UNLOADING MECHANISM USING SHAPE MEMORY ALLOY ACTUATORS FOR BALL BEARINGS – SPACE QUALIFICATION TESTS SEQUENCE

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ABSTRACT

Duration of space missions is intended to increase in upcoming years and consequently, the need of instruments that could carry out longer missions is crucial for space constructors today. Thus, bearings must have a better lifetime in orbit to perform their function as long as possible.

Designing a ball bearing includes inevitably an antagonist problem. On the one hand, the stiffness of the bearings must be high during the launch phase to resist in harsh mechanical environments (vibrations, shocks, …); on the other hand, the resistive torque of the bearings must be the lowest possible to ensure a long operating lifetime, which leads to the need of a low stiffness. The unloading mechanism using shape memory alloy actuators for ball bearings presented here is a project led by the CNES and ADR to propose a technical solution to this issue. It is a bi-stable system which can preload and unload a ball bearing in order to match requirements of both phases (launch and mission in orbit).

This document presents both the development and the working principle of the mechanism, and qualification processes performed on two BreadBoard Models.

1 INTRODUCTION

The unloading mechanism is an idea of the CNES which has been developed by ADR. The innovative concept of this system is the use of a shape memory alloy actuator to preload and unload the ball bearing. The use of SMA materials (here a Titanium-Nickel alloy) instead of another type of actuators present a major advantage for a space application: it is a compact and lightweight design compared to an electromagnet or a motor, and it creates no microvibrations during the unloading process in orbit.

This system is more reliable and cheaper than a previous one (called BAPS for Adjustable Bearing Preload) because its mechanical design and working principle is far less complex. Furthermore, ADR has already worked on a similar system for cryogenic application, so they only had to adjust the materials considering their CTE. They have unprecedently clear view of numerous spacecraft applications using ball bearings.

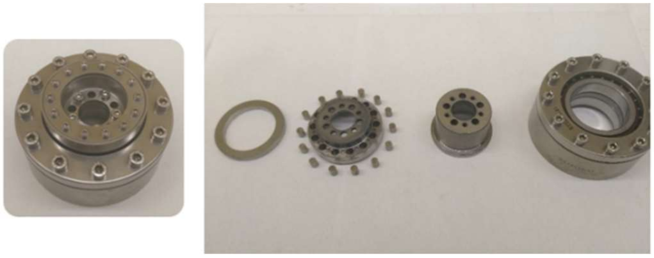


Figure 1 – Views of the unloading mechanism using SMA actuators

The aim of this study is to perform a space qualification tests sequence following the standard practice to evaluate the ability of the system to survive severe conditions. This qualification sequence has been carried out on two breadboard models (BBM) provided by ADR. Most of these tests have been fulfilled using the test facilities of CNES's mechanism department.

Both BBM are composed of two ball bearings which are oil-lubricated. The bearings are in B-type configuration. They are preloaded at 2,000 MPa (Hertz pressure in the bearings) by fourteen shape memory alloy actuators which apply a load on the interior ring of the bearing. This configuration is called hard preload[(1)]: An axial load (here applied by the SMA cylinders) creates a deflection in the ball bearings. By knowing the stiffness of the bearing, it is possible to adjust a suitable Hertz pressure. This configuration provides a higher bearing stiffness and a more predictable behaviour than a soft preload. Thus, it becomes possible to adapt the stiffness of the bearing if the deflection varies.

Currently, the BBM are not designed for any specific mission because there is no any customer yet for its application. For the tests sequence, common requirements for space mission of a mechanism using a ball bearing (such as scan mechanism) have been chosen to qualify the mechanism. In this way, it will be easier to prove to potential customers that the mechanism is liable to be qualified.

2 SPECIFICATIONS AND DESIGN DESCRIPTION

2.1 Specifications / Performances

The performances specified for the working conditions of the system are listed in table 1.

