Rotordynamic Design Considerations for a 23 MW Compressor with Magnetic Bearings

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This paper describes a very demanding magnetic bearing application. A multi-stage centrifugal compressor, with a maximum discharge pressure of 130 barg, driven by a variable speed synchronous electric motor, with an operating speed range from 600 rpm to 6300 rpm. The requirements for the rotordynamic performance of the compressor are defined. Special engineering tools, covering the mechanical aspects as well as the electronic controller for the magnetic bearing system, are needed to comply with the requirements. The results of the calculations and an experimental verification are given.

1 INTRODUCTION

The industry, through it's increasing demand for advanced, versatile but robust rotating machinery, requires the application of modern, environmentally friendly technologies. Active magnetic bearings are gaining more and more acceptance and are becoming a serious feature for the industry to consider. A very prominent example is a compression unit that features a centrifugal compressor, driven by a 23 MW variable speed synchronous electric motor, both using active magnetic bearings. Since dry gas seals are used in the compressor, it follows that this unit is completely dry, therefore no lubrication or seal oil is needed. The paper covers the rotor dynamic aspects of the compressor and is focused on the analytical methods applied during the design phase. An experimental verification of the rotor response measured during extensive shop testing is included.

2 DESCRIPTION OF THE NATURAL GAS PLANT AND THE COMPRESSOR

The Groningen natural gas field, which is located in the northern part of The Netherlands, is operated by NAM B.V., a shareholder company of Shell Nederland B.V. and Esso Holding Company Holland Inc. The field has 29 production installations (clusters) spread over the province of Groningen (see Figure 1).

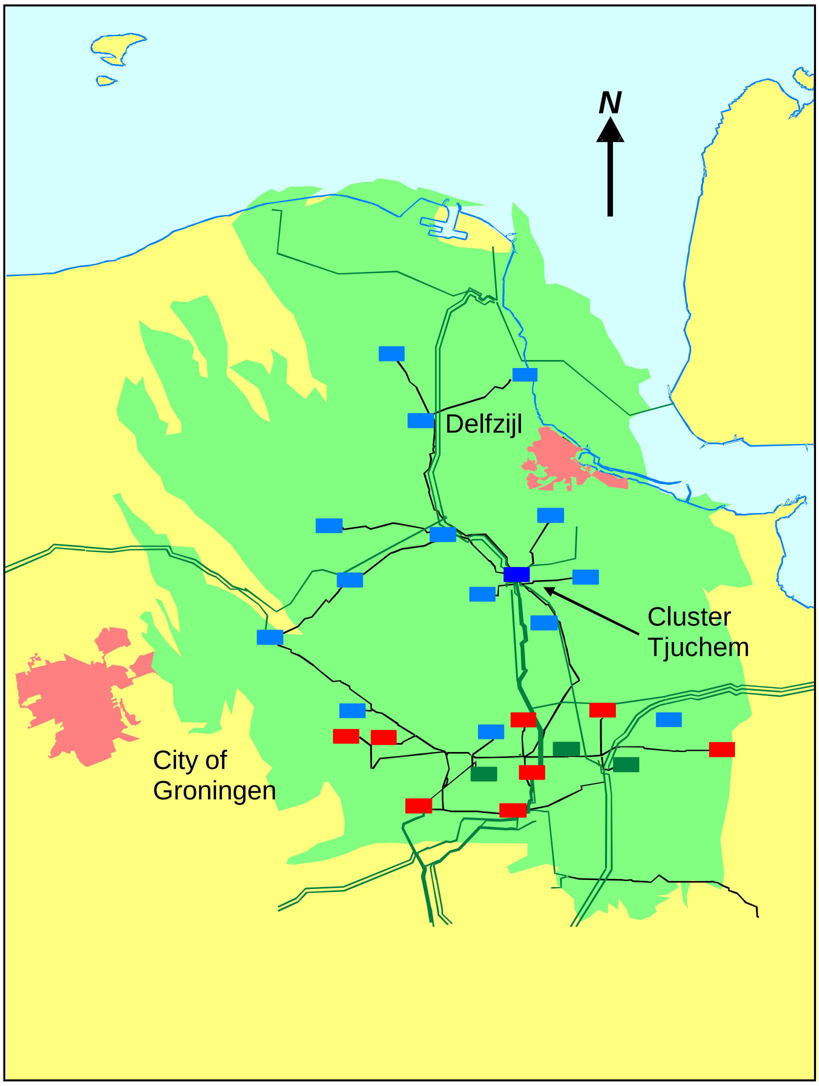


Fig. 1 Natural gas Field of Groningen

The Groningen Field is used as a peak flow supplier with continuous production swings, mainly due to national daily and seasonal demands. (see Figure 2).

