Electroviscous Journal Bearing for Active Control of Vibration of an Out-of-Balance Rotor

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1 Introduction

Two journal bearings support a rotor on oil films. Vibration of the rotor, caused by mass imbalance, is reduced by adjusting the bearing forces through voltage induced changes in the viscosity of an electroviscous (otherwise known as electro-rheological or ER) oil. Such oils are modelled as Bingham, rather than Newtonian, fluids when a strong enough electric field is applied. The effect of applying control forces at the bearings using this idea is investigated.

2 Rotor Model

The rotor used for the simulation is shown in Figure 1. It is based on that used by Burrows and Sahinkaya [2] with a less extreme mass imbalance distribution:

$$\begin{array}{c}[(4.0,0^o)\ \ (3.3,73^o)\ \ (9.6,196^o)\\ (2.9,118^o)\ \ (11.9,7^o)\ \ (2.3,266^o)\\ (9.3,303^o)\ \ (1.6,116^o)\ \ (2.7,160^o)]\end{array}$$

where the pairs represent mass (g) at the periphery and angle for the nine stations. The displacements in the x and y directions at station i are denoted by x_i and y_i respectively for i running from 1 up to 9. The rotor bearing system is modelled (Burrows and Sahinkaya [2]) by:

$$M\ddot{q} + C_e\dot{q} + (K_s + K_e)q = Bf \tag{1}$$

where

$$q = \begin{bmatrix} x_1 \\ \vdots \\ x_9 \\ y_1 \\ \vdots \\ y_9 \end{bmatrix} \text{ and } f = \begin{bmatrix} \cos\omega t \\ \sin\omega t \end{bmatrix}; \tag{2}$$

ω represents the speed of rotation in radian/s. M and K_s are the mass and structural stiffness matrices of dimensions 18×18. The bearing forces are accounted for by introducing damping and stiffness matrices C_e and K_e respectively of dimensions 18×18. The out-of-balance forces are obtained by introducing the matrix B of dimensions 18×2. The structures of these matrices are given below as they have implications for the design of the controller.

$$M = \begin{bmatrix} M' & 0 \\ 0 & M' \end{bmatrix}, \tag{3}$$

where $M' = diag(m_1, \ldots, m_9)$, and

$$K_s = \begin{bmatrix} K'_s & 0 \\ 0 & K'_s \end{bmatrix}, \tag{4}$$

where K'_s is a 9×9 tri-diagonal matrix (the only non-zero elements are of the form $k_{i,i-1}$, $k_{i,i}$ or $k_{i,i+1}$). The only non-zero elements of C_e and K_e are in positions $(1,1)$, $(1,10)$, $(9,9)$, $(9,18)$, $(10,1)$, $(10,10)$, $(18,9)$ and $(18,18)$. Their values depend on the speed of rotation (Holmes [3]) and on the rheological properties of the ER lubricant. Since the control is to be implemented by modifying the rheology of the lubricant, the values of the elements of K_e (and, in principle C_e) will be altered, according to the control action desired, in the simulation. The same rotor was studied earlier (Metcalfe and Burdess [5]) where interest centered on the closed loop control strategy with the assumption that the control forces would be applied at a separate station near the centre of the rotor by, for example, an electromagnetic actuator.

3 Rheology

An ER fluid is modelled as a Newtonian fluid in the absence of an electric field and as a Bingham fluid when an electric field is applied. The relation between shear stress, τ, and shear rate, $\dot{\gamma}$ is

$$\tau = \eta\,\dot{\gamma} \tag{5}$$

for the Newtonian fluid and

$$\tau = \tau_y + \eta_p\,\dot{\gamma} \tag{6}$$

for the Bingham fluid when it flows, with no flow occurring for $\tau < \tau_p$; η_p is the plastic viscosity and τ_y the yield stress of the Bingham fluid. A simple model