Table 1 – Standard specifications for trade-off

<table>
<tr><td>Mechanical environment - vibrations</td><td>
<table>
<tr><th>Axe</th><th>Fréquence (Hz)</th><th>PSD (g^2/Hz)</th><th>Pente (dB/Oct)</th></tr>
<tr><td rowspan="3">Z</td><td>20-100 Hz</td><td>NA</td><td>3 dB/Oct</td></tr>
<tr><td>100-300 Hz</td><td>0.15</td><td>NA</td></tr>
<tr><td>300-2000 Hz</td><td>NA</td><td>-5 dB/Oct</td></tr>
<tr><td rowspan="3">X, Y</td><td>20-100 Hz</td><td>NA</td><td>3 dB/Oct</td></tr>
<tr><td>100-300 Hz</td><td>0.2</td><td>NA</td></tr>
<tr><td>300-2000 Hz</td><td>NA</td><td>-5 dB/Oct</td></tr>
</table>
</td></tr>
<tr><td>Mechanical environment - shocks</td><td>
<table>
<tr><th>Fréquence</th><th>Niveau qualification (g)</th></tr>
<tr><td>100 Hz</td><td>20g</td></tr>
<tr><td>100 - 1000Hz</td><td>Linéaire log-log</td></tr>
<tr><td>1000 – 10000Hz</td><td>1000g</td></tr>
</table>
</td></tr>
<tr><td>Thermal environment – Operational</td><td>Working temperature: -100/+100 °C</td></tr>
<tr><td>Thermal environment – Stockage</td><td>- Working temperature in preload configuration : -100/+80 °C
- Working temperature in unloaded configuration : -150/+150°C</td></tr>
<tr><td>Vacuum degassing</td><td>All materials used for the manufacturing should respect ECSS:
- Total Mass Loss < 1%
- Collected Volatile Contaminant Material < 0.1%</td></tr>
<tr><td>Cleanliness class</td><td>Class 100,000</td></tr>
</table>

2.2 Working principle of Shape Memory Alloys

A shape memory alloy is a material capable to fully recover its initial dimensions when it is warm above a specific temperature called austenitic transformation temperature. At low temperature, SMA can be deformed elastically in its martensitic state. Then, when the deformed SMA is warmed up, the austenitic transformation takes place and the SMA recovers its initial dimension due to the change of its crystallographic structure[4]. This process is described on the figure 2.

Shape memory alloys have two shape-memory effect: the one-way memory effect and the two-way memory effect. The working principle of the first one has been described above. The two-way shape memory effect is the effect that the material "remembers" a shape in the cold case (martensitic state) and in the hot case (austenitic state). It can be obtained through the application of an external load during heating or if a training process has been applied to the material.

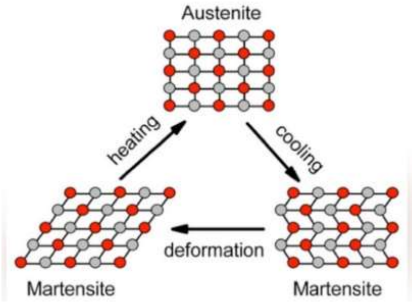


Figure 2 – Working principle of Shape Memory Alloy

This property of SMA (which is not the only one) has been used to create SMA actuators that have been mounted in the unloading system of the mechanism. Thus, the system is not dependent of the poor precision of the two-way shape memory effect. The TiNi (Titanium-Nickel) actuators have been manufactured with a precision of ±0.01 mm. The actuators have been grinded by ADR to fit in their drill hole.

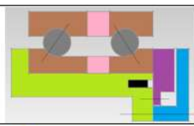
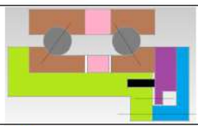
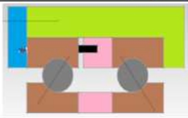
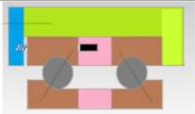
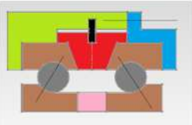
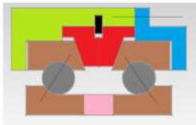
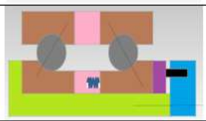
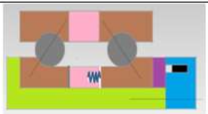
2.3 Trade-off proposed by ADR

ADR has defined several criteria to define the feasibility of the different technical solutions considered for the unloading system of the mechanism. The main criteria used to realise this trade-off are listed below :

- Contact pressure on SMA actuators
- Easy assembly
- Easy manufacturing
- Compactness
- Easy rearming (for unloading system)
- Backlash

The table 2 presents the working principle of the technical solutions which have been considered in both case: preloaded and unloaded states.

Table 2 – Technical solution for the unloading system

	Preloadead configuration	Unloaded configuration
Solution 1		
Solution 2		
Solution 3		
Solution 4		

Based on the list of criteria presented before, the technical solution chosen by ADR to manufacture is the system n°4 (without spring in the bearing spacer). It is the more interesting one for this application because it is compact, easy to manufacture and easy to assemble.

2.4 Design description of the unloading mechanism of ball bearings

Based on the technical solution n°4 presented above, ADR has manufactured the mechanism which includes:

- A pair of ball bearings mounted in back-to-back configuration
- The unloading system composed of a part similar to a gun barrel that contains the fourteen SMA actuators. This system is mounted on the interior shaft of the mechanism to exert the preload load on the interior ring of the bearing through the SMA actuators.

In the preloaded configuration, the fourteen SMA actuators apply a load on a spacer which is in contact with the interior ring of the bearing. This preload creates a Hertz pressure of 2,000 MPa on the balls of the bearing.

