Design and Control of Hybrid Magnetic Bearings for Maglev Axial Flow Blood Pump

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Abstract - This paper introduces the development of hybrid magnetic bearings (HMBs) for a maglev axial flow blood pump. The design criteria for the radial, axial and current stiffness of the HMB are proposed and the HMB is accordingly designed through theoretical calculation, 3-Dimensional Finite Element Analysis (3-D FEA) and experimental verification. Based on the system identification result of the HMB, a proportional, integral, and derivative (PID) controller, low pass filter and lead compensator have been designed for the HMBs of the axial flow blood pump. Open and closed loop transfer functions of the system were determined and experimental results show that the developed maglev pump can spin stably in air with speeds up to 20,000 rpm and in water with speeds up to 9,000 rpm, which validate the design and control of HMBs of the maglev pump. The magnetic bearing system adequately supports the pump impeller and is well suited for this application.

I. INTRODUCTION

A. Active Magnetic Bearing

Active magnetic bearings are a novel mechatronic product that produce contact-free electromagnetic force to support rotors or other loads. Compared to traditional bearings, active magnetic bearings (AMBs) offer advantages such as: no friction, low heat generation, no required lubrication, quiet operation, and fast and stable rotation due to active control. Because of these reasons, active magnetic bearings have been highly favored in artificial blood pumps in order to eliminate the problems such as material wear, heat generation, bearing maintenance, platelet aggregation, and thrombus growth, which may otherwise be caused by traditional bearings [1][2][3][4][5].

A typical active magnetic bearing (AMB) system usually consists of an actuator, a position sensor, a controller and an amplifier (Fig. 1). The sensors monitor the rotor position and send this information to the controller. The controller determines required control effort and sends the control signal to the magnetic actuators via amplifier. The magnetic actuators convert the current supplied by the amplifier into magnetic forces exerted onto the rotor, which act to adjust the rotor position and provide enough damping to the rotor.

Fig. 1. Control diagram of active magnetic bearings

The controller is critical to the active magnetic bearing system and it can determine the performance of the AMBs [4][5][6][7][8][9][10]. Many control schemes have been investigated. Some of these are LQG/LTR [11], H∞ [12], sliding mode [13], adaptive feed-forward [14], and PID [15]. Among these, PID (proportional, integral, and derivative) control is the most widely used in magnetic bearings system because of its simplicity and robustness [16][17]. In order to reduce high-frequency noise and increase system phase margin, PID control is sometimes combined with low pass filters and lead compensators. This combination has been shown to be very effective in controlling AMBs [18].

B. Magnetically Levitated axial flow blood pump

In this paper, hybrid magnetic bearings (HMBs) have been developed for axial flow blood pump. A hybrid magnetic bearing is a magnetic bearing where one or more permanent magnet rings are added to deliver bias magnetic flux for the electromagnets. In HMBs, the bias-flux is provided by the permanent magnets and therefore it can decrease the system power consumption. The impeller of the blood pump is suspended by magnetic fields, actively controlled by radial HMBs in the radial axes and passively controlled in the axial axis. Hall Effect sensor arrays (HESAs) are used to sense the rotor displacements and communicate these to the controller.

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The cross section view of the developed maglev axial flow blood pump is shown in Fig. 2. The impeller is levitated by the two HMBs on two ends and driven by a brushless permanent magnet dc motor in the middle. Two HESAs are mounted on sides of HMBs to measure the rotor position by sensing the positions of the HESA magnets which are contained within the rotor.

Based on preliminary prototypes, two criteria have been established for the design of the HMBs: one is that the radial strength of the HMBs should be high enough to pull the impeller/rotor back from the wall to the levitation center; the other one is that the axial stiffness of the HMBs should be high enough to hold the impeller from touching the pump front cavity at maximum speed and pressure rise. In order to guide the design process to satisfy the two criteria, 3-D finite element method (FEM) has been utilized which, combined with experimental verification, can help designers analyze the magnetic fields, find out its magnetic flux loop, and calculate the electromagnetic forces with different dimensions or configurations of magnetic bearings [5][19][20][21].

PID control has been used successfully for the HMBs of this axial flow blood pump. A low pass filter is added to the sensor outputs to reduce high-frequency noise and act as an anti-alias filter. System identification was performed on the magnetic bearings system of the pump both in air and in fluid to find out its frequency response. Based on the experimental system identification results of the magnetic bearings, a lead compensator was designed for the HMBs of the pump in air in order to increase the system performance.

