# SIMULATION-BASED DESIGN METHOD OF ACTIVE MAGNETIC BEARINGS ACCORDING TO THE CRITERION OF DYNAMIC STIFFNESS (DDS)

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## ABSTRACT

Applications of magnetically supported rotating machinery are expected to prove, that real loads can be controlled reliably by active magnetic bearings (AMB) during normal operation and disturbances. The basis for the optimal design is firstly the knowledge about the rotor dynamic behavior during operation and secondly the specific load acting on the rotor. Thereby, static and dynamic shares as well as their superpositions must be considered, and also specific process demands (accuracy of rotor position, maximum allowable rotor displacement).

Up to now it has ben usual to design active magnetic bearings on the basis of the maximum static load, expected from the application. Often, the dynamic stiffness (as frequency-dependent ratio between disturbance input force and output displacement) is considered only during the start-up phase. At that time the used controller and power amplifier are already determined. Adjustments are merely possible in the setting range of the controller. The tuning of the loop components are not always optimal. In addition, the expenditures for start-up increase.

The paper describes a simulation-based method for the AMB Design on the parameter Dynamic Stiffness (DDS). The method is also applicable to the theoretical proof of the reliability performance during normal operation and disturbances.

The simulation tool MLDyn was developed by IPM. The dynamic stiffness is determined as a function of the frequency of the real disturbance forces. A necessary dynamic stiffness is introduced as a criterion for the reliability performance and an allowable shaft displacement as the weighting criterion. The proposed DDS method is presented for a process pump in a power plant.

## INTRODUCTION

Applications of magnetic supported rotating machinery are expected to prove, that real loads can be controlled reliably by magnetic bearings during normal operation and disturbances. The basis for the optimal design is firstly the knowledge about the rotor dynamic behaviour during operation and secondly the specific loads acting on the rotor.

Attention should be paid to:

- static and dynamic loads during normal operation and disturbances,
- specific process demands (accuracy of rotor position, absolute and relative maximum allowable rotor displacement).

To make optimal use of the advantages of magnetic bearings it is necessary to know the possible characteristic loads of the machines.

For pumps forces can be determined resulting from

- cavitation,
- loads at disturbances based on pressure changes (sudden boiling at negative transients),
- waterslaps and
- erosion on the working machine.

The paper describes a simulation-based method for the AMB Design regard to the parameter Dynamic Stiffness (DDS). In the design phase the method is also applicable to the theoretical proof of the reliability performance of active magnetic bearing systems. This is based on the Simulation Tool MLDyn developed by IPM. **Necessary dynamic stiffness** is introduced as a criterion for reliability and **allowable rotor displacement** as a criterion for quality.

The proposed DDS method is shown for a process pump in a power plant.

**SIMULATION TOOL MLDyn AND MAGNETIC BEARING TEST FACILITY FLP 500**

The simulation tool MLDyn was especially developed for theoretical investigations in the field of active magnetic bearing systems. MLDyn contains all components of the AMB control loop (Fig. 1).

MLDyn is characterized by
- modelling the magnetic bearing control loop for completely active magnetically supported rigid rotors,
- a modular type of construction and an easy exchange of components of the control loop,
- emergency operation for imbalances and unit loads,
- the possibility of configuration for any magnetic bearing systems by adjustment of characteristic parameters and structures,
- verification at pilot plants.

Application fields are
- investigations of the dynamics of rotors,
- loop investigations (transients, numbers of revolutions, critical operating situations, start-up and shut-down operations),
- investigations of the control loop stability,
- support in controller design,
- preparation of experiments.

