ACTIVE BALANCING SYSTEM FOR THE COPERNICUS IMAGING MICROWAVE RADIOMETER (CIMR) INSTRUMENT: SYSTEM DESIGN AND BREADBOARDING RESULTS

Tharek Mohtar, Andrea Moroni, Alessandro Bursi, Andrea Galbiati, Isacco Pretto, Matteo Spinelli

OHB Italia, Via Gallarate 150, 20151 Milano, Italia, Email: tharek.mohtar@ohb-italia.it

ABSTRACT

The Copernicus Imaging Microwave Radiometer (CIMR) satellite will be equipped with a conically scanning microwave radiometer imager. The conical scan concept is based on a rotating antenna assembly (reflector and feed cluster) that continuously rotates around the satellite nadir axis scanning the Earth's surface. The instrument will provide observations of sea-surface temperature, sea-ice concentration, and sea-surface salinity.

The large dimensions of the instrument's deployable reflector combined with the deployed reflector stiffness make it impossible to balance the system, with the needed accuracy, during the on-ground integration phase. The resulting unbalance affects the performance and life of the instrument's rotating mechanism. Moreover, it reduces the pointing performance of the satellite.

In this regard, OHB is developing a dedicated balancing system to solve this issue. The Active Balancing System Assembly (ABSA) aims at balancing the CIMR instrument at discrete points in time (in particular after initial instrument spin-up) by using a customizable number of linear mechanisms called Mass Position Devices (MPD). Each device positions a tungsten alloy counter-mass using a ball screw/linear guide arrangement. A redundant Brushless DC gearmotor drives the ball screw transmission. The MPD is equipped with a novel Hold Down and Release Mechanism (HDRM) based on a High Output Paraffin Actuator (HOPA).

This article describes the design and development of the MPD. It also summarizes key design drivers (mass, loads, envelop) and relevant trade studies. Special attention is given to the description of the linear transmission and the HDRM.

Finally, the article presents the results of dedicated breadboarding testing activities. Elegant breadboards were built to demonstrate the suitability of the mechanism against the launch vibration loads. Moreover, it also shows the results of the reduced life test of the linear transmission. The test campaign revealed several key design improvements.

1 INTRODUCTION

The prime manufacturer of the CIMR satellite is TAS-I while OHB-I is responsible for the instrument. The CIMR instrument is a passive conical scanning total power radiometer operating at multiple frequency ranges from 1.4135 GHz to 36.5 GHz.

The conical scan concept is based on a rotating antenna assembly (reflector and feed cluster) that continuously rotates around the satellite nadir axis scanning the Earth surface, and the reflector pointing direction is selected to look at the Earth with the required Observation Zenith Angle (OZA). In particular, the rotor is composed of an irregular hexagonal crate mounted on ball bearings. The crate is equipped with a deployable boom. This boom carries an also deployable reflector.

While the satellite flies along its orbit (Along Track direction, in red in Fig. 1), the rotation of the antenna allows to cover the Earth surface with measurements along a circular sector (light blue in Fig. 1) with acquisition of contiguous pixels in the Across Track (or Along Scan) direction. At the following revolution the antenna looks at another circular sector, moved forward due to the translation of the satellite; the level of overlap of the new sector with the past one depends on the speed of the satellite and on the speed of the rotation of the antenna: a higher rotation speed will guarantee more overlap, while a low rotation speed might produce gaps between following acquisitions.

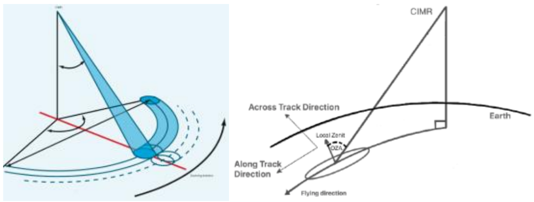

Figure 1. Conical scan principle

The instrument is composed of a rotating part (where antenna and feeds are located) and a fixed one connected to the satellite platform. Fig. 2 shows the CIMR architecture.

