# Development of Oil-Free Turbo-Chillers Equipped With Magnetically-Levitated Compressors

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***Abstract*— The chiller market has increasingly adopted oil-free technology due to the ease of maintenance and the use of variable-speed direct drive. Oil-free bearings also eliminate the possibility of contaminating refrigerant with lubricating oil, resulting in the prolonged use of refrigerant and avoiding performance degradation. In this paper, the development of oil-free chillers which have been recently introduced in the market is described. The centrifugal compressor of the chiller is equipped with active magnetic bearings: two radial bearings and one thrust bearing. We present some results of performance validation tests.**

## I. Introduction

As with any other sectors, high efficiency and low power consumption are the primary factors in heating, ventilation, and air-conditioning (HVAC) industry. Maintaining high efficiency even under partial load is very important and is a technological trend in the HVAC sector.

One solution to meet this need is to use oil-free turbo machines for compression or ventilation, coupled with variable-speed drives adopting inverter technology. Oil-free turbo-machines use either aerodynamic bearings (mostly foil bearings) or active magnetic bearings (AMB). Limited load capacity of gas bearings forces them to be used in smaller machines, whereas large capacity machines use AMBs for complete oil-free, non-contact operation. The turbo-chillers with magnetic bearings have the added benefit of higher reliability and maintainability because critical signals indicating the machine state are already available for the control of magnetic bearings.

In this paper, we describe a turbo-chiller equipped with AMBs, which is recently launched and commercially available. All the critical components of the magnetic bearing system are developed in-house, including sensors, amplifiers, and controllers. A series of oil-free chillers have been developed. This paper focuses on two models: 500RT and 1000RT capacity machines.

## II. System Description

Shown in Fig. 1 is the picture of the oil-free turbo-chiller described in this paper. The main components of the chiller include a condenser, an evaporator, an economizer and a compressor. The schematic of the compressor is illustrated in Fig. 2. It is a two-stage centrifugal compressor suspended by AMBs: two radial bearings and one thrust bearing. The radial bearing located at the left of the motor is called the motor bearing. The other radial bearing is called the impeller bearing.

Figure 1. Picture of LG Electronics 1000RT oil-free chiller with active magnetic bearings.

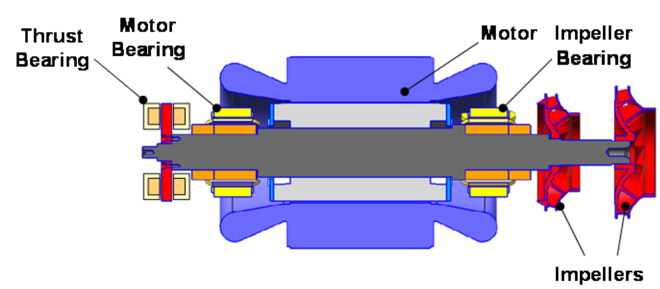



Figure 2. Schematic of compressor rotor

Total of eight different models have been planned, half of which are already in the market. Table 1 summarizes the capacities of the models. The models with doubled capacity are also available by cascading two compressors. In this paper, we focus on two types of machines: 500RT and 1000RT.

TABLE I. LG Electronics Oil-Free Turbo Chiller Series

| Capacity (usRT) | 340 | 380 | 440 | 500 | 600 | 700 | 800 | 1000 |
|---|---|---|---|---|---|---|---|---|
| COP | 6.06 | 6.03 | 6.09 | 6.08 | 6.16 | 6.14 | 6.10 | 6.17 |
| Motor Power (kW) | 220 | 240 | 280 | 340 | 400 | 450 | 550 | 670 |

## III. SYSTEM DESIGN

### A. AMB Design

The design requirements of the two compressors are shown in Table 2. The radial bearing is a standard eight-pole type. The thrust bearing is a double-acting, fully-electromagnetic type. The results of the design are summarized in Table 3. Figure 3 displays the pictures of the radial and thrust magnetic bearings manufactured for 500RT machines.

