ROTOR/AUXILIARY BEARING DYNAMIC CONTACT MODES IN MAGNETIC BEARING SYSTEMS

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ABSTRACT
Magnetic bearings systems usually incorporate auxiliary bearings so that rotor/stator contact is prevented. The rotor must then make contact with the auxiliary bearings if any clearance boundaries are exceeded. It is then possible for a rotor with a given unbalance condition to exhibit either contact free levitation or to become trapped in various contact modes. For passively mounted auxiliary bearings, the contact modes are determined by the system design parameters. However, it is now feasible to mount the auxiliary bearings in an active system using piezoelectric actuators. If closed loop control is applied, the mounting (e.g. stiffness/damping) characteristics may be varied on-line. The potential also exists to apply open-loop actuation to the auxiliary bearing to make the clearance domain variable. For example, appropriate synchronous translation of a circular auxiliary bearing may cause it to appear as having a different radius or to be elliptical to the rotor. The question that needs to be addressed is whether this is useful in the destabilisation of contact modes so that contact free levitation may be restored. This paper will investigate whether this is a feasible option.

INTRODUCTION
The benefits of equipping rotating machinery with magnetic bearings are well documented [1] in terms of high speed operation, low friction/wear and active rotor control. Auxiliary bearings are also included in parallel with the magnetic bearings to ensure that rotor-stator clearances are maintained. The auxiliary bearings are not required under normal conditions of operation; however, they must be present to deal with faults, conditions when magnetic bearing load capacity is exceeded, and other abnormal circumstances. In principle, the magnetic bearing components are protected, but an auxiliary bearing must be properly designed to cope with the expected non-linear dynamics. These may cause high levels of mechanical and thermal stresses over short or long duration periods. Hence the effective operational life of an auxiliary bearing is very dependent on the expected rotor dynamic responses that may arise from the contact dynamics.

The analysis of the non-linear dynamics arising from rotor/stator contact has received significant attention over a number of years [2–7]. When a rotor interacts with an auxiliary bearing, the rotor dynamics can involve a number of characteristic responses. These depend on the bearing design, support stiffness and damping, material characteristics at the contact interface, the rotor unbalance condition, and also the initial conditions prior to contact. The response types may be:

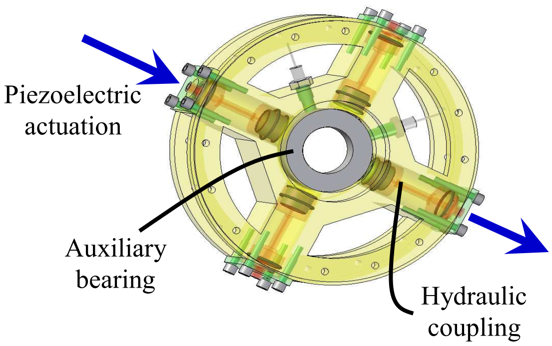
(a) persistent in a number of rub/bounce type contact modes

or

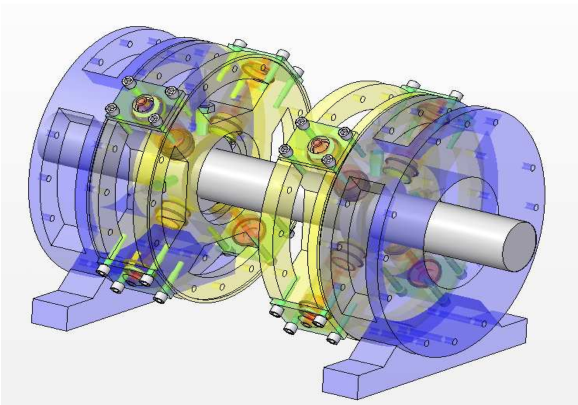
(b) transient leading to normal contact free rotor motion

In this paper it will be taken that the condition of the rotor dynamics is such that it is possible for the rotor to operate in contact free motion. In other words, response type (b) is possible. Response type (a) may then occur if the system is affected by a disturbance input. For example, a high acceleration base excitation could cause the magnetic bearing load capacity to be exceeded. Another condition might include a temporary fault. Control functionality is assumed to remain active; hence the case of total power/control loss leading to rotor drop is excluded from consideration.

