ENGINEERING FOR ROTORS SUPPORTED ON MAGNETIC BEARINGS: THE PROCESS AND THE TOOLS

J. Schmied,[1] F. Betschon[2]

INTRODUCTION

The application of magnetic bearings in rotating machinery is connected with an engineering process, which requires special software tools and which involves several parties (the original equipment manufacturer, the magnetic bearing manufacturer, sometimes the end user of the machine and a consultant accompanying the engineering).

The technical questions comprise classical rotordynamics such as the location of critical speeds, the necessary damping of mechanical natural modes as well as control engineering. Due to their electrical and electronic components magnetic bearings have a different behaviour than conventional mechanical bearings (oil bearings, roller bearings), which has to be taken into account in the design.

Due to their unique properties (no oil required, high circumferential speeds, low losses) magnetic bearings begin to spread more and more into industrial applications and it is time to offer engineering design tools, which allow to handle rotors on magnetic bearings with the same standard as conventional rotors concerning efficiency and predictability. Beside being efficient and reliable these tools must be open and flexible, since we are still in an early phase of industrial applications and the product has not yet reached a completely mature state. That is why the practical experience is still growing fast and the tools must be able to consider this.

In the following, the technical questions are described more detailed and engineering software tools are introduced, which fulfil all the mentioned requirements. They are well proven, efficient, open and flexible concerning the mechanical questions as well as the controller design.

The use of the engineering tools is demonstrated for a specific application: A high speed motor running above the natural frequency of the free rotor's first bending mode at a maximum speed of 120'000rpm. The example does not cover all features of the tools, however it is quite challenging due to the high speed.

THE TECHNICAL QUESTIONS IN MAGNETIC BEARING APPLICATIONS

In all applications with or without magnetic bearings, the allowable rotor vibrations are limited. Reasons for the limits can be small clearances, bearing capacity limits or limited driving torque when crossing critical speeds. In many applications such as pumps or compressors, design specifications exist (e.g. the API) for the rotor damping and the location of critical speeds.

[1]Delta JS AG, Technoparkstrasse 1, 8005 Zurich, tel: ++41/1/445 3130, fax: ++41/1/445 3139, e-mail: deltajs@swissonline.ch.
[2]ETH Zurich.

In addition to this the allowable vibration limits are specified. Some of these rules can also be used as a guideline for magnetic bearing applications. However, they do not cover all aspects.

Basic questions in the case of any rotor design are: Where are the critical speeds? Are critical speeds within the operating speed range? How many critical speeds must be crossed during the run up of the rotor? What is the necessary damping of the natural modes of the rotor in order to ensure safe run up and operation? What is the necessary stiffness of the bearing? How can the necessary damping and stiffness of the bearing be achieved?

These questions have higher significance in magnetic bearing applications, especially because of their limited bearing capacity. Oil bearings and also roller bearings can easily cope with excessive bearing loads at least for a short while. In case of magnetic bearings this will lead to a contact of the rotor with the auxiliary bearings, which are ball bearings or bushings with a low friction surface. Such contacts should not occur too often. That is why loads which can occur during a rotors lifetime have to be considered more carefully.

Fig. 1 shows the capacity limits as a function of the frequency of a bearing load. The specific capacity at low frequencies (below f_{limit}) normally are already lower than in case of conventional bearings. An oil bearing can cope with specific loads of 30bar, whereas magnetic bearings have their limits around 5-10bar, depending on the material. This limit is reduced even further at higher frequencies (above f_{limit}) due to the limited voltage of the power amplifiers (see e.g. /1/).