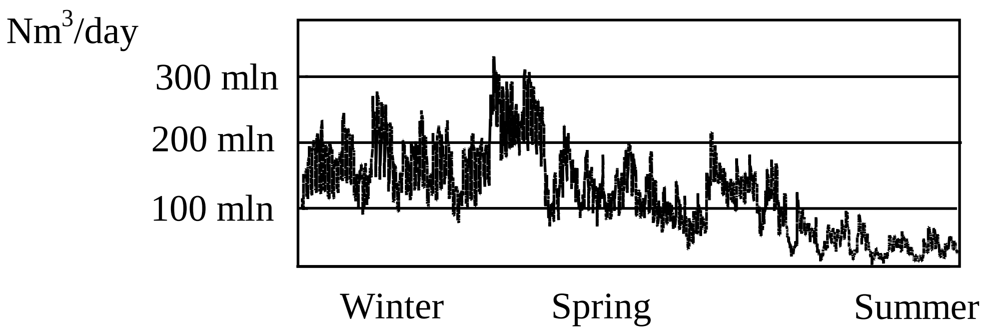


Fig. 2   Gasproduction over 6 Months in Groningen

Contractual commitments to deliver sufficient capacity at a supply pressure of 65 barg to the grid, require that compression units be installed in the near future. Feasibility studies concluded that compression upstream of the gas conditioning plant on all clusters would be the most economical solution. See Figure 3 for a schematic overview of the gas treatment plant.

The project is a good application for active magnetic bearings, since they facilitate extension of the operating speed range of the unit all the way down to 10%, and also allow for remote monitoring. The operating principle of magnetic bearings has been explained in various technical papers. The present limitations are also documented [1].

A cross section of the compressor is shown in Figure 4. The initial compressor is a 5-stage configuration. In the future the compressor will be revamped to an 8-stage configuration.

Tandem dry gas seals are installed at the shaft ends. Active magnetic journal bearings are applied in combination with self lubricated, auxiliary bearings. These auxiliary bearings are located at carefully chosen locations on the rotor, to prevent damage to rotor and stator parts during emergency rundown situations.

An active magnetic, double acting axial bearing is installed at the Non-Drive-End (N.D.E.) of the compressor. The power is supplied through a dry diaphragm type coupling.

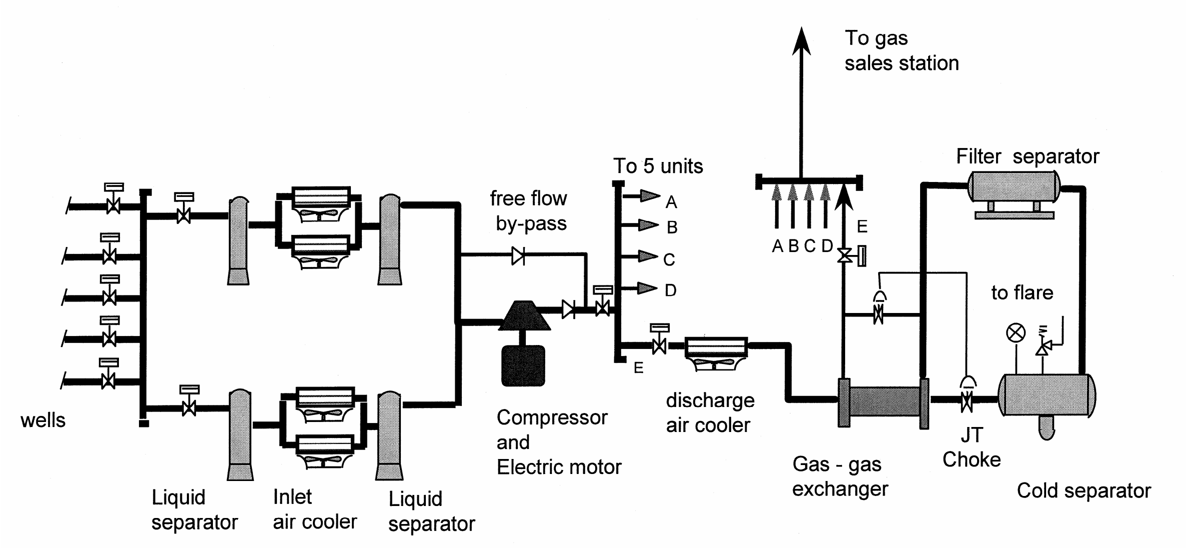


Fig. 3 Process Flow Scheme of Gas Treatment Plant, with Compression Unit

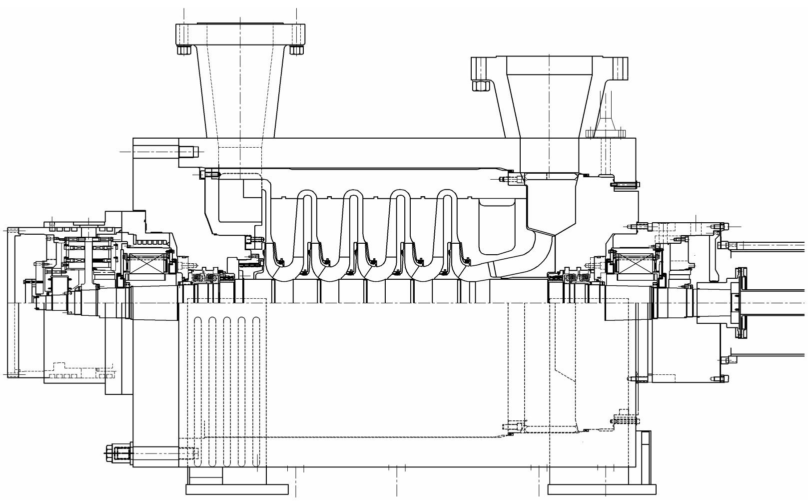


Fig. 4 Cross Section of the Compressor (Initial Configuration)

