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Design and Experiment of 300-HP-Class Turbo Compressor with Hybrid Magnetic Bearings

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Abstract

High-capacity and high-efficiency turbomachinery, such as turbo blowers and turbo compressors, have been developed recently. In order to keep up with these trends, magnetic bearings are being increasingly applied to turbomachinery instead of air foil bearings. In this study, hybrid magnetic bearings are applied to a 300-HP-class turbo compressor, and a no-load test and full load test were performed to evaluate the performance of the designed magnetic bearings. The rotor speed could be increased up to 51,000 rpm in the no-load test, and 45,000 rpm in the full load test. The results indicated a low level of rotor vibration and stable performance of the magnetic bearings in both the tests. Owing to the cooling system that uses water and air, the problem of thermal expansion of the rotor in the axial direction. could be avoided. The estimated aerodynamic thrust force is approximately 407 N at 45,000 rpm under the full load condition; thus, it is verified that the thrust magnetic bearings have sufficient load capacity.

Keywords : Hybrid magnetic bearings, Turbo compressor, Load test, Rotor dynamics, Thrust force

1. Introduction

High-capacity and high-efficiency turbomachinery, such as turbo blowers and turbo compressors, have been developed recently. In order to keep up with these trends, magnetic bearings are being increasingly applied to turbomachinery instead of air foil bearings.(Park, et al, 2013a, Park, et al. 2014, Park, et al 2015) The employment of magnetic bearings to turbomachinery is essential because the density of the working fluid is higher, whereas the viscosity is lower under the operating environment of supercritical CO_2 (S-CO_2). (Lee, et al. 2014, Seo, et al. 2015) In this study, hybrid magnetic bearings are applied to a 300-HP-class turbo compressor, and experiments are performed to evaluate the performance of the designed magnetic bearings under load condition.

2. Design of Hybrid Magnetic Bearings for Turbo Compressor

Fig. 1(a) shows the configuration of the 300-HP-class (225 kW) turbo compressor. It consists of upper and lower impellers, a shaft, magnetic bearings, and a PMSM (permanent magnet synchronous motor). It is designed to have a rated speed of 50,000 rpm, flow rate of 0.76 kg/s, pressure ratio of 3.5, and an aerodynamic efficiency of 82%. The motor stator is located in the middle of the compressor housing, which has a water cooling channel to cool down the motor stator. In addition, because the temperature of the rotor as well as the motor stator increases during high speed operation, the inlet and outlet are designed to supply cooling air with a maximum flow rate of 0.3 kg/s for cooling both the rotor and motor stator. The upper and lower radial bearings are located above and below the motor, respectively, and the thrust magnetic bearing is located between the lower radial bearing and lower impeller. The rotor is placed vertically so that the effect of gravitational force on the radial magnetic bearings can be minimized, and the required force for the radial magnetic bearings can be reduced. The thrust magnetic bearing is designed to have a load capacity of 2,000 N based on the type of AM-HMB (axially magnetized-hybrid magnetic bearing) as shown in Fig. 1 (b), and the radial magnetic bearings are designed to have a load capacity of 200 N based on the hybrid homo-polar type magnetic bearing, as shown in Fig. 1(c).(Park, et al 2013b, Park, et al, 2014) The design parameters for thrust and the radial magnetic bearings are listed in Tables 1 and 2, respectively. Here, l_p is the axial thickness of the ring magnet, A is the area of the upper or lower pole faces, g_0 is the air gap length between the upper pole face and the thrust collar, g_1 is the air gap length between the outer side of the thrust collar and the inner side of the ring magnet, and N_t is the number of coil turns for thrust magnetic bearing. T_r is the thickness of the radial permanent magnet, A_r and g_r denote the pole face area and the air gap length between the radial stator core and the rotor core, respectively, and N_r is the number of coil turns of the radial magnetic bearing.