After the thermal activation of the SMA actuators beyond the austenitic transformation temperature, the mechanism is in the unloaded configuration where the spacer is not in contact with the SMA actuators anymore, but with the surface of the “gun barrel” part as illustrated in figure below. The Hertz pressure on the balls of the bearing is only 500 MPa.

The figure 3 shows the working principle of the mechanism.

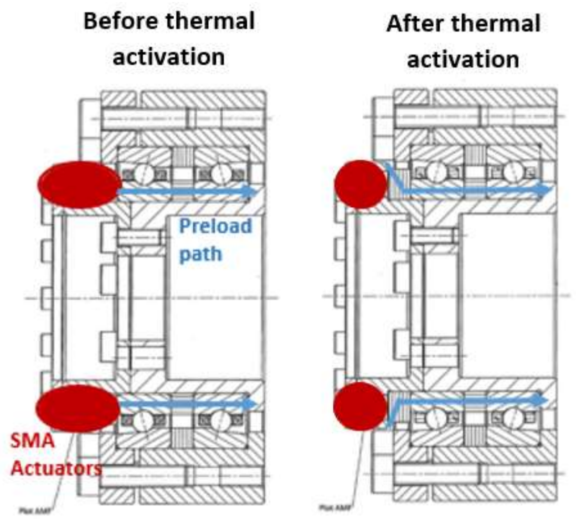


Figure 3 – Working principle of the mechanism

Besides, the characteristics of the mechanism in terms of resistive torque are the following :

Table 3 – Performances of the component

	Resistive Torque (cN.cm)	Bearing stiffness (N/µm)
Preloaded conf.	220	85.1
Unloaded conf.	50	48.3

In the unloaded configuration, the resistive torque is significantly lower than the one of a classic bearing for space mission. Therefore, this improvement will allow a longer operating lifetime in orbit for the bearing and less electrical power for the motor.

2.5 Retrofit process

As ADR has provided two BBM, it has been decided to refit one of them with ball bearings lubricated with Molybdenum disulphide (MoS_2). Spacers for a preload of 1,500 MPa has been used during the retrofit process instead of the initial ones of 2,000 MPa.

3 MECHANICAL ANALYSIS

3.1 Mechanical environment

The mechanism shall be capable to support a payload such as a mirror or reflector and to handle the mechanical environment of the launch phase. Thus, a ballast of 1 kg. has been added to the mechanism to reproduce the payload behaviour and as a consequence a consistent dynamic behaviour of mechanism’s rotor. It is primordial to describe the configuration and the natural frequency of the system in operating conditions. Indeed, in the RBSDyn study and during the vibration testing, the interior shaft will be fixed, and the exterior rotor is free to move. Thus, by increasing the mass, the natural frequency of the system will decrease.

3.2 RBSDyn model

RBSDyn is a software utilized to compute the loads and Hertz pressures exerted in a bearing by considering inertial properties of an axisymmetric model. RBSDyn calculates the load distribution inside each ball bearing in quasi-static and dynamic configuration. It enables the determination of the specifications and especially the qualification levels of random vibrations for the vibration test.

The model of the unloading mechanism of ball bearings is presented on the figure 4.

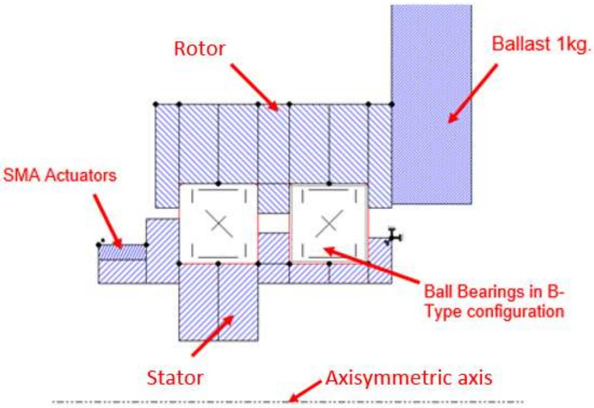


Figure 4 -RBSDyn model

RBSDyn enables the computation of the natural frequencies of a model as well. The first natural frequency for each BBM corresponds to an axial mode. This frequency can be identified with a tap-test and gives an information on the stiffness of the bearing. This will be helpful to characterize the system after each test.

3.3 Calculation of qualification levels

First of all, the computation of the qualification levels for the vibration test has been realised for both 1,500 MPa and 2,000 MPa BBM. These levels have been calculated the same way for both models, therefore only one method will be presented here, and the results are gathered in the next section.

The qualification level of random vibration corresponds to the Power Spectral Density (PSD). It describes how power is distributed over a range of frequencies. According to ECSS, the range of frequencies that must be studied for a vibration qualification is [20 - 2000] Hz. PSD is given in g^2/Hz, but it is commonplace to use the value of its root mean square (RMS) to describe the level of vibrations over the range of frequencies considered.