TABLE II. DESIGN REQUIREMENTS

| Model | 500RT | 1000RT |
|---|---|---|
| Max Operating Speed (rpm) | 18,000 | 10,500 |
| Radial Bearing Load (N) | 1,120 | 3,000 |
| Thrust Bearing Load (N) | 5,000 | 10,000 |

TABLE III. AMB DESIGN RESULTS

| Design Parameters | 500RT | | 1000RT | |
|---|---|---|---|---|
| | Radial | Thrust | Radial | Thrust |
| Max Current (A) | 15 | 15 | 15 | 30 |
| Actuator Gain (N/A) | 141 | 2445 | 410 | 2174 |
| Open-loop Stiffness (N/μm) | -1.8 | -14.7 | -4.4 | -52.2 |

Figure 3. Pictures of radial and thrust magnetic bearings

### B. Position Sensors

Position sensors are also developed. Using printed coils on flexible substrates, inexpensive eddy-current sensors are designed and manufactured. The system has the total of 10 sensors, all of which are differentially operated. The thrust sensors also measure the rotational speed. Figure 4 shows the pictures of the radial sensor holder and the sensor coil on flexible substrate.

The axial gap sensor not only measures the gap but also detects the rotation direction which is utilized by the system controller.

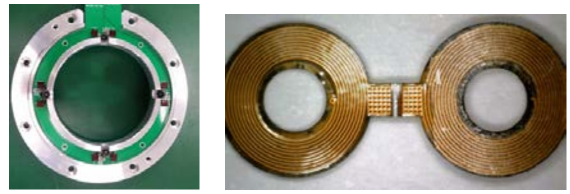


Figure 4. Pictures of the sensor holder and the sensor coil

### C. Amplifiers and Digital Controller

Power amplifiers are of half-bridge PWM type as shown in Figure 5. In order to reduce the power consumption, three-state switching strategy is used. The controller consists of three DSPs, two of which are used to process the sensor signals and to implement the transconductance operation for current control. The main controller runs the suspension control algorithm and communicates with the system controller. The main controller has the sampling frequency of 7 kHz for the suspension control and 21 kHz for current control.

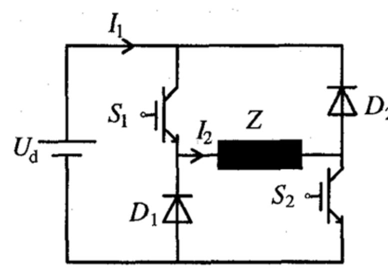



Figure 5. Half-bridge switching topology employed by the power amplifier

## IV. SYSTEM IDENTIFICATION

Sine sweep tests are performed to identify the dynamics of the actual machines. Fig. 6, 7, and 8 show the measured plant transfer functions of motor, impeller, and thrust bearings, respectively. The plant transfer function contains the dynamics of transconductance amplifiers, sensors, electromagnets, and rotor. The plant transfer functions for radial bearings show the bending modes of the rotor: the first bending at 287~288 Hz and the second bending at 523~534 Hz. As for the plant transfer for thrust bearing, no flexible modes are visible. Also, no interactions from radial motions can be found.

Using the measured plant transfer functions, dynamic models of the system are identified using the algorithm by Sanathanan and Koerner (SK algorithm) [1]. Sequential identification strategy is employed, where the rigid-body dynamics is first identified in the low-frequency region, and then the amplifier dynamics is identified in the frequency from 50 Hz to 300 Hz. Lastly, the flexible modes are identified in the high frequency range. Using the identified models, bearing controllers are designed. The controller is a lead-lag type with one or two notch filters to boost the damping at the flexible modes.

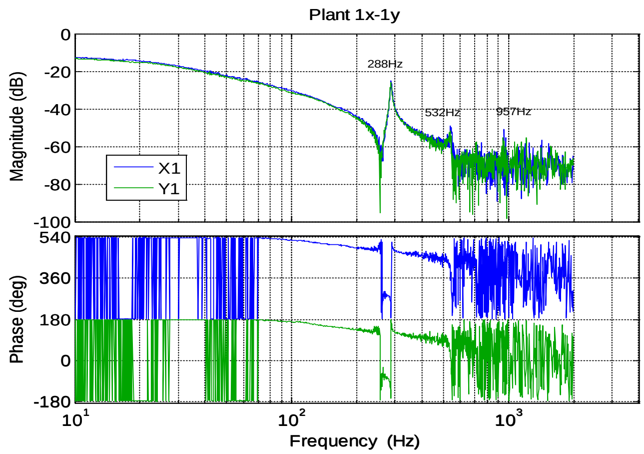



Figure 5. Identified plant transfer function (1000RT, motor bearing)

