Rotordynamic Design and Control of Three-Stage Centrifugal Compressor with Magnetic Bearings

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Abstract

Demand for turbo chillers is increasing for cooling facilities such as large buildings and data centers, and the application of oil-free technology, known for its eco-friendly high efficiency, is expanding to large scale projects. In this study, a large capacity magnetic bearings and rotor dynamics design are described for the development of an oil-free turbo chiller with 2400-3000 USRT performance. First, the rotor and magnetic bearings supporting the three-stage impeller of the large capacity compressor were designed and developed. A constant speed motor rotating at 50 Hz was applied, and the shaft was designed to be long and heavy because of the high load aerodynamic design at low rotation speed. Therefore, the focus was placed on designing sufficient separation margin for the first bending mode. Second, it was important to design the stability of the magnetic bearings with notch filters and PID position control. It is necessary to verify the stability of the bearing support over a wide load range, as efficiency must be maintained not only under maximum load but also during load fluctuations. The system identification of magnetic bearings was used to optimize the controller, and the control stabilization was satisfied through the operation data. Based on these studies, the operation of large capacity turbo chillers was stabilized and the product was successfully commercialized. The chiller introduced in this study was installed in the Middle East and proved its performance by applying a large capacity magnetic bearing for the first time in the oil-free turbo chiller industry.

Keywords: *Centrifugal compressor, Oil-free turbo chiller, Active magnetic bearings, Rotor dynamics*

1. Introduction

The magnetic bearing chiller is an efficient, reliable and cost-effective solution for cooling applications. It utilizes a non-contact bearing system which eliminates the need for lubrication, resulting in reduced maintenance costs and improved efficiency. For this reason, in the HVAC market, demand for products with various capacities to which magnetic bearings are applied is increasing as an alternative to existing oil bearing chillers and screw chillers. In particular, in Middle Eastern countries, where it is difficult to utilize cooling towers due to water shortage, there is a high demand for large air-cooled chillers instead of water-cooled chillers. If oil is used in an air-cooled chiller, oil accumulation in the dry condenser is likely to cause failure. Therefore, in order to build a district cooling system in the Middle East, zero water consumption solutions and oil-free technology are required. That is the need for expansion of oil-free magnetic bearing technology, and it will be of great help to the development of the air-cooled turbo chiller industry in the Middle East. LG Electronics participated in the development of a 2400-3000 USRT class magnetic bearing turbo chiller for installation in Middle Eastern countries. Based on the results of commercialization of a 500-1000 USRT class water-cooled turbo chiller (Noh et al., 2018), we challenged a large capacity air-cooled chiller. In general, expensive high voltage inverter motors are used above 1000 USRT, and constant speed systems using gears have the disadvantage of requiring a separate oil system for gear cooling and lubrication. We tried to implement a system driven directly by a constant speed motor, utilizing a new aerodynamic design that could increase the compression capacity at low constant speed. The goal is to build a constant speed system that does not use a high voltage inverter and develop an oil-free compressor equipped with a large capacity magnetic bearings. With this oil-free technology applied, it is expected to expand its technological advantage and market share in the air-cooled chiller market. In addition, because oil is not used, gear loss and heat exchange loss are reduced, so energy will be saved.

The configuration of the 3000 USRT class oil-free turbo chiller is shown in Fig. 1. The chiller size has a length of 9,355mm, a width of 3,912mm and a height of 4,218mm. Two compressors are fixed on the heat exchanger, and the total weight is about 67 tons. Two compressors are connected in series, and it is a system that compresses a total of five-stages from a two-stage compressor to a three-stage compressor. The three-stage compressor has a longer shaft and heavier impeller weight than the 2-stage compressor. The design and control of rotor bearings with three-stage impellers was challenging. Since the first bending mode is closer compared to the rotor speed of 50Hz, we focused on ensuring the separation margin and stabilizing control according to the new design of the 3-stage compressor.

