Critical Performance of Turbopump Mechanical Elements for Rocket Engine

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It is generally acknowledged that bearings and axial seals have a tendency to go wrong compared with other rocket engine elements. And when those components have malfunction, missions scarcely succeed. However, fundamental performance (maximum rotational speed, minimum flow rate, power loss, durability, etc.) of those components has not been grasped yet. Purpose of this study is to grasp a critical performance of mechanical seal and hybrid ball bearing of turbopump. In this result, it was found that bearing outer race temperature and bearing coolant outlet temperature changed along saturation line of liquid hydrogen when flow rate was decreased under critical pressure. And normal operation of bearing was possible under conditions of more than 70,000 rpm of rotational speed and more than 0.2 liter/s of coolant flow rate. Though friction coefficient of seal surface increased several times of original value after testing, the seal showed a good performance same as before.

Key Words: High-speed Bearing, Axial Seal, Turbopump, Liquid Rocket Engine

1. Introduction

In JAXA, the development of reusable sounding rocket is in progress (Fig.1)\(^1\). This rocket vertically takes off, reaches to 100 km altitude, lands vertically on the launch site and is launched again within several days. This rocket has 4 engines, each engine thrust is about 31 kN, propellants of this engine are liquid hydrogen and liquid oxygen (Fig.2). This engine was proposed applying high reliability design method which is composed of high accuracy lifetime estimation and envelope of operational limit of each component. Especially, an accurate estimation of lifetime of a turbopump bearings and grasp of operational limit of shaft seals are needed. Purpose of this study is to grasp operational limit of mechanical seals and hybrid ball bearings of turbopumps\(^2\).

![Fig.1 Reusable sounding rocket](image1)

2. Results and discussion of bearing tests

2.1. Bearing tester and test bearing

Figure 3 shows the configuration of the bearing tester\(^3\). The rotating shaft was supported by two test bearings and was driven by an air turbine. A bellows-type gas actuator pressurized by helium gas imposed a thrust load on the test bearings. The pressurized bearing-cavity was sealed with a floating-ring seal. The bearing temperature at the outer-race surface was measured with a thermocouple (Au+0.07 Fe-chromel). Furthermore, the leakage from the floating-ring seal located at the turbine side was sealed with a helium-purged seal.

![Fig.2 Rocket engine for reusable sounding rocket](image2)

![Fig.3 Configuration of the bearing tester](image3)
Figure 4 shows the configuration of an angular contact, hybrid ceramic 25-mm-bore ball bearing. The test bearings had SUS 440C (AISI 440C) rings, as well as Si$_3$N$_4$ balls, and a single outer land-guided retainer with elliptical shaped ball-pockets. This single-guided bearing could reduce the bearing torque to about one-half of that of the conventional double-guided bearing, and more adequate cooling at the outer raceway was obtained, so that high self-lubricating performance and high durability were achieved. The bearings were self-lubricated with a thin film of PTFE transferred from retainer consisting of glass cloth-reinforced PTFE (glass cloth, 45 wt.%; PTFE, 55 wt.%). The retainer was chemically etched with hydrofluoric acid (HF) to improve self-lubricating performance.

2.2. Experimental conditions

This rocket engine is employed expander bleed cycle, and efficiency of fuel (liquid hydrogen) turbopump has an large impact on efficiency of rocket engine system. Especially, bearing coolant flow rate was important, because in small turbopump, the coolant flow rate directly affects the volumetric efficiency of the turbopump. It was tested to grasp critical performance under conditions of bearing cooling flow rate of 0.055~0.3 liter/s (measured by turbine flow meter), test area pressure of 0.4~1.3 MPaG, bearing axial load of 980 N (calculated from pressure of bellows gas actuator and test area pressure), and rotational speed of 40,000~90,000 rpm.

2.3. Bearing test results

Figure 5 shows bearing temperature and saturation temperature in test area vs. cooling flow rate. Saturation temperature was a boiling point of liquid hydrogen under test area presser. Ranging from 0.3 to 0.1 liter/s, bearing temperature showed a liquid phase temperature of coolant at 70,000 rpm. But in case of rotational speed at 90,000rpm, bearing temperature increased sharply to 40.5 K under flow rate of 0.15 liter/s. This temperature exceeded saturation temperature. It was consider that bearing cooling performance of liquid hydrogen was decrease under this condition because it exceeded the saturation temperature and changed into gas hydrogen. Figure 6 shows power loss of bearing which was calculated by the difference of coolant inlet and outlet enthalpy. It was consider that sudden decrease of power loss was caused by decrease of a coefficient of viscosity. At those test conditions, coolant (liquid hydrogen) was boiling and changed from liquid phase into gas and liquid mixture phase. Although bearing outer race temperature was increased, bearing outlet temperature was degrease, that is, generation of heat of bearing was more than cooling capacity. In case of rotational speed of 90,000 rpm, bearing outlet temperature was over saturation temperature under flow rate of 0.15 liter/s. This condition was poor cooling condition.

Figure 7 shows flow rate that bearing outlet temperature agreed with the saturation temperature, and that power loss of bearing decreased abruptly. (In case of 90,000rpm, it was consider that power loss was already decreasing.) In a word, flow rate of a dotted line (Fig.6) was an assumed operational limit of coolant flow rate. In this result, it was presumed that in case of 40,000 rpm, limited flow rate was about 0.15 liter/s, and that of 90,000 rpm, limited flow rate was about 0.23 liter/s.

Figure 8 shows hybrid ceramic bearing after tests. Total operational time was 54 minutes. Contact surface of inner and outer race showed no extreme roughness and it was judged that both of them were in good condition. But, micro cracks occurred on the surface of Si$_3$N$_4$ balls. Those cracks were assumed to be caused by thermal shock$^6$. The ball pocket and rotational guide surface of retainer showed the evidence of hard contact. And color of that contact surface changed into black.