3 REQUIREMENTS FOR THE ROTORDYNAMIC PERFORMANCE.

The challenge for the design engineer for such an application is impressive and unique, since there is a lack of design consensus represented in the accepted industrial guidelines and standards. API 617 [2] deals only with fluid film bearings and contains no specific guidelines and acceptance criteria as reference data for both customer and supplier. Appendix J of this standard does contain generalised application considerations for active magnetic bearings. The application of active magnetic bearings mandates that the design engineer expands his horizon into the area of Mechatronics. The rotordynamics of a compressor rotor, with traditional fluid film bearings, is limited to the tuning of the interaction between the rotor and the oil film in the bearings. Active magnetic bearings however require that the rotor, the bearings and also the bearing control system are properly accounted for, i.e. rotor, sensors, controller, amplifiers and electromagnets.

Delaval Stork has built up significant experience in the application of dry-dry technology (dry gas seals and active magnetic bearings) over the years, on both beam type and overhung type centrifugal compressors [3, 4]. Based upon this in-house expertise the following considerations for dry-dry rotordynamic performance have been developed:

1. The allowable vibration level at each radial magnetic bearing shall not exceed twice the API 617 para 2.9.5.5 limits throughout the operating speed range. Active magnetic bearings have an inherent lower stiffness compared to that of traditional fluid film bearings. This means that vibration levels for a given unbalance will be higher at the bearing location. Assuming these higher vibration levels, the design engineer can better optimise the control loop algorithm which in this case is quite essential given the extremely large operating speed range. Maintaining the vibration limits specified in API 617 would mandate high stiffness from the active magnetic bearing control system with inherent high dynamic currents to the electromagnets. This introduces the risks of a sensitive control loop and saturation of the power amplifiers even during small displacements.
2. The amplification factor for all modes within the operating speed range shall preferably be less than 2.5. This means that on a case to case basis, prudent judgement should be given to this design criterion. Specifying an amplification factor of 2.5 (equals a damping ratio of 20%) as maximum, introduces constraints for the design engineer in the optimisation of the control loop.
3. The bearing control system shall provide adequate damping for all modes to allow stable operation for all the conditions specified. This mandates not only positive damping over a large frequency range but also exact knowledge of all excitation mechanisms, with labyrinth induced cross coupling being the most dominant [5]. Unlike fluid film bearings, active magnetic bearings are not as forgiving in overload situations.
4. All modes which can be excited during operation shall be clearly observable and controllable, i.e. the rotor shall yield sufficient amplitude at the various bearing sensor locations.
5. The bandwidth of the control loop should be as small as possible, i.e. a low frequency roll-off is preferred. This makes the bearing control system less sensitive for natural frequencies above the roll-off frequency. In this way, all rotors, each of which may exhibit slight differences in natural frequencies, due to manufacturing tolerances, are not limited to use in one machine. The rotor bundles are not assigned to a specific field cluster, but

may travel from one cluster to another, as part of the maintenance concept adopted for the project. This approach eliminates the need for on-site tuning.

4 TOOLS FOR THE ROTORDYNAMIC ANALYSES

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, the controller and amplifiers, and the combined rotor bearing system. The following combination of programmes was used for the rotordynamic analyses. For the structural part the comprehensive machine dynamics programme MADYN was used, and for the mechatronic part MEDYN (mechatronic system dynamics) was used [6]. The latter are functions, which were developed using the mathematical software package MATLAB. The tasks of MADYN and MEDYN as well as the communication between the two programmes are summarised in Figure 5.

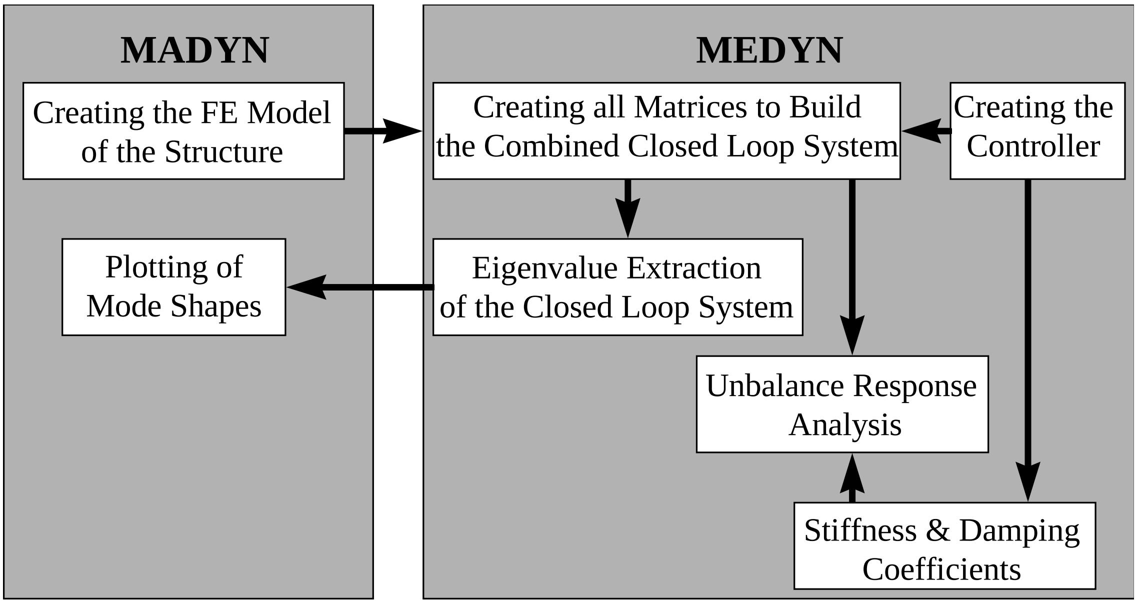


Fig. 5 Tasks of MADYN and MEDYN

In the structural part, the following effects are considered:

- Gyroscopic effects
- Rotor fluid interaction, such as labyrinth induced excitations
- Dynamics of flexible parts mounted on the rotor, such as disks

In addition to this the tools may also be used to account for the flexibility of stator parts, although this was unnecessary in this project, since all stator parts were sufficiently rigid.

The mechatronic part includes the modelling of:

- Non-collocation of sensors and actuators
- Negative stiffness of magnetic actuators
- Digital controllers
- Time delay for the digital controllers

- Characteristics of the hardware components of magnetic bearings, such as sensors and amplifiers
- Separate sensors as well as separate controllers for displacement and velocity
- Coupling of bearings by means of the controllers, such as separate controllers for the tilting and translation modes of rotors, which may be useful in the case of symmetric or almost symmetric rotors.

The mechatronic part also has a design tool for the controller, which allows combining standard controller components. The following components are available in analog as well as digital form:

- Base component, which is a modified PID controller with bandwidth limited phase lead cells
- First order filter
- Second order filter with a parallel proportional part
- Second order all pass filter
- General second order filter
- Notch filters
- Analogue Butterworth filters in case of digital controllers for anti-aliasing

5 BASIC ROTORDYNAMIC BEHAVIOUR

The rotor model used for all rotordynamic analyses is shown in Figure 6. The basic data of the rotor are as follows:

- Weight 2100 kg
- Total length 3845 mm
- Bearing span (between actuators) 2674 mm
- Maximum continuous speed 6300 rpm

The triangles in this figure indicate the positions of the bearing actuators. Each bearing has two sensors at each side of the actuator. Finally, the outboard sensors were used due to reasons that will be explained in section 6.

The disk at node 8 is the axial bearing disk. Its one nodal diameter vibration mode had to be considered in the rotor model, since its frequency of 490 Hz is in the range which must be considered for the magnetic bearing tuning (frequencies up to approx. 1000 Hz, also see section 6). To model this mode the disk is fixed to the rotor with a rotational spring. The stiffness of the spring is adjusted to yield the frequency of 490 Hz.

The investigation of the basic rotordynamic behaviour comprises the analysis of the natural modes for different bearing stiffnesses at different speeds. Their frequencies are shown in Figure 7. This diagram shows which modes are in the operating speed range, their separation margin from this speed range, and gives guidelines for the controller design, i.e. to show at which frequencies the controller must provide damping. Negative bearing stiffnesses are included, since the magnetic bearing can have such stiffness values at certain frequencies. In this application the first bending mode is in operating speed range, and requires special attention.

Figure 8 shows the first four bending mode shapes and the disk mode at zero speed. The long vertical lines indicate the actuator position and the short lines the sensor positions.

It can be seen that the disk mode and the rotor modes interact. In the second and higher bending modes the flexible disk tilts more than a rigid disk would. This has the effect of considerably increasing the gyroscopic effect. As a consequence the frequency difference between forward and backward whirl increases as well as the frequency regions where the bearing must provide damping.

