On the Coupling of AMB Supported High Speed Rotors

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ABSTRACT

Magnetically supported rotors usually are designed very compactly in order to maximize the value of bending eigenfrequencies. However, with increasing power output the rotating equipment becomes larger, and therefore the coupling of magnetically supported rotors should be considered. We will present some practical considerations as well as some control theoretical aspects of the coupling of AMB supported rotors.

In a conventional drive setup the shaft of the drive is coupled to the load specified by the industrial process; for example a pump or a compressor. The system components are frequently designed independently of each other, and by using an appropriate coupling device eventual problems with the overall dynamics might be overcome. However, this modular approach, which is very effective in most low speed applications, cannot be applied with high speed drive systems. The main reason for this inconvenience is that the dynamic behaviour of the single rotors is changed by the coupling. In the case of magnetic bearings the controller layout should be based on the coupled system. Otherwise the dynamic behaviour of the overall system may not satisfy the requirements, and in the worst case instability may occur.

A SHORT HISTORY OF COUPLINGS

The coupling is one of the oldest and most important design elements of mechanical engineering. Historically, rotating equipment was first connected by means of rigid flanges (Figure 1). However, since misalignment due to the initial assembly or to operating conditions always occur, rigid couplings may lead to unacceptable bearing forces. The need for flexible couplings was already recognized by the Greeks which are said to have invented the first flexible coupling, i.e. the universal joint, around 300 B.C. (Figure 2). The industrial revolution, and especially the invention of the automobile, precipitated the development of couplings. In the last decades the development of couplings for high powers and speeds was a major topic of coupling research motivated by demanding industrial requirements [3]. As a result of the early evolution today's coupling technology is highly developed. Therefore the use of magnetic bearings will not have a significant effect on coupling research, but magnetic bearings, as will be shown in the following sections, will surely influence the criteria for coupling selection. For a detailed overview on couplings and its history refer to [2].

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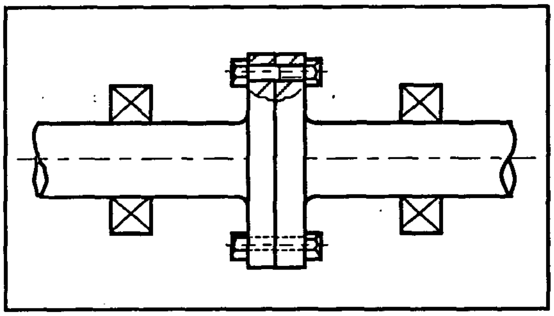

Figure 1: Rigid coupling of two shafts

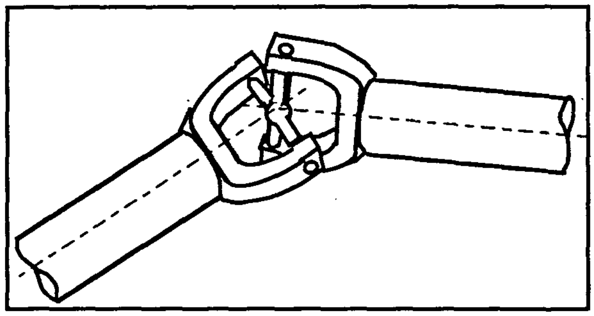

Figure 2: The universal joint - the first flexible coupling

BASIC FUNCTIONS OF COUPLINGS

The main function of a coupling is to transmit torque from the drive to the driven component of a rotating equipment.

If *rigid couplings* are used, the shaft axes should be aligned very precisely during the initial assembly of the rotating equipment. Otherwise, the rotating equipment may be subject to large moments and forces. Note that misalignment of the rotor shafts can also be caused by the flexure of the structure under load or thermal expansion during operation. In the case of conventional bearings, rigid couplings should only be used if very little misalignment is expected, and if the equipment is strong enough to accept the generated moments and forces. *Flexible couplings*, not only satisfy additional technical requirements by means of accommodating misalignment and axial displacements (Figure 3), but in many applications they are used for commercial convenience. Very often, especially in low-speed applications, the drive and the load machine are designed independently of each other and by different manufacturers, so that the relative position of their shaft ends do not necessarily match in any way. By using flexible couplings any resulting inaccuracies may be overcome, thus expensive modifications of the rotating equipment can be avoided.