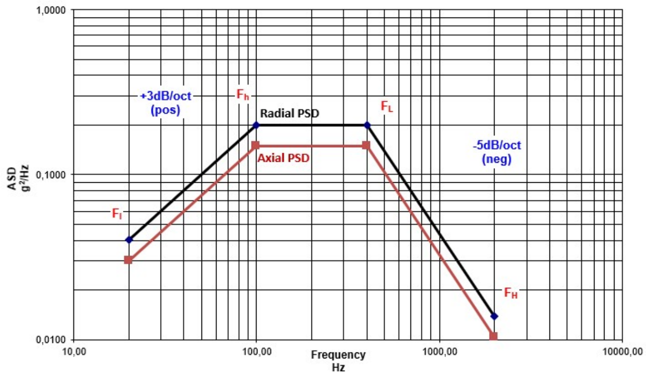


Figure 5 – Axial and radial PSDs for vibration testing

As there is no specification for the PSD because the project has no any customer yet, the purpose of these vibration tests is to inject a PSD superior to 10 gRMS which is equivalent to a generic level of PSD. The aim of this computation is getting *the maximum PSD for the vibration test without risks for the ball bearing*. The final goal of the vibration test will be to prove that the SAM device does not limit the mechanical strength of the mechanism. A margin of safety is defined to ensure the mechanical handling of the ball bearings:

$$M(\%) = P_{admissible} / (P_{calculated} \times C_{safety}) - 1$$

Handling of the ball bearings for a radial load (BBM 2,000 MPa):

The maximal Hertz pressure is 3,087 MPa and refers here to the mean Hertz pressure to which is added three standard deviations. The dynamic response of the bearings is given on the figure 6.

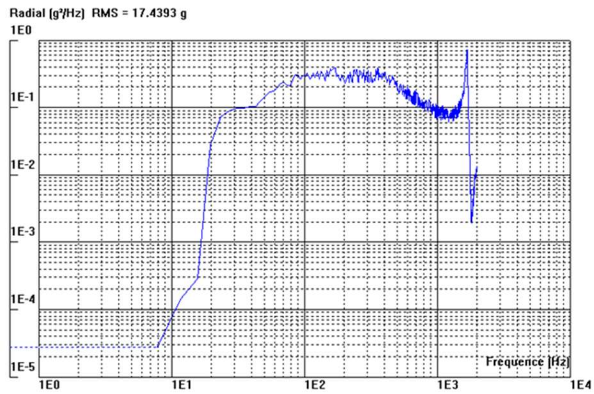


Figure 6 – Frequential response for radial axis

Handling of the ball bearings for an axial load (BBM 2,000 MPa):

The 3σ-Hertz pressure is 3,237 MPa.

3.4 Synthesis

The results of the calculation in random vibration are specified in table 3.

Table 4 – Margin of safety for vibration testing

Reference			Units	WSP11528	Margin of safety
Mechanism preloaded at 1,500 MPa	Radial	Random vibration Pressure Max	MPa	3158	16%
	Axial	Random vibration Pressure Max	MPa	3305	11%
Mechanism preloaded at 2,000 MPa	Radial	Random vibration Pressure Max	MPa	3083	18%
	Axial	Random vibration Pressure Max	MPa	3237	13%

4 QUALIFICATION PROCESS

4.1 Qualification tests sequence & Test bench

The first BBM preloaded at 1,500 MPa has been submitted to the entire tests sequence described below (Diagram 1). The second BBM preloaded at 2,000 MPa will only be submitted to hazardous tests (vibrations and shocks).

Diagram 1

Retrofit process
*
Rolling track running-in + Ground lifetime
*
Random vibration testing
*
Thermal cycling in preloaded configuration
*
Thermal activation under vacuum
*
Thermal cycling in unloaded configuration
*
Flight lifetime

** means Functional test*

To perform the whole sequence, a test bench has been designed and manufactured to measure the resistive torque C_r during the tests:

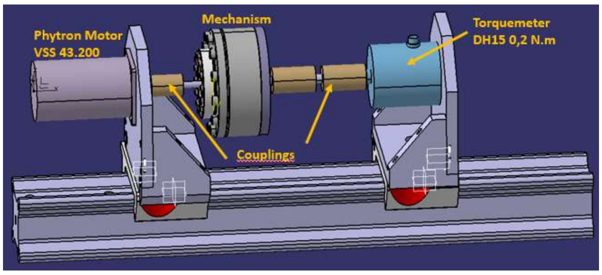


Figure 7 – CAD model of the test bench

4.2 Characterisation of axial modes

Tap-tests have been realised in both preloaded and unloaded configuration for each mechanism (dry lubrication at 1,500 MPa and oil lubrication at 2,000 MPa). The results show expected behaviour: axial mode in unloaded configuration (SMA flange released) are the same for both components. In preloaded configuration, axial mode of the oil-lubricated mechanism is higher because the preload is higher, so is the stiffness of the bearing.

