MULTI-SCREW FOCUSING MECHANISM FOR DMC3 SUB-1M IMAGER

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ABSTRACT

This paper describes the focus mechanism for Surrey Satellite Technology Ltd.'s (SSTL) SSTL300 platform based sub-1m (SSTL300-S1) resolution optical imager. SSTL's distinctive strategy of qualifying COTS components is employed in the design, utilizing heritage components from other mechanisms and judiciously integrating new COTS components. This approach will allow the focus mechanism to be conceived, designed, built and qualified within 18 months and cost effectively. The robust nature of the solution means that this design can be applied to imagers across a wide variety of vibration and thermal profiles.

The resulting focus mechanism design employs a stepper motor, planetary gearbox, and ball screw/ball nut to linearly position the 500g focusing lens. The lens is mounted in a carriage with three integral bushing shafts which travel along six linear ball bushings. The novel aspect of the design is the use of three synchronized ball-screws to support and simultaneously drive the carriage.

1. INTRODUCTION

1.1. SSTL Corporate Philosophy/Design Approach

SSTL remains a pioneer in affordable small-satellite technologies; giving customers access to space for Earth Observation (EO) or Science Missions, who may otherwise be unable to afford such advanced capability. SSTL is able to deliver high-performance systems at low-cost through its unique design philosophy. Essentially, the SSTL philosophy for new product developments involves design with large margins allowing only limited analysis to be suitable, early test programmes to commence, and thorough qualification to take place, whilst responding rapidly to lessons learnt.

Where possible, SSTL engineers strive to use Commercial off the Shelf (COTS) components, which are qualified through intensive testing by SSTL to give high confidence for their use in the space environment. Inevitably certain parts are too complex and critical for COTS components to be suitable; in which case before accepting to use expensive space-rated parts, SSTL encourages performing small in-house modifications to COTS components to validate them for the space environment. This allows cost-effective options to remain: this strategy is demonstrated in the S1 Focus Mechanism. It is through SSTL's engineering experience and appreciation of historical lessons learnt, that such high reliability is achieved at comparatively low-cost. The corporate structure of SSTL with substantial horizontal and vertical integration allows development programs to remain rapid in timescales.

1.2. Application

The SSTL300-S1 imager is the next evolution from the 2.5m resolution imager flown on Nigeria-Sat2. In both imagers on-orbit focusing is performed by adjusting the position of a focus lens linearly along the lens optical axis.

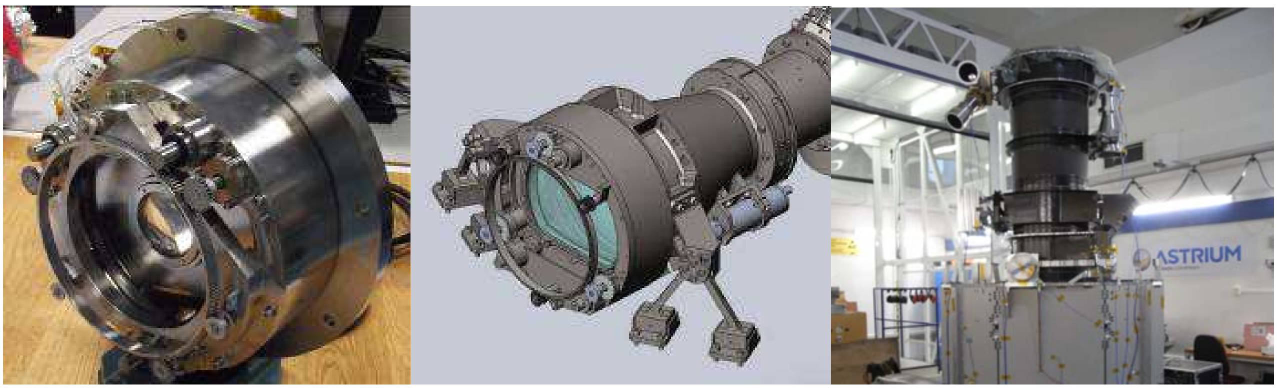

Figure 1. DMC3 Imager Focus Mechanism EM (L),PFM Model with RLA Assembly (C), and RLA Mass Dummy on SQM Imager at top right (R)

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The S1 Imager has more stringent requirements for the focus lens' alignment, linear position, and stability, as well as a more challenging environment and very different location on the imager. This required development and qualification of a new focus mechanism design integrated into one of the optical sub-assemblies on the imager, the Relay Lens Assembly (RLA), a location depicted in Figure 1.

The focusing lens is the final lens on the RLA along the optical path. This configuration was traded off against other locations, including at the focal plane; however, the logistical challenges and criticality of the thermal solution proved more onerous than the mechanical environment at the RLA. As a result the focus mechanism is fully integrated into the RLA.