LIMITS AND SELECTION OF HIGH-SPEED COUPLINGS

The true maximum speed limit at which a coupling can operate cannot be divorced from the connected equipment. Since the same coupling may be used for different types of equipment, it is very difficult to rate a coupling. In the following only the coupling without

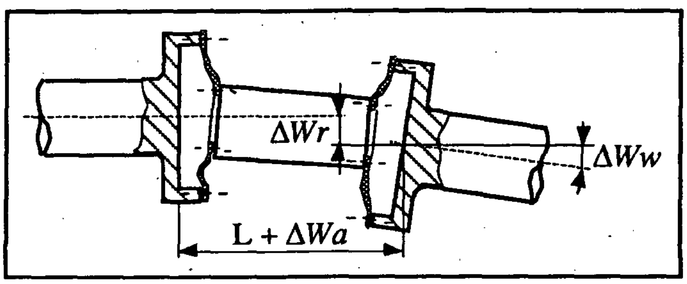

Figure 3: Flexible metallic membrane coupling with radial and angular (ΔWr, ΔWw) misalignment and axial displacement (ΔWa).

the connected equipment is considered. The design equations for different coupling ratings are described in [2]. The following considerations can affect a coupling's maximum speed capabilities: *centrifugal stresses, torque, amount of misalignment, lateral critical speed of the coupling (for long couplings), forces generated from unbalance, heat generated from flexing of mating parts (elastomeric couplings).*

The Figure below which was originally presented by *Neale* [3] shows the approximate performance boundaries of high speed couplings in function of speed and torque. A similar and more recent diagram can be found in [2], and it can be seen that these boundaries were not significantly changed, which indicates that the speed limits of couplings have already been reached. Among hundreds of different types of couplings only a few are suitable for high speed applications.

Contoured disc and *multiple membrane coupling* belong to the group of metallic membrane couplings. These couplings do not need lubrication and rely on the flexure of one ore more metallic membranes for misalignment and axial movement (see Figure 3). *Gear couplings* tolerate misalignment by means of adequate tooth profiles. As a drawback, high speed gear couplings need a continuous oil-flow lubrication. *Quill shaft couplings* belong to the group of rigid couplings. Compared to flanged rigid couplings they do accommodate for some misalignment through the flexing of their long and slender shafts. The maximum speed of quill shaft couplings is mainly limited by the strength of the shaft, and is therefore far away of the other coupling types. The shaft end connection of quill shaft couplings which may be combined with disc couplings needs careful design. Usually for a given speed and torque the system designer must select one among several coupling types. In order to select a satisfactory coupling, it is essential to regard it as part of the whole machinery system, and not to view it merely as a joint of the shafts ends. This is especially valid for magnetic bearings.

