Heat generation of ball bearing for rocket-engine turbopump in low-temperature gas hydrogen

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Abstract. Cryogenic fluid such as liquid hydrogen, liquid oxygen, and liquid methane is used for rocket propellant because of their performance. Cryogenic fluid turbopump equipped in rocket-engine is started to rotate after precooling bearings is completed. The ball bearing used for the turbopump have worse lubricating performance because the lubricant is not liquid material such as oil which coagulate at cryogenic temperature but solid material. Polytetrafluoroethylene (PTFE) is used for the lubricant in Japanese rocket engines [1]. One of the glass transition temperature of PTFE is around 100 K, and it is considered that the strength of PTFE become weak discontinuously above 100 K and poor lubrication problem would occur in bearings. Ball bearings for turbopump which need to perform high stiffness at high rotation speed don’t meet the performance suddenly when they have any slight damages. Therefore, bearings should be precooled before the engine start so that poor lubrication problems would not occur. However, there is little knowledge of the ball bearing performance in the environment of gas coolant. If ball bearings for turbopumps have good performance in the environment, turbopump can be started to rotate under the insufficient precool environment. It contributes reduction of time and propellant for the precool, and improvement of rocket performance and diversification of space transportation mission are expected. In this study, bearing rotational tests in gas coolant are conducted in the several conditions of the rotational speed and the thrust load in order to clarify the bearing performance for rocket engines in gas coolant.

1. Introduction
Cryogenic fluids such as liquid hydrogen, liquid oxygen, and liquid methane tend to be used for rocket propellant because their performances as the rocket propellant are higher than other propellants. Cryogenic fluid turbopump equipped in rocket-engine is started to rotate after precooling bearings is completed. The ball bearing used for the turbopump have lower lubricating performance because the lubricant is not liquid material such as oil which coagulate at cryogenic temperature but solid material. Polytetrafluoroethylene (PTFE) is used for the lubricant in Japanese rocket engines [1]. One of the glass transition temperature of PTFE is around 100 K, and it is considered that the strength of PTFE become weak discontinuously above 100 K and poor lubrication problem would occur in the contact areas in the bearings [2]. Ball bearings for the turbopump which need to perform high stiffness (100,000 ~ 200,000 N/mm) at high rotation speed (20,000 ~ 100,000 rpm) don’t meet the performance suddenly when they have any slight damages such as wear and crack. Therefore, bearings should be precooled before the engine start so that poor lubrication problems would not occur.
However, there is not enough knowledges of the ball bearing performance in the environment of gas-phase coolant where is higher temperature than liquid-phase coolant. If ball bearings for turbopumps have good performance in the environment, turbopump can be started to rotate under the incomplete precooled environment where the propellant is still gas-phase. It contributes reduction of time and propellant for the precool operation of the engine reignition, and improvement of rocket performance and diversification of space transportation mission are expected [3, 4].

In several country such as France, Korea, and Japan, rocket engines using electric pump cycle which is famous for RocketLab’s Rutherford engine are under development [5, 6]. In the electric pump, it is necessary to make mechanical losses minimum as possible because electric pump has much less power than turbine driving pump. Thus, though liquid-cooled bearings are used in the cryogenic turbopump with turbine, the use of gas-cooled bearing in the electric pump is considered. Figure 1 shows the power losses of rotating shafts in the propellants. The loss is about 4 kW in the condition of 60,000 rpm in LCH₄ or 120,000 rpm in LH₂. This loss cannot be negligible amount because electric pump is considered to use for small rocket engines.

In this study, bearing rotational tests in gas coolant are conducted in the several conditions of the rotational speed and the thrust load in order to clarify the bearing performance for rocket engines in gas coolant.

![Figure 1. Power loss of rotating shafts in the propellants](image)

2. Bearing tester

Figure 2 shows the cross section of the bearing tester. In this tester, two bearings (bearing A and B) can be rotated in the same condition of rotation speed, bearing load, coolant flow rate. The main shaft is driven by an electric motor. Coolant is fed between the bearings and drained in the downstream of shaft seals. Hydrogen gas is used for the coolant, and the gas is fed at the temperature of about 100 K. Thrust load is applied using the double bellows cartridge set between the outer races in which GHe can be supplied.
Figure 2. Bearing tester schematic
TBa: Temperature of bearing A outer race, TBB: Temperature of bearing B outer race,
TBaO: Temperature of the downstream flow of bearing A,
TBbO: Temperature of the downstream flow of bearing B,
PBRG: Pressure of the bearing room, TFI: Temperature of the upstream flow of bearings

3. Test bearings
Table 1 shows the parameter of bearings used in this research. The bearing is an angular contact ball bearing with ceramic balls [7]. The inner diameter is 25 mm. The bearing is self-lubricated bearing with glass reinforced PTFE retainer [8]. In order to clarify the influence of inner race curvature on bearing performance, the curvature of the two bearings is different. The curvature of bearing A is larger than the one of bearing B. The outer race curvature and contact angle are the same between the two bearings.