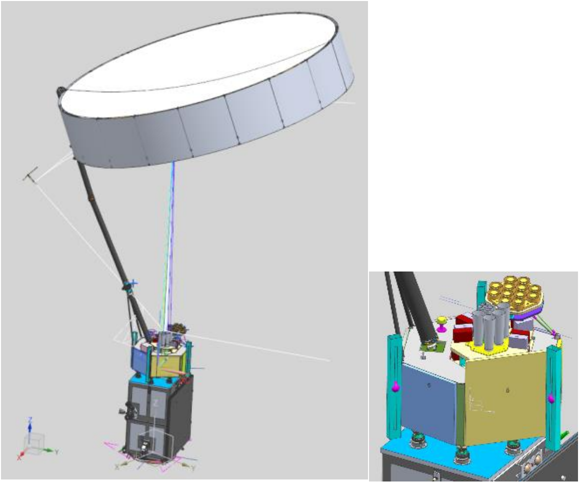

Figure 2. CIMR Satellite (left) and CIMR Instrument rotating part (right, reflector not shown)

The CIMR instrument is dominated by a 7.4 meters in diameter deployable mesh antenna that is mounted on a deployable boom. The length from the base of the boom to the top of the mesh antenna is 10.8 meters.

A rotor unbalancing could cause problem/degradation for pointing performance, operational life, scan motor functionality and spacecraft AOCS.

During the SMAP mission [1], an approach based on tests and simulations was used to perform the scanning radiometer rotor balancing on ground. However, due to the CIMR size of the rotating crate, reflector, and the boom, no full ground characterization can be performed in deployed configuration. Hence, the baseline following mission PDR is to correct the effects of the unbalances and the perturbations across the mission in flight through a dedicated balancing system operating in orbit.

A sensitivity analysis has been performed at instrument level to identify the balancing needs. The outcome of this analysis showed that the contribution of the static unbalancing is negligible, while the dynamic unbalancing shall be controlled.

To fulfil this principle, the ABS has been designed with masses that can move along linear guides with a compensation capability depending on the mass of the moving counter-masses, on the position of the linear guide with respect to the CoG of the Instrument rotation axis and on the length of the linear guide (i.e. stroke that the mass can run).

A complex trade off taking into account all the contributions both at instrument and ABSA level (balancing compensation range, max counter-mass value allowed for structural reasons, maximum linear guide number for S/C mass saving, allowed linear guides position and CoG vertical position on the crate, relative ABSA control electronic complexity etc) led to a system configuration based on 3 linear guides accommodated on 3 vertexes of the instrument crate.

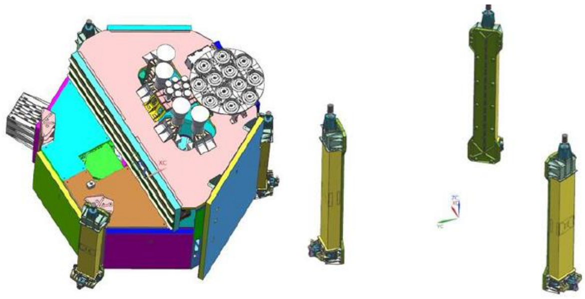

Figure 3. MPD configuration at Instrument level

2 DRIVING REQUIREMENTS

The ABSA MPD consists of an assembly grouping two main functions, mainly separated one from the other: CW movement along the linear guide by means of gearmotor actuator and the HDRM holding the CW during launch.

According to this idea, the driver requirements are:

- Balancing range that fixes the length and the mass value (being the distance from the rotation axis fixed by crate design. Thus, the MPD are placed on the perimeter of the crate to be as far as possible from the rotation axis.
- Time needed to move the counter-masses in the required position for balancing.
- Stringent mass (excluding counter-mass) and volume.
- Inertial load compatibility: at launch the mass shall be fixed via an HDRM to withstand the inertial loads.
- Accessibility requirement. The MPD shall be installed externally to the rotating crate to allow a late modification/installation of the subsystem in case of counter-mass weight last stage tuning due to instrument development outcome. In addition, the accessibility of the HDRM area for refurbishment is limited and complex mainly at S/C level.
- High TRL: the design shall be heavily driven by the heritage to minimize the development risks. For this reason, all base components were selected with at least TRL6.

In addition, ABSA is being designed with high flexibility, aiming at minimizing the fixed mass. Moreover, the possible to be able to adapt the moving masses based on Instrument-tested unbalances is of utmost importance.