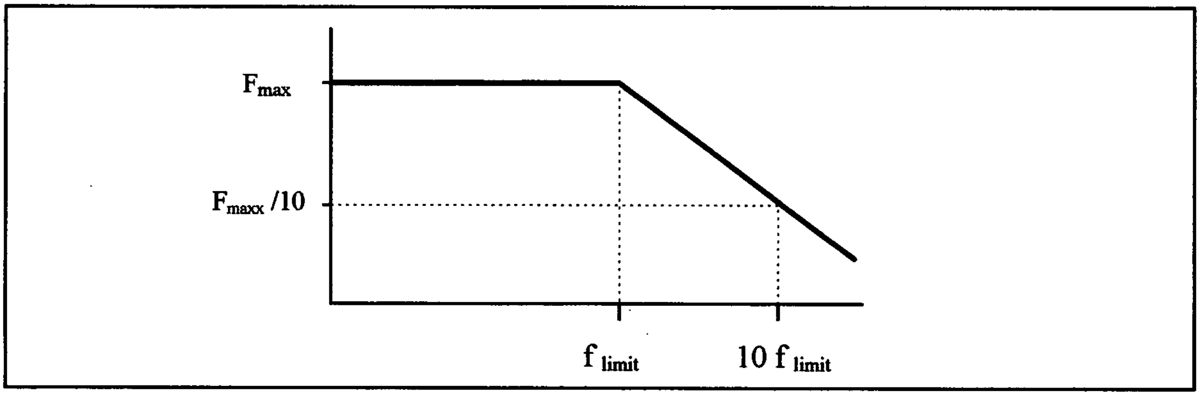

Figure 1. Force Limits of a Magnetic Bearing as a Function of Frequency

An optimal damping is mandatory for any robust rotor design. In most applications the damping has to be provided by the bearing. In order to enable a magnetic bearing to do this, all natural modes, which must be well damped (normally all modes below the maximum operating speed), must also be observable and controllable. Sufficient damping force must be provided by an appropriate controller design.

Fig. 2 shows a typical magnetic bearing transfer function consisting of the sensor, the amplifier and a controller with a PID part and a second order filter. In order to provide damping the phase angle ϕ of the transfer function must be either in the region

$$0^\circ < \phi < 90^\circ,$$

or

$$-180^\circ > \phi > -270^\circ \text{ (corresponds to } 90^\circ < \phi < 180^\circ)$$

In the latter case the bearing has a negative stiffness force. This does not lead to an instability at high frequencies.

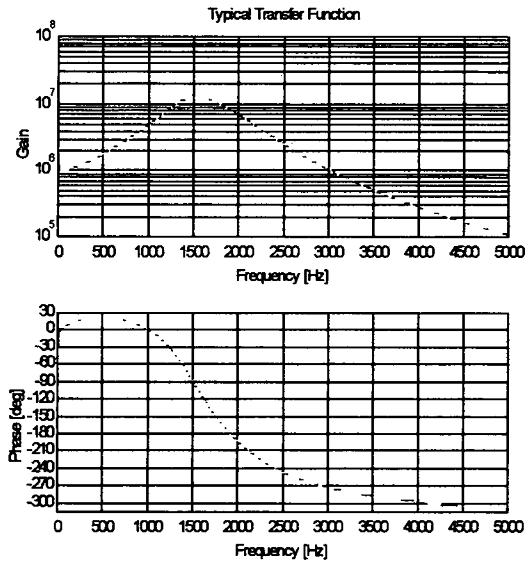

Figure 2. Typical Magnetic Bearing Transfer Function

The shown transfer function provides damping forces up to frequencies of 1000Hz. The running speed for this transfer function would be preferably below 1000Hz, in order to damp all natural modes during run up and run down. For frequencies between 1900Hz and 2900Hz the transfer function also provides a damping force. This is to ensure the stability of higher natural modes. The amplitude level in this frequency range is limited by a filter, which is advantageous because of electrical noise, which should not load the bearing unnecessarily. The damping is achieved with a phase below -180°, which is the only way, since the amplifier and sensor have a phase loss with increasing frequency. This phase loss is increased, in cases where a digital controller is used.

Besides sufficient damping, a sufficient stiffness force can also be important for some applications, e.g. pumps or compressors must resist high fluid forces, which normally have very low frequencies.

The properties of a rotor may change considerably with speed due to gyroscopic effects or fluid forces. Some rotors may have different properties when operating in different modes. A robust controller design must take this into account.

THE ENGINEERING SOFTWARE TOOLS CONCERNING MAGNETIC BEARING APPLICATIONS

Software tools suited for the engineering of magnetic bearing applications must comprise a structural part describing the rotor as well as a mechatronic part describing the magnetic bearing and the combined rotor bearing system. For the structural part the comprehensive machine dynamics programme MADYN /2/ is used. For the mechatronic part tools were developed using the mathematical software package MATLAB /3/. The latter are components of the package MEDYN (mechatronic system dynamics). The tasks of MADYN and MEDYN as well as the communication of the two programmes are summarised in fig.3.

