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Control of a Rubbing Rotor Using an Active Auxiliary Bearing

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Abstract

A new approach to control a rubbing rotor applying an active auxiliary bearing is presented. A two-phase control strategy has been developed, which guarantees a smooth transition from free rotor motion to the state of full annular rub, in case of an operation state which causes rotor rubbing. The designed control system is able of both, avoiding high impact forces and stabilizing the rotor during rubbing. For the experimental verification of the designed control system a test rig has been realized. The experimental results show a drastic reduction of the contact forces, as well as a reduction of the rotor deflection.

Keywords: Actire auxiliary bearing; Rubbing rotor; Two-phase control

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1. Introduction

The control of rubbing phenomena in rotating machinery is of practical interest for the prevention of structural damage since there is the possibility of serious failures up to complete destruction of the system if the impact forces are not taken under control. In rotating machinery, increased efficiency is often achieved by tightening operation clearances, and so, if a machine is not operating under normal conditions, the stationary and rotating elements are in danger of coming into contact. Rotor-to-stator rubs may occur under several operation states, such as a sudden increase of the unbalance, an extreme outside excitation or the passing of a resonant frequency during the speeding-up or coastdown. Auxiliary bearings or back up bearings are used in rotor systems to prevent direct contact between rotor and casing when the rotor response is too large. Thus conventional auxiliary bearings limit large response amplitudes.

In this paper a control concept using an active auxiliary bearing is proposed which reduces the rubbing severity. The control force is applied by the auxiliary bearing, which is attached to the foundation via two unidirectional magnetic actuators, Fig. 1.

The advantages of this concept are the following: If the rotor system runs in the usual way, the active auxiliary bearing does not take effect, so the original design of the rotor system can be kept to largest extent unchanged. Additionally, the auxiliary bearing does not only limit a too large response amplitude of the rotor and prevents the rotor/blades and the casing/seals from direct contact, but also effectively reduces the rubbing severity and especially avoids the occurrence of destructive rubbing instabilities.The capability of existing auxiliary bearings, i.e. as safety bearings in active magnetic bearing systems or as run-

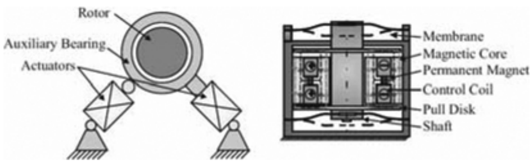

Fig. 1. Rotor with active auxiliary bearing (left), Electromagnetic actuator (right).

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through resonance support, can be well extended from this concept by introducing active control.

Several control structures have been proposed in the literature for the control of rubbing rotors. Feedback linearization was used in (Al-Hiddabi, et al.) in a drill string to suppress the lateral and torsional vibration of the system, dynamic inversion was used to track a desired bit speed. Numerical simulations showed the elimination of the torsional vibration and reduction in the lateral vibration. The problem of a Jeffcott rotor control under rubbing was treated in (Jiang, J. et al., 2003). The author developed a controller based on stability analysis of the synchronous annular rub to reduce the intensity of the rubbing. Feedback linearization and direct feedback were used to minimize the transient contact forces of the rotor stator system (Ulbrich, H., 1999) by activating the controller before the occurrence of the first contact.

In (Chavez, Al. et al., 2005) sliding control and cross coupled feedback in a rotor system driven by a power limited motor was used to reduce the impact forces and to decrease the lateral and torsional vibrations. On the other hand, numerous interesting research publications about impacts in robotics have been published in the last decade (Brogliato, B. 1999 ; Brogliato, B. et al., 1997 ; Tornambè, A. et al., 2003), but the problem of controlling impacts is still open, due to sudden changes of the equations of motion when there is a switch from a state of no contact to a condition of contact.

2. Feedback control concept

A control strategy has been developed, which guarantees a smooth transition from free rotor motion to the state of "full annular rub." The feedback controller assures a permanent contact with low contact forces. To keep the principal purpose in mind, the control scheme also has to limit the rotor amplitude, as a passive auxiliary bearing does.