If the magnetic bearing control action is not adaptive, transition from type (b) to type (a) responses may be possible. The explanation for this phenomenon is that the plant dynamics undergo a non-linear change from a non-contact condition to a contact condition. In fact, a stable closed loop system without contact may become either a stable or an unstable closed loop system when contact dynamics are present. The problem then is that the magnetic bearing control action may be ineffective in returning the rotor to a contact free state. This leads one to ask whether another form of control



(a) Auxiliary bearing actuation section



(b) Integration into rotor/magnetic bearing system

FIGURE 1: Active auxiliary bearing system design

action could be useful to return the rotor to an operational state.

CONCEPT – ACTIVE AUXILIARY BEARING

The concept of moving an auxiliary bearing has been considered by Ulbrich *et al.* [8]. The objective was to minimise the contact forces when loss of levitation occurs. Electromagnetic actuators capable of strokes up to 1 mm were used to move the auxiliary bearing and effective control was demonstrated.

In this paper, the concept is considered for system in which magnetic bearing control is fully functional. If the rotor should make contact with the auxiliary bearing, any auxiliary bearing actuation would be intended to destabilise a contact mode. The rotor would then “fall back” to a contact free state. The requirement is for actuation strokes that are a fraction of the radial clearance associated with a typical auxiliary bearing. Hence a system based on piezoelectric actuators has been conceived. Research is being undertaken at the University of Bath, where the system has been designed [9] and is being manufactured. Figure 1 shows the designed components. The features of the system include:

(a) Piezoelectric actuators that are resiliently mounted on the outer casing of the rotor/magnetic bearing system. Inertial loads acting on the system may then be reduced at the piezoelectric actuators.
(b) A closed circuit hydraulic coupling that allows force to be transmitted from the piezoelectric actuators to the auxiliary bearing. The coupling has been designed to allow high force (~ 10 kN) transmission normal to the bearing, but low resistance to tangential movement of the bearing.
(c) The ability to operate the piezoelectric actuators in a push-pull sense to reduce bearing distortional stresses. The hydraulic couplings may be linked to allow actuation along a single axis using either one or two actuators. The use of two orthogonal axes allows full movement in the radial plane.
(d) A controller to operate the actuators. Inputs may include auxiliary bearing displacement, rotor displacement and hydraulic coupling pressure.

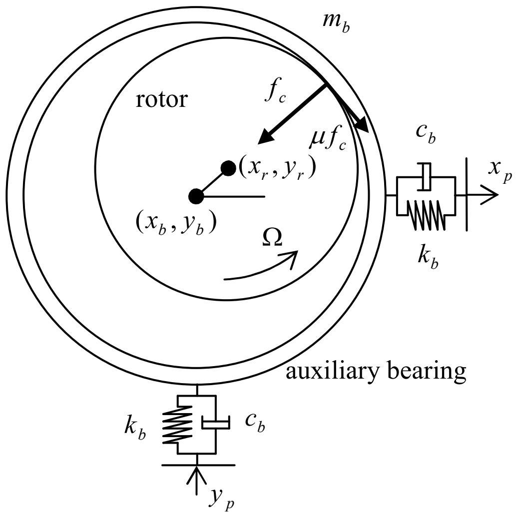


FIGURE 2: Active auxiliary bearing model

At this stage in the research, the auxiliary bearing controller design is still to be formulated until a complete understanding of the system dynamics is ascertained. Towards this end, it is instructive to compare rotor dynamic responses with and without auxiliary bearing actuation.

SIMPLIFIED DYNAMIC MODEL

Consider a nominally rigid auxiliary bearing of mass m_b. The bearing is actuated through radial spring (k_b) and damper (c_b) coefficients that are representative of the hydraulic couplings. The piezoelectric actuator system effectively displaces the coupling by (x_p, y_p), hence generates forces that are applied to the bearing. Figure 2 shows the condition when rotor/auxiliary bearing contact occurs. The rotor displacements (x_r, y_r) are also dependent on the auxiliary bearing deflections (x_b, y_b). For the purpose of this paper, the rotor will be represented as a simple disc of mass m_r, which may be unbalanced with eccentricity e.