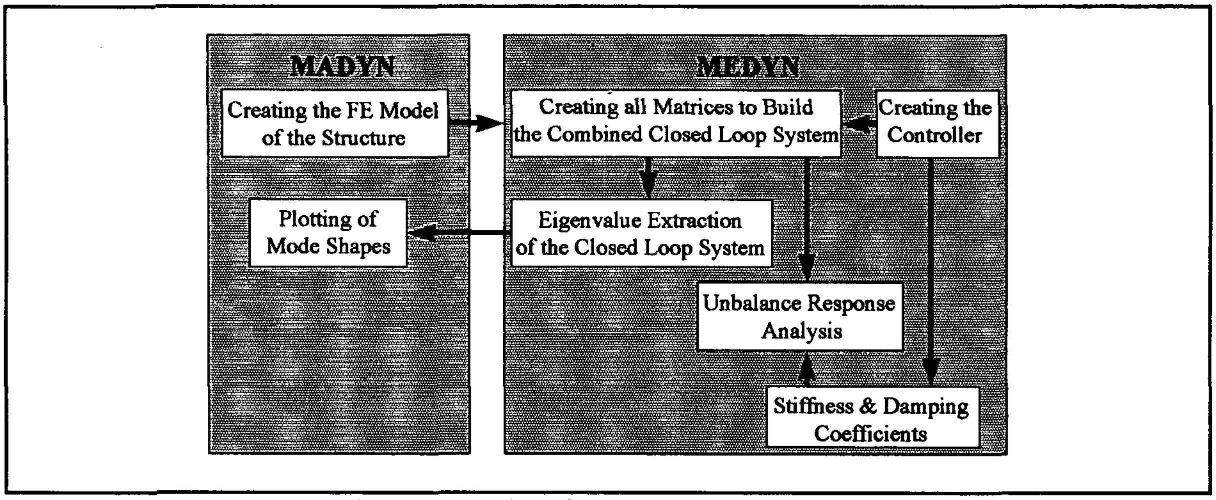

Figure 3. Tasks of MADYN and MEDYN

In the structural part, the following effects can be considered:

- Gyroscopic effects,
- rotor fluid interaction, which is important for pump and compressor applications,
- the flexibility of stator parts,
- the dynamics of blades, which plays a role in some applications such as turbomolecular pumps.

The mechatronic part allows the analysis of

- the non collocation of sensors and actuators,
- the negative stiffness of magnetic actuators,
- analogue as well as digital controllers,
- a time delay for digital controllers,
- the characteristics of the hardware components of magnetic bearings, such as sensors and amplifiers,
- separate sensors as well as separate controllers for displacement and velocity,
- the coupling of bearings by means of the controllers, such as separate controllers for the tilting and translation modes of rotors, which can be useful in case of symmetric or almost symmetric rotors.

The mechatronic part also has a design tool for the controller, which allows the user to combine standard controller components. At present the following components are available in analogue as well as digital form. In the latter case the analogue functions are transformed with a first order hold transformation.

- A base component, which is a modified PID controller with bandwidth limited phase lead cells,
- a first order filter,
- a second order filter with a parallel proportional part,
- a second order all pass filter,
- notch filters,
- analogue butterworth filters in case of digital controllers for anti aliasing.

New components can easily be included in the future if necessary.

A typical procedure for the engineering is as follows: The structural part is modelled in a similar way as in case of conventional bearings. The basic behaviour of the rotor can then be studied. This includes the analyses of natural frequencies of the rotor at standstill and at speed for different bearing stiffness coefficients, as well as the analyses of damping ratios of natural modes for different bearing damping coefficients. Knowing the basic behaviour, i.e. the variation of natural frequencies with speed and the ideal bearing stiffness and damping coefficients a controller can be designed. The actual behaviour of the combined system (closed loop behaviour) can then be studied with the mechatronic part. This procedure is demonstrated in detail for an example in the next section.

AN EXAMPLE: A HIGH SPEED SYCHRONOUS MOTOR

The rotor of the high speed synchronous drive with its stator parts is shown in fig.4. Its length is 181mm and it weighs 680g. Its maximum speed is 120'000rpm. The middle part of the rotor is the motor part with permanent magnets held by carbon fibre.