![Figure 4: The configuration of hybrid ceramic bearing](image-url)
3. Results and discussion of axial seal test

3.1. Seal tester and test seal system

Figure 9 shows the configuration of the axial seal tester. This tester is designed to test seal system under cryogenic conditions. In this study, wear amount of mechanical seal nose and surface roughness was measured.

Figure 10 shows the configuration of the mechanical seal and its photographs. A material of mechanical seal nose was a hard carbon (carbon + PTFE). This mechanical seal is used for liquid oxygen turbopump. This mechanical seal performance improves when inlet pressure is higher than outlet pressure. And under this condition, wear amount was increased.

Figure 11 shows a helium purge seal (segmented seal). This helium purge seal can separate high-temperature hydrogen gas and high-pressure liquid oxygen by a wall of helium gas (Helium gas flowed from the openings between two segmented seals.). A material of segmented seal was a soft carbon (carbon + PTFE). In this test, helium purge pressure was 0.3MPa. A contact surface of segmented seal had Rayleigh step, and squeeze load to main shaft was reduced. But lubrication condition of this seal sliding surface was boundary lubrication. A seal ring of a turbine gas seal was segmented seal. And lubrication condition of this seal sliding surface was gas (turbine gas) lubrication.

3.2. Experimental conditions

The tested mechanical seal used welding metal bellows. Therefore acceptable maximal differential pressure was low. In these tests, inlet seal pressure was 1 MPaG (surface pressure: about 0.2 MPa). Cooling flow rate of bearing and seal was 0.4 liter/s. Maximum rotational speed was 30,000 rpm. But liquid nitrogen was used in these tests instead of liquid oxygen and liquid hydrogen.
3.3. Seal test results

Figure 12 shows the result of rotational speed, mechanical seal leak rate and helium purge leak rate. Rotational torque increase at the point of sliding distance of about 80 km, and rotational speed became unstable. It was considered that those are caused by an increase of surface roughness of mechanical seal nose. But mechanical seal leak rate was 0.0 liter/min and did not increase, and seal performance was maintained.

Rotational torque decreased at the point of sliding distance of about 450 km. And flow rate of helium purge seal increased at the point of sliding distance of about 460 km. These evidences showed that, helium purge seal was broken at this sliding distance.

Figure 13 shows wear amount of mechanical seal nose, surface roughness of mechanical seal nose (Rq) and surface roughness of contact surface of mating ring (Rq). Surface roughness of new mechanical seal nose was 0.09 μm, and it decreased to 0.06 μm after experiment with sliding distance of 100 km (running-in). Surface roughness of new mating ring was 0.01 μm. Those of mating ring and mechanical seal nose were the same when sliding distance was about 100 km. Thereafter surface roughness of both mating ring and seal nose increased simultaneously with sliding length. And both...
of them increased to 0.08 μm at sliding distance of about 500 km. There is no difference in surface roughness of tested in continuation operation and that of tested with clean up every 100 km. It was consider that wear debris on contact surface was a cause of torque variation.

Moreover real surface pressure was increased by the existence of wear debris and wear amount of mechanical seal nose suddenly increased.

Figure 14 shows a photograph of mechanical seal nose and mating ring after tests (about 500km). If the contact surface of mechanical seal was cleaned up every 100 km, seal nose height was over 0.8 mm and this mechanical seal can reuse (Fig.14 (b)). In this condition, wear of mating ring was moderate and uniform. But in case of continuation operation, mechanical seal cannot reuse because the seal nose was almost worn out (Fig.14 (c)). In this condition, wear of mating ring was severe.

Figure 15 shows SEM (Scanning Electron Microscope) photograph and EPMA (Electron Probe Micro Analyzer) photograph of mating ring after tests (test of Fig.13 and Fig.14 (c)). An element analyzed by EPMA was carbon. A carbon transferred from mechanical seal nose to mating ring. And many carbons transferred outside (high velocity) more than main shaft side (low velocity). A mechanical seal nose leans by the distribution of temperature and pressure, because the pressure decreased along the flow, on the contrary, temperature increased along the flow. And the contact pressure of the inner side of the nose carbon differs from that of the outside. But this difference of contact pressure is disappeared during running-in. It was considered that a difference of carbon transfer quantity was caused by sliding distance or sliding velocity.

Figure 16 shows a photograph of helium purge seal after tests (test of Fig.13 and Fig.14 (c)). Cryogenic liquid flowed to a helium purge seal from test area when wear amount of mechanical seal nose was over 1 mm, and seal ring was cracked. In this result, this seal system stopped functioning as axial seal for liquid oxygen turbopump.

4. Summary
One of critical performance of hybrid ceramic bearing (flow rate) was measured in liquid hydrogen. Normal operations of bearings are possible under conditions of less than 70,000 rpm of rotational speed and more than 0.2 liter/s of coolant flow rate.

In the case of continuous operation without any cleanup of contact surface, carbon part of mechanical seal would be worn away over 150 km of sliding distance. Wear rate was accelerated by the existence of wear powder on the contact surface. This duration was not sufficient for reusable rocket engine. But a life cycle of mechanical seal extended tremendously when a contact surface was clean up every 100 km.

Fig.13 Wear amount and surface roughness of mechanical seal nose and mating ring.

Fig.14 Photograph of mechanical seal nose and mating ring after tests.

(a) New mechanical seal and mating ring.
(b) Tested mechanical seal and mating ring.
(c) Tested mechanical seal and mating ring.
(Continuation operation)
References