When contact with the rotor occurs ($f_c > 0$), the equations of motion for the auxiliary bearing are embedded in the complex form

$$\begin{aligned} m_b(\ddot{x}_b + i\ddot{y}_b) + c_b(\dot{x}_b + i\dot{y}_b) + k_b(x_b + iy_b) \\ = f_{bx} + if_{by} + (1 + i\mu) f_c e^{i\theta} \end{aligned} \tag{1}$$

where $f_{bx} + if_{by} = k_b(x_p + iy_p) + c_b(\dot{x}_p + i\dot{y}_p)$, f_c is the contact force, μ is the effective coefficient of friction between the rotor and auxiliary bearing, and $\theta = \tan^{-1}((y_r - y_b)/(x_r - x_b))$. The corresponding equations of motion for the rotor are

$$\begin{aligned} m_r(\ddot{x}_r + i\ddot{y}_r) = f_{mx} + if_{my} \\ -(1 + i\mu) f_c e^{i\theta} + m_r e\Omega^2 e^{i(\Omega t + \phi)} \end{aligned} \tag{2}$$

Here, the magnetic bearing is considered to be fully functional with force components (f_{mx}, f_{my}), Ω is the rotor speed and ϕ is a phase angle associated with the unbalance.

As a test case, an eight-pole radial magnetic bearing in a differential driving mode is used. The magnetic bearing forces, including a simplified saturation limit, are written in the form

$$f_{mx,y} = -\frac{k_m}{k_f} \tanh k_f \left(\frac{(V_0 + V_c)^2}{(c_m + z)^2} - \frac{(V_0 - V_c)^2}{(c_m - z)^2} \right) \tag{3}$$

where $z = x_r$ or y_r. Here, the effective magnetic bearing clearance, c_m, is less than the nominal auxiliary bearing clearance, c_r. The coefficients k_m and k_f determine the maximum force capacity and incorporates the voltage gain of the system. The magnetic bearing controller is assumed to use PID feedback of rotor displacements to set the voltage V_c. Since static forces acting on the rotor will be compensated for by the integral action, these are not included in the equations of motion.

A Hertzian contact model is assumed for simulation purposes. The rotor will be in contact with the auxiliary bearing if

$$\delta r = \sqrt{(x_r - x_b)^2 + (y_r - y_b)^2} \geq c_r \tag{4}$$

According to Hertzian theory, the length of the contact zone ($2a$) is related to the normal contact force by

$$a = \sqrt{4R^* f_c / \pi E^* l_b} \tag{5}$$

where effective radius and elasticity parameters are included in the expressions

$$\left. \begin{aligned} R^* = R_b(R_b - c_r)/c_r \\ 1/E^* = (1 - \nu_r^2)/E_r + (1 - \nu_b^2)/E_b \end{aligned} \right\} \tag{6}$$

The non-linear contact force/deflection relation is then contained within

$$\delta r - c_r = \frac{f_c}{\pi E^* l_b} \left(\frac{2}{3} + \ln \frac{4R_r R_b}{a^2} \right) \tag{7}$$

An inverse numerical procedure may be used to determine the force/deflection relation explicitly for use in equations (1) and (2).

The time variation of (x_r, y_r) will show the rotor orbit motion in a fixed frame of reference. It is also useful to view rotor motion in a forward synchronously rotating frame, since the rotor synchronous response may be related to a stationary unbalance vector. The rotor orbit in the rotating frame will be seen through coordinates (u_r, v_r), where

$$u_r + iv_r = (x_r + iy_r) e^{-i\Omega t} \tag{8}$$

AUXILIARY BEARING MOTIONS

Open-Loop Considerations

There are a number of options available for consideration. These are related in the first instance to a non-contacting condition in which the rotor unbalance is driving a circular forward whirl orbit:

$$x_r + iy_r = a_r e^{i\Omega t}, \quad a_r < c_r \tag{9}$$

The following cases are not exhaustive, but give partial insight into the range of actions that may be taken

Case 0: No action. In this case the residual gap before any contact is $c_r - a_r$.

Case 1: The auxiliary bearing is made to execute a synchronous forward circular whirl orbit:

$$x_b + iy_b = \pm a_b e^{i\Omega t} \tag{10}$$

The residual gap before contact is then $c_r - a_r \pm a_b$.

Case 2: The auxiliary bearing is made to execute an elliptical synchronous orbit ($x_b^2/a_b^2 + y_b^2/b_b^2 = 1$):

$$x_b + iy_b = \tfrac{1}{2}(a_b + b_b)e^{i\Omega t} + \tfrac{1}{2}(a_b - b_b)e^{-i\Omega t} \quad (11)$$

The residual gap before contact is then $c_r - a_r + a_b$ in the x – direction and $c_r - a_r + b_b$ in the y – direction.

Case 3: The circular whirl of Case 1 is made non-synchronous:

$$x_b + iy_b = \pm a_b e^{i(\Omega-\omega)t} \quad (12)$$

Here, the residual gap before contact varies dynamically over the range $c_r - a_r - a_b$ to $c_r - a_r + a_b$.

