The CRISM Motor/Encoder Assembly and Diaphragm Bearing Assembly Design

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Abstract

This paper will describe the thin section angular contact bearings and WS$_2$ dry film lubrication used on the compact Reconnaissance Imaging Spectrometer for Mars (CRISM) motor/encoder and diaphragm bearing assemblies.

Introduction

CRISM will use targeted observations to search for evidence of aqueous activity and to characterize the geology and composition of surface features on Mars (Figure 1). Global measurements acquired repeatedly throughout the Martian year will provide information on atmospheric water vapor, CO, and aerosols complementary to that from other MRO instruments.

The Optical Sensor Unit (OSU) consists of an optical system, a cryogenic system, and focal plane electronics gimbaled about a single axis to allow scanning over $\pm 60^\circ$ from nadir. Its mechanical design builds on proven technology from previously successful APL instrument designs. The base housings are fabricated from titanium that provides high stiffness and thermal isolation within the same component. The gimbal bearings are a precision assembly designed to operate in a -60°C environment. The gimbal is driven directly by a brushless DC motor paired with a BEI 20-bit incremental position encoder. The encoder disk is co-mounted directly to the bearing shaft beside the motor rotor and the read heads are mounted to the bearing housing alongside the motor stator. The electrical signals and purge are passed through a twist capsule in the center of the motor/encoder bearing assembly. A second bearing pair is mounted in a parallel diaphragm bearing housing that provides high stiffness in the lateral directions to the gimbal axis and flexibility along the gimbal axis to compensate for differential expansion of the instrument and spacecraft. The spectrometer housing is passively cooled to -90°C using a flexible link to the anti-sunward radiator. The anti-sunward radiator passes through the center of the diaphragm bearings in a thermally isolated mount, and thus rotates with the OSU; its FOV is independent of gimbal position.

Figure 1. CRISM Instrument

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CRISM Thermal Design

The CRISM thermal design provides both active cooling of the IR focal plane to cryogenic temperatures to reduce dark current and passive cooling of the spectrometer housing to -80°C for low background. Simultaneously it keeps the electronics section near -40°C (Fig. 2). Cryogenic cooling is provided by three Ricor K508 integral Stirling cryocoolers. The multi-cooler configuration requires “thermal switching” between coolers. A cryogenic diode heat pipe assembly consisting of heat pipes and a thermally isolating mounting assembly connects the active cooler with the focal plane while isolating if from the two dormant coolers. Each of the three diode heat pipes is connected to the focal plane on one end and to a cooler on the other. The focal plane electronics are mounted in the bottom of the gimbal housing and are maintained at -40°C. The housing along with tantalum plates provide EMI shielding and minimize cable length to the focal planes. Table 1 lists the CRISM expected flight operating temperatures.

Table 1. CRISM Flight Operating Temperatures

<table>
<thead>
<tr>
<th>Component</th>
<th>Min.</th>
<th>Max.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Telescope</td>
<td>-75</td>
<td>-25</td>
</tr>
<tr>
<td>Spectrometer Housing</td>
<td>-110</td>
<td>-78</td>
</tr>
<tr>
<td>Optical Bench</td>
<td>-75</td>
<td>-25</td>
</tr>
<tr>
<td>VNIR FPA</td>
<td>-60</td>
<td>-20</td>
</tr>
<tr>
<td>IR FPA</td>
<td>110</td>
<td>120</td>
</tr>
<tr>
<td>Cryo-Coolers</td>
<td>-25</td>
<td>20</td>
</tr>
<tr>
<td>Diode Heat Pipes</td>
<td>95</td>
<td>115</td>
</tr>
<tr>
<td>Electronic Boards</td>
<td>-25</td>
<td>20</td>
</tr>
<tr>
<td>Motor/Encoder</td>
<td>-40</td>
<td>10</td>
</tr>
<tr>
<td>Shutter Motor</td>
<td>-71</td>
<td>0</td>
</tr>
<tr>
<td>Integrating Sphere</td>
<td>-71</td>
<td>0</td>
</tr>
<tr>
<td>Anti-Sunward Radiator</td>
<td>-130</td>
<td>-90</td>
</tr>
</tbody>
</table>

Figure 2. CRISM Thermal Zones
CRISM Bearing System

The CRISM instrument rotates the OSU ±60° from nadir. The OSU weighs 20.9 kg (45.9 lbf) and is supported by the motor/encoder (Figure 3) and the diaphragm (Figure 4) bearing assemblies. The motor-encoder side used a thin-section angular-contact duplex pair mounted back to back and was designed to take the non-axis moments, the entire thrust load, and its share of the radial loads associated with launch event (Figure 5). These bearing were designed for a 266.9-N (60-lbf) axial preload. The diaphragm side used a thin-section angular-contact duplex set mounted face to face and was designed to take only the radial loads associated with the launch event. The diaphragm bearings were designed for a 66.7-N (15-lbf) axial preload.