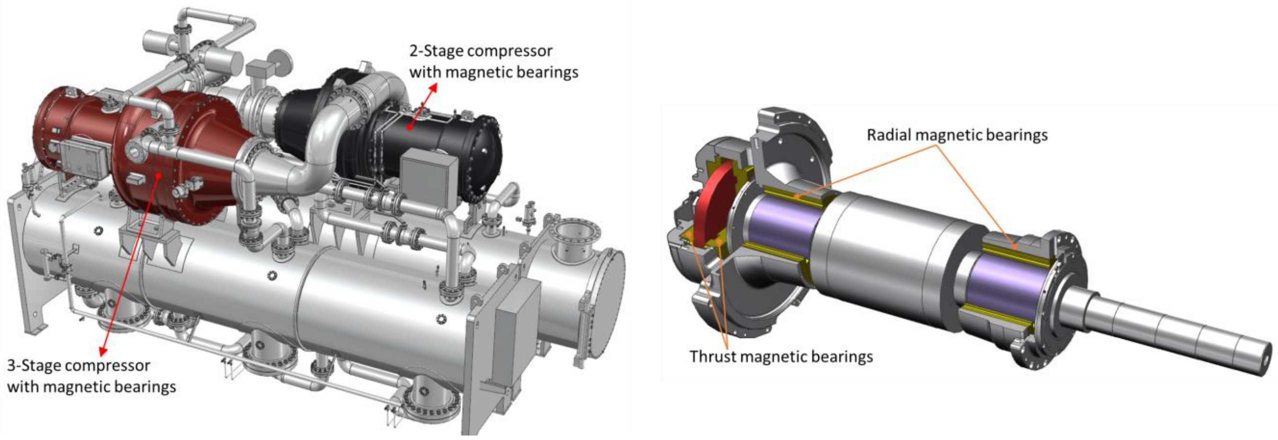

Figure 1 LG Electronics 3000 USRT oil-free turbo chiller with two compressors operated in series (left), and schematic view of rotor-bearing structure for 3-stage compressor (right)

2. Rotor-Bearing design

The shaft has three impellers, and the total length is about 2500mm. The total weight of the rotating parts is 1.3 ton. The mode shape of the rotor is shown in Fig.2. As a result of the rotor stability analysis, the first bending mode was found to be 74 Hz (backward) and 85 Hz (forward), and it was finally confirmed that it had a separation margin of approximately 48% for a constant speed of 50 Hz. In the initial rotor design, the separation margin was not suitable, but the design was modified by reducing the stacking length of the radial magnetic bearing several times. In addition, several modifications were made to reduce weight in the impeller design, while maintaining structural stiffness. As a result, the weight of each impeller was 45 kg, and the magnetic bearing stacking length was designed to be 284 mm. The outer diameter of the magnetic bearing rotor is 248.8 mm, and the bearing load design is 14kN. The load distribution of the radial magnetic bearings supporting both ends of the shaft was calculated as 4900N (NDE) and 7800N (DE).

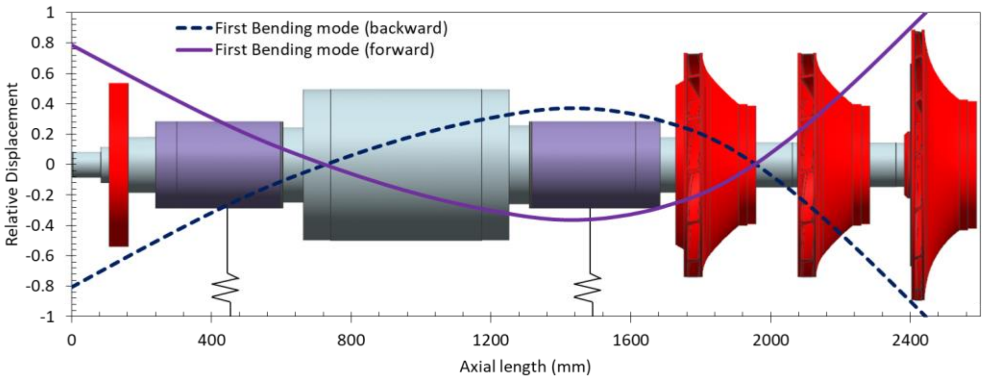

Figure 2 Mode shape of rotor: 1st bending mode 74 Hz (backward), 85 Hz (forward)