Case 4: The elliptical orbit of Case 2 is made non-synchronous:

$$x_b + iy_b = \tfrac{1}{2}(a_b + b_b)e^{i(\Omega-\omega)t} + \tfrac{1}{2}(a_b - b_b)e^{-i(\Omega-\omega)t} \quad (13)$$

In this case the auxiliary bearing orbit "tumbles" relative to the rotor orbit.

Closed Loop Considerations

The system will have the capability to measure auxiliary bearing and rotor radial displacements together with the internal hydraulic coupling pressure, p. The displacements of the piezoelectric actuators in a feedback strategy may then be written in the form

$$\begin{bmatrix} x_p \\ y_p \end{bmatrix} = \mathbf{F}\left(\begin{bmatrix} x_b \\ y_b \end{bmatrix}, \begin{bmatrix} \dot{x}_b \\ \dot{y}_b \end{bmatrix}, \begin{bmatrix} x_r \\ y_r \end{bmatrix}, \begin{bmatrix} \dot{x}_r \\ \dot{y}_r \end{bmatrix}, p \right) \quad (14)$$

There are a range of options available to configure the characteristics of the system. For example:

(i) Use of the auxiliary bearing states in a proportional feedback sense to allow variations of the stiffness (k_b) and damping (c_b) characteristics to be made.
(ii) Use of the difference between the auxiliary bearing and rotor states to allow the contact stresses to be controlled.
(iii) Use of the rotor states to anticipate contact and manoeuvre the auxiliary bearing in advance of a contact event.
(iv) Use of p to control the forces applied to the auxiliary bearing.

This list is not exhaustive, for example, other optimal multi-input, multi-output control strategies may be devised. The simulation examples presented in this paper will investigate some of the potential benefits.

EXAMPLE SIMULATIONS

The rotor/auxiliary/magnetic bearing data appropriate to the system shown in Figure 1 are given in Table 1.

TABLE 1: System data

$m_r = 4.25$ kg, $R_r = 15.0$ mm, $e = 0.1$ mm, $\phi = 0$
$m_b = 0.18$ kg, $R_b = 15.4$ mm, $l_b = 10.0$ mm
$c_b = 2500$ Ns/m, $k_b = 6\times10^7$ N/m
$E_r = E_b = 2.1\times10^{11}$ N/m^2, $\nu_r = \nu_b = 0.3$
$c_r = 0.5$ mm, $c_m = 0.8$ mm
$\mu = 0.05$, $\Omega = 1000$ rad/s

The magnetic bearing can deliver a maximum radial force of 2500 N with a current gain of 241 N/A. The PD controller gains were chosen to give a levitated rotor natural frequency of around 70 Hz (440 rad/s) with a damping ratio of 0.1. The integral gain was sufficiently low so as to have negligible influence at this frequency.

It is supposed that the system is affected by a sufficiently large disturbance that leads to rotor/auxiliary bearing contact. The initial conditions of the auxiliary bearing are that it is centralised in a state of rest. Initially, the rotor just touches the auxiliary bearing and has an initial velocity V at a variable angle of incidence β. Hence it is appropriate to set

$$\begin{gathered} x_b(0) + iy_b(0) = \dot{x}_b(0) + i\dot{y}_b(0) = 0 \\ x_r(0) + iy_r(0) = -c_r, \quad \dot{x}_r(0) + i\dot{y}_r(0) = Ve^{i\beta} \end{gathered} \quad (15)$$

An initial velocity of $V = 0.3$ m/s could be interpreted as being appropriate to a 10g base acceleration input.

Temporary Contact (V = 0.3 m/s, β = –135 deg.)

Figure 3 shows the rotor response when the angle of incidence is β = –135 deg. The rotor is seen to experience a single bounce-like contact, induced by the initial conditions, but the following rotor dynamics are within the clearance circle. The return to contact free levitation is seen in the stationary reference frame (Figure 3(a)), with a nearly circular final synchronous forward whirl orbit. When viewed in a synchronously rotating reference frame (Figure 3(b)), the final steady state orbit is represented around a single point that lags the unbalance force vector by nearly 180 deg.

Persistent Contact (V = 0.3 m/s, β = 135 deg.)

