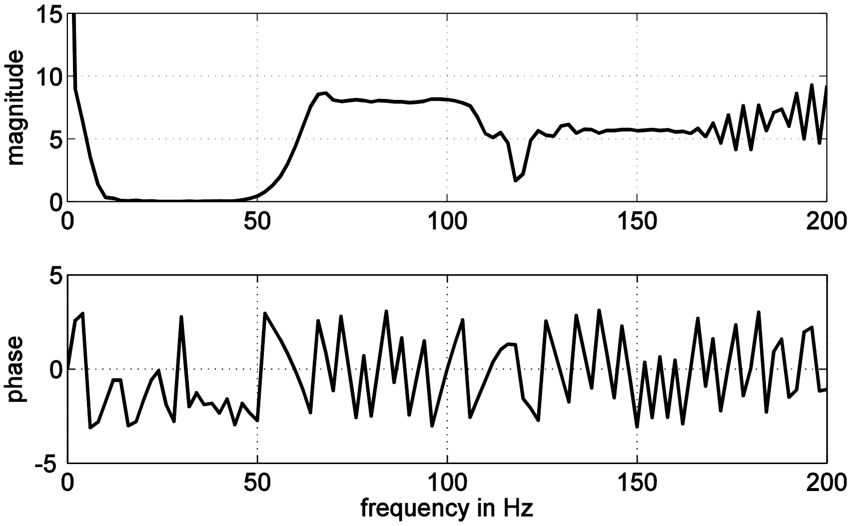

Figure 6: Frequency spectrum of windowed excitation.

Now, a single impulse has the obvious disadvantage that the resulting relevant vibration response time signal is too short. A suitable arrangement of several impulses in one measurement may do better. However, it has been emphasized in [2] that with impulsive excitation, multiple impacts should be avoided because they generate "holes" in the excitation spectrum, resulting in divisions by zero when frequency response functions are calculated. A series of impulses can therefore not be used in a broadband measurement, but may well be suited for a narrow-band test. Other than with unintended multiple hammer impacts, the maximum value of an impulse is a sort of an optimum in the present experiments and should therefore stay constant, while the time between two impulses may vary.

The shape of a single excitation impulse is displayed in Fig. 2. The bearing coil current is assumed to generate a proportional magnetic force. Figure 3 shows the time signal of the selected excitation. The time between two impulses is gradually reduced. Instead of exciting a certain fundamental frequency, this aims at covering a frequency band. The resulting frequency spectrum can be seen in Fig. 4. A usable frequency band emerges between 70 and 105 Hz, but the magnitude is still not very smooth. By the application of a window as displayed in Fig. 5, the time signal is distorted in a way that results in a reasonable shape of the frequency spectrum in the excitation frequency band. Figure 6 shows the frequency spectrum of the windowed excitation. For the calculation of frequency response functions, the same window is applied to all vibration responses so that the window does not change any modal damping values.