Table 1. the design parameters of test bearings

|                   | Bearing A | Bearing B |
|-------------------|-----------|-----------|
| Inner diameter    | 25 mm     | 25 mm     |
| Outer diameter    | 52 mm     | 52 mm     |
| Material (inner race / outer race / ball) | SUS440C/ SUS440C/ Si3N4 | SUS440C/ SUS440C/ Si3N4 |
| Ball diameter     | 5/16 in   | 5/16 in   |
| Number of balls   | 10        | 10        |
| Inner race curvature | Larger | Smaller  |
4. Measurement
In this test, following measurement items were recorded at 10 Hz. Temperature was measured using thermo couple which was calibrated with LN\textsubscript{2}. The flow rate of GH\textsubscript{2} supply was measured using thermal mass flow meter. It was confirmed by flow tests that the GH\textsubscript{2} flow rate of bearing A is almost the same as one of bearing B. The bearing load applied to bearings were calculated using the differential pressure between bearing room pressure and GHe pressure inside the double bellows cartridge. The relationship between applied load and the differential pressure was calibrated. The driving power of the electric motor was measured using a power meter. The rotational torque was estimated by formula (1). Note that in this formula power supplied to the motor is converted to the shaft power and the electrical power loss in the motor is not considered.

\[ T = \frac{P \cdot 60}{2\pi \cdot NF} \]  
\( T \): Shaft torque[Nm], \( P \): Motor power[W], \( NF \): Rotational speed[rpm]

The energy difference of the coolant between upstream and downstream of the bearing was calculated by formula (2) and (3). Enthalpy of the coolant is calculated by REFPROP [9].

\[ \Delta W_A = (H_{AO} - H_{In}) \cdot \frac{m_F}{2} \]  
\[ \Delta W_B = (H_{BO} - H_{In}) \cdot \frac{m_F}{2} \]

\( \Delta W \): Energy difference of the coolant between upstream and downstream of the bearing [J/s], \( H_O \): Enthalpy of the coolant at the downstream of bearing [J/kg], \( H_{In} \): Enthalpy of the coolant at the upstream of bearing [J/kg], \( m_F \): Flow rate of coolant supply [kg/s]

5. Test condition
Table 2 shows the test condition. Rotational speed was changed in the range of 10,000 rpm to 30,000 rpm. Thrust load applied to the bearings was changed in the range of 1,000 N to 2,000 N. In the condition of the thrust load of 2,000 N, bearings were tested up to 20,000 rpm because heat generation was large. Flow rate was changed by 3 steps in each condition of the rotational speed and thrust load in order to investigate the influence of flow rate on the bearing temperature.

| Table 2. Test condition |
|-------------------------|
| Bearing coolant         | Gas hydrogen |
| Rotational speed        | 10,000 / 15,000 / 20,000 |
|                        | 25,000 / 30,000 rpm     |
| Thrust load             | 1,000 / 1,500 / 2,000 N |
| Coolant flow rate       | 570 ~ 1,150 NL/min      |
| Duration of rotation    | 8,490 s                |

6. Results and discussion
Figure 3 shows a typical historical data of the rotating tests. As mentioned above, each condition of the rotational speed, the coolant flow rate and the thrust load is changed in the tests. Gas hydrogen was used as the bearing coolant cooled in the upstream of the tester at the temperature of 100 K to 120 K, and the temperature changed influenced by the coolant flow rate.
6.1. Bearing temperature

Figure 4 and 5 show the relationship between the rotational speed and bearing temperature in the thrust load condition of 1,000 N and 1,500 N respectively. The data of each plot in the graph is 100 points averaged data of historical data. Bearings were tested in the condition of bearing temperature up to about 200 K in this research. In the condition of 1,000 N load, bearing temperature is less than 200 K in each flow rate condition. Bearing temperature decreases with the increase of the coolant flow rate in the same condition of the rotating speed and thrust load. In the condition of 1,500 N load, bearing temperature is higher at the same rotational speed.

