DESIGN, BUILDING AND TESTING OF A SUN CALIBRATION MECHANISM FOR THE MSI-VNS INSTRUMENT ON EARTHCARE

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ABSTRACT

TNO has developed a mechanism to perform sun and dark calibration as a module of the Visible-NIR-SWIR Optical Unit (VNS) in the context of the ESA EarthCARE mission. This paper will address the conceptual and detailed design and modelling approach of the mechanism. Finally the production and testing of the Life Test Model (LTM) will be presented. The rotating part of the mechanism (calibration carousel) is the supporting structure of the instrument calibration diffusers. By rotating the carousel either the instrument nominal, sun calibration or dark calibration/safe modes can be selected. The calibration carousel is suspended in (a.o.) hard preloaded angular contact bearings and driven by a Phytron stepper motor. FE Modelling has been used to derive the bearing- and motor forces and accelerations. These analysis results were used as input to the CABARET analyses performed by ESTL (UK). Using the analysis results the bearing stress, stiffness, gapping and friction torque were predicted. A flight representative Life Test Model (LTM) has been manufactured assembled and was successfully subjected to ground cycles testing, vibration-, thermal vacuum- and life cycle testing.

1. INTRODUCTION

The EARTHCARE satellite mission objective is the observation of clouds and aerosols from low earth orbit. The key spatial context providing instrument within the payload suite of four instruments is the Multi-Spectral Imager (MSI) [1]. The MSI is intended to provide information on the horizontal variability of the atmospheric conditions and to identify e.g. Cloud type, textures, and temperature. It will form earth images at 500m Ground Sample Distance (GSD) over a swath width of 150km; it will image earth in seven spectral bands: one Visible, one Near-IR (NIR), two Short-Wave IR (SWIR) and three Thermal IR (TIR). The instrument therefore comprises two optical modules:

- The Thermal IR (TIR) optical unit [5].
- The Visible-NIR-SWIR (VNS) optical unit [3].

The development of the overall MSI instrument has been reported earlier [6], [7].
wavelength range. The focal length (22.2mm) of the telescopes is based on the orbit height, ground sampling distance requirement and the 25µm detector pixel pitch.

The in-orbit operating temperature of the VIS/NIR and SWIR-1 detectors is 300K, while the operational temperature of SWIR-2 is 235K. The SWIR-2 radiator panel, with a surface of 0.175m², is designed to reach a temperature well below the target. An active thermal control circuit stabilizes the SWIR-2 detector temperature at 235K.

In order to achieve the required radiometric accuracy of 10% absolute and 1% relative, the VNS will be regularly calibrated in-orbit. Dark measurements will be performed for offset corrections during eclipse of the orbit; sun calibration will be performed for response calibration during passes over the South Pole region. During the sun calibration the sun light will illuminate a pair of Quasi Volume Diffusers (QVD). The QVDs are mounted on a rotating carousel (see Figure 4), which is used to switch between dark calibration, sun calibration and earth operational viewing modes.

Finally the Sun Calibration Baffle is implemented in order to prevent straylight during calibration. The baffle ensures that all reflections on the Spacecraft and the Earth are fully blocked.

2. CMA DRIVING REQUIREMENTS AND CONSTRAINTS

The overall design of the calibration mechanism was driven by the following functional and environmental requirements:
- Angular reproducibility of the carousel < 0.2deg
- Thermal operational design temperature of -30…+50°C
- Random vibration loads. The input level of the unit is ~10 grms and the response at the I/F with the mechanism is ~20 grms
- Motorization margin (MoS on friction torque)
- Micro-vibrations induced by the mechanism (the drive motor in particular)

Additionally, the design of the mechanism was further constrained by the available envelope and the calibration approach of the VNSOU. The carousel of the mechanism should enable two sets of diffusers (one set for calibrating every orbit and one set for monthly calibration), a dark calibration and a Nadir viewing operational position. This requires the need for four mechanism positions and with the available envelope results in a required 360° rotation of the carousel.