The response of Figure 3 is changed dramatically when the angle of incidence is on the other side of the normal (β = 135 deg.). Figure 4(a) shows that the rotor endures a number of bounce-like contacts before settling into a forward synchronous whirl rub mode. The rotor motion also has a significant higher frequency harmonic component at 4Ω = 4000 rad/s. This is excited by non-isotropy in the magnetic bearing field as the rotor passes the four pole pairs during each rotation. It is also noted that the natural frequency of the combined rotor/auxiliary bearing mass oscillating on the spring, k_b

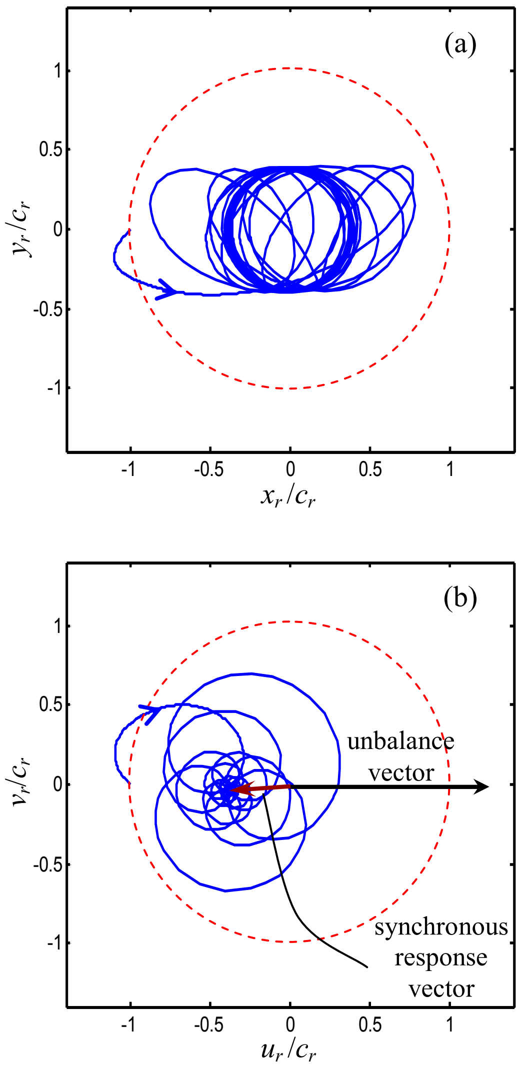


FIGURE 3: Rotor response after contact with auxiliary bearing (- - - -). (a) View in a stationary frame; (b) View in a synchronously rotating frame. Initially, V = 0.3 m/s, β = -135 deg.

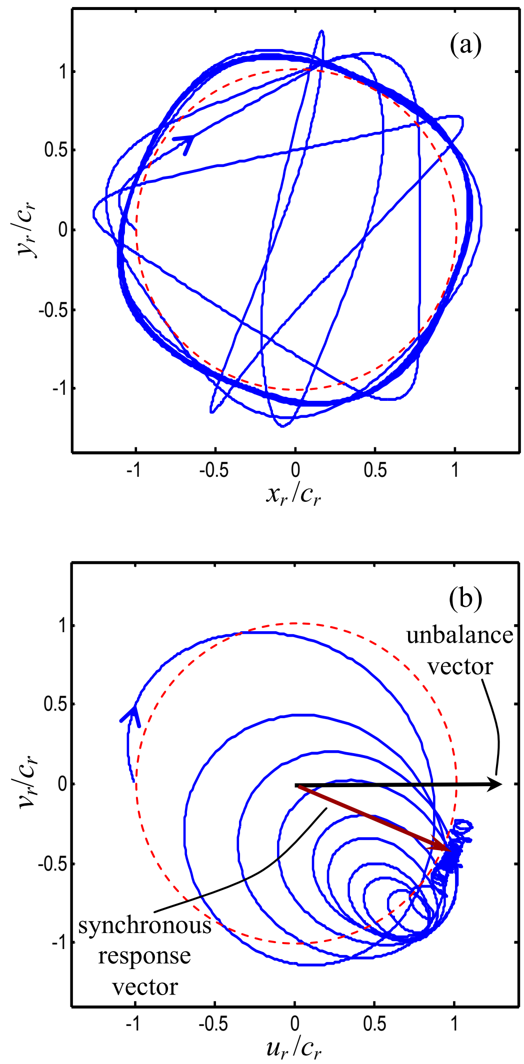


FIGURE 4: Rotor response after contact with auxiliary bearing (- - - -). (a) View in a stationary frame; (b) View in a synchronously rotating frame. Initially, V = 0.3 m/s, β = 135 deg.

is around 3700 rad/s. When viewed in the synchronously rotating frame, it is seen how the rub orbit synchronous vector moves around a single point that lags the unbalance force vector by around 30 deg. (Figure 4(b)). In other words the contact dynamics induce nearly a 150 deg. phase shift in the synchronous response compared with the no-contact dynamics.

