# Magnetic Excitation of a Three-Bearing Rotor

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## ABSTRACT

Exciting mechanisms, such as the flow induced clearance or aerodynamic excitation and the hydrodynamic bearing excitation cause self-excited vibrations of the turbomachinery rotor. By a magnetic simulation of the flow-induced nonconservative forces, the stability behavior and stabilizing measures can be investigated, on the bases of arbitrary rotor configurations. The magnetic bearing is used to excite a three-bearing rotor. The natural frequencies are varried by the direct stiffness of the magnetic bearing. Stiffening of the shaft does not stabilize the multi-bearing rotor generally. When the magnetic excitation counteracts the journal bearing excitation, the stability of the rotor is improved for the first and the second vibration mode. The experimental results are compared to theoretical results obtained by finite element calculation.

## INTRODUCTION

As the working pressure and the running speed of high-perfomance turbomachinery is increased, a reliable prediction of the stability limit is required. In the past the comparison between experimental and theoretical results was mostly restricted to rotors carried in two radial bearings. A substantial problem for experimental investigation was the complicated generation of flow-induced nonconservative forces with sufficient reproducibility. This problem can be solved effectively by using a magnetic bearing /1/.

There are two main exciting mechanisms acting on the rotor of turbomachines: the flow-induced clearance excitation, leading to self-excited vibrations of the rotor as soon as a certain load is exceeded /2, 3/, and the oil-whip, which is induced by hydrodynamic bearings. For the latter case, instability occurs if a threshold speed is exceeded. These two exciting mechanisms are acting together in many cases. On the other hand, it is also possible to stabilize the rotor by magnetically counteracting the excitation of the journal bearings.

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First, the exciting mechanisms and the simulation procedure will be presented briefly. Then the three-bearing laboratory rotor and the experimental procedure will be explained, and, in a final step, the stiffness of the rotor as well as the rotational direction of excitation of the magnetic bearing are varried and the stability limits are compared to the results of a finite element calculation.

## EXCITING MECHANISMS AND SIMULATION PROCEDURE

Exciting mechanisms occuring in turbomachinery are induced by lateral forces which transfer rotative energy into the rotor bending vibrations. In case of instability, the vibration caused by unbalance is overlaid by a vibration close to an eigen-frequency of the rotor.

The clearance excitation is caused by non-uniform blade tip leakage losses and assymmetric pressure distribution. If only the displacement-dependent linearized excitation constant $q$ is taken into consideration, the cross-coupling force $F^Q$ is given by

$$\begin{pmatrix} F_x^Q \\ F_y^Q \end{pmatrix} = \begin{pmatrix} 0 & -q \\ q & 0 \end{pmatrix} \begin{pmatrix} x \\ y \end{pmatrix} \tag{1}$$

It is rather complicated to generate these exciting forces in the experimental field with suffucient reproducibility, especially when certain parameters are to be varried systematically.

Based on the assumption of small vibrations, the dynamic properties of the lubrication film of journal bearings are described by four stiffness and four damping coefficients. The bearing force is obtained by the following eq.

$$\begin{pmatrix} F_x^B \\ F_y^B \end{pmatrix} = -\begin{pmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{pmatrix} \begin{pmatrix} x \\ y \end{pmatrix} - \begin{pmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy} \end{pmatrix} \begin{pmatrix} \dot{x} \\ \dot{y} \end{pmatrix} \tag{2}$$

As the running speed of the rotor is increased, unstable vibrations occur when the damping becomes insufficient to dissipate the energy resulting from the nonconservative cross-coupling forces. The stiffness matrix can be splitted in a conservative symmetric part and a nonconservative skew-symmetric part, which is analogue to the clearance excitation (eq. 1; see for example /4/).

The magnetic bearing used to simulate the clearance excitation is identical to the bearing used by *Ulbrich* /1/. The well-known linear current-force characteristic is obtained by a differential arrangement and due to the bias current the bearing exerts a force on the displaced rotor, even without a control current.

$$F_x^M = k_i i_c + k_s x \tag{3}$$

A signal conditioning as outlined in fig. 1 permits the magnetic generation of circulatory forces. The commercial design eddy current probes are located in the same differential arrangement as the horseshoe-magnets.