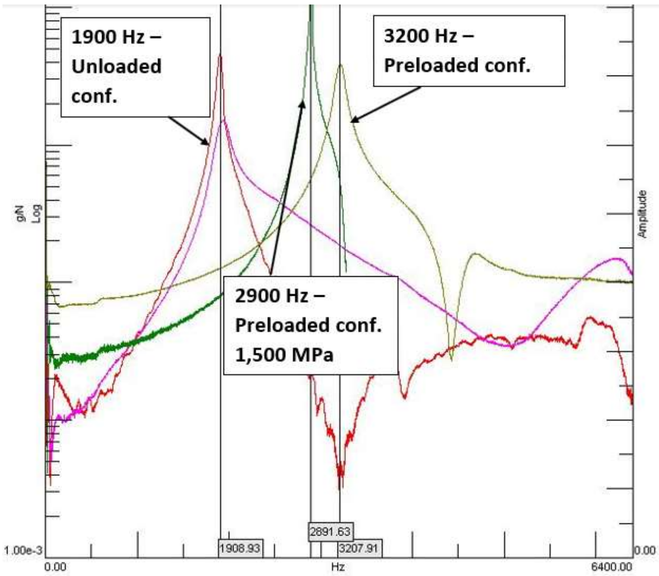


Figure 8 – Axial modes: (pink & red at 1900 Hz) Unloaded configurations; (green at 2900 Hz) Preloaded configuration at 1,500 MPa; (yellow at 3200 Hz) Preloaded configuration at 2,000 MPa

4.3 Ground Lifetime

The specifications of the lifetime test are the following:

- 650,000 revolutions
- Rotation speed : 60 rpm
- Nitrogen chamber
- Room temperature and pressure

They correspond to the lifetime requirements of another space mission of the CNES.

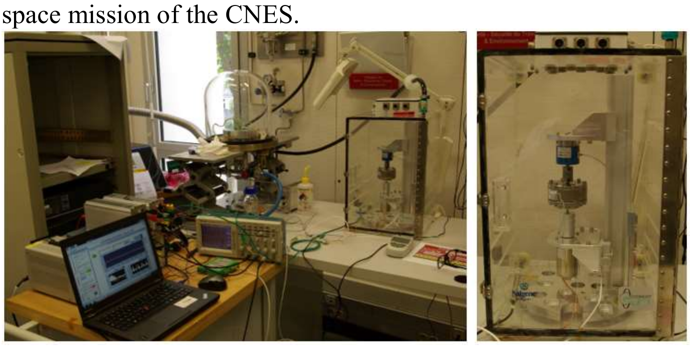


Figure 9 – Test bench for ground lifetime

The first part of the lifetime has been carried out with 100,000 revolutions. On the figure 10, the resistive torque shrinks significantly from 400 cN.cm to 150 cN.cm during the first revolutions. This is due to the running-in of the rolling track lubricated with MoS_2.

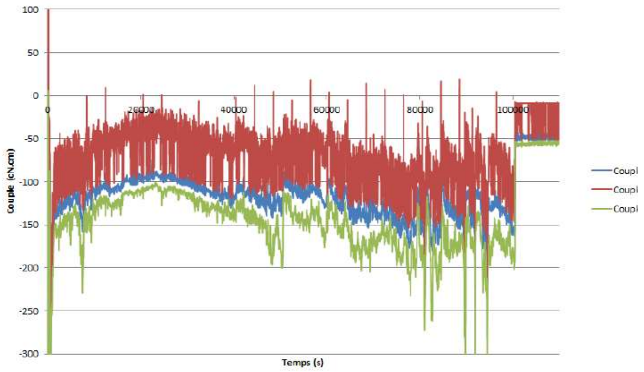


Figure 10 – Evolution of the resistive torque the first lifetime

The resistive torque is more or less steady. The second part of the lifetime shows that the resistive torque grows thick and fast. It has reached its initial value of 400 cN.cm. For the rest of the test, it has been steady around this value.

4.4 Vibration testing

Vibration testing has been carried out on a shaker testing machine in a clean room. The test on axial axis (Out Of Plane configuration) is described on figure 11.