1.3. Design Drivers

The fundamental purpose of the focus mechanism is to position the focusing lens along the optical axis. In order to ensure that the earth image is focused with the focal plane's required depth of focus, the lens must be positioned within 5µm. Due to various factors affecting the focal length of the imager, it must be able to position the lens along a 10mm length of travel. The required life and total travel are small, driven mostly by ground cycles.

In doing so, the lens must not be perturbed in any other degree of freedom, either during focusing or due to test, launch, micro-vibration, or thermo-elastic loads. The parallel shift of the lens optical axis (referred to as decentration) must be within 5µm of the nominal position. The tilt of the lens optical axis must be within 15 arc-seconds of the nominal orientation – also expressed as 10µm Total Indicated Run-out on the lens surface.

The lens must also be able to be installed after the mechanism has already been qualified; the sequence of assembly has to consider the large size of the lens (148mm diameter).

Requirements	
Mass	**<5kg**
Linear Position Resolution	**5 µm**
Sensor Accuracy	**+/- 2µm**
Linear Travel	**10mm**
Optical Alignment	**10µm Decentration (dia) 10µm TIR (15 arc-sec) Tilt**
Design Load	**60g axial, 30g transverse**
Temp. Range	**-20°C to +50°C**

The design load used for the mechanism is 60g quasi-static in the optical axis, and 30g in the transverse axes. Also, the mechanism must provide independent feedback on focusing movement, not require launch locks, and hold the focus position while unpowered.

2. ENGINEERING MODEL CONCEPT

The mechanism concept settled upon for an engineering model utilizes a stepper motor-driven ball screw to drive the lens carriage along three linear ball bushing shafts. The design allowed two different configurations to be tested: a single screw configuration which drives the carriage at one edge of the lens, and a multi-screw configuration which drives the carriage at three points roughly equally spaced around the lens. The motion of the three screws is synchronized by a three pinion and ring gear set. The EM mechanism is built largely to a flight standard, but using COTS tribological components and eliminating costly custom modifications allowed the EM to be assembled at a low cost.

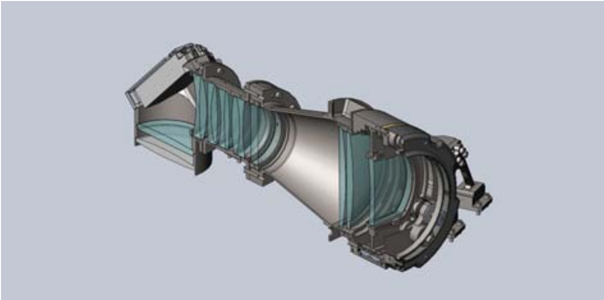

Figure 2. RLA with Optics Model

2.1. Actuator

As a result of the requirement to position the lens with a high resolution and to do so over a comparatively large distance, concepts from other focus and linear positioning mechanisms quickly fell by the wayside. In order to avoid windup, low friction and high stiffness along the line of action are required. For this reason a 440C stainless steel ball screw with a 2mm lead pitch was selected. The accompanying ball nut has an integral thread that easily clamps on to the lens carriage. In keeping with SSTL's ethos, both the ball screw and its supporting spindle bearings are COTS components, with the ball screw undergoing post-machining and lubrication under the direction of SSTL.

To drive the ball screw, a stepper motor with an integrated planetary gearbox is coupled to the lead screw using a beam coupling. The stepper/planetary is a commercial vacuum-rated motor gearbox of the same product line as those which SSTL has successfully utilized on its line of Antenna Pointing Mechanisms. Alterations to adapt the motor/gearboxes to withstand launch loads are well understood and undertaken for flight motors on this mechanism.

2.2. Linear Motion

Ideally, an application for a high resolution linear positioning mechanism would utilize flexures to eliminate hysteresis and friction. Unfortunately, the length of travel requirement discouraged their use as 10mm is approximately 5 times too large a displacement for a reliable flexure of the load capacity and stiffness required. Therefore linear rails were the preferred solution.

Typically in space applications profile rail is preferred due to their moment load capacity and the ability to preload them heavily. However, in our case, the length of the linear bearing carriages required extending the structure unnecessarily. Also, accommodating the mounting of profile rails and aligning them to the optical axis of the RLA would be quite difficult. If the rails were even slightly off axis, a movement of 5mm could carry the lens out of alignment and violate the 5µm decentration requirement.

As a result, 440C stainless steel linear ball bushings with Delrin retainers were selected along with 440C shafts. This allows for an end-supported configuration that is much more easily manufactured than one featuring profile rail.

2.3. Structure

The focus lens has a diameter of 148mm, and is truncated at two sides with a width of 110mm. It is first installed into a titanium lens cell, and then subsequently the cell is bonded into place in the titanium carriage.

