BIAS CURRENT OPTIMISATION AND FUZZY CONTROLLERS FOR MAGNETIC BEARINGS IN TURBO MOLECULAR PUMPS

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ABSTRACT

Magnetic bearings are commonly used in turbo-molecular pumps because of their suitability for high speed and contamination free operations. A five-axis turbo molecular pump with two radial and one axial magnetic bearing is considered in this paper. The effect of the bias current on the energy consumption for both unidirectional and differential control current strategies is studied. It is shown that optimising the bias current according to operational conditions is beneficial. A differential current control gives better efficiency compared with a unidirectional approach when the bias current is variable. The optimum bias current is a function of the desired bearing stiffness and the rotor vibration amplitudes. The desired bearing stiffness is a function of rotor vibration levels or transmitted forces. A fuzzy logic controller is set up to resolve the conflicting control requirements. A simulation involving a non-linear magnetic bearing model is used to verify the suggested control and optimisation techniques.

INTRODUCTION

A turbo molecular pump (TMP) is a gas transfer pump, which operates by the interaction of a molecule and a moving surface [1]. The pump performance depends on the rotor speed, and it is ideally suited to the use of magnetic bearings, which provide contactless and friction free support of the rotor. These bearings are suitable for high speed operation and do not require lubrication, which may contaminate the product; they can operate in hostile environments at high temperatures and in a vacuum [2]. The use of magnetic bearings makes these pumps attractive for hydrocarbon free operation, pumping of corrosive gases and low vibration applications, such as required in nuclear physics, surface science experiments, mass spectrometry, semiconductor fabrication and electron beam lithography. However, magnetic bearings are inherently unstable [3], and require feedback in order to function properly. They can act as fully controllable actuators and sensors, and usually incorporate a processor. Therefore these bearings can be used for sensing, control and identification.

The opposing poles generate a combined bi-directional force. Although this force is a nonlinear function of the control current and the clearance, linear approximations are usually made for small rotor vibrations around the central static operating point. A bias current is supplied to the poles to improve linearity and to satisfy the operational requirements of the power amplifier. However, this entails energy consumption even if no force generation is required. The first part of the paper studies the effect of the bias current on the energy consumption and dynamic performance of the magnetic bearing as a function of the orbit size. Two strategies for achieving the demanded control force are compared [4].

An optimum combination of the bias current and feedback parameters can be selected to achieve a desired bearing stiffness to minimise energy consumption. One of the requirements of the controller is that rotor-stator contact should be avoided. Magnetic bearings have limited force capacity, and retainer (or auxiliary) bearings are usually incorporated to protect damage to the magnetic bearing poles and laminations. If a rotor touches an auxiliary bearing, it enters a highly nonlinear dynamic state, the controller is ineffective, and the system has to be shut down [5]. This condition may arise as a result of a fault in the magnetic bearing drive circuit, a loss of power, or internal and external transient effects. Therefore, the first requirement is to prevent rotor vibrations exceeding a certain limit. Another requirement may be to minimise the transmitted force, so that the pump works with minimum noise and vibration while minimising the energy consumption.

These conflicting requirements can be addressed by a fuzzy logic controller (FLC) [6]. Hung [7] combined a PID controller with FLC for adjusting the linear controller signal to compensate nonlinear effects. Lieberts [8] combined a PID controller with an adaptive fuzzy controller for non rotating levitation systems. Hong *et al.* [9] developed a robust fuzzy vibration controller to adjust the stiffness properties of bearings subjected to harmonic disturbances. Youzhi Xu and Kenzo Nonami [10] obtained a fuzzy model of a system by using a fuzzy neural network, and designed a sliding mode controller. Because of the computational efficiency requirements for real time applications, a first order Takagi-Sugeno type FLC is used in this analysis. The FLC considers the conflicting requirements and the measured state of the system and selects a suitable effective bearing stiffness. This stiffness is then realised by selecting the optimum combination of the bias current and the proportional feedback gain to minimise energy consumption. A simulation example is provided to demonstrate the effectiveness of the proposed technique.