3. CALIBRATION MECHANISM CONCEPTUAL DESIGN

In order to fulfil all the driving requirements a number of conceptual design trades were made early in the program:
- Bearing Suspension Trade
- Mechanism Positioning Trade

3.1. Bearing Suspension Trade

Ideally a symmetrical bearing suspension of the carousel is preferred (Figure 5). This provides the highest stiffness and results in an even distribution of bearing forces during launch. Unfortunately, the available envelope does not allow the implementation of this concept.

To handle this constraint an a-symmetrical bearing support was implemented, as shown in Figure 6. The ratio of ~ 1:2 results in a 2² = 4 times lower stiffness (stiffness in principle scales with the square of a distance) felt locally at the CoG of the carousel and therefore a factor 2 times lower eigenfrequency when compared to the symmetrical concept. This reduced eigenfrequency is not beneficial for the structural integrity of the mechanism, since this is close to the excitation frequency of the main optical housing. For that reason and the fact that the second (the right one of the two) bearing has to withstand higher loads, a hard
A preloaded duplex bearing is placed near the carousel. For thermal robustness (i.e. CTE matching) the bearings are mounted in a bush of the same material (SS 440C). The drive motor is chosen to be a stepper motor, equipped with its own bearings and connected to the OU housing (fixed world). In order to prevent an over determined system, the carousel and the motor shaft are coupled using a flexible coupling. This coupling is stiff in rotation (around the centre line) and weak in the other directions.

3.2. Mechanism Positioning Trade

In order to achieve the required positioning accuracy combined with the need to allow for four positions and a 360° degree rotational freedom, three positioning concepts were considered. The first concept consisted of the application of mechanical end stops. This concept allowed the accurate positioning for two positions only and did not allow for a 360° rotation. Because at least three of the four positions needed a high positioning accuracy this option was disregarded.

The second concept consisted of counting the steps of the stepper motor and thus keeping track of the position of the mechanism. This concept was disregarded based on the inherent risk of “losing” steps during operation of a stepper motor and the resulting uncertainty in the achievable accuracy.

The final concept consisted of the application of optical switches combined with unique sections (so called binary or grey code). This concept fitted all the requirements (see Figure 13).

4. MODELLING OF THE MECHANISM

The above described conceptual design is supported by modelling activities. Herein, the link is made with the design drivers as mentioned before. The focus in this section is applied on the description of the modelling activities of the following design drivers:

- Random vibration loads and the resulting bearing loads
- Bearing gapping, stress and friction torque

4.1. Random Vibration analyses

A dedicated FE model has been made of the Optical Unit Assembly (OUA). This includes the Calibration Mechanisms Assembly (CMA). The modelling of the bearings is considered to be worth mentioning. Since a modal and random vibration analysis makes use of a constant stiffness, a Hertzian stiffness (which is inherent due to the application of ball bearings) is not suited. For that reason, a linear bearing stiffness is introduced in the model (see Figure 7).
stress ($\sigma$) of the ‘bearing FEM struts’ and multiplication with the cross section of the struts (A), i.e. $F = \sigma \times A$. Both methods are verified against each other and lead to comparable numbers.

4.2. Bearing analyses

With the bearings forces due to random vibrations as input (together with a.o. the bearing geometry), analyses have been performed on bearing gapping, stiffness, stress and friction torque. In order to have sufficient motorization margin, a relatively low bearing preload of ~50 N is chosen. The resulting friction torque is used for the motorization margin calculation and, following the guidelines from the ECSS on mechanisms [9], resulted in a margin of safety of $\text{MoS}_\text{torque} = 0.1$. With the preload known, the bearing stiffness can be calculated which is used as input for the random vibration analyses. Note the iterative behaviour of the analyses on the bearing stiffness since it depends on the forces due to random vibrations.

The maximum allowable bearing gapping is set to 20$\mu$m. This number is established based on both the research conducted by ESTL (as described in [8]). The duplex bearing was found to be the most critical and for that reason, only this bearing is described in this article. $\text{Figure} \ 8$ shows the results of a CABARET analysis on axial gapping of the duplex bearing. The horizontal axis shows the axial bearing force in [N]. On the left vertical axis the bearing stress [MPa] is seen. The right vertical axis shows the axial displacement of the inner bearing ring [$\mu$m].

