

Development of highly efficient hybrid magnetic bearing and application to ultra-low temperature fluid pump

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Abstract

Highly efficient hybrid type magnetic bearing is proposed in this paper. It is intended to apply the proposed magnetic bearing to ultra low temperature fluid pump. Traditional low temperature pump is supported by a ceramic ball bearing which is damaged by the low temperature liquid. Non-contact support is highly requested. However, the standard electro-magnet type magnetic bearing consumes high electric power causing heat to the liquid. This paper proposes highly efficient hybrid type radial magnetic bearing which can operate with very low power consumption. Also the pump is canned by FRP pipe which has thick wall to separate the stator from the liquid. The designed magnetic bearing has 3 mm wide air gap. The fabricated magnetic bearing shows good dynamic properties with high force factor.

Key words : Magnetic bearing, Actuator, Digital control, Rotary machinery, Mechatronics, Centrifugal pump

1. Introduction

Traditional ultra-low temperature pump uses ceramic ball bearing in the liquid (Barber Nichols Inc.). Hence there are several problems of short bearing life time, pollution of powdered ceramic and heat transfer to the liquid.

The standard electro-magnet type magnetic bearing consumes high electric power and is not applicable (Schweitzer and Maslen, ed., 2009). In this paper highly efficient magnetic bearing is proposed for supporting the pump impeller in ultra-low temperature liquid. A centrifugal pump impeller is supported by a hybrid (HB) type radial magnetic bearing and the shaft is set in vertical position as shown in Fig. 1. The top of the shaft is supported by a standard ball bearing and motor, since the low temperature liquid is under this level. The stator of HB magnetic bearing is better to be separated from the fluid. A special designed fiber reinforced plastics (FRP) pipe can be used in such a low temperature which has thick wall of 2-mm thickness (Fujiwara Co.). The designed magnetic bearing has 3 mm wide air gap which has good dynamic characteristics with high force factor (Kakihara, et al., 2005), (Kodama, et al., 2006), (Martin, et al., 2004), (Kakihara, et al., 2006), (Okada, et al., 2009), (Miyazawa, et al., 2009), (Okada, et al., 2011). Highly efficient magnetic bearing is realized by