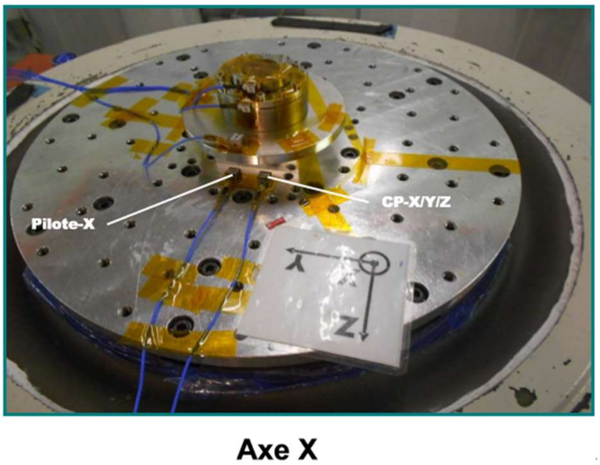


Figure 11 –Axial configuration for vibration testing

For each case, the results are the following:

Table 5 – Results in both configuration

Capteur	Superposition Associée définissant l'axe de vibration	Fréquence Initiale (Hz)	Amplitude Initiale (g, g/g, g² or g²/g²)	Fréquence Finale (Hz)	Amplitude Finale (g, g/g, g² or g²/g²)	Dérive en Fréquence (%)	Dérive en Amplitude (%)
Lest-X	SUP-02-X	893	73.7	866	86.6	-3.0 %	17.4 %
Lest-Z	SUP-01-Z	1 032	25.7	1 023	22.6	-0.9 %	-12.1 %
Rotor-X	SUP-02-X	893	8.8	862	8.0	-3.5 %	-8.5 %
Rotor-Z	SUP-01-Z	1 032	22.5	1 023	19.6	-0.9 %	-12.9 %
Stator-haut-X	SUP-02-X	893	66.8	866	79.5	-3.0 %	19.0 %
Stator-haut-Z	SUP-01-Z	1 032	26.1	1 023	23.1	-0.9 %	-11.6 %

The frenquency-shift and amplitude-shift measured are in line with the success criteria which state that frequency-shift should be inferior to 5% and amplitude-shift inferior to 20% of the initial values.

Modal comparison (table 5) shows that the RBSDyn model and the vibration testing are coherent for the axial mode. Yet, the overturning mode measured during the test does not correspond to the RBSDyn predictions.

Table 6 – Modal comparison

Preload 1,500 MPa	RBSDyn	Vibration testing
Axial mode	956 Hz	866 Hz
Overturning mode	1457 Hz	1023 Hz

A study to explain the differences between the RBSDyn model and test results has not been achieved yet. The CNES is still adjusting the RBSDyn model. Furthermore, the Grms response of the mechanism was superior to the one predicted by RBSDyn. The risk to pass the PSD levels at 0 dB has been accepted and the system has not been damaged.

4.5 Thermal cycling in preloaded configuration

The specifications of thermal cycling test are the following:

- Nitrogen chamber
- Room pressure
- Temperature range : -50°C/+45°C
- 8 cycles

The cycles must respect the ECSS of thermal cycling(3). The resistive torque has been measured at different temperature stages during the first cycle and the last one. The figure 12 described the thermal interfaces of the test bench for thermal environment:

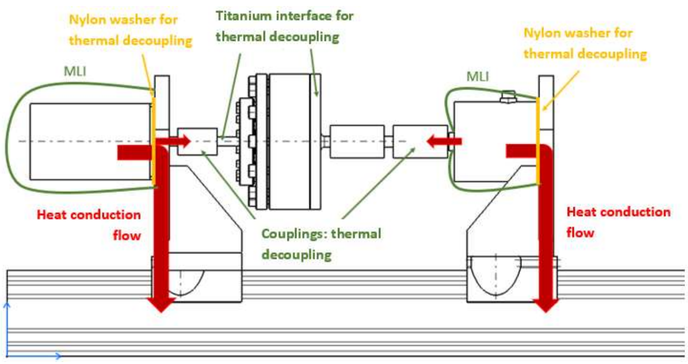


Figure 12 – Thermal study of the test bench

Figure 13 – Test facilities for thermal cycling

Table 7 – Peak-to-Peak results at each stage

Stage temperature	Rotational speed (rpm)	Peak-to-Peak value (cN.cm)
20°C	60	870
	3	980
35°C	60	850
	3	1000
45°C	60	900
	3	920
-5°C	60	1100
	3	1180
-20°C	60	1200
	3	1170
-35 °C	60	1080
	3	1080
-50°C	60	1080
	3	1080

The peak-to-peak amplitude at the temperature stages does not vary much due to dry lubrication. The little change observed for the cold temperature stages is explained by the fact that the torquemeter has not been calibrated for these temperatures.

4.6 Thermal vacuum : activation test

This test is the major issue of the qualification sequence. It proves that the mechanism is functional in operating conditions, that is why the test is carried out in a vacuum chamber (figure 14). The specifications of this test are the following:

- Temperature of the chamber : -50°C
- Pressure : 10^{-6} torr
- Power supply of the warmers: 12 W

Figure 14 – Test facilities for thermal vacuum environment

Austenitic transformation is not immediate[4]: during heating, it begins when $T=A_s$ (the temperature at which the austenitic transformation starts) and the SMA is fully austenitic when $T=A_f$ (the temperature at which the austenitic transformation is complete).