Preloading

Angular contact bearings should be preloaded as lightly as necessary to achieve the desired results. A duplex pair is a pair of bearings that have a pre-determined amount of preload built into them. This was accomplished by grinding the inner or outer ring a sufficient amount to eliminate all internal clearance within the bearing commonly referred to as the preload offset. There are, however, several disadvantages to preloading bearings:
- Increased running torque
- Sensitivity to differential thermal expansion
- Sensitivity to misalignment
The CRISM duplex bearing pairs were separated by titanium spacers so that the preload offset would remain constant over temperature. However, the difference between the 440C inner and outer rings and the titanium shaft and housing resulted in a reduction of clearance as temperature decreased (Figure 6). A reduction of clearance results in a decrease of the contact angle. However, the bearings are only going to experience substantial axial loads during the launch. The bearings were tested to -196°C and continued to rotate freely. The few disadvantages of preloading are more than offset by the following advantages:

- Reduces axial and radial runout of the rotating shaft. Required for the encoder disk to read head alignment
- Reduces the shaft deflection under load and improves its assembled stiffness
- Removes free play in the bearing set, keeping the bearing set loaded in-order to avoid skidding of the balls
- Minimizes the peak stresses that occur during the maximum loading events by ensuring the load on the bearings is shared by more balls in each bearing
- Decreases bearing noise

In addition to the axial preload, the CRISM bearings employed a light interference fit, 12.7 µm (0.0005 in) in the bearing/shaft fit and 15.2 µm (0.0006 in) in the bearing/housing fit.
Figure 5. CRISM motor/encoder spindle

Figure 6. CRISM motor/encoder bearing clearance
CRISM Bearing Lubrication

The thermal analysis of the CRISM motor/encoder and diaphragm bearing assemblies indicated that they would operate at -40°C and -90°C respectively. The analysis indicated that the diaphragm bearing assembly would be too cold for oil or grease lubrication. Our tests showed that -40°C was beyond the acceptable temperature range for Pennzane synthesized hydrocarbon based oils and greases and -90°C was beyond the accepted range for perfluorinated polyether based Brayco oils and greases. It was desired to use identical lubrication in both bearing assemblies for the following reasons:

- To have nearly identical bearings and identical lubrication in both bearing assemblies
- To simplify bearing/lubrication testing
- Once the bearings and lubrication were selected, an single qualification test could be applied to both assemblies

The three choices of dry film lubrication that were considered were:

1. Ion plated lead
2. Sputtered MoS\(_2\) (molybdenum disulfide)
3. WS\(_2\) (tungsten disulfide)

Ion-plated lead bearings were successfully used in the Compact Remote Imaging SPectrometer (CRISP) tracking mirror assembly on the Comet Nucleus TOUR (CONTOUR) spacecraft. The CRISP tracking mirror assembly bearings were a Barden precision angular contact duplex pair with a bronze cage. They were purchased through BEI as part of their motor/encoder assembly. This was done since they were the same bearings used in the BEI motor/encoder and they had used an identical set previously in the SABER\(^3\) instrument on the TIMED spacecraft. However, we were unable to procure ion-plated lead thin-section bearings at the time. If it were possible to procure them in the future, we would definitely recommend trying them.

There is a great deal of literature touting the benefits of sputtered MoS\(_2\). Dry film lubrication tests for MoS\(_2\) and WS\(_2\) were conducted for the CRISP cover and release mechanism. Sputtered MoS\(_2\) is supposed to work great in vacuum, however, moisture absorption can cause severe performance degradation. Most of the CRISP cover and release mechanism qualification testing was conducted in ambient conditions including a test on NASA’s Low Gravity Experiment aircraft (the “Vomit Comet”). Tests comparing sputtered MoS\(_2\) and WS\(_2\), showed WS\(_2\) to be clearly superior to sputtered MoS\(_2\) for this application. Sputtered MoS\(_2\) seemed to exhibit tremendous stiction in tests conducted in ambient conditions where as stiction was virtually undetectable with WS\(_2\). In fact, we believe that the cover and release mechanism would not have worked at all using sputtered MoS\(_2\) in ambient conditions. Sputtered MoS\(_2\) was also used in the bearings for the MDI\(_S\) instrument on the MESSENGER spacecraft launched in 200X. In addition to the well known issues of sputtered MoS\(_2\), this instrument suffered two additional issues:

1. Smoothness
2. The sputtered MoS\(_2\) coating is a hard and brittle coating

The MDI\(_S\) instrument was able to meet its performance requirements with the sputtered MoS\(_2\) coated bearings. However, the initial flight assembly had to be replaced and a new assembly built up because the sputtered coating had become cracked and resulted in large unacceptable torque spikes. Based on our experiences, we would NOT recommend sputtered MoS\(_2\) coatings.