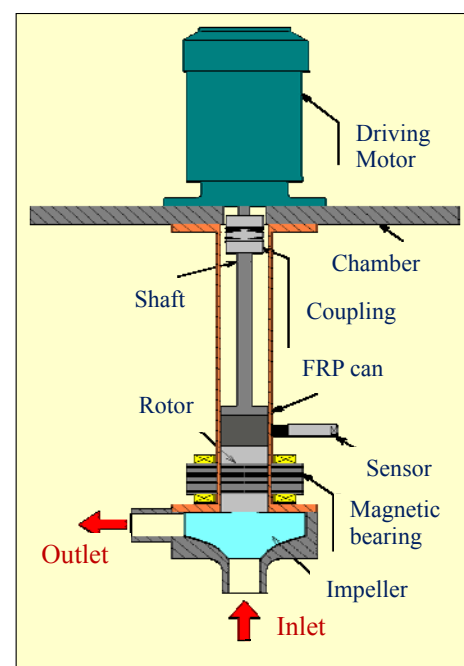


Fig. 1 Schematic of ultra-low temperature fluid pump

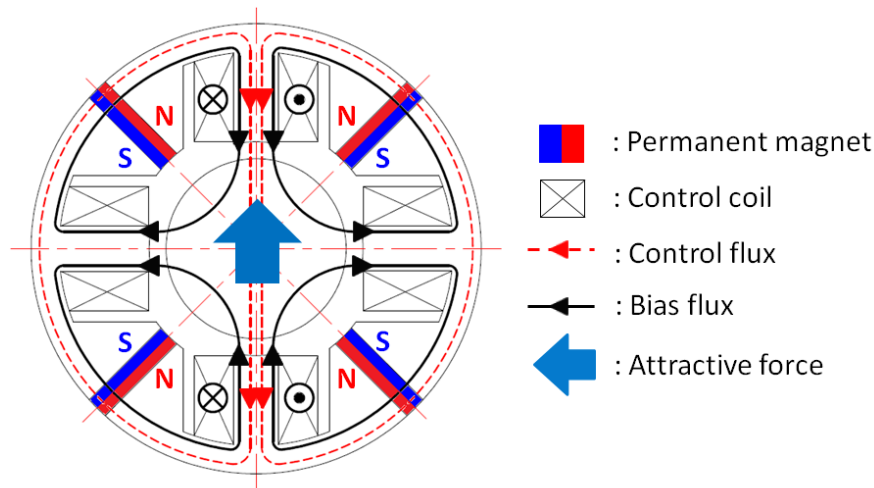


Fig. 2 Principle of proposed radial magnetic bearing

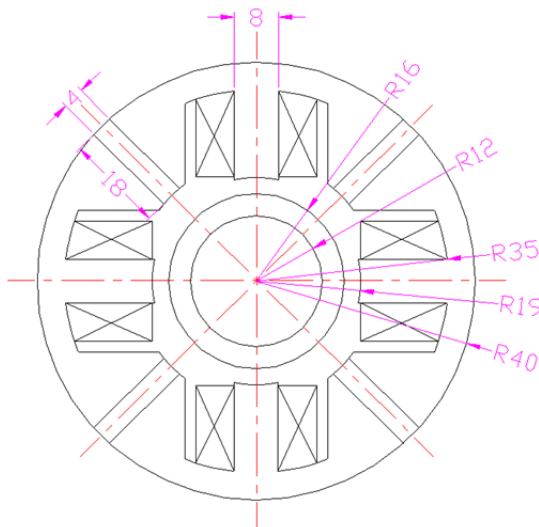


Fig. 3 Analytical model of radial AMB

Table 1 Physical parameters of radial AMB

Item	Value, Material
Air gap	3 mm
Permanent magnet	Neodymium (NMX-48CH) ≈ 1000 kA/m
Coil turns	400 turns/slot
Core material	Electromagnetic silicon sheet 35A300

using strong bias flux produced by wide area neodymium permanent magnets (PMs) and high copper ratio coils by using pressed winding technology made by Selco Co., Ltd. (Selco Co.).

2. Principle of highly efficient magnetic bearing

Principle of the proposed magnetic bearing is shown in Fig. 2. The stator has four main poles with pressed coil windings. Between them sub-poles are arranged which has strong and wide area neodymium PMs to produce high bias flux as shown by the solid black lines. The upper and lower control windings are connected series. To produce upward force the current of two coils are flowing as shown by the arrow symbols which produce the dotted red control flux lines. The control current increases the upper air gap flux while decreases the lower air gap flux causing upward force as shown by the big blue arrow. Similar control is applicable to the left and right coil windings to control the left and right direction. Hence radial two directions are actively controlled.

3. FEM analysis and design

The proposed magnetic bearing is designed through 2D FEM analysis. Then the model is optimized through the analysis.

3.1 Analytical model

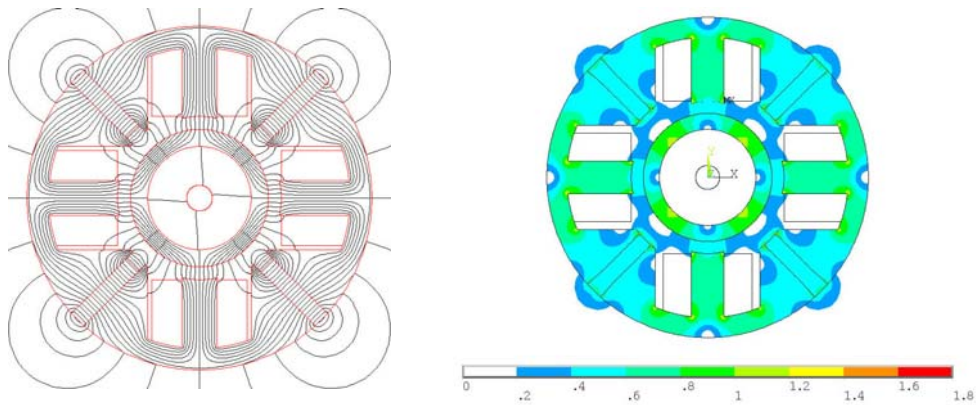


Fig. 4 Bias flux lines and flux densities (Control current 0 A)

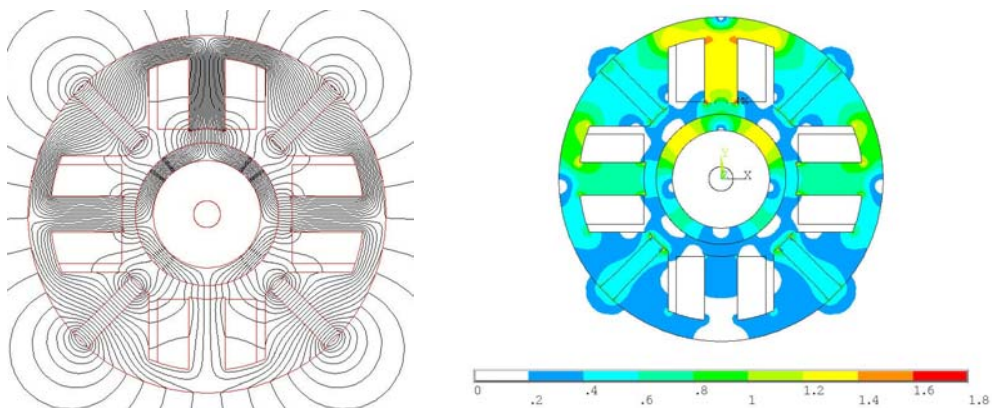


Fig. 5 Flux lines and flux densities (Control current 2 A)

