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# Practical Controller Design for Rotors on Magnetic Bearings by Means of an Efficient Simulation Tool

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**Abstract**

This paper describes in detail by means of a realistic example and an efficient simulation tool the whole procedure of the magnetic bearing controller design. In a first step the suitability of the rotor is assessed. The objectives for the actual controller design to ensure a robust system, which is suited for the application, are then explained. In a first step of the design a controller structure, which is a series of proven filters with stable poles, is defined. The strategy to select the filters is determined by the rotor properties and the hardware characteristics of the bearing. The parameters of the controller are then optimized to fulfil the objectives. The optimization can be automated by the definition of suited goals, which are closely related to the objectives. Finally the performance of the resulting rotor bearing system is assessed.

## 1 Introduction

In the last years magnetic bearings again observe an increasing interest in the gas compression industry. The main motivation to use magnetic bearings is the design of hermitically sealed compressors, which require bearings without lubrication.

Besides allowing oil free machines magnetic bearings have further advantages: They are almost frictionless and therefore have low power losses and practically no wear; they allow high speed, which is limited only by the stresses in the lamination; they can be remotely controlled and are a good monitoring device, since the bearing force can be measured by the current.

However, magnetic bearings also have important restrictions. Their load capacity corresponds to only about one tenth of the capacity of oil bearings. The dynamic capacity is further limited due to the power limit of the amplifiers (see for example [1]). The engineering of rotors on magnetic bearings therefore requires considerable effort.

An important part of the engineering is the controller design. Many concepts for the controller design such as H-infinite, sliding mode control and μ-synthesis are discussed in the literature (see for example [2]). In this paper a controller design procedure is described, which basically consists of two steps. In a first step a controller is designed by a series of proven stable filters. The selection of the filters hereby depends on the rotor and magnetic hardware properties. This approach is widely used in the industry. In a second step the parameters of the controller with a given structure are optimized. This process can be automatized by defining appropriate goals and utilizing an optimization algorithm. The automated optimization helps to considerably reduce the effort for the design and to ensure, that the parameters for a given controller structure are optimally set. The combination of defining the controller structure with engineering insight followed by an automated parameter optimization with suited goals is new to the authors' knowledge.

## 2 Description of the rotor and the magnetic bearings

The example rotor is shown in figure 1. It is a compressor rotor supported on two magnetic bearings. The blue triangles mark the actuator positions and the blue fat lines the sensor positions. Sensor and actuator are non-collocated, which is taken into account for all analyses.

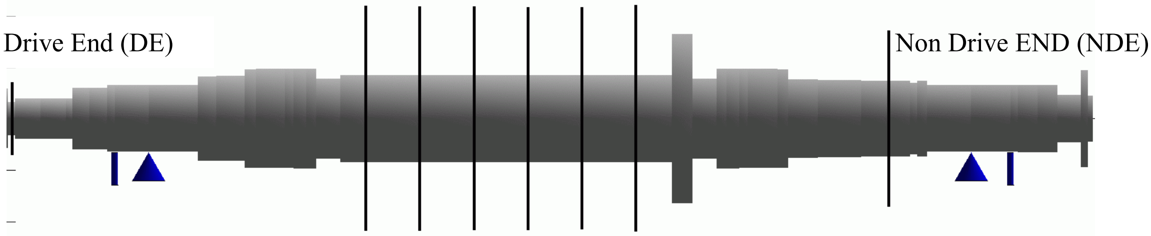



**Figure 1:** Example rotor, weight 550kg, maximum speed 12’600rpm.

The magnetic bearings have a capacity of 5000N per axis. The axes are arranged under 45°, i.e. two axes contribute to carrying the weight load of the rotor. The dynamic capacity of the bearing drops below the static capacity above a frequency of 200Hz due to amplifier saturation (because of the limited power of the amplifier and the inductance of the bearing the current is limited).

The dynamic characteristic of the bearings including the amplifier, the sensor, the zero-order-hold-behaviour (see [3]) and the processing time of the digital signal processor is shown as a force/displacement transfer function in figure 2. The sampling rate is 13 kHz causing a zoh delay of half the sampling rate of 38.5 µs. The processing time of 20 µs adds to this delay. A further delay corresponding to about 124 µs is caused by the switched amplifiers, which includes the sampling. Thus the main effect of the hardware characteristic is a phase loss according to these delays. In order to create a damping force, the phase loss must be compensated by the controller.

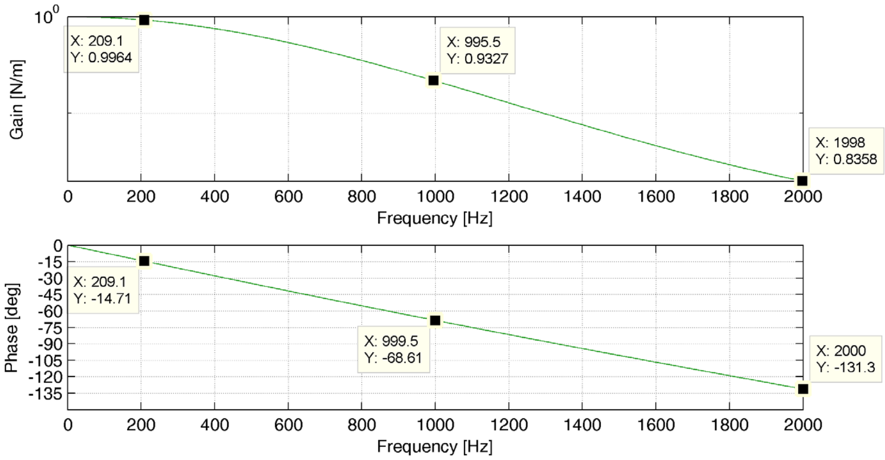



**Figure 2:** Transfer function of the bearing hardware including digitization effects.

## 3 Basic analyses of the rotor, suitability of the rotor

Basic analyses of the rotor without detailed modelling of the magnetic bearing are carried out to support the controller design and to assess the suitability of the rotor for magnetic bearings.

