SWARM DEPLOYABLE BOOM ASSEMBLY (DBA) DEVELOPMENT OF A DEPLOYABLE MAGNETOMETER BOOM FOR THE SWARM SPACECRAFT

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ABSTRACT

The Swarm programme consists of 3 magnetically clean satellites flying in close formation designed to measure the Earth's magnetic field using 2 Magnetometers mounted on a 4.3m long deployable boom.
Deployment is initiated by releasing 3 HDRMs, once released the boom oscillates back and forth on a pair of pivots, similar to a restaurant kitchen door hinge, for around 120 seconds before coming to rest on 3 kinematic mounts which are used to provide an accurate reference location in the deployed position. Motion of the boom is damped through a combination of friction, spring hysteresis and flexing of the 120+ cables crossing the hinge. Considerable development work and accurate numerical modelling of the hinge motion was required to predict performance across a wide temperature range and ensure that during the 1st overshoot the boom did not damage itself, the harness or the spacecraft.
Due to the magnetic cleanliness requirements of the spacecraft no magnetic materials could be used in the design of the hardware.

Keywords: Deployable Boom, Cable / Harness friction, Spring Hysteresis, Kinematic Mounts, Torque Margin, Magnetic Cleanliness.

1. INTRODUCTION

The need to fit all 3 Swarm spacecraft within the circular shape of a single fairing resulted in 3 spacecraft, each with a cross sectional shape that would fit into a 120° sector of a circle. With the spacecraft basically triangular in shape the only room available for the boom in the stowed position was the small segment of the circle between the flat outward facing surface of the spacecraft and the circular fairing.

Once the spacecraft separates from the launch vehicle the 2 Magnetometers have to be deployed away from the spacecraft body in order to position the instruments as far away as possible from the spacecraft body and avoid any influence on the instruments by stray magnetic or electrical fields emanating from the spacecraft. The Swarm DBA (Deployable Boom Assembly) consisting of a 4.3m long CFRP tube and a bespoke hinge assembly, is designed to perform this function by deploying the CFRP tube plus the instruments mounted on it. Initial proposals to use a damper to control the motion were rejected once the meaning of 'magnetically clean spacecraft' was fully understood by the team - the spacecraft wasn't even allowed to use stainless steel wire thread inserts! This resulted in a design based upon a novel dual pivot hinge mechanism that relied upon bearing friction, spring hysteresis and natural damping within the harness to damp out the oscillating motion and bring the boom to a stop at the desired position.

Right from the outset it was apparent that the design of the hinge would be dominated by the 50 or so wires crossing the hinge line. When this number more than doubled to 120 cables and was also to include fibre optic lines, blanketing to protect from the corrosive effects of Atomic Oxygen plus Kapton tape wrapping and earth braiding it was clear that understanding the performance of the harness would be key to developing a design that had sufficient torque margin to meet ECSS requirements over the operational temperature range.

2. DESIGN OVERVIEW

The DBA consists of 3 main elements:

- ***The CTA*** - Composite Tube Assembly - a 4.3m long CFRP composite tube with integral Titanium end fitting and mounting points used to attach the 2 instruments. The interfaces to the HDRMs were integral to the instrument attachments.
- ***The Deployment Hinge*** - used to provide the energy required to move the CTA plus instruments from the stowed to the deployed position and accurately position the instruments relative to the spacecraft body. It also includes a system that constrains the harness in position against the launch vibration whilst not interfering with the deployment process and also preventing the harness from being damaged during the deployment.
- ***The HDRMs*** - 3 HDRMs (Hold Down Release Mechanisms) are used to support/retain the DBA in the stowed position for launch and upon command release the DBA and allow it to deploy.

This paper concentrates on the design, development and qualification of the Deployment Hinge Assembly because it was developed for the Swarm programme. However for completeness short descriptions of the CTA and the HDRMs are also provided.

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CTA and HDRMs

The CTA: The CTA was originally planned to be a constant cross-section (circular), constant wall thickness CFRP tube with brackets clamped to it to provide locations / interfaces for the instruments and the HDRMs. One instrument is mounted approximately half way along the tube and the other at the opposite end from the Hinge Mechanism. The central instrument mount has 2 HDRM interfaces and the tip instrument interface has a single HDRM interface.
In the end because of the extremely tight space and weight restrictions imposed the CTA ended up as a rectangular cross section tube with cross section and wall thickness that both tapered along the length of the tube. In addition the interface brackets for the instruments had to be integral to the tube design and therefore had to be built into the tube as the CFRP was laid up.

The HDRMs

These were a modification of a low shock, non-explosive, 'Off-the-Shelf' catalogue actuator from NEA Electronics, Inc of Chatsworth, California, USA to provide interfaces to suit the particular installation on the Swarm spacecraft.