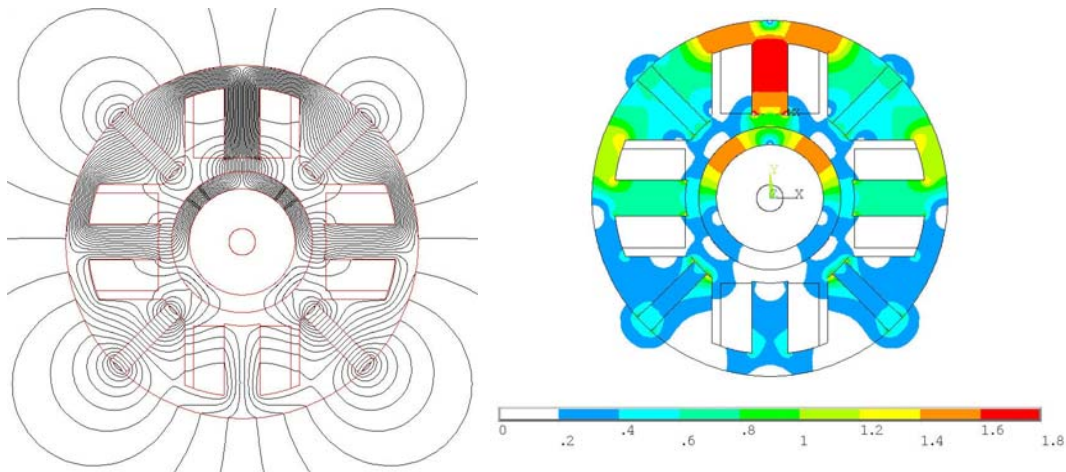


Fig. 6 Flux lines and flux densities (Control current 4 A)

Analytical model and the parameters are shown in Fig. 3 and Table 1, respectively. Analysis is carried out by using the commercial FEM code ANSYS. Through analysis the model is optimized to get enough force factor. The model shown in Fig. 3 is the optimized one.

3.2 Analytical results

Analytical flux lines and flux densities are shown in Fig. 4 (Control current 0 A), in Fig. 5 (Control current 2 A), and in Fig. 6 (Control current 4 A), respectively. Bias flux is well formed from the permanent magnets as shown in Fig. 4. When the control current is 2 A, the upper pole flux is increased near the saturation level around 1.3 T while

