DESIGN, DEVELOPMENT AND QUALIFICATION OF THE FILTER WHEEL ASSEMBLY (FWA) OF THE ASPIICS INSTRUMENT FOR ESA PROBA-3 MISSION

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ABSTRACT

This paper describes the mechanism design, development and qualification process of Filter Wheel Assembly for PROBA-3. Starting from the presentation of the driving requirements, the lessons learnt from the early design process which resulted in some innovative redesign to protect the stepper motor and sensitive filters. The paper presents the final design solution implemented (including hard and soft preloaded bearings, a dry lubricated sliding I/F and a novel motor coupling) and the performance achieved. The development approach has included detailed analyses and testing of qualification models, both completed successfully. Selected designers' insights are presented to provide guidelines and recommendations about problematic areas of similar mechanisms. The paper focuses on the challenges faced during the mechanism development e.g. stiffness, sliding prevention and motorization margin evaluation with a step profile acceleration without force/torque measurements. It is important to note that the FWA device passed complete qualification and its flight model is already delivered for integration with the Instrument.

INTRODUCTION

Filter Wheel Assembly (FWA) is a critical mechanism which is part of a Coronagraph instrument called ASPIICS for the PROBA-3 mission. PROBA-3 is an ESA mission devoted to in-orbit demonstration of novel formation flying techniques, with two satellites precisely positioned at a relative distance of 150 m. In particular, PROBA-3 will carry a scientific payload (ASPIICS, integrated under CSL Liege), in which the FWA is responsible for quickly and precisely positioning six optical filters in the optical beam of the detector. The device is developed in the Space Research Centre of Polish Academy of Sciences (CBK PAN), at the Laboratory of Mechatronics and Satellite Robotics (LMRS) to be exact.

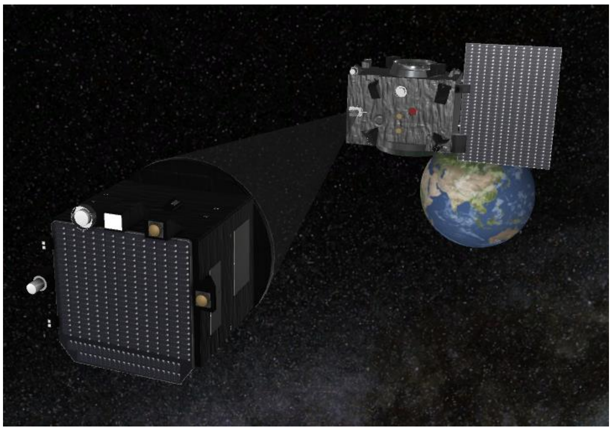

Figure 1 Render of PROBA-3 Mission spacecrafts on orbit.

The CBK designed Filter Wheel Assembly is a relatively small
- mass: 0,908kg
- envelope: 120x130x110mm

mechanism of critical functionality for the ASPIICS Instrument.

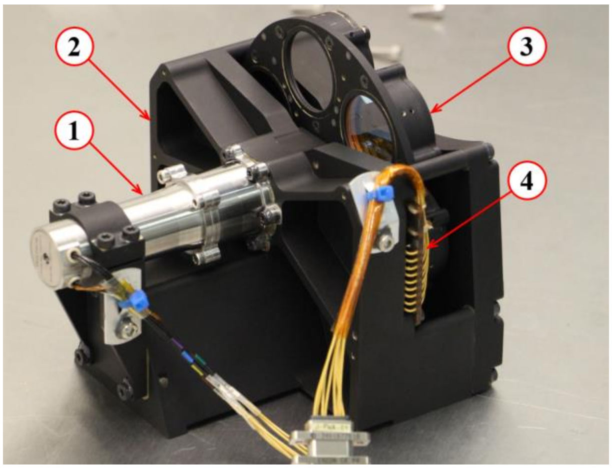

Figure 2 FWA FM during integration. 1-Stepper Gearmotor, 2-Primary Structure, 3-Filter Wheel with Filters, 4-Position Sensor Electronics.

In the next chapters we introduce first the main driving requirements of the design, then we go through the design description with particular focus on bearing system and motor gear-wheel coupling.

REQUIREMENTS

FWA was to meet, among others, the following set of requirements:

- Storing of 6 filters, clear aperture ∅27 mm, ∅28 diameter, 5mm thickness
- Mass <1.2 kg
- FWA first significant vibration mode ≥400Hz when hard mounted to an infinitely rigid interface.
- Random loads QL (X, Y, Z): 20, 27, 15 GRMS total.
- Sine load QL: 30g, 5-100Hz
- Life time: 90 000 cycles

Most challenging of these requirements were those regarding the resistance to vibrations environment, as gearmotors are typically rated for 30GRMS, which in our case is equal to level of excitation for the whole FWA unit.