Deployment Hinge Design

The Hinge design is based around a novel dual pivot design, similar to a restaurant kitchen door. However, instead of separate torsion springs around the 2 pivots, 3 compression springs and a cable/pulley system are employed to provide the torque necessary to pull the DBA towards the top dead centre (TDC) or deployed position.

The following 2 diagrams show the hinge - minus the CTA - in the deployed and stowed positions. The DBA is mounted onto the spacecraft in the deployed position and the action of moving to the stowed position compresses the springs by wrapping an anchored cable around the primary drum. The stowed position is approximately 157º from TDC, equivalent to a spring compression of approximately 110mm, which together with the deployed pretension of 56mm generated a starting torque of 57Nm around the primary pivot from the 3 springs.

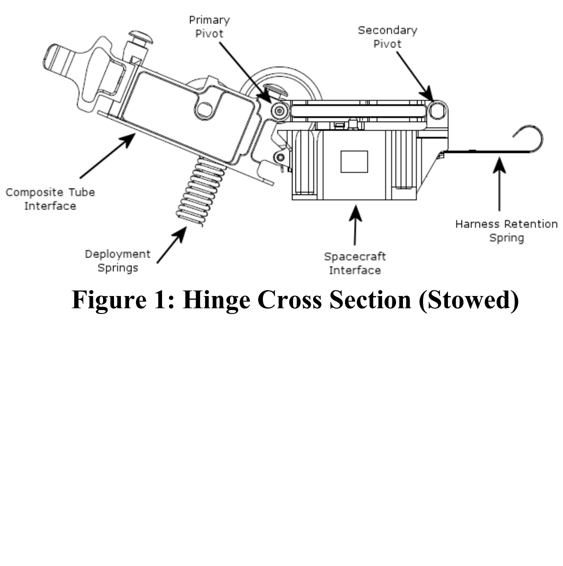


Figure 1: Hinge Cross Section (Stowed)

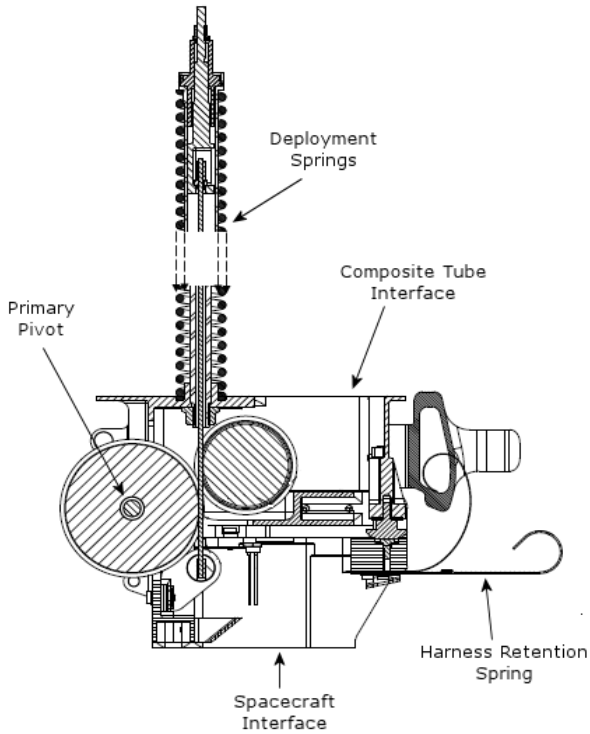


Figure 2: Hinge (TDC/Deployed position)