The instrumentation with thermocouples and warmers is described on figure 17. The thermocouples allow the measure of the temperature gradient in the mechanism, and especially in the "gun barrel" part where are located the SMA actuators (in order to check that SMA transition temperature is reached). The warmers are fixed on the surface of the "gun barrel".

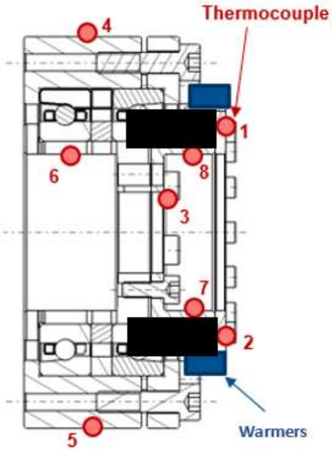


Figure 17 – Instrumentation for thermal vacuum activation

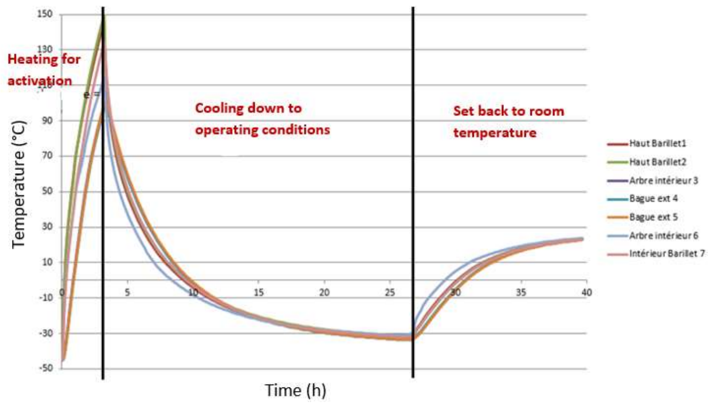


Figure 15 – Evolution of the temperature during the thermal vacuum activation test

Firstly, the pressure is reduced to 10^{-5} torr in the vacuum chamber. Then, the temperature has been lowered down to -50°C.

At this moment, the warmers are powered on the warming phase start at t = 0. Three hours of warming have been required to reach 140°C in the "gun barrel". The graph on the figure 15 shows the evolution of the temperature during the entire test.

When T = 140°C, the power supply of the warmers is shut down and the temperature of the mechanism decrease slightly to -35°C. The three hours required to reach A_s can be reduced with a better thermal isolation.

Finally, the vacuum chamber has been set back to room temperature and pressure.

The functional test (tap-test and measurement of the resistive torque) proves that the SMA have been activated properly. Indeed, the stiffness of the bearing and the resistive torque are lower (figure 16 and 17).

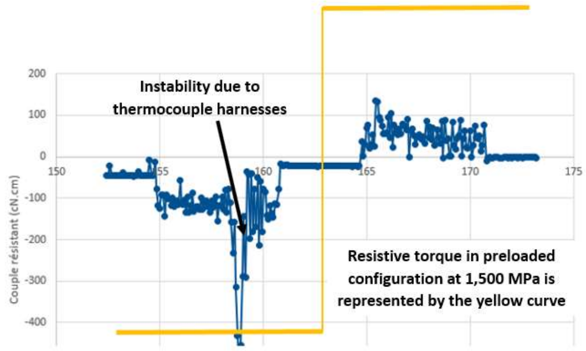


Figure 16 – Resistive torque measured after activation (unloaded configuration)

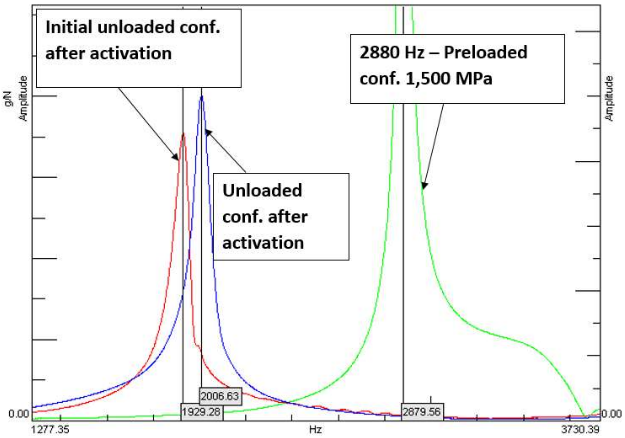


Figure 17 – (green) Axial mode in preloaded configuration before activation; (red) Axial mode in unloaded configuration before tests; (blue) Axial mode after activation

Therefore, the ability of the mechanism to unload the bearing has been proved in operating conditions. Furthermore, it can be seen on the figure 16 that the resistive torque is far below than the initial one (represented in yellow on the graph). It means that the mechanism is capable to *extend the lifetime of ball bearings*.