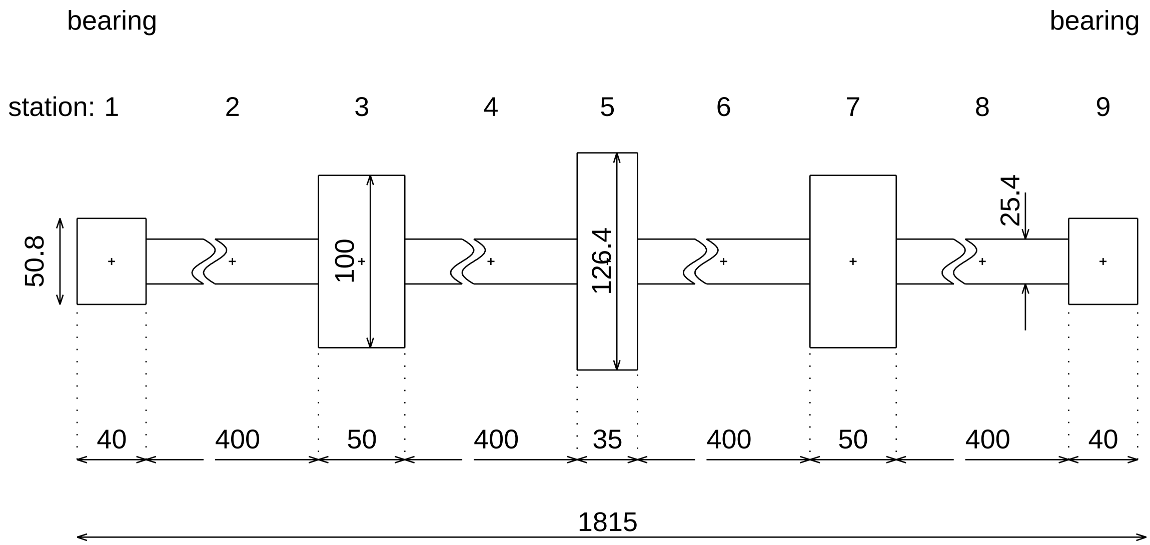

FIGURE 1. Rotor-bearing model.

of the effect of the applied electric field is to assume that the viscosity is unaltered, $\eta_p = \eta$, and that the yield stress is proportional to the applied voltage gradient, V above some threshold voltage V_0.

$$\tau_y = \alpha(V - V_0) \tag{7}$$

For a typical ER fluid (Stangroom [6]) we might expect $V_0 = 1\ \mathrm{kV\ mm^{-1}}$ and, with τ measured in kPa, the constant of proportionality would have a value around $\alpha = 2$. In our preliminary investigation of application to control we assume that the change in bearing force is proportional to the applied electric field.

Calculation of the bearing force for a journal bearing where the voltage is applied across a segment of the gap has been undertaken to estimate the magnitude of the effect that is achievable. The configuration considered (Figure 2) was one where the segment between angular positions ϑ_1 and ϑ_2 has a slot cut across the bearing so that the gap where the field is applied is large enough to avoid "bridging" by the particles (which are suspended in the ER fluid) when the electric field is turned on. A bearing with mean gap 0.5mm and slot depth 0.5mm has been considered (so that the mean gap where the ER fluid is activated is 1mm).

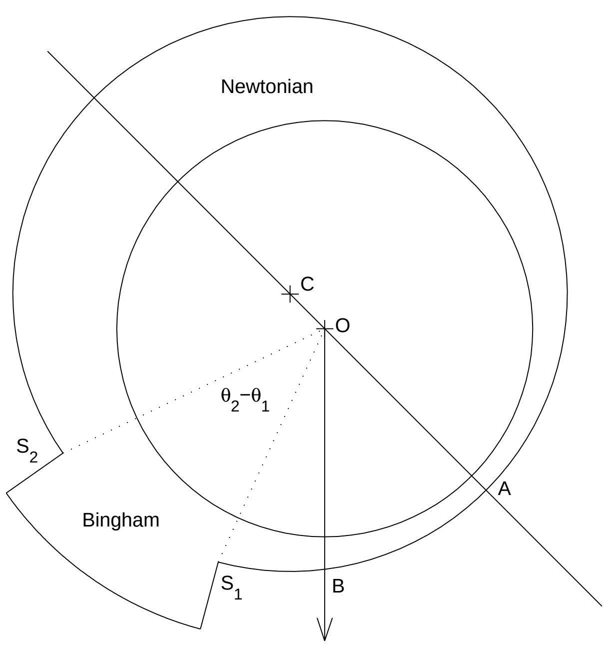

FIGURE 2. Schematic of the journal bearing.

Traditional lubrication theory (for a long bearing) is used for the flow around the gap except across the slot. For the slot, the Bingham fluid equation is used, which means that shear flow of the lubricant takes place in layers near the solid surfaces of bearing and shaft while a plug of fluid occupies the centre of the gap (where the shear stress is less than the yield stress of the fluid at the relevant voltage gradient). For simplicity the transition between the Newtonian and Bingham fluid regions is ignored. Experience with ER fluids suggests that this will be reasonable, since the changes in rheology with electric field happen very rapidly. A typical value achieved for the load enhancement (the percentage increase in bearing force above that for a lubricant which is Newtonian everywhere) is 9% for $\vartheta_2 - \vartheta_1 = \pi/6$, $\omega = 200\ \mathrm{s^{-1}}$, $\eta = 0.05\ \mathrm{Pas}$ and $\tau_y = 8.5\ \mathrm{kPa}$.

