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Evaluation of Lead Coated Ball Bearing Using Frequency Analysis in Extreme Conditions

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This study presents the evaluation of deep groove ball bearings for turbo pump unit under extreme conditions using frequency analysis. We conducted the evaluation test for two types of the deep groove ball bearing considering the operating time at the rated speed. The maximum rotational speed is 11,000 rpm and the experiment used water to make it similar to the actual environment. The test results included the vibration signal to confirm stability of the ball bearing. And, the test confirmed the stability of ball bearing using the frequency analysis in real time. On the other hand, the ball bearing was analyzed the inner surface for the wear trends. Furthermore, we confirmed the optimization clearance of the cryogenic ball bearing and this study conducted the rotating test during the target time, successfully.
Keywords: PTFE cage wear, cryogenic environment, Ball bearing

1. Introduction

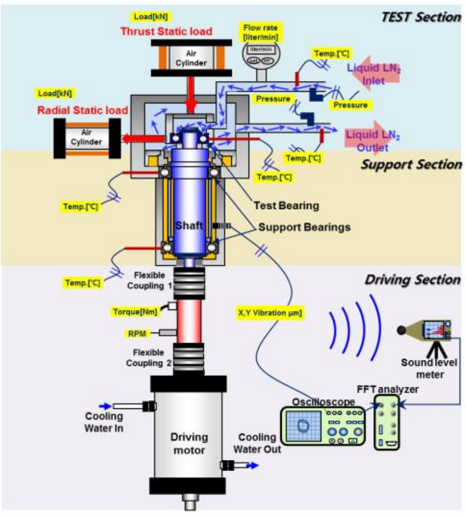
Unlike standardized rolling bearings, non-standardized bearings are generally operated under extreme conditions, and technical paradigms such as design, fabrication, and evaluation are significantly different from standardized rolling bearings. In particular, the ball bearings used in turbo pumps for space launch vehicles are unable to lubricate oil or grease because the working fluid flows through the clearance of the ball bearing in the turbo pump. In addition, it receives a large load in the axial and radial directions, and it rotates at a relatively high speed in the ball bearing at the same time. Thus, the evaluation of performance and reliability is a very important procedure for ball bearing operating in extreme conditions.

Kingsbury [1, 2] was the first to analyse the variations in the torque of ball bearings as well as the cage whirling motion associated with the fluctuations in the torque based on analytical and experimental data. In addition, studies have shown that the frequency of the radial motion increases with an increase in the cage whirling motion. Donald W.Wisander et al [3] were investigated lead coating applied to aluminium-bronze cages as a lubricant in liquid hydrogen. They stated that the a lead formed an effective lubricant transfer film on the balls and in the race grooves. Gupta [4] researched analytical approach of cage motion in a solid-lubricated ball bearing with the computer program ADORE. From this investigation, motion effects of cage instability related with cage pocket and guide land clearances as considering the whirl ratio, time-averaged wear rate and the shape of the whirl orbit. Munro et al. [5] performed tests to measure the torque on ball bearings using a bronze cage coated with lead both in vacuum and under atmospheric conditions. The ball bearing torque decreased under light loads because of coulombic friction. On the other hand, the test results demonstrated that the lead coating did not afford a distinct advantage under heavy loads.

In this study, the evaluation of ball bearings and technical studies were conducted using frequency analysis for turbo pump ball bearings operating in extreme environment.