3 MECHANISM DESIGN DESCRIPTION

The MPD (Fig. 4) is composed of two main subsystems. The first subsystem is the linear transmission. The function of this subsystem is the positioning of the counter-mass along the mechanism stroke. The stroke is equal to 868 mm while the maximum moving mass is 9 kg. Given the dimensions of the crate, the maximum balancing range of each MPD is 5.8 $kg \cdot m^2$. The second subsystem is the HDRM. This assembly protects the mechanism against the launch loads by holding the counter-mass in position. The same system has also the objective of releasing the mass once in-orbit.

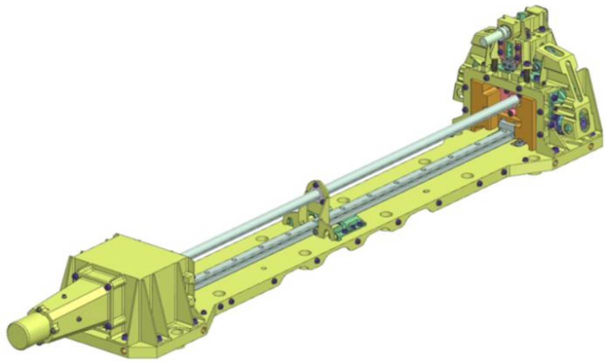

Figure 4. MPD. The sheet metal cover that protects the mechanism is hidden in the figure to show the internal components.

3.1 Linear transmission

An aluminium base support is equipped with a linear rail on which a tungsten alloy counter-mass slides. The linear rail also prevents the mass from rotating. The mass is moved by a ball screw transmission. The ball screw is connected to a normally closed brake to maintain the counter-mass position in orbit. During operation, the ball screw is rotated by a BLDC gearmotor through a gear transmission. The linear position of the counter-mass is measured by redundant HE sensors combined with redundant mechanical limit switches. Fig. 5 shows the baseline layout for the MPD.

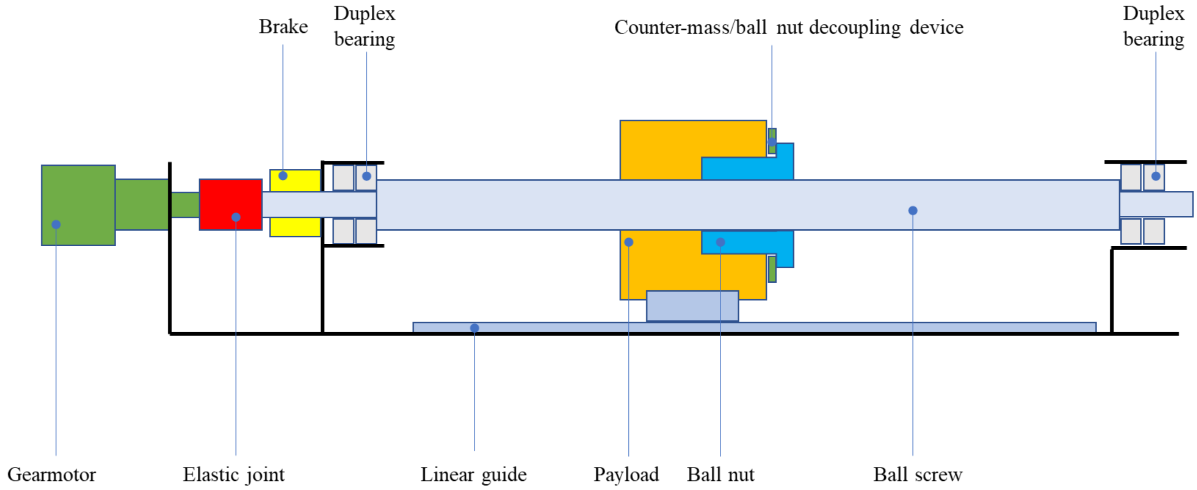

Figure 5. MPD schematic layout

The ball screw is mounted on a fixed duplex bearing on one side and a floating duplex bearing on the other. The floating duplex bearing allow the screw to expand, and contract based on the thermal load. Both bearings are mounted in back-to-back configuration. They are hard preloaded using locking nuts fastened to the ball screw ends.

The radial load on the ball screw shall be minimized to avoid bending. The structural analyses shown radial forces applied on the ball nut due to the deformation of the MPD structure. For this reason, the ball nut has been decoupled from the counter-mass. Thus, the connection between the ball nut and the counter-mass is accomplished with an elastic coupling. The elastic coupling has a large torsional stiffness about the rotation ball screw axis and relatively low stiffness values along and about all the other axes.