DESIGN DESCRIPTION

FWA is designed in co-axial layout, with filters equally spaced on plane perpendicular to rotation axis (Fig.3). Filter wheel is driven by Phytron PhySpace 19-3 stepper motor with 7:1 reduction VGPL22 planetary gear. Torque is transferred from gearmotor shaft (1) to coupling(3) by interference fit on feather key, and then by pin(7) to the wheel shaft(5). Rotating disc structure is mounted to wheel shaft by 6 M3 screws and positioned by fitted diameter. Wheel shaft is rotating on back-to-back duplex bearing pair(4) near the motor, and soft preloaded by wave spring(8) bearing(9) on the opposite side of the wheel. Axial displacement of shaft regarding duplex bearings is disabled by bearing nut (2). Bearing (9) is allowed to slide along wheel shaft. Labyrinth seals are introduced in form of stacks of precise washers(6).

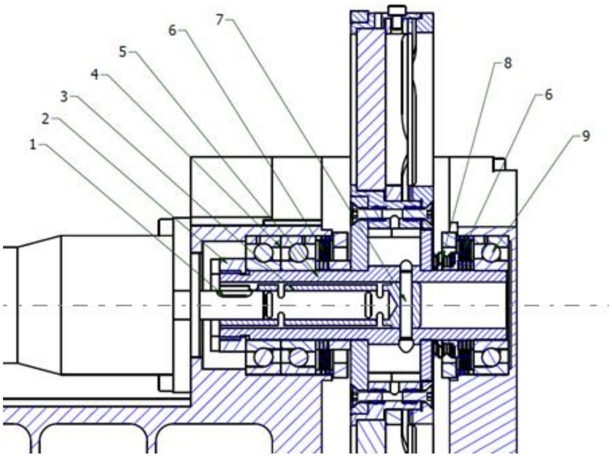

Figure 3 FWA Mechanism layout.

BEARING SYSTEM DESIGN

FEM calculations of preliminary model (Fig. 4) showed, that axial loads on the bearings (ca. 1kN 3σ then) were too high to use soft-preloaded system, as this would require preload force of same value, plus margins of safety. Simple switching to hard-preload was not an option, as it would not be possible to achieve enough precise distance between outer rings of bearing on the opposite sides of the wheel. Instead, two-way efforts were made:

1. Reduction of loads on bearings via structural changes.
2. Changes in the bearing system, so it could withstand higher loads.

Structural changes were focused on reducing weight of Filter Wheel (as it directly reduces loads on bearings) and increasing stiffness of both Filer Wheel and Primary Structure. Eventually, they led to more the twice reduction of bearing loads.

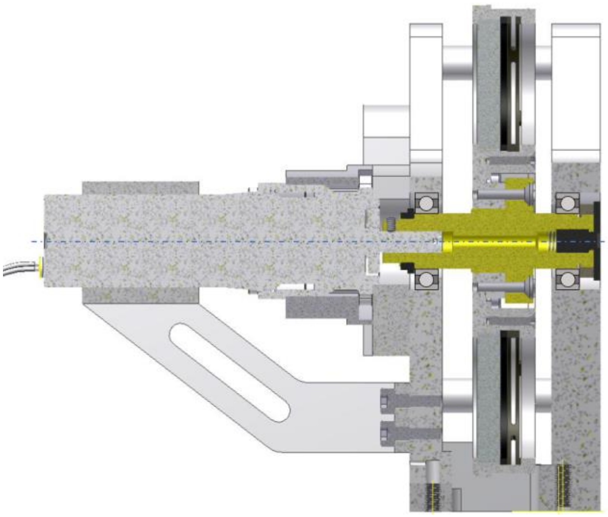

Figure 4 FWA Preliminary model layout.

Bearing system was changed, as bearing on the motor side was replaced with hard preloaded duplex pair of GRW SV7901A-2Z angular contact (25deg) bearings. This allowed for accommodation of axial forces much higher than preload value, as for hard preloaded set gapping occurs at 2.8 times preload force, and even then it can still be acceptable as resultant gap is usually rather small.

Bearing calculations

Important part of the FEM analysis was detailed modelling of the bearings. It was modelled in a way which allowed to extract forces in bearings which occur during dynamic loads, mainly random vibration.