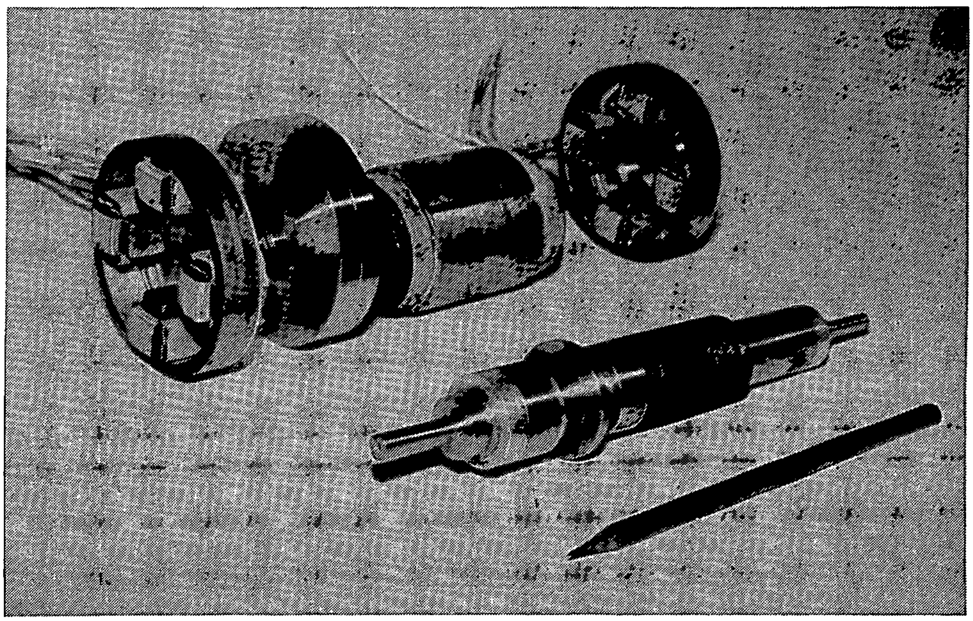

Figure 4. Rotor and Stator Parts of the High Speed Synchronous Drive

THE BASIC ROTORDYNAMIC BEHAVIOUR

The rotor model is shown in fig.5.

The first three bending modes of the free rotor at standstill can be seen in fig.6. The long vertical lines indicate the actuator locations and the shorter vertical lines the sensor locations.

The natural frequencies and the range of the frequencies as the rotor starts rotating up to its maximum speed are listed in table 4.1.