The selected motor is a Maxon dual winding BLDC

EC32 flat combined with a GP22UP planetary speed reducer.

Regarding the brake, it is necessary to keep the counter-mass in position after the balancing operation and to avoid unwanted movements during on-ground testing. The brake should only work in static conditions. Only the motor applies torque to stop the counter-mass. The selected type of component is a normally closed (power off brake) electromagnetic brake. When power is applied, a solenoid attracts an armature toothed disc that is normally in contact with a fixed toothed disk (Hirth coupling). By separating the armature from the disk, the shaft connected to the brake is free to rotate.

The ball screw and linear guides are made of stainless steel lubricated by Braycote 601. Both components are preloaded with oversized balls.

3.2 HDRM

The HDRM is based on a HOPA coupled with a high stiffness mechanical transmission (refer to Fig. 6). The HOPA has a long flight heritage conferring it the maximum TRL.

Regarding the mechanical transmission, the counter-mass is blocked by means of two identical coupling surfaces at both sides of the counter-mass and by the slider/rail on CW bottom part. The slider is going to withstand most of the loads. The contact with the counter-mass is provided by a hard stop (at the aluminium bracket interface) combined with a second interface (at the proper counter-mass surface) and an adjustable preloading interface.

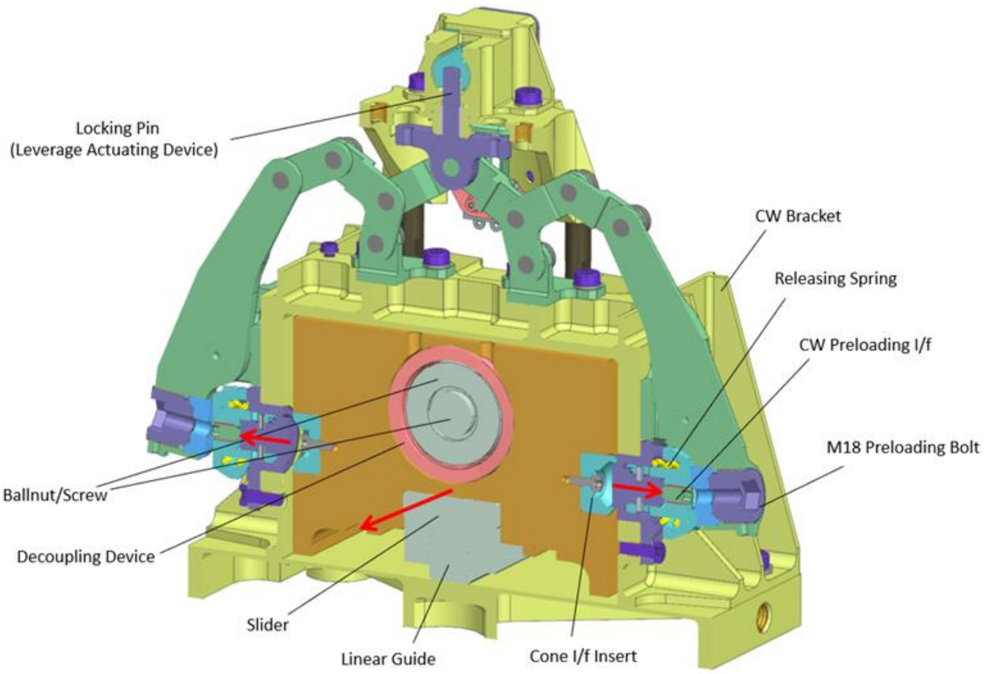

Figure 6. HDRM design

The main purpose of these interfaces is to prevent the counter-mass from moving along the direction of the ball screw and to keep the orthogonal forces and torques applied to the linear rail in the allowable ranges (e.g.: this is the reason for the symmetric design). On the other hand, the slider and linear rail helps the HDRM in constraining the CW subjected to loads orthogonal to the axis of the ball screw.

In addition, the HDRM also relies on the elastic coupling mounted at the counter-mass interface with the ball nut, such that no load (much lower than what the ball screw can withstand) is applied from the mass to the ball screw during the launch.

The release mechanism is mounted on top of the leverage, upon a bracket thermally decoupled from the MPD through a set of thermal titanium washers able to optimize the HOPA working functions. The actual operation of the mechanism is allowed by the release of a special locking pin opportunely guided into the bracket.