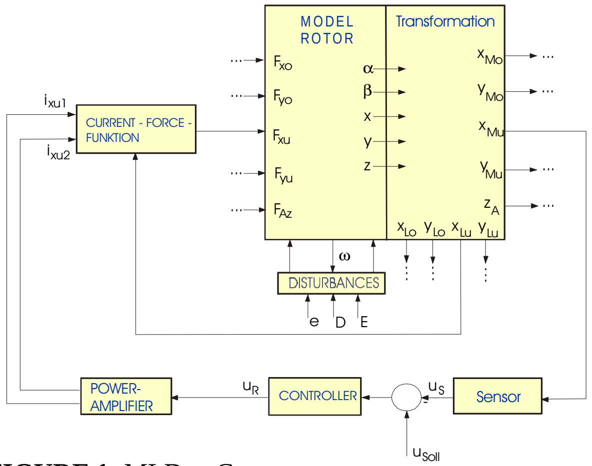


**FIGURE 1:** MLDyn Components

For the verification of MLDyn the test facility FLP 500 (Fig. 2) is used [Ref. 1]. Fig. 2 shows the principal mechanical construction of this test facility.

The FLP500 is a vertical machine with an asynchronous motor (nominal power of 240 kW) speed controlled by a frequency converter. The rotor with a mass of 1300 kg is fixed completely by means of redundant axial and radial magnetic bearings. The maximum rotational speed is 7200 rpm.

The axial magnetic bearing is located in the upper part of the machine. The upper radial bearing is mounted below the axial bearing and the lower radial bearing below the asynchronous drive. All components are installed in a vessel with a diameter of 1m, a height of about 3 m and a mass of 7000 kg. Table 1 gives a survey of the most important parameters.

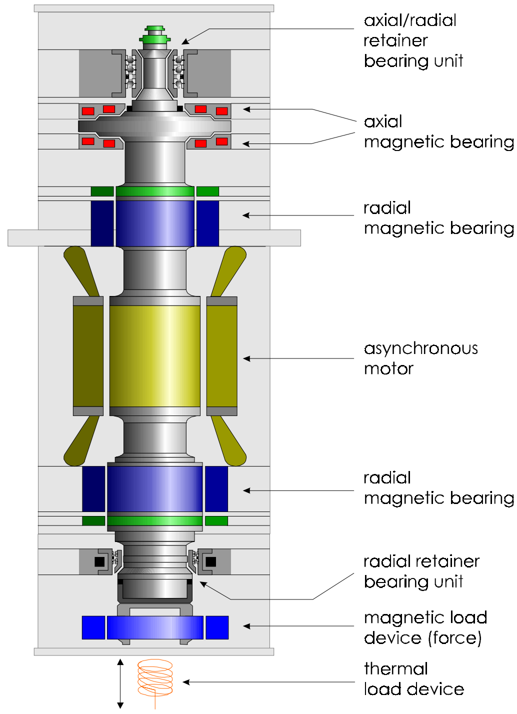


**FIGURE 2:** Magnetic Bearing Test Facility FLP 500

**TABLE 1:** Characteristic Parameters of FLP 500

| | |
|---|---|
| MASS OF FOUNDATION | 47 t |
| TEST VESSEL DIMENSION<br>- DIAMETER<br>- HEIGHT<br>- MASS<br>ROTOR MASS | <br>1 m<br>3 m<br>7 t<br>1.3 t |
| CLEARANCE OF RETAINER BEARING<br>- AXIAL/RADIAL<br>- RADIAL | <br>0.6 mm<br>0.4 mm |
| AIRGAP OF MAGNETIC BEARING<br>- AXIAL<br>ABOVE<br>BELOW<br>- RADIAL | <br><br>1.0 mm<br>1.2 mm<br>1.0 mm |
| MAGNETIC BEARING LOAD<br>MAX. LOAD<br>- AXIAL<br>- UPPER RADIAL I<br>- LOWER RADIAL II<br>PULSE WIDTH MODULATING AMPLIFIER<br>- VOLTAGE<br>- MAX. CONTROL CURRENT<br>- FREQUENCY | <br><br>124,000 N<br>6,240 N<br>10,380 N<br><br>160 V<br>50 A<br>17 kHz |
| NOMINAL POWER OF ASYNCHRONOUS DRIVE | 240 kW |

## METHOD FOR THE PROOF OF RELIABILITY PERFORMANCE BASED ON THE CRITERION OF DYNAMIC STIFFNESS

### LOAD CAPACITY AND STIFFNESS

Analogous to conventional bearings at magnetic bearings load capacity and stiffness can be defined.