4 Frequency response functions

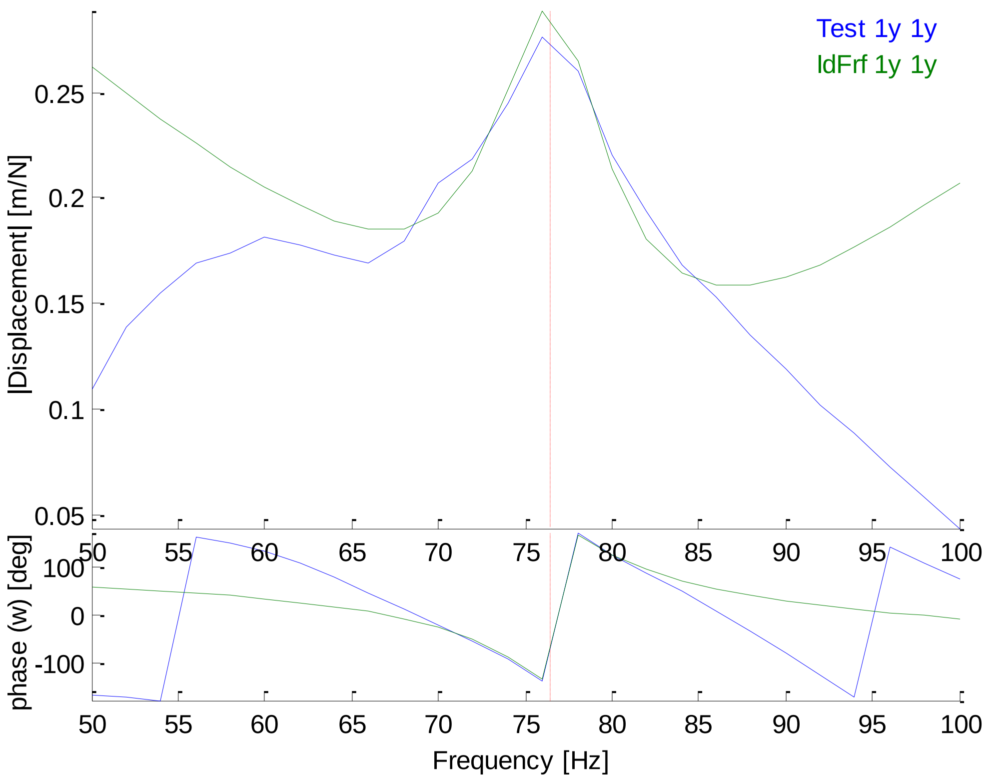

Figure 7: Measured and estimated point frequency response function, excitation at bearing 1 in direction y, 0 rpm, PID controllers with small differential part.

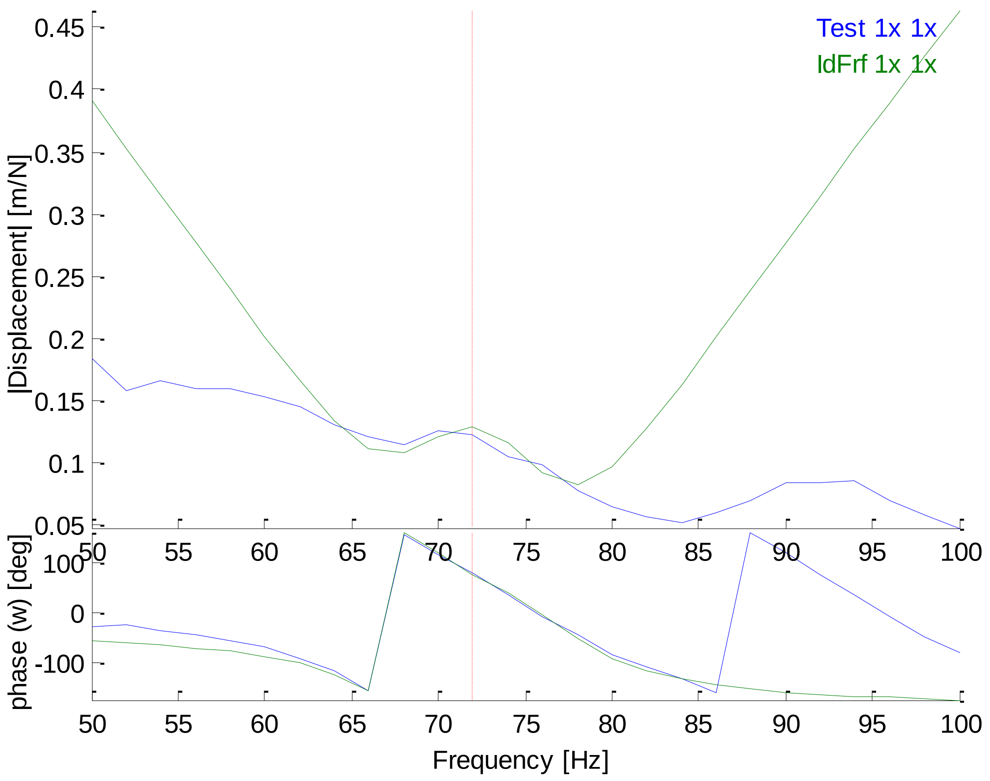

Figure 8: Measured and estimated point frequency response function, excitation at bearing 1 in direction x, 60000 rpm, PID controllers with small differential part.