The natural modes of the rotor in free condition provide valuable information. They reveal, which modes can be influenced by the bearing and which modes will most likely be in the operating speed range, since the resulting magnetic bearing stiffness typically is very low. At the same time it is useful for the controller design to know, at which bearing stiffness values the frequencies of the natural modes start to shift to higher values and to what extent the natural frequencies can shift due to the gyroscopic effect. For these reasons an eigenvalue analysis at maximum speed with a stiffness variation of the bearing starting at a low value is carried out in a first step. The natural frequencies up to a frequency of 2000 Hz are shown in figure 3. The shown maximum frequency is much higher than the maximum speed, because the high modes must be considered in the controller design. The frequency pairs in the figure belong to the backward (lower frequency) and forward (higher frequency) whirling natural modes of the different shapes. Since the frequency at zero speed is between these pairs and the forward mode frequencies rise, whereas the backward mode frequencies decrease with speed, the figure reveals the range of the natural frequency for each mode.

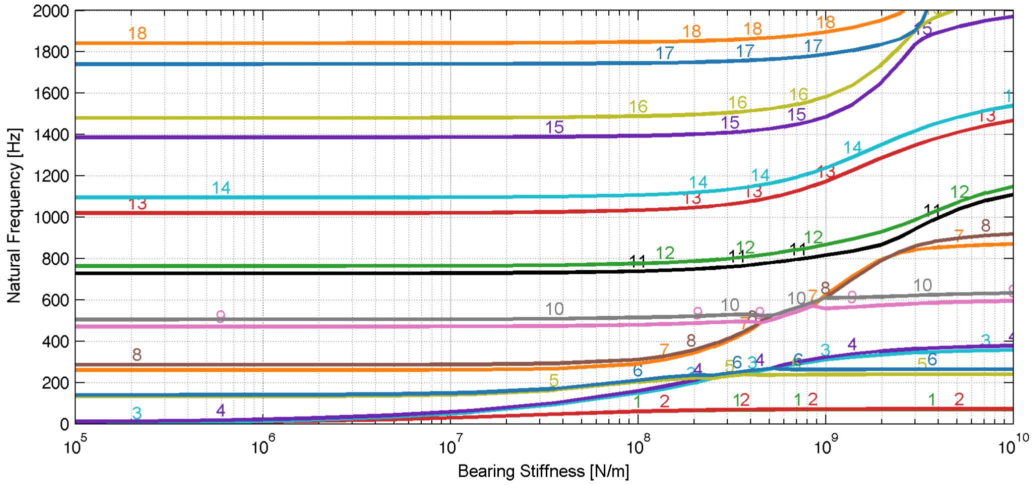



**Figure 3:** Frequency of natural modes for a stiffness variation.

The natural modes in the stiffness variation diagram up to 1000 Hz can be seen in figure 4 for a stiffness of $10^4$ N/m, i.e. for the practically free rotor. The shapes allow assessing, which mode can be influenced by the bearing. A node at the sensor location means that the mode is not observable and a node at the actuator means it is not controllable. In both cases the bearing cannot influence the mode, i.e. cannot damp it, but also not excite it.