The static load capacity $F_{max}$ is the maximum load capacity for a static force over an unlimited time period. It is stated by

- maximal forces of the application
- the geometry of the arrangement
- the design of the machine

The load is limited by the maximal receivable ampere turns, i.e.restricted by coil temperature and voltage limit.

The stiffness S is given by the negative position stiffness $K_S$ of the electromagnet and the controller gain.

Different from the conventional bearing the AMB - stiffness is dependent on the operating frequency ω of the disturbance forces [Ref. 2]. Therefore it there is a difference between static and dynamic stiffness.

The dynamic stiffness for a closed loop is

$$Sd(s) = \frac{1}{Gz(s)} = \frac{F_{Dis}(s)}{X(s)} \qquad (1)$$

where

s - Laplace-Operator
Gz(s) - disturbance transfer function
$F_{Dis}(s)$ - disturbance force dependent on frequency
X(s) - displacement

The static stiffness is the limit of Sd for t→∞ and s→0 by a static force $\Delta F_{Static}$ different from the working point.

$$S_{static} = \lim_{s\to 0} Sd(s) = \frac{\Delta F_{static}}{X} \qquad (2)$$

The attainable maximum stiffness is dependent on the frequency behaviour of the control loop. The proof of the reliability performance requires an analysis of the whole frequency spectrum of the disturbance forces.

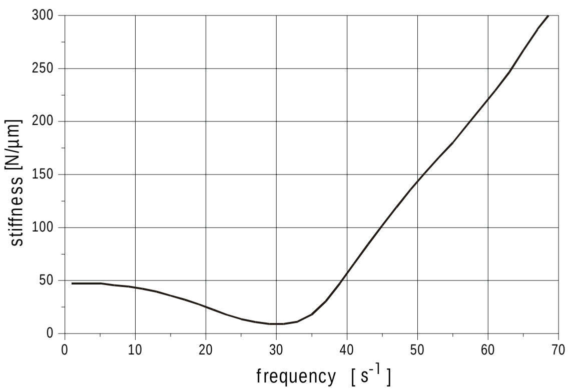


**FIGURE 3:** Dynamic stiffness versus the frequency of the upper radial AMB FLP 500 with $I_0$ = 6.45 A and $F_{Dis}(\omega) = \hat{F}\cdot\sin(\omega t)$, $\hat{F} = 1000\,N$

In Figure 3 the stiffness of a radial bearing is shown as an example (test facility FLP 500) as a function of the disturbance frequency. Only up to 5 Hz the stiffness is 50 N/µm. If FLP 500 operates with 1.800 RPM (30 Hz) the stiffness is nearly 20 N/µm. Due to the rotor mass inertia the stiffness increases from 32 Hz.

### ALGORITHM

Depend on the expected application requirements, the necessary AMB force FD is determined. This is followed by an adjustment of AMB components like sensors and amplifier and a first optimization of the control loop. Based on the required AMB feature, a necessary stiffness Sn is defined, which the bearing has to be adapted to:

$$Sn = \frac{F_{max}}{X_{al}} \qquad (3)$$

where $F_{max} = F_D + F_{Res}$

$F_D$ - design force
$F_{Res}$ - reserve force (safety allowance)
$X_{al}$ - allowable rotor displacement

The allowable rotor displacement $X_{al}$ is given by constructive (i.g. seals) and operative (positioning accuracy) requirements of the application and is introduced as the quality criterion, which has to be secured by the AMB system.