Simplified models of the bearing were used for the analyses. Each bearing was modeled by the two RBE3 elements simulating outer and inner bearing raceway. Those RBEs were connected by the zero-length CBUSH elements by the dependent nodes (Figure 5). Each of the CBUSHes was given axial, radial and tilting stiffness and those values are dependent on the assumed preload. Hard preloaded Duplex bearings were simulated as a single bearing with the resultant stiffness.

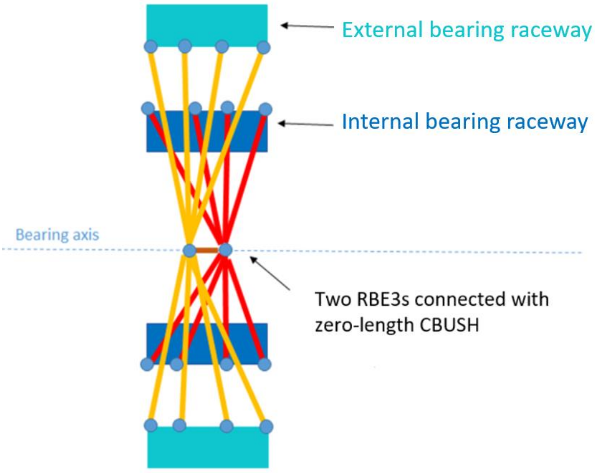

Figure 5 Bearings FEM schematics

Hard preload of 71N was assumed and GRW provided us with the estimated values of the bearings stiffness (Table 1)

Table 1 Bearing stiffness values from GRW

Input Data			
Bearing	A	A	B
Bearing Type left	2x SV7901A-2Z E (Duplex)	2x SV7901A-2Z E (Duplex)	SV7901A-2Z E
preload	71 N	71 N	35 N
F axial	-	300 N	-
F radial	-	-	-
Results			
Axial stiffness	59 N/µm	59 N/µm	28 N/µm
Radial Stiffness	118 N/µm	95 N/µm	45 N/µm
Tilting Stiffness	6,3 Nm/mrad	5,7 Nm/mrad	1,1 Nm/mrad
pmax	1194 MPa	1981 MPa	962 MPa

Soft preloaded bearing was modeled in the same way as shown in Figure 5, but independent nodes of the RBE3 connected to the shaft had no axial direction constraint. That way effect of sliding of the internal raceway along the shaft could be simulated. Additionally, soft preload spring was introduced into the model by the ring of solid elements with resultant stiffness of the spring (Figure 6).

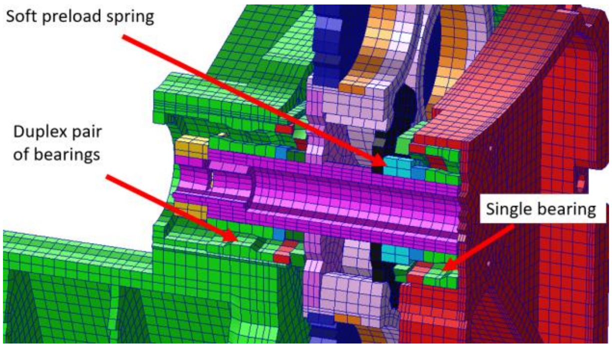

Figure 6 FE model of the FWA - filter wheel subassembly, shaft and bearings, soft preload spring.

With those values of the stiffness and model philosophy, whole set of the analyses was conducted - including random vibration. According to the analysis results, random vibration caused the highest loads in bearings taking into consideration 3-sigma values. Table 2 and Table 3 show forces obtained from the random analysis for the duplex pair and for the single bearing. Analysis assumed excitation in single axis at once but for more conservative approach it was assumed that the maximum radial load is the resultant of the Y and Z loads.

Table 2 Duplex pair reaction forces

Excitation / Force [N]	X (Axial)	Y	Z	Radial
X	449.048	61.297	46.715	77.069
Y	4.835	15.941	0.531	15.950
Z	11.1944	2.025	13.018	13.175

Table 3 Single bearing reaction forces

Excitation / Force [N]	X (Axial)	Y	Z	Radial
X	3.846	79.124	60.116	99.371
Y	0.215	19.987	0.722	20.000
Z	0.075	2.245	16.471	16.623

Forces in bearings allowed to determine gapping that may occur during the launch. The most critical load case in terms of the axial gapping is the one with the highest axial load – and that is X excitation. For that case 3-sigma axial force in the duplex pair reaches equals 449 N. Also, for that case the radial load is the highest for the duplex pair (77N).
To estimate value of the resultant gap plots were used (Figure 7, Figure 8, Figure 9). They were derived from the ESTL report [1] which is a result of the CBK minor consultancy request. Plots were obtained in the

CABARET software.