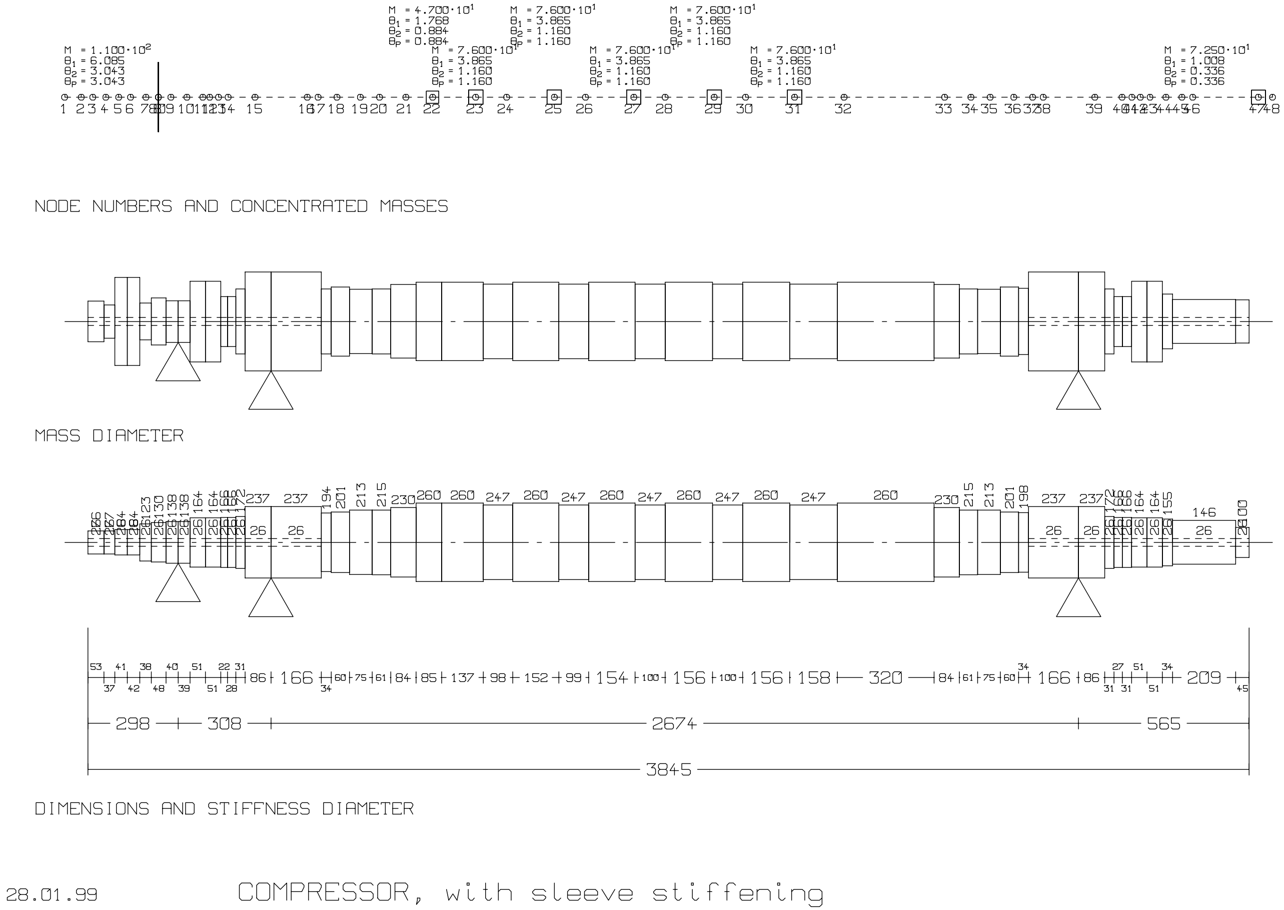


Fig.6 Rotor Model

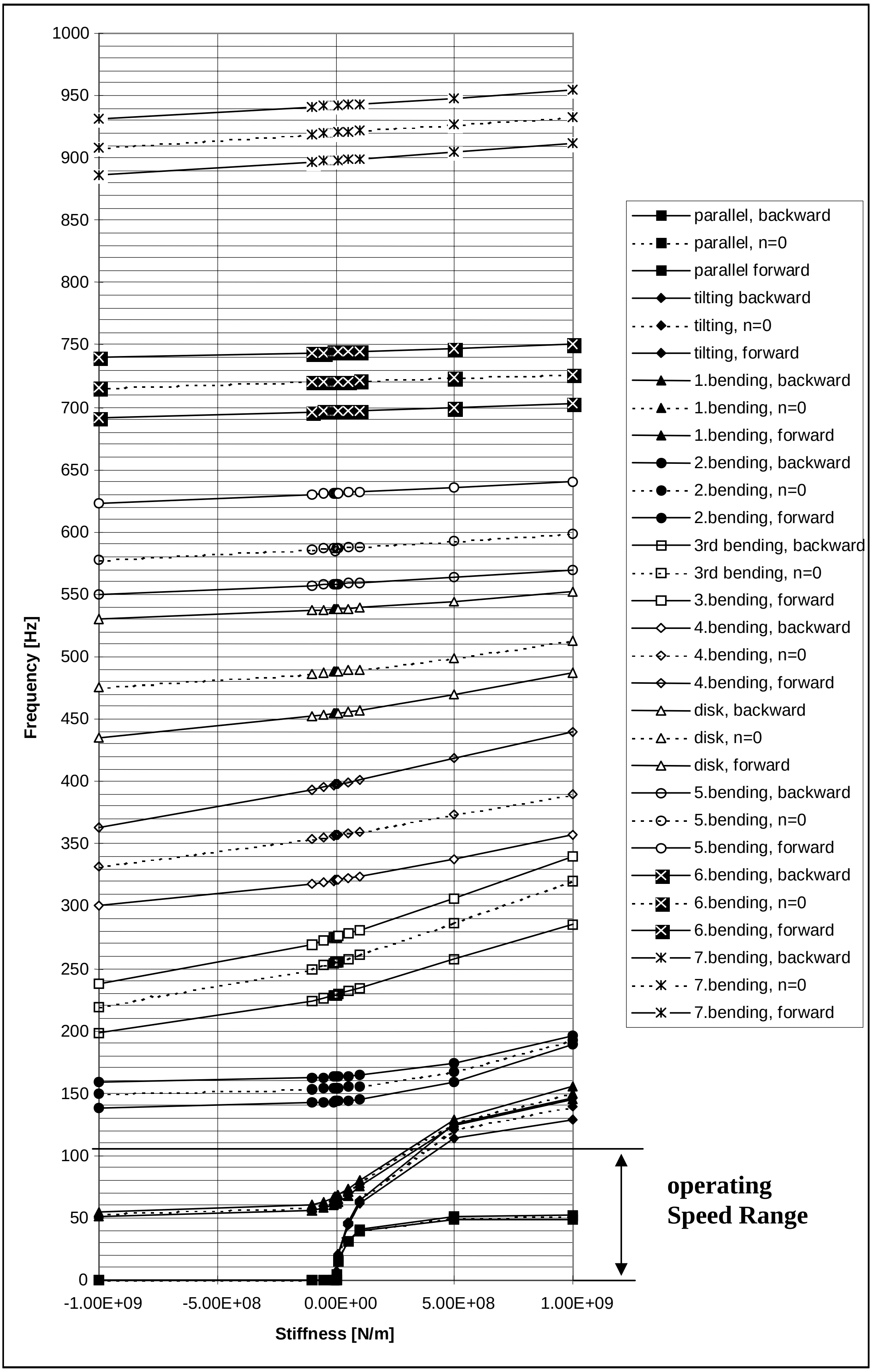
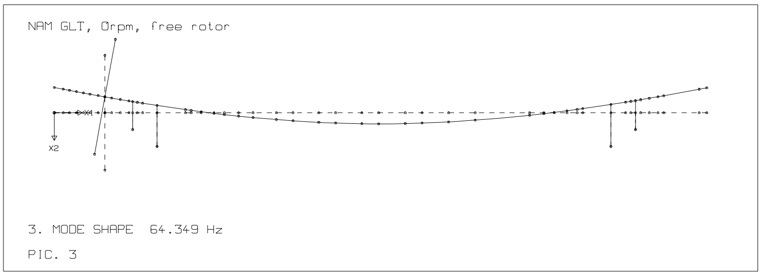
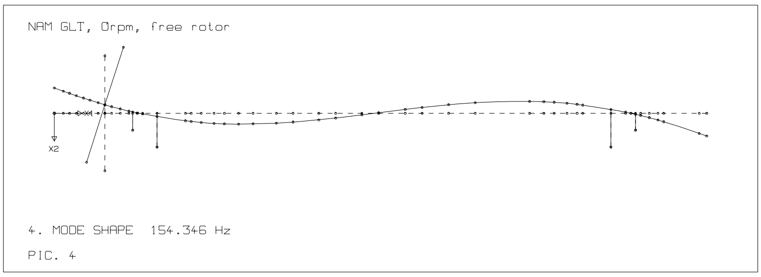
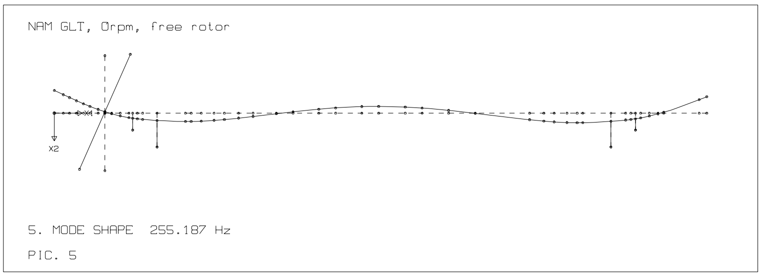
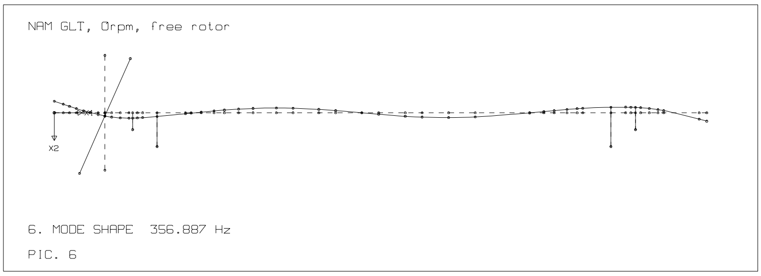


Fig.7 Natural Frequencies as a Function of Bearing Stiffness and Speed









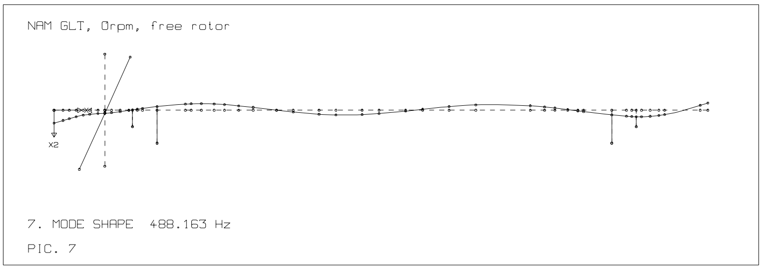


Fig.8: Natural Modes of the Free Rotor at Standstill

6 THE CONTROLLER DESIGN

The aim of the controller design is to provide good damping for all modes in the speed range of the machine and sufficient damping for higher modes in order to keep them stable. The bearing stiffness must also be high enough for the rotor to resist fluid forces, which normally have low frequencies. Additionally the controller must not create high currents (i.e. high forces) at high frequencies in order to prevent saturation of the power amplifiers due to noise (also see section 3).

Fig. 9 shows the magnetic bearing transfer function (sensor displacement to bearing force) including the sensor, the controller, the amplifier and the actuator. The actuator has a magnetic pull, which can be modelled as a spring with negative stiffness (k_s=1.57 10^7 N/m) and which has to be compensated for by the controller. The pull is not included in the transfer function as shown.

