FILTER WHEEL ASSEMBLY: A LONG LIFE SPACE-MECHANISM WITH HARD MOUNTED BEARINGS

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ABSTRACT

For long lifetime space mechanisms, a soft-mounted bearing arrangement is commonly used. This arrangement allows maintaining a low preload within the bearings also in presence of thermal gradients and guarantees that the Hertz pressure between the balls and the races remains constant [1]. The bearing wear is thus minimised and the bearing lifetime performance is enhanced. The drawback of this standard configuration is the need for a launch lock to constrain the gapping within the bearings under vibration loads and avoid the generation of indents. In order to minimise the mass and system complexity, an alternative bearing arrangement was selected for the Filter Wheel Assembly (FWA). Contrarily to preceding projects, the bearings were hard-mounted in this application. This paper presents the considerations taken for the bearing design to guarantee that the mechanism is able to reach the expected lifetime requirements despite the rigid bearing mounting approach.

1. INTRODUCTION

1.1. Filter Wheel Assembly Design Overview

The multi-viewing, multi-channel, multi-polarisation imager (3MI) currently being developed for the Meteorological Operational Second Generation satellites will enhance the future Earth observation performance through its capability to capture pictures at different viewing angles, wavelengths and polarization directions. The Filter Wheel Assembly (FWA) is the core subsystem of the 3MI and enables this capability through a compact filter wheel disc carrying optical filters for the bandwidth and polarization axis selection.

Figure 1. Filter Wheel Assembly (FWA)

The disc is supported and driven by a high precision mechanism (Fig. 2) which guarantees accurate disc positioning and rotation throughout the entire mechanism life-time of 57 Mio. revolutions. The bearing arrangement guiding the shaft is thus a key mechanism design feature, especially due to the absence of a launch lock which would protect the bearings from high loads during launch.

1.2. FWA Bearing Arrangement

A novel bearing mounting approach was used for this long life space mechanism: The FWA utilises two bearing pairs mounted in a x-configuration which combines the advantages of hard-mounted bearings that can sustain high launch loads and soft-mounted bearings that increase the mechanism's robustness with respect to temperature variations and thermal gradients.

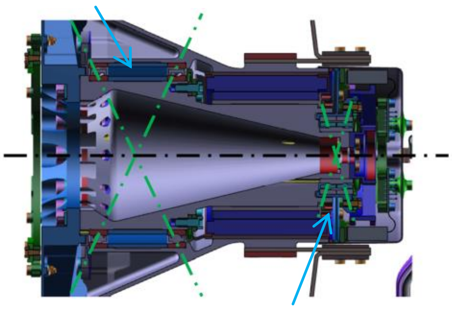


Figure 2. Bearing Arrangement

The bearings are placed in a double x-configuration which guarantees that the rotation axis is determined by two rotation points only and that these rotation points are not affected by temperature changes or thermal gradients. Indeed, the load generated by the membrane ensures that at least one of the front bearings and one of the rear bearings remains preloaded, even if thermal gradients within the bearings causes the loss of preload within the bearing pair itself as illustrated in Fig. 3.

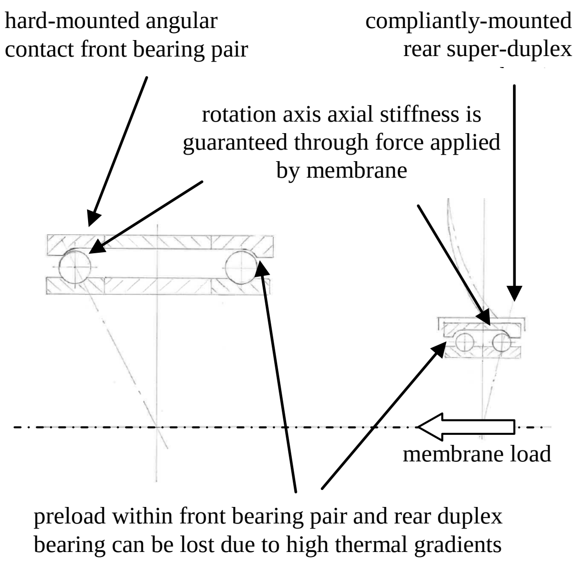


Figure 3. Bearing Loading in Presence of High Thermal Gradients

In order to optimise the bearing performance for the FWA application, careful consideration of the ball bearing preload, race design, cage design and lubricant selection was required. For a bearing the capability to sustain high launch loads is contradictory to operation over an extended lifetime. Thus, detailed analyses and extensive testing was necessary to find the most appropriate bearing configuration. The FWA mechanism bearings design was based upon:

- An in-depth review of lubricant heritage for high lifetime space mechanisms
- A bearing geometry optimisation to enhance lifetime, but also avoid race/land-override under the launch loads
- Optimisation of the cage design to eliminate instability during operation and excessive wear
- Optimisation of the bearing mounting approach to control preload in the hard-mounted bearing arrangement

Each of those aspects is discussed in detail in chapter 2.

2. BEARING DESIGN OPTIMISATION FOR LONG LIFETIME

2.1. Lubricant Heritage

Lubricant candidates suitable for bearings in long-life space application include lead, sputtered MoS_2 combined with PGM-HT cage and Nye2001a.

Lead has substantial heritage for space applications and is often the preferred lubricant due to its ability to withstand greater ground operation in air than MoS_2 films [2]. Lifetimes of 10^9 revs have been achieved with lead in vacuum and accelerated testing can be carried out with lead lubricated bearings with factors of 10 or more. On ground, the life of lead-lubricated bearings tends to be limited by cage wear and debris generation in the 1g environment. In-flight cage wear and therefore debris generation is not normally a limiting factor. Whilst lead is prone to oxidation and an attendant change in frictional behaviour, in contrast to the performance of MoS_2, the tribological performance of lead at bearing level is rather less sensitive to its storage and operational environments. In general, lead-lubricated bearings can be operated in air for up to 100’000 revs at slow speed (< 100 rpm) without permanent degradation and will recover on subsequent operation in vacuum.

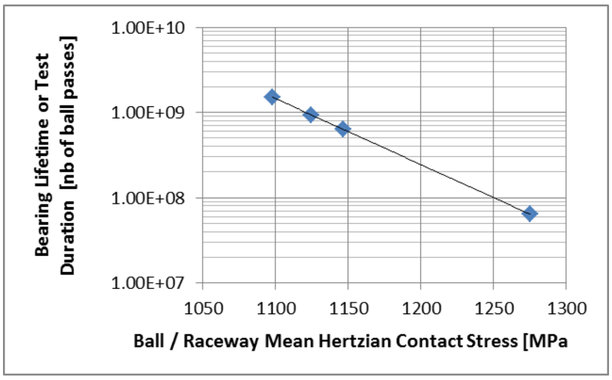


Figure 4. Lifetime vs. Mean Hertz Stress for Lead lubricated Bearings with Bronze Cage [6]

The second long-life space lubrication for bearings includes **MoS_2 coated balls and raceways plus PGM-HT cages**. This combination has been widely used and both heritage and test data are available [3]. In addition, commercial availability and lead times are good and the only drawback is the requirement to perform ground testing in a nitrogen purge or vacuum as air operation degrades the lubricant film. Indeed, in air MoS_2 can react chemically with moisture and oxygen resulting in the formation of molybdenum trioxide. Two reactions are possible:

$$2MoS_2 + 4H_2O + O_2 = 2MoO_3 + 4H_2S \quad (1)$$

$$2MoS_2 + 4H_2O + 9O_2 = 2MoO_3 + 4H_2SO_4 \quad (2)$$

MoO_3 is a poor lubricant, however the tendency is for limited oxide growth - limited to thicknesses of about 100 Angstroms. This limited thickness arises because the surface oxide forms a passivation layer whose presence acts as a barrier to the further oxidation of the underlying MoS_2. This thin oxide layer yields a higher friction than un-oxidised MoS_2 such that when components so-lubricated are operated in vacuum after air exposure the torque is initially higher. However, the thin oxide is quickly removed by wear in vacuum and low-friction is restored. The use of dry nitrogen gas to purge the bearing during ground testing is nonetheless highly recommended.

