# HIGH PRECISION DUPLEX BEARING WITH THERMAL OFF-LOAD DEVICE FOR THE NIRSPEC WHEEL SUPPORT MECHANISMS

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## ABSTRACT

Austrian Aerospace has the privilege to develop the wheel support mechanisms for the Filter Wheel and the Grating Wheel of the Near Infrared Spectrograph (NIRSpec) Instrument of the James Webb Space Telescope (JWST).
One of the key components of the wheel support mechanisms are the bearings which shall

- have highest accuracy,
- have low friction torque during operation,
- withstand considerable launch loads and
- withstand cryogenic operating temperatures of 30K.

The chosen bearing is a custom-made hard preloaded flanged duplex ball bearing in back to back configuration with a novel thermal off-load device.

## 1. INTRODUCTION

The NIRSpec Instrument will be the spectrograph of the James Webb Space Telescope in the wavelength range of 0.6 to 5 μm. The instrument is equipped with two wheels, a filter wheel to place one of eight filters, and a grating wheel to place one of seven gratings or a prism within the optical path. The gratings shall be placed with highest accuracy, for the bearing repeatability a target value of better than 1 arcsec is planned, which results in a radial repeatability of better than 0.1 μm for the selected bearing.

The friction torque shall be minimized during operation at low temperature to prevent excessive heat input by the motor dissipation, and to allow minimizing the conducted heat over the required cross sections of the cryogenic harness. Operating temperature is 30K. Launch temperature of the mechanism is ambient; friction torque at ambient is allowed to be higher compared to the cryogenic value. However testing at ambient temperature must be possible. Due to the low number of rotations (<100000) life time is not an issue for the bearing and its lubrication. For the mechanism a cantilever design was requested by the customer which implied a short distance between the bearing races of the employed duplex bearing. The wheels are developed by Carl Zeiss Optronics GmbH, Germany (reference [2]).

## 2. DESCRIPTION OF THE MECHANISM

Two identical wheel support mechanisms are used on the NIRSpec Instrument; one supports the filter wheel, and the other one supports the grating wheel. The wheel support mechanisms shall provide safe and accurate support of the wheels on the optical bench plate which is made of Silicon Carbide (SiC). An enclosed motor actuates the wheel rotation. Figure 1 presents the design:

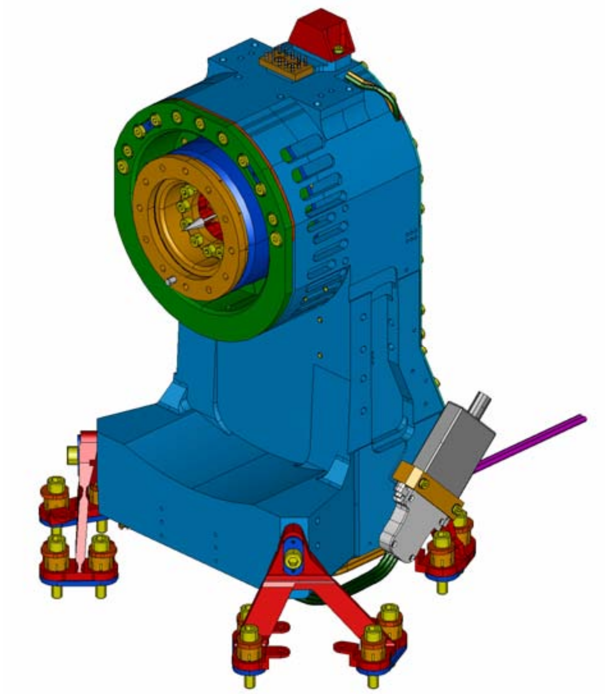

*Figure 1. NIRSpec Wheel Support Mechanism.*

The main structure is connected to the optical bench plate by Kinematic Mounts. The main structure supports the central bearing. It encloses the cryogenic motor type C116 manufactured by Tieftemperaturlabor / Freie Universität Berlin (TTL/FUB). The following main parameters apply for the mechanism:

| | |
|---|---|
| Design Concept | Cantilever Design |
| Main Structural Material | Titanium Alloy |
| Overall Dimensions | 322 x 264 x 193 mm |
| Mass | 6.660 kg |

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## 3. BEARING TYPE, LAYOUT AND DESIGN

### 3.1. Bearing Design

The bearing is a ball bearing in back to back configuration. It consists of an outer ring with 2 separate races, 2 rows of balls, and 2 inner rings that are clamped together with 12 bolts of Titanium alloy. Invar segments are placed between the inner rings.