![Figure 8: The results of a CABARET analysis on axial gapping of the duplex bearing](image)

The duplex consists of two individual bearings (A and B). For each bearing both the stress and the axial deflection are calculated. When the axial force is increased, the stress increases in bearing B and decreases in bearing A. At some point the stress in one of the two bearings becomes zero (@ +200N for bearing A). From this point on, this bearing will start gapping. The maximum force from the random vibration analyses is found to be 765 N (3-$\sigma$). The resulting calculated axial gapping (at 765N) is 20 $\mu$m (on the right in the figure) which is just within the allowable limit.

The maximum bearing stress is determined with a similar analysis (radial gapping) and yields 2300 MPa. The allowable limit is 3200 MPa (including a safety factor of 1.25) resulting in a design with sufficient margin for strength.

The same CABARET models have been used to derive the friction torque in the bearings which is used as input for the angular reproducibility calculations.

5. DETAILED MECHANISM DESIGN

The basis of the mechanism design is the carousel. It consists of: the carousel housing (~$\varnothing$150mm), the cover plate, two sets of diffuser mounts and the shaft as can be seen in $\text{Figure} \ 9$. The shaft is manufactured from (hardened) stainless steel (SS) 440C and the remaining structural parts are made of aluminium.

![Figure 9: The detailed design of the carousel](image)

The bearing suspension is shown in $\text{Figure} \ 10$. The hard preloaded duplex bearing and the single bearing can be clearly seen. The single bearing is soft preloaded by a bush-spring. Both preloads (hard and soft) are designed to be ~50N. This bush-spring is manufactured from SS using Wire Electro Discharge Machining (Wire EDM). A nut at the left provides rigid connection of all the three inner bearing rings. All three bearings are mounted with a SS bush. The flange (at the right) is mounted on the aluminium housing. This flange provides both a rigid I/F with the aluminium housing and (from thermo-mechanical point of view) acts as a force frame to minimize the radial shrinkage locally at the bearings.
The complete drive train is shown in Figure 11. The drive motor is chosen to be a Phytron stepper motor (Ø52mm, see Figure 12). It is designed to be cold redundant (i.e. two sets of coils are implemented which are mechanically separated). The I/F flange of the motor is placed as close as possible to the CoG of the motor to make it more robust for launch loads. A flexible coupling is shown which connects the motor shaft to the carousel shaft.

An opto-coupler is a C-frame shaped part with a light source and a light sensitive receiver. Dedicated vanes of the carousel (see the cover plate in Figure 9 and Figure 13) pass through this C-frame providing a ‘dark’ or ‘light’ signal. By implementing a form of grey-code in these flanges the system is capable of identifying each of the four unique positions.

Mechanical limit switches are applied to be able to separately identify the mechanism dark or closed position. These are implemented to allow the space craft (S/C) to separately detect correct functionality and safe positions of the mechanism (see Figure 14). The levers follow a cam profile that is fixed (rotation locked by a parallel key) to the carousel shaft. The dark or closed position covers a significant angular range and therefore does not require and accuracy positioning. These switches are implemented redundant.
6. LTM PROGRAMME

In order to demonstrate that the CMA mechanism will survive the environmental loads and will show no performance degradation, a Life Test Model (LTM) has been built and tested (Figure 15). A short description of LTM test programme is as follows:

- Characterisation on components during LTM CMA assembly
- Ground testing
- Vibration testing
- TV testing
- Life cycles
- Destructive analyses

6.1. Mechanism characterisation

Prior to the actual LTM testing, a characterisation on the mechanism components has been performed as a part of the LTM CMA assembly. Several measurements took place during the assembly and integration process of the LTM. Two tests characterize the functioning of the mechanism:

1. A running torque test was performed on the bearing assembly
2. An optical test was performed to assess the angular positioning reproducibility.

A photograph of the setup is shown in Figure 16. On the left, the bearing assembly is mounted on a rigid block. The shaft is driven by a DC motor (at the right). In between a torque measurement device is installed. An optical encoder is used (not shown) to demonstrate the angular reproducibility.