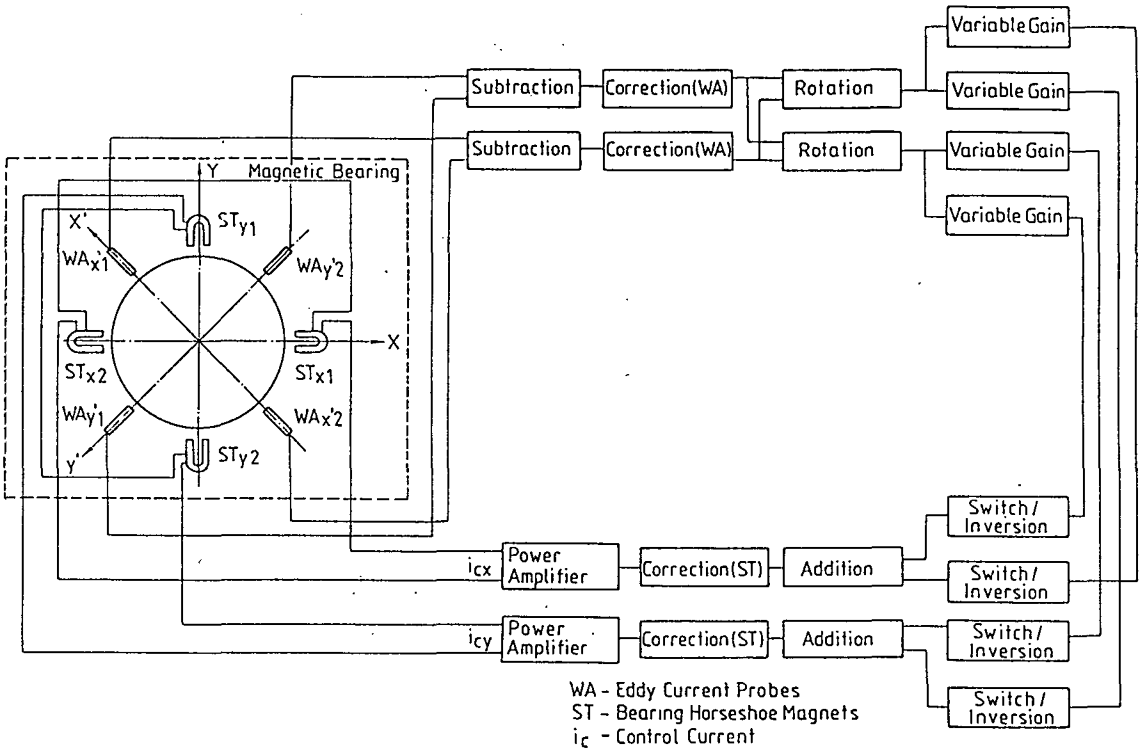


**Fig. 1** Schematic of the magnetic bearing controll system

$$\begin{pmatrix} F_x^M \\ F_y^M \end{pmatrix} = \begin{pmatrix} r & -q \\ q & r \end{pmatrix} \begin{pmatrix} x \\ y \end{pmatrix} \tag{4}$$

$$q = k_i V_l$$

$$r = k_i V_d + k_s$$

The amplification and conversion of a displacement proportional voltage into a control current by analogous components is symbolized with $V_d$ and $V_l$. By varying $V_l$, a known excitation is adjusted. The critical speeds of the rotor and the vibrational modes can be influenced by $V_d$. The sign of the conversion factors and, in consequence, the direction of the force can be changed by an inversion.

## EXPERIMENTAL INVESTIGATIONS AND COMPARISON TO THEORY

The experimental investigations were performed on a rotor carried in three journal bearings (fig 2). The test rig should represent a multi-bearing turbomachine rotor with a light-weight high-pressure shaft section and a weighty low pressure section. The journal bearings are of the three-lobe type. The magnetic bearing is located in the same light-weight rotor section as the electromagnetic shock-exciter. The pulse force of the shock-exciter is generated by unloading a condenser battery via a coil. For further details of the laboratory rotor and the instrumentation see /5, 6/.