WS\(_2\), Dicronite, was chosen for the CRISM motor/encoder and diaphragm thin section bearings. However, we were unable to locate any specific examples or find any heritage on WS\(_2\)-lubricated bearings used in space applications in the available literature. However, based on our previous success with WS\(_2\) on the CRISP instrument, we were optimistic that WS\(_2\) could also work in bearings. We also knew that the bearing companies offered it as an option and that the WS\(_2\) coaters coated bearings. We decided to conduct our own WS\(_2\) coated thin-section Teflon-toroid bearing test.
Identical sets of sputtered MoS$_2$-coated and WS$_2$-coated thin-section Teflon-toroid angular contact duplex-pair bearings were purchased simultaneously (Figure 7). Prior to running the tests, an initial set of pictures using a scanning electron microscope were taken of both types of bearings (Figure 8 through Figure 10). Figures 8 through 10 show that MoS$_2$ appears to have a much rougher surface than WS$_2$. We believe this is why the MDIS bearings never "felt" smooth.

Figure 7. WS$_2$ and MoS$_2$ life test bearings

Figure 8a. WS$_2$ Coating

Figure 8b. MoS$_2$ Coating
Following the test of the WS$_2$-coated bearings in which 57952 cycles were completed, we were concerned that the bearings seemed much rougher than they were prior to the test. After a visual inspection of the bearings, a significant amount of debris in the bearings was found. Post-test pictures were taken with a standard microscope, Figure 11 through Figure 13. There were several types of contamination found:

- White particles
- Brown particles on the toroids
- Brown film on the raceways

We were concerned that the contamination found in the bearings following the test would present a real problem, both in terms of lifespan and smoothness. However, upon review of the data, the rougher feel was not adversely affecting the motor control system. The bearings were disassembled and further inspected. We could find no evidence of wear in the raceways or the WS$_2$ coating. The life test was also significantly longer than the expected lifetime of the instrument at Mars.

**Bearing Test Conclusion**

The WS$_2$ coating is more than adequate for a slow moving oscillating gimbal requiring dry film lubrication. We were satisfied enough with the WS$_2$ performance that we did not test the sputtered MoS$_2$ bearings, we had found the solution we were looking for. The Teflon toroids would likely continue to break down and contaminate the bearings with additional particles that the instrument may no longer be able to rotate as precisely as is required. We believe the Teflon toroids will ultimately be the life limiting factor of these
bearings. Although this may be an advantage for continuously rotating higher speed mechanisms acting as an additional lubricant.

Figure 11. Post life test bearing

Figure 12. Post life test bearing raceway

Figure 13. Post life test bearing raceway

CRISM Bearing Anomalies

Bearing Ball Spacing
The APL Space Department has used different sizes of thin-section Teflon-toroid-spaced bearings on several programs and tests. One feature that we have always noticed with these types of bearings is the potential for a "large" gap to occur between a ball and toroid (Figure 14).

We attribute both the high running torque and the poor feel to the non-uniform ball-toroid spacing. We believe that non-uniform ball-toroid spacing can result in some various amounts of pressure between all the toroids and balls. This can result in unpredictable friction between the ball and toroid causing an increase in the running torque. It may also result in a stick-slip situation between the ball and toroid resulting in torque spikes or non-smooth rotation. Additionally, it could lead to rapid wear of the toroid. The toroid wear particles could wind up in the raceways as contamination, also causing anomalous
torque spikes. Grease and oil lubricated bearings may exhibit the same problems as the CRISM dry film lubricated bearings to a far less noticeable degree due to the grease or oil between the ball and toroid. The evidence for this conclusion with the CRISM bearings is two fold:

1. The significant amount of Teflon contamination found in the bearings following the run-in procedure described in Appendix 1
2. The drastic change in torque level and smoothness following the high-pressure air blow through cleaning procedure described in Appendix 1