SYSTEM DESCRIPTION AND MODELLING

The simulated system consists of a magnetically levitated turbo-molecular pump, a backup pump and a dSPACE digital signal processor connected to a PC as shown in Figure 1. A rotary pump is used to provide the required pressure before the TMP is started. The TMP consists of a vertical shaft acted upon by two radial bearings providing four control forces, and one thrust bearing providing lift (Figure 2). There are also retainer bearings incorporated within all three magnetic bearings to prevent rotor contact. The rotor is driven by a brushless DC motor, which is placed between the upper and lower radial magnetic bearings. There are two pairs of inductive sensors along the shaft to measure the displacement of the rotor at two points in orthogonal radial directions, and one sensor at the bottom to measure the vertical position of the rotor in the axial direction. The sensor signals are sampled by a DS2002 ADC and fed into the DS1003 Power PC board (digital signal processor) that performs control of the system. The control signals for the magnetic bearings are applied through pulse width modulation amplifiers via DS2103 DAC. A host PC is connected to the dSPACE's modular hardware for rapid control prototyping (RCP) and data acquisition.

The rotor mass m = 4.5 kg with a transverse moment of inertia of I_T = 0.021kgm^2, and a polar moment of inertia of I_P = 0.011 kgm^2. The axial positions of the two radial bearings measured from the centre of gravity of the rotor are b_1 = 0.023 m and b_2 = –0.075 m for the upper and lower magnetic bearings respectively. The radial sensor positions are a_1 = 0 m and a_2 = –0.056 m from the centre of gravity of the rotor. All three magnetic bearings have the following common data; magnetic permeability of vacuum μ_0 = $4\pi 10^{-7}$ H/m, number of coils per pole N_a = 250, geometric factor γ = cos(360^o/16)=0.92388, air gap clearance g_0 = 0.00028m. The pole areas are 160×10^{-6} m^2 and 80×10^{-6} m^2 for the upper and lower magnetic bearings.

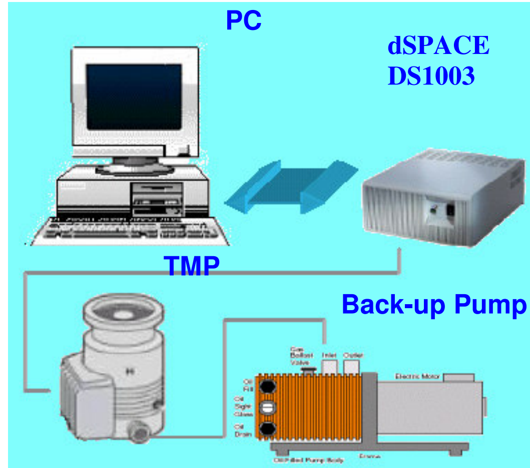

FIGURE 1: Schematic view of the system

An earth fixed XYZ axis system, with the vertical Z axis pointing upwards, is centred at the static position of the centre of gravity of the rotor. The linear motion of the rotor mass centre of gravity is described by x, y and z coordinates, and the angular displacement between the rotor and the XZ and YZ planes are denoted by θ_x and θ_y respectively. Including the gyroscopic effects, the equation of motion of the rotor for a constant rotational speed of Ω can be written as follows:

$$\mathbf{M}\ddot{\mathbf{q}} + \Omega\mathbf{C}\dot{\mathbf{q}} = \mathbf{Q} \tag{1}$$

where $\mathbf{M} = diag(m, m, I_T, I_T, m)$ and

$$\mathbf{C} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & I_P \\ 0 & 0 & 0 & -I_P & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix}$$

$$\mathbf{Q} = [F_x, F_y, M_x, M_y, F_z]^T, \ \mathbf{q} = [x, y, \theta_x, \theta_y, z]^T \tag{2}$$

The vectors **q** and **Q** are the generalised coordinates and the generalised input forces F_x, F_y and F_z, and moments M_x and M_y. The measurement vector $\mathbf{q_O}$ and the positions at magnetic bearing locations $\mathbf{q_M}$ can be obtained as:

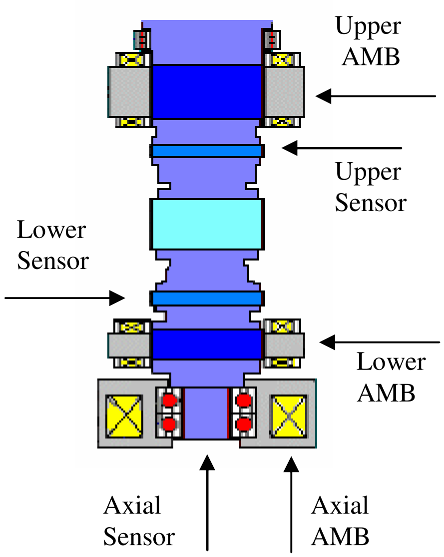

FIGURE 2: Schematic view of the TMP rotor, magnetic bearings and sensors.

$$\mathbf{q_O} = \mathbf{H_O}\,\mathbf{q} = \begin{bmatrix} 1 & 0 & a_1 & 0 & 0 \\ 0 & 1 & 0 & a_1 & 0 \\ 1 & 0 & a_2 & 0 & 0 \\ 0 & 1 & 0 & a_2 & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}\mathbf{q},\quad \mathbf{q_M} = \mathbf{H_M}\,\mathbf{q} \qquad (3)$$

where $\mathbf{H_M}$ has the same structure as $\mathbf{H_O}$ except that parameters a_1 and a_2 are replaced by b_1 and b_2 respectively.

The following equation is used to model the force generated by the two opposing poles of the magnetic bearing along the line of poles [2]

$$f = \frac{\mu_0 A N_a^2}{4}\gamma\left[\frac{I_1^2}{(g_0 - s)^2} - \frac{I_2^2}{(g_0 - s)^2}\right] \qquad (4)$$

where s is the displacement, I_1 and I_2 are the currents in the opposing poles. The forces generated by the magnetic bearings can be converted to the generalised inputs as

$$\mathbf{Q} = \mathbf{H_f}\mathbf{U} = \begin{bmatrix} 1 & 0 & 1 & 0 & 0 \\ 0 & 1 & 0 & 1 & 0 \\ b_1 & 0 & b_2 & 0 & 0 \\ 0 & b_1 & 0 & b_2 & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}\begin{bmatrix} f_{1x} \\ f_{1y} \\ f_{2x} \\ f_{2y} \\ f_{3z} \end{bmatrix} \qquad (5)$$

An unbalance mass of M_e (kgm) positioned at an axial distance of d from the centre of gravity of the rotor and with an angle of φ radians from a reference marker on the shaft will produce the following generalised input vector.

$$Q = M_e\Omega^2 \begin{bmatrix} \sin(\Omega t + \varphi) \\ \cos(\Omega t + \varphi) \\ d\sin(\Omega t + \varphi) \\ d\cos(\Omega t + \varphi) \\ 0 \end{bmatrix} \qquad (6)$$

Equations (1)-(6) define the rotor response at a given rotational speed.

CONTROL CURRENT GENERATION

The current or force demand to the magnetic bearing is given by a control voltage to each magnetic bearing transconductance amplifier. A PID controller is used to generate the control force demand. The integral action is to set the static position of the rotor at the bearing centre, and the derivative action provides damping to improve stability. The most significant component of the controller is the proportional action, which should be large enough to offset the inherent negative stiffness properties of the magnetic bearing. If the control current is proportional to the displacements then

$$I_c = -Ks \qquad (7)$$

where K is the proportional controller gain. This control current is sent to the opposing poles, which operate either in a differential or unidirectional mode.