Instead of the typical installation of the linear bushings into the carriage, the focus mechanism has the bushing shafts installed in the carriage. This is for two reasons.

Firstly, the linear bushings are designed to have two bushings along each shaft to react moment loads. Installing two bushings on the carriage would have a smaller distance between the two bushings, but would again unnecessarily extend the length of the structure, with the corresponding increase in mass.

Secondly, it is of paramount importance is that thermo-elastic distortions are minimized across the focus mechanism structure. For this reason, the main housing, end plate and carriage are all titanium and largely axi-symmetric; the other structural elements driving the position of the carriage, the bushing shafts and the bushings themselves, are 440C stainless and only a mismatch of 3.4 ppm/°C. Since the only CTE mismatch (the length of the shafts) is compensated by the linear bushings; if the shaft was mounted to the end plate and housing, it would impart a load on the structure at the operating temperature of -2.5°C which is thus avoided.

The structure is designed so that the bearing bores (for the ball screw spindle bearings) and the bushing bores can be match machined, thereby ensuring their concentricity to within a few microns. The fit of the housing and end plate is retained upon disassembly and assembly with six tapered dowels fitted before match machining takes place.

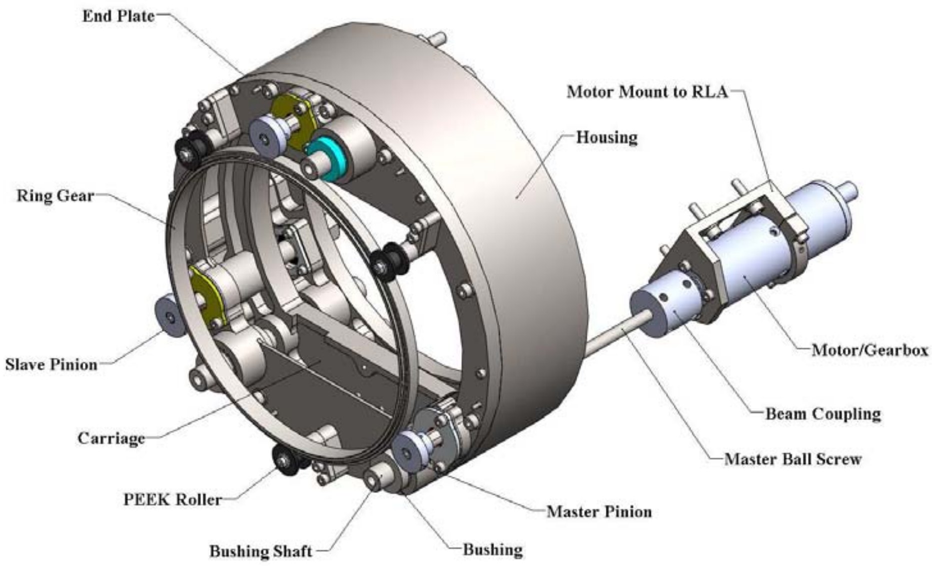

Figure 3. PFM Focus Mechanism Components

2.4. Sensor

A further requirement on the focus mechanism is that when a focusing movement is commanded, independent feedback can confirm that the movement has taken place. This had to be measured linearly; an input or output encoder on the motor/gearbox would not suffice. To that end a vacuum-rated non-contacting linear encoder was selected and implemented in a location on the mechanism expected to see <1krad TID. Selecting a vacuum-rated EEE part provided it is effectively shielded is in keeping with SSTL's COTS approach.

3. THE CASE FOR MULTIPLE SCREWS AND A GEAR SET

The initial concept comprised of the lone ball screw coupled to the motor/gearbox. Calculations and accumulated wisdom suggested that this concept would meet all of the linear positioning and optical alignment requirements, and posed little threat of thermo-elastic loads.

The chief concern was whether the first structural natural frequency would prove to be high enough. The RLA is mounted on flexures and had predicted sine loading along the optical axis of greater than 40g. If the focus mechanism's amplification factor was greater than 1.5, the design load would be exceeded. Furthermore, at frequencies below 300Hz, the focusing lens could be susceptible to micro-vibration inputs, which while small, could easily result in a shift of the optical path equivalent to a pixel at the imager detector.

FEA combined with bearing analysis yielded a small margin between the RLA and Focus Mechanism modes, but far too small for comfort. The first mode for the single screw design, a moment about the eccentric location of the ball screw, was unavoidable if one ball nut was constraining the carriage. The line of action must be outside the lens, as the clear aperture is projected along the optical axis in both directions; unless a prohibitively large counterweight is used, the carriage centre of gravity will be >74mm from the ball screw axis.