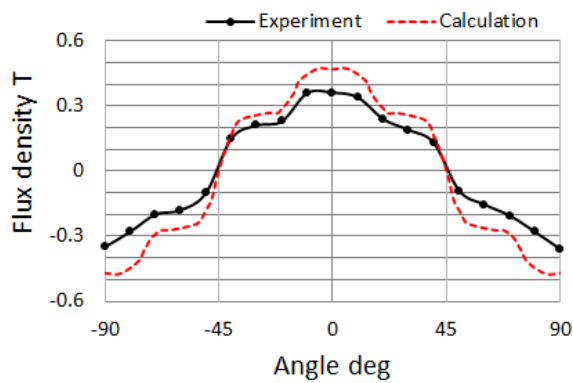


Fig. 7 Flux densities in air gap

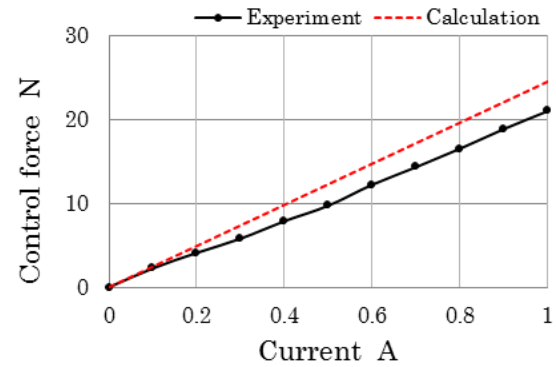


Fig. 8 Control force characteristics

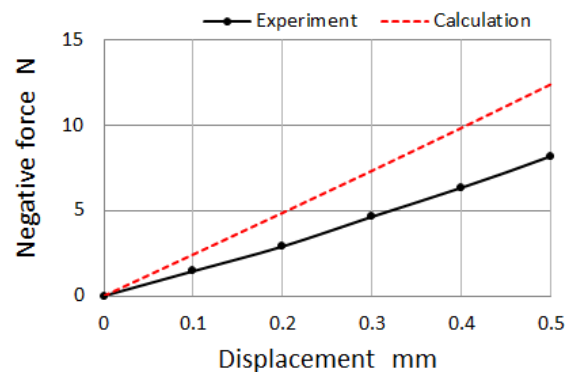


Fig. 9 Negative force characteristics

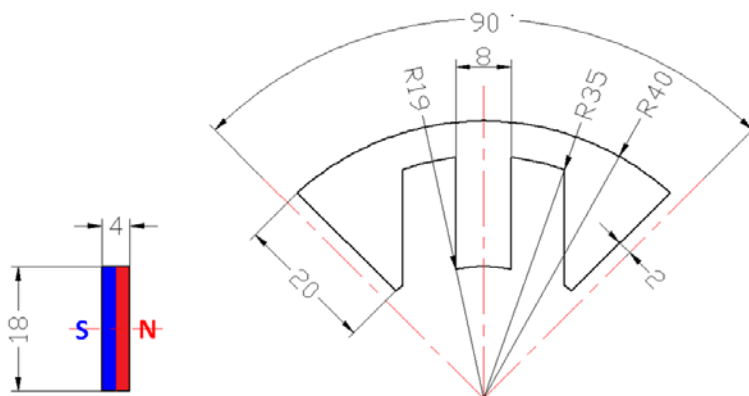


Fig. 10 Designed stator core and permanent magnet



Fig. 11 Photo of assembled stator

the lower pole flux is lower than 0.1 T as shown in Fig. 5. When the control current is 4 A, the upper pole flux is over 1.6 T while the lower flux lines are inverted. This means that relatively small current can control this magnetic bearing.

Analyzed air gap flux density, control force and negative force characteristics are shown by the dotted red lines in Figs. 7, 8 and 9, respectively. The air gap flux density in Fig. 7 shows enough bias flux with the maximum bias flux of 0.48 T. Also the control force is considered enough high with the force factor of 26.1 N/A in Fig. 8 compared with the negative force factor of 25.4 kN/m in Fig. 9. These values are considered enough high compared with the wide gap of 3 mm. These graphs also include the experimental results mentioned later.

Even though the maximum bias flux is high, the air gap flux change is relatively smooth as shown in Fig. 7. The rotor core is made of laminated silicon sheet, hence the rotor rotating loss may be small compared with the liquid viscosity loss.

4. Experimental magnetic bearing and static characteristics



Fig. 12 Fabricated magnetic bearing and flux density measurement

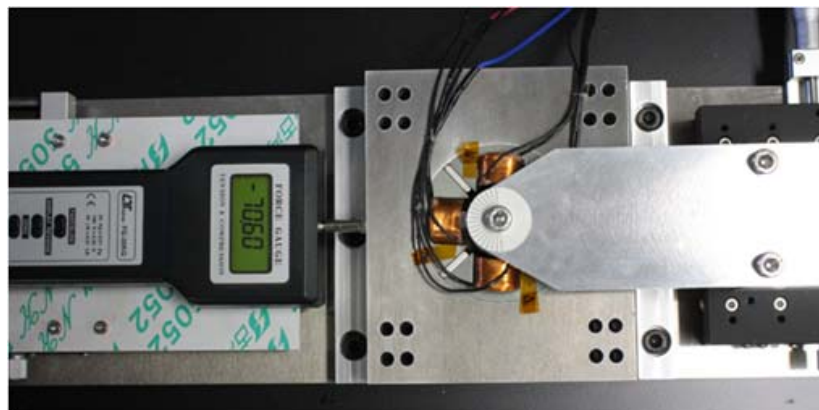


Fig. 13 Photo of control and negative force measurement

To confirm the capability of the proposed magnetic bearing the designed magnetic bearing is fabricated and tested.

4.1 Experimental setup

The designed stator core and PM are shown in Fig. 10 while the photo of assembled stator is shown in Fig. 11. The stack thickness of the cores and PMs are 25 mm.

The stator has PMs, control coils and sensors which produce electromagnetic flux change. The stator is better to be separated from the ultra-low temperature liquid. But any metal can may affect adversely to dynamic change of flux. A special FRP pipe can be used in such low temperature liquid which has 2-mm thickness (Fujiwara Co.). The stator is designed to have 3 mm wide air gap.