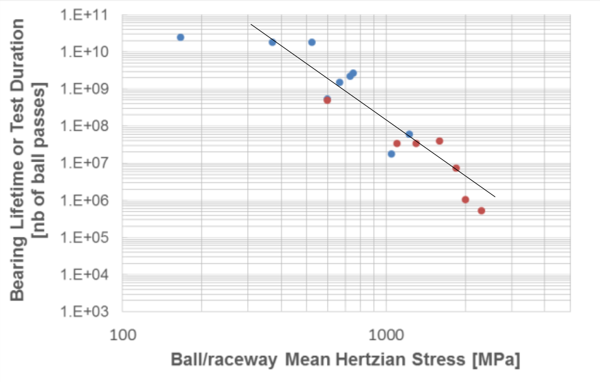


Figure 5. Lifetime vs. peak Hertz Stresses for Bearings with self-lubricating cages (orange dots: MoS_2)

Fluid lubricants are widely used in space, particularly PFPE-based oils and greases such as Fomblin Z25 and Braycote 601EF, and synthetic hydrocarbons such as Nye 2001a. An important limitation of PFPE fluids is their tendency to degrade when used on bearing steels under boundary lubrication conditions. The mechanism which leads to the degradation and eventual failure is as follows: Fluorine is released from the oil by the action of repeated shearing under high contact stresses and reacts with iron in the steel to form FeF_3. Despite being chemically inert to most materials, PFPEs suffer catalytic attack by FeF_3 (and other metal fluorides) which in turn leads to the breakdown of further molecules and the release of yet more fluorine: a chain reaction ensues and oil degradation is rapid. In ball bearings this results in higher and more erratic torque and enhanced wear. One of the very few space compatible fluid lubricants not based on PFPE and thus not prone to the above described chemical degradation and suitable for long lifetime applications is **Nye2001a**. Its viscosity increase at low operating temperatures resulting in higher ball bearing torques, the need to include labyrinth seals to minimize evaporation losses and it's limitation in performing accelerated testing for applications with nominal operation in the boundary regime represent significant drawbacks for this kind of lubrication. The use of Nye2001a was therefore not considered further for the FWA.

2.2. Bearing Race Design

The bearing races design drivers are the axial load applied on the bearings and the targeted cage clearances. In addition to the Hertz pressure applied on the race itself, careful consideration has to be taken to avoid that race override can occur due to the applied axial loads. Indeed, if the axial load is too high, the contact ellipse between the ball and the raceway has no longer the form of an ellipse, but is truncated as shown in Fig. 6. The risk of damaging the balls being higher in this situation which has therefore to be avoided [1].

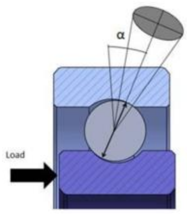
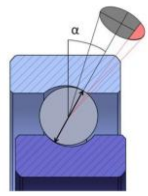
Nominal contact ellipse	Truncated contact ellipse
	

Figure 6. Bearing Axial Load Limit

2.3. Cage Design

In order to optimise the bearing performances, careful consideration of the ball bearing cage design is also required. This is essential to avoid instability during nominal operation and during the accelerated testing required for qualification. Such cage instability could result in excessive wear and high or erratic torques. The FWA bearing cages were optimised for the selected ball bearing and lubricant in terms of dimensions, mass, materials and pockets. Additional factors which were considered comprise cage roundness achievable, cage to land clearances, ball pocket clearances and the effects of thermal expansion and contraction.