Reproducible Torque Spikes
Another type of torque spike was also noticed with the CRISM bearings that could be easily reproduced based on the operation of the spindle. During the run-in procedure in Appendix 1, unidirectional rotation resulted in extremely smooth running torque. However, CRISM was intended to oscillate ±60°. A reproducible torque spike occurred following a change in direction of rotation. Rotating the spindle backwards and forwards, sometimes referred to as “safe cracking”, torque spikes, equal to or greater than the nominal running torque, would result within several degrees of rotation following the reversal. A ball can never remain rolling between surfaces that form an angle to each other. All angular-contact ball bearings create an angle between the two raceways. Therefore, as the bearings rotate, the balls produce a gyroscopic motion in addition to rolling. Pressure or drag friction between the toroid and the adjacent balls also seems to deflect the toroids based on the direction of rotation. Thus, the gyroscopic motion of the balls and pressure between toroids and adjacent balls appears to cause the toroids to align themselves based on the direction of rotation. This behavior appears to be attributable to the vast majority of toroids aligning themselves (Figure 15). When the spindle reverses, as do the balls, the toroids flip and align themselves in the opposite direction (Figure 16). As the toroids flip and align themselves in the opposite, direction, a torque spike resulted. This effect was significantly reduced following the cleaning procedure. Once the balls and toroids became more evenly spaced, the drag friction between the toroid and adjacent balls was reduced, thus reducing the ability of the toroids to align themselves. This was also noticed following the Christmas holiday. As the bearings sat over the holiday, pressure between the balls and toroids either slightly re-spaced the balls or caused the Teflon to cold flow resulting in bearings the felt much better after having sat for an extended period. However, following another run-in, the bearings quickly resorted to their pre-holiday behavior.
Figure 15. Toroids in “down” position  
Figure 16. Toroids in “up” position

Unfortunately, we never photodocumented this effect, thus, it was extremely difficult to find pictures of the flight hardware demonstrating this effect clearly. Figure 15 clearly shows all the toroids uniformly aligned in the “down” position. Figure 16 shows most of the toroids in aligned in the “up” position, including most of the ones at the top of the bearing. This effect can not be reproduced with a single un-mounted bearing.

CRISM Bearing Assembly Procedure

The CRISM bearings were removed from the manufacturer’s packaging and visually inspected to verify that they conformed to the documentation and were marked properly (Figure 17). Once we were sure that the bearings were correctly marked and free of contamination, they were assembled as follows:

1. The bearings and spacers were stacked and aligned per drawing and documentation (Figure 18)
2. The bearings and spacers were placed in an assembly fixture (Figure 19) specifically designed to keep them aligned during assembly
3. The bearing retainer was placed on the assembly (Figure 20)
4. The bearing shaft (Figure 21) was cooled in liquid nitrogen
5. The cooled bearing spindle was quickly assembled in the bearing assembly fixture. Weights were placed on the assembly to ensure that it remained seated against the top bearing.
6. The bearing assembly was quickly placed in a N₂ purged vessel and allowed to equilibrate for 24 hours (Figure 22)
7. The spindle bearing assembly was then visually inspected and checked for “feel”. It was noted that there was a significant amount of pressure on the spacers between the two bearings on both the Engineering Test Unit (ETU) and the flight unit. The ETU and the flight assembly were the first bearing assemblies with spacers that we had assembled using the liquid nitrogen technique. Previously, the bearings were duplex pairs without spacers. We were concerned that with the shaft being significantly cooler than the bearings and spacers, that as it warmed and expanded, there would be very little pressure between the bearings and spacers or worse, a gap. However, that never materialized
8. The entire spindle bearing assembly was then cooled in liquid nitrogen
9. The cooled spindle bearing assembly was quickly assembled in the titanium motor/encoder/bearing housing
10. The assembly was then purged and allowed to equilibrate for 24 hours
11. The assembly was visually inspected and checked for “feel”
12. We were not happy with the “feel” of the flight assembly and took the steps outlined in Appendix 1

There were two main issues with the flight bearing assembly, 1) high running torque; 2) torque spikes that made the assembly not feel smooth as it rotated. We were confident that contamination was not the problem from pre- and post-assembly inspections causing either of these problems.
Figure 17. CRISM flight bearings
Figure 18. Bearings stacked and aligned
Figure 19. CRISM bearing alignment fixture
Figure 20. Bearing assembly
Figure 21. CRISM bearing shaft
Figure 22. Purging bearing assembly
Appendix 1

Table 2 lists the operations and measurements made to the flight motor/encoder bearing assembly. Most measurements were an average running torque. Some measurements were a peak torque when there was a significant torque spike to the running torque. On several tests where many measurements were made, an average value is reported and denoted by and Avg. in the Torque column.