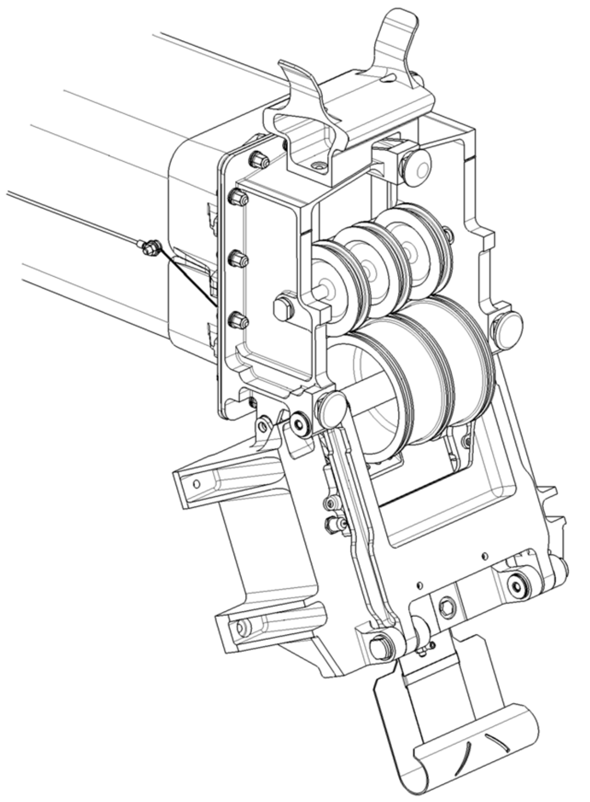


Figure 3: 3D View of Hinge in Stowed Configuration

Releasing the DBA allows the force generated in the compressed spring to rotate the DBA towards the TDC position. In the process of doing so, energy is transferred from the compressed springs into momentum in the moving DBA. At the TDC position rotation transfers from the primary pivot to the secondary pivot and the DBA continues towards the overshoot position. As the DBA rotates about the secondary hinge the momentum stored in the moving DBA is now transferred into recompressing the springs via the anchor cables down the centre of the deployment springs, recompressing the springs. Because the cable anchor point is offset towards the primary pivot the overshoot angle is approximately 1/3rd of the stowed angle, thereby limiting the amount the harness is flexed during the overshoot. At the end of the overshoot the force developed in the now re-compressed springs returns the DBA back towards the TDC position, where the DBA again overshoots towards the original stowed position. This cycle of oscillations back and forth continues until the DBA comes to rest at the TDC position.

Each oscillation is smaller than the previous as energy is lost through:

- Harness resistance
- Spring hysteresis
- Bearing friction
- Spring guide tube friction
- Kinematic mount friction

The DBA oscillates for around 120seconds before coming to rest.

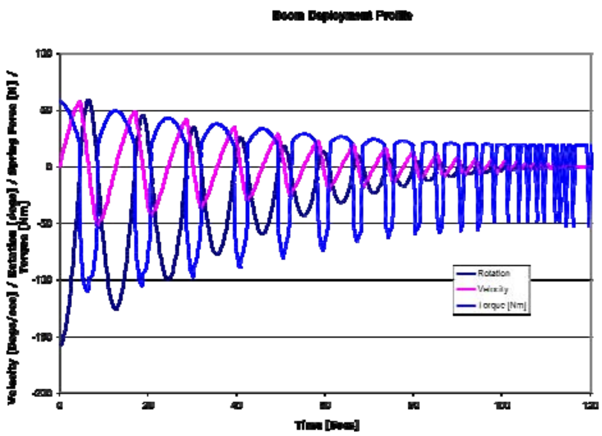


Figure 4: Deployment Profile (Prediction)

The above figure shows a prediction of the DBA deployment profile starting from a stowed position of ~157°, overshooting to ~55°and then oscillating back and forth until the DBA stops at TDC. It was produced using the VBA for Excel model that included allowances for measured values of:

- Harness resistance
- Spring hysteresis and guide friction
- Bearing friction
- K-mount friction

Harness Management System

Control of the loop of harness crossing the hinge line during both launch and deployment also proved to be a key driver for the hinge design.

Initial modelling of the dual axis system showed that with the boom stowed at an angle of 157° from the deployed position the initial overshoot would be around 55º. Therefore the harness routing and management system would have to cope with this wide dynamic angular range whilst also ensuring proper support during launch.

3. DEVELOPMENT PROGRAMME

Having settled on a design based around a stored energy, naturally damped, dual pivot design it was clear that understanding the performance of the harness as the boom oscillated back and forth between the stowed and overshoot positions, was key to correctly sizing the springs required to provide the energy for deployment. A follow on from this was to design an effective means of restraining the harness bundle in the stowed position against the launch vibration. This restraint system not only had to restrain the harness in the stowed position it had to be designed such that it would not interfere with the natural flexing of the harness as the boom oscillated back and forth during the deployment sequence. In order to meet these 2 objectives a development programme was instigated to a) for the predicted operating temperature range measure the resistance of the harness as it flexed and b) by testing a representative length of cable on the vibration facility, develop a system that would properly support the cable during launch vibration.

To support this work, especially understanding the impact of the harness resistance, a mathematical model of the deployment sequence was developed using VBA in Excel. This model proved to be very useful throughout the programme and accurately predicted the deployment profile of the final mechanism.

Harness Sample

The starting point for this development work was to produce a sample harness that realistically represented the flight harness in terms of the number of wires, their size and their flexibility. There was a risk that the final harness could be substantially different from the test sample but progress had to be made on the design using the available knowledge otherwise the DBA would end up on the critical path and drive the programme. An outcome of this work would be to develop an understanding of the parameters that would affect the stiffness of the assembled bundle of wires and cables and then define back to the prime team how the harness bundle should be configured to minimise its impact upon the overall deployment performance.