For lead lubrication, leaded bronze cages are used. However, leaded bronze is ductile and therefore not an ideal material for snapover (or crown) cages of the type used in deep groove ball bearings. During assembly, the cage ribs can deform permanently as the cage is pressed over the balls which can result in the cage being poorly retained. Wear during operation can also affect cage retention. One-piece full ball pocket cages were therefore used for the FWA application and include the following design particularities:

- The cage and ball bearing lands and ball pockets clearances are optimized based on the experience from ESTL, and considering thermal expansion and minimizing the cage mass. The inner cage clearance is thus 0.2 mm, while the outer cage clearance is at least 0.8 mm [5]
- Alternating holes and slots have been implemented in the larger bearing cages to allow the balls more freedom of movement to reduce the effects of ball crowding and ball to cage forces, thus minimising cage hang-up

For the **MoS_2 lubricated** bearings, consideration was given to ensuring there is as much material present as possible to prevent premature cage wear-out. Also, the clearances were optimised to ensure that stabilisation was provided by guidance at the outer ring at higher temperatures and the inner ring at the lower temperatures due to the difference in coefficients of thermal expansion of steel and PGM-HT. The effects of the thermal environment upon clearances are presented in Fig. 7.

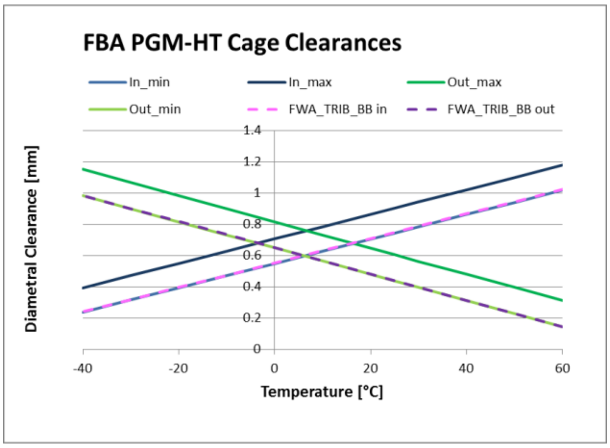


Figure 7. Cage Clearance vs. Temperature

2.4. Bearing Mounting Approach

As described in section 1.2, the FWA bearings pairs were mounted in a double X configuration. While the rear bearing was designed as a super-duplex bearing, high precision spacers were used to position the two front bearings at the correct distance w.r.t each other. The stiffness of those spacers and their manufacturing tolerance were chosen such as to minimize the impact on the bearing preload and the bearing alignment. Indeed, a height difference of 1µm between the inner and outer spacers would lead to a bearing preload change of 42N. These considerations highlighted the need to also accurately control the bearing clamping force in order to avoid uneven loads applied on the inner and outer bearing spacer which would have a direct impact on the bearing preload.

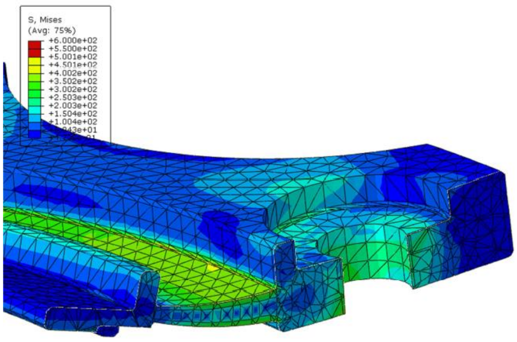


Figure 8. Bearing Clamp Ring Design and Analysis

A special bearing clamping system (Fig. 8) was designed in order to improve the clamping load accuracy from 50% with classical bolt clamping to 10% through the use of the dedicated 'spring clamp rings' (Fig. 9 and Tab. 1). The load generated by the clamp rings is controlled by the selection of suitable shims which define how much the clamp ring is deformed after it has been bolted to its counterpart.