The accessible dynamic stiffness is calculated by MLDyn in consideration to the AMB design selected.

$$Sd(\omega) = \frac{F_D + F_{Dist}(\omega)}{X(\omega)} \qquad (4)$$

Figure 4 is a qualitative representation of Sd(ω). The result at the point of intersection between the simulated dynamic stiffness and the necessary stiffness Sn is the maximum frequency $\omega_{max}$ for acting disturbance forces. This frequency limits the allowable operating range and therefore it is the design limit. Above the design limit ($\omega > \omega_{max}$) the dynamic stiffness is below the necessary stiffness. The perfect operation of the drafted AMB is not guaranteed. On the basis of an analysis of the real frequency spectrum at a machine the dominating frequency $\omega_{Op}$ is determined as well as the operating stiffness $S_{Op}$. The criterion of the reliability performance of the AMB is

$$\mathbf{S_{Op} > Sn} \qquad (5)$$

If the relationship (5) is not fulfilled in a first step a controller fine tuning follows. If the tuning is not successful in a next step the AMB parameter must be determined again. The whole algorithm is shown in Figure 5.

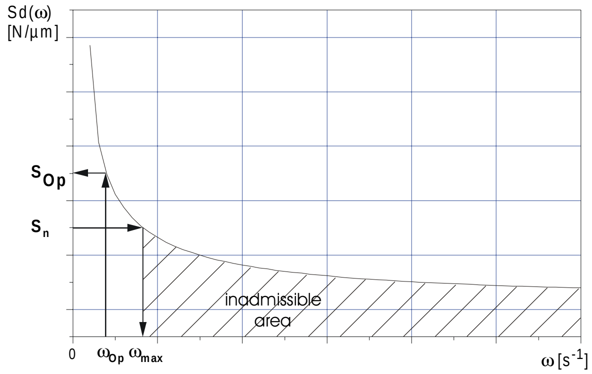


**FIGURE 4:** Closed loop dynamic system stiffness versus frequency

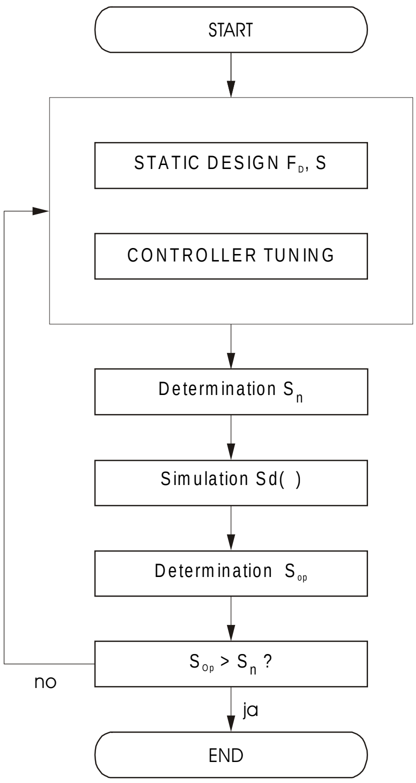


**FIGURE 5:** Algorithm to proof of the reliability performance for AMB

## DESIGN OF AN ACTIVE MAGNETICALLY SUPPORTED PUMP

The described algorithm is exemplary for the magnetic bearing design and proves its function in process type design at a cooling water pump in a conventional power plant.

Figure 6. shows the construction of the single-stage single-suction centrifugal pump with flying supported impeller in principle.