Harness Thermal Performance

Having decided that harness flexibility would be a key driver in sizing the deployment springs, determining the stiffness variation across the operating temperature range was the first key development task. A simplified test rig was produced with the correct geometry and with the harness attached a series of tests were performed to measure the harness resistance in both directions at ambient and hot and cold extremes. The following graphs show the results of those tests:

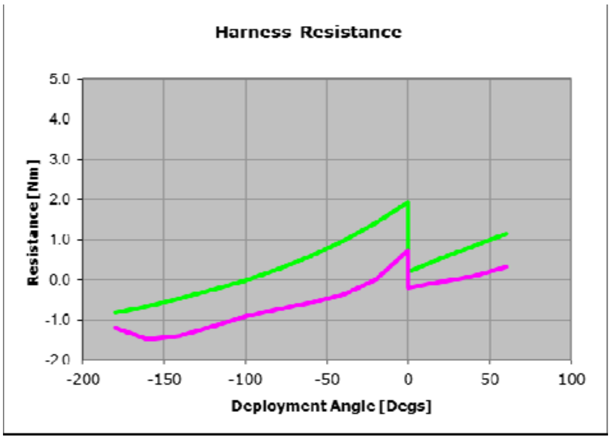


Figure 5: Harness Resistance (Hot)

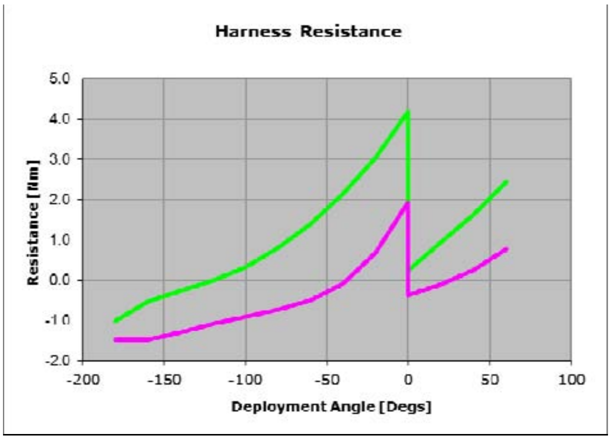


Figure 6: Harness Resistance (Cold)

For each graph the green curve is from the stowed to the TDC position, or zero degrees, and then on to the over shoot position - the cerise curve is the return stroke. For the math model is was assumed that the curve or loop would still be followed as the overshoot reduced with each successive swing and not some intermediate lower value.

Although not shown there is little difference between ambient (21°C) and hot (+60ºC), but there is as expected around a 2-fold increase at cold (-60°C) conditions. The other point worthy of note is that the harness resistance, as the hinge pivots around the primary hinge, initially is very low, negative in fact, which means that it was assisting deployment, becoming positive, opposing deployment as the DBA approaches the TDC position. As the hinge changes from pivoting about the primary hinge to pivoting about the secondary hinge the harness resistance sees initially a step change, dropping to around zero and then builds up steadily as the boom approaches the maximum overshoot position. A similar, but with reduced value, profile is recorded for the return stoke with peak values around half of those seen on the forward stroke. The difference between these 2 curves represents the energy lost or absorbed in flexing the harness as the boom oscillates back and forth.

With the peak harness resistance value occurring at TDC, when the deployment torque is reducing to a minimum value, sizing the springs to accommodate this profile proved to be taxing. As this also means that the maximum deployment torque is available when the harness resistance is at its lowest there is a considerable excess in energy at the start of the stroke that is translated into stored momentum in the rotating boom. The overshoot resistance profile, with increasing resistance with increasing deployment angle resulted in a design which naturally absorbed this excess energy, minimising the overshoot and hence the amount of flexing the harness had to endure. As mentioned the radius from the anchor cable to the primary pivot was around 1/3rd of the equivalent distance to the secondary pivot, this ratio effectively restricted the 1st overshoot angle to around 1/3rd of the stowed angle.

The harness tests in the thermal chamber were performed a number of times to test different harness routing profiles and also to investigate the effects of changes as the harness definition evolved through the development programme.

Figure 7: Harness Resistance Test Rig

The above photo shows a view from above of the harness bundle, in the Beta Cloth sheath, and of the simplified hinge test rig.

Initially the harness only consisted of 50 cables, which quickly grew to around 120 cables but this value and the harness build up, varied for several months before stabilising. In addition a Beta Cloth outer layer was added late in the development programme and a BeCu spring - as part of the Harness Management System - was added following a series of development test performed on the vibration table. All of these changes had to be characterised to ensure that the ECSS rules of deployment margin could be maintained.

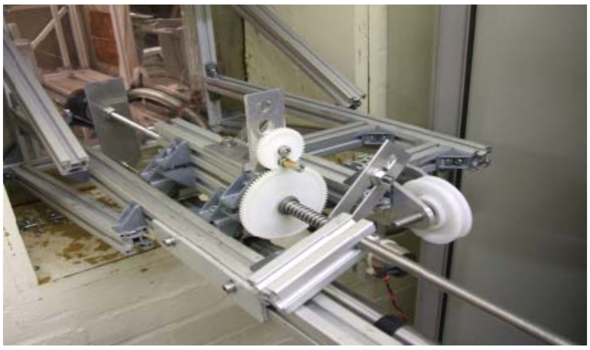


Figure 8: Harness Resistance Test Rig (Outside Test Chamber)

This photo shows the load cell and LVDT used to monitor the harness movement and resistance from outside of the test chamber. A system of pulleys and weights was used to bias the system to one side to ensure that the load cell was always positively loaded.