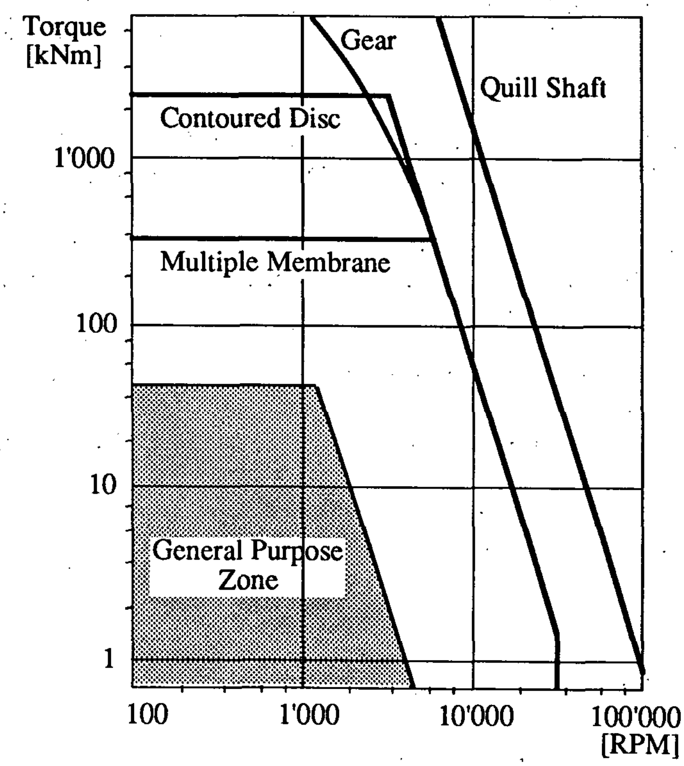

Figure 4: Approximate performance boundaries of high speed couplings

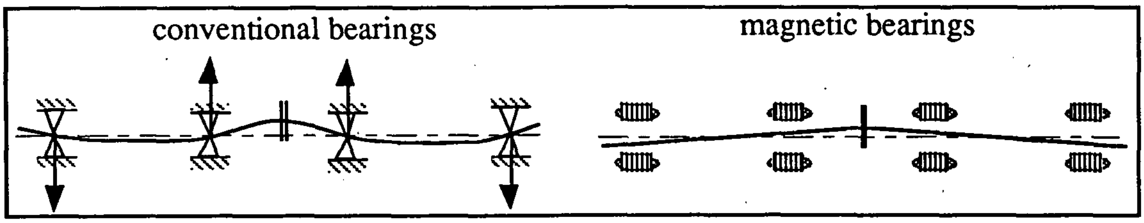

Figure 5: Magnetic bearings tolerate misalignment of the rotor shafts

In the case of conventional bearings, axial and angular freedom of the coupling are essential to reduce the bearing forces due to misalignment of the rotor shafts. By using *magnetic bearings* these forces can be avoided by a force-free rotation around the main inertia axis. Figure 5 shows the effect of a shaft misalignment for conventional and magnetic bearings if a rigid coupling is used. The conventional bearings will impose the rotation axis by flexing the structure, and thus leading to bearing forces. Magnetic bearings allow the structure to rotate around it's main inertia axis without force reaction, thus rigid couplings may be a suitable alternative for flexible couplings.

High speed rotating machinery often operates close to bending critical speeds of the system and therefore the effects of couplings upon the vibrations has to be considered for the coupling selection [4]. If gyroscopic and non-conservative effects are neglected, small lateral displacements from the equilibrium position of two coupled rotors can be described by the following equation:

$$\begin{bmatrix} M_1 & 0 \\ 0 & M_2 \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} D_1 & 0 \\ 0 & D_2 \end{bmatrix} \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} + \begin{bmatrix} K_1 & 0 \\ 0 & K_2 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} L_1 & 0 \\ 0 & L_2 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} F_1 \\ F_2 \end{bmatrix} \tag{1}$$

In the above equation index 1 (resp. 2) stands for rotor 1 (resp. rotor 2), x is the vector containing the lateral rotor displacements and angular rotations, M is the mass matrix, D is the inner damping matrix, K is the stiffness matrix, L is the feedback for the AMB control and F are the forces generated by the unbalance. The dark fields in equation (1) indicate where the rotordynamics is affected after coupling. The two main effects of couplings on the dynamic behaviour of the coupled system are:

- the eigenfrequencies and eigenmodes are affected by the mass, the damping and the stiffness of the coupling
- the vibration levels are influenced by the amount of coupling unbalance

In equation (1) it is assumed that the controller feedback L remains the same after coupling. Therefore the question arises: *"Should the controller layout be based on the coupled system?"*.