In a first approach a cascade control is used, see Fig. 2, with $\mathbf{q}_a$ the measured position of the auxiliary bearing and $\mathbf{q}_{ad}$ the desired one, $\mathbf{q}_r$ are the coordinates of the rotor, $\mathbf{f}_c$ are the desired control forces, $\mathbf{i}_c$ the control current to drive the actuators and p a boolean variable to activate and deactivate the control system. The activation routine observes the movement of the rotor. In case of a sudden arising unbalance the controller can be activated automatically if the rotor response is too large. The trajectory computation provides the target trajectory for the auxiliary bearing and should assure a smooth transition from free rotor motion to a state of permanent contact, which can be formulated as:

$$\mathbf{q}_{ad} = \underset{(\ddot{\mathbf{q}}, \dot{\mathbf{q}}, \mathbf{q})}{arg\ min} \begin{cases} \mathbf{g}_N \\ \dot{\mathbf{g}}_N \end{cases}, \tag{1}$$

with arg min $f(x)$ = value of x that minimizes $f(x)$, $\mathbf{q}$ the generalized coordinates of the rotor and the auxiliary bearing, $\mathbf{g}_N$ the distance between the contact points and qad the target trajectory for the auxiliary bearing. The relative velocity of the contact point in tangential direction $\dot{\mathbf{g}}_\mathrm{T}$ will not be taken in consideration in Eq. (1) because a non-sliding contact would cause a backward whirl, which is not wanted.

In order to define the essential geometrical variables a cross section of the auxiliary bearing and the rotor is shown schematically in Fig. 3. The origin of the coordinate system coincides with the center of the undeformed rotor, $\mathbf{r}_r$ is the position vector to the center of the deformed rotor (in the cross section) and $\mathbf{r}_a$ to the center of the auxiliary bearing. The air gap in the auxiliary bearing is called $\boldsymbol{\delta}_0$ and $\mathbf{r}_N$ represents the vector from the center of the auxiliary bearing to the center of the rotor. Additionally we introduce the polar angles φ_a and φ_r of the vectors $\mathbf{r}_a$ and $\mathbf{r}_r$. The desired position of the auxiliary

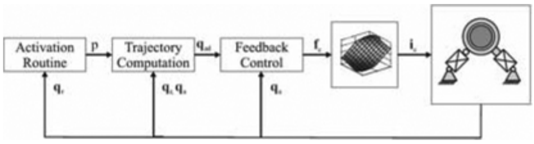

Fig. 2. Cascade control.

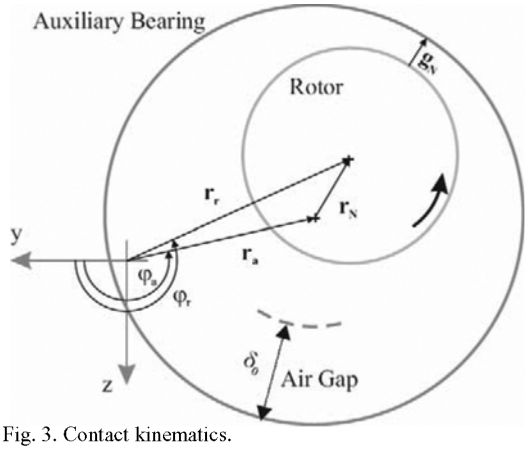

Fig. 3. Contact kinematics.

bearing is chosen in a way that the contact point coincides with the point of the surface of the rotor which is farthest from the origin of coordinate system, fig. 3. This means that

$$\varphi_{a\ desired} = \varphi_r \tag{2}$$

Furthermore the desired polar radius $|\mathbf{r}_{N\ desired}|$ is needed to determine the desired position of the auxiliary bearing according the equation

$$\mathbf{r}_{a\ desired} = \begin{pmatrix} \varphi_a \\ |\mathbf{r}_a| \end{pmatrix} = \begin{pmatrix} \varphi_a \\ |\mathbf{r}_a| - |\mathbf{r}_{N\ desired}| \end{pmatrix} \tag{3}$$

In the first control phase (free rotor motion), which means $|\mathbf{r}_N| \leq \delta_0$, the target position of the auxiliary bearing follows

$$\begin{aligned} /\mathbf{r}_{N\ desired}/ &= /\mathbf{r}_N/ - \int (/\dot{\mathbf{r}}_N/ . v_{pmax})dt \\ &= /\mathbf{r}_N// - \int (/\dot{\mathbf{r}}_N/ - A\, e^{a/\mathbf{r}_t/})dt \end{aligned} \tag{4}$$

where v_{pmax} is the maximum relative velocity of the contact points. The constant factors A and α are chosen in such a way that first impact is kept small but also that the amplitude of the rotor does not exceed δ_0, to meet the principal purpose of an auxiliary bearing.

In case of contact, which means $|\mathbf{r}_N| = \delta_0$, the desired distance $/\mathbf{r}_{N\ desired}/$ follows:

$$/\mathbf{r}_{N\ desired}/ = \delta_0 + \frac{1}{K_P} f_{perm} \tag{5}$$

with f_{perm} the desired contact force during the permanent contact and $\frac{1}{K_P}$ a conversion coefficient. Note that it is necessary to choose a desired contact force which is large enough to ensure a permanent contact despite of elements of uncertainty of the measurement and control system.