The release mechanism is composed of a hinged lever (unhook lever) in contact with a plane surface. This surface is part of a moving piston (HOPA piston) that can translate inside a cylindrical cavity. The contact between the locking pin and the unhook lever plus the contact between the lever and the HOPA piston keeps the lever in place avoiding its rotation. In turn, the HOPA piston is forced in position by a compression spring (HOPA spring) designed to withstand the launch loads preventing an unwanted translation of the HOPA piston (Fig. 7).

During the release, the HOP actuator pushes the HOPA piston against the HOPA spring and friction forces (Fig. 8). This force moves the piston and thus the contact interface for the unhook lever. Once the interface is not in contact with the tip of the lever, the lever rotates about its hinge pushed by the locking pin. The locking pin is being pushed by the release springs placed close to the counter-mass interface (Fig. 9).

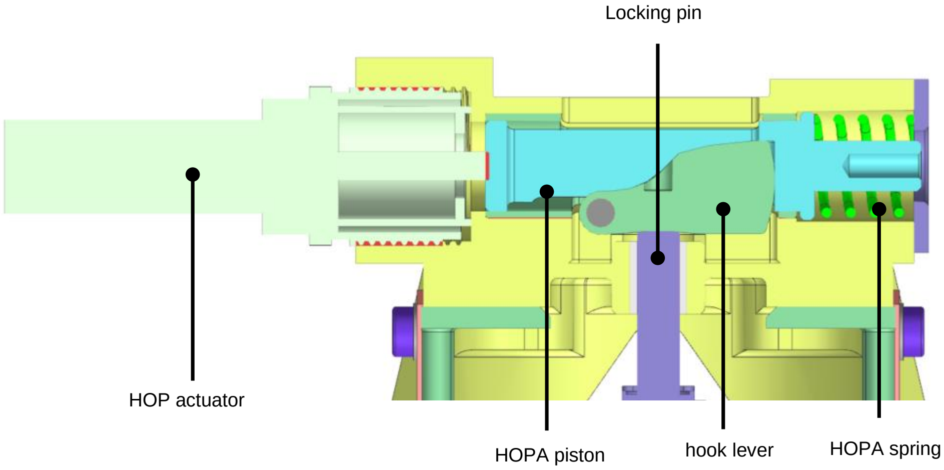

Figure 7. HDRM release device cross-section in locked position

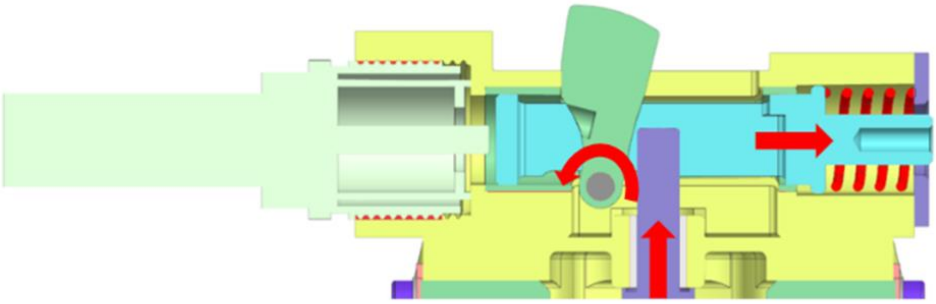

Figure 8. Release device in released configuration

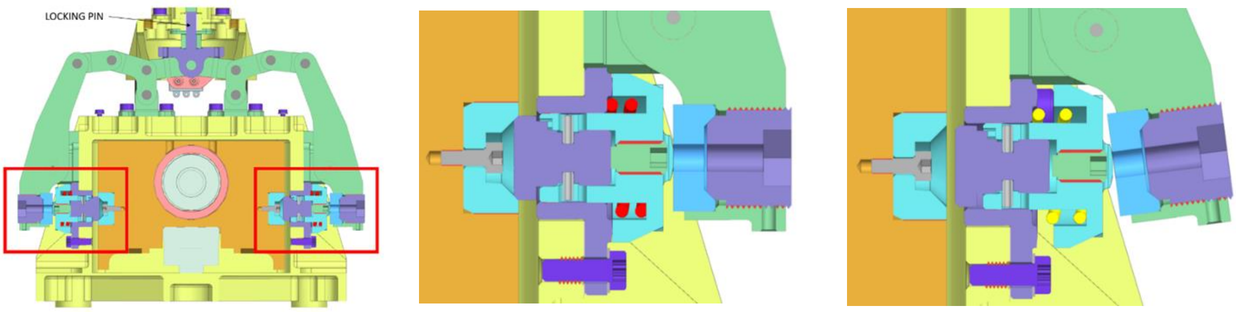

Figure 9. HDRM Release springs

4 BREADBOARD TESTING ACTIVITIES

The objective of the breadboard (BB) was to demonstrate the suitability of a set of critical function of the design and to get early feedback to implement and improve in the EQM design. In particular, the selected topics are the HDRM system to sustain the launch environment, suitability of the lubrication on the guide of the linear transmission and the applicability of a dedicated motion control law on the linear transmission through a dedicated control electronics (ABSA CE). Since the functionalities are fully decoupled (the former is working during launch when the latter is not and vice versa), they could be fully separated into two different breadboards design and test campaigns. In this paper, the main treated topics are the HDRM and its environmental tests and, with the same breadboard, the life of the selected linear guide in TVAC after the vibration tests.