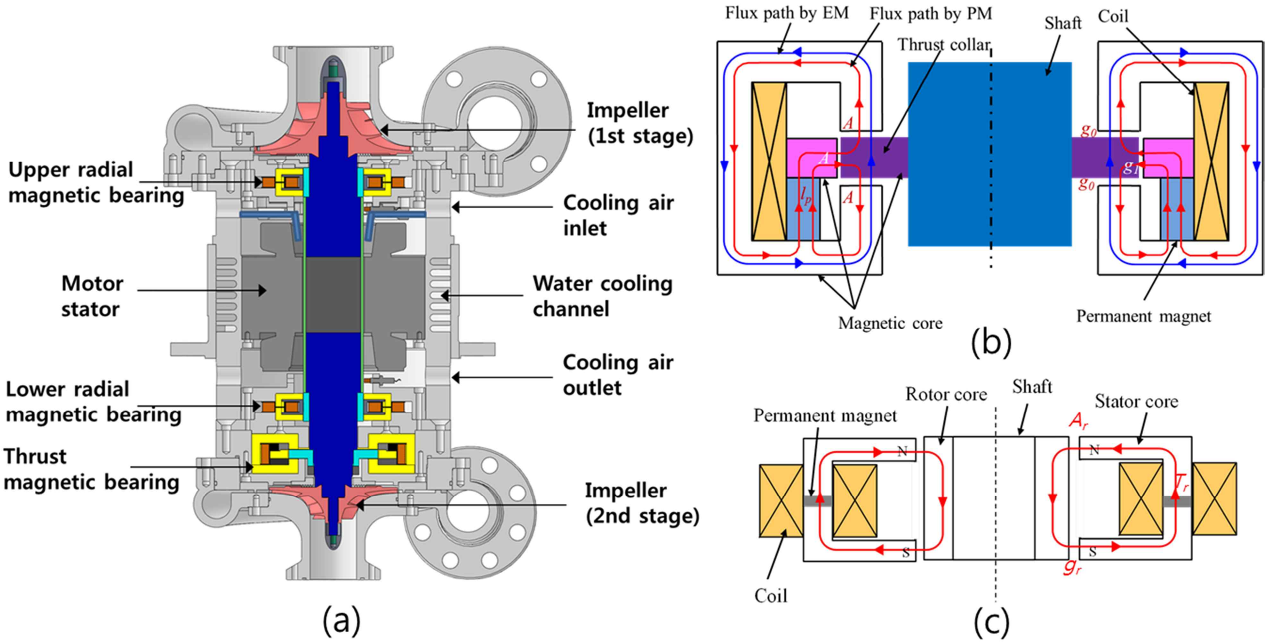


Fig. 1 Configuration of 300-HP-class turbo compressor and hybrid magnetic bearings

Table 1 Design parameters for thrust magnetic bearing

Item	Value
Load capacity [N]	-2,000–2,000
l_p [mm]	10
A [mm^2]	2,950
g_0 [mm]	0.6
g_1 [mm]	1.5
N_t [turn]	100
Current gain, K_{ti} [N/A]	230
Position gain, K_{tx} [N/m]	-8,57$\times 10^4$
Permanent Magnet	NdFeB (N40SH)

3. Rotor dynamic analysis

The parameters of the design rotor are listed in Table 3. The frequency of the free-free first bending mode is predicted to be at 1,269 Hz. Rotor dynamic analysis was performed using commercial program, xlrotor, and the predicted damped natural frequency map is shown in Fig. 2. It is predicted that the critical speed at the rigid body modes would be less than 5,000 rpm, and that at the backward first bending would be 70,000 rpm, which has a sufficient margin from the rated speed of 50,000 rpm.

Table 2　Design parameters for radial magnetic bearing

Item	Value
Load capacity [N]	-200–200
T_r [mm]	1.0
A_r [mm^2]	260
g_r [mm]	0.4
N_r [turn]	100
Current gain, K_{ri} [N/A]	35
Position gain, K_{rx} [N/m]	-3,34$\times 10^5$
Permanent Magnet	NdFeB (N40SH)