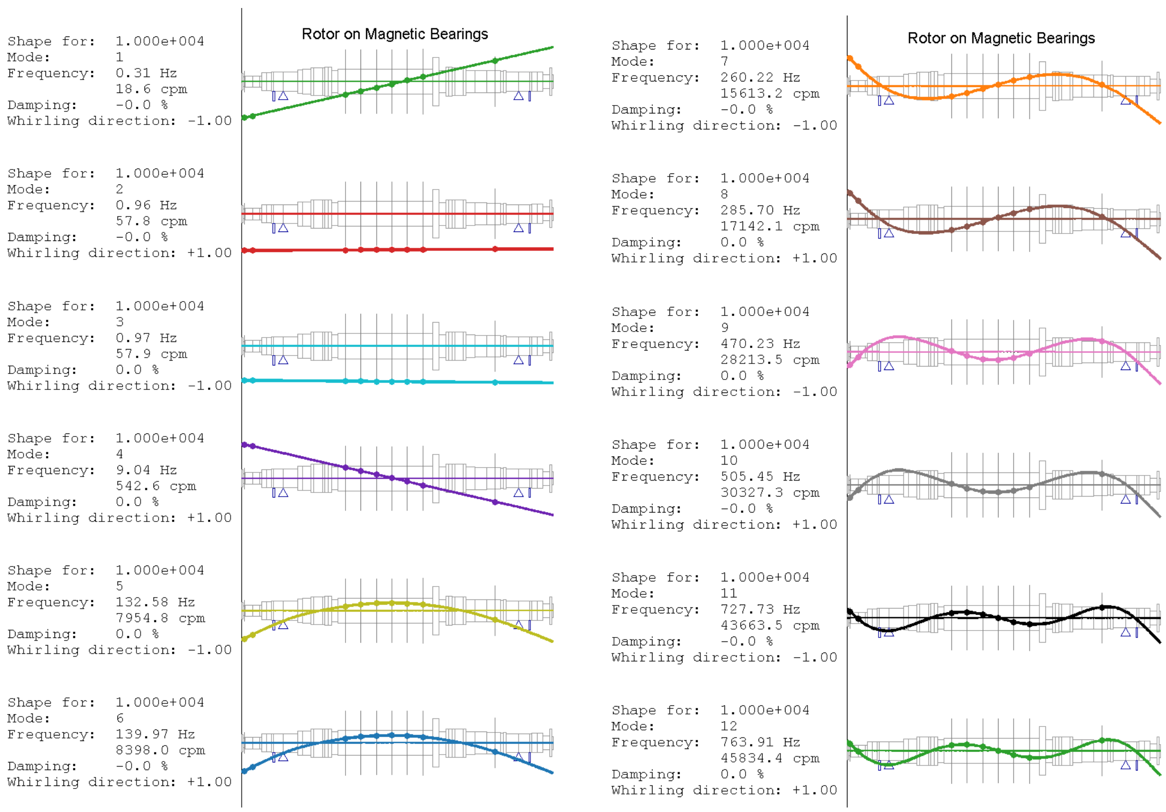



**Figure 4:** Natural modes at 12’600 rpm for a stiffness of $10^4$ N/m (practically free).

The first bending mode (mode 5, 6) is below the maximum speed of the rotor. For this mode a good observability and controllability is essential, since it should be well damped. The second bending mode (mode 7, 8) is above maximum speed. This mode should have a sufficient separation margin. It must not necessarily be well damped, but has to be stable. The same applies for the third and higher bending modes. It can be seen, that the second bending mode is not observable at the DE bearing and that the third and fourth mode are not well controllable and observable, respectively, at the NDE bearing.

The observability and controllability can be better recognised in the report shown in table 1 for all modes to 2000 Hz. The values at the sensor and actuator nodes correspond to the deflection of the shape at these locations relative to the maximum deflection. Moreover the angles dA1 and dA2 between the sensor and actuator deflection are shown. An angle of 180° means, that the mode has a node between sensor and actuator.

**Table 1:** Observability and controllability report.

Bearing 1: Shaft 1 Station 09 DE Bearing Sensor
Bearing 1: Shaft 1 Station 10 DE Bearing Actuator
Bearing 2: Shaft 1 Station 41 NDE Bearing Sensor
Bearing 2: Shaft 1 Station 40 NDE Bearing Actuator

| Mode | Freq. [Hz] | s1-09 | a1-10 | dA1 | s1-41 | a1-40 | dA2 | |
|---|---|---|---|---|---|---|---|---|
| 1 | 0.31 | 0.80 | 0.74 | 0 | 0.79 | 0.72 | 0 | Rigid body modes |
| 2 | 0.96 | 1.00 | 0.99 | 0 | 0.96 | 0.96 | 0 | |
| 3 | 0.97 | 0.94 | 0.94 | 0 | 0.99 | 0.99 | 0 | |
| 4 | 9.04 | 0.74 | 0.68 | 0 | 0.85 | 0.78 | 0 | |
| 5 | 132.58 | 0.46 | 0.34 | 0 | 0.70 | 0.56 | 0 | 1st bending |
| 6 | 139.97 | 0.47 | 0.34 | 0 | 0.70 | 0.56 | 0 | |
| 7 | 260.22 | 0.04 | 0.12 | 180 | 0.52 | 0.32 | 0 | 2nd bending |
| 8 | 285.70 | 0.00 | 0.15 | 0 | 0.51 | 0.29 | 0 | |
| 9 | 470.23 | 0.33 | 0.43 | 0 | 0.29 | 0.01 | 0 | 3rd bending |
| 10 | 505.45 | 0.32 | 0.43 | 0 | 0.27 | 0.01 | 180 | |
| 11 | 727.73 | 0.49 | 0.52 | 0 | 0.07 | 0.25 | 180 | 4th bending |
| 12 | 763.91 | 0.53 | 0.57 | 0 | 0.06 | 0.26 | 180 | |
| 13 | 1020.12 | 0.81 | 0.71 | 0 | 0.12 | 0.43 | 0 | 5th bending |
| 14 | 1095.23 | 0.81 | 0.71 | 0 | 0.16 | 0.47 | 0 | |
| 15 | 1386.19 | 0.72 | 0.42 | 0 | 0.31 | 0.56 | 0 | 6th bending |
| 16 | 1480.61 | 0.75 | 0.45 | 0 | 0.35 | 0.58 | 0 | |
| 17 | 1741.09 | 0.50 | 0.10 | 0 | 0.45 | 0.59 | 0 | 7th bending |
| 18 | 1842.34 | 0.53 | 0.13 | 0 | 0.49 | 0.60 | 0 | |

The Campbell diagram of the practically free rotor is shown in figure 5. It can be seen, that the 2nd bending forward mode has a sufficient separation margin of 38.5%. The API standard 617 for compressor [4] for example requires a separation margin of 26% for modes with low damping.

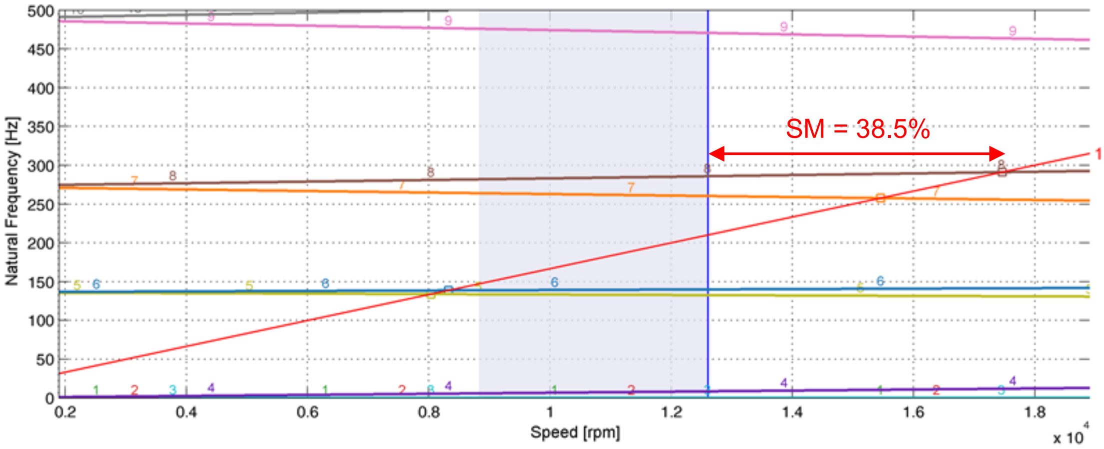



**Figure 5:** Campbell diagram of the free rotor.

In order to check the achievable damping for the first bending mode, which is below maximum speed, a damping variation for the practically free rotor is carried out. The natural frequencies and damping ratios of the modes up to the second bending mode are shown in figure 6. It can be clearly seen, that the 1st bending mode can be highly damped. The maximum damping ratio is 50% for a damping coefficient of about 100'000 Ns/m, whereas according to API 617 [4] only 20% damping ratio (corresponding to an amplification factor of 2.5) is necessary to run the rotor at the critical speed. This would be achieved for a coefficient of 35'000 Ns/m. The result of this analysis is another measure for the observability and controllability of modes. The second bending mode cannot be well damped, since it is much worse in this regard. However, due to the separation margin this is not necessary.