4 Control Strategies

We have made a preliminary investigation into the feasibility of controlling out-of-balance vibration of the rotor through changes in the rheology of the oil film that might be induced by an electric field. The control force is taken to be that for a Newtonian lubricant multiplied by a load enhancement factor, c, depending on the applied voltage.

Two simple control strategies (bang-bang control and feedback control) are presented here. The behaviour of the system is calculated with a FORTRAN program to solve the differential equations governing

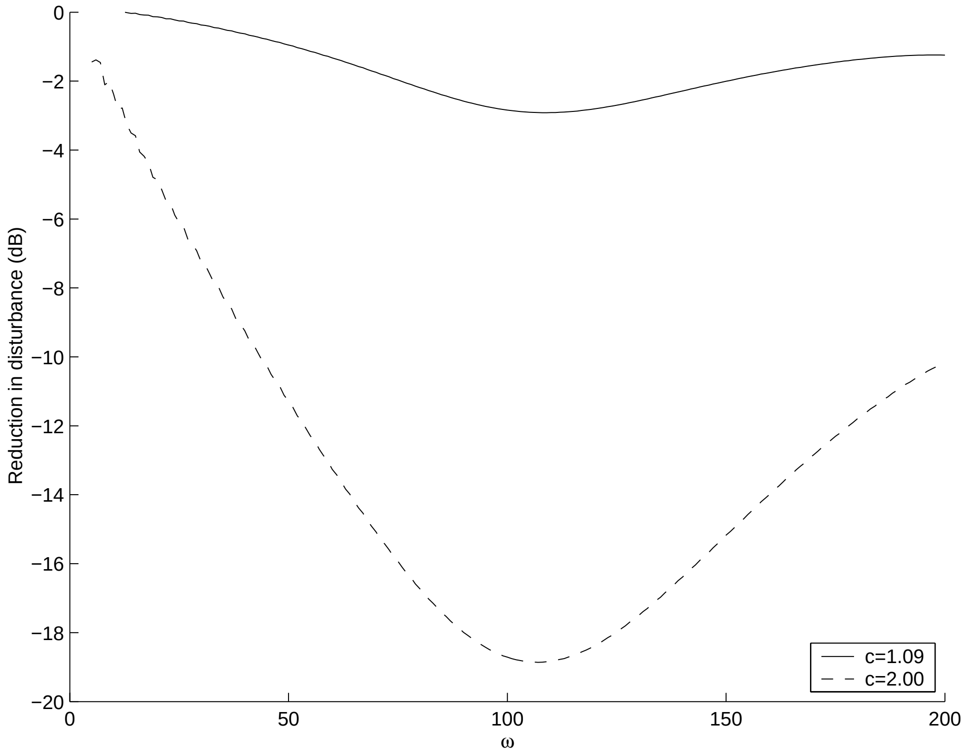

FIGURE 3. Bang-bang controller performance.

the system, using a Runge-Kutta-Merson routine from the NAG FORTRAN library. The simulation is continued over a period of two seconds, which is long enough for the transient response to become negligible. The frequency response of the system enters the simulation both directly through out-of-balance term Bf in Equation (1) and through changes to the bearing damping and stiffness matrices with ω, which are calculated following Holmes [3].

4.1 Bang-bang Control We assume that application of the electric field increases all the equivalent damping and stiffness elements by the constant factor c. The control strategy is to apply the electric field if vibration exceeds some predetermined critical level. When the vibration is reduced below the critical level the electric field can be turned off. The critical level could be, for example, a root mean square displacement of 15% of the minimum gap in steady running (the gap at point A in Figure 2). The simulation model predicts substantial reductions in vibration which we show in Figure 3. The graphs show the effect of values $c = 2$ and $c = 1.09$ over the frequency range $\omega = 10$ to $\omega = 200$. The reduction in disturbance is plotted as $20\log_{10}(\mathrm{SSQ}(\omega)/\mathrm{SSQ}_0(\omega))$, where

$$\mathrm{SSQ}(\omega) = \sum_{j=1}^{j=9}\left(x_j^2 + y_j^2\right) \tag{8}$$

and $\mathrm{SSQ}_0(\omega)$ is the corresponding value for no control action ($c = 1$).

The value $c = 2$ is chosen to show the effect of doubling the bearing forces; the mean square vibration is reduced to a fraction of about 0.11 of its level without the field applied at a rotation frequency of around 100 radian/s. The value $c = 1.09$ is chosen to show that an appreciable effect can be obtained even with the 9% load enhancement which has been conservatively predicted (see Section 3 above). The mean square vibration is reduced in this case to about 0.71 (29% reduction). The first critical rotation frequency with no control action is 240 rad/s; with $c = 2$ this exceeds 400 rad/s.