4.2 Measured static characteristics

First, the flux density in air gap is measured by using a gauss meter (Lake Shore 421, MNT-4E02-VH) as shown in Fig. 12. The magnetic fluxes are measured in angular step of 10 degrees the values of which are shown by the dots with solid line in Fig. 7. The maximum flux density is about 0.36 T which is about 75 % of the calculated value. The measured flux densities are not symmetrical. This is considered due to the assembling error, but this is not serious for practical use. Next, the control and the negative forces are measured by using the measuring setup as shown in Fig. 13. For centralized shaft and measuring the force factor, forces were measured by increasing the current in steps of 0.1 A as shown by the dots with solid line in Fig. 8. Also for measuring negative force factor, forces are measured after shaft displacements in step of 0.1 mm as shown by the dots with solid line in Fig. 9. The force factor is about 21.1 N/A which is 88 % of the calculated one. The negative force factor is 16.4 kN/m which is 68 % of the calculated one. The measured values are lower than the calculated ones. This is mainly due to the three dimensional flux leakage which is not considered in 2D analysis. Even though the measured values are lower than the calculated ones they are still high enough to control the pump impeller.

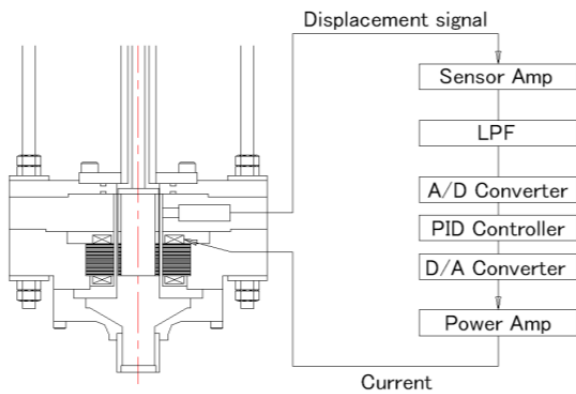


Fig. 14 Control system

Table 2 PID gains of levitation control

	Radial Value	
	x - direction	y - direction
K_p A/m	10	10
K_i A · s/m	0.0001	0.0001
K_d A/s · m	0.03	0.03

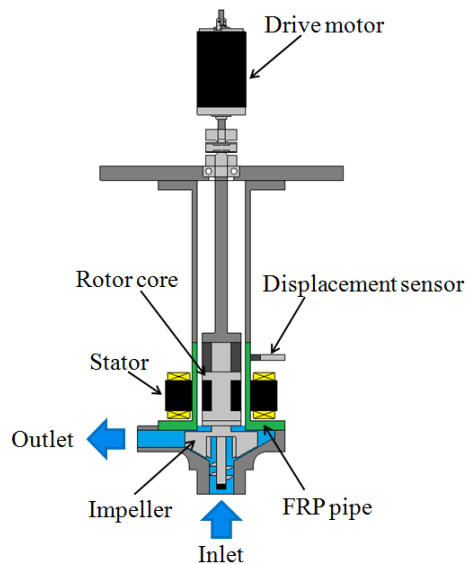


Fig. 16 Control object (Pump and AMB)

Table 3 Pump specification

Item	Value	Unit
Fluid	Liquid nitrogen	-
Suction pressure	101	kPa
Suction temperature	75	K
Flow		
Minimum flow	7	L/min
Maximum flow	12	L/min
Pump pressure head	8.75	m
Rotation speed	3200	rpm

5. Control system

For the levitation control we use dSPACE based digital PID controller. The control system is shown in Fig. 14.