The bearing provides a damping force if the phase angle is in the following range:

$$0 \prec \varphi \prec 90^o$$

or

$$-180^o \succ \varphi \succ -270^o \, .$$

In the latter case the bearing stiffness is negative, which does not lead to instabilities at high frequencies. Damping is provided for frequencies below 125Hz and for frequencies above 270Hz. Between 125Hz and 270Hz a negative damping force is created. This is mainly due to a second order filter with its frequency at 160Hz, which has two functions:

1. To reduce the amplitude at high frequencies.
2. To drop the phase, which has a tendency to decrease with increasing frequency (due to the amplifier characteristic and the digital controller), below 180^o into a positive damping region.

The filter was introduced in the frequency range of the second bending mode, because this mode has a node close to the outboard sensors. In fact, the node is even slightly inside the sensors (see figure 8). The negative damping force provides low rotor damping for this mode, which is sufficient. This is also the reason, why the outboard sensors were chosen as active sensors.

At around 18 Hz the phase is increased by an extra filter. This is to increase the damping of the rotor rigid body modes, which are in this frequency range and which are mostly affected by the destabilising labyrinth seal forces.

The overall controller transfer function is an eighteenth order polynomial that is designed using the bearing manufactures controller design software. To fully optimise the controller for this application the transfer function is synthesised with complex rather than simple poles and zeros. Digital controller hardware is essential to the implementation of this type of controller transfer function.

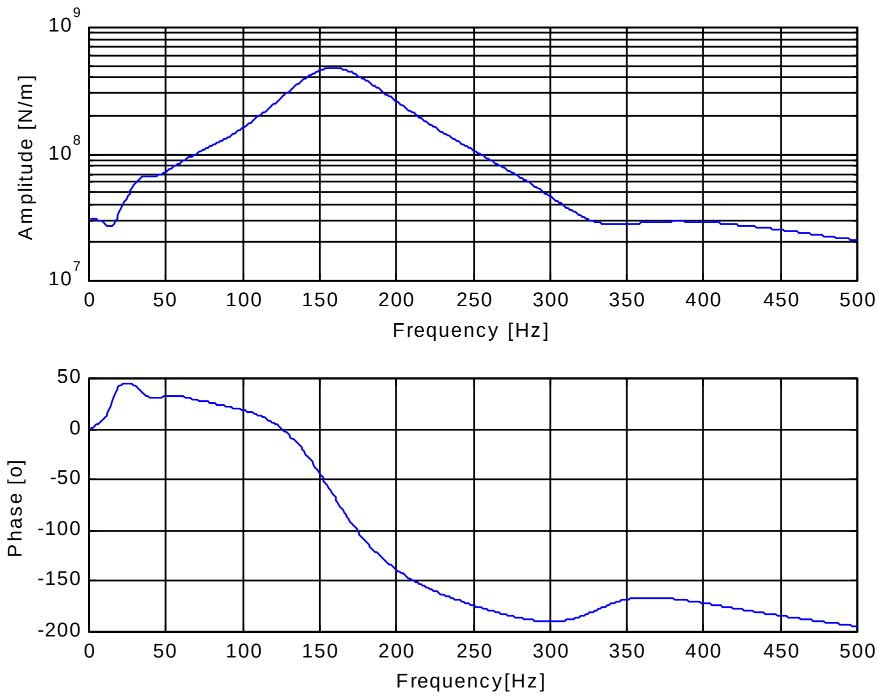


Fig.9 Magnetic Bearing Transfer Function (without magnetic pull)

7 ACHIEVED ROTORDYNAMIC PERFORMANCE

The eigenvalues (natural frequency and damping ratio) of the combined rotor bearing system at nominal speed and standstill are listed in Table 1. The table contains all eigenvalues which can be assigned to the rotor. Additionally some eigenvalues are caused by controller poles, which interact with the rotor. This interaction can change their frequency and damping ratio. The eigenvalues caused by the controller with a damping ratio below 20% are also shown in the table.

All modes below the maximum speed are very well damped. The first bending mode, which is within the operating speed range, has a damping ratio of 20%. This covers the API 617 requirement for compressors, although its application to compressors supported on magnetic bearings is controversial.

Fig.10 shows the calculated and measured bearing response of the DE bearing to an unbalance of the magnitude of G2, i.e. 3300 gramm-millimeters is applied at the thrust disk and 2700 gramm-millimeters at the coupling, at the same angular position. This distribution yields a good excitation for the first bending mode. The maximum force below the maximum speed of 6300 rpm remains below the bearing dynamic capacity of 20000 N peak-peak.

The agreement between measurement and calculation is good.

The calculated force at higher speeds (where the compressor is not required to operate) increases due to the controller dominated pole at 130 Hz, which has relatively low damping. Also the measured force remains at a high level up to maximum speed for the same reason.