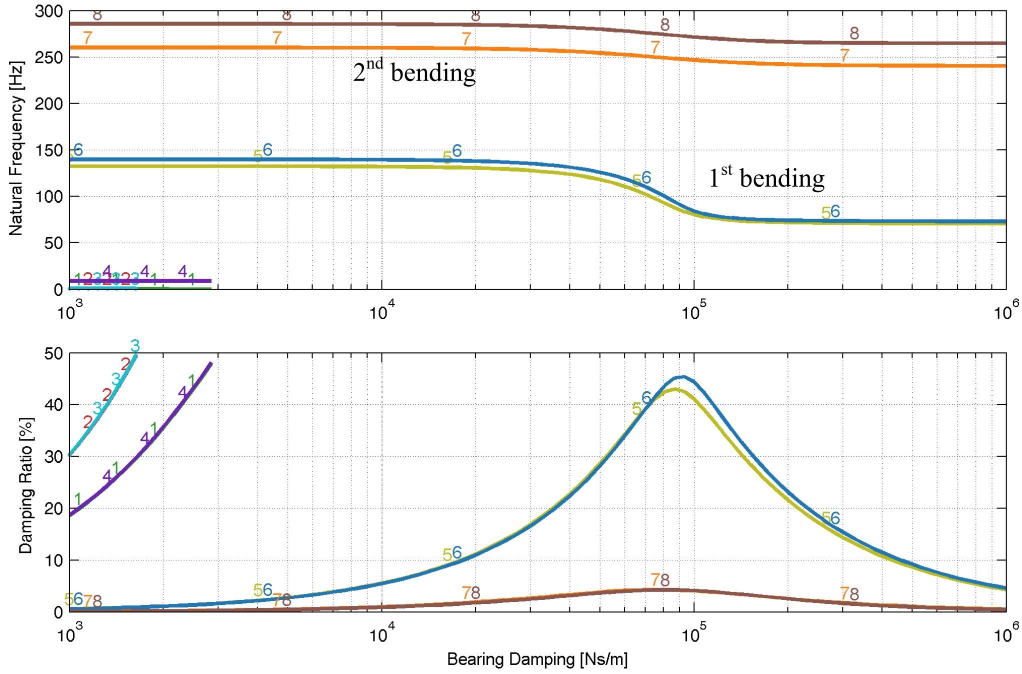



**Figure 6:** Frequency and damping ratio of the free rotor for a damping variation.

## 4. Controller design

### 4.1 Design objectives and difficulties in the design process

The objectives of the controller design are as follows:

1.) The system shall be stable considering all operating conditions. To ensure stability, fluid forces may have to be included, which can play a role in high pressure compressors. Structural damping mainly caused by the friction of shrunk parts can be considered. The value depends on the rotor design. For the present compressor a damping ratio of 0.1% is on the safe side.

2.) The system must be robust, i.e. it should remain stable in case of small model changes or deviations between model and real system. The robustness is checked by the sensitivity as defined in ISO 14839-3 [5]. According to this standard the sensitivity shall be less than 3.0 for newly commissioned machines and less than 4.0 for unrestricted operation. In this example a structural damping of 0.1% is also considered in the sensitivity analysis.

3.) Modes in the operating speed range must be well damped. For the present type of machine API 617 can serve as a guideline, which means that the damping ratio for these modes should be 20%.

4.) Dynamic bearing capacity shall not be reached at realistic unbalance levels, as they can occur in the field, considering all other loads. Other loads are mainly static (weight and side loads from the flow) or have low frequency (excitations from the flow).

Due to the magnetic pull, magnetic bearings are inherently unstable. Therefore the controller in the first place has to overcome this negative stiffness by providing sufficient positive stiffness. Typically the resulting stiffness should have the magnitude of the magnetic pull.

As we can see in figure 2, the bearing hardware has a characteristic with continuous phase loss, which means loss of damping. The controller therefore has to overcome this phase loss. For modes with high frequencies this is difficult, since the phase loss increases with frequency. In figure 7 the relation between phase angle, damping and stiffness is explained.

A controller providing too much stiffness pushes the natural frequencies to higher frequencies with higher phase loss, i.e. less or negative damping. Creating phase lead, i.e. overcoming phase loss and creating damping, also increases the gain (a pure damping force linearly increases with frequency). In combination with the phase loss this leads to unwanted high stiffness at high frequencies. Figure 3 helps to decide, what stiffness is acceptable to avoid problems due to this effect.

Because of this side effect when creating damping, it is not possible to achieve damping of any level by the magnetic bearing. In the present case the observability and controllability of the rotor would theoretically allow to create a damping ratio of the first bending far above 20%. However, for the mentioned reasons it does not make sense to aim for more than 20% damping ratio.

High gain, which is a result of aiming for high damping, also increases the sensitivity and the bearing forces. The design objective 3 therefore is contradictory to the objectives 2 and 4. To ensure stability including fluid forces may also require a high gain. Objective 1 therefore can also be contradictory to objectives 2 and 4. Moreover high gain can cause problems due to noise, saturating the amplifiers. For these reasons it can be difficult to find an optimum for all goals and compromises may be necessary.