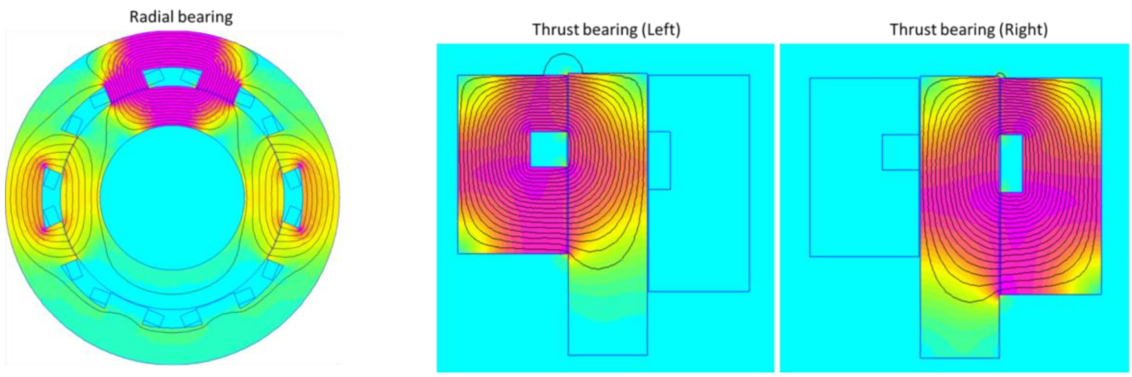

Figure 3 Finite element analysis of magnetic bearing, maximum pulling force per pole: Radial bearing (14kN), Thrust bearing (60kN)

The radial bearings supporting both ends of the shaft were designed to be 14kN, and the safety factor was designed to be more than 1.5 in consideration of the DE Side Bearing to which a lot of load is applied. The bias current is 6A and the maximum current of each pole is 15A. The thrust bearing was designed to support 60kN when 15A was applied, and a safety factor of 2 was applied by predicting the maximum axial load generated from the impeller. Both radial and thrust bearings are designed with an air gap of 1 mm. Fig. 3 shows the finite element analysis results for each bearing.

3. Control design

The magnetic bearing is a system that achieves stabilization through active control, and the performance of the controller as well as the hardware must be evaluated. The design of the controller determines the stiffness through dynamic analysis and designs a controller in the form of a phase lead that obtains stability. Along with this, a notch filter is applied for flexible mode control and a control program is produced. In order to verify the stability of the control design, the system response must be analyzed through the excitation test of the sine wave function. The sine sweep tests were conducted with 900 points from 10 Hz to 1000 Hz. The frequency response results allow identification of the dynamics of the built bearing system. Fig. 4 shows the plant transfer function results for one axis of radial bearing and thrust bearing, respectively. In the plant for radial bearings, the bending modes of the rotor can be identified, the first mode is around 80 Hz and the second mode is around 220 Hz. In the case of the plant for the thrust bearing, no flexible modes appear, but we can see that there are some modes near 150 Hz and 400 Hz. System identification allows more accurate notch filter and PID control design. Fig. 5 shows the sensitivity of each bearing. Based on ISO standard (ISO14839-3, 2006), the sensitivity level satisfies less than 3. The reason that the data before and after controller tuning is almost the same is because the controller was optimized while system identification was repeated several times. Based on this, the stability and reliability of bearing control can be confirmed.