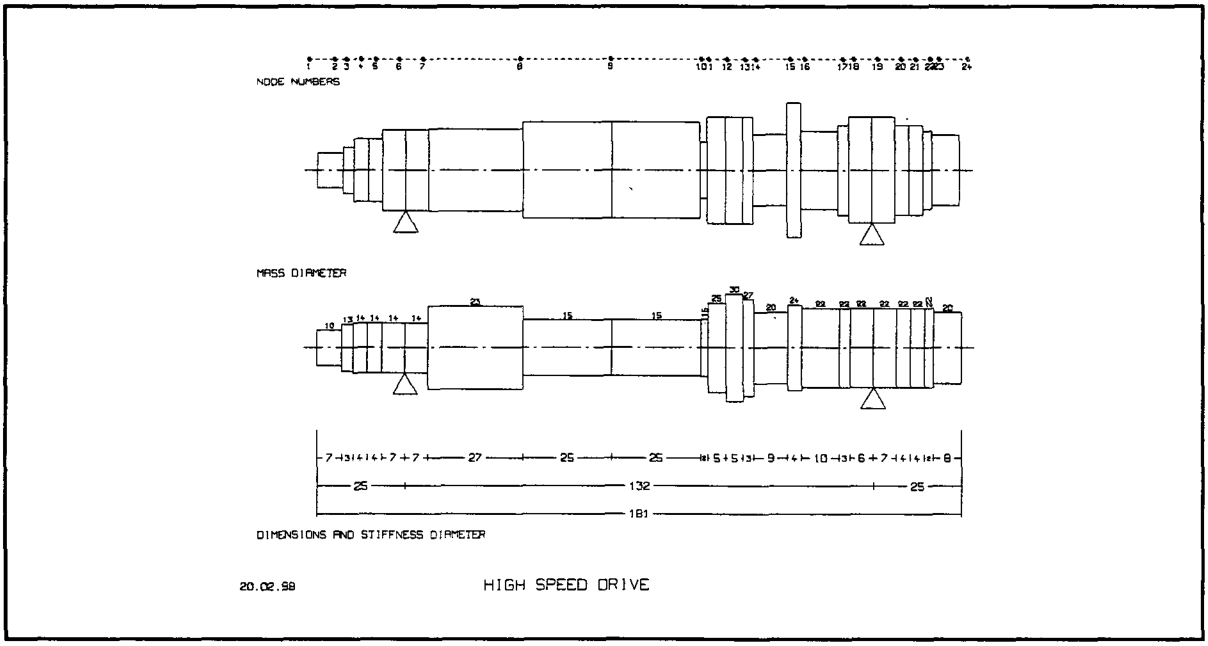

Figure 5. Rotor Model

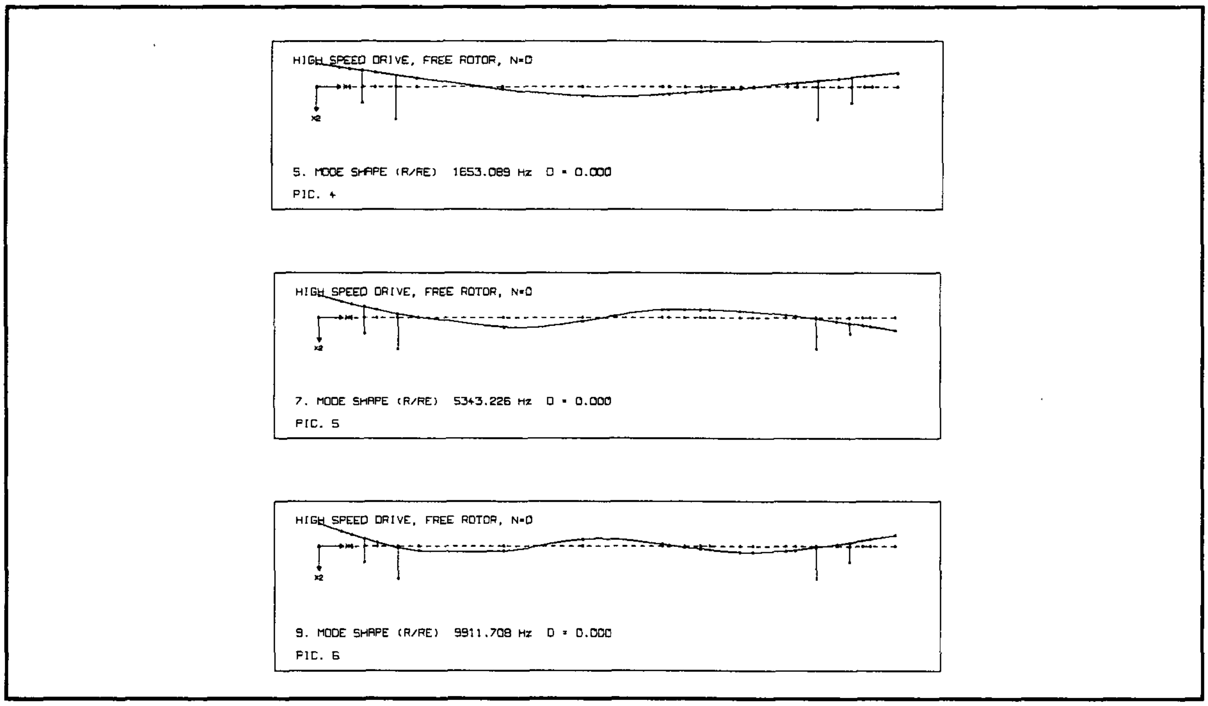

Figure 6. Natural Bending Modes of the Free Rotor at Standstill

Table 4.1. Natural Frequencies of the Free Rotor at Standstill and at Max. Speed (120'000rpm)

Mode	Standstill	max. speed, backward whirling	max. speed forward whirling
Tilting	0 Hz	0 Hz	81.3 Hz
1st Bending	1653 Hz	1538 Hz	1776 Hz
2nd Bending	5343 Hz	5134 Hz	5559 Hz
3rd Bending	9911 Hz	9594 Hz	10234 Hz

The natural frequencies of the first three modes at standstill as a function of the bearing stiffness are shown in fig. 7. In the analysis a spring has been applied at the actuator location.

The magnetic bearings each have a negative stiffness of about 53'000 N/m. The motor also has a negative stiffness in the same order of magnitude. Hence the stiffness of the bearings created by the controller should be at least around 2-3 10^5N/m in order to safely compensate these negative stiffnesses

The natural frequency and the damping ratio of the first three modes at standstill as a function of the bearing damping coefficient for a stiffness of 10^6N/m can be seen in fig. 8. Considerable damping ratios for the first bending mode are achievable. A coefficient of 250Ns/m yields 30%.

Fig. 7 and 8 are very helpful for the controller design, as will be shown in the next section.