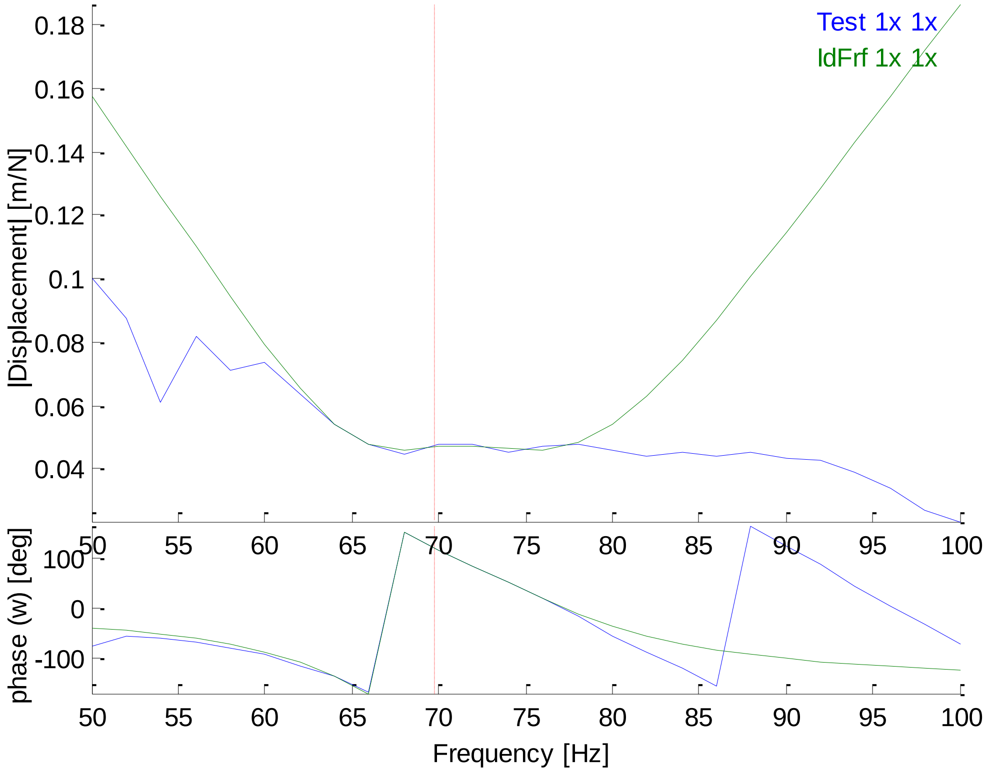

Figure 9: Measured and estimated point frequency response function, excitation at bearing 1 in direction x, 60000 rpm, PID controllers with large differential part.

The excitation frequency band between 70 and 105 Hz enables the investigation of the first pair of modes. As it was found in [5], these modes are likely to be excited from bearing 1, but may not respond to an excitation from bearing 2. The excitation is therefore confined to bearing 1. Excitation directions x and y are used in separate tests; if the system is rotating, the resulting modal parameters should agree. Modal tests are performed at rest and at rotational speeds of 20000, 40000, and 60000 rpm. Furthermore, two versions of the active magnetic bearing position controller are compared. In each modal test, four vibration responses are measured. Excitation signal and vibration responses are windowed, frequency response functions are calculated by Discrete Fourier Transform and averaged over four measurements. The identification is done by the Matlab Structural Dynamics Toolbox [1].