Figure 5(a) shows the auxiliary bearing deflection induced by the contact dynamics. Figure 5(b) shows the contact force variation with time. Note that the contact force levels reach 6000 N before settling to the mean synchronous component at around 1750 N. This is consistent with the rotor centre of mass orbiting with a radius that is slightly larger than the radial clearance c_r. These contact force levels are driven by the inherent unbalance force at amplitude 425 N.

The results of Figures 3 – 5 provide insight into the dynamic behaviour of the system with and without contact. It would be undesirable to allow contact to persist when a normal operating condition without contact is possible. It would appear that one option is to use active control to prevent the rotor synchronous response vector from undergoing such a large phase change due to contact. If this cannot be prevented due to control lag, the control action should reverse the phase change to destabilise the established contact mode.

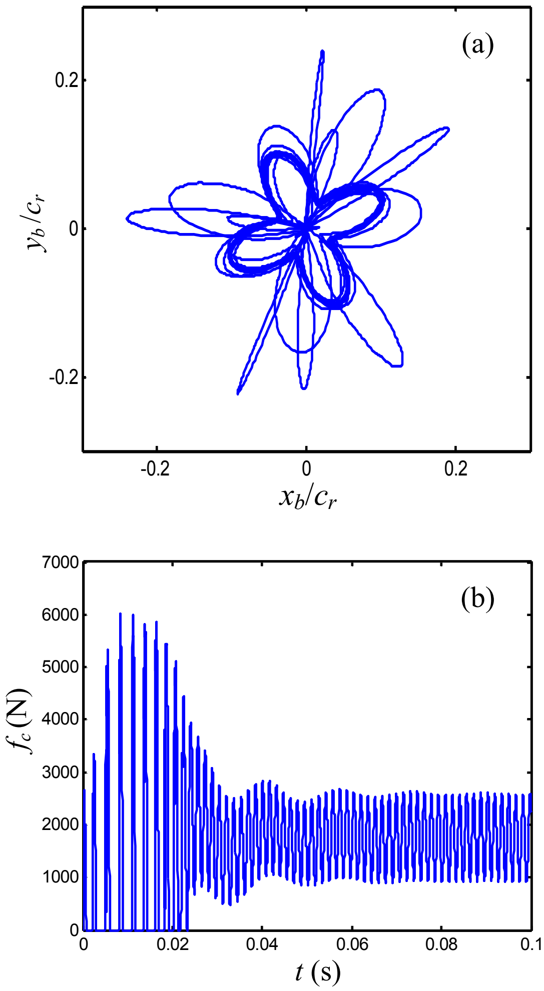


FIGURE 5: Auxiliary bearing deflection (a) and contact force (b) showing persistent contact. Initially, $V = 0.3$ m/s, $\beta = 135$ deg.

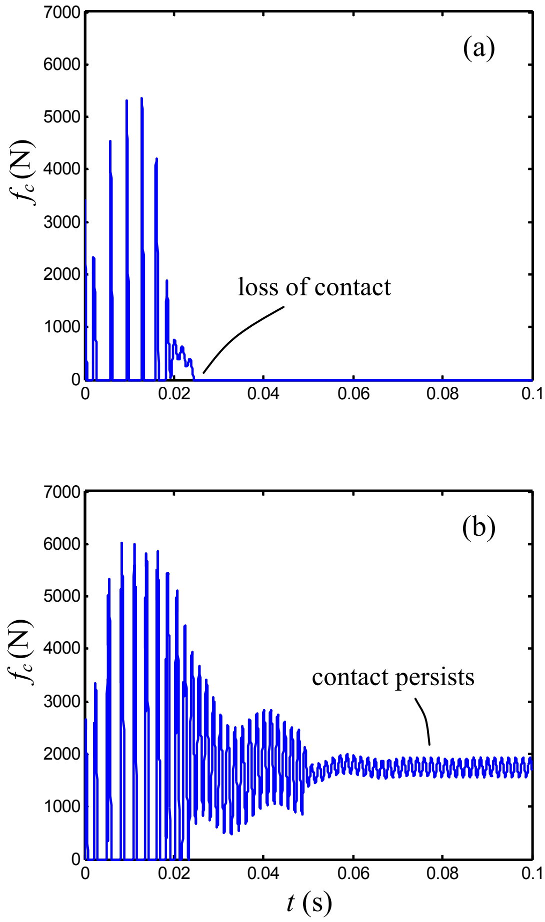


FIGURE 6: Contact force variation. Bearing damping c_b changed from 2500 Ns/m to 17500 Ns/m at (a) $t = 0$ s; (b) $t = 0.05$ s. Initially, $V = 0.3$ m/s, $\beta = 135$ deg.