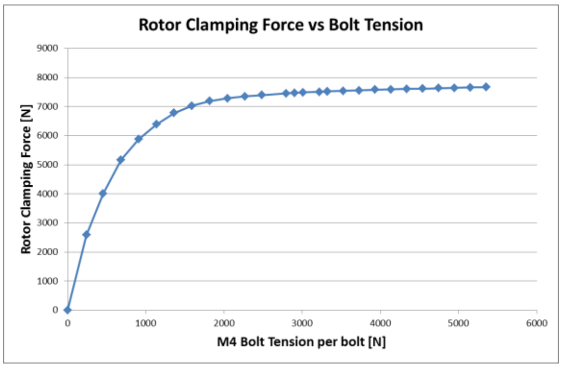


Figure 9. Clamp Ring Load vs. Bolt Pretension

Table 1. Bearing Clamping Accuracy

element	Load Variation with Clamp Ring	Bolt Load Variation
Min	7070 N	12 x 2270 N
Nom	7512 N	12 x 3323 N
Max	7882 N	12 x 5356 N
Max load variation	812 N	3086 N

The stiffness of the clamp ring/structure interface is an order of magnitude higher than the bearing stiffness, which guarantees a sound load path for vibration load despite the presence of a 'soft clamp ring'. The repeatability of the clamping force and the stiffness ratios were assessed with non-linear contact analysis (Fig. 10) and verified during the vibration test campaign.

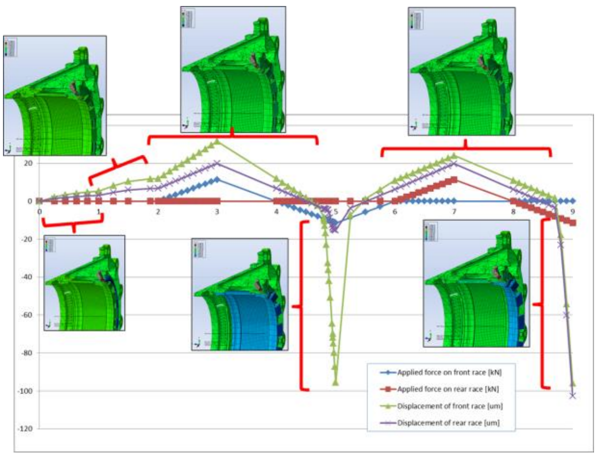


Figure 10. External Force & Displacement on Rotor Load Path

The clamp ring deformation measurement which is linearly proportional to the load applied by the clamp ring on the bearing spacers (Fig. 11) allows furthermore verifying the clamping load after integration and avoiding thus any unintended preload variation.

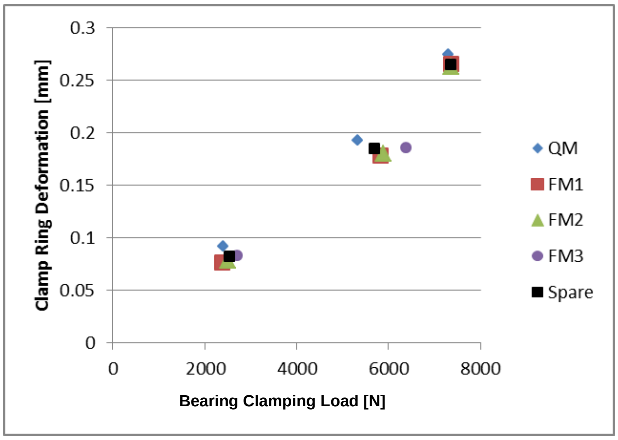


Figure 11. Clamp Ring Load vs. Deformation

With a moderate shimming accuracy, it is thus possible to reach an acceptable preload accuracy. For the FWA, the preload variation induced by the manufacturing and integration tolerances can be up to 69N and is mostly driven by the bearing races and spacer manufacturing tolerances (Tab. 2).

Table 2. FWA Elements affecting the Front Bearing Preload

element	design uncertainty	resulting preload variation
Bearing preload gap	±0.5µm	± 21 N
Spacer height	±0.5µm	± 21 N
Outer clamping load accuracy	± 1000 N	± 18 N
Inner clamping load accuracy	± 500 N	± 9 N

3. BEARING LIFETIME PERFORMANCE VERIFICATION

3.1. Lead vs. MoS_2 Lubrication

As both lead lubrication + bronze cage and MoS_2 + PGM-HT cage were initially considered good lubricant/cage options for the FWA bearings, breadboard tests have been carried out in an early project phase to validate their performance within the FWA bearing races. Two thin-section bearings with a large diameter, one coated with lead and one coated with MoS_2, were submitted to a partial life test as programmatic constraints did not allow performing a full life test. The bearing characteristics are given in Tab. 3.