Spring Hysteresis and Spring Guide Friction

Three compression springs in parallel hidden within the CFRP tube were used to provide the energy to deploy the DBA and then once it had stopped oscillating and come to rest at the TDC position provided the force necessary to ensure that the DBA registered positively in the 3 K-mounts between the fixed and moveable brackets that make up the hinge. With a free length of over 470mm the springs exceeded the acceptable ratio for buckling and internal guides had to be provided to ensure that the springs would not buckle.

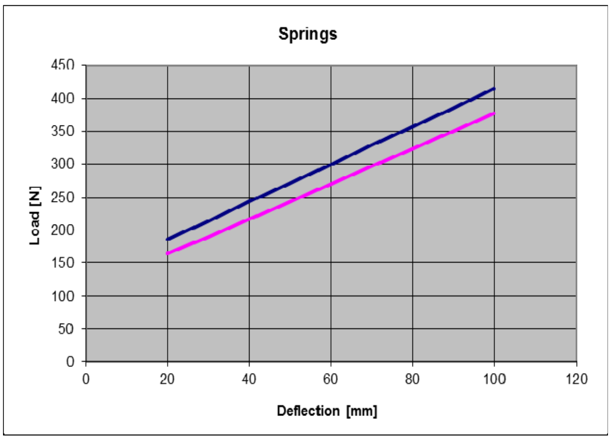


Figure 9: Spring Hysteresis Plot

Using a development spring and EM versions of the proposed spring guides the friction of the guides and the spring hysteresis were both measured using a tensile test machine.

The previous plot shows the results of these tests. These values were incorporated into the Excel VBA model as a change in spring stiffness (slope) and intercept as the DBA motion changed from compressing to expanding the springs.

Bearing Friction

The primary and secondary pivots consisted of plain shafts running in plan bearings, consisting of an outer metallic shell lined with a low friction sleeves; these we fitted in both the fixed and moving bracket halves to ensure redundancy in the event of seizure.

For the primary and secondary pivots a series of tests were performed to measure the stiction and dynamic friction and its variation with temperature and load. Again the data from these tests was added to the Excel VBA model to improve its overall fidelity.

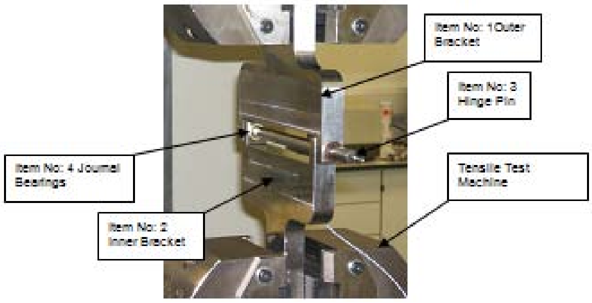


Figure 10: Bearing Friction Test

The test sample was loaded into a tensile test machine and with different loads applied across the hinge the torque required to rotate the hinge pin was measured. Because of the redundant liners the test measured the worst case highest friction whereas in operation the hinge will rotate on the surface with lowest friction - using the test results in the deployment analysis therefore gives a small positive margin to our advantage.

The tests were performed at hot, ambient and cold conditions to determine the full variation of friction in the hinge elements.

The following 2 plots show the variation between bearing friction at hot (+60°C) and cold (-60°C). As with the harness bundle the variation between ambient and hot is small. The plots also show how the bearing friction coefficient is lowest when the bearing loading is highest.

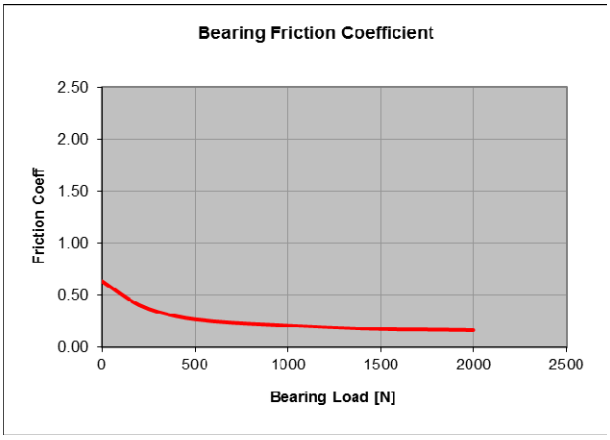


Figure 11: Bearing Friction (Hot)