Figure 2 presents a cross-section of the bearing:

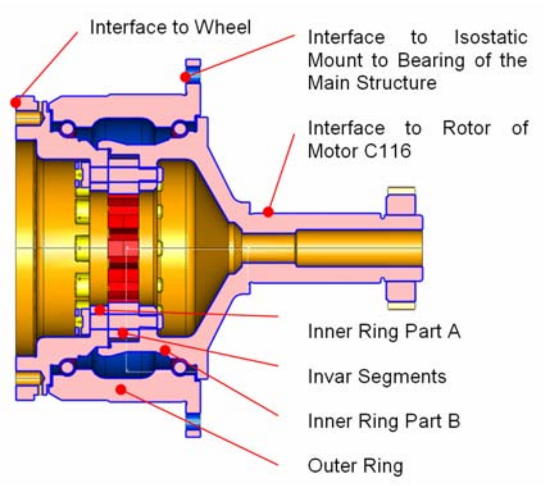


*Figure 2. Bearing Cross Section.*

Bearing rings and balls are of the same material to prevent thermal impact. Bearing pre-load is applied by clamping the inner ring halves with the bolts. The pre-load is adjusted by modification of the contact area between the inner ring parts.

The main parameters of the bearing assembly are provided in the table below:

| | |
|---|---|
| Overall Dimensions | Ø 103 mm x 112 mm |
| Pitch Circle Diameter | 65.1 mm |
| Center Plane Spacing | 31 mm |
| Quantity and Size of Balls | Ø 4.7625 mm (2 x 33 off) |
| Ring and Ball Material | AISI 440C |
| Cage Material | Duroid 5813 |
| Lubricant | Sputtered $MoS_2$ by ESTL, applied on the races. |
| Mass | 1.480 kg |
| Part Count | 102 |
| Bearing Supplier | ADR |

### 3.2. Accuracy

The bearing design was mainly driven by the demanding accuracy requirement.

A super duplex design was chosen for highest bearing precision: A single bearing assembly has approximately a run-out of a factor of 10 worse compared to a super duplex bearing. Therefore to reach the same angular misalignment a 10 times greater bearing distance would be required. Such a design does not fit within the provided envelope of the cantilever design.

The bearing is made of AISI 440C with special heat treatment for cryogenic applications as it was not possible to use special steel like Cronidur 30 due to programmatic reasons. The main structure is made of Titanium alloy; the wheels are made of an Aluminium Zinc alloy. To prevent deformation of the bearing due to different thermal expansion of the different metals during cool-down flanged interfaces to the main structure of the mechanism and to the wheels with isostatic mounts are foreseen. The bearing has no typical cylindrical bearing seat but just centering diameters instead. The interface to the rotor of the motor which is realized as a hollow shaft is integrated into the bearing design.

A high precision bearing supplier, ADR, has been selected as development partner to ensure highest manufacturing quality. Relatively thick walled rings prevent deformation of the bearing during the grinding process. Manufacturing of both bearing races of the rings is done on the same setting to minimize run-out. Steel balls of highest precision (Grade 3) are used.

The allowable peak Hertz'ian stresses are limited to a value so that no permanent deformation of the balls and races of more than 0.1 µm occurs. This means to limit the peak Hertz'ian stress to a value of less than 2360 MPa. To safe-guard the solid lubricant during long term storage the Hertz'ian stress under pre-load only is less than 1500 MPa. Hard pre-load is applied. The pre-load is dimensioned so that nominally no gapping of the balls occurs during random vibration to prevent deterioration of the balls, the races or the lubricant.

As the cryogenic temperatures do not allow fluid lubricants, dry lubrication was selected. Sputtered MoS2 by ESTL is applied on the races. The layer thickness was controlled to stay within a ± 0.1µm tolerance range. Thickness measurements confirmed the layer thickness and uniformity.