In each condition of the thrust load, the temperature sensitivity of bearing A to the rotational speed is almost the same as one of bearing B. Therefore, it is considered that heat generation of both bearings is almost equivalent because friction force on the contact area under the equivalent thrust load is almost the same unless lubrication film does not break. Though temperature of bearing A tends to be slightly higher than one of bearing B, the temperature difference is considered to be influenced by the heat input of the tester from the outside because it does not change a lot regardless of the increase of the rotational speed.

Table 3 shows the maximum contact pressure in the bearing contact area which is calculated by the basic method described in [10]. Because the inner race curvature of bearing A is larger than one of bearing B, the maximum contact pressure of bearing A inner race is higher than bearing B. On the other hand, the outer race contact pressure of both bearings is almost equivalent because the outer race curvature of both bearings is the same. Considering the test results, PTFE lubrication film is not broken in the contact pressure up to 2.68 GPa under the GH2 cooling in the temperature of about 120 K. Though bearing tests in the load condition over 2,000 N are not conducted in this research, bearing A whose maximum contact pressure is larger would have poor lubrication caused by break of lubrication film in the higher thrust load condition.
Though the maximum contact pressure of bearing B is smaller than one of bearing A, the sliding speed in the inner race contact area is higher because the contact area of bearing B is larger than one of bearing A. Therefore, bearing B would have larger friction caused by balls’ slip and spin in the higher speed condition. In this research, the maximum rotational speed is 30,000 rpm and it is considered that the influence of friction on heat generation is too slight to raise the bearing temperature. In the higher rotational speed condition, bearing B whose friction is larger would have fatal damages caused by poor lubrication.

Figure 4. Bearing temperature in the condition of 1,000 N

Figure 5. Bearing temperature in the condition of 1,500 N
Table 3. Maximum contact pressure in the bearing contact area

|          | Bearing A |          | Bearing B |
|----------|-----------|----------|-----------|
|          | Inner race | Outer race | Inner race | Outer race |
| 1,000 N  | 2.19 GPa   | 1.29 GPa  | 2.05 GPa   | 1.31 GPa   |
| 1,500 N  | 2.47 GPa   | 1.44 GPa  | 2.30 GPa   | 1.46 GPa   |
| 2,000 N  | 2.68 GPa   | 1.56 GPa  | 2.49 GPa   | 1.57 GPa   |

6.2. Energy increase of bearing coolant

Figure 6 and 7 show the relationship between the rotational speed and energy increase of bearing coolant in the load condition of 1,000 N and 1,500 N. The cooling ability to keep bearing temperature under 200 K is about 450 W and 610 W in the load condition of 1,000 N and 1,500 N respectively. The energy difference increases with the flow rate in the same condition of the rotational speed and thrust load. It is considered that the increase of the energy difference is caused by the change of heat transfer coefficient. Also, bearing temperature tends to decrease with the increase of the flow rate because the coolant ability is higher in the higher flow rate condition.

Figure 6. Energy difference of bearing coolant in the condition of 1,000 N
7. Bearing inspection

After the series of rotational tests for this research, bearings were inspected in detail. There are no fatal damages in bearings. Table 3 shows rotation tolerance of bearing A. There is no great difference of rotation tolerance between before and after tests, and this bearing can be used more. Figure 9 and 10 show the groove shape of bearing A. The wear depth in the inner race and outer race is less than 1 µm and the wear amount is acceptable to this ball bearing. It is considered that this bearing lubrication worked properly in the test conditions.

| Table 3. Rotation tolerance of Bearing A |
|-----------------------------------------|
| Before tests | After tests |
| Inner race radial runout | 1 µm | 1 µm |
| Inner race axial runout | 4 µm | 3 µm |
| Outer race radial runout | 1 µm | 1 µm |
| Outer race axial runout | 4 µm | 2 µm |

Figure 7. Energy difference of bearing coolant in the condition of 1,500 N
Figure 8. Ball roundness of bearing A

Figure 9. Inner race groove shape of bearing A

Figure 10. Outer race groove shape of bearing A
8. Conclusion
The bearing tests in several conditions of rotational speed and thrust load were conducted in order to clarify the performance of bearings for rocket-engine turbopump under gas coolant use. Following is a list of accomplishments in this research. Note that the bearing performance under gas cooling environment such as incomplete fuel turbopumps and electric motor pumps will be considered using the knowledges obtained in this research.

- There are no fatal damages in bearings tested in the conditions of this research.
- Bearing temperature increases with the rotational speed and thrust load.
- Bearing temperature keep lower in the higher flow rate condition.

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