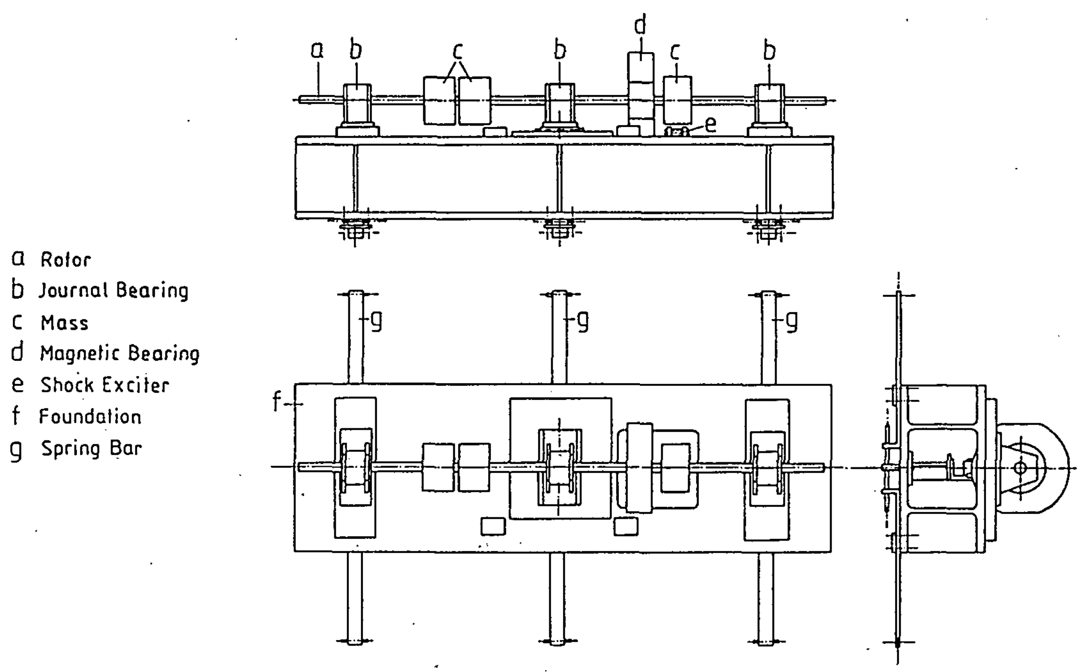


**Fig. 2** Three bearing laboratory rotor

The main scope of the investigations was the stability behavior of the rotor and its improvement, as well as the comparison to the results of a finite element calculation. The stability limit is marked by a change in the vibration amplitude and the orbit shape at a certain running speed and for a certain excitation by the magnetic bearing. At the same time, the system damping of one natural frequency becomes zero, and this unstable natural frequency can be observed with the signal analyzer. The damping was measured out of the decay curve of the perturbed rotor.

The damping of the first and the second natural frequency is shown in fig. 3. As the damping of the second natural frequency becomes zero, the second mode is affected by instability. The calculated damping values and stability limits, which are marked by curves, were performed with various bearing coefficients /7, 8, 9/. The deviations between experiment and calculation are contained within the range of deviations obtained by calculation with different bearing coefficients.