Differential Mode: The control current I_c is superimposed on the bias current I_b such that the control current is applied to the opposing pole with an opposite sign.

$$\begin{aligned} I_1 &= I_b + I_c = I_b - Ks \\ I_2 &= I_b - I_c = I_b + Ks \end{aligned} \qquad (8)$$

Inserting equation (8) into (4) and linearising force expression about the static position of $s = 0$ gives the following equivalent bearing stiffness coefficient

$$K_{eq} = -\left.\frac{\partial f}{\partial s}\right|_{s=0} = \frac{4\alpha}{g_0^3} I_b (K g_0 - I_b) \qquad (9)$$

Unidirectional Mode: Only the pole in the direction of the force receives the control current. This allows for

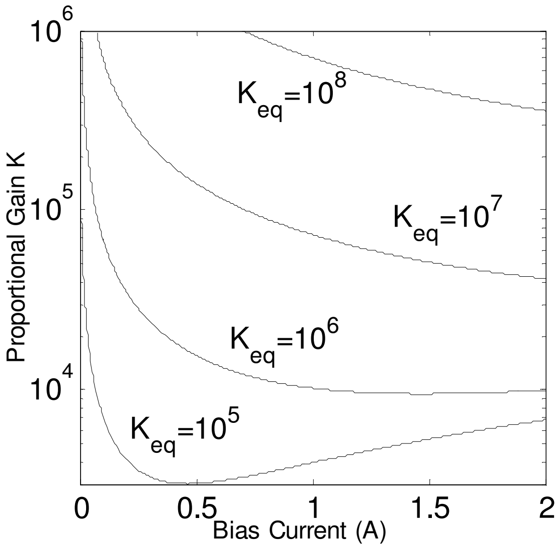

FIGURE 3: Constant bearing stiffness curves as a function of K and I_b

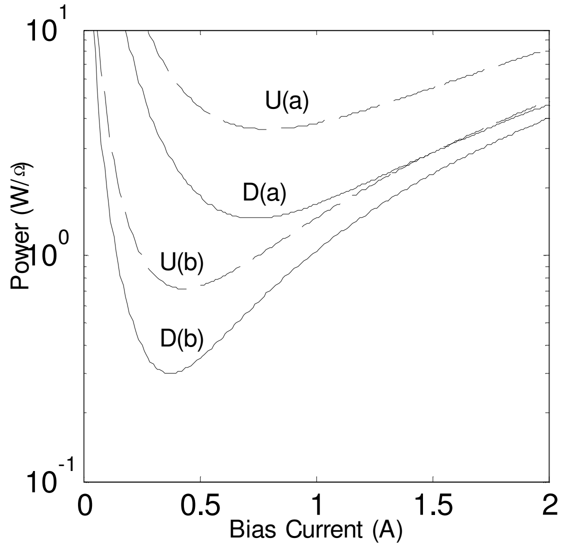

FIGURE 4: Power consumption (W/Ω) as a function of the bias current, constant K_{eq}=10^6 N/m. Vibration amplitude: (a) $0.4g_0$, (b) $0.1g_0$. Modes: (U) Unidirectional, (D) Differential

small bias current settings at the expense of increasing the nonlinearity. The lower limit of the bias current is set by the properties of the power amplifiers. The current on the opposing poles are:

$$I_1 = \begin{cases} I_b + 2|I_c| & \text{if} \quad I_c > 0 \\ I_b & \quad\quad I_c < 0 \end{cases}$$
$$I_2 = \begin{cases} I_b & \text{if} \quad I_c > 0 \\ I_b + 2|I_c| & \quad\quad I_c < 0 \end{cases} \tag{10}$$

Assume a steady state synchronous sinusoidal motion resulting from an unbalance force. Then inserting equations (7) and (10) into equation (4) and linearising around $s = 0$ gives the same equivalent bearing stiffness expression as in the differential control current mode in equation (9).

