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EVALUATION OF VIBRATION IN TURBOMACHINERY FOR SUPERCRITICAL CO_2 SYSTEM SUPPORTED BY HYBRID MAGNETIC BEARINGS

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Recently, supercritical CO_2 (S-CO_2) system for power generation is receiving attentions as a new technology to replace nuclear power generation technologies. S-CO_2 technology has advantages in high efficiency, compact size compared to the existing steam power generations. Under the operating environments of S-CO_2, the density of the working fluid is higher, whereas the viscosity is lower, the employment of the magnetic bearings to the turbo machines is essential. In this study, the hybrid magnetic bearings are applied to turbo machines such as a pump-driving turbine and a turbo compressor for S-CO_2 system, of which rated speed is 21,000 rpm and 50,000 rpm, respectively. The experiments are performed to evaluate the vibration of each turbo machine under no-load and load condition. The noticeable vibration component was only synchronous vibration for both machines, and their magnitude was less than 2 um and 18 um, respectively. Based on the evaluated vibration, it was verified that the hybrid magnetic bearings could support the turbo machines for S-CO_2 system.
Keywords: S-CO_2, turbo compressor, magnetic bearings

1. Introduction

Supercritical CO_2 system for power generation is receiving attentions because it has advantages in high efficiency, compact size compared to the existing steam power generations [1-7]. The S-CO_2 power generation system requires several turbo machines, such as a power turbine, a pump-driving turbine, and a compressor. Under the operating environments of S-CO_2, the density of the working fluid is higher, whereas the viscosity is lower, the employment of the magnetic bearings to the turbo machines is essential [8]. In this study, the hybrid magnetic bearings are applied to turbo machines such as pump-driving turbine and turbo compressor for S-CO_2 system, of which rated speed is 21,000 rpm and 50,000 rpm, respectively. The hybrid magnetic bearings were designed according to the load specification of each turbo machine, and the test rigs were prepared for each machine to evaluate the stability of rotation. The following chapters 2 and 3 present the design procedure and experimental results for the pump-driving turbine and the turbo compressor, respectively.

2. 143 kW class pump-driving turbine

Figure 1 shows the configuration of the 143 kW class pump-driving turbine. It consists of a turbine impellor, a pump impellor, a shaft, a thrust magnetic bearing and two radial magnetic bearings.

It is designed to have a pumping power of 143 kW at the rated speed of 21,000 rpm. The upper and lower radial bearings are located right inside the turbine and pump impellor, respectively, and the thrust magnetic bearing is located in the middle of the shaft between the two radial bearings. 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 force acting on the rotor-shaft was predicted by the flow analysis. As a result, it was predicted to be 6.9 kN from pump to turbine direction and 0.5 kW from turbine to pump direction, respectively. Therefore, it was predicted that a net thrust force of 6.4 kN would be applied in the direction from pump to turbine. Based on this prediction, the thrust magnetic bearing was designed to have the load capacity of 10,000 N based on the type of AM-HMB (Axially magnetized-Hybrid magnetic bearing) as shown in [9-11], and the radial magnetic bearings are designed to have the load capacity of 200 N based on the hybrid homo-polar type magnetic bearing as shown in [10-12]. The design results show that the diameter of the shaft is 70.2 mm, the diameter of the thrust collar is 176 mm, the length of the rotor-shaft is 467 mm, and the outer diameter of the thrust bearing is 356 mm. The critical speed analysis for the designed rotor-shaft was performed, and the first forward and backward bending mode critical speeds were predicted at 69,000 rpm and 45,000 rpm, respectively. The bending mode critical speeds are far enough away from the rated speed of 21,000 rpm. Figure 2 shows the radial magnetic bearings, thrust magnetic bearing, and rotor-shaft. The impact test was performed for the rotor-shaft, and it was found that the frequency of the free-free first bending mode was 1,166 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°.