The following photograph (figure 3) presents the bearing assembly:

*Figure 3. Bearing Assembly.*

**3.3. Thermal Off-Load Device**

During launch a high pre-load is required to prevent gapping, whereas during operation a low pre-load is desired to minimize friction torque. For off-loading of the bearing after launch active pre-load systems are known, e.g. the Bearing Active Preload System (BAPS) by ESTL. As the launch of the NIRSpec Instrument will be at ambient temperature, and operation will be at cryogenic temperature of 30K, a thermal off-load device was realized that uses the temperature difference as means of actuation.

Between the inner rings 12 segments of Invar M93 are placed, so that during cool-down of the bearing the pre-load of the bearing is reduced as the outer ring shrinkage exceeds the shrinkage of the inner ring assembly: The axial contact point distance of the outer ring is determined by the outer ring which is of one piece of bearing steel. The axial contact point distance of the inner ring is determined by a stack of the inner rings of bearing steel and the Invar segments of a defined thickness. Pre-load is set during manufacturing at room temperature. When the bearing is at low temperature the reduction of the axial contact point distance of the outer ring exceeds the reduction of the axial contact point distance caused by the differences in thermal expansion.

A segmented design of the Invar parts was preferred instead of a full ring to prevent bimetallic effects in radial direction. The clamping bolt is positioned in the center of each segment. Special attention is given to the accurate positioning of the segments in the bearing by use of a dedicated alignment and insertion tool for the integration of the segments between the inner rings.

The thermal off-load device was dimensioned to reduce the bearing friction torque to at least 25% of the ambient value, but not less than 10% of the ambient value as for accuracy reasons no clearance during operations is allowable. Target value is 20%.

Influence of the thermal off-load device on the accuracy is minimised by combined machining of the 12 segments of each bearing. Bolt scattering is minimised by manufacturing provisions such as defined tensioning sequence, bolts of same batch, threads of same quality and manufacturing by same tool, and $MoS_2$ lubrication of the bolts. The impact of bolt scattering is moreover minimised by design by creating a stiff design in the bolted area, and a weaker structure connecting this area with the race area, which again has a stiff design. See these "waists" of the inner rings in the bearing cross section of figure 2.

The following photograph (figure 4) presents the Invar segments aligned on a central mandrel that is removed after inner ring assembly:

*Figure 4. Invar Segments on Mandrel.*

Centering of the inner ring halves during assembly is done by the compressed balls itself; the cylindrical fit of the inner rings is mainly used during manufacturing, and remains as a limiter with loose fit. However these potential sliding areas (due to different thermal expansion) are also coated with $MoS_2$ to minimize risk of adhesion in case of fretting. Environmental conditions allow leaving the Invar without protective coating.

As the off-load device is sensitive to dispersions of the thermal expansion of the AISI 440C material of the rings, and of the thermal expansion of the Invar M93, samples of the readily heat treated material batches were taken and their thermal expansion was measured under vacuum with a Michelson laser interferometer measurement system (ASTM Standard E 289 - 95).

A patent of this thermal off-load device is filed (reference [1]).

Alternative design candidates for thermal off-load devices were investigated: One was a design using rings of different metals: It shows that among available bearing materials the coefficient of thermal expansion (cte) differs significantly. Therefore using different material for the inner ring and for the outer ring would lead to different pre-load at different temperature, as desired. Alternatively also the balls could be manufactured of a different type of bearing steel compared to the ring material. However these solutions were identified in being sensitive to dispersions of the cte in terms of knowledge, batch variation, un-isotropic behaviour, or manufacturing impacts, and therefore discarded.