FEEDBACK INTERPRETATION FOR THE COUPLING

This section provides a feedback interpretation for the coupling of rotors. Consider two rotors, say rotor 1 and 2, which are to be coupled mechanically by some coupling device. We will show that the coupling leads to a *feedback interconnection* of two transfer matrices

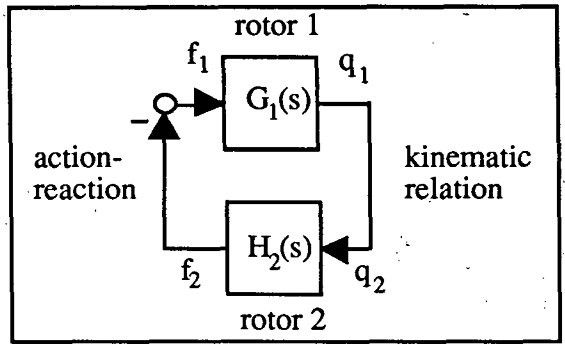

Figure 6: Coupling leads to feedback interconnection

$G_1(s)$, $H_2(s)$, where $G_1(s)$ (resp. $H_2(s)$) describes the dynamic compliance (resp. stiffness) of rotor 1 (resp. rotor 2). Let us start with the simple case of a fictitious coupling device which provides a completely *rigid* coupling and has negligible mass. Regarding kinematics, this means that the displacement vectors q_1, q_2 (each consisting of linear and angular deviations) are equal on each side of the coupling device, i.e. $q_1 = q_2$. Furthermore, the law of action and reaction tells us that both bending torque and transversal force on each side of the coupling are opposite, i.e. $f_1 = - f_2$. Thus, the coupling of two rotors amounts to the feedback setup shown in Figure 6. This feedback interpretation also holds in the case of real coupling devices with non-zero mass and finite stiffness. In the latter case the characteristics of the coupling device can be included in either $G_1(s)$ or $H_2(s)$. Therefore, Figure 6 remains the same even for real coupling devices.

STABILITY OF COUPLED ROTORS

Since we know that the coupling of two arbitrary mass-spring-damper systems (where all masses, springs and dampers assume *positive* values) is stable again, one might guess that the coupling of two arbitrarily stable systems must generally remain stable. But in fact it is widely known in the control community that this is generally *not* the case. Consider the mechanical example shown in Figure 7. A straightforward computation of the characteristic polynomials of each of the uncoupled subsystems reveals that both uncoupled subsystems S_1 and S_2 are asymptotically stable. However the coupling, that is a rigid connection between points A and A', causes the overall system to become unstable. Of course, the reason for this is the presence of one *negative* damper in subsystem S_1. Subsystem S_1 is asymptotically stable, but it is *not* passive in the energy based sense of system theory[1] [1]. The above example illustrates that, without an additional assumption regarding passivity, the feedback interconnection of two stable systems will *not* necessarily preserve stability[2]. One might criticize the above example of being purely academic because seemingly negative dampers do not exist. But actually there are many known effects in rotor dynamics that behave similar to a negative damper, e.g. inner damping in rotating machinery, following

[1] i.e. the transfer function of subsystem S_1 is not *positive real*, meaning that the real part is not positive for all frequencies.

[2] the set of stable transfer functions merely fulfills the mathematical axioms of a *ring*. Generally, the inversion leads out of this set since stable transfer functions may still have zeros in the right half plane. Note that every feedback interconnection leads to linear fractional transforms (LFT) and that the inversion is part of an LFT.