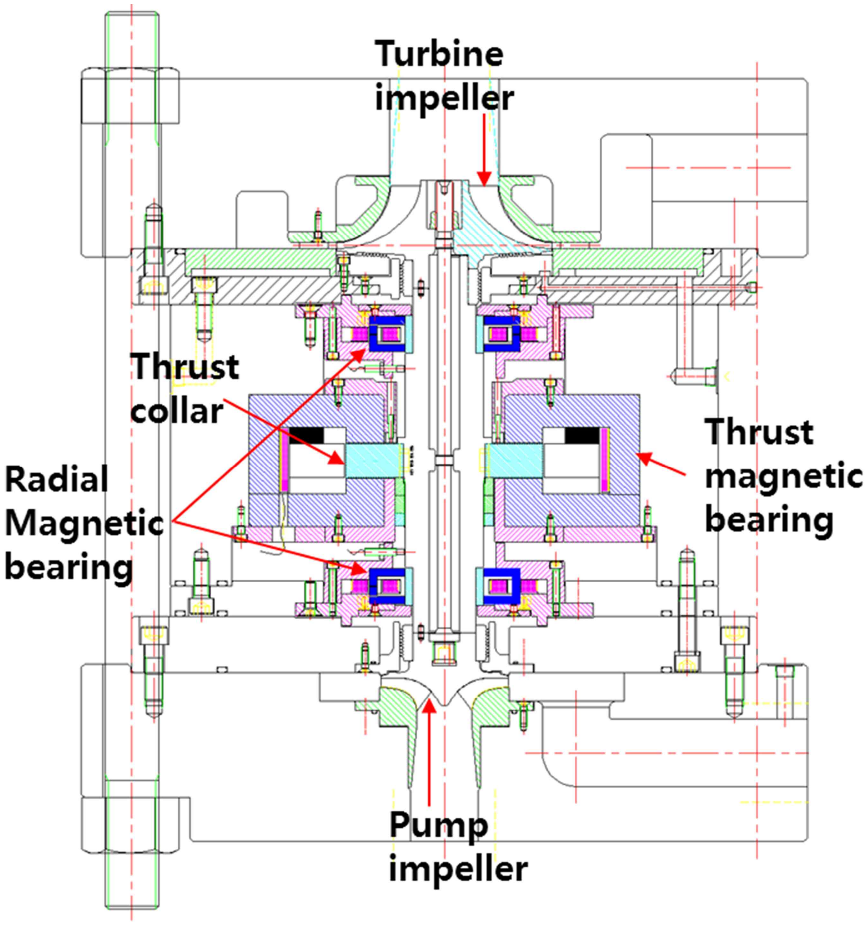

Figure 1: Configuration of Pump-driving Turbine

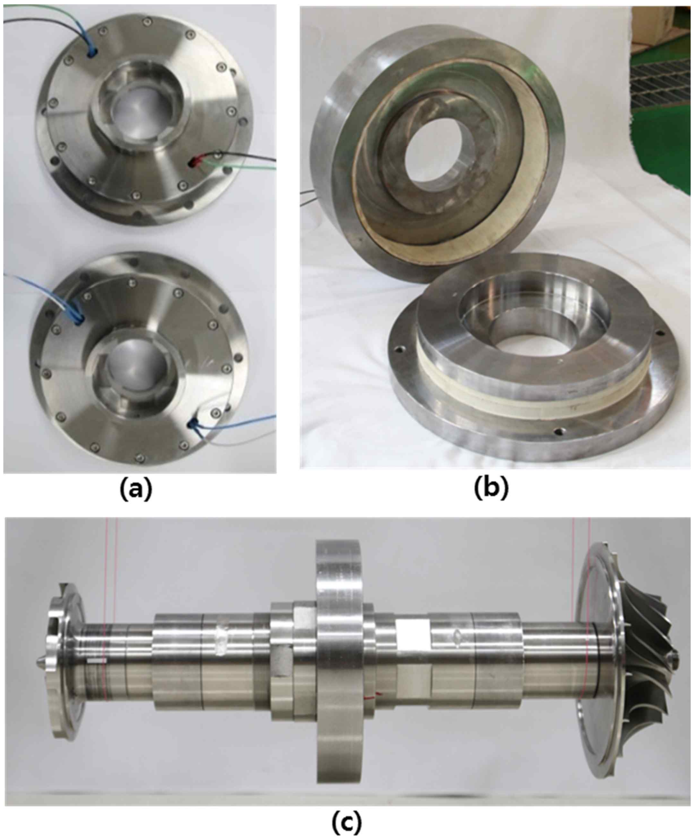

Figure 2: Manufactured (a) Radial Magnetic Bearings, (b) Thrust Magnetic Bearing, (c)Rotor for Pump-driving Turbine