The torque measured with the setup (~5 Nmm peak-to-peak) is equal to within 10% of the torque measured by ESTL. The angular reproducibility is measured to be ~0.05° which is well within the budget of ~0.20°.

6.2. Ground testing

After the mechanism characterisation, the LTM is fully integrated. The next step was to perform the ground testing. A number of 16,000 cycles have successfully been carried out in dry nitrogen (GN₂). The GN₂ is representative for the PFM and is required due to degradation of the MoS₂ coatings (of the bearings) in (humid) air. The starting current, which is used as a sanity check of the mechanism, is measured at every 4000 cycles and was <120 mA which is well within the requirement of < 280 mA.

6.3. Random vibration testing

The random vibration loads that the LTM has to withstand are 21, 19 and 14 grms for the X, Y and Z axis respectively. The duration is 6 minutes for each axis. This covers the total accumulated life time of unit test, instrument test, S/C level test and launch. The vibration campaign initiated with low level sine and -12 dB random runs. Based on these runs both model correlation and 0 dB predictions could be made. Unfortunately, for the initial design, a relatively poor correlation was observed. Figure 17 shows both the modelled (green dotted line) and the measured (solid line) transmissibility of the accelerometer on the motor. The first significant eigenfrequency deviates with approx. 20 %. As a result, the predicted response levels in the mechanism were unacceptably high. This applied for the motor, the bearings and structural parts. Since notching would only help to a certain extent, it was decided to carry out a redesign.
The redesign started with correlating the FE model with the measurements in order to gain insight in the cause of the deviations in dynamic behaviour. The cause is found in a.o. contact definitions and stiffness assumptions in the FE model.

After a proper correlation, design solutions were implemented and verified by analysis. The solution was found in the application of laminated stiffeners. These stiffeners should result in both a stiffer housing and in the introduction of additional damping. A laminated stiffener consists of three plates of aluminium which are mutually adhesively bonded with EC2216. A total of three laminates are applied. Next, the laminates are adhesively bonded to the existing CMA housing (Figure 18).

With the updated mechanical design, a second vibration campaign has been carried out. The resulting correlation (approx. 5% in eigenfrequency) is seen in Figure 19. Based on this result, the vibration test has continued including full (unnotched vibration levels). The LTM successfully survived these loads.

6.4. TV testing

During the Thermal vacuum (TV) test, a total of 8 cycles over a temperature range of -30°C…+70°C were applied. At the extreme temperatures, functional checks of the CMA have been performed. The starting current at the extremes is < 120 mA. This is the same current as observed during the ground testing. This shows the a-thermal behaviour of the CMA. After visual inspection (after the TV testing) it was concluded that no visual damage occurred.

6.5. Life cycle testing

A total of 47,000 cycles have been performed in vacuum. Periodically, starting currents have been measured showing similar behaviour as observed during ground and TV testing. No degradation in performance was observed.

6.6. Destructive Physical Analyses (DPA)

After all environmental tests a DPA was performed by the suppliers of the tribological components. The results from the DPA of the motor bearings (performed by Phytron) are:

- The motor bearings show no wear debris or degradation on the raceways and the ball
- The retainer shows slight running marks of the balls, a little bit more on the rear bearings than on the front bearings

The results from the DPA of the mechanism bearings (performed by ESTL) are:

- The bearings of the mechanism show no damaged and remain in good condition; some minor marks caused by integration of the bearings into the LTM were noted.
- Although mean and peak torques levels have increased after life test, the torque measurements do not exhibit any evidence of anomalous behaviour
- Overall the bearings were in good condition and additional lifetime can be expected as
lubricant was still present on the raceways

There is no significant or unexpected wear or degradation observed during DPA. The worst case variation or degradation of the peak torque throughout the life test is less than 50% and therefore considered as acceptable. Furthermore, the motorization margin is well within the requirement. The results are compliant with [9].

7. CONCLUDING REMARKS

A dedicated mechanism (VNS LTM) has been designed, built and tested. The VNS LTM successfully passed all the tests thereby maintaining the required performance and is therefore considered qualified for the MSI program.

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