Active Change of Mounting Characteristics

The purpose of making changes to the spring/damper characteristics associated with the auxiliary bearing is to influence the system contact dynamics so as to cause a return to contact free levitation. In principle, the proposed active system would allow on-line changes. However, the changes may induce mixed results due to the non-linear nature of the dynamics. Moreover, it was found that a significant change in the effective damping from $c_b = 2500$ Ns/m to 17500 Ns/m was required to cause appreciable differences.

Suppose this change is made before the initial contact with $\beta = 135$ deg. Figure 6(a) shows that the persistent contact of Figure 5 is prevented from building up. However, suppose now that change in the damping characteristic is made only after the forward rub mode has become established. Figure 6(b) shows that the forward rub contact mode still persists, though the 4Ω harmonic component is attenuated. Hence, one must not expect a single change in mounting characteristic to always be successful in returning the rotor to contact free levitation.

It should also be recognised that the suggested increase in damper rate is significant and may exceed the capability of the piezoelectric actuators. It may also result in increased energy dissipation rates by the actuators, for which heating effects may be another limitation.

Open-Loop Options

It is seen from Figure 5(a) that the auxiliary bearing experiences deflections due to contact up to 25% of the radial clearance, particularly when the transient bounce-like rotor motion occurs. This raises the question of whether the piezoelectric actuators could be used to

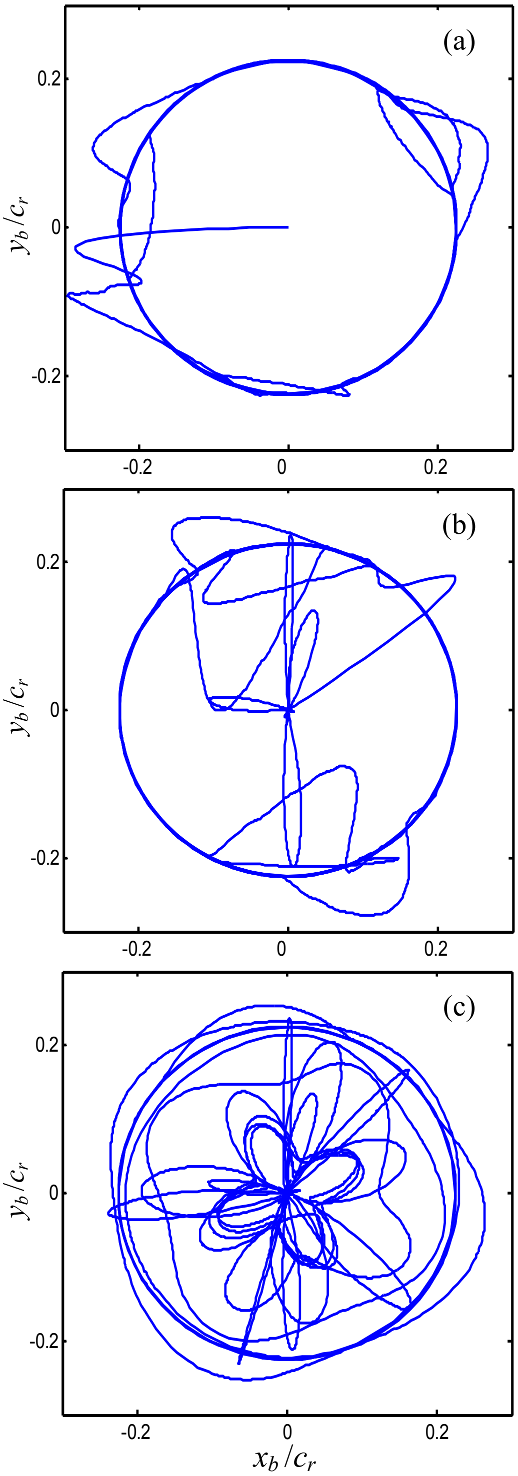


FIGURE 7: Auxiliary bearing deflection with additional actuated forward synchronous motion initiated at (a) t = 0 s; (b) t = 0.01 s; (c) t = 0.05 s. Initial contact at t = 0 s with V = 0.3 m/s, β = 135 deg.

apply appropriately phased circular orbit motion (see equation (10)) on the auxiliary bearing that would lead to loss of contact.