Table 1: Natural Frequencies and Damping Ratio of the Rotor Bearing System

Mode	Frequency n=0	Damping Ratio n=0	Frequency n=6300rpm	Damping Ratio n=6300rpm
rotor parallel	12.6 Hz	25.8%	-12.6 Hz +12.6 Hz	25.8 % 25.8 %
rotor tilting	14 Hz	28.8%	- 13.8 Hz + 14.1 Hz	28.8 % 28.9%
controller pole 35.8Hz, D=0.4	35.8 Hz	18.5%	- 35.7 Hz + 35.9 Hz	18.3 % 18.7 %
rotor 1.bending	81.2 Hz	19.8 %	- 77.3 Hz + 84.7 Hz	21.0 % 19.2 %
controller pole 161Hz, D=0.2	130 Hz	6.8 %	- 129 Hz + 131 Hz	7.8 % 6.5 %
controller pole 161Hz, D=0.2	131 Hz	9.5 %	- 131 Hz + 132 Hz	12.3 % 15.5 %
rotor 2.bending	155 Hz	1.4 %	- 143 Hz + 164 Hz	1.5 % 1 %
rotor 3.bending	250 Hz	0 %	- 223 Hz + 272 Hz	0 % 0 %
rotor 4.bending	355 Hz	0 %	- 320 Hz + 396 Hz	0 % 0 %
rotor disk mode	488 Hz	0 %	- 454 Hz + 538 Hz	0 % 0 %
rotor 4.bending	587 Hz	0 %	- 558 Hz + 631 Hz	0 % 0 %

+ forward whirling, -backward whirling
D= damping ratio

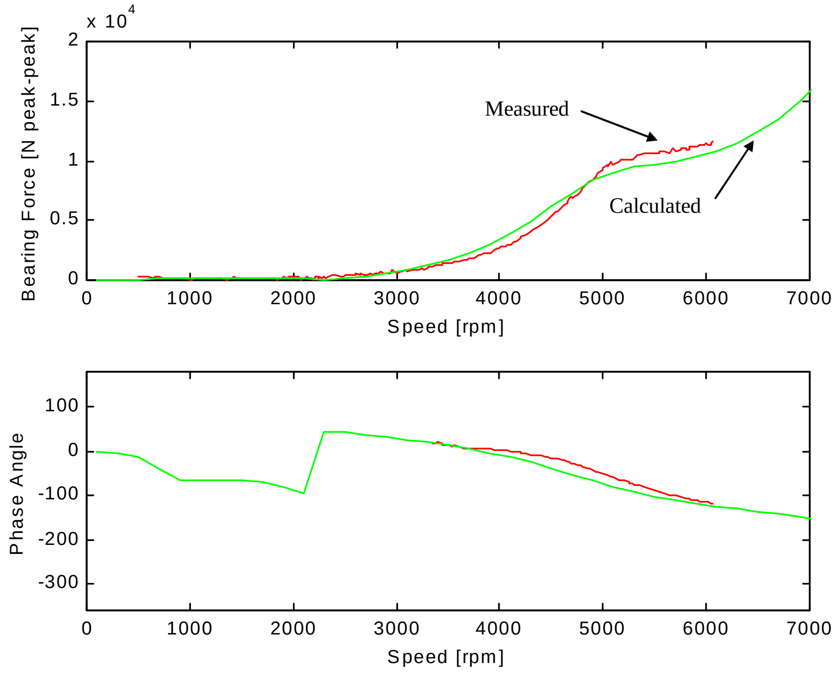


Fig.10 DE Bearing Response to an Unbalance G2 (peak-peak amplitude)

8 CONCLUSIONS

Traditional rotordynamic analysis programs for rotors supported by oil lubricated bearings, are not sufficient to completely predict the rotordynamic behaviour of rotors with active magnetic bearings. For accurate predictions, a comprehensive analysis including the complete bearing control system is necessary. The rotor design analysis, described in this paper, uses a mechatronic part that was programmed in MATLAB. In the design phase of the rotor bearing system, the consequences of changes of the controller or the rotor can easily be evaluated.

For active magnetic bearings in compressors, the axial bearing design usually results in a relatively large disk. The dynamic effects of this disk have to be taken into account.

The agreement between a measured and calculated unbalance response is excellent (see Figure10).

Although the rotordynamic guidelines in the API 617 standard may be applied for magnetic bearing applications, it is felt that these criteria are less suitable for magnetic bearings. Inhouse criteria have been developed. Future issues of the API 617 guideline will be able to benefit from the knowledge of active bearing applications now becoming available.

REFERENCES

[1] Present limits of operation of product lubricated and magnetic bearings in pumps. M.K. Swann, J. Watkins and K.R. Bornstein, Proceedings from the 14-th international Pump Users Symposium, 1996.

[2] API 617, Centrifugal Compressors for Petroleum, Chemical, and Gas Service Industries, 6-th edition, February 1995.

[3] Active Magnetic Bearings on Compressor Applications. J.K. Koo, S.B. Bennett and E.A. Bulanowski, Presentation at the Vibration Institute, May 22, 1986.

[4] Application of Magnetic Bearings and Dry gas Sealing in Axial Inlet process Compressors. C. W. Pearson, H. J. Aarnink, J. Magee and J.G.H. Derkink, Proceedings of the 2nd International Symposium on Magnetic Bearings, July 12-14, 1990, Tokyo.

[5] Dynamic Labyrinth Coefficients from a High Pressure Full-scale Test Rig using Magnetic Bearings. Norbert G. Wagner and Klaus Steff, Proceeding at the 8th Workshop on Rotordynamic Instability Problems in High Performance Turbomachinery, May 6-8, 1996 Texas A & M University.

[6] Engineering for Rotors Supported on magnetic Bearings. J. Schmied, F. Betschon, Proceedings of the 6th International Symposium on Magnetic Bearings, August 5-7, 1998, Boston.