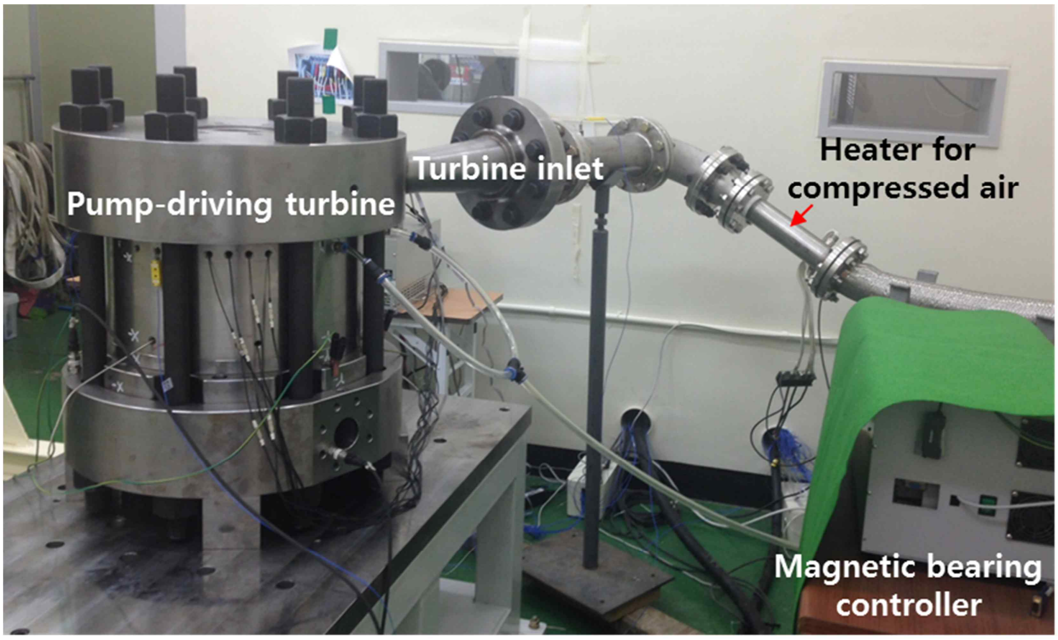

Figure 3: Experimental setup for Pump-driving Turbine under High Temperature Condition

As shown in Fig. 4, the experimental setup was constructed to evaluate the rotational stability of the pump-driving turbine under high temperature condition. At the time of the experiment, the loop which can supply S-CO_2 was not prepared, and the turbine was driven by supplying the hot air. A heater was installed in the middle of the air supply pipe to heat the supplied air to 250 °C. Figure 4 shows the waterfall plot of the vibration measured while increasing the speed of the pump-driving turbine to the rated speed of 21000 rpm. The unbalance responses of the upper and lower magnetic bearings at 21,000 rpm were less than 2 μm and 0.2 μm, respectively, and the axial vibration was less than 0.4 μm. The radial and axial vibrations remained constant and stable during the long time operation for more than 6 hours. The surface temperature of the magnetic bearing was monitored while performing the long running test. After 4 hours from the test start, the temperature increased up to 100 °C, but the temperature was maintained constantly thereafter. Based on these experimental results, it was confirmed that the pump-driving turbine with hybrid magnetic bearings can be operated for a long time even under high temperature condition.