2. Test facility and boundary condition Introduction

The ball bearings mounted on the turbo pump are largely divided into two types of ball bearings for oxidize and fuel pumps. In this paper, a study on ball bearings for fuel pumps was conducted. The deep groove ball bearings used in the experiment have inner diameters of 50mm and 80mm respectively, and SUS440C which has great hardness and abrasion resistance is used as inner, outer race and ball material. The surface except the ball is soft metal coated with lead. And, aluminium-bronze was used as the material of the cage. Figure 1 presents a schematic view of the ball bearing test rig. The test rig could be driven at up to 11,000 rpm using a DC motor and provided a load of up to 20 kN using a pneumatic cylinder. The test rig was divided into three main parts along the axial direction. The first part was a 50 kW (driven at 11,000 rpm) DC motor capable of rotating at up to 11,000 rpm; water was used for cooling the motor. The second part was a middle chamber, which included a main shaft supported by ball bearings at both ends. The last part was the test chamber containing the test ball bearing. All the shafts of this part were connected through a flexible coupling to minimize misalignment. A staggered labyrinth and a lip-type seal were mounted between the oil lubrication chamber and the test chamber to minimize fluid mixing. Eddy current displacement sensors were fixed on the middle part in the oil lubrication chamber at intervals of 90° to measure the vibration of the main shaft. The rotational component data were analyzed using a real-time fast Fourier transform analyzer. The other test data (temperature, pressure, torque, and flow rate, among others) were stored using a NI data acquisition board and the program LabVIEW, which allowed for real-time monitoring.



(a) Schematic view of test rig

(b) Photograph of test rig

Fig. 1. Schematic view and test apparatus of test rig

Table 1: Design parameter of test bearing (6210, 6216) and cage

Design Parameter		No. 6210	No. 6216
Race	Bore diameter, [mm]	50	80
	Material	SUS440C	SUS440C
	Soft metal coating material	Lead	Lead
	Precision grade	P4	P4
Cage	Material	Aluminum-bronze	Aluminum-bronze
	Guidance type	Outer	Outer
Boundary condition	Speed, [rpm]	11,000	11,000
	Load(axial/radial), [kN]	7/5	13/10
	Working fluid	Water	Water
	Operating time, [s]	2,100 ≤	2,100 ≤

3. Test results of non-standard bearing coated with solid lubricant

Fig. 2 shows the waterfall plot with bore diameter of 50 mm at the end of rotating test. The horizontal axis indicates the frequency range, the diagonal line indicates the time, and the vertical axis indicates the vibration amplitude. In fig. 2, the most prominent component is outer race ball pass frequency (BPFO), which represent specific frequency component of the outer race. In addition, general ball bearings also have frequency components related to the inner ring and balls, such as Inner race ball pass frequency (BPFI) and ball spin frequency (BSF). It shows different values depending on the number of revolutions, the number of balls, and the geometry of the bearing, and it can be easily calculated formally. The cage of 50mm inner diameter bearing used in this experiment is bronze material and has higher abrasion than SUS440c. Therefore, it can be confirmed that the ball is rotated and FTF is generated due to continuous friction with the cage. However, since the change of the signal is not large at the end of driving, it is judged that the friction has occurred within the normal range. Also, when the bearing was disassembled after the experiment, it was confirmed that the components of the cage were distributed in both the ball, the inner and outer races. This suggests that the bronze component of the cage is increasingly worn to provide coulomb damping to the bearing, and it reduces the friction between the ball and the race due to the bronze film. In addition, the outer ring of the actual bearing is fixed. Thus, the outer race is continuously subjected to a specific load due to the axial and radial load. This causes damage to the outer race raceway surface, which appears as BPFO. This results in a BPFO associated with the outer race, but the absolute magnitude does not change with time.