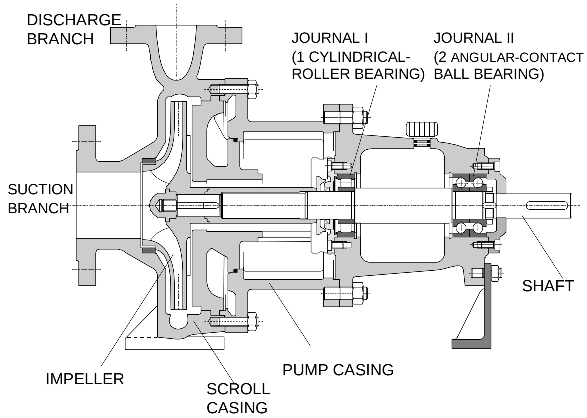


**FIGURE 6:** Construction of the cooling water pump [Ref. 3]

The procedure to design the magnetic bearings was as follows:

1.calculation of mass and moment of inertia for the original rotor
2.static design of the magnetic bearings for the original rotor
3. recalculation of mass and moment of inertia for the rotor with radial bearing rotor cores and thrust disk of axial bearing
4.static design of magnetic bearings taking 3 into consideration of 3.
5. numerical simulation to verify the design with the above described method

The foundation for determining the bearings' load capacity are the axial and radial thrust and the geometrical proportions at the pump. Table 2 shows the parameters of the magnetic bearings conceived.

**TABLE 2:** Parameters of the magnetic bearings conceived

| Parameter | Radial Bearing AMB I not impeller sided | Radial Bearing AMB II impeller sided | Axial Bearing |
|---|---|---|---|
| Load Capacity $F_D$ [kN] | 2,65 | 4,06 | 30,4 |
| Lifting Force $F_L$ [kN] | 1,23 | 1,89 | 14,1 |
| Air Gap [mm] | 0,6 | 0,6 | 1 |
| Retainer Bearing Clearance [mm] | 0,3 | 0,3 | 0,5 |
| Power Amplifier: Continuous Current[A] Output Voltage[V] | 14<br>140 | 14<br>140 | 14<br>140 |

The magnetic support used in radial direction consists of two heteropolar magnetic bearings with eight poles, whereas in axial direction it has two ring magnets and a thrust disk. Figure 7 shows the geometrical positions of the magnetic bearing at a pump.

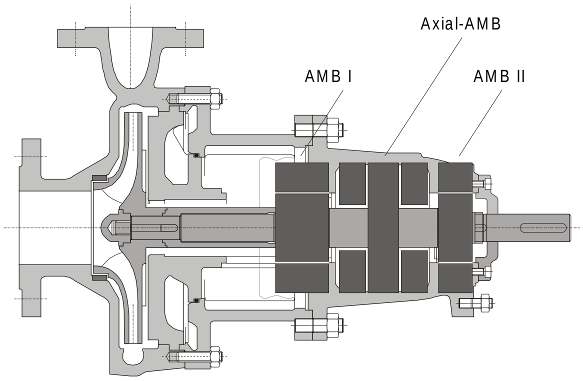


**FIGURE 7:** Disposition of the magnetic bearings at the pump

The allowable rotor displacement Xal as quality criterion is given by the inserted seals at this pump. The geometrical proportions result in a value $X_{al}$ of 40 µm in the radial bearings [Ref. 3].
Under consideration
- of the optimal arrangement of the magnetic bearing,
- of the force at the bearing point subject to the position of the shaft mass centre,
- of the hydraulic force Fhyd at the impeller and
- of a safety factor Y from 10 %

the necessary stiffness $Sn_{AMB\,I}$ at the radial AMB I amounts to

$$Sn_{AMB\,I} = \frac{F_{max}}{X_{al}} = \frac{F_D + \Psi \cdot F_{res}}{X_{al}} = 61{,}4\,N/\mu m \qquad (6)$$

$\Psi$ - safety factor

The magnitude of the disturbance force was selected in reference to [Ref. 3] with 20% of the axial load for 0-175 Hz. The result of the simulation is shown in Figure 8.