In order to evaluate any degradation of the linear guide, the force needed to translate the slider was measured using a dynamometer (refer to Fig. 10). This measurement was performed after integration. Subsequently, it was performed after vibration testing. Finally, it was done after the reduced life test.

Figure 10. friction measurement

4.1 Breadboard description

The main components of the HDRM are included inside this BB: contact interfaces, links in the kinematic chain, hinges, springs, and the flight representative linear guide. In addition, materials and lubrication are also representative of the FM design. The HOPA is not included in the BB since already proven during past projects. The mechanism actuation is performed using a threaded shaft instead.

To be noted that, due to the advanced activities with respect to the main development of the instrument, the main requirements on the ABSA counter-mass and its balancing capabilities were not fully frozen. For this reason, the BB has been designed in order to be able to accommodate a counter-mass much larger than the nominal one (FM nominal mass 7 kg, FM max mass 9 kg, BB 12 kg) with the clear purpose of providing a very worst case of the system. This of course led to increase preloading force in the HDRM and to verify stress margins on all the HDRM parts.

Moreover, the ball screw is not included as it is decoupled from the HDRM thanks to the elastic coupling.

During the breadboarding activities, also the reset device has been manufactured and tested.

4.2 Vibration testing

Before the vibration test, the force needed to move the linear guide slider was measured. The stroke was divided into five zones. The force was measured several times on each zone. In this case, the minimum recorded force was 1.5 N and the maximum value was 4.3 N.

During the HDRM preload application, strain gauges were mounted on the mechanical transmission to measure the actual applied preload. These same gauges were used after the test to verify the preload reduction after the environmental tests.

Fig. 11. shows the BB on the shaker. During this test, both sine and random loads were applied, along the three axes.

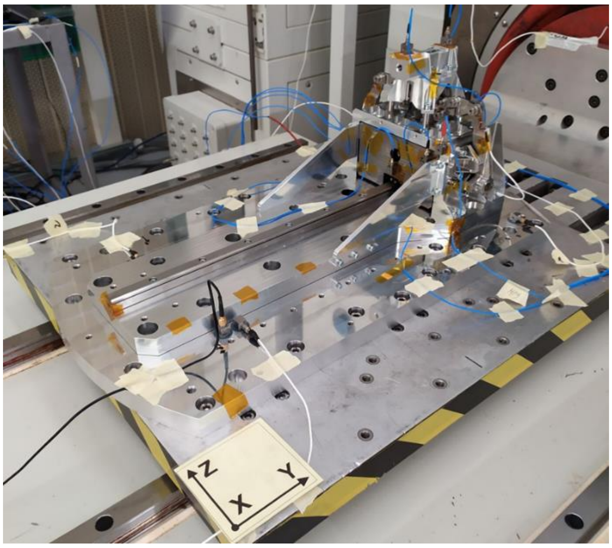

Figure 11. BB vibration test setup (Y axis)

Tab. 1 shows the sine levels applied as well as the sweep rate.