Figures 7, 8, and 9 give a few examples of measured and estimated frequency response functions. For the rotor at rest, active magnetic bearing PID controllers with a small differential part, and the excitation applied at bearing 1 in direction y, Fig. 7 shows the identification of a resonance at 77 Hz.

For the rotor rotating at 60000 rpm, active magnetic bearing PID controllers with a small differential part, and the excitation applied at bearing 1 in direction x, the identification of the first backward gyroscopic mode at 72 Hz is shown in Fig. 8; the first forward gyroscopic mode also appears in the measured curve. As the MDOF estimator finds it difficult to curve fit the measured frequency response functions, only SDOF estimates are used. For close modes with high damping, this may not be satisfactory and could be avoided by a higher frequency resolution of the frequency response functions, which usually improves the situation for the MDOF estimator. A better placement of the excitation frequency band could also help; these matters are further discussed in Subsection 5.4.

For the rotor rotating at 60000 rpm, active magnetic bearing PID controllers with a large differential part, and the excitation applied at bearing 1 in direction x, the identification of the first backward gyroscopic mode at 70 Hz is displayed in Fig. 9. Both backward and forward resonances are less distinct than in Fig. 8, which indicates that modal damping is generally higher.

5 Identification results

5.1 Natural frequencies

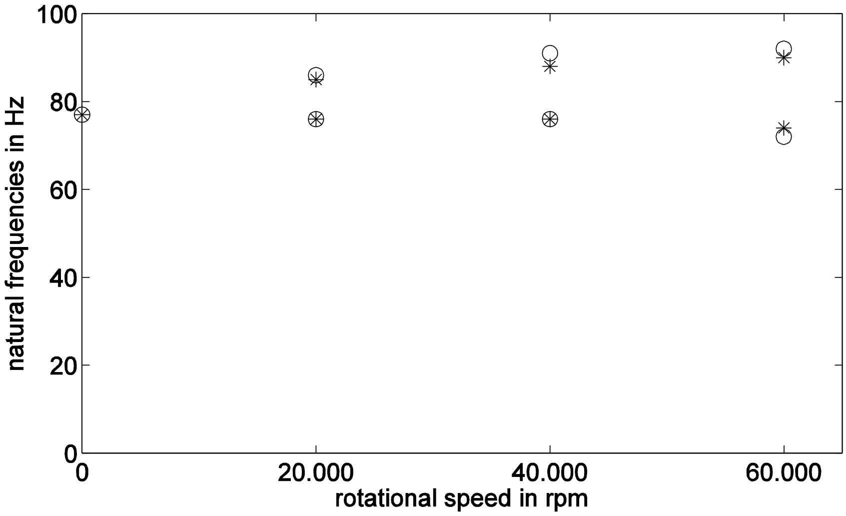

Figure 10: Natural frequencies depending on the rotational speed, PID controllers with small differential part, excitation at bearing 1, directions x (o) and y (*).