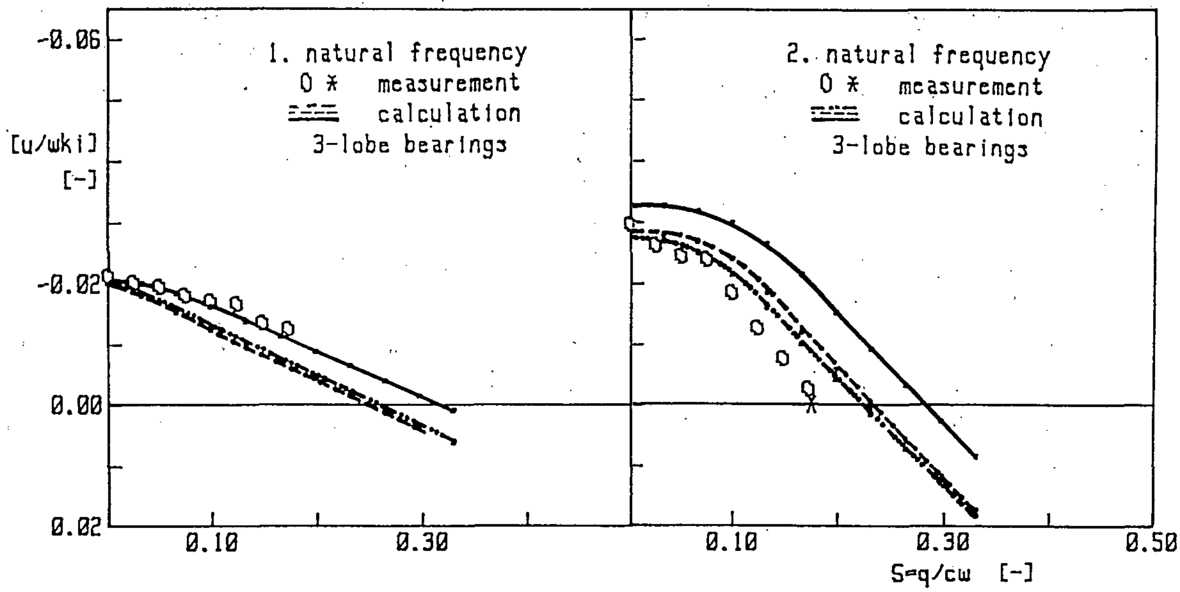


**Fig. 3** Damping of the first and the second natural frequency versus excitation. ($\omega_{k1}$=420 rad/s; $\omega_{k2}$=725 rad/s; $c_w$=1,516 N/mm, rotor stiffness; $u$ - damping; $\Omega$=293 rad/s; bearing data /7/ ——, /8/ –··–, /9/ - - -)

By varying the direct stiffness of the magnetic bearing, the natural frequencies and the vibrational modes change. When other exciting mechanisms are neglected, that mode becomes unstable, which is more deflected and, as a consequence, more excited by the magnetic bearing. Whereas the first mode is more pronounced when the magnetic bearing weakens a shaft section, the second mode is more bended when the shaft section is stiffened.

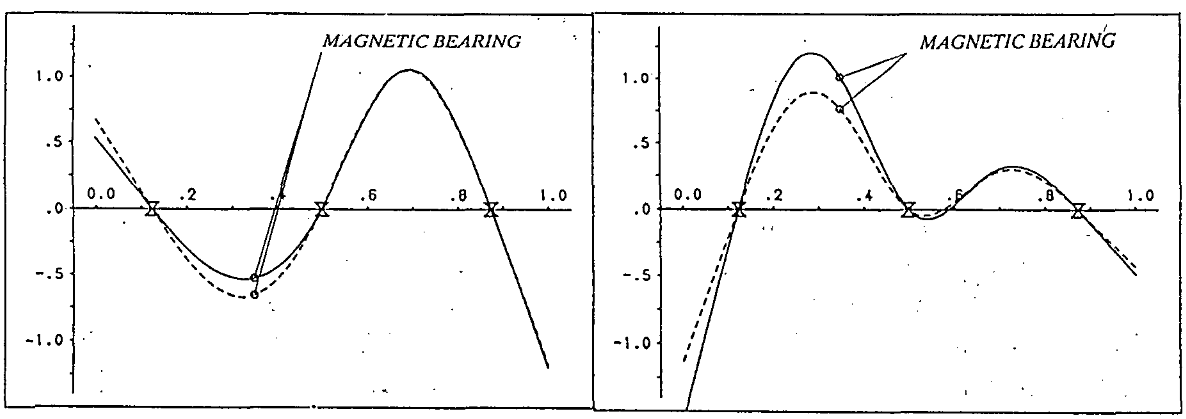


**Fig. 4** The first and second bending mode of the rotor as a function of the direct stiffness of the magnetic bearing. ($r$=-335.5 N/mm ——; $r$=410.5 N/mm - - - )