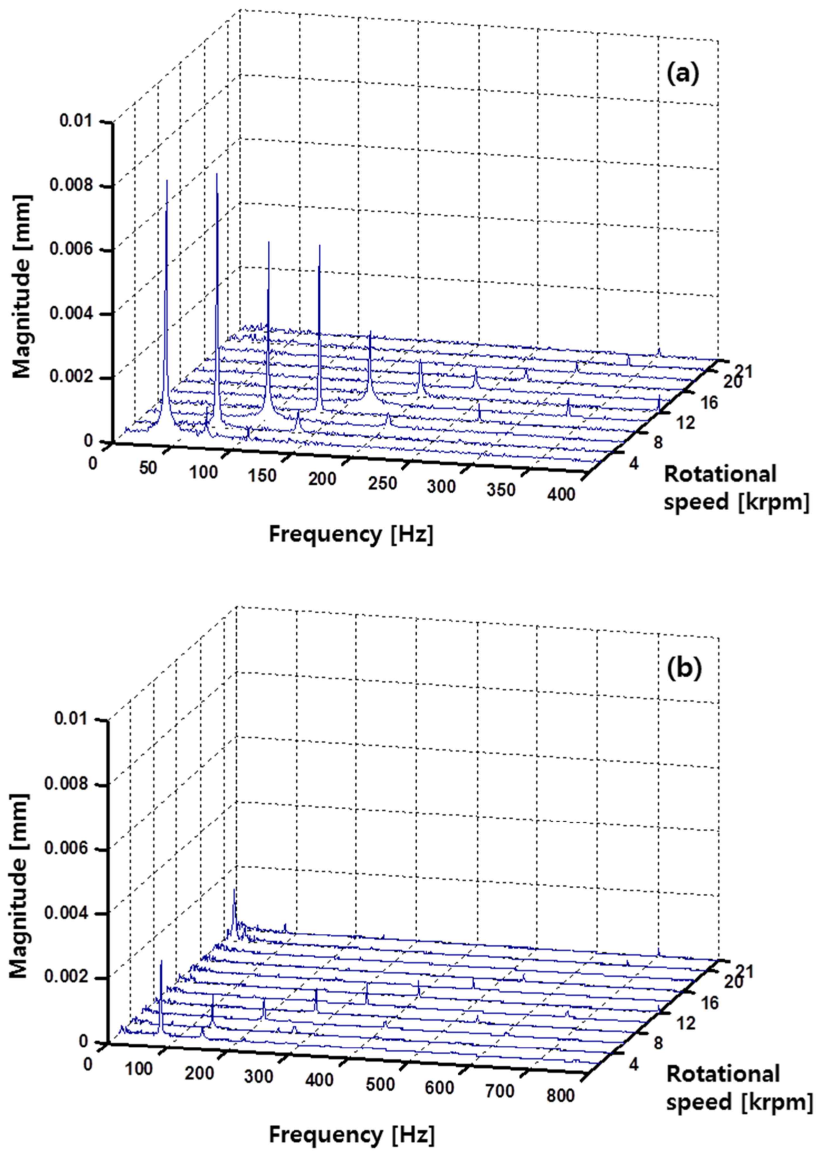

Figure 4: Waterfall Plot of vibrations in Pump-driving Turbine up to 21,000 rpm. (a) Radial Vibration in Lower Radial Bearing, (b) Axial Vibration

3. 225 kW class turbo compressor

Figure 5 shows the configuration of the 225 kW (300 HP) class turbo compressor. It consists of upper and lower impellors, 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. 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 impellor. The rotor is placed vertically for the same reason as the pump-driving turbine. The thrust magnetic bearing is designed to have a load capacity of 2,000 N, and the radial magnetic bearings are designed to have a load capacity of 200 N. The topologies of the thrust and radial magnetic bearings are the same as for the pump-driving turbine. The design results show that the diameter of the shaft is 70.2 mm, the diameter of the thrust collar is 104 mm, the length of the rotor-shaft is 525 mm, and the outer diameter of the thrust bearing is 191 mm. The critical speed analysis for the designed rotor-shaft was also performed, and the first forward and backward bending mode critical speeds were predicted at 84,000 rpm and 68,000 rpm, respectively. The bending mode critical speeds are far enough away from the rated speed of 50,000 rpm.

