Performance study of electric pump unit bearings of spacecraft thermal control systems

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Abstract. This paper presents the methodology and test results of rolling bearings of low-flow electric pump units used to ensure circulation of the thermal liquid in the active thermal control system of spacecraft. A multi-position stand has been developed for testing bearings within 40 days in a thermal liquid at a speed of 6000 rpm with the possibility of registering the separator rate speed. As a performance criterion, the clearance in the bearing is used. The conditions of the onset of pitting are investigated.

1. Introduction
The most important part of various purposes spacecraft is a life support system, in particular, thermal control systems, on the performance of which their active life (12 years or more) largely depends. To ensure the circulation of the thermal liquid in the active thermal control system, low-flow electric pump units are used with ball bearings of the electric motor. The main condition for ensuring heat removal from spacecraft instruments throughout the entire active life of increased capacity spacecraft is the durability of the bearings. Operational data on the failure pattern is not enough to select the criterion for the ball bearing performance of these mechanisms. Basically, the failure of ball bearings occurs due to fatigue failure of the working surfaces (pitting), chafe caused by slippage. The results of statistical processing of experimental bench-scale and full-scale tests of aircraft engine bearings in a low-viscosity liquid environment (kerosene, cryogenic components, water, etc.) show that their main damage is working surfaces runout [1]. The paper [2] indicates the bearing friction surfaces state and the vibration severity interrelationship. Bearing runout is associated with micro-slip of balls and raceways [3].
Numerous publications [4–9] have been devoted to this issue in recent years. It was shown in the paper [10], that wearing process of bearing races is observed in the area located close to the sides of the contact zone. The main results relate to larger bearings and heavily loaded bearings. The papers [11–13] summarize the experience over the past 30–40 years and contain recommendations for the selection of ball bearings, assembly designs, and conceptual rules and recommendations for assembly, preloading, and inspection are developed. From the literature, it is known that mainly tests of bearings are carried out as part of a full-scale electric pump unit. In the paper [14], the results of tests of several electric pumps with a rotation frequency of 4000 rpm and a resource of up to 44766 hours are presented. It is shown that there is a clearance increase in bearing during long-term tests. According to the estimates of the authors, the axial force value in the electric pump unit is 2 ... 5N and the radial force is 1N.
The aim of the present work is to study the performance of high-speed electric pump bearings according to the wearing process criterion.

2. Study of research and test procedure

As the test object used a ball bearing having the following characteristics:

- Inner diameter - 6 mm;
- Outer diameter - 19 mm;
- Width - 6 mm;
- Ball diameter - 3.969 mm;
- Number of balls in the bearing - 6 pcs;
- Dynamic loading capacity - 2.17 kN;
- Static loading capacity – 1.16 kN.

The bearings operate in a low-temperature thermal liquid environment, the basis of which is 2,2,4-trimethylpentane. To reduce the runout of bearings, antiwear additives are included in the thermal liquid.

For testing, a test bench was made, figure 1. The test bearings are located in a 4-position sealed chamber 2 filled with a thermal liquid. Figure 1 shows the chamber with the cover removed.

**Figure 1.** Test bench (a) and support node drawing (b): 1 – stand base; 2 – four-position test chamber; 3 – electric motors; 4 – drive system; 5 – temperature control device. The arrows show the acting forces on bearing

Four test modules are used to test multiple bearings simultaneously. Each test module consists of a support unit, figure 1b, with a shaft for mounting the test bearing, a mechanical seal with silicon carbide (SiC) rings, and loading devices in the form of spiral springs.

**Figure 2.** The measuring diagram of the separator rate speed: 1-tested bearing; 2-support; 3-fibre optic sensor; 4-chamber wall; 5-spring; 6-shaft drive.
On the test bearing 1 outer ring, figure 2, mounted on the shaft 6, a support 2 is fixed for transmitting force from the tension and compression springs. The support 2 has a groove for attaching the tension spring. The groove plane passes through the middle of the bearing 1, thus providing a centrally acting radial force. Support 2 has a rod for centring the compression spring 5. In addition, several holes are made on the end surface of the support to reduce weight and freely supply thermal liquid to the friction zone.

The radial and axial forces of a given value are created respectively by a spiral tension spring (not shown) and a coiled compression spring 5. The springs are pre-calibrated on a calibration device.