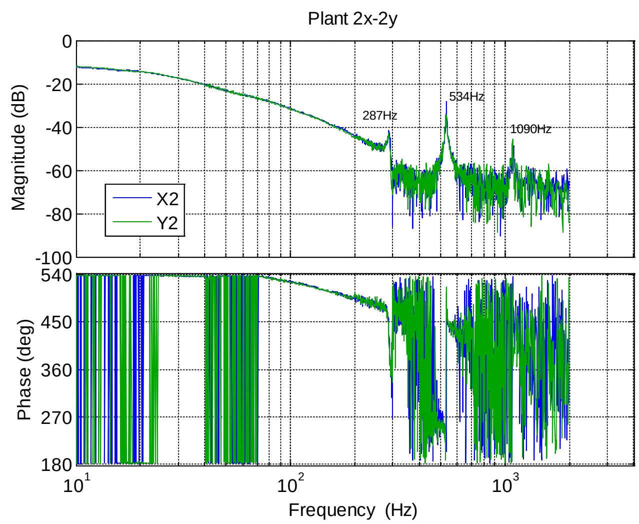



Figure 6. Identified plant transfer function (1000RT, impeller bearing)

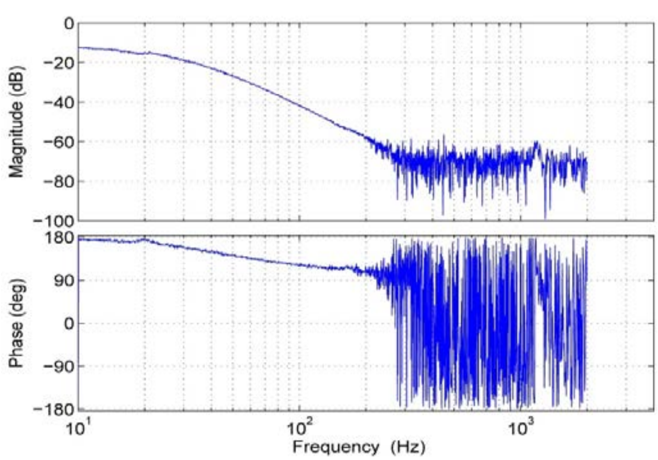
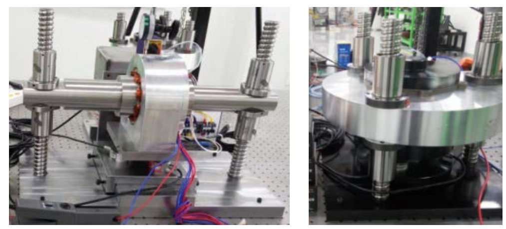



Figure 7. Identified plant transfer function (1000RT, thrust bearing)

## V. PERFORMANCE VALIDATIONS

In order to validate the performance of active magnetic bearings and the turbo-chiller, various tests have been performed.

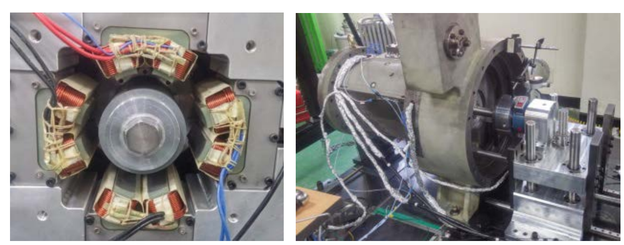
First, the static load capacities of the bearings are measured and verified to be in agreement with the design. Figure 8 shows the test setups for measuring the static load capacities.



(a) (b)

Figure 8. Measurement setups for static bearing load capacities: (a) radial and (b) thrus bearingt.

Dynamic load capacities of the bearings are also measured to see if they satisfy the design requirements. In order to apply the force while the rotor is spinning at the rated speed, specialized non-contact actuators have been designed and implements. Figure 9 displays the pictures of the test setups. Sinusoidal excitations at a specific frequency are applied to the rotor and the resulting vibrations are measured.



(a) (b)

Figure 9. Test setups for measuring the dynamic bearing load capacities: (a) radial and (b) thrust bearing.

The results of dynamic tests are shown in Figure 10 and 11 in the form of dynamic stiffness. For the radial bearing, the load capacity is maintained up to around 50 Hz and starts to increase due to the stiffening of the rotor. As a reference, the requirements of dynamic load for G2.5 is also displayed in Figure 10, which clearly shows that the radial bearing satisfies the requirement.

As for the thrust bearing, the results show that the bearing can produce the dynamic load capacity up to around 3 Hz. Since the most of the thrust disturbances like surge are fairly slow, the bearing satisfies the design requirements. Increasing the integral gain of the bearing controller can improve the load capacity, if necessary.