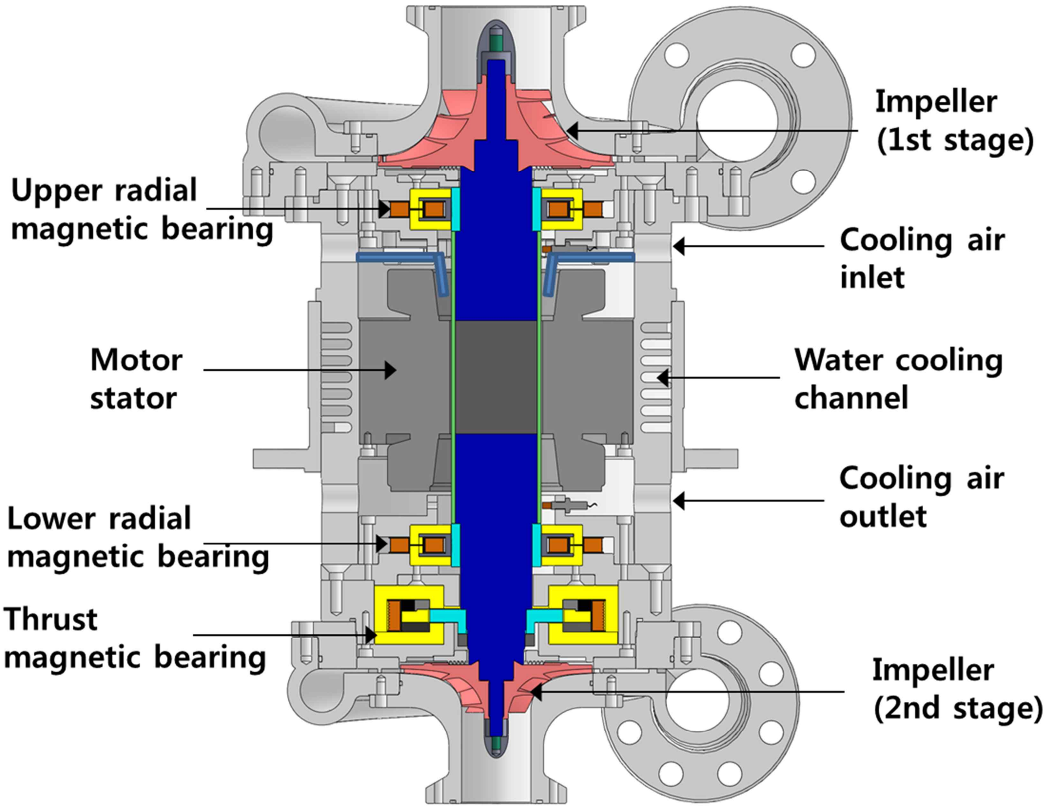

Figure 5: Configuration of 300 HP class Turbo Compressor

Figure 6 shows the radial magnetic bearings, thrust magnetic bearing, and rotor-shaft. The impact test was performed for the rotor-shaft, and it was found that the frequency of the free-free first bending mode was 1,225 Hz, which is similar to the predicted result. The magnetic bearing controller was optimized to meet the same criteria as for the pump-driving turbine. 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.

Before the load test with impellors, the no-load test without impellors (with shaft only) was performed 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.