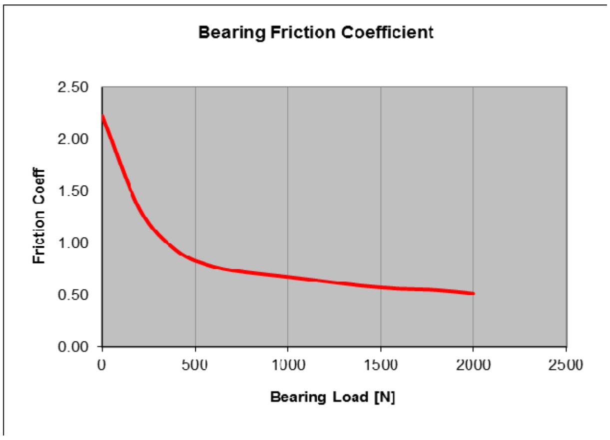


Figure 12: Bearing Friction (Cold)

This is to our advantage as the harness resistance is lowest when the spring compression/force is at its highest. As the deployment springs are under compression at the TDC position - equivalent to around 480 N (total for 3 springs) - the hinge design does not enter into the region of a rapidly increasing friction coefficient at the start of the plots.

Harness Management System

With the Boom rotated through approximately 157º when stowed we were presented with a large loop of harness that had to supported during launch to make sure it didn't move into an unacceptable position as a result of the launch vibration. Using the same fixture that we had employed to measure the harness resistance in the thermal chamber we configured the assembly on the vibrator and mocked-up a collection of guides.

The following figure shows the hinge and harness in the stowed configuration. The large loop of harness running from out under one side of the fixed bracket to the first tie down position on the side of the CFRP tube can clearly be seen. Initial tests on the vibrator identified the need for lateral guides to prevent the harness from moving sideways and a 'Harness Guide' - identified in the above figure - was added. The 'Harness Guide' consisted simply of 2 guide fingers, one either side of the harness that prevented it from moving under lateral vibration. The guides can also be seen more clearly in Figure 3 above.

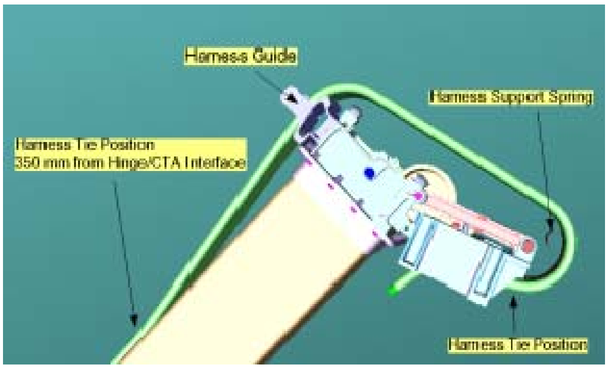


Figure 13: Harness Routing (Stowed)

Tests in the other axes showed that these fingers were not sufficiently long to ensure that the harness remained within their envelope and with restrictions of the available space under the fairing we introduced a cantilever BeCu spring - Harness Support Spring in Figure 13 - that put the harness under tension in the stowed configuration. This was found to be sufficient to ensure that the harness stayed in place during vibration testing.

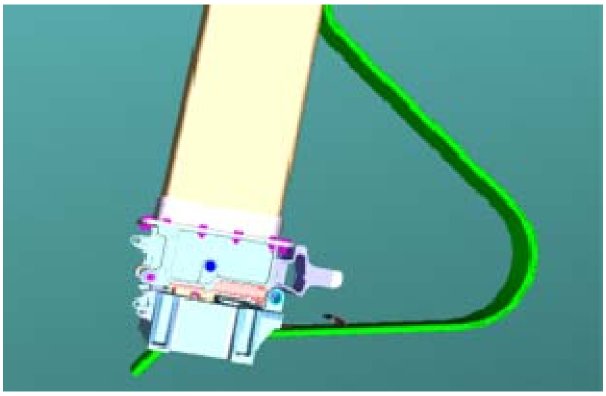


Figure 14: Harness Routing (Deployed)

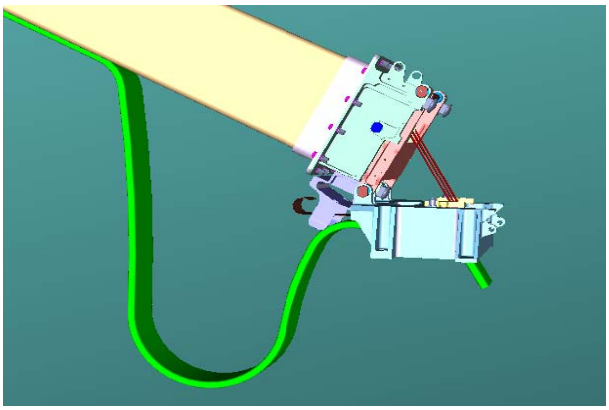


Figure 15: Harness Routing (Overshoot)

A series of deployment tests at full DBA level showed that the spring and guide did not interfere with the deployment of the DBA and the harness exited and entered the guides without damaging the harness.

Because of the addition of the BeCu spring to tension the harness we had to revisit the tests performed in the thermal chamber to determine the impact, if any, on the deployment profile of the hinge.