Another investigated alternative solution was to equip the inner ring with two circumferential grooves, one on the inner diameter and another one on the outer diameter. The Invar segments would be inserted at elevated ring temperature so that later they are clamped in any expected environment of the instrument. The symmetrical design prevents rotation of the races during cool-down. Figure 5 presents the stress distributions and the deformation of an inner ring cross section of this design candidate, verifying that the applying stresses are below the allowable stresses of the concerned materials:

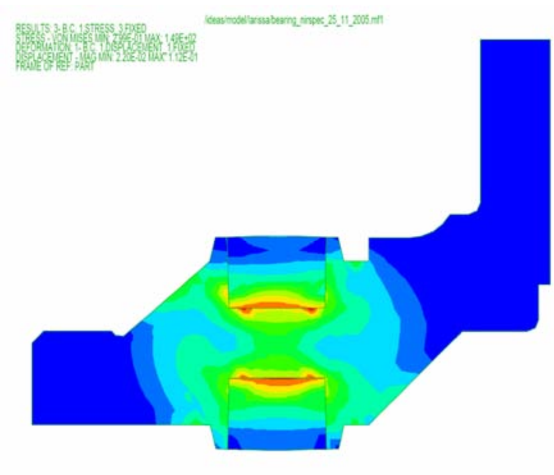


*Figure 5. Alternative Solution Stress Plot.*

This design candidate was considered promising, however it was not finally selected, as unpredictable setting effects between the inserted segments on the highly loaded contact areas during temperature cycling could not be excluded, and as the manufacturing sequence of grinding the circumferential grooves, then inserting the segments into the heated inner rings, and after that the grinding of the races was not practicable and could not be accommodated in the schedule.

### 3.4. Stress-Corrosion Cracking Considerations

Stress-Corrosion Cracking (SCC) is critical for the AISI 440C bearing material which has low resistance to stress-corrosion cracking according to ECSS-Q-70-36A. For programmatic reasons it was not possible to use less sensitive material as e.g. Cronidur-30 which is a Nitrogen alloyed stainless steel for corrosion resistant fracture tough bearings that was already used for cryogenic aerospace mechanisms (reference [3]). The following measures to minimize the risk of SCC are taken: The highest grade of raw material quality is used for the bearings. A special heat treatment for cryogenic application is applied. The rings are dye penetrant tested after final machining to detect possible failures on the ring surfaces. Then the bearing rings and balls are passivated to improve their corrosion resistance. The ring surfaces are shot-peened to reduce residual stresses.

The bearing will not be exposed to corrosive environment, but it will be exposed exclusively to cleanroom or inert gas environments, or to vacuum. Exposure time to air will be minimized.

### 3.5. Optimisation of the Internal Geometry

To optimize the internal geometry of the bearing, as the race radii, the ball diameter and number, the contact angle, the central plane distance and to predict the bearing friction torque the CABARET software (supplied by ESTL) was used. However a smaller ball size was preferred for accuracy reasons, because small balls have smaller manufacturing tolerances and an averaging among the balls within the bearing more likely equalizes dimensional tolerances.

The most critical load for the bearing among the static and the vibration load cases was investigated, and design optimizations on the structure were performed to minimize these loads. The worst case load combinations were taken as conservative approach. A damping factor of $\zeta = 2\%$ (therefore an amplification $Q = 1 / 2\zeta = 25$) for the structure was assumed. The $3\sigma$ values of the axial load, the radial load, and the moment load on the bearing are assumed to act at the same time, combined with the worst case load directions as the bearing load is also depending on the combination of the load directions. This is a conservative standard assumption, leading to a worst case pre-load and bearing friction torque. For the sputtered $MoS_2$ coating a coefficient of friction of 0.05 was applied (reference [4]). An optimum solution was found providing low friction torque and at the same time high stress margins under load and under pre-load only, while preventing gapping of the bearing.

The results were compared with the results of the ADR proprietary software and found in good accordance.

## 4. BEARING MOUNT

### 4.1. Isostatic Mount

The bearing is connected to the main structure using an isostatic mount of Titanium alloy, allowing compensation of the different thermal expansion of the Titanium alloy and the bearing steel while maintaining high stiffness of this connection. Deformation of the bearing due to thermal stresses is minimized. The design consists of a slotted, thin walled cylindrical structure. Figure 6 presents the isostatic mount:

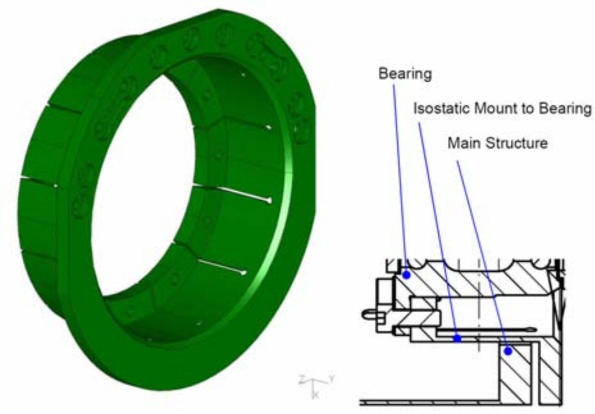


*Figure 6. Isostatic Mount to Bearing.*

### 4.2. Mechanism Design to Low Bearing Loads

For the mechanism a cantilever design was requested by the customer. For the bearings such a design leads to an unfavorable combination of loads as axial loads, radial loads, and moment loads act on the bearing. Generally, when a cantilever is accelerated in lateral direction, the upper end does not only deflect lateral, but there is also a rotation of the upper end of the cantilever. For the support mechanism of the wheel a sophisticated design was developed, which eliminates rotations perpendicular to the bearing rotation axis during random vibration in its main modes in X, Y and Z direction to minimize bearing loads (bending moments) and so to allow low pre-load and following low bearing friction torque.

For the Y direction, which is the direction of the bearing axis, an eccentricity of the bearing mount on the main structure was introduced, so that when the main structure is bent, the bearing mount provides equivalent counter-bending. The eccentricity was realized by a slit in the contact area between bearing mount and main structure. The following detail (figure 7) presents this design feature:

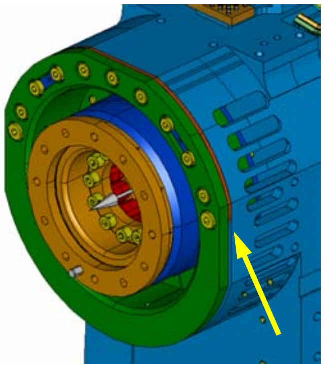

*Figure 7. Bearing Mount Eccentricity.*

The deformation plot (figure 8) presents the main mode shape in Y-direction. It shows that the wheel is not rotated by the Y-acceleration, therefore applying very low moment load on the bearing:

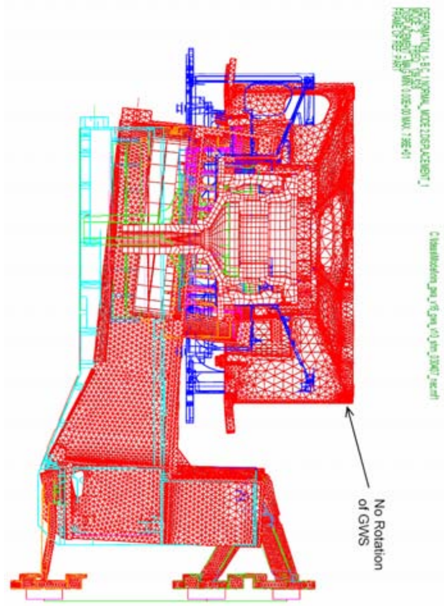


*Figure 8. Mode Shape Y-direction.*

This compensation requires high bending stiffness of the main structure to limit the amount of necessary counter-bending which influences the first fundamental Eigen-frequency of the assembly. This first fundamental Eigen-frequency is specified to be greater than 140 Hz.

A similar concept is followed for the Z-direction which is the lateral direction of the mechanism. By tuning the lateral stiffness of the isostatic mounts of the mechanism, and by designing a cantilever with high torsional stiffness, also for the Z-direction a pure translation, without rotation of the wheel around the vertical X-axis could be realized, as shown in the following deformation plot (figure 9):