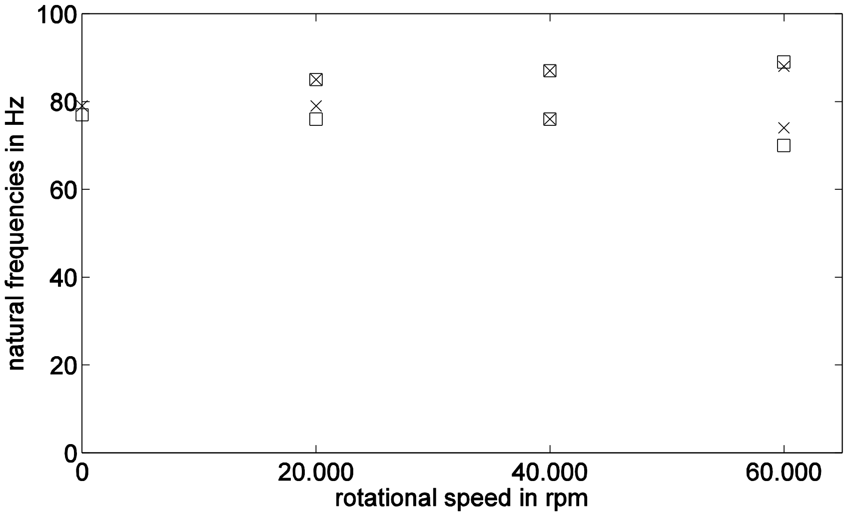

Figure 11: Natural frequencies depending on the rotational speed, PID controllers with large differential part, excitation at bearing 1, directions x (□) and y (×).

Figures 10 and 11 show an overview of natural frequencies extracted from the present measurements. Identification results agree fairly well between excitation directions x and y, which gives good confidence in the accuracy of the results. Natural frequencies hardly differ between active magnetic bearing PID controllers with small and large differential part. Two natural frequencies can be distinguished from 20000 rpm upwards, which demonstrates the onset of the gyroscopic split between the first backward and forward natural frequencies. In [5], this effect was only captured at 60000 rpm.

5.2 Modal damping

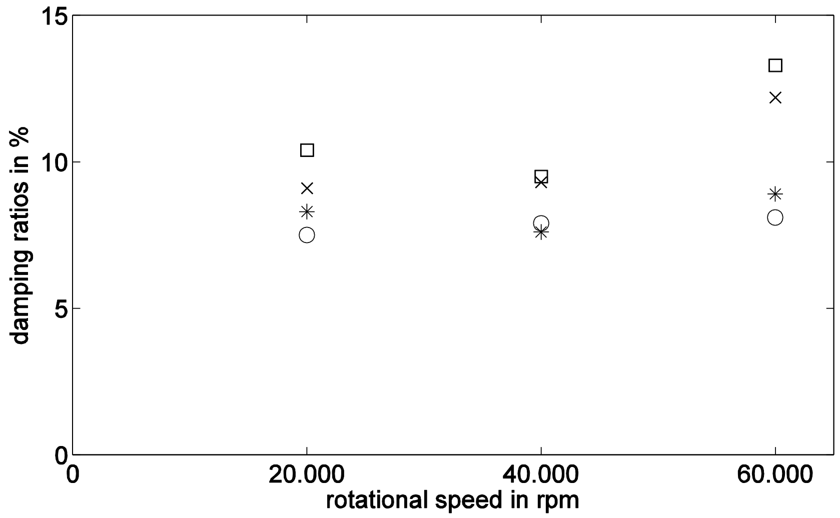

Figure 12: Modal damping of first backward gyroscopic mode depending on the rotational speed, PID controllers with small differential part, excitation at bearing 1, directions x (o) and y (*); PID controllers with large differential part, excitation at bearing 1, directions x (□) and y (×).