A required equivalent bearing stiffness can be obtained by various combinations of the bias current I_b and the proportional controller gain K satisfying equation (9). Figure 3 shows contours of constant equivalent stiffness values. If the bias current is fixed by design then there is obviously only one proportional feedback value to give the required bearing stiffness. In the case of a variable bias current design considered here, this freedom can be used to select a bias current to minimise the energy consumption, and then choosing the feedback gain to realise the bearing stiffness. The selection of I_b is limited because the resulting sinusoidal control current must lie between an upper and lower limit. The upper limit is determined by the current limitation of the power amplifiers and the coils. The lower limit is set by the transconductance amplifiers as they can not function properly below a certain current. The maximum current limit is taken as 2 A in the simulated system considered here.

BIAS CURRENT OPTIMISATION

A steady state synchronous response of the rotor at a constant rotational speed of Ω can be represented as:

$$s = S\sin(\Omega t + \theta) \tag{12}$$

where S is the amplitude of the synchronous response, and θ is the phase of the response with respect to a reference point on the shaft. The power consumption P can be expressed as.

$$P = \frac{R}{2\pi} \int_{\Omega t=-\theta}^{2\pi-\theta} (I_1^2 + I_2^2) d(\Omega t) \tag{13}$$

where R is the coil resistance, and I_1 and I_2 are given by equations (8) or (10). The power is a function of the vibration amplitude S, and the bias current I_b for a given equivalent bearing stiffness K_{eq}. Therefore by using the measured value of S and the corresponding value of K_{eq}, it is possible to select I_b and K to minimise equation (13). Since K_{eq} does not change, the system response should not change.

Figure 4 shows how the power consumption of opposing coils changes as a function of the bias current for both modes of operation for an equivalent bearing stiffness of 10^6 N/m. If the bias current is variable, then the differential mode is superior to the unidirectional

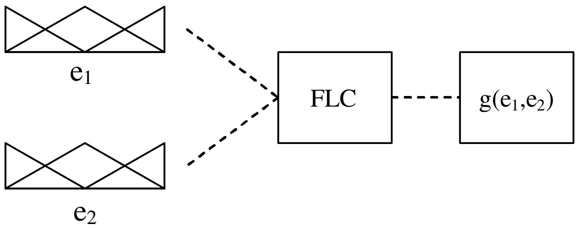

FIGURE 5: Fuzzy Logic Controller to select bearing stiffness.

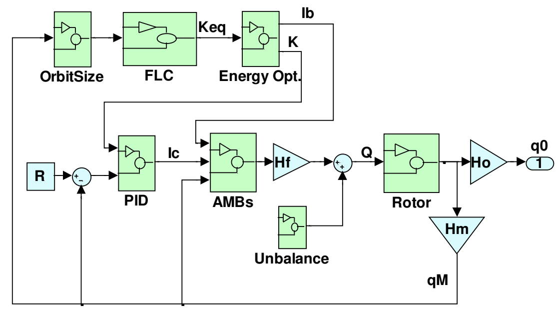

FIGURE 6: Control System Block Diagram

mode at both low and high vibration levels. Therefore a variable bias current control in differential mode is used in the proposed controller. This argument would not be valid if constant bias current controllers are used. In a constant current differential mode, the bias is set at the half of the upper current limit, which is 1 A in this example. In the constant bias current unidirectional mode, the bias current can be set to a much lower value, say at 0.4 A, making it more energy efficient at low vibration levels than the differential mode. The high vibration amplitude results for the orbit size of $0.4\times g_0$, as shown in graphs (a) in figure 4, represent an extreme case. The rotor vibrations should not reach to this level under normal operating conditions. The retainer bearings are not co-located with magnetic bearings and have smaller clearances. Therefore, the controller should not allow the orbit to grow beyond 40 percent of the magnetic bearing clearance, and in any case the fuzzy logic controller should demand highest possible effective bearing stiffness when the orbit approaches this limit. The results also show that even under this extreme condition, significant energy savings are possible under the variable bias current control controllers.