The BeCu spring force was determined by comparing test results with and without the spring.

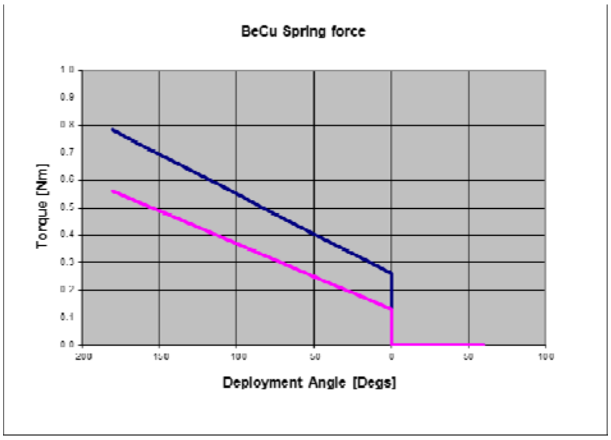


Figure 16: BeCu Spring force (Cold)

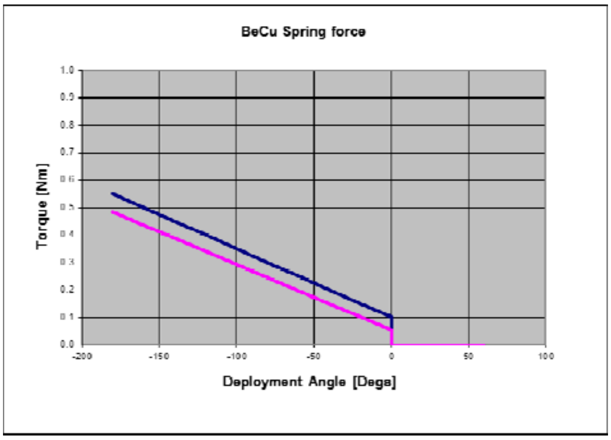


Figure 17: BeCu Spring force (Hot)

Data was only computed for the first part of the deployment sequence as the spring plays no part once the DBA is past TDC.

Kinematic Mounts

The Titanium K-mounts were originally coated with Apticote 450, but this was found not to be robust enough to survive numerous deployments and was eventually replaced with Keronite. Through a series of trials this was found capable of surviving around 10 deployment cycles. Since ECSS requires factorising ground cycles by 4 and in orbit by 10 this removes any possibility of ground testing. Therefore, instead of looking for a better coating it was decided to replace the concerned parts after final ground testing by new parts so that the applied factor of 10 related solely to the single in-orbit deployment.

4. QUALIFICATION/FLIGHT PROGRAMME

The DBA underwent a full qualification programme including a life test consisting of 74 deployments plus an additional 65 deployments for the K-Mount surface coating. The qualification vibration test was particularly difficult as the 4.3m long DBA required an equally long test rig which had to be very deep to achieve the necessary stiffness. The resultant test set-up was at the limit of the vibrator capacity preventing us from performing a random level vibration and instead relying upon the sine and quasi-static vibration to qualify the DBA for the launch vibration.

Because of the size of the overall DBA thermal qualification testing was broken down and the Hinge, CFRP tube and HDRMs were all qualified individually at the required temperatures.

Post life test the QM hinge was dismantled and visually inspected and found to be in very good condition overall. During this inspection it was noted that there was evidence of contact between the pin of the primary axis and the housing in an area where there wasn't a plain bearing lining the bore. This was traced to an underestimation of the load applied to the pin and hence the amount of deflection that was occurring. The DBA had performed nominally throughout all the life test deployments and there was no evidence of degraded performance. This was due to the redundant design of the hinge where bearing liners were provided in all elements of the hinge, so that should any part of the hinge seize then it would still be free to rotate within the unaffected part of the hinge. Nevertheless a modification was performed on the flight hinges to remove the problem.

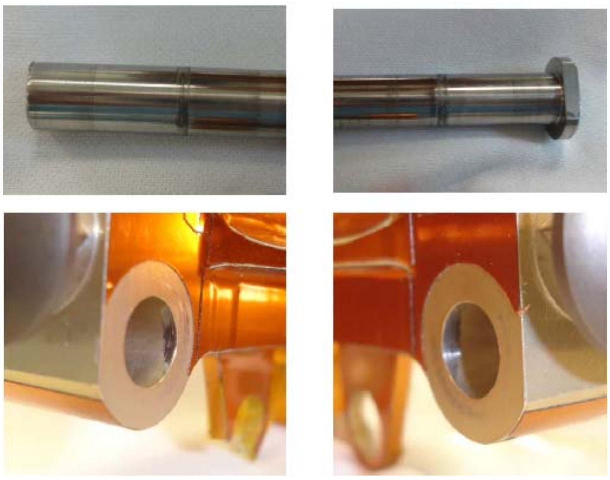


Figure 18: QM Post Life Test Inspection

For the flight units a test was devised at hinge equipment level to ensure the build quality of the hinge at the supplier's premises prior to integration into the DBA. The test consisted of a short boom fitted with a small inertia and mounted on a pedestal such that the short boom rotated in the vertical plain and finished with the DBA at TDC in the vertical direction. This removed the need for an offload rig, keeping the cost down, and allowed the Excel VBA model to be adapted to represent the particular configuration, allow for the influence of gravity and then to predict results under these conditions. Without an offload rig there could be no argument about it influencing the test results.