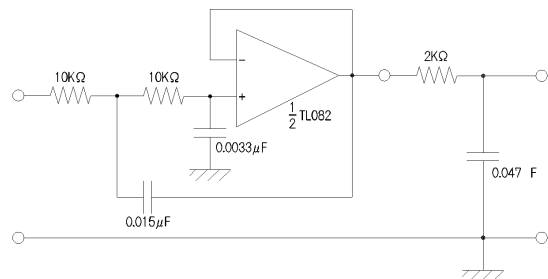


Fig. 15 Third order anti-aliasing filter

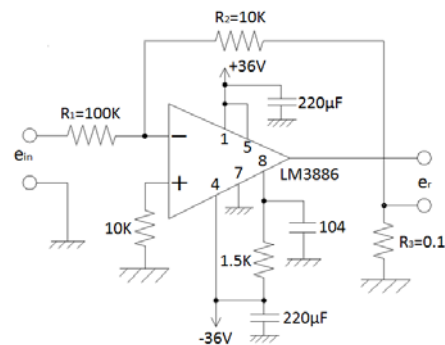


Fig. 17 Power amplifier circuit



Fig. 18 Photo of magnetically levitated pump

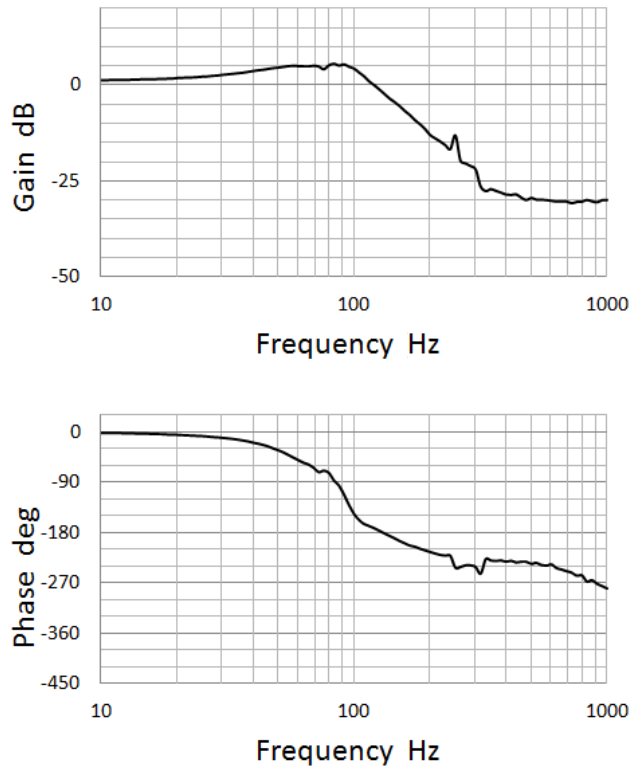


Fig. 19 Frequency response in x-direction

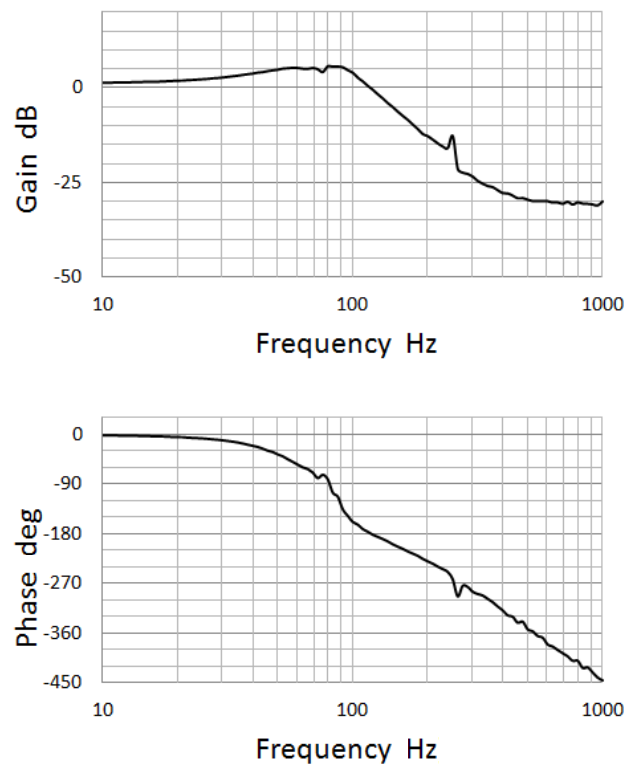


Fig. 20 Frequency response in y-direction

Radial x-, y-displacements are measured by eddy current sensors (AEC-5509) and put into the dSPACE (DS-1104) via third order anti-aliasing filters (shown in Fig. 15) and A/D converters. The control gains are determined experimentally the values of which are shown in Table 2. Sampling interval used is $\tau = 0.1$ ms and the derivative time constant is 0.8 ms. The controlled plant is shown in Fig. 16 which is composed of pump and radial magnetic bearing. The calculated actuating signals are put out through D/A converters to the analog power amplifiers which are shown in Fig. 17. Thanks to high force factor of the proposed magnetic bearing the power amplifier consumes very low power.

6. Levitation test of the pump

To confirm the developed magnetic bearing a simple pump setup is fabricated and the levitation test is carried out. The lower part of pump shaft is successfully levitated without mechanical contact. The low temperature liquid is not go up to the mechanical coupling nor to the motor. The clearance between the upper shroud of pump and the pump housing is about 0.5 mm which produces enough fluid resistance for sealing the liquid. Hence mechanical sealing is not necessary.

6.1 Pump test setup

The developed pump is shown in Fig. 18. This is planned to apply to the liquid nitrogen. However, water pumping is tested due to the university laboratory limitation. Due to the same limitation the pump pressure head is limited to 0.7 m while the actual pump has the nominal head of 8.75 m. The nominal pump parameters are shown in Table 3. Even though the tested pressure is low the levitation of the pump is stable. The driving motor used in this experiment is a DC motor (Maxon 148877).