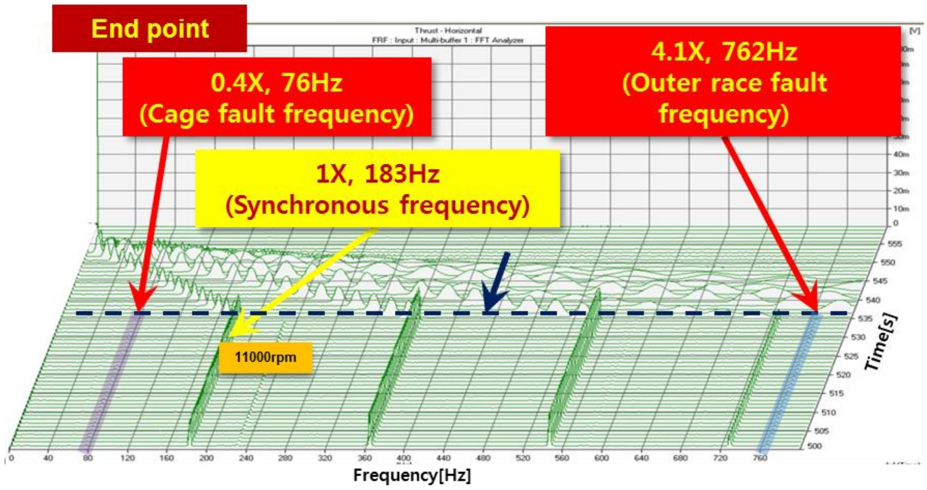


Fig. 2. Schematic view and test apparatus of test rig

Fig. 3 shows the waterfall plot with bore diameter of 80 mm at the end of rotating test. In fig. 3, the most prominent component is BPFO, which represents specific frequency components of the outer race. The load applied in the test of the bore diameter 80mm bearing is 13, 10kN in the axial and radial direction. This is an example where a relatively large load is applied, compared to other bearings, and the DN number reaches 880,000 due to the large bearing bore. In addition, it can be judged that a large change in the outer ring raceway due to the severe test conditions in which only the supply of the water is performed without additional lubrication. Thus, this result is shown as BPFO and its 2X component in Fig. 3. In addition, it can be confirmed that the double component of BPFO is branched. This indicates that the damage is increasing on the outer raceway surface due to the large load and high speed rotation condition. Also, the increase in BPFI at the end time indicates that the inner ring raceway surface is also damaged. After the end of the experiment, it was confirmed that these signals were due to the unspecified wear of the inner and outer races through the disassembly of the bearings. Particularly, it was found that the surface of the outer ring had a large flaking phenomenon, and the effect of extreme conditions without lubrication was directly confirmed on the surface hardening and fracture of the ball bearing.

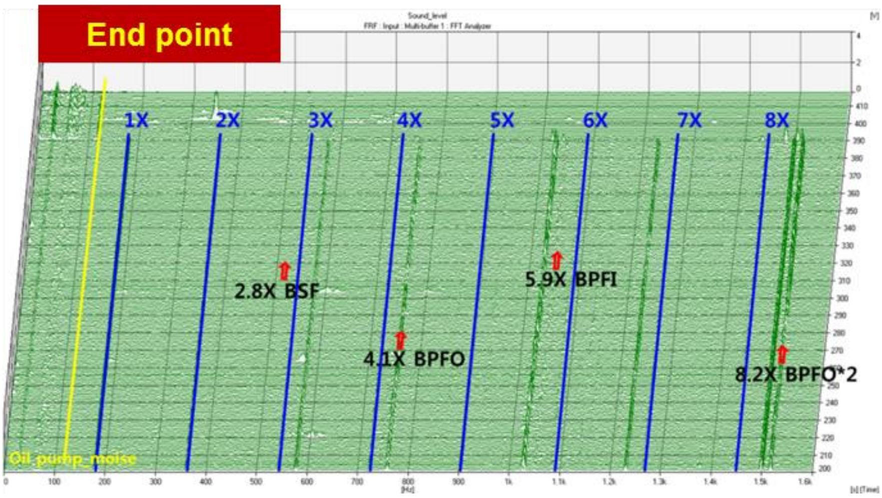


Fig. 3. Schematic view and test apparatus of test rig

4. Conclusions

In this study, the test evaluation was conducted for two types of the ball bearing under specific conditions operating in extreme environments. In each experiment, the stability of the bearing was evaluated by analysis of the specific frequency that occurs according to the geometry of the ball bearing. Fracture mechanics in bearings through specific frequency analysis investigated in extreme environments without lubrication system. From these results, the stable driving test was performed satisfying specific load condition, operation speed and time. It is considered that optimization of bearing geometry value is essential for stable drive.

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