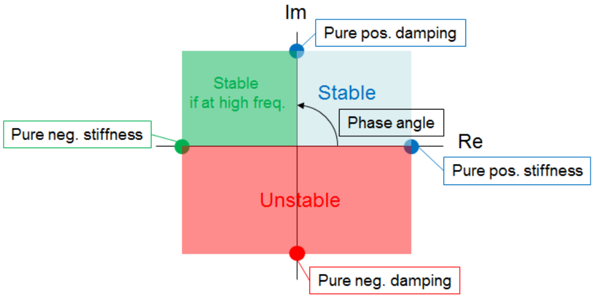



**Figure 7:** Relation between phase angle, damping and stiffness.

### 4.2 Proven filters for the design

In order to achieve the controller objectives, filters with suited properties are combined in series. In the following the properties of the most important available filters and the purpose they serve are described.

To provide good damping in the speed range of the machine, the controller has to provide phase lead. A suited controller building block with phase lead cells is shown in figure 8. It is called "base block" in the software MADYN 2000 (see [6]). The formula to the right of the figure is the transfer function with $s$ as the Laplace operator. In the graph $P$ is set to zero. Phase lead is achieved by setting the frequencies $f_{n1}$ and $f_{n2}$ lower than $f_{d1}$ and $f_{d2}$. It can be seen, that the phase lead causes a gain increase. The second term with s in the denominator is an integrator. By setting $P_{n1}$ to zero and using $P$ instead, a PID controller can be realised. The phase lead and the gain increase then are less controlled. Therefore using $P_{n1}$ is preferable in many cases.

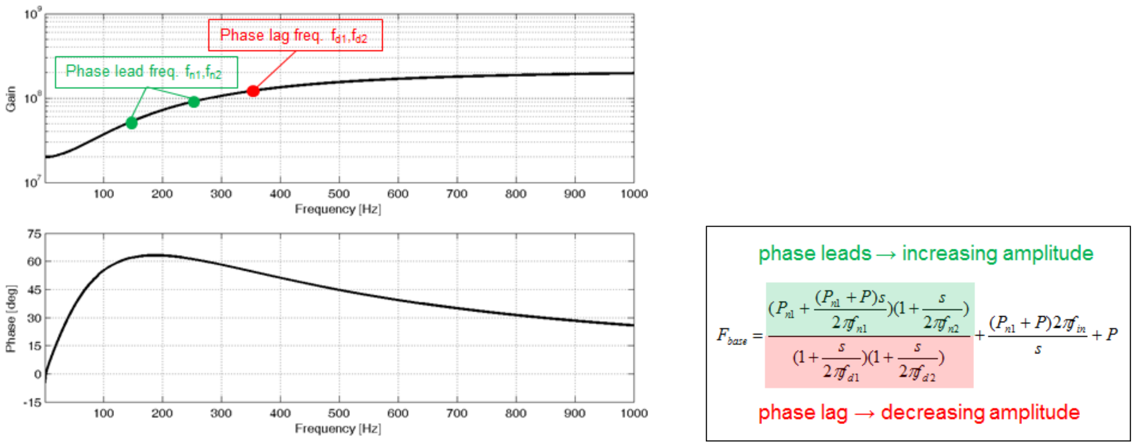
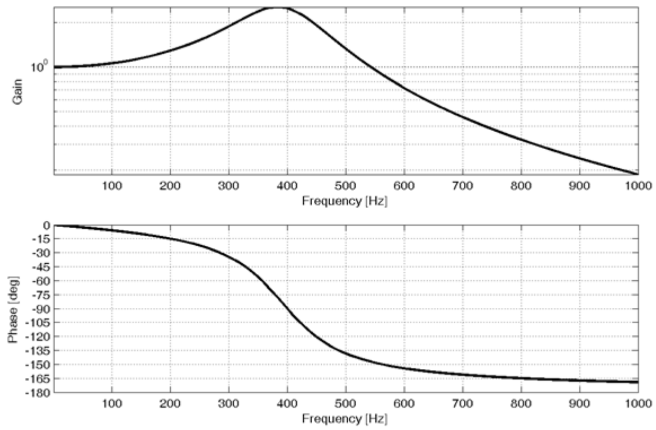



**Figure 8:** Base controller building block with phase lead cells, $P$=0, $f_{n1}$=150 Hz, $f_{n2}$=250 Hz, $f_{d1}$=$f_{d2}$=350 Hz

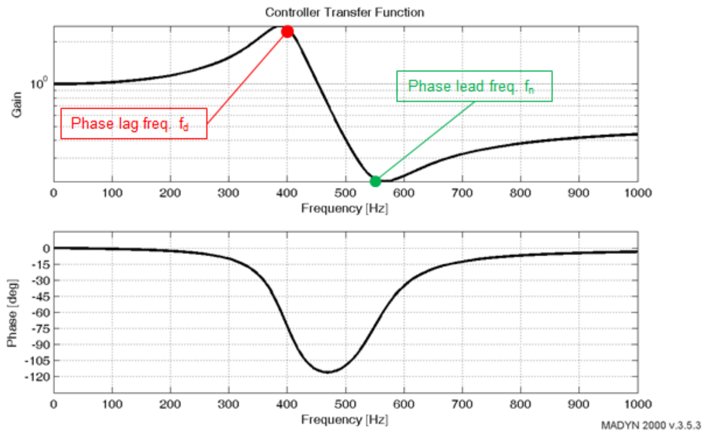
In the higher frequency range it can become very difficult to maintain a positive phase angle. Therefore the phase will be dropped intentionally in a suited frequency range to a value below -180˚, where the controller provides damping again, however, combined with negative stiffness (see figure 8). In the high frequency range, the negative stiffness force does not cause instability; it will only reduce natural frequencies slightly. For the phase roll off a second order filter as shown in figure 9 can be used. The lower the damping parameter $D$ is set, the faster the phase can be dropped. However, a low damping parameter also causes a high gain in the resonance of this filter. A suited frequency range for the drop means, that there are no well observable and controllable natural modes of the rotor in the range with a phase angle between zero and -180°.