This fact is demonstrated in fig. 4 which contains the calculated first and second bending mode of the rotor in rigid bearings for two distinct values of the direct stiffness $r$. The possition of the magnetic and the journal bearings on the rotor and the relative magnitude of the amplitude can be recognized. It can be shown that contrary to the two bearing system a stiffening of the shaft and the lift of the natural frequencies do not stabilize the multi-bearing rotor generally /6/.

When both exciting mechanisms act in the same direction as the whirling rotor the stability threshold declines with increasing speed to the point where instability is caused by pure oil whip. Thus, the stability of the rotor is improved when the magnetic excitation counteracts the bearing excitation. Both, the first as well as the second bending mode can be stabilized by this simple procedure, as demonstrated in fig. 5a and 5b.

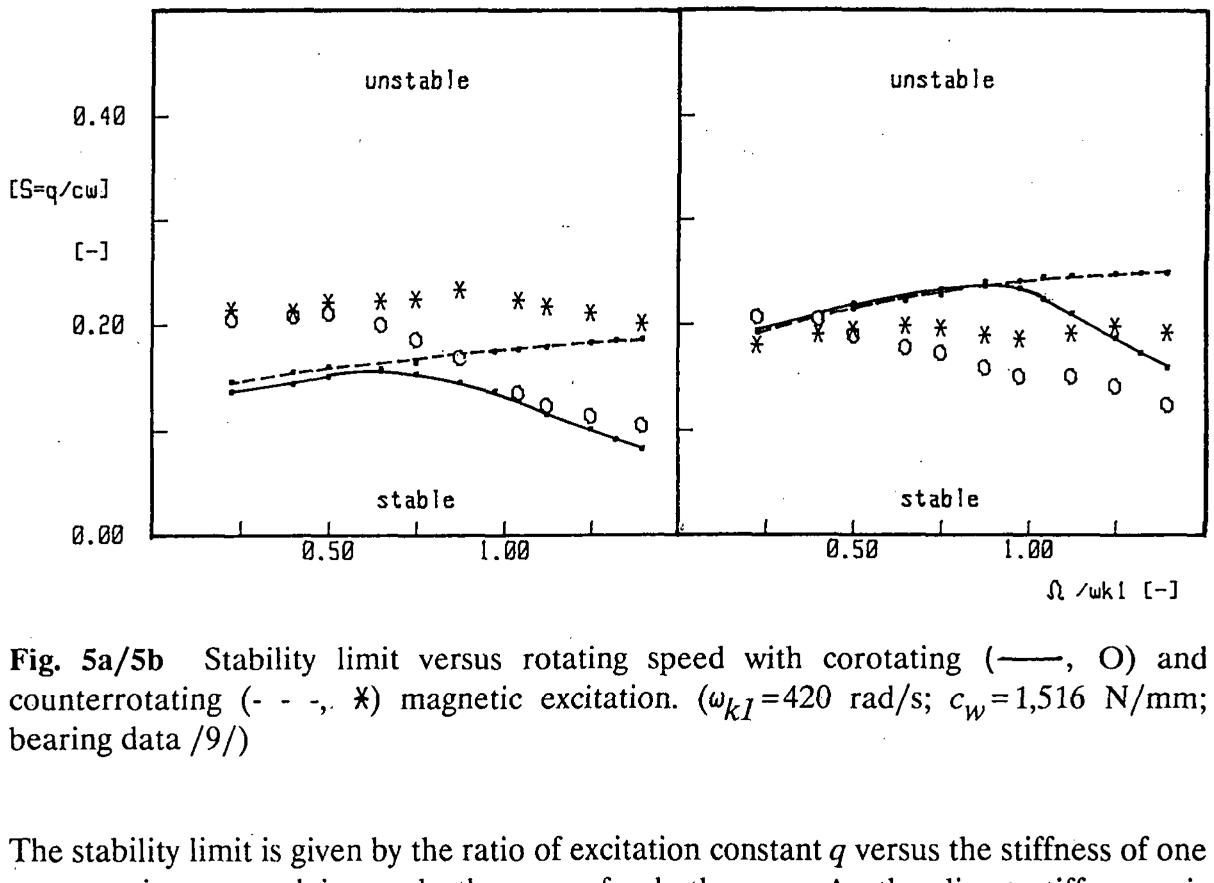


**Fig. 5a/5b** Stability limit versus rotating speed with corotating (——, O) and counterrotating (- - -,. ✱) magnetic excitation. ($\omega_{kl}$=420 rad/s; $c_w$=1,516 N/mm; bearing data /9/)

The stability limit is given by the ratio of excitation constant $q$ versus the stiffness of one rotor-section $c_w$ and is nearly the same for both cases. As the direct stiffness $r$ is increased the instability changes from the first to the second mode and the stability limit declines to previous values. The area above the measured limits, which are labeled with single symbols, and the calculated results, which are marked by curves, symbolizes the unstable region of the rotor. The direct stiffness $r$ is held constant for the case when instability is related to the first ($r$=224.0 N/mm) or the second ($r$=-335.5 N/mm) bending mode respectivly. Here, too, the coincidence of the measured and calculated results is satisfactory.