4.7 Thermal cycling in unloaded configuration

The specifications are strictly the same as above for the thermal cycling in preloaded configuration.

The results are also similar. Therefore, the same conclusions can be made: the resistive torque of the mechanism is not temperature dependant. The errors are due to the calibration of the torquemeter.

4.8 Synthesis

The graph below (figure 18) summarised the evolution of the stiffness and the resistive torque of the bearing after each test. It proves that there was not any major change in the configuration of the mechanism during the whole qualification sequence.

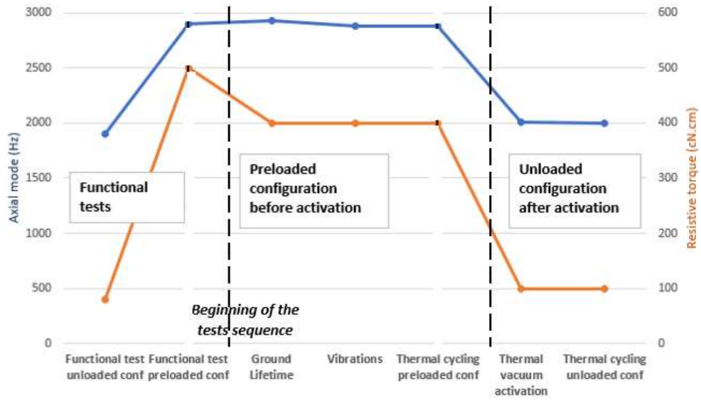


Figure 18 – Evolution of the resistive torque and the axial mode (bearing stiffness) during the tests sequence

5 LESSONS LEARNED

5.1 Correlation between models and functional tests before tests sequence

Before the beginning of the test sequence, functional tests have been performed in order to determine whether the RBSDyn model was well-correlated to the results of these tests. The qualitative behaviour in terms of modes predicted by the model was correct but the values of these modes did not match (error of 35%).

Due to planning constraints, the vibration testing has been running with a model which was not adjusted correctly. Thus, the results of the vibration testing could not have been interpreted fully yet, especially because currently, the RBSDyn model is still being adjusted.

Therefore, it is crucial to readjust the model to understand what did not work in order to avoid this issue again with future modelling on RBSDyn. Furthermore, it is absolutely necessary to adjust the model in order to match the functional test (tap-test, test on a Kistler table, …) to be sure that the results of the tests sequence (vibrations and shocks tests) will be relevant.

5.2 Difficulties to define generic specifications

As it has been explained in the introduction, it was central to draw up generic specifications for the tests to best represent the operating conditions of a space mission.
The ECSS(2)(3) have been helpful to define generic specifications for the thermal cycling tests and shocks test. For the lifetime tests, the specifications have been chosen from a previous space mission where designing constraint on ball bearings have been encountered.

Moreover, it is important to underline that during the second test sequence with hazardous tests, the limits of the ball bearing have been reached. It means that the mechanism has been submitted to the maximal generic specifications.

However, for vibrations testing and thermal vacuum activation, the specifications were not generic because strong and particular hypothesis have been made. Moreover, RBSDyn modelling has implied to design a specific ballast for vibration testing, which is specific to this mechanism.

A discussion between space constructors to define more precisely generic specifications for standard tests would help to lead space qualification tests sequence on mechanisms which have not any mission yet.

It still remains many improvements to do for vibrations testings but engineers must rely on ECSS. Besides, it must be enlightened that a space mission is and will always be specific. Therefore, it is quite unlikely that space engineers will define generic requirements for ball bearings systems. That is why it was not possible to have generic specifications for the thermal vacuum activation test.

6 CONCLUSION

The qualification tests sequence has successfully proved that the mechanism can handle classical space environment (vibrations, shocks, vacuum, …) at typical qualification levels. During the risk mitigation tests on the second BBM, the rolling track of the bearing have been damaged. However, after these tests, the unloading system is still fully functional in operating conditions (activation of SMA actuators in thermal vacuum), which is an encouraging success for the qualification of the component.

Therefore, this system could be a viable option to the designing constraints of ball bearings for space applications. Indeed, the bi-stable system respond to engineer's antagonist need to have a high bearing stiffness for the launch phase and a low one in orbit. Both tests sequences have also demonstrated that the SMA actuators are not a restricting factor regarding the drawing up of the requirements of the mission.

Eventually, this study can attest to any potential customers that the unloading mechanism for ball bearings is working in operational conditions and is liable to qualify for a space mission.

7 REFERENCES

[1] Collective authors: Technics and Technologies for Spacecraft, Volume 4 – Spacecraft bus (2014)

[2] ECSS-E-ST-32: Structural general requirements (2008)

[3] ECSS-E-ST-31: Thermal control general requirements (2008)

[4] Etienne Patoor & Marcel Berveiller: Technology of shape memory alloys, Hermes (1994)