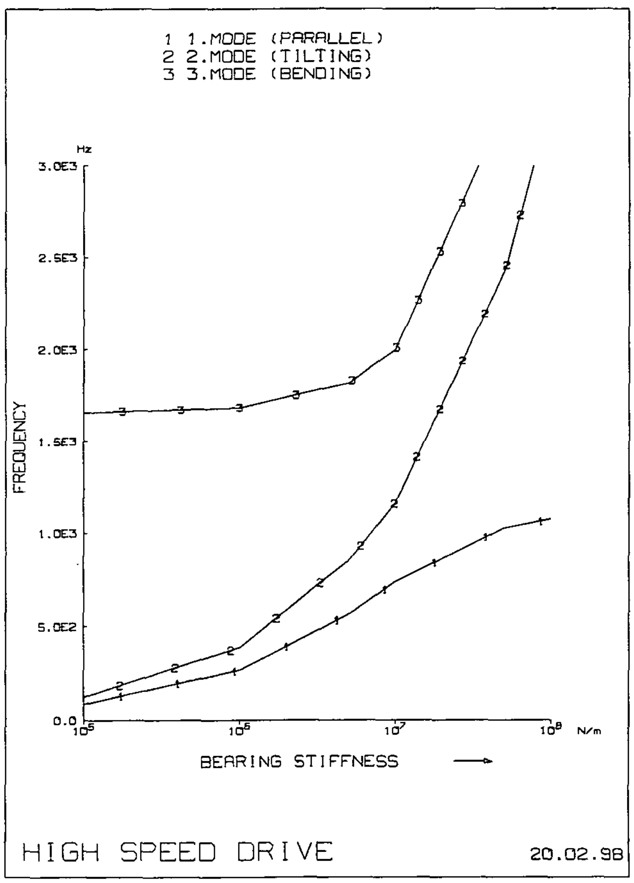

Figure 7. Natural Frequencies of the first three Modes vs. Bearing Stiffness

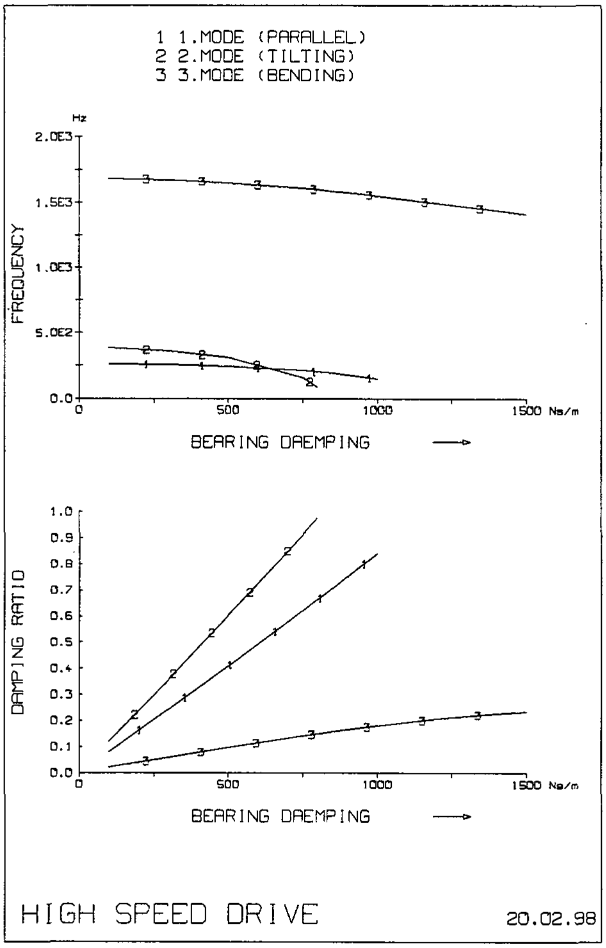

Figure 8. Frequency and Damping Ratio of the first three Modes vs. Bearing Damping ($k = 10^6$N/m)

THE DESIGN OF THE CONTROLLER

The bearings have to provide the damping force to be able to cross the first bending mode of the rotor at around 1650 Hz. It has to be created in spite of a phase loss of the amplifier (i.e. the current controller) of more than 45° at this frequency and a time delay of 71μs due to the analogue to digital transformation as well as the calculation in the signal processor (i.e. time between the input of the position signal and the output of the current) causing 40° phase loss. The zero order hold behaviour of the digital controller with a sampling rate of 10'400Hz adds another 30°.

Due to these phase losses it was not possible to achieve sufficient damping force at 1650Hz with a normal PID controller. Enough damping could only be achieved by means of a second order filter, which lowered the phase sufficiently below 180°. At the same time a phase lead at lower frequencies had to be maintained in order to damp the rigid body modes.

The transfer function of the bearings is shown in fig.9. Both bearings have the same function. The amplifiers are approximated with a second order filter. The detailed functions of all components in the s domain are listed in the appendix. The following forms of the transfer function are shown in fig. 9:

1. Solid: The digital form of the original analogue controller transformed by a first order hold (foh) transformation. This transformation yields a good approximation of the analogue transfer function. This is the way the controller is programmed in the signal processor and the way it is considered in the closed loop analyses (see next section).
2. Marked "o": The analogue form of the transfer function including the antialiasing filter of first order at 5200Hz. It can be seen, that the filter almost has no influence in the range up to 2000Hz.
3. Marked "x": The analogue form including a time delay of 96μs. Although a time delay of only 71μs could have been achieved, it was decided to output the controller current signal at the end of one cycle of the sampling rate. This measure increased the damping force for the bending mode, since it lowered the phase even more below 180°.
4. Marked "+": The digital form of the transfer function transformed by a zero order hold (zoh) transformation.

The transfer functions 2 to 4 are only plotted for design purposes of the controller, in order to see the influence of the considered effects. They are not used to create the closed loop model for the next section.

The actual closed loop model is built by transforming the rotor and the analogue parts of the bearing with a zero order hold transformation and combining them with the digital controller transfer functions transformed by first order hold (solid curve).