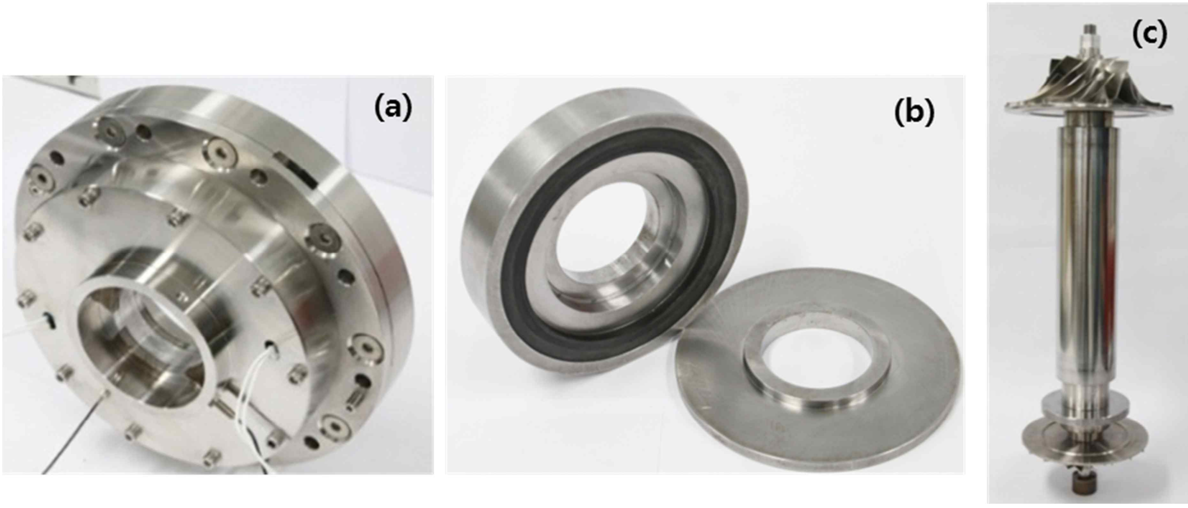

Figure 6: Manufactured (a) Radial Magnetic Bearings, (b) Thrust Magnetic Bearing, (c)Rotor for 300 HP class Turbo Compressor

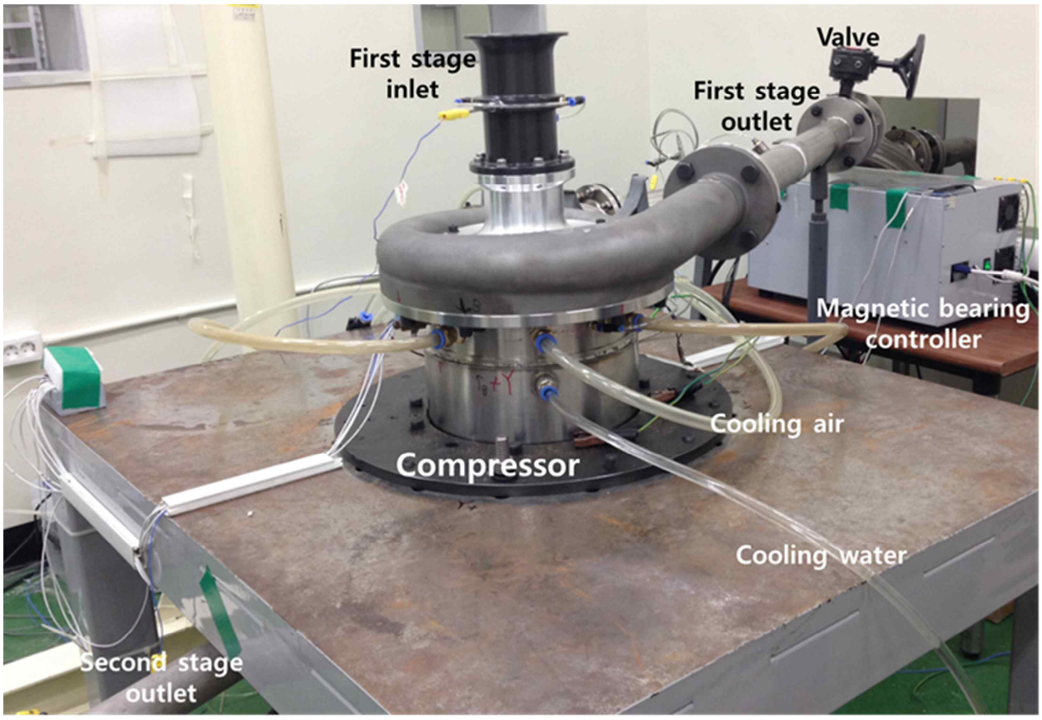

Figure 7: Experimental setup for 300 HP class turbo compressor

The experimental setup for the full load test is shown in Fig. 7. The motoring test was performed under load condition for a rotor with impellor. 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. The flow rate of cooling air was controlled to maintain the temperature of the motor stator below 100℃. 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 50,000 rpm, and the motor input power (Inverter output power) at this speed was 250 kW). 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 50,000 rpm were less than 12 μm and 18 μm, respectively, and the axial vibration was less than 4 μm, as shown in Fig. 8. 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.