FUZZY LOGIC CONTROLLER

The selection of a K_{eq} value depends on the objective function of the controller. During steady state constant speed operations, it is desirable to minimise the transmitted forces, hence the equivalent bearing stiffness should be small (i.e. soft bearing). The rotor may be allowed to vibrate within certain limits. However, soft bearing can cause the rotor vibrations to exceed the safety levels if there are sudden or gradual changes in operating conditions, such as a change in unbalance, transient or steady external excitations, faults in the transducers or signal processing, drive electronics etc [11]. In this case, it would be desirable to increase bearing stiffness until the rotor vibrations are reduced to an acceptable safe level. Obviously, it is important to avoid rotor contact with the retainer bearings. These conflicting requirements and constraints can be handled by a fuzzy logic controller, which selects the optimum equivalent bearing stiffness value. A first order Takagi-Sugeno (TS) type fuzzy controller is used because of its computational efficiency. The Multi Input Single Output rule has the form:

$$\{ \text{ If } f(e_1 \text{ is } A_1) \text{ and } (e_2 \text{ is } A_2) \quad \text{Then } y=g(e_1,e_2) \} \qquad (14)$$

where e_1 and e_2 are inputs, and A_1 and A_2 are term sets, and y is the output of the FLC, f() is a logical function and g() is the function of the inputs [6].

The inputs of the TS fuzzy supervisory control (FSC) are the orbit size and changes in the orbit size. The output is the equivalent stiffness value for each magnetic bearing. Each input has three triangular shape membership functions as shown in figure 5. This controller is added to the rotor and magnetic bearing modelling as shown in Figure 6. Local PID controllers act on the rotor position signals corresponding to magnetic bearing locations. The FSC gives a request for K_{eq} to the energy optimisation block, which calculates the optimum values of the bias current and the proportional gain to achieve the required equivalent bearing stiffness.

The simulation is run at a constant operational speed of Ω = 30,000 rpm. An unbalance of $0.75\text{x}10^{-5}$ kgm is suddenly added to the shaft at the location of the upper magnetic bearing. Table 1 shows the results after all control parameters converged to their steady state values. At this level of unbalance, the controller has set the bearing stiffness at 1.875×10^5 N/m with rotor vibrations of 6×10^{-6} m, which is about 2% of the magnetic bearing clearance. The above stiffness value is obtained by setting the bias current to 0.34 A and the proportional feedback gain of 4950. This requires a power of about 0.9 W per axis. When the system is run under constant bias current setting, the power requirements of the differential and unidirectional modes are 7.72 and 1.53 Watt per axis respectively. As expected the unidirectional mode is more efficient than the differential mode under constant bias current settings. However, variable bias current FLC provides the most efficient solution with power requirements reduced by 88% and 41% compared with constant current differential and unidirectional modes

TABLE 1: Experimental results at 30,000 rpm with an unbalance of 0.75×10^{-6} kgm.

Bias Current	Variable	Constant	
Mode	Fuzzy Differential	Differential	Unidirect.
K_{eq} (N/m)	187,500	324,000	207,000
I_b (A)	0.34	1	0.43
K	4950	5500	4750
S (μm)	6	5	4.3
P (W)	0.90	7.72	1.53

respectively. Further runs have been performed by suddenly increasing the out of balance to 1.65×10^{-6} kgm, where the controller is settled for 2×10^{5} N/m bearing stiffness with I_b = 0.38 A, K = 4,900 resulting an orbit of 1.45×10^{-5} m (about 5% of the clearance) and power consumption of 1.14 W per axis.

CONCLUSIONS

The paper describes the application of a variable bias current fuzzy logic controller to minimise energy consumption under changing operating conditions and conflicting controller requirements for a magnetically levitated and controlled turbo molecular pump. It has been shown that an optimum bias current for a given bearing stiffness and rotor vibration level exists. A significant energy savings can be achieved by varying the bias current and the proportional feedback gain without affecting the system response. Furthermore, the setting of the equivalent bearing stiffness through an FLC can provide robustness and adaptability against sudden or gradual changes in operating conditions or development of faults. A dynamic simulation has demonstrated the effectiveness of the proposed algorithm. Significant energy savings are predicted compared with the use of constant bias current differential and unidirectional mode controllers. The ability of the controller to cope with a sudden increase of unbalance has also been demonstrated.

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