A powerful filter to influence the phase angle in a limited frequency range is the general second order filter shown in figure 10. In the example the phase is dropped in a limited frequency range. Setting $f_d$ higher than $f_n$ would cause a phase increase. The phase can also be influenced by different $D_n$ and $D_d$ values.




$$F_{filt2} = \frac{P1}{1+\frac{2D}{2\pi ff}s+\frac{s^2}{(2\pi ff)^2}} + P2$$

**Figure 9:** 2nd order filter, $P2$=0, $ff$=400 Hz, $D$=20%.




$$F_{filtgen2} = P\frac{\frac{s^2}{(2\pi f_n)^2}+\frac{2D_n s}{2\pi f_n}+1}{\frac{s^2}{(2\pi f_d)^2}+\frac{2D_d s}{2\pi f_d}+1}$$

**Figure 10:** General 2nd order filter, $f_d$=400 Hz, $f_n$=550 Hz, $D_n$=$D_d$=10%.

### 4.3 Design strategy for the present compressor

The observability and controllability of the natural modes of the rotor shown in figure 4 and 5 offer the following strategy: The second bending mode is not observable on the DE side. Therefore the phase angle of the DE bearing can be rolled off in the frequency range of 200 to 450 Hz. On the NDE the 3rd and 4th bending mode is either not observable or controllable. Therefore the phase angle of this bearing can be rolled off in the range of 300 Hz to 1000 Hz.

A further requirement for this rotor is a minimum stiffness of the bearings. The reason is that the bearing must resist sufficiently the fluid forces in the machine (self-excitation in seals, forced excitation due to flow separation). An experience based rule for this type of compressor is, that the first natural frequency in closed loop condition should not be lower than 35 Hz. The parameter $P_{n1}$ of the base controller was therefore set to 20 kN/mm, which is much higher than necessary due to the magnetic pull of 2.6 kN/mm of the bearings.

### 4.4 Performance of a first controller

With the strategy described in paragraph 4.3 and the type of controller building blocks described in paragraph 4.2 a first controller is designed. The resulting bearing transfer functions as well as the eigenvalues and the sensitivity at maximum speed are shown in figure 11. It can be seen, that the system is stable. The fat lines of the transfer function indicate the frequency ranges, where the bearing provides damping (if there is not a node between sensor and actuator). It can be seen, that at any frequency either of the bearing damps. Also the design strategy explained above can be clearly recognised.

The first bending mode has a frequency of 193 Hz and a damping ratio of 12%, which is much lower than the goal of 20% for modes in the speed range. The maximum value for the sensitivity is below the required value of 3.0 for newly commissioned machines.

An unbalance distribution exciting the 1st bending mode and the response of the bearing forces to this unbalance can be seen in figure 12. The maximum force is almost 1000 N. Considering the weight and the load capacity this would only allow an unbalance of G3 until the bearing would saturate, which is not very robust. A reasonable unbalance limit is G5 considering fouling and thermal bows, which may occur in the field.