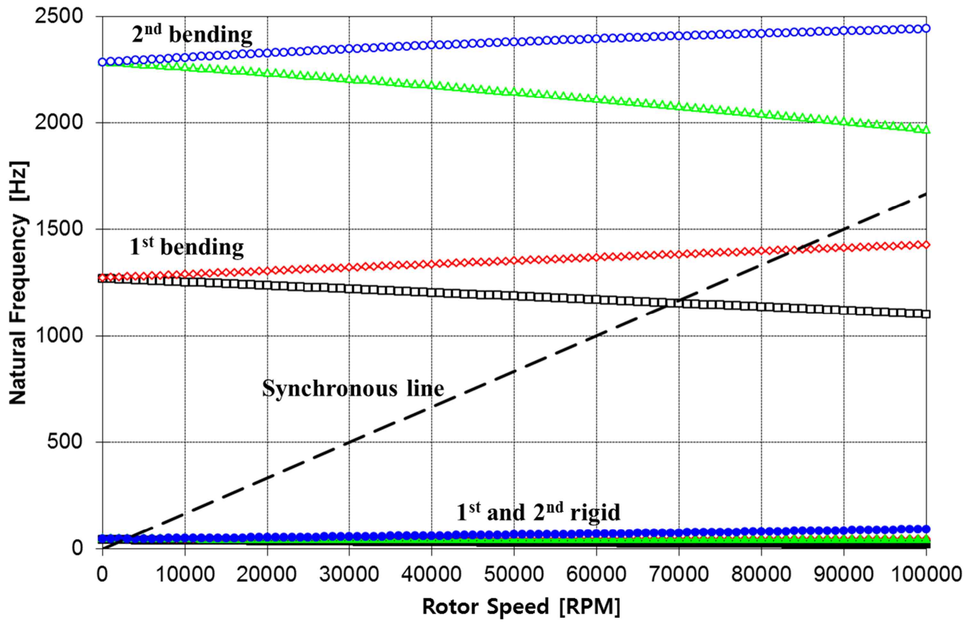


Fig. 2 Predicted damped natural frequency map

Table 3　Design parameters for rotor

Item	Value
Length of rotor [mm]	525
Outer diameter [mm]	70.2
Length of magnet [mm]	88.5
Outer diameter of magnet [mm]	64
Material of shaft stud	SUS304
Material of sleeve	Inconel718
Material of impellers	Ti-6Al-4V
Mass of rotor [kg]	13.0

4. Fabrication of turbo compressor with hybrid magnetic bearings

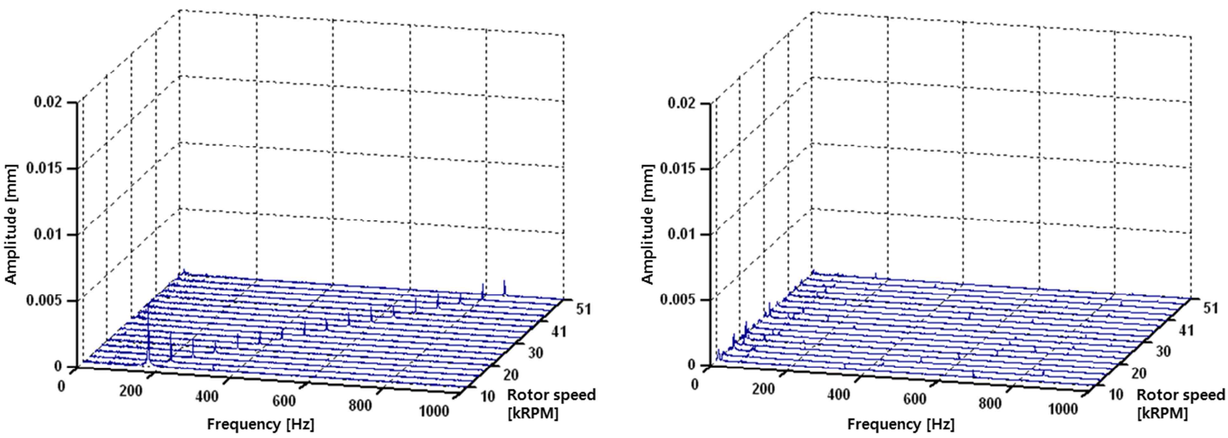
Fig. 3 shows the fabricated hybrid magnetic bearings and the rotor. The impact test was performed for the rotor, including the impellers, and it was found that the frequency of the free-free first bending mode was 1,236 Hz, which is similar to the predicted result. The magnetic bearing controller is composed of a PID (proportional-integral-derivative) controller, lead compensator, low pass filter, and UFRC (unbalance force rejection controller) with a sampling frequency of 10 kHz. The PID gains were optimized to levitate the rotor stably and achieve a phase margin greater than 20°. The load capacity of the thrust magnetic bearing was evaluated by axially stacking weights on the rotor while the rotor is fully levitated. As the load is increased up to 1,100 N, the current supplied to the thrust magnetic bearing was also linearly increased to 4.5 A. Based on this result, it can be predicted that the maximum load capacity of the thrust magnetic bearing at a current of 10 A would be more than 2,000 N, which is the required load capacity.