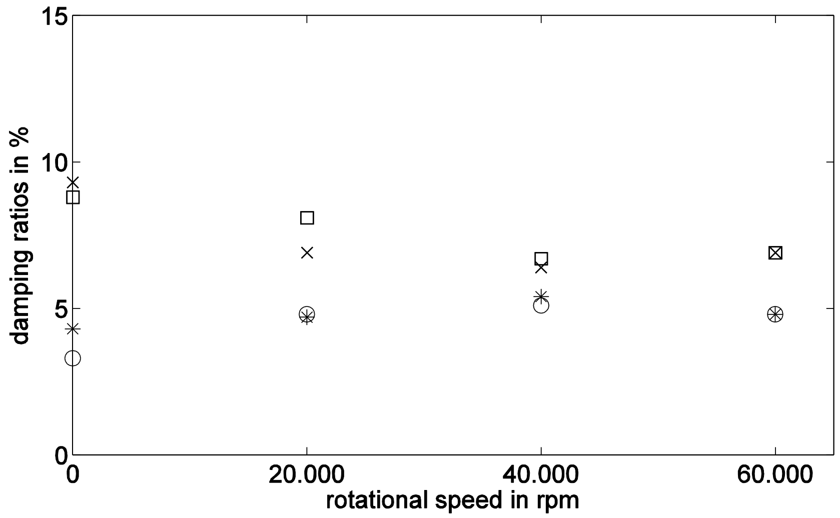

Figure 13: Modal damping of first forward gyroscopic mode depending on the rotational speed, PID controllers with small differential part, excitation at bearing 1, directions x (o) and y (*); PID controllers with large differential part, excitation at bearing 1, directions x (□) and y (×).

Figure 12 shows the damping ratios of the first backward gyroscopic mode. For active magnetic bearing PID controllers with a small differential part, they lie between 7.5 and 8.9 %. For active magnetic bearing PID controllers with a large differential part, they range from 9.1 to 13.3 % and are thus always higher than for comparable measurements with small differential part. This was to be expected since an increase in the differential part of the active magnetic bearing PID controller leads to a higher bearing damping. In most cases, identified damping ratios match fairly well among excitation directions x and y.

Figure 13 contains the damping ratios of the first forward gyroscopic mode for active magnetic bearing PID controllers with small and large differential parts. As a reference, modal damping ratios at rest are also included; here, the deviation between excitation directions x and y can be explained by an anisotropy in the relevant bearing damping. For the rotating system, identified damping ratios mostly agree as they should; again, higher bearing damping leads to higher modal damping ratios.

Although in all comparable cases, the modal damping of the first backward gyroscopic mode is higher than the modal damping of the first forward gyroscopic mode, the backward mode is less stable under practical operating conditions. This cannot be explained by the present measurements and must have to do with the nature of the self-excitation from process forces.

5.3 Mode shapes

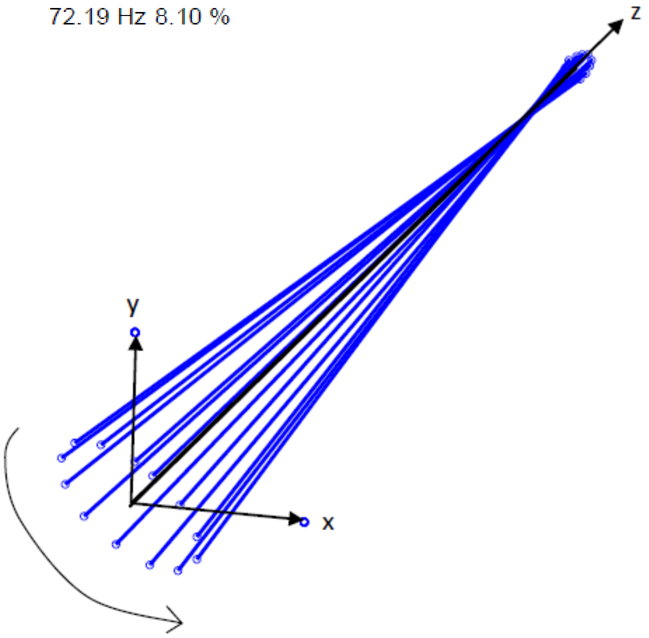

Figure 14: First backward gyroscopic mode shape at 60000 rpm, PID controllers with small differential part, excitation at bearing 1, direction x.