The system data of Table 1 are assumed and if no auxiliary bearing control is applied, the results would be the same as in Figures 4 and 5. Simulations were then undertaken in which open-loop actuator forces capable of inducing forward synchronous whirl of the auxiliary bearing at just over 20% of the radial clearance were applied. Figure 7 shows the effect of initiating the auxiliary bearing actuation with delays of 0 s, 0.01 s, and 0.05 s. The varying levels of transient contact induced activity before the circular motion becomes established without further contact are seen. The procedure is always successful in establishing return of the rotor to contact free levitation. This is evidenced by Figure 8, which shows how the contact forces are attenuated to zero when the circular orbit motion becomes established. Of course, this would need some advanced knowledge of the unbalance phase, which could be estimated by on-line testing under normal operating conditions. Also further study is required to establish whether the procedure is robust under all possible contact modes, steady or transient, particularly those involving flexural rotor displacements.

CONCLUSIONS

This paper has examined how contact modes may become established in a functioning rotor/magnetic/auxiliary bearing system. A system design for actuating the auxiliary bearing has been introduced. The following conclusions are noted:

1. A nominally stable rigid rotor under magnetic bearing levitation may be driven into persistent auxiliary bearing contact, depending of the initial conditions associated with the disturbance imposed.
2. Closed loop changes to the auxiliary bearing characteristics may or may not be successful in recovering contact free levitation. For the system considered, increased damping did not cause the forward synchronous rub mode to be destabilised.
3. Open-loop imposition of auxiliary bearing orbit motions may have potential in destabilising contact modes. For the system considered, this procedure was successful in all applications. In practice, the motion could be applied then stopped when contact free conditions are restored.

The potential benefits of actuating auxiliary bearings have been shown in a simulation study. A proposed system is under construction and the intention is to test the procedures in an experimental programme. The results will be reported in due course.

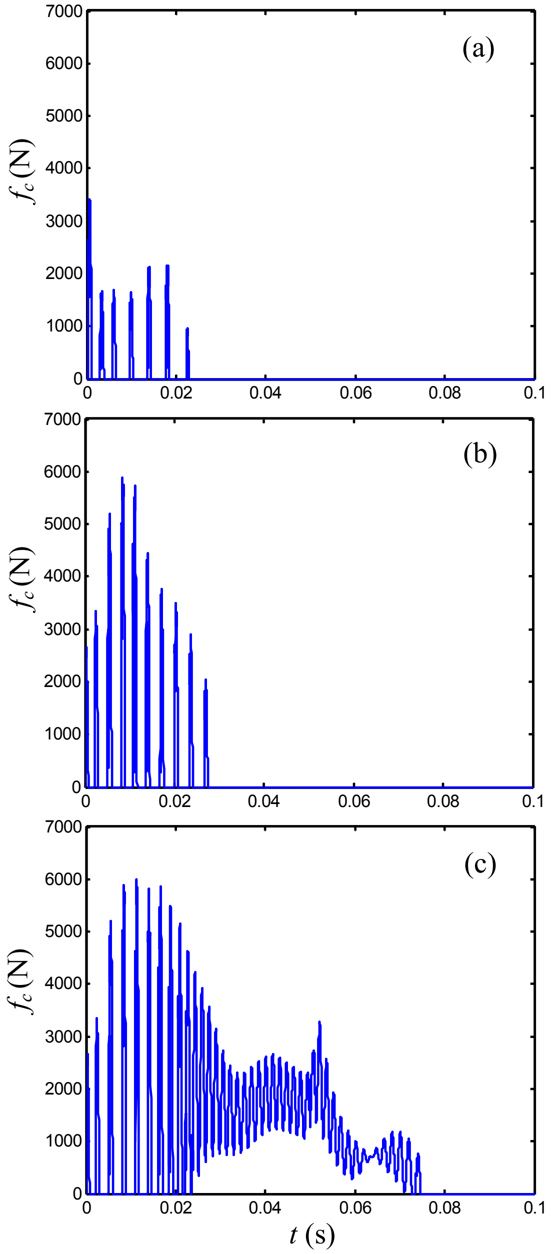


FIGURE 8: Contact force variation under actuated forward synchronous motion of auxiliary bearing initiated at (a) t = 0 s; (b) t = 0.01 s; (c) t = 0.05 s. Initial contact at t = 0 s with V = 0.3 m/s, β = 135 deg.

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