(a) (b) (c) (d)

Fig. 3 Manufactured components. (a) Radial magnetic bearing, (b) Thrust magnetic bearing, (c) Rotor with impellers, (d) Assembled turbo compressor

5. Experimental results of turbo compressor

First, the motoring test was performed under no-load condition for a rotor without an impeller to evaluate the structural strength of the shaft. The rotational speed of the shaft can be stably increased up to 51,000 rpm, and it was verified that the shaft had a good structural strength, and the magnetic bearing can stably support the shaft across the full operating range. The unbalance responses of the upper and lower magnetic bearings at 51,000 rpm were less than 1 μm and 4 μm, respectively, and the axial vibration was less than 0.5 μm, as shown in Fig. 4.



(a) (b)

Fig. 4 Waterfall plots of rotor vibrations in turbo compressor at a speed of 51,000 rpm under no-load condition. (a) Upper radial vibrations, (b) Axial vibrations

Next, the motoring test was performed under load condition for a rotor with impellor. The experimental setup is shown in Fig. 5. Under the load condition, as the rotational speed increased, the motor input power increased rapidly, and the temperature of the motor stator and rotor also increased. Because the thermal expansion of the rotor in the axial direction owing to the eddy current loss of the motor can cause the impeller to crash into the shroud touch resulting in an accident, it is very important to cool down the temperature of the motor stator and rotor.(Park, et al. 2015) The flow rate of cooling air was controlled to maintain the temperature of the motor stator below 100℃. For additional protection in this experiment, a laser displacement sensor was installed at the upper volute and targeted the top of the rotor to monitor the thermal expansion of the rotor in the axial direction. The aerodynamic load was controlled by adjusting the valves, which are installed at the upper and lower air outlet pipes. Under the full load condition, the rotor speed could be increased up to 45,000 rpm, and the motor input power at this speed was 310 HP (231 kW). The measured motor input power with respect to the rotor speed is shown in Fig. 6. The thermal expansion of the rotor in the axial direction was maintained at approximately 0.1 μm at this speed. The unbalance responses of the upper and lower magnetic bearings at 45,000 rpm were less than 8 μm and 13 μm, respectively, and the axial vibration was less than 5 μm, as shown in Fig. 7. The aerodynamic thrust force can be calculated by multiplying the current input to the thrust magnetic bearing with the current gain, 230 [N/A] listed Table 1. The current input to the thrust magnetic bearing was increased from -0.64 A at standstill to 1.13 A at 45,000 rpm, and it can be estimated that the aerodynamic thrust force at 45,000 rpm acts upward with a magnitude of approximately 406 N. Based on these results, it is verified that the designed hybrid magnetic bearings could support the turbo compressor stably, and the thrust magnetic bearing has a sufficient load capacity,.