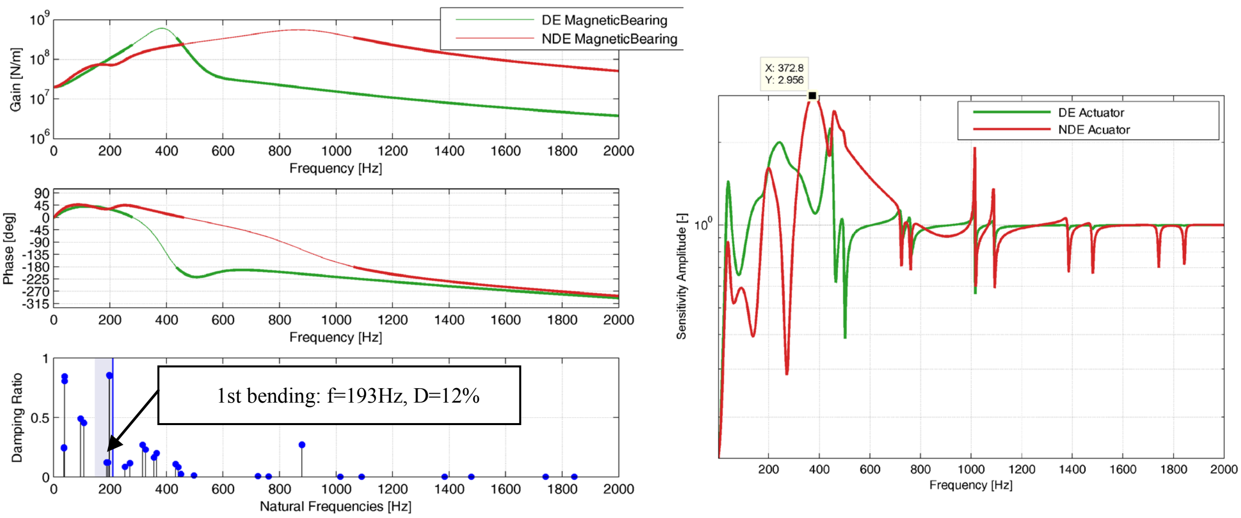



**Figure 11:** Bearing transfer functions, eigenvalues and sensitivity at maximum speed with a first controller

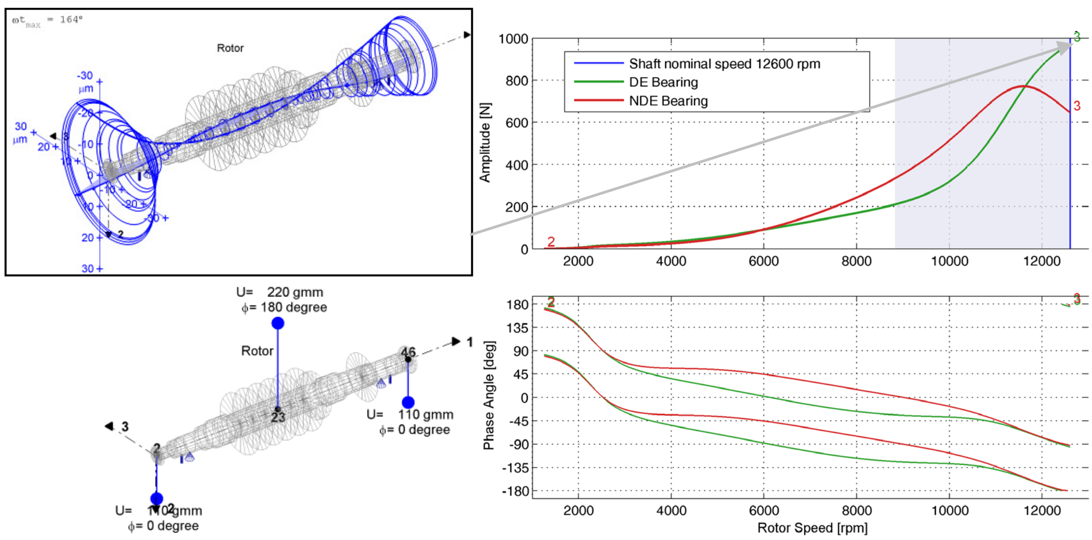



**Figure 12:** Unbalance load case (magnitude G1) and bearing force responses with a first controller

### 4.5 Parameter Optimization

As described above the performance of the first controller is not satisfactory in the following respects: The damping of the 1st bending mode is too low (12% instead of 20%) and the bearing forces to an unbalance response are too high. The maximum force should be reduced to approximately 50% in order to have a satisfactory robustness.

Numerical optimization of the controller parameters is carried out in order to achieve all goals. Totally 20 high-level parameters of both (DE and NDE) controllers are optimized. These are all parameters except the $P_{n1}$ parameter of the base block, which ensures a frequency of the parallel mode of above 35 Hz, and the integrator of the base block. The structures of the two controllers remain unchanged and their transfer function is a polynomial of 7th order (without hardware according to figure 2).

The optimization is carried out by maximising the value of a score function. This function must yield a quality score assessing the deviation from goals, which should be achieved. The following goals covering the general requirements for a robust system can be defined in MADYN 2000:

1.) Damping goal of all natural modes in a wide frequency range. The goal can be a function of the frequency, i.e. the goal damping of natural modes in the operating speed range for example can be higher than for other modes.
2.) Bearing force goal, i.e. the bearing forces should remain below a certain limit.
3.) Gain goal, i.e. the gain should remain below a certain limit. There are no strict criteria regarding the gain. Nevertheless this goal was introduced, since a low gain helps to keep the sensitivity low and because high gain can cause amplifier saturation due to noise.
4.) Sensitivity goal, i.e. the sensitivity should remain below a certain limit.

The score function uses relative values for goal deviations allowing combination of goals with different scaling for different goals. The scaling is also different for the cases, that a goal is excelled or not fulfilled. The weight for excelled goals is set much lower, i.e. it is not the aim of the optimization to exceed goals, for example to achieve higher damping than necessary.

For the optimization of the present system the damping goal, a bearing force goal and gain goal are applied. The bearing force goal is set to 500 N and the gain goal to 500 kN/mm. A factor of ½ is used for the gain goal, i.e. damping and force, are regarded as more important. The sensitivity goal is not applied, since it is usually automatically fulfilled, if the gain is not too high and if the damping goal is fulfilled. This implies that the damping goal function is chosen accordingly, i.e. the damping goal of the modes above the operating speed range has a certain level.

The used damping goal function is shown in figure 13. In the upper operating speed range the goal is set to 17%. The reason is that force and damping goal may be contradictory and it was decided to slightly release the damping goal, since the force goal is more important. In the low frequency range the goal is set to 20% in order to have robustness against self-excitation from fluid forces in the seals. Above the speed range the damping goal linearly decreases from a rather high value of 5% to 0% at 1000 Hz. The function in this range is chosen in order to keep the sensitivity low.

The algorithm used for the optimization is the “interior point” algorithm provided by MATLAB, see [7].

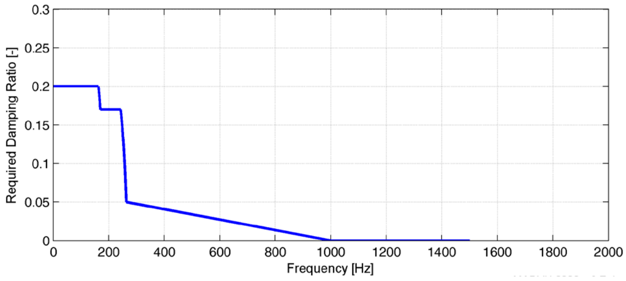



**Figure 13:** Goal function for the damping ratio

## 5. Performance of the optimized controller

The bearing transfer functions of the optimized controllers as well as the eigenvalues and the sensitivity at maximum speed can be seen in figure 14. The eigenvalues are shown together with the damping goal and the transfer functions with gain goal. It can be seen, that the system is stable and the first bending now has a damping above 20%. Not all set goals are fulfilled; eigenvalues with a damping below the goal are red. The gain goal is also not fulfilled in the whole frequency range. However, it is not necessary to completely fulfil these goals, since the damping goal above the speed range and the gain goal were set to receive a low sensitivity. The peak sensitivity is further reduced compared to the first designed controller from 2.96 to 2.45. Both values are below the limit of 3.0. The damping of the parallel mode at 37 Hz is also below the goal of 20%. This goal was set in order to keep the damping high and not to really achieve it. The purpose of a high damping here is the robustness against fluid forces, for which the set stiffness already accounts (→ frequency above 35 Hz).