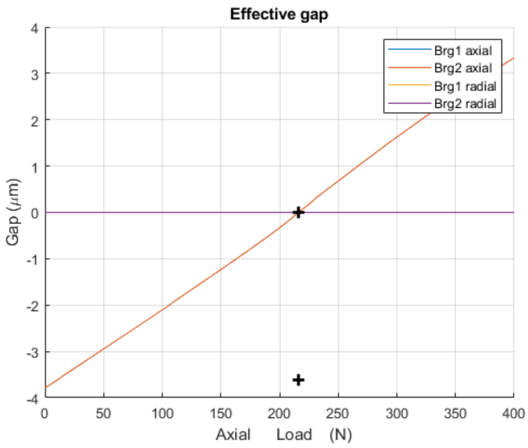

Figure 7 Gap dependence on axial load for duplex bearing – at ca. 215 N load ball lift-off occurs.

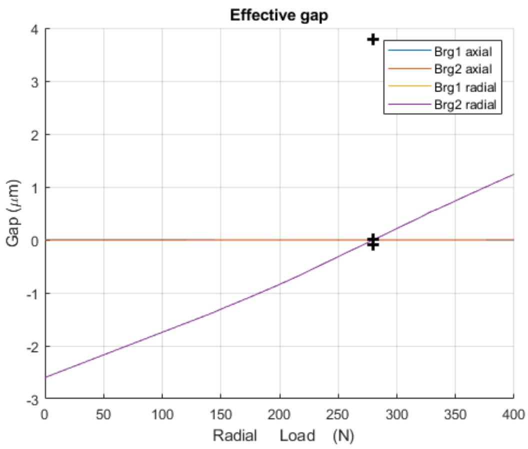

Figure 8 Gap dependence (for least loaded ball) on radial load for duplex bearing

As it is shown in those plots, there is almost linear dependence between load values and the effective gap. Therefore, from the Figure 7 it can be determined (by the linear extrapolation of the plot) that the gapping would reach value of 4um. Also, from the Figure 8 it can be derived that there will be no radial gapping on the duplex pair of the bearings (radial load is less than 280N).

For the single bearing, dependance between radial load in the bearing and the effective gap is not linear (Figure 9). Highest reaction force in the single bearing reaches almost 100N which gives us 5.5 um of the gapping.

None of the obtained gapping values exceeds 20um which is a maximum tolerable value set by the rule of thumb.

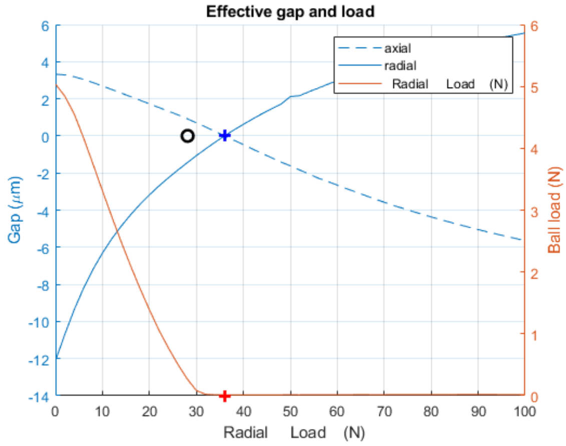

Figure 9 Soft preloaded (35N) bearing: Ball load, axial and radial gap in function of radial load. Plot for least loaded ball.

Also, bearings' raceways were analyzed in terms of the Hertzian stresses. For the applied bearing limit axial load for the max allowable Hertzian stress is 1320N (ESTL calculation). That gives us Margin of Safety which is equal 2.9.

Knowing the 1320N values as the limit axial load we have also calculated the fatigue life for bearings, assuming bearing loaded only with preload (which is 71N).

$$L_{10} = \left(\frac{C}{P}\right)^3 = \left(\frac{1320}{71}\right)^3 = 6\,426 \times 10^6$$

Reqirement [FWA-8101] states, that FWA must sustain 90 000 cycles. Assuming worst scenario (1 cycle = 1 rotation), this gives us 71 400 higher L_{10} life than required. Obviously, this means that FWA bearings will never reach fatigue failure, as most likely first to fail will be lubrication.

GEAR – WHEEL COUPLING

Gear output shaft could be directly connected to filter wheel shaft, if proper measures (quite feasible in this case) regarding coaxial mounting were taken. However, to perform FEM calculations of such system, stiffness matrix of gear shaft regarding gear housing is necessary, as it imply crucial information about distribution of loads between main bearing system and gear bearings, that have much lower load capacity. Unfortunately, manufacturer does not provide necessary values, and after failed attempts to achieve them, it was chosen to implement translationally compliant coupling.