This paper introduces the design and control of hybrid magnetic bearings for axial flow blood pump application. Two design criteria are proposed and 3-D FEA and experimental verification are utilized to guide the design process. The controller was modeling and implemented digitally, based on the system identification results of HMBs. Experimental results of the developed maglev axial flow blood pump with HMBs are reported and these results verify the validity of both the HMB structure and controller design.

II. METHODS

A. Structure Design of HMB

Preliminary design of HMB - The schematic of HMBs of the axial flow blood pump is shown in Fig. 3. In order to provide passive axial stability, the axially magnetized stator and rotor permanent magnets are sandwiched between the stator and rotor big and small end iron (Fig. 3). The magnetization of stator and rotor permanents magnets are opposite one another so that a closed magnetic flux path is formed between the two. The coils are wound around the leg of the stator big end iron, which is designed to provide the main electromagnetic force.

Preliminary design of HMB - The electromagnetic force of magnetic bearing is dependent on magnetic air gap, the number of coils, and the pole face area, according to

\[ F = \frac{\mu_0 N^2 I^2 A_x}{4 g^2} \]  

(1)

where \(\mu_0\) is the magnetic permeability of the air, \(N\) is the number of coil turns, \(I\) is the coil current, \(A_x\) is the cross-section area of the flux, and \(g\) is the air gap distance between the electromagnet and the rotor [6].

Due to the constrains imposed by providing enough space for the blood to flow between rotor and stator, biocompatible packaging of the magnetic components, and an impeller clearance of 0.2 mm, the magnetic air gap of the HMBs is set as 2.5 mm, which is much larger than the magnetic gap for most applications.

The mass of the pump impeller is 0.15 kg. (1G = 1.44 N). The axial thickness of the big iron within both the rotor and stator is designed as 13 mm. Gauge 23 wire (maximum current of 4.7 A) with 48 turns is used for the electric coils. Using these numbers and equation (1), the current needed for one magnetic bearing to levitate the rotor is about 2 A. The material of N4467, which has the residual flux density of 1.33 Tesla and relative permeability of 1.0555, with length of 6 mm is chosen for the permanent magnets of HMBs. M19 iron was chosen for the AMBs with magnetic relative permeability of 8000.

Design criteria of HMB - The relationship between electromagnetic force to current supplied for an AMB, as
defined by equation (1), can be linearized over small
displacements. This electromagnetic force \( F \) can be
expressed as the sum of forces resulting from control current
\( i \) and rotor displacement \( x \) [6]:

\[
F = K_i i + K_x x
\]  
(2)

where \( K_i \) is the force-current factor (current stiffness) and \( K_x \)
is the force-displace factor, position stiffness (radial or
axial).

We determined out the passive radial stiffness of the
motor is about -6 N/mm based on the earlier empirical
measurements. A summation of forces that would be present
when the rotor is at its maximum radial displacement, equation (3), indicates that the current stiffness and radial
stiffness of the HMB should allow the HMB to pull the
impeller back to center when it is stuck on the wall
\[
0.147 \times 9.8 < K_x \times 2 \times 4.7 + (K_x \times 2 - 6) \times 0.2
\]  
(3)

Equation (3) can be rearranged to

\[
23.5 K_i + K_x > 6.6
\]  
(4)

The design variables most easily modified are the thickness
and inner diameter of the small end iron, with which the
HMB should satisfy the equation while at the same time
keeping the pump stable in the axial axis.

When the pump is running in water, there is pressure rise
across the pump, which makes the impeller move axially forward. The maximum allowable travel is 1 mm for the
impeller and the maximum pump pressure is 133 mmHg
(17.7 kPa) with flow rate of 6L/m @7K rpm of rotational
speed. The cross section area of the impeller head is 2e-4 m².
Therefore the fluid force caused by the pump head pressure is

\[
F = P \times A = 3.5N
\]  
(5)

When the axial pump is placed with front downwards, the
gravity force (1.44N) must be taken into account, and
therefore the total possible maximum axial force is 1.5+3.5
= 5 N. The axial stiffness of the motor is found out to be 2.4
N/mm by 3-D FEA and verified by empirical measurement.
The axial stiffness of the HMBs should be high enough to
prevent the impeller from touching the pump cavity on the
front at this situation, and therefore it should satisfy the following equation

\[
2 \times K_x + 2.4 > 5
\]  
(6)

Optimal thickness and ID of the stator end iron

The small stator iron affects the force capacity of HMB, and we concentrate on the axial thickness and inner diameter
(ID) of the small stator iron.