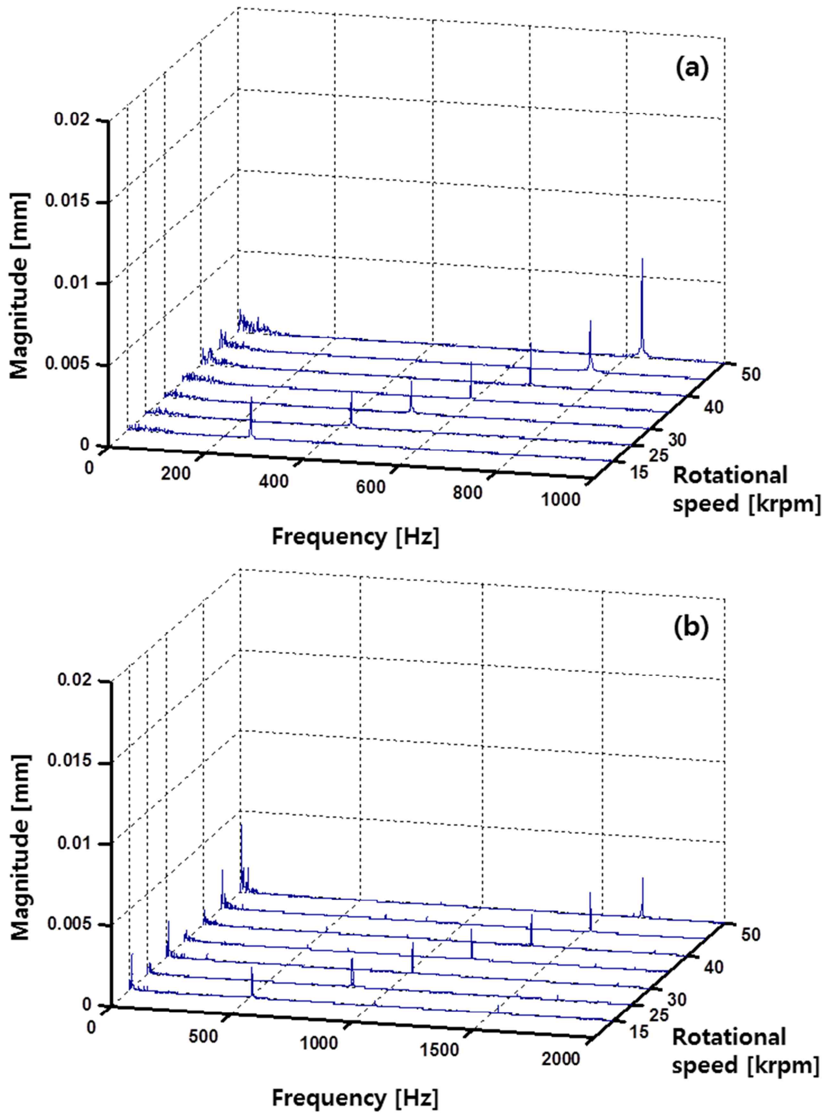

Figure 8: Waterfall plot of vibrations in Turbo compressor up to 50,000 rpm. (a) Radial vibration in upper radial magnetic bearing, (b) Axial vibration

4. Conclusion

In this study, the hybrid magnetic bearings are applied to turbo machines such as the 143-kW-class pump-driving turbine and the 225-kW-class turbo compressor for S-CO_2 power generation system, of which rated speed is 21,000 rpm and 50,000 rpm, respectively. The experiments are performed to evaluate the vibration of each turbo machine under load conditions. The noticeable vibration component was only synchronous vibration for both machines, and there magnitude was less than 2 um and 18 um, respectively. Based on the evaluated vibration, it was verified that the hybrid magnetic bearings could stably support the turbo machines for S-CO_2 power generation system.

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