Table 3. Bearing Characteristics for Breadboard Test

	Bearing 1	Bearing 2
Type	Two single angular contact bearings, face-to-face	
Outer diameter	100 mm	
Inner diameter	80 mm	
Thickness	2 x 10 mm	
Angular contact	25 degrees	
Preload	300 N	
Lubricant	Lead	MoS_2
Cage	Bronze	PGM-HT

Both bearings were subjected to vibration tests and a thermal vacuum cycling prior to their operation at accelerated speed over an extended time. The bearings completed 22 Mio revs at speeds up to 435rpm without failure prior to disassembly for inspection.

The friction torque of those bearings was found to be rather sensitive to the acceleration sequence and thermal environment. Indeed, slight variation of the friction torque would generate a thermal gradient in the bearing which would temporarily further amplify the friction torque as illustrated in Fig. 12.

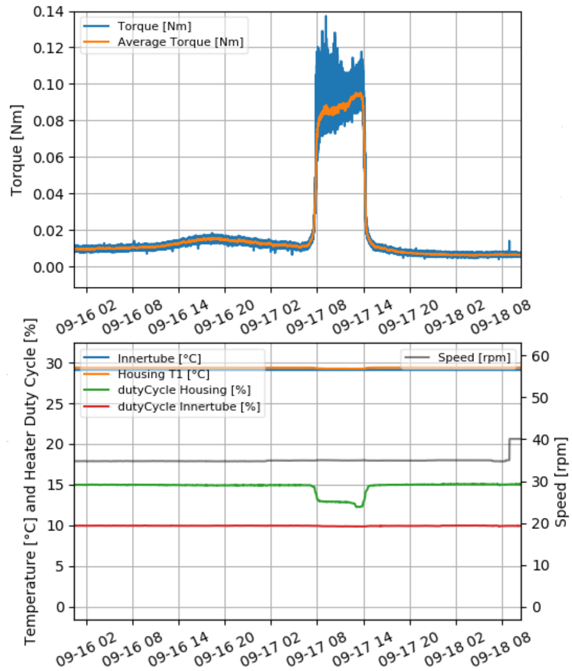


Figure 12. Friction Torque Increase due to Rapid Acceleration

The hard-mounted bearings have however some self-regulating capability to counteract this effect and avoid significant preload variation over an extended duration. Indeed, in the presence of a positive thermal gradient (i.e. inner race temperature > outer race temperature), the preload within the bearing is temporarily increased which leads to an increase of the friction torque that heat’s up the bearing outer race more than the inner race, reduces the thermal gradient within the bearing and thus their preload. The bearings would thus recovery to a nominal friction torque within a few hours.

The inspection of the bearings done at the end of the breadboard tests showed some wear within the cage pockets and on the cage riding side (i.e. inner diameter for bronze cage, outer diameter for PGM-HT cage) as illustrated in Fig. 13. This wear generated a significant amount of debris for the lead lubricated bearing which might be critical for applications with optics in proximity. The MoS_2 lubricated bearing generated considerably less debris which reduces the risk of contamination of other parts.

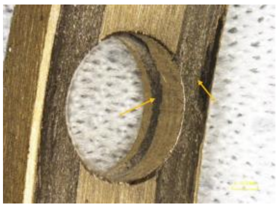
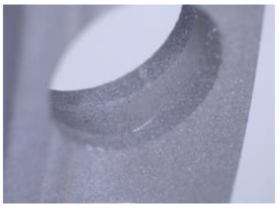
Bearing 1	Bearing 2
	
	

Figure 13. Cage Wear and Debris after 22 Mio Revs

The success criteria for bearing life tests consists usually to define the bearing failure as the wearing through of two-thirds of the widths of the cage ball-pockets separators. After 22 Mio revs, the cage pocket separators were worn by less than 13%. An extrapolation of their wear shows that the bearings could potentially reach a lifetime above 100 Mio revs.