6.2 Stability test of levitated pump

First, the frequency responses are measured without rotation. The rotor is levitated with the filled water. The sinusoidal signal of ± 0.03 mm magnitude from 10 to 1000 Hz is put into the error signal and the corresponding

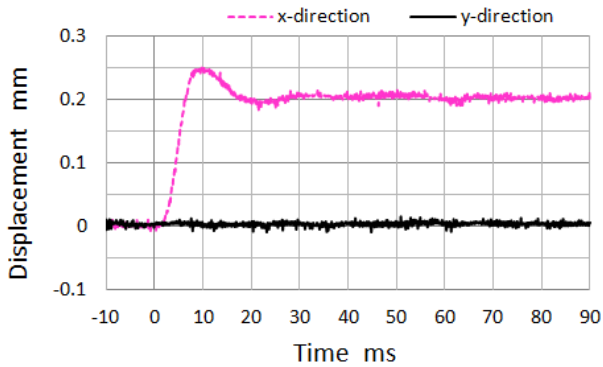


Fig. 21 Step response (x-direction excitation)

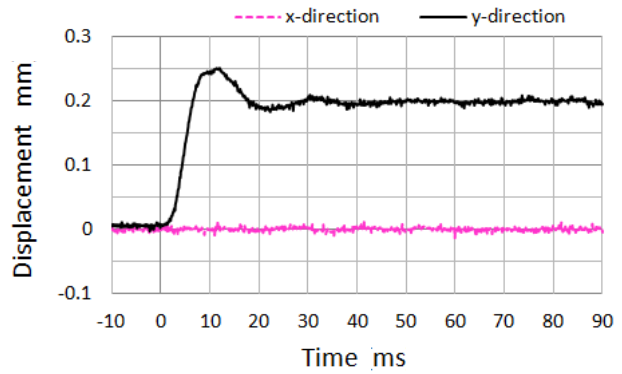


Fig. 22 Step response (y-direction excitation)



Fig. 23 Photo of pumping test

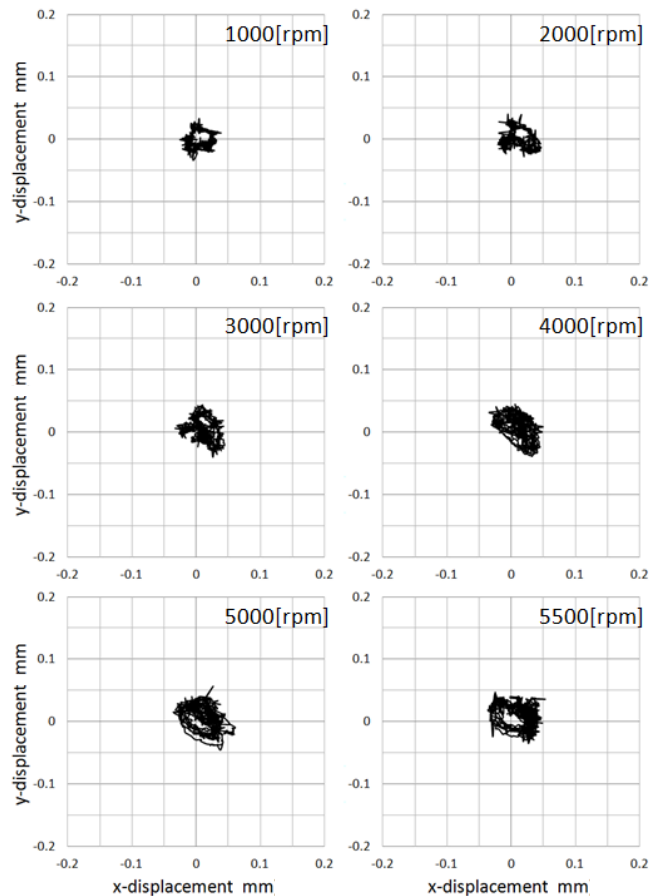


Fig. 24 Orbital trajectory during pumping test

levitated displacement is measured. The frequency response analyzer (NF FRA5095) is used for measurement. The frequency responses of x- and y-directions are shown in Figs. 19 and 20, respectively. The maximum peak is about 5 dB around 85 Hz. The responses are almost well reduced with the PID controller except of the small peaks around 75 Hz. This is due to the poor support stiffness of the stator housing with narrow shafts. This might be improved by improving the support stiffness. Also the phase difference between Figs. 19 and 20 occurs over the frequency range of 300 Hz and the corresponding gains are lower than -25 dB. This may be caused due to the higher frequency difference of electrical properties in the x- and y-directional stator coils. This difference is not serious because of high frequency and the low gain region.