## CONCLUSIONS

The magnetic bearing permits the investigation of arbitrary rotorsystems when bearing and clearance excitation act together. It is very easy to vary certain parameters deliberatly in order to identify rotordynamic parameters such as the stability behavior and, likewise, to test measures which improve stability. Whereas stiffening of the shaft is not always advantageous, the magnetic counteracting of exciting mechanisms can be efficient. The qualitative prediction of the rotor behavior by finite element calculation is good, while the quantitative prediction of stability limits is highly sensitive to journal bearing coefficients.

## REFERENCES

/1/ *Ulbrich, H.*:New Test Techniques Using Magnetic Bearings. Magnetic Bearings. Proceedings of the First International Symposium, Switzerland (1988), Schweitzer, G, (Editor), Springer Verlag, pp. 281/288

/2/ *Thomas, H.-J.*: Instabile Eigenschwingungen von Turbinenläufern, angefacht durch die Spaltströmung in Stopfbuchsen und Beschauflungen. Bull. de'AIM 71 (1958) H.11/12, pp. 1039/1063

/3/ *Ehrich, F. F.*: A State of the Art Survey in Rotordynamics - Non-Linear and Self-Excited Vibration Phenomena. Preprints of the Second International Symposium on Transport Phenomena, Dynamics and Design of Rotating Machinery, Honolulu, Hawaii (1988), S. 1/23

/4/ *Adams, M. L., Padovan,J.*: Insights into Linearized Rotor Dynamics. Journal of Sound and Vibration, Vol. 76 (1981), pp. 129/142

/5/ *Kwanka, K.*: Laufstabilität eines dreifach gelagerten Rotorsystems bei Anregung durch nichtkonservative Querkräfte. Fortschr.-Ber. VDI, Reihe 11, Nr. 141 (1990)

/6/ *Kwanka, K.*: Stability Behavior of a Three Journal Bearing Rotor System Excited by a Lateral Force. Preprints of the Third International Symposium on Transport Phenomena and Dynamics of Rotating Machinery, Hawaii (1990), pp. 101/114

/7/ *Glienicke, J.:* Experimentelle Ermittlung von statischen und dynamischen Eigenschaften von Gleitlagern für schnellaufende Wellen. Fortschr.-Ber. VDI, R.1, Nr.22 (1970)

/8/ *Schaffrath, G.*: Das Gleitlager mit beliebiger Schmierspaltform - Verlagerung des Wellenzapfens bei zeitlich veränderlicher Belastung. Dissertation TH Karlsruhe (1967)

/9/ *Someya, T.* (Editor): Journal Bearing Databook. Springer Verlag (1989)