To linearize the strong nonlinearity of the electromagnetic actuators in the whole operating range a nonlinear feed forward mechanism is applied. The nonlinear compensation current field $i_c(f_c, q_a)$ cancels the nonlinear effects by taking the influence of the displacement on the actuator force into account. Thus, the system will get a linear force characteristic to the control current independent to the displacement. Accordingly it is possible to use a PID controller with variable coefficients to control the electromagnetic actuators, which follows the equation:

$$\mathbf{f}_c = K_P\,\mathbf{e} + K_D \frac{d\mathbf{e}}{dt} + K_I \int \mathbf{e}\, dt \tag{6}$$

where $\mathbf{f}_c$ is the control force, K_P, K_D and K_I are coefficients and $\mathbf{e}$ the tracking error:

$$\mathbf{e} = \mathbf{q}_{ad} - \mathbf{q}_a \tag{7}$$

The coefficients of the control law are chosen for the linearized system for both operation states (separated or in contact) and were optimized using numerical simulations. Note that in case of contact the integral part of the controller is set to zero.

3. Test rig

The system consists of an elastic rotor with one disc, which is mounted on two isotropic ball bearings. The auxiliary bearing is attached to the foundation via two unidirectional magnetic actuators. The air gap between the rotor and the auxiliary bearing is 0. 3 mm. A magnetic bearing is used to create realistic excitations to the rotor such as a sudden arising unbalance, so that the rotor comes into contact with the auxiliary bearing. A direct current disc-servomotor allows a rotational speed up to 3500 rpm.

Several sensors are used to gather information, as it is shown in Fig. 4. There are two eddy current displacement sensors to measure the position of the rotor beside the auxiliary bearing. The same sensors are installed inside the actuators. Load washers in each actuator are measuring the actuator forces, from which the contact forces are determined indirectly. With the help of accelerometers the load of the bearings are recorded.

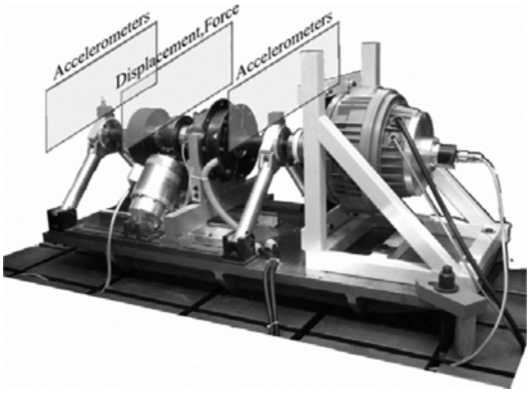

Fig. 4. Test rig.

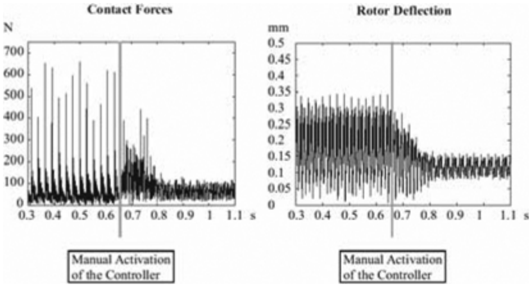

Fig. 5. Experimental results.

4. Experiments

In Fig. 5 the contact forces (left) and the rotor deflection (right) during rotor rubbing are shown. The rotor is running at a constant subcritical speed and the electromagnetic bearing applies an unbalance which causes rubbing between the rotor and the auxiliary bearing. For this experiment a manual activation mode of the feedback controller is chosen. Up to the time of 0. 65 s the controller is deactivated and there is a state of partial rub with multiple impacts and contact forces up to 700 N. At time 0. 65 s the controller is activated and within a tenth of a second the rotor is stabilized and the contact forces are reduced up to 80 per cent. At the same time the rotor deflection is significantly reduced as it is shown on the right of Fig. 5.

5. Conclusions

In this paper a new concept is proposed to reduce the negative effects of rubbing phenomena in rotating machinery. Therefore a control concept for an active auxiliary bearing has been developed. The auxiliary bearing is attached to the foundation via two powerful electromagnetic actuators, which have been developed at the Institute of Applied Mechanics, TU Munich. A two phase feedback control, which has been designed using numerical simulation, assures a smooth transition from the state of free rotor motion to a permanent contact in the desired state "full annular rub." The applied control concept leads to a drastic reduction of the impact forces and a stabilization of the rotor system.

The future research work will focus on the development of an adaptive control which is able to compensate measurement errors. In addition, there are still uncertainties relating to the friction coefficient, which has a great influence on the process of rubbing. The measurements will also be expanded on supercritical rotors. Furthermore there is the possibility to use the control system to take influence on torsional vibrations.

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