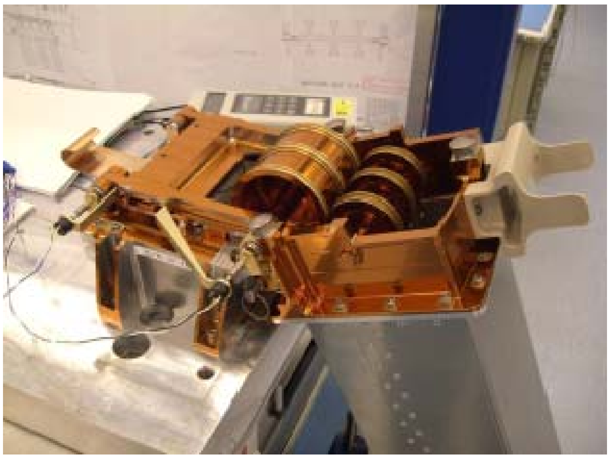


Figure 19: Flight Hinge Unit Deployment Test

Note the small potentiometers attached to the primary and secondary pivots to record motion.

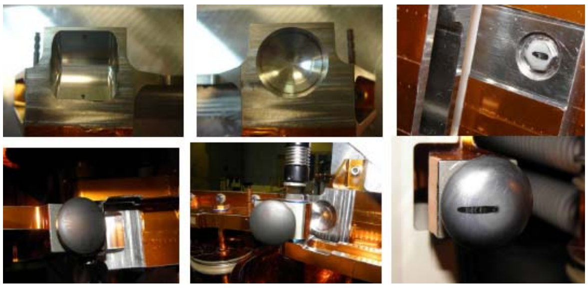
5. LESSONS LEARNT

A number of problems were encountered during the Swarm DBA programme.

a) ***Spring Manufacturer*** - Because of the restriction on the use of steel and iron on the Swarm spacecraft we were directed into using Titanium springs. The Hinge manufacturer selected a reputable spring manufacturer, who unfortunately had limited experience in the manufacture of Titanium springs. As a result it was noticed that the springs were relaxing as the testing progressed and performance was declining. **Recommendation:** When choosing a spring manufacturer, ensure that they have experience in the selected material and specifically in the heat treatment process.

b) ***Spring Cable Anchor*** - The deployment springs are compressed by the action of moving the DBA from TDC to a position away from TDC. This is achieved via a cable through the springs to the fixed bracket. The end of the cable in the fixed bracket passes through a hole in a shaft and is terminated in a crimped ferrule that prevents the cable returning back through the hole. During the overshoot phase of the deployment this shaft has to rotate to match the changing line of action of the cable. During early testing it was discovered that the shaft wasn't rotating and the cable was flexing at the exit from the hole in the shaft. This was cured by recessing the hole deeper into the shaft, moving the ferrule contact face away from the centre line of the shaft thereby generating sufficient torque to ensure that the shaft rotated with the changing line of action of the cable. **Recommendation:** Always ensure that sufficient actuation margin is provided, even for the smallest detail.

c) ***K-Mounts Wear*** - The K-mounts are used to register the 2 bracket halves at TDC and ensure the required deployment accuracy is achieved. The original design used Apticote 450 as the K-Mounts surface treatment. After some trial and error a protective coating of a dry lubricant; Keronite, was found to be an acceptable solution. However with an overshoot angle of 55° it was always going to be difficult to prevent wear between these sliding surfaces. The following photos show the wear recorded on the QM K-mounts after completion of the life test



From these it can be seen that the one closest to the secondary pivot, which was a ball on flat surface, had the worst wear - the single wear mark evident in the bottom right hand image. The ECSS for mechanisms states that sliding surfaces should be avoided, this is not always possible.
Recommendation: Ensure that the surface treatment for sliding surfaces, especially when under high Hertzian pressure, is thoroughly researched and early testing is introduced to develop alternatives.

d) ***Primary Shaft Wear*** - This was the result of a missing assessment of shaft deformation under load. Fortunately. Fortunately the practise of providing redundant bearing surfaces in all hinge elements offset the potential for this to have become a single point failure.
Recommendation: More intensive review of deflections vs. load and motions during the review process should have identified the shortcoming of the design and avoided the error.

6. SWARM SPACECRAFT

Swarm is due to be launched from Plesetsk using a Rockot launcher in Dec 2013.