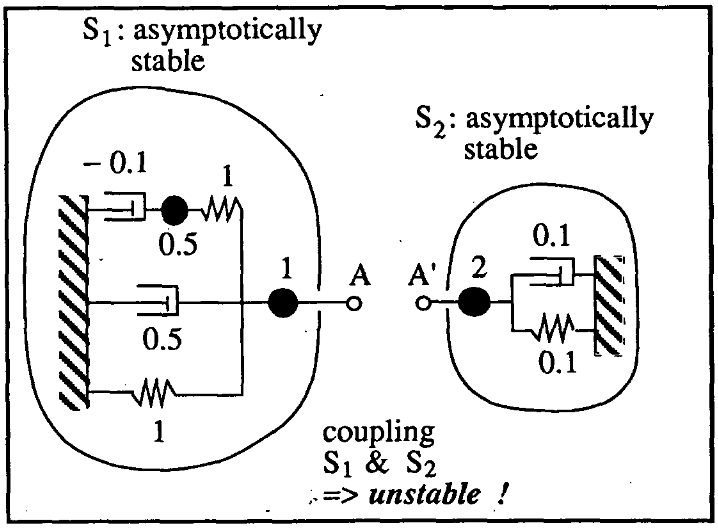

Figure 7: An example of unstable couplings

loads etc. In the case of AMB suspended rotors, *active* control may cause non-passivity of the rotor system. Even if the coupling of two stable rotors does *not* necessarily imply instability, severe "performance degradation" can be expected. Think of a "nice stability region" D in the s-plane where D keeps some distance from the imaginary axis. Then, we may repeat the above reasoning, i.e. the eigenvalues of the coupled rotor system may leave D even if the eigenvalues of each uncoupled subrotor lie in D.

COUPLING OF A MAGNETICALLY SUPPORTED DRIVE SYSTEM

In this section the above considerations on couplings are illustrated for the coupling of a magnetically supported drive system consisting of an electric drive and a turbocompressor, as shown in Figure 8. Let's assume that the drive and the compressor were designed independently of each other to allow a satisfactory operation for the uncoupled system at a speed of 30'000 RPM (500 Hz). A finite element analysis of the rotors (see Figure 8) shows that the eigenfrequencies are far away enough from the operational speed and that the sensor

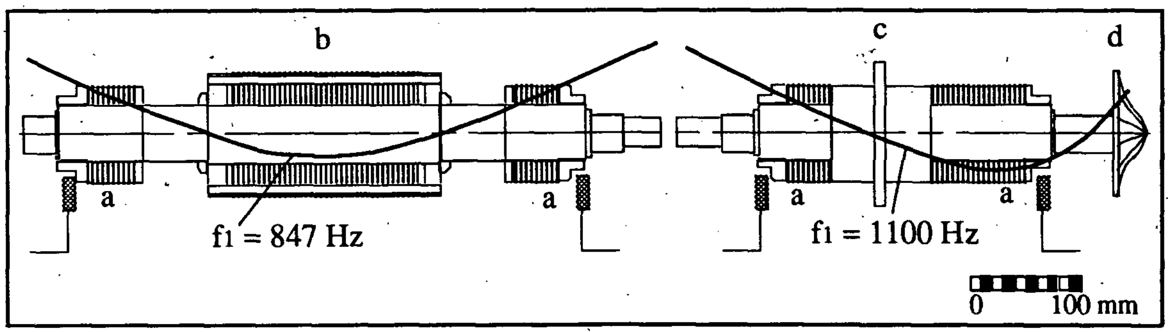

Figure 8: Electric Drive and Compressor with its first bending eigenmodes
a) radial AMB with sensor c) thrust disk for axial AMB
b) electric motor unit d) impeller