The test bearings were loaded with radial and axial forces using springs before the start of the experiment. The following nominal forces were established based on the results of the paper [1]: radial force - 1N, axial force - 8N. The tests were carried out at a thermal liquid temperature of 35 ... 40°C.

To seal the inner chamber from the environment and prevent the evaporation of the thermal liquid during the experiment, mechanical seals with silicon carbide rings were used.

To rotate the shafts of the test modules, four DC motors with permanent magnets with a power of 0.3kW DC 12-48 V are used, which are structurally similar to the pump electric motor.

In the study of the main excitation frequencies during the bearing operation, the separator rate speed $n_s$, which is calculated by the formula (1):

$$n_s = \frac{n_0}{2} \cdot \left(1 - \frac{d_w}{D_m}\right)$$

$n_0$ - rotor speed; $d_w$ is the diameter of the balls; $D_m$ is the diameter of the circle passing through the middle of the rolling elements.

A change in $n_s$ during the experiment can indicate the occurrence of micro-slip in the bearing and wearing process. To measure the separator rate speed and the distance from the housing to the separator, the test chamber is equipped with fibre-optic sensors 3, figure 2, which are installed at a distance $\Delta$ from the bearing chamber, as well as pulse sensors that measure the speed of the bearing inner ring. The ratio between the measured frequencies affects the slipping speed of the balls and, accordingly, the likelihood of wear. At the beginning and at the end of the experiment, the clearance in the bearing was measured.

3. Results

At the first stage, commercial bearings of the usual accuracy class were tested at nominal values of radial and axial forces, respectively 1N and 8N. One bearing was loaded with a radial force of 1 N and an axial force of 20 N. During the experiment, the temperature was maintained at 38.50 °C, a rotation speed of 6000 rpm. After 100 hours of testing, the experiment was terminated due to the noise and the thermal liquid temperature rise in the test chamber at 10 °C. Visual inspection of the samples showed that pitting is observed on the bearing raceway with increased axial load, figure 3. The failure pattern was fatigue. Local crumb formation of a metal fragment from the inner ring surface has occurred. Cracks were located perpendicular to the direction of movement of the balls. Transverse cracks were also visible in front of the local fracture zone. No traces of damage to the raceways were observed on the remaining bearings.

At the second stage, we tested ball bearings manufactured using special technology. The experiment time was 40 days, the rotation speed was 6000 rpm, and the temperature was 38.50 °C. The radial and axial forces for each bearing were set in accordance with Table 1. The radial force had a nominal value, and the axial force was increased 2, 3, and 4 times, respectively.

| Bearing number | 1 | 2 | 3 | 4 |
|----------------|---|---|---|---|
| Radial force, N| 1 | 1 | 1 | 1 |
| Axial force, N | 8 | 16| 24| 32|

Table 1. The values of the radial and axial forces of the bearings in the experiment
Figure 3. Image of pitting on the surface of the bearing raceway. The arrow indicates the direction of rolling of the ring.

The graph in figure 4 shows the $n_S/n_R$ ratio change characterizing the sliding change in the ball bearing No. 4 during the experiment. It is seen that its monotonous increase occurs.

Figure 4. Change in $n_S/n_R$ during the experiment
Inspection of the bearings after the experiment did not reveal significant damage to the friction surface in all four samples. The clearances in the bearings changed by 1-4 microns, table 2. The runout around the circumference of the outer ring of the bearing after the experiment increased by 1-1.5 microns.

**Table 2.** The results of the measurement of clearance in the bearings

| Number of bearing | Radial clearance before experiment, µm | Radial clearance after experiment, µm |
|-------------------|----------------------------------------|---------------------------------------|
| 1                 | 14-16                                  | 16-20                                 |
| 2                 | 15-17                                  | 18-20                                 |
| 3                 | 14-16                                  | 16-19                                 |
| 4                 | 16-17                                  | 16-18                                 |

The results obtained are the basis for subsequent studies of the ball bearings performance as part of full-scale units, as well as for a more detailed study of chamber precession and wearing process.

**4. Conclusions**

1. A four-position test bench has been developed with an individual loading system for the tested bearings and registration of the separator rate speed.
2. When the axial load on the commercial bearing increases to 20N, pitting occurs after 100 hours of experience.
3. Bearings made by special technology, after testing for 40 days, have an increase in radial clearance of 1-4 microns.

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