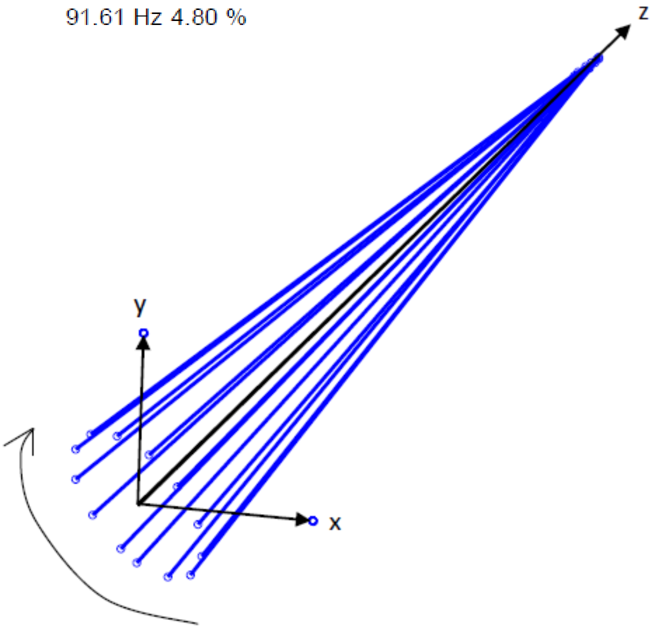

Figure 15: First forward gyroscopic mode shape at 60000 rpm, PID controllers with small differential part, excitation at bearing 1, direction x.

The first backward and forward gyroscopic mode shapes are conical in all measured cases. Extracting the first forward gyroscopic mode appears to be a special problem of the given configurations as it was never successful in [5]. In the present measurements, mode shape identification works best for PID controllers with a small differential part and the excitation applied at bearing 1 in direction x; at 40000 and 60000 rpm, both backward and forward mode come out with good quality. Figs. 14 and 15 show the results for the rotor rotating at 60000 rpm.

5.4 Accuracy

The accuracy of the identified modal parameters is obviously limited by the frequency resolution of the frequency response functions, which equals 2 Hz. At a sampling frequency of 10 kHz, which is required by a sufficient resolution of the excitation current impulse, this results from the measurement time of 500 ms, a limit given by the signal processor if four position signals are to be measured simultaneously. The measurement time could be extended by acquiring each position signal separately, thus increasing the frequency resolution. Also, the excitation frequency band could be re-adjusted to better fit the individual natural frequencies, and more averages could be taken. However, the present results clearly show the expected tendency of the identified modal damping for a given change in the active magnetic bearing position controller, and this is as much as can be expected within the resources given for the present experiments. Confidence in the results is gained by the extent to which the identified modal parameters agree between excitation directions x and y.

6 Conclusion

In this paper, a series of excitation current impulses has been used for the modal parameter identification of a gyroscopic rotor in active magnetic bearings. As opposed to broadband excitation from a single impulse, the frequency band of interest is confined to a narrow range that covers the first backward and forward gyroscopic modes of the rotor for rotational speeds up to 60000 rpm. These restrictions lead to significant improvements in the accuracy of the results. Natural frequencies identified from excitation directions x and y are in good agreement, and in some cases, backward and forward modes come out clearly. The quality of the results could still be enhanced by a better adjustment of the frequency band, a higher frequency resolution, and a more suitable MDOF estimation algorithm.

By the application of the same window to both excitation and response, modal damping ratios are left unchanged. Compared to measurements with damping exaggerated by exponential windows, this makes it easier to measure the onset of the gyroscopic split. Modal damping ratios identified from excitation directions x and y are in good agreement. Although the first backward gyroscopic mode is prone to instability under practical operating conditions, its measured damping ratio is always higher than the damping ratio of the respective forward mode.

Modal damping ratios of the first backward gyroscopic mode have been compared between active magnetic bearing PID controllers with small and large differential parts. Not surprisingly, large differential parts lead to higher modal damping. For more advanced versions of the active magnetic bearing position controller, the influence on the modal damping of the first backward gyroscopic mode will be less obvious. The resistance of the system against self-excitation can then be assessed by modal damping identification. Under operating conditions in which self-excitation is present, this serves to reduce the energy consumption of the active magnetic bearings and also reduces the risk of touching the backup bearings.

References

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