Qualification Level - Y axis		Qualification Level - X and Z axes	
Freq. [Hz]	**Acc. [g]**	**Freq. [Hz]**	**Acc. [g]**
25	5	25	15
50	32	120	15
120	32		
Sweep Rate	2 oct/min	**Sweep Rate**	2 oct/min

Table 1. Vibration test sine level

On the other hand, Tab. 2 shows the notched random levels. The levels were notched because the application of the full random qualification levels cause the 3σ accelerations at MPD CoG to exceed the Quasi-static loads.

Notched X axis (5.65 gRMS)		Notched Y axis (8.3 gRMS)		Notched Z axis (3.85 gRMS)	
Frequency [Hz]	PSD [g^2/Hz]	Frequency [Hz]	PSD [g^2/Hz]	Frequency [Hz]	PSD [g^2/Hz]
20	0.001	20	0.00688	20	0.004
100	0.030	100	0.17	100	0.020
350	0.030	200	0.17	200	0.020
370	0.002	310	0.095	310	0.01472
470	0.002	330	0.005	330	0.0016
490	0.027	430	0.0050	430	0.0016
2000	0.010	450	0.0579	450	0.01135
		2000	0.008	2000	0.004
gRMS	5.65	gRMS	8.30	gRMS	3.85
Duration	2 min/axis	Duration	2 min/axis	Duration	2 min/axis

Table 2. Random vibration test Qualification Notched levels

Inspections just after the test, on site, have been conducted and no issue was reported.

Preload qualitative verification has been carried out by checking the preloading features rotation (marked at the beginning of the test). Also, general marking on all the screws have been verified.

The counter-mass was still locked at its own position and release mechanism lever cannot move around its axis as expected.

After the UUT came back to OHB-I facility is has been subjected to the following tests which can be considered as an additional inspection:

- Preload verification through the strain gauges reading: preload still available gives the evidence that the mechanism could withstand the vibrations.
- Release test: gives the evidence that the mechanism is working, and vibrations did not affect it.
- Visual inspection of the contact surfaces: no evident damages can be detected on the contact surfaces gives the evidence that the mechanism could withstand the vibrations.
- Friction on linear guide: The measurement of friction after vibration resulted in forces ranging from 1.6 to 3.8 N. Thus, no significant changes were identified from this point of view.

After the release, the hook lever (refer to Fig. 7) suffered a minor damage. This damage was generated by the contact between the lever and the locking pin. When the load is released, the hook lever rotates rapidly towards its end stop. Once it impacts whit the stop, it recoils toward the moving locking pin. Between the start and the end of the movement of the locking pin, the lever can clash against the pin several times. These clashes generate high stress point on the lever resulting in the dents shown in Fig. 12.

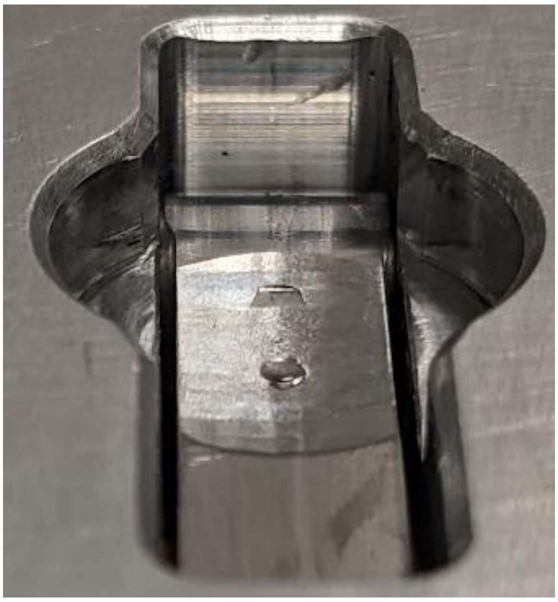

Figure. 12. Dents on the hook lever after release

The damage was deemed not critical for the HDRM functioning. To demonstrate it, 10 additional releases were successfully performed with the same lever.

4.3 Early Reduced life test

After the vibration tests, the BB was equipped with a linear transmission for a partial life test of the linear guide both in ambient and in TVAC (Fig. 13). The aim of this test was to early test the system in order to advance debugging issues on a technology with little available heritage for space applications (Linear guide, applied lubricants, sliders, etc.) as a de-risking activity. The linear transmission is based on a commercially available high helix screw. Due to the TVAC chamber dimensions, the actuator was connected to the screw at 90 degrees through a couple of bevel gears.