Next, the step response test is carried out. The step signal of 0.2 mm magnitude is added to the error signal and the corresponding displacements are measured and recorded by the digital oscilloscope (Yokogawa DL 708E). The x- and y-directional step responses are shown in Figs. 21 and 22, respectively. Figure 21 is the response of x-directional

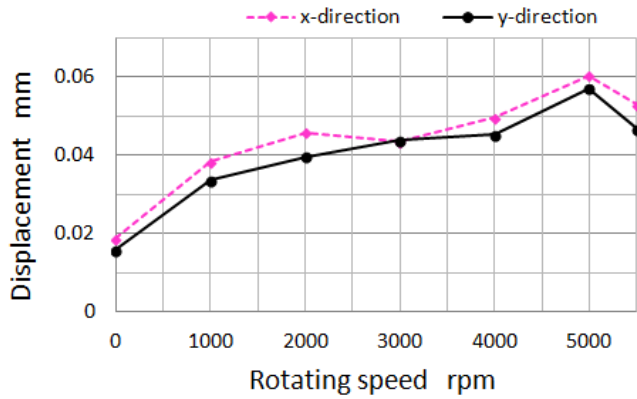


Fig. 25 Vibration amplitude during orbital trajectory measurement

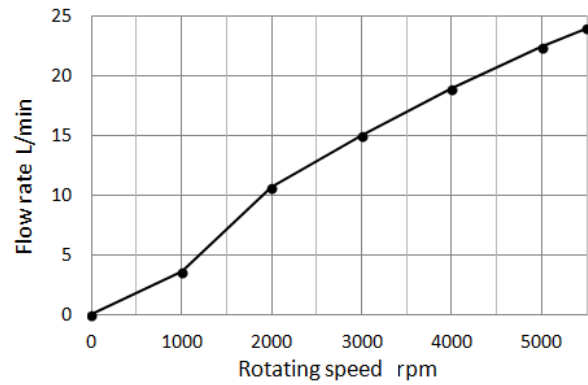


Fig. 26 Flow rate versus rotating speed

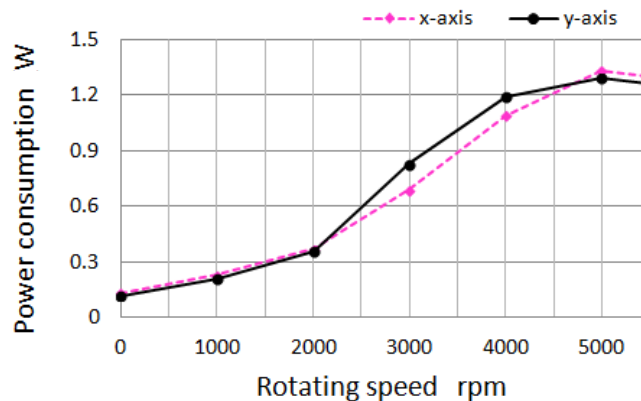


Fig. 27 Power consumption of magnetic bearings versus rotating speed

excitation while Fig. 22 is the response of y-directional excitation. Both responses are well reduced within 30 ms. The cross coupling between x- and y-directions is not recognized in these responses. These responses show good and stable levitation characteristics.

7. Pumping test results

Stable levitation is realized during the pumping test as shown in Fig. 23. The unbalance responses from 1,000 to 5,500 rpm are measured and the orbital trajectories are recorded as shown in Fig. 24. The levitated rotation is stable under the maximum vibration of 60 μm . From these responses the x- and y-directional vibration amplitudes are measured and recorded as shown in Fig. 25. The displacements are well controlled by the proposed PID controller. The flow rate versus rotating speed is shown in Fig. 26 while the x- and y-directional power consumptions of magnetic bearings are shown in Fig. 27. The nominal flow rate of 12 L/min. is recorded about the rotating speed of 2500 rpm. Vibration level is under 60 μm up to 5500 rpm. The x- and y-directional power consumptions of magnetic bearings are lower than 1.3 W. This means high capability of the developed magnetic bearing to apply to the ultra-low temperature fluid pump.

8. Concluding remarks

In this paper highly efficient magnetic bearing is developed which is planned to apply to ultra-low temperature fluid pump. The developed magnetic bearing can support the pump impeller successfully up to 5500 rpm over the rated operating speed of 3200 rpm. The power consumption recorded is under 1.3 W which is considered very low level. Further research work is continuing to apply the developed pump to high pressure output and actual fluid such as liquid nitrogen or liquid helium.

Acknowledgement

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