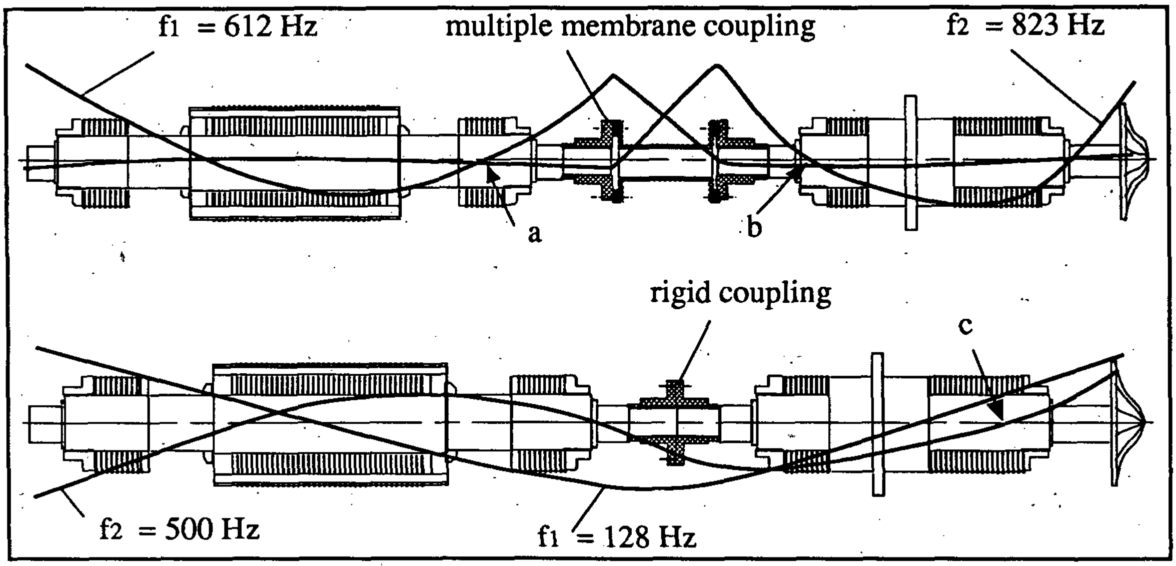

Figure 9: Influence of a flexible and a rigid coupling on the overall system dynamics

and the bearing planes are situated in such a way that the first bending eigenmodes are good observable and controllable. In the following the influence of different coupling types on the system dynamics is shown. Only the first two bending eigenmodes of the coupled system are considered and damping effects are neglected.

By using a flexible coupling the dynamic behaviour of the rotors remains rather independent of each other, see Figure 9 above. In comparison with the uncoupled system the bending eigenfrequencies have clearly decreased and the nodes of the eigenmodes have changed their position in an unfavorable manner. The first bending eigenmode affecting the drive is hardly controllable because a node (a) is situated in the middle of the right bearing plane. The second eigenmode affecting the compressor is hardly observable because a node (b) is situated in the left sensor plane. The flexible coupling influences the dynamic behaviour of the coupled system mainly by its weight. In fact, if the coupling mass were negligible, the elastic behaviour of the rotors would remain unchanged.

If a rigid coupling is used, the dynamic behaviour of the drive and the compressor cannot be dealt with separately any longer, see Figure 9 below. Now the first two bending eigenfrequencies are located within the operating frequency-range and the second bending eigenfrequency is unfortunately just equal to the operating speed of 500 Hz. Additionally, a node (c) of the latter eigenmode is situated between the strong magnetic bearing and the sensor plane near the impeller, which obviously makes an operation for this example impossible.

In the above example the system configuration has been chosen in unfavorable manner, and therefore by using different rotors or couplings the system dynamics may be improved. However, it was clearly shown that even if a flexible coupling is used, the coupled system dynamics can change unfavorably.

CONCLUSIONS

In this paper several practical and control theoretical aspects for the coupling of magnetically supported rotors were shown, leading to the following conclusions:

- Today's couplings are highly developed and couplings for high speeds and powers are available.
- For AMB applications stiff couplings may be an alternative to flexible couplings, since misalignment is tolerable by means of *"force-free"* rotation.
- Mass and stiffness of couplings can considerably affect the dynamic behaviour of the coupled system components.
- The vibration levels of the coupled system are influenced by the amount of coupling unbalance, and therefore the couplings as well as the coupled rotor shafts should be well balanced.

The authors' main conclusion on the coupling of AMB-supported high speed rotors may be summarized as follows: *The controller layout should be based on the coupled system and the effects of couplings may already be considered in the design of the uncoupled system. However, a modular system-design (i.e. a controller layout and design based on the uncoupled system) might be feasible, but a severe degradation of the system dynamics may occur.*

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