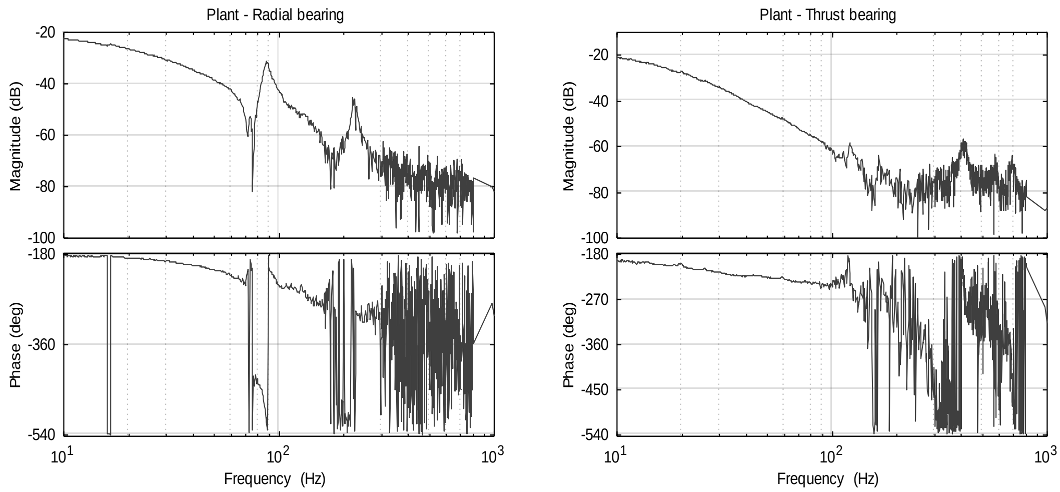

Figure 4 Identified plant transfer function: radial bearing (left), thrust bearing (right).

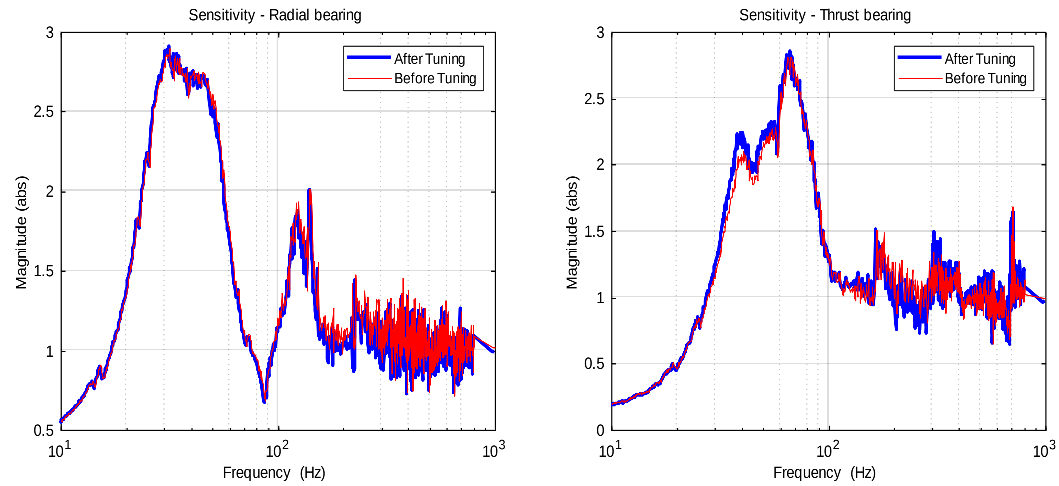

Figure 5 Measured Sensitivity: radial bearing (left), thrust bearing (right).

4. Operation test

The compressor motor operates at constant speed at 50Hz. The refrigerant goes through the two-stage compressor to the three-stage compressor, and the largest load is applied to the three-stage compressor. We observed the vibration of the three-stage compressor in detail, and as a result, it was confirmed that the vibration of the bearing was very good under full load. Fig. 6 shows the raw data and spectrogram of each bearing vibrations. The data sampling rate is 5 kHz, the air gap is 1 mm in the radial direction, and the clearance of the backup bearing is 0.5 mm. In radial bearings, the NDE vibration is slightly larger than the DE side vibration. However, the maximum vibration magnitude does not exceed within 0.1mm. This represents a vibration magnitude of less than 20% of the backup clearance. This result showed a smaller vibration than the criterion of ISO14839, indicating that the magnetic bearings are robust in refrigerant compression systems with load fluctuations or large loads.