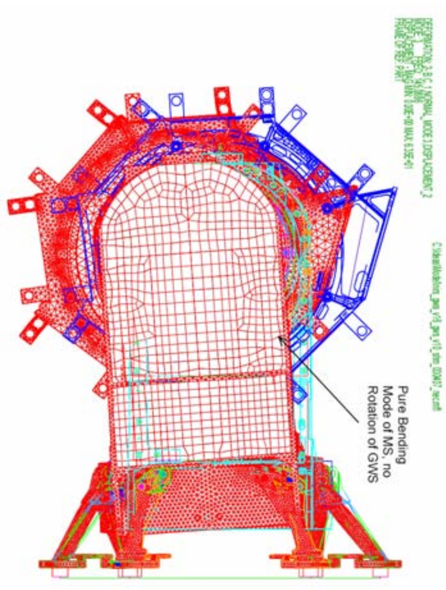


*Figure 9. Mode Shape Z-direction.*

Also for the X-direction which is the vertical direction a concept of counter-bending was implemented into the design by controlled weakening of the interfaces between to the rear side kinematic mounts to the optical bench and the main structure. A cut-out on the rear side of the main structure controls the counter-rotation. Figure 10 presents this cut-out, shown without the kinematic mounts:

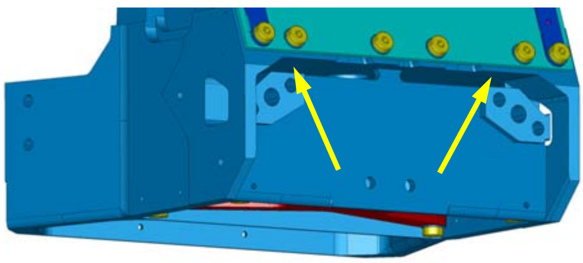

*Figure 10. Cut-Out at Kinematic Mount Interface.*

The following deformation plot (figure 11) shows the related deformation of the structure. Also in this direction the wheel experiences considerable translation, while the rotation of the wheel is minimized. The plot also shows that the translation is mainly in vertical direction, with low lateral component, so that not only the moment load on the bearing but also the axial load on the bearing is limited. (The slotted features on the main structure, close to the bearing interface are not related to the minimization of the bearing loads but are implemented to lower this Eigen-frequency for load reduction of the optical components of the wheel.)


*Figure 11. Mode Shape X-direction.*

An additional design constraint was to keep the rotation center of the rotational modes of the wheel at the bearing center, so that these rotational modes do not cause axial or radial loads on the bearing, as schematically shown in the following plot (figure 12):

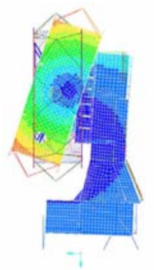

*Figure 12. Mode Shape of Wheel Rotation.*

First vibration tests on the structural thermal model of the mechanism showed very good correlation between the prediction by the used analysis model and the measured results, confirming the analysis technique.

With the described measures the loads on the bearing could be dramatically reduced during this design optimization: The bearing friction torque at ambient temperature could be reduced by more than 80%, from 210 Nmm to 40.5 Nmm predicted value.

## 5. TEST FACILITY FOR CRYOGENIC TESTS

The following test sequence is planned for the bearings: After ambient temperature run-in in vacuum a high precision friction torque measurement at ambient temperature at the bearing supplier ADR is planned. A friction toque measurement at cryogenic temperature will be performed by Austrian Aerospace GmbH. Then repeatability measurements will be conducted by Carl Zeiss Optronics GmbH.

For the cryogenic friction torque measurement at 30K a dedicated test-setup was developed. The test-setup is schematically shown in the following figure 13:

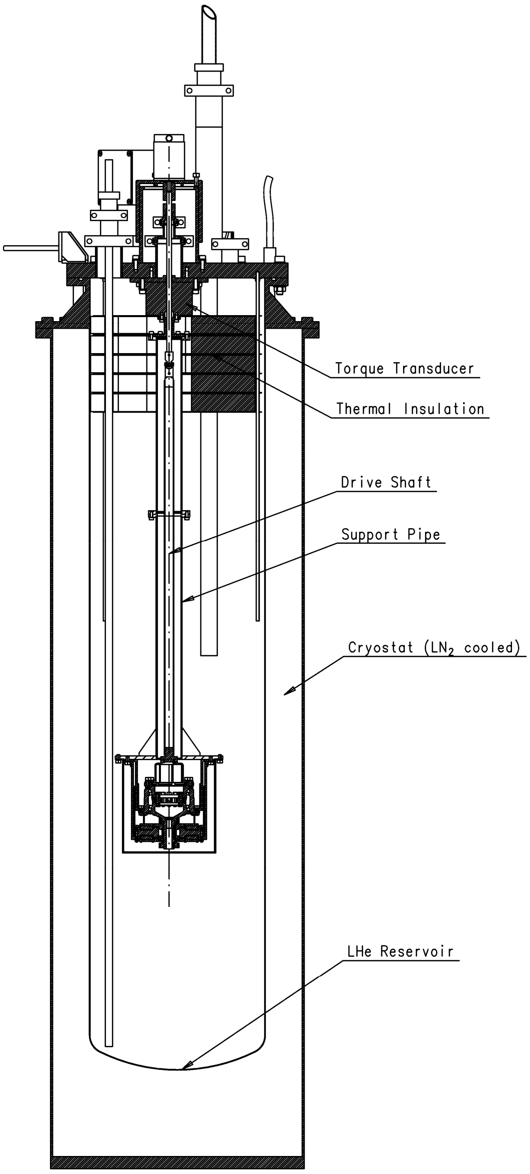


*Figure 13. Cryogenic Friction Torque Test Setup.*

The setup is placed in a cryostat. On the cover of the cryostat an auxiliary motor is placed, that rotates the drive shaft via a rotational feed-through. The drive shaft actuates the bearing that is mounted in a temperature controlled housing which is supported by the support pipe. On the upper end of the support pipe a torque transducer is placed so that the reaction torque of the bearing is measured. To separate the torque transducer from possible motor noise the torque transducer is mounted separately from the auxiliary motor. The bearing is mounted on isostatic mounts to minimize deformation of the bearing due to thermal stresses.

*Figure 14. Interiour Parts of Test-Setup.*

Cooling medium is liquid Helium, the test atmosphere is Helium gas. As an auxiliary motor a stepper motor is used to allow simple adjustment of angular velocity and rotation count. For thermometry Cernox type sensors by Lakeshore, US., are used. The torque transducer (figure 15) is based on deformation measurement of a roational spring body by a Linear Variable Differental Transformer (LVDT) principle to allow long term measurements.

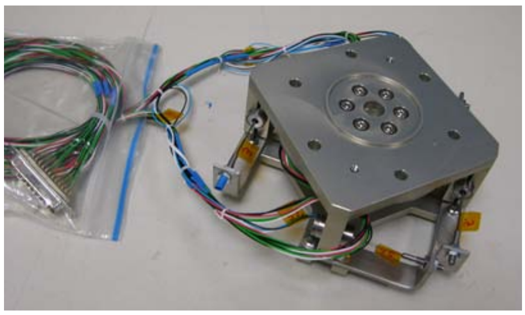

*Figure 15. Torque Transducer.*

## 6. STATUS OF BEARING DEVELOPMENT

The bearings were manufactured, coated and assembled.

*Figure 16. Assembled Bearing.*

The test campaign has just started in July 2007. The first test results will be presented during the symposium.

## 7. SUMMARY AND CONCLUSION

A highly accurate bearing was developed in the course of the development of the support mechanism for the wheels of the NIRSpec instrument. To minimize the bearing friction torque two design features were implemented:

- A novel thermal bearing off load device was implemented, which shall reduce the bearing friction torque at low temperature to 20% of its ambient value (see reference [1]).
- An optimized design of the mechanism which is optimized for minimum bearing loads by tuning of the mode shapes was established.

The resulting low friction torque of the bearing of less than 10 Nmm predicted value at operating temperature of 30K helps to minimize dissipated and conducted heat due to motor operation.

## 8. REFERENCES


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## 9. REMARK & ACKNOWLEDGEMENT


The support structure of the NIRSpec wheels, developed by Austrian Aerospace GmbH is part of the filter wheel assembly and the grating wheel assembly, both developed by Carl Zeiss Optronics GmbH. The NIRSpec instrument is developed by Astrium GmbH, Germany under ESA contract as ESA contribution to the James-Webb-Space-Telescope.

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