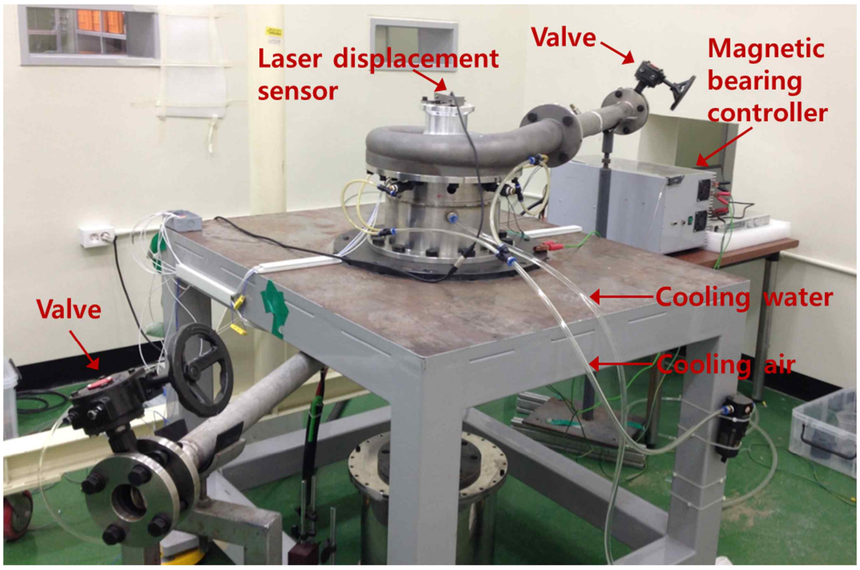


Fig. 5 Experimental setup under load condition

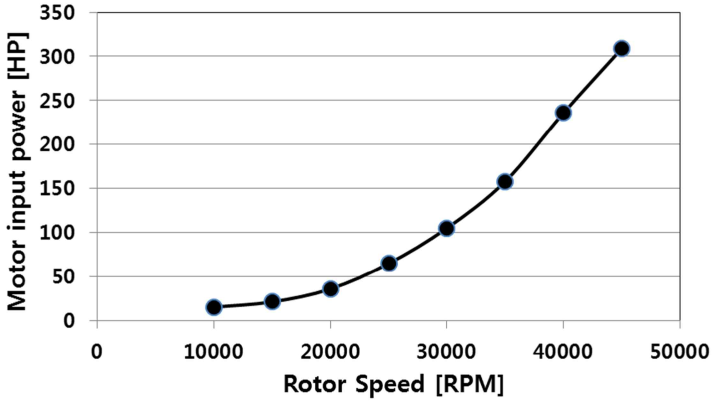
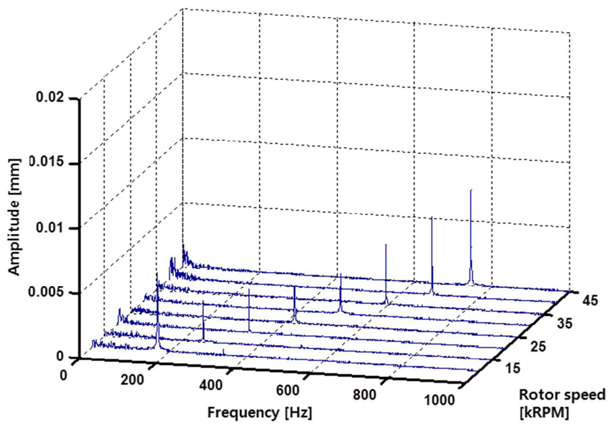
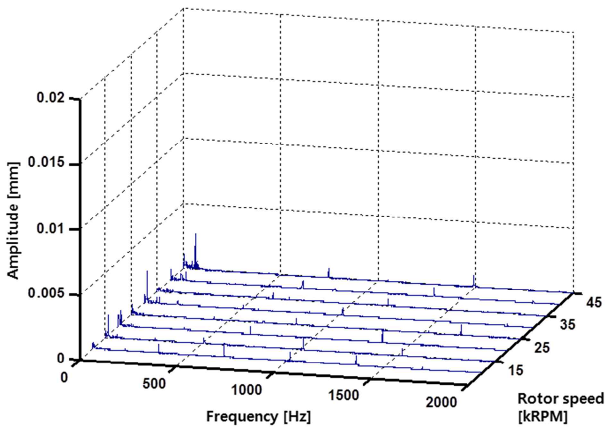


Fig. 6 Measured motor input power with respect to rotor speed



(a)



(b)

Fig. 7 Waterfall plots of vibrations in turbo compressor up to 45,000 rpm under load condition. (a) Upper radial vibrations, (b) Axial vibrations

6. Conclusion

In this study, hybrid magnetic bearings were applied to a 300-HP-class turbo compressor, and no-load and full load tests were performed to evaluate the performance of the designed magnetic bearings. The rotor speed could be increased up to 51,000 rpm in the no-load test, and 45,000 rpm in the full load test. The results indicated a low level of rotor vibration and stable performance of the magnetic bearings in both the tests. Owing to the cooling system that uses water and air, the problem of thermal expansion of the rotor in the axial direction could be avoided. The estimated aerodynamic thrust force is approximately 407 N at 45,000 rpm under full load condition; thus, it is verified that the thrust magnetic bearing has a sufficient load capacity. In the future, the performance of the turbo compressor would be evaluated by controlling the valves, and studies to improve the completeness of the compressor would be continued.

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