4.2 Feedback Control We assume that the electric field can be adjusted so that the increase in the damping at each bearing, C_e, is proportional to the sum of the absolute values of the displacements from the dynamic equilibrium of the rotor at that bearing. Thus C_e becomes $C_e(1+k(|x_1|+|y_1|))$ for the journal bearing at station 1 and $C_e(1+k(|x_9|+|y_9|))$ for the journal bearing at station 9. The vibration reduction achieved through gains $k = 10$, $k = 100$ and $k = 1000$ are shown in Figure 4. The gain of 10 raises the first critical frequency from 240 to slightly above 300.

5 Discussion

The simulation results reported here are sufficiently encouraging for the proposed vibration controller to be considered for practical applications. The magnitude of force obtainable using the electroviscous bearing has been shown to be adequate to provide the required control action. Detailed modelling of the lubrication flow has been undertaken and results of this will be reported elsewhere.

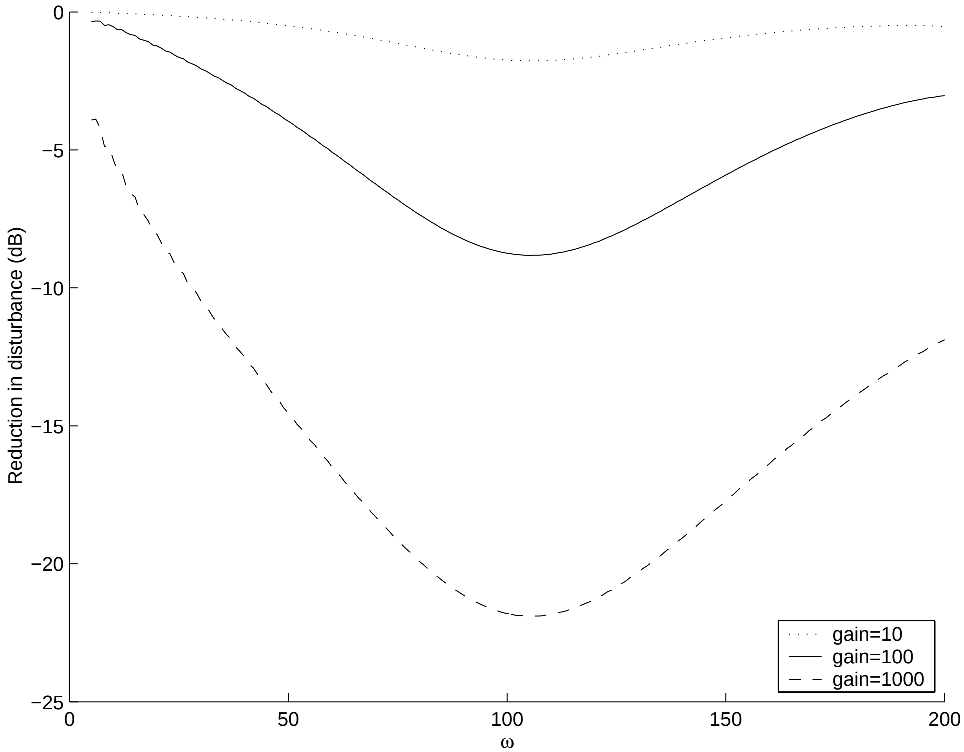

FIGURE 4. Feedback controller performance.

The model of the rotor does not incorporate any damping other than that supplied by the oil film, and the control strategies are stable over the range of gains and rotation speeds considered. They provide impressive damping. Control action also has the benefit of raising the first critical rotation frequency from 240 rad/s to above 400 rad/s for the bang-bang control. This would be of value in applications requiring variable speed rotation which occasionally exceeds the first critical rotation frequency.

The next phase of the research project is to establish the precise relationship between the applied electric field and the elements of the matrices C_e and K_e, and to develop appropriate control algorithms.

It would be feasible to use displacements from other stations in the feedback control as additional inputs to the controller. However, care must be taken to ensure that this does not lead to instability through spillover (Balas [1]).

There is scope for applying optimal control and H_∞ control. An alternative is to use adaptive algorithms (see, for example, Metcalfe and Burdess [4]).

Whilst further theoretical and simulation studies are needed, the bearing design will clearly be crucial. It is the authors' opinion that theoretical studies are not a substitute for practical experimentation, and that testing the algorithm on a real rotor is an important part of further work.

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