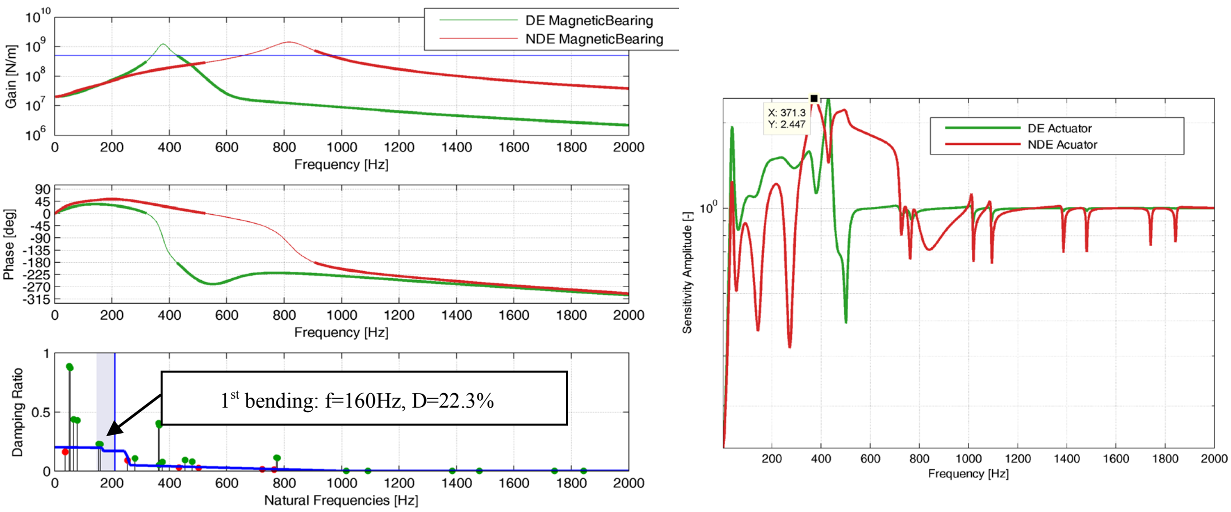



**Figure 14:** Bearing transfer functions, eigenvalues and sensitivity at max. speed with the optimized controller

The bearing forces due to an unbalance exciting the first bending are shown in figure 15. The goal of 500 N is exactly fulfilled and the system now is sufficiently robust against unbalances, which may arise in the field caused for example by fouling or thermal bending.

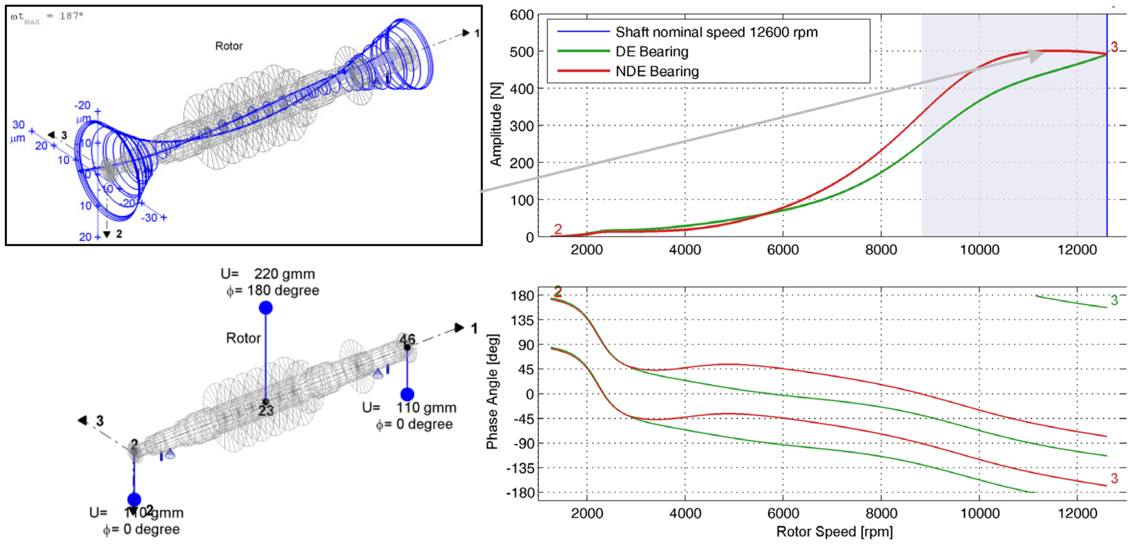



**Figure 15:** Unbalance load case (magnitude G1) and bearing force responses with the optimized controller

## 6 Conclusion

A procedure for practical magnetic bearing controller design is described for a compressor supported on two magnetic bearings. The objectives are stability, robustness, damping of modes in the operating speed range and low bearing force responses to unbalance.

In a first step a controller is designed by combination of proven, stable building blocks. The strategy for the design is mainly determined by the observability and controllability of modes up to a frequency range to 2000Hz, which is far above the speed range of the machine. In the present case this first design yields a stable system with sufficient robustness. However, the damping of the first bending mode in the speed range is too low and the maximum bearing force response is too high.

By means of an automated optimization using a damping goal function, a gain goal and a force goal the damping of the first bending was increased to a value above the required 20% and the maximum bearing force response is reduced to 50%, i.e. the limit for allowable unbalance is doubled. Moreover the robustness is further improved.

This combination of designing the structure of a controller with engineering insight and numerically optimizing the parameters is powerful and saves a lot of engineering effort. Of course the optimization cannot fulfil physically impossible goals. Therefore it is important, that the controller structure is good and that the initial controller makes sense. The better the initial controller, the better will be the possible final result of an optimization. The selection of the goals and the weight factors are also important and require some engineer’s intuition, especially because goals can be contradictory

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