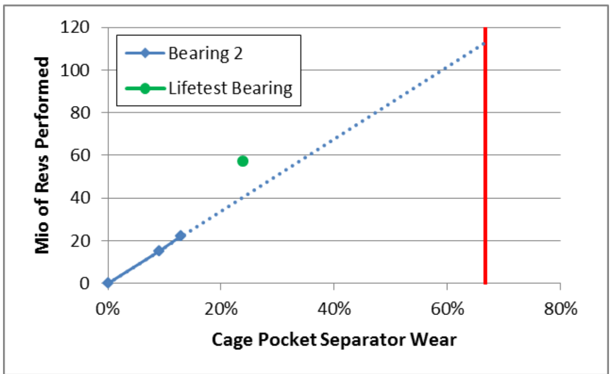


Figure 14. Bearing Lifetime Extrapolation

3.2. Filter Wheel Assembly Life Test

As final step for the bearing selection, a full life test was performed with bearings integrated into the FWA life test model. This model is identical to the flight model, with exception of the disc that does not include any optics, but consists of a mass and inertia dummy.

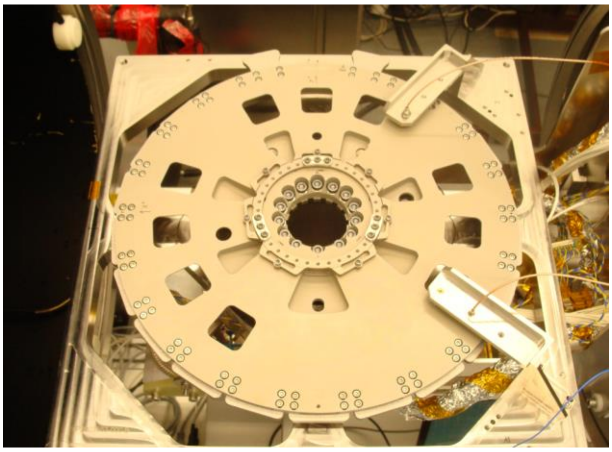


Figure 1. Filter Wheel Assembly Life Test Model

The bearings selected for the formal life test were MoS_2 lubricated and fitted with PGM-HT cages due to their advantage of lower debris generation. After completion of the FWA assembly, the mechanism was submitted to an environmental load campaign, including vibration, shock and thermal cycling in vacuum. During the life test, the mechanism was operated at an accelerated speed of 210 rpm. Higher speeds could not be reached due to limitations of the driving electronics. The bearing friction torque, the mechanism motorisation margin and positioning accuracy performance were measured in regular intervals throughout the whole life test. The results presented in Tab. 3 show that the mechanism's performance remained very stable throughout the entire life test and that there was no significant bearing degradation, which was confirmed by the absence of any friction torque increase or motorisation margin decrease.

Table 4: FWA Performance throughout Life Test

performance	**nominal value**	**variation observed during lifetest**
average friction torque	40 mNm	37 – 43 mNm
motorisation margin	2	3.95 – 4.18
positioning accuracy	0.12°	0.04 - 0.06°
Wobbling	0.003°	< 0.001°

A cage pocket separator wear of 24% was measured on the bearings after they had completed 57 Mio revs, which corresponds to even less wear than extrapolated from the initial breadboard measurements shown in Fig. 14 and confirms that the bearings can be operated over a further extended lifetime.

4. CONCLUSION

The experience gained throughout the FWA development shows that a hard-mounted bearing configuration can be successfully operated over an extended lifetime. As this configuration is however more sensitive to manufacturing tolerances and thermal gradients, the design features affecting the bearing preload have to be considered in a more detailed manner than within a soft-mounted bearing configuration. With a suitable mounting and assembly approach, this sensitivity can be reduced and the hard-mounted bearing configuration becomes an interesting design solution for applications, where the use of a launch lock is discarded due to complexity and mass constraints.

5. REFERENCES

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