In order to find out the optimized thickness of the stator
small end iron, its thickness was varied from 0 to 4 mm in
1 mm increments and its ID was adjusted to 5 values between
20 mm to 38 mm. The corresponding radial, axial, and
current stiffness of each variation of the HMB was
determined through 3-D FEA and experimentally [5]. The

| TABLE I |
|-------------|------------------|-----------------|
|             | RADIAL, AXIAL AND CURRENT STIFFNESS OF THE DESIGNED HMB |
| 3-D FEA Result | Empirical Data |
| Radial stiffness (N/mm) | -5.9 | -5.1 |
| Axial stiffness (N/mm) | 3.2 | 3.1 |
| Current stiffness (N/A) | 0.87 | 0.87 |

B. Controller Design of HMB

The transfer function of the PID controller that is used on
the HMB for one direction is as follows

\[
G_p(s) = k_p + \frac{k_i}{1 + s T_d} + \frac{k_d}{s}
\]  
(7)

where \( k_p \) is the proportional coefficient, \( k_i \) is the differential
coefficient, \( k_d \) is the integration coefficient, and \( T_d \) is the time
constant of the differentiator. By controlling the coil currents
through PID controller, the position of the rotor can be controlled.

In order to reduce the high-frequency noise brought by the
HESAs, a low pass filter is designed and its transfer function is as follows

\[
G_{lp}(s) = \frac{k_{lp}}{T_{lp}s + 1}
\]  
(8)

where \( k_{lp} \) is the gain of the low pass filter, \( T_{lp} \) is the time
constant of the low pass filter.

In order to increase the phase margin of the system, a lead
compensator [23][24][25] is designed and implemented according to

\[
G_{ld}(s) = k_{ld} \frac{T_{ld}s + 1}{\alpha T_{ld}s + 1}
\]  
(9)

where \( k_{ld} \) is the gain of the lead compensator, \( T_{ld} \) is the time
constant of the lead compensator. The ratio \( \alpha \) is obtained by

\[
\alpha = \frac{1 - \sin \phi_m}{1 + \sin \phi_m}
\]  
(10)

where \( \phi_m \) is the maximum phase lead. The time constant \( T_{ld} \) is obtained by

\[
T_{ld} = \frac{1}{\omega_m \sqrt{\alpha}}
\]  
(11)

where \( \omega_m \) is the frequency at which there is the maximum
phase lead. \( \omega_m \) should be identified such that the magnitude
of the open loop system \( G_{ol} \) should satisfy
\[
|G_{el}(j\omega_m)K_{el}| = \sqrt{\alpha} \tag{12}
\]

Since the magnitude of the lead compensator at high frequency is high based on equation (9), this can magnify high frequency noise and an additional low pass filter with high cutoff frequency is added to reduce the filter gain at high frequency. The final transfer function of the lead compensator including this filter is

\[
G_{ld}(s) = k_{ld} \frac{T_{ld}s + 1}{(\alpha T_{ld}s + 1)(T_{ldlp}s + 1)} \tag{13}
\]

where \(T_{ldlp}\) is the time constant of the additional low pass filter.

C. System Identification of HMB

The reference position of rotor was supplied with a multi-frequency signal, and then its corresponding rotor displacements are acquired. These time domain reference and displacement signals were then transformed into frequency domain signals by Fast Fourier Transform (FFT), and the system transfer function in frequency domain was obtained by system output (rotor displacement) over system input (reference position). A broadband SPHS [22] (Appendix), multi-frequency test signal with bandwidth of 1 to 300Hz and fundamental frequency of 1 Hz was used to perturb the suspended rotor.

III. RESULTS

A. Hardware

A system consisting of an xPC Target and Matlab with Simulink is used as the rapid prototyping tool for the development of the digital controller of the HMBs. The sampling frequency of the digital controller is set as 5 KHz. One NI 6024E card is used as the A/D data acquisition card, and one NI 6602 counter is used as the PWM generator card with switching frequency of 20 KHz. Allegro A1321 Hall Effect Sensors are used for the pump as position sensors, which have the output sensitivity of 5mV/G. The photograph of the experimental setup is shown in Fig. 4.

B. PID controller

The same PID controller is used both in air and in water levitation. Due to the fluid force at high speed, integrator is not used for the pump running in fluid. Experimental PID gains are listed in Table II.