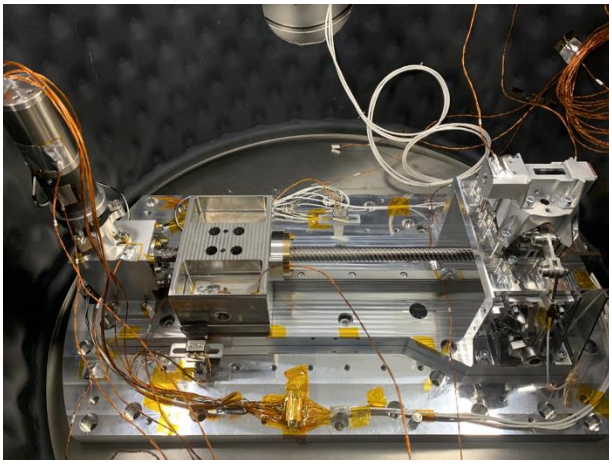

Figure 13. BB for reduced life test inside TVAC

Due to the EQM gearmotor unavailability, the actuator used was a stepper gearmotor rated for ultra-high vacuum. The lower maximum speed of the electric gearmotor combined with the hight lead of the selected screw allowed the movement of the mass using similar velocities and accelerations as the ones selected for the QM.

During the test, the starting current of the motor was measured to verify any deviation. The minimum starting current remained unchanged along the whole test.

After testing, the friction was again measured. The values ranged from 0.7 to 1.2 N.

5 LESSONS LEARNT

The BB test campaign confirmed most of the design features. Moreover, it also suggested improvements for the EQM design. Hereafter a brief list of them.

Confirmations:

1. Leverage kinematic works properly, stand alone and under full preload,
2. Contact surfaces showed neither damage nor scratches, then they are correctly dimensioned,
3. Test mass is correctly held during the test by the applied preload and by the slider,
4. Release mechanism correctly release with the factored motorization margins (measured).
5. Release springs correctly work.
6. Friction on the slider is largely below the maximum allowable value. Moreover, it did not change significantly between assembly, vibration testing, and reduced life test.

Improvements for EQM design and tests:

1. Hook lever of the release mechanism shall be modified to avoid damages during the release, as well as the release mechanism chassis for the same reason.
2. Few minor improvements in the HDRM design.
3. Strain gauges shall be included into EQM and FM design, for integration and test purposes (preloading activities) since it is the most reliable way to apply the correct preload to the HDRM.
4. Few minor modifications on the MGSEs used for integration and test, as well as to the needed tools and instruments.
6. Integration and test procedure step-by-step have been improved.

6 CONCLUSIONS

The implemented design has been driven by a high demanding flexibility in most of its functionalities since it had to cope with several requirements which were not frozen yet. The tight schedule of the instrument and S/C led to achieving a design of the ABSA that could be suitable and reliable in the early stage of the project, many months before the requirements freezing held at the PDR. Thus, counter-mass envelope and mass value, structural behaviour affected by the counter-mass itself, mass budget as well, the gearmotor selection based on allowed speed and acceleration, control loop and thermal dissipation etc have been traded off such that a wide range of options could be selected.

With the focus on the BB activities, even the adaptation of the FM design to the BB had to consider a good level of confidence to be reached and acceptable worst-case scenario had to be tested. The tests confirm the TRL6 and served as invaluable de-risking activities.

As already demonstrated by past projects as well as the experiences of the space mechanism design community, the manufacturing and testing of breadboards is a must to avoid unexpected issues during the qualification campaign. The extra effort and expenses pay off during qualification resulting in an overall reduction of risk and costs.

7 REFERENCES

1. M. Mobrem, S. Kuehn, C. Spier and E. Slimko, "Design and performance of Astromesh reflector onboard Soil Moisture Active Passive spacecraft," 2012 IEEE Aerospace Conference, Big Sky, MT, USA, 2012, pp. 1-10, doi: 10.1109/AERO.2012.6187094.

ACKNOWLEDGEMENT AND DISCLAIMER

Programme of the European Union Copernicus co-funded with esa

Views and opinion expressed are however those of the authors only and the European Commission and/or ESA cannot be held responsible for any use which may be made of the information contained therein. The authors would also like to thank Philipp Eggers (ESA) for his contributions to the ABSA project.