Due extreme limitations in space, as implementation of coupling was not expected, no standard solution would fit and special one was necessary. Volume for coupling was found not in axial direction (distance along rotation axis), but in radial one – inside hollow shaft of filter wheel, so coupling took a form of long but small diameter

(7mm) tube with proper cutouts. There was no access to gear shaft in mounted position, so coupling could not use any tightening screw, and had to rely on pretension clamping on shaft key. As transmitted torques were to be relatively small, polymer (PEEK) was selected for coupling material.

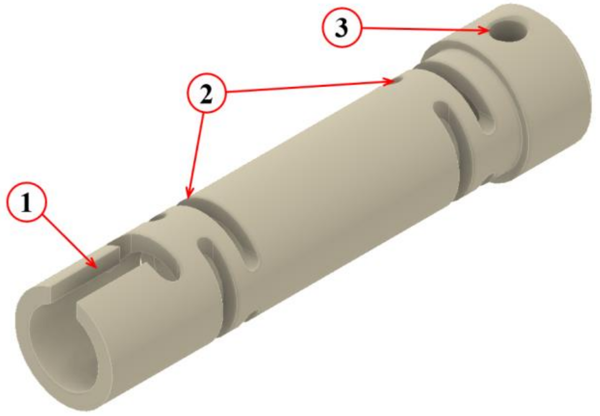

Figure 10 Coupling: 1-Jaws with tight fit on gear shaft, 2-Cuouts, stiff for torsion, compliant for bending, 3-Output pin hole.

TESTING

Throughout its test campaign, the FWA passed the complete qualification including:

- Vibration testing both at Unit and Instrument level.

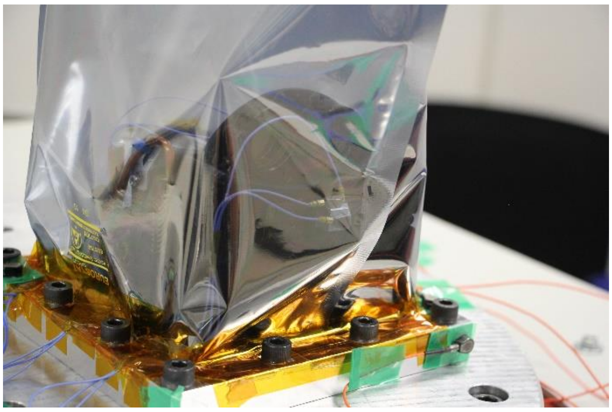
Figure 11 FWA FM Vibration Testing at AstroFein

- TVAC campaign, including both Thermal Cycling and Life Test of 12 days in total, during which the FWA performed 184000 full cycles. 1 cycle of operation consists of going through all the 6 filters, so it corresponds with a single revolution of the Wheel.


Figure 12 FWA QM inside CBK TVAC

- Bakeout before delivery to CSL for integration with Instrument

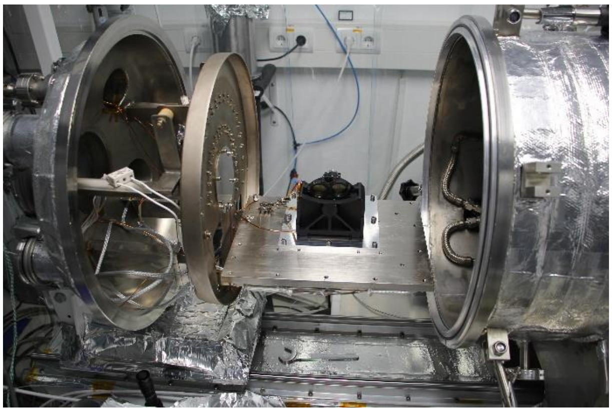
Figure 13 FWA FM in preparation for Bakeout at ESTEC

CONCLUSIONS

Filter Wheel Assembly (FWA) went through the complete cycle of development of the flight mechanism. Joint efforts of LMRS team supported by CSL and ESA experts led to the development of efficient mechanism that is passed the complete design review loop and successfully demonstrated its full compliance with requirements of the mission.

At the moment of writing of this paper, the Flight Model of the FWA is already delivered to CSL and ESA.

Figure 14 Part of the LMRS team involved in the FWA project presenting FWA Qualification Model.

REFERENCES

1. Vortselas A. (2019). *MEC-ESTL-TN-0070-CBK Minor Consultancy*, ESTL