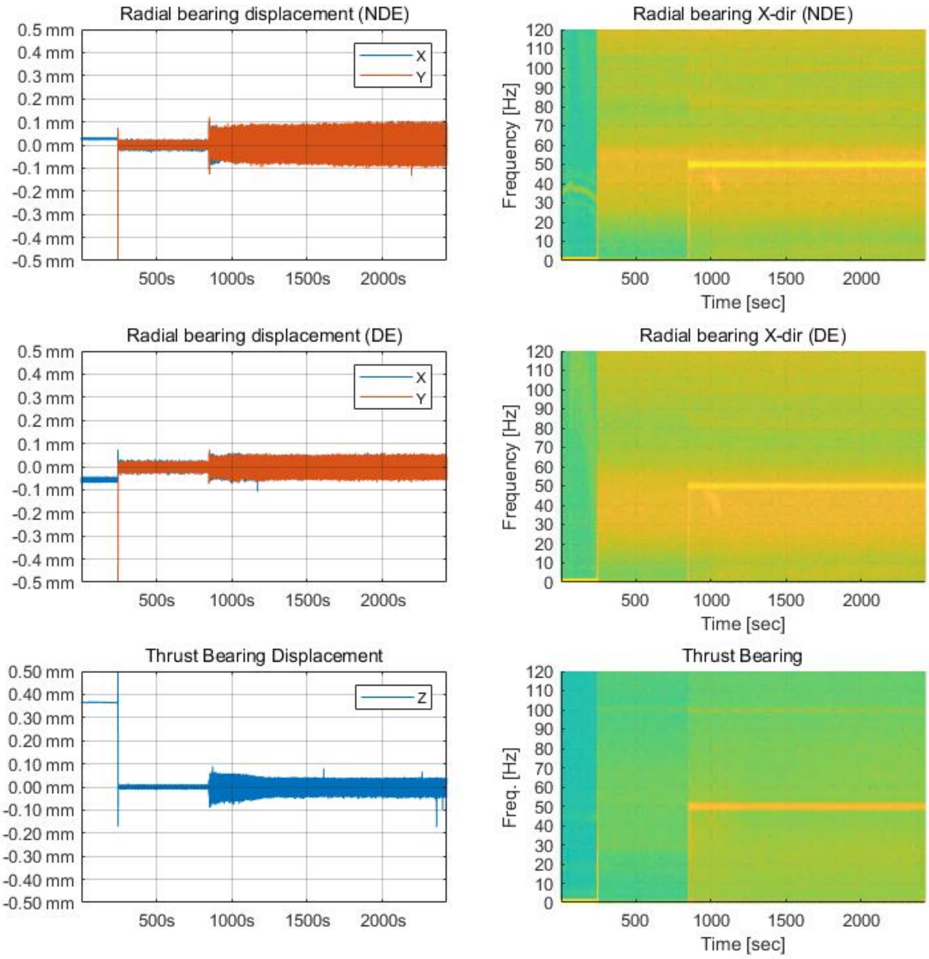

Figure 6 X, Y, Z direction vibrations (raw data) and spectrogram plot of each bearing according to rotation test.

5. Conclusions

In this study, a magnetic bearing system for a three-stage compressor was designed, which required a longer shaft and higher load capacity than a two-stage compressor. Focus on design to stabilize rotor dynamics and verify bearing load capacity. Control optimization is also derived through magnetic bearing design and identification of systems corresponding to high loads. As a result of field operation, the stability of the large-capacity magnetic bearing was confirmed by analyzing the vibration data under high load. With this, LG Electronics achieved its goal of developing an air-cooled oil-free turbo compressor with large capacity magnetic bearings. Based on the results of this study, we can expect to expand the technology of turbo chillers with large capacity magnetic bearings.

References

Noh MD, Jung J, Lee N, Jang S, Lee K and Kim S (2018) Development of Oil-Free Turbo-Chillers Equipped With Magnetically-Levitated Compressors, Proceedings of ISMB 16, Beijing, China.

ISO Standard 14839-3 (2006), Mechanical Vibration – Vibration of Rotating Machinery Equipped with Active Magnetic Bearing: Part 3 – Evaluation of Stability Margin.