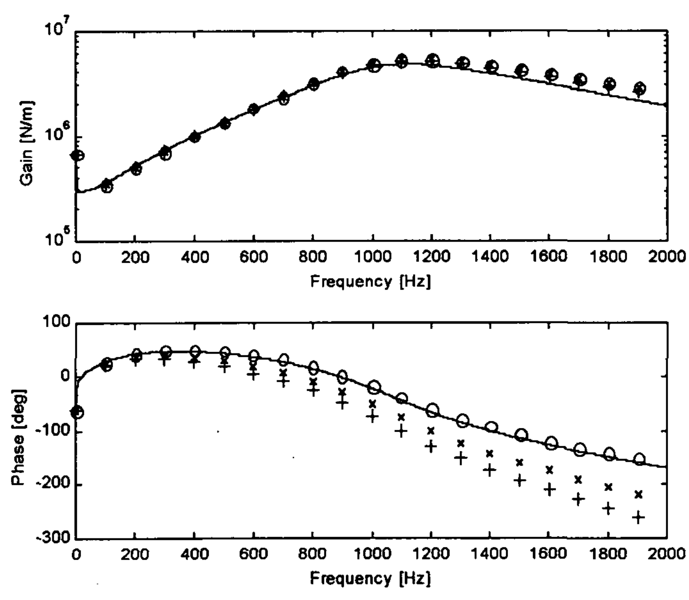

Figure 9. Bearing Transfer Function ("-" foh, "o" incl. filter, "x" incl. filter & delay, "+" zoh)

THE CLOSED LOOP BEHAVIOUR

The frequencies and damping ratios of all frequencies below 5000Hz are listed in table 4.2. Due to aliasing, rotor modes, which are above 5000Hz will appear in the analysis below 5000Hz. These modes are excluded in the table as well as some high damped modes which arise due to the additional controller co-ordinates. In the analyses a structural damping is considered (proportional to the stiffness matrix), which produces a damping ratio of 1.5‰ at 1500Hz. Damping ratios exceeding these values are caused by the bearing.

Table 4.2. Eigenvalues (Damping Ratio and Frequency) of all Modes below 5000Hz at Standstill and full Speed (=2000Hz)

Mode No.	Frequency n=0	Damping Ratio n=0	Frequency n=2000cps	Damping Ratio n=2000cps
1: parallel	144 Hz	40.2%	- 142.6 Hz + 144.6 Hz	40.5 % 40.3 %
2: tilting	547 Hz	16.7 %	- 111.5 Hz + 562.9 Hz	84.8 % 3.6 %
3: 1st bending	1492 Hz	8.6 %	- 1378.2 Hz + 1498.5 Hz	2.4 % 23.4 %
4: 1st bending	1730 Hz	21.0 %	+ 1807.2 Hz	8.7 %
5: 2nd bending	-	-	- 4840.7 Hz	0.6 %

+ forward whirling, - backward whirling

At standstill all modes are well damped. At maximum speed the damping of the forward whirling tilting mode as well as the backward whirling bending mode decrease. However, the backward mode can not be excited by unbalance and the tilting mode only looses damping, when the speed is much higher than its frequency.

The bending mode appears two times. This is due to the frequency dependence of the bearing characteristic.

Fig. 10 shows the response of the left bearing force to an unbalance of G1 (according to ISO 1940) in the middle of the rotor. The maximum bearing force when crossing the bending mode is about 6N. The maximum dynamic bearing capacity is 30N.

The behaviour at standstill could well be confirmed by measured transfer functions with excitation from the bearings. At the time of publication a speed of 1600cps has been reached, which is very close to the bending critical speed. Higher speeds were not yet possible due to problems in the drive.