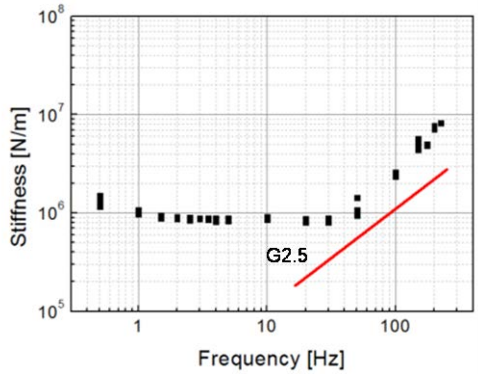



Figure 10. Results of dynamic load capacity measurements for radial bearing.

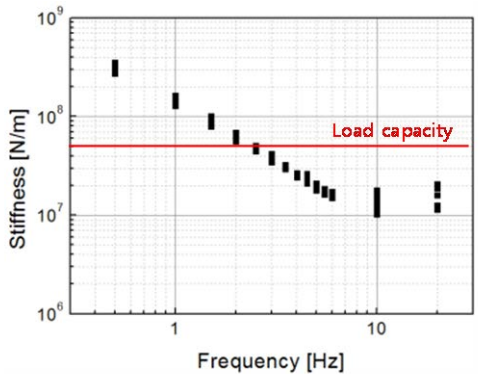



Figure 11. Results of dynamic load capacity measurements for thrust bearing.

In order to verify the reliability of the backup bearings, a drop test is also performed. The test procedure is as follows. While the machine is running at the rated speed, the power to the bearing is suddenly cut off. After letting the machine run on the backup bearings for 30 seconds, the power is restored. The total of 50 drops have been tried. Figure 12 shows the results of the dropt test. Although a slight increase in the radial clearance is observed, it is no more than 10% of the initial clearance, which satisfy the ISO requirement.

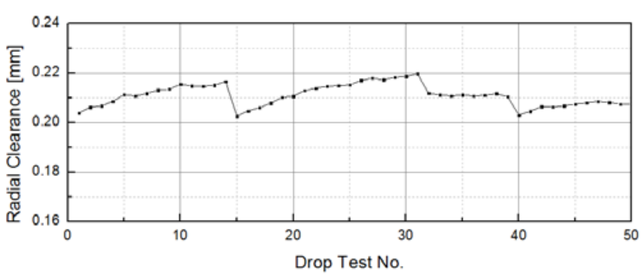



Figure 12. Drop test results

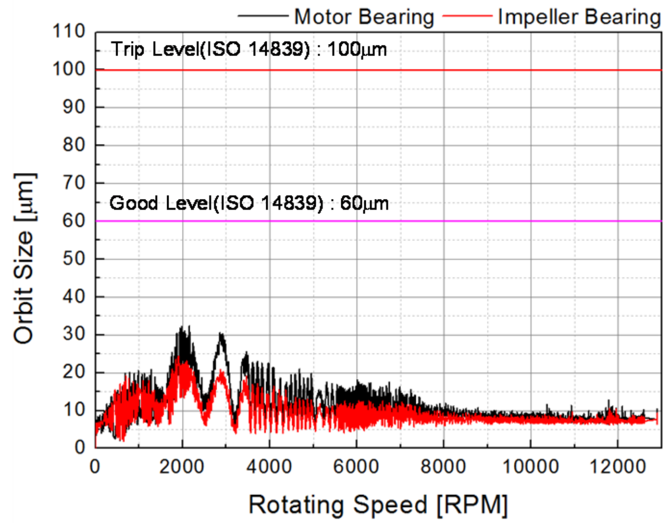



Figure 13. Full load test (1000RT)

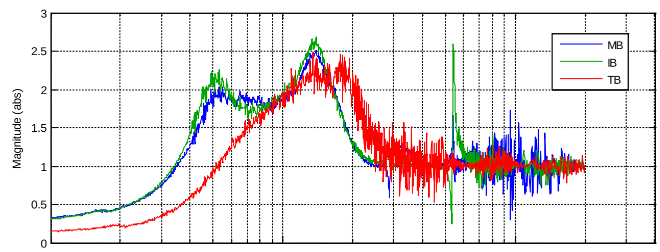



Figure 14. Measured sensitivities (1000RT)

Fig. 13 shows the results of a run-up test. The orbit size is maintained below 30 μm, which is much smaller than the criterion of ISO14839.

The sensitivities of the AMB system are measured. As shown in Fig. 14, sensitivities of all bearings are below 3, which satisfies the ISO requirements for stability [2].

## REFERENCES

[1] C. K. Sanathanan and J. Koerner, "Transfer function synthesis as a ratio of two complex polynomials," IEEE Trans. Automatic Control, vol. 8, pp. 56-58, 1963.

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