<table>
<thead>
<tr>
<th>TABLE II</th>
<th>PID CONTROLLER GAINS FOR AIR AND WATER</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(K_p)</td>
</tr>
<tr>
<td>In Air</td>
<td>3</td>
</tr>
<tr>
<td>In Water</td>
<td>4</td>
</tr>
</tbody>
</table>

C. Low Pass Filter

In order to reduce the high-frequency noise from the HESA and at the same time not to affect the HMBs system performance, the cutoff frequency of 700 Hz is set for the low pass filter with the gain of 1, and the transfer function of the low pass filter is

\[
G_p(s) = \frac{1}{0.0003s + 1} \tag{14}
\]

D. System Identification Results in Air and in Water

Fig. 5 below shows the system identification result: closed loop system transfer functions both in air and in water, from which we can obtain their open loop system transfer functions in MATLAB which is shown in Fig. 6.

E. Lead Compensator

Based on the system identification results shown above, the phase margin of the open loop system is 13 degree for the air and 42 degree for the water. To have good system performance, the phase margin of system should be larger than 40 degree [25]. Therefore a lead compensator is added to PID control for the HMBs working in air, but the lead compensator is not necessary for the system in water.

Since the DC gain of the system in air is 1.8, the \(K_{ld}\) of the lead compensator is set as 1. \(\phi_m\) is chose as 30 degree for the air as the maximum phase lead of the compensator. So based on equation (10), we can obtain \(\alpha = 0.33\). Then according to the equation (12), and based on the experimental system transfer function shown in Fig. 6, \(\omega_m\) is indentified as 220 degree. And according to equation (11), we have \(T_{ld} = 1.25\) ms. Usually the pole of the additional low pass filter is chosen 10 times of the first pole of the lead compensator and therefore \(T_{ldlp}\) is equal to 4.18e-5 s.
So finally based on equation (13), we can obtain the transfer function of the lead compensator in air, as shown in equation:

\[
G_{ld\_air}(s) = \frac{1.25e^{-3s} + 1}{1.74e^{-8s^2} + 4.59e^{-4s} + 1}
\] (15)

**F. System Performance in Air and in Water**

Firstly, PID control of HMBs without lead compensator was tested in air and it was found out the rotor/impeller hit the surrounding wall (displacement of 200 micron) when its speed was increasing to about 7,000 rpm, no matter what the PID gain set. The designed lead compensator as introduced above was added to the PID control and the rotor could operate at and above 7,000 rpm. Error! Reference source not found. shows the rotor displacements and current consumption of the HMBs system both in air (with lead compensator) and in water (without lead compensator), from which it can be seen that by adding the lead compensator, the maglev impeller of the pump can pass its natural frequency stably in air and after that the rotor displacements become smaller with the increase of the rotor rotational speed, all of which matches the system transfer function shown in Fig. 5. The current consumption of the HMB system in air and water (Fig. 8) shares the same trend as the rotor displacements: the maximum current consumption occurs at the natural frequency and after that the current consumption decreases as a function of speed.

Due to the power limitation of the motor, the pump can only spin up to 9,000 rpm in water, while the normal operational speed of heart pump is around 5,000 rpm. It can be seen that without lead compensator the pump rotor can past its natural frequency in water as predicted by the system identification result shown above (about 8,000 rpm), and after that speed, the rotor displacements and HMBs’ current consumption decreases.
IV. CONCLUSIONS

In this paper the design and control of HMBs for axial flow blood pump have been introduced. Based on preliminary design and assisted by 3D FEA and experimental verification, the HMBs that can safely satisfy the design criteria are presented. The low pass filter is added to the PID controller in order to remove the high frequency noise from the position sensors. Through the system identification of the HMBs, the lead compensators are designed for the air operation of the pump so that the maglev pump system can have enough phase margins. System identification results also show that lead compensator is not necessary for the water operation of the pump. Experimental results show that the maglev impeller of the pump can spin stably at speeds up to 20,000 rpm in air and 9,000 rpm in water, which validates the design and control of the pump.

The method of parameter estimation is currently underway to be applied on the maglev pump to find out its stiffness and damping coefficients of different fluids and their influences on the pump performance.

APPENDIX

The Schroeder Phased Harmonic Sequences (SPHS) is a signal that contains different frequency components with lowest possible peak value. This low-peak periodic signal $s(t)$ has any given power spectrum $P_s$ where $P_s$ is the ratio of the power at $\omega = \omega_0$ to the total power. i.e.\[ \sum_{i=1}^{N} P_i = 1 \] SPHS is constructed as:\[ s(t) = \sum_{i=1}^{N} \frac{P_i}{2} \cos(i\omega_0 t + \phi_i) \] \[ \phi = \phi_0 - 2\pi \sum_{i=1}^{N} P_i, i = 1..N \]

REFERENCES