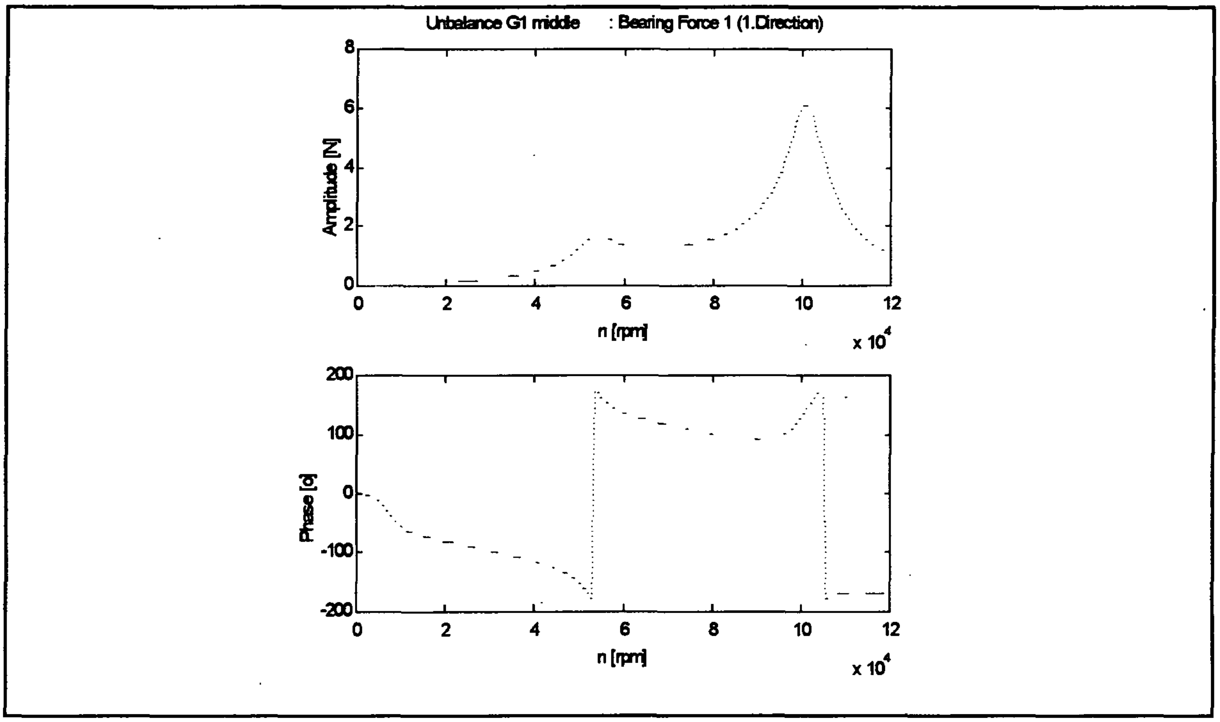

Figure 10. Bearing Force of the Left Bearing due to Unbalance G1 in the Rotor Middle

SUMMARY

The unique properties of magnetic bearings (no lubrication required, high circumferential speeds, low losses..) lead to their wide industrial application. For this reason a demand for engineering tools arises, which allow to treat rotors with magnetic bearings with the same standard as conventional rotors concerning efficiency and predictability. At the same time these tools must be flexible and adaptable to the experience, which rapidly grows with the rising number of applications.

In this paper tools are introduced, which are based on the comprehensive rotordynamic programme MADYN and on a specially developed MATLAB toolbox MEDYN. MADYN covers the structural part, i.e. the modelling of the rotor and stator as well as the analysis of the basic rotordynamic behaviour such as the influence of the rotational speed, of the bearing damping and stiffness coefficients on the natural modes of the rotor. MEDYN covers the electronic and controller part, i.e. their modelling, as well as the analyses of the combined system, such as its natural modes and its unbalance response behaviour.

The use of the tools is demonstrated in a very demanding example: A high speed synchronous motor running above its first bending mode at a speed of 120'000rpm. The bending mode with a frequency of 1'650Hz was damped by means of a digital controller with a sampling rate of 10'400Hz, i.e. the controller had to cope with considerable phase losses.

REFERENCES

/1/ G. Schweitzer, A. Traxler, H. Bleuler: Magnetlager, Grundlagen, Eigenschaften und Anwendungen berührungsfreier, elektromagnetischer Lager. Springer Verlag 1992.
/2/ H.D. Klement: Berechnungen der Eigenfrequenzen und Stabilität von Rotoren mit MADYN. VDI Berichte Nr. 1082, 1993.
/3/ The Math Works Inc.: MATLAB The Language of Technical Computing. User Description, 1996.

ACKNOWLEDGEMENTS

The authors wish to thank Sulzer Electronics, who are a partner in this project, for the excellent cooperation.

APPENDIX

For the transfer functions the following components are used:

$$F_{base} = \frac{P_{n1} + \frac{(P_{n1}+P)s}{2\pi f_{n1}}\left(1+\frac{s}{2\pi f_{n2}}\right)}{\left(1+\frac{s}{2\pi f_{d1}}\right)\left(1+\frac{s}{2\pi f_{d2}}\right)} + \frac{(P_{n1}+P)2\pi f_{in}}{s} + P$$

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

The amplifier characteristic for both bearings is simulated with F_{filt2} with the following parameter:

P1 = 1, ff = 2200 Hz, D = 0.5.

The controllers for both bearings consist of F_{base} and F_{notch}.
Fbase has the following parameter:

f_{n1} = 150 Hz, f_{n2} = 1000 Hz, $f_{d1} = f_{d2}$ = 3000 Hz , f_{in} = 2 Hz, Pn1 = 0, P = 300'000 N/m .

The parameter of F_{filt2} for the controller are:

P1 = 1, ff = 1100 Hz, D = 0.3.