The solution was to support the lens carriage in the same fashion at multiple points, as one would do as a matter of course in a static assembly. Two identical ball screws are driven by the master ball screw via a ring gear and pinion set. Intuitively, a concept with three ball nuts fixed to the same carriage poses a great risk for one or more screws becoming unsynchronized and jamming.

Upon closer inspection, however, the limiting factor in implementing such a design would not be the threat of

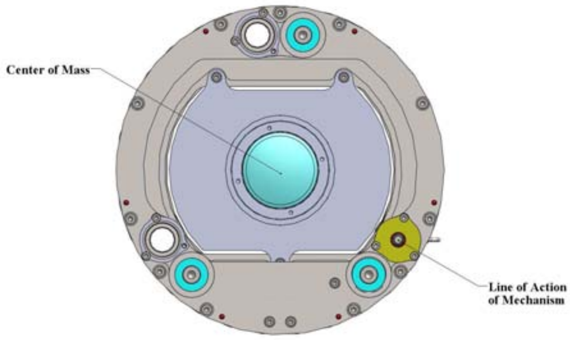

Figure 4. Single Screw Configuration – Moment Load

jamming – it would be inducing a tilt in the carriage. The tilt requirement, expressed as 10µm TIR or 15 arc-seconds, could also be expressed as a distance, or axial misalignment between two ball nuts. The distance from the ball nuts to the optical axis is approximately 20% greater than the radius of the lens, and so a 12µm axial misalignment would exceed the tilt requirement. The load induced by such a misalignment (if borne by only one nut, a conservative case) would be well within the operating load of the ball screw.

Based on tolerance analysis of the bearing bore locations, component run-out, and particularly gear transmission and thermal error, the multi-screw configuration was a viable concept. The similarity of the two concepts meant that an Engineering Model could be manufactured that could be configured and tested in the single screw and multi-screw variants using the same basic hardware only excluding the ring gear, pinions, and driven ball screws in the case of the former.

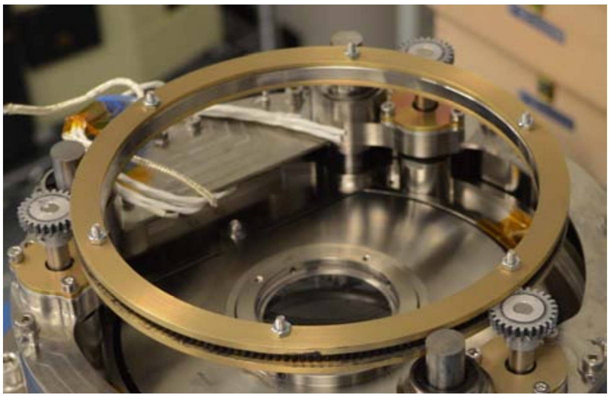

Figure 5. Gear and Pinion Assembly

4. ENGINEERING MODEL TEST CAMPAIGN

The EM test campaign consisted of the following elements:

- Functional Test – verifying the linear positioning capabilities
- Optical Alignment Test – ensuring that the decentration and tilt are within requirements throughout the travel of the mechanism at ambient, hot and cold temperatures.
- Vibration Test – sine and random vibration tests in three axes.
- Thermal Life Test – verifying life at -7.5°C, operation and torque margins at -20°C and +50°C.
- Micro-vibration Test – <10N vibration inputs verifying no shift in frequencies at small amplitudes.

The functional test verifies the linear positioning capabilities, including the step resolution, backlash, and repeatability as well as encoder and limit switch function. Absolute accuracy is not a required test, as knowledge of the position of the focusing lens relative to the best focus position is determined based on the image quality. The linear position is determined using a 0.1µm resolution Heidenhain GSE linear encoder. In order to verify the spring-loaded probe does not preload unduly the mechanism, the non-contacting mechanism encoder is checked for correlation with the Heidenhain encoder.

The optical alignment tests are conducted on the Opticentric MOT300, an optical element metrology system which characterizes the location and orientation of optical elements relative to a reference axis; the cylindrical datum of the focus mechanism is aligned to this axis within 0.2µm using the integrated tip tilt table.

Fitted into the focus mechanism is a GSE lens with a mass dummy representative of the flight focusing lens. The Opticentric measures the location of this lens' two focal points in Cartesian coordinates. Careful consideration of these values determines the decentration of the lens with a resolution of <500nm. Based on the known focal length's of each surface of the lens, the tilt relative to the reference axis is calculated with <0.5 arc-second accuracy. These measurements were taken before and after exercising the mechanism through linear profiles encompassing the entire 10mm travel length; this was conducted at 20°C and repeated at 50°C (hot case) and 2°C (approximate operating temperature) to verify the absence of large thermo-elastic distortions.

The sine and random vibration tests were undertaken to qualification levels.