### Table 2. Bearing Assembly Operations

<table>
<thead>
<tr>
<th>Date</th>
<th>Operation</th>
<th>Torque mN-m (oz-in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>12/04/2003</td>
<td>Initial torque measurements</td>
<td>111.6 (15.8)</td>
</tr>
<tr>
<td>12/16/2003</td>
<td>Run-in @ 60 RPM, 19 minutes (1140 Revolutions)</td>
<td>111.6 (15.8) nominal 190.7 (27.0) peak</td>
</tr>
<tr>
<td></td>
<td>Run-in @ 60 RPM, 25 minutes (1500 Revolutions)</td>
<td>143.0 (20.25) Avg.</td>
</tr>
<tr>
<td></td>
<td>Cool off</td>
<td>381.3 (54.0)</td>
</tr>
<tr>
<td>12/23/2003</td>
<td>Run-in @ 60 RPM, 22 minutes (1320 Revolutions)</td>
<td>254.2 (36.0)</td>
</tr>
<tr>
<td></td>
<td>Loosen retaining nut</td>
<td>158.9 – 190.7 (22.5 – 27.0)</td>
</tr>
<tr>
<td></td>
<td>Tighten retaining nut, 14.2 N-m (10.5 ft-lb)</td>
<td>158.9 – 190.7 (22.5 – 27.0)</td>
</tr>
<tr>
<td></td>
<td>Tighten retaining nut, 20.3 N-m (15 ft-lb)</td>
<td>254.2 – 286.0 (36.0 – 40.5)</td>
</tr>
<tr>
<td></td>
<td>Loosen retaining nut, 17.0 N-m (12.5 ft-lb)</td>
<td>190.7 (27.0)</td>
</tr>
<tr>
<td></td>
<td>Tighten retaining nut, 19.0 N-m (14.0 ft-lb)</td>
<td>190.7 (27.0)</td>
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<tr>
<td></td>
<td>Run-in @ 60 RPM, 20 minutes (1200 Revolutions)</td>
<td>508.5 (72.0)</td>
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<tr>
<td></td>
<td>Loosen retaining nut</td>
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<td></td>
<td>Tighten retaining nut, 17.0 N-m (12.5 ft-lb)</td>
<td>286.0 (40.5)</td>
</tr>
<tr>
<td></td>
<td>Loosen retaining nut</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Run-in @ 60 RPM, 20 minutes (1200 Revolutions)</td>
<td>158.9 (22.5)</td>
</tr>
<tr>
<td></td>
<td>Tighten retaining nut, 17.0 N-m (12.5 ft-lb)</td>
<td>190.7 (27.0)</td>
</tr>
<tr>
<td>1/5/2004</td>
<td>Loosen retaining nut and re-measure torques</td>
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</tr>
<tr>
<td></td>
<td>Set-1</td>
<td>238.7 (33.8) Avg.</td>
</tr>
<tr>
<td></td>
<td>Set-2</td>
<td>317.8 (45.0) Avg.</td>
</tr>
<tr>
<td></td>
<td>Changed torque measuring method for all future measurements.</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Peak torque measurements</td>
<td>208.3 (29.5) Avg.</td>
</tr>
<tr>
<td></td>
<td>Tighten retaining nut, 10.2 N-m (7.5 ft-lb)</td>
<td>240.8 (34.1) Avg.</td>
</tr>
<tr>
<td></td>
<td>Run-in @ 60 RPM, 20 minutes (1200 Revolutions)</td>
<td>182.2 (25.8) Avg.</td>
</tr>
<tr>
<td></td>
<td>Peak torque measurements</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Best &quot;feel&quot; so far</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Tighten retaining nut, 13.6 N-m (10.0 ft-lb)</td>
<td>317.1 (44.9) Avg.</td>
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<tr>
<td></td>
<td>Run-in @ 60 RPM, 20 minutes (1200 Revolutions)</td>
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</tr>
<tr>
<td></td>
<td>Peak torque measurements</td>
<td>148.3 (21.0) Avg.</td>
</tr>
<tr>
<td></td>
<td>The best these bearings have ever &quot;felt&quot;</td>
<td></td>
</tr>
<tr>
<td>1/6/2004</td>
<td>Removed retaining nut, rotor, and spacer</td>
<td></td>
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Acknowledgements

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References