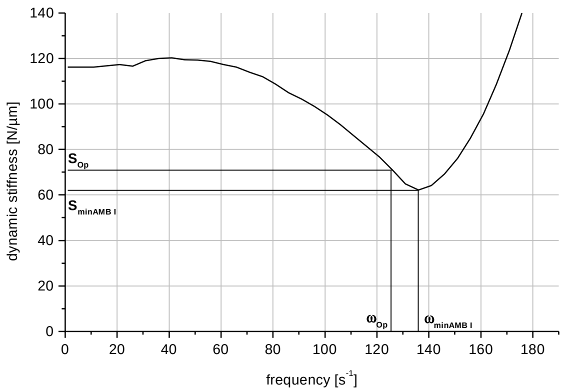


**FIGURE 8:** Simulated Dynamic Stiffness versus Frequency AMB I

The minimum of the simulated dynamic stiffness is $S_{min}$ = 63 N/µm at 136 Hz. This is above the necessary stiffness $Sn_{AMB\,I}$.
The dominated pump frequency has been determined by long-term measurements in a power plant. The pressure and the radial acceleration have been recorded. Figure 9 shows the characteristic amplitude frequency spectrum measured.

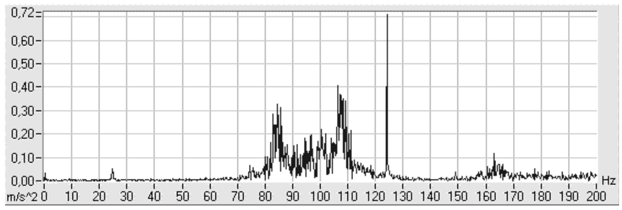


**FIGURE 9:** Amplitude spectrum of the radial acceleration

The value of the dominant operating frequency $\omega_{Op}$ is 125 $s^{-1}$ and thus equal to the frequency of the pump shovels. By means of the dominant operating frequency $\omega_{Op}$ the operating stiffness $S_{Op}$ is determined as shown in Figure 8.

$$S_{Op} = Sd(125s^{-1}) = 71{,}2\,N/\mu m \qquad (7)$$

The simulated dynamic stiffness is larger then the necessary stiffness $Sn_{AMB\,I}$ (Fig. 8) in the complete frequency range. The criterion of the reliability performance of the AMB is fulfilled. The radial magnetic bearing conceived complies with specific process demands.
For the other magnetic bearings the calculations have been performed respectively. The results are shown in table 3.

**TABLE 3:** Results of Simulation

| | Radial Bearing AMB I not impeller sided | Radial Bearing AMB II impeller sided | Axial Bearing |
|---|---|---|---|
| $X_{al}$ [µm] | 40 | 40 | 100 |
| Sn [N/µm] | 61,4 | 13,1 | 300 |
| $S_{Op}$ [N/µm] | 71,2 | 37,5 | 861 |
| $\omega_{Op}$ [$s^{-1}$] | 125 | 125 | 125 |
| $S_{min}$ [N/µm] | 63 | 34,2 | 371 |
| $\omega_{min}$ [$s^{-1}$] | 136 | 136 | 1 |
| $K_R$ | 8 | 5 | 1,3 |
| $T_N$ [s] | 0,065 | 0,07 | 0,5 |
| $T_V$ [s] | 0,02 | 0,012 | 0,005 |

## CONCLUSIONS

The simulation-based method described makes possible to design magnetically supported machines and/or to prove the function of the AMB conceived. A condition for using this method is the numeric simulation performed with simulation tool MLDyn. The method allows the optimal design of magnetic bearings according to the specific loads already during the planning stage. This allows a efficient planning and a design of magnetic bearings exceeding the static design used up to now.

## REFERENCES

[Ref. 1] ISO/CD 14839 in preparation
Mechanical vibration - Vibration of active magnecic bearing equipped rotating machinery

[Ref. 2] Schweitzer,G.; Traxler, A.; Bleuler, H.
Magnetlager - Grundlagen, Eigenschaften und Anwendungen berührungsfreier, elektromagnetischer Lager
Springer Verlag Berlin Heidelberg, 1993

[Ref 3.] Autorenkollektiv
Technisches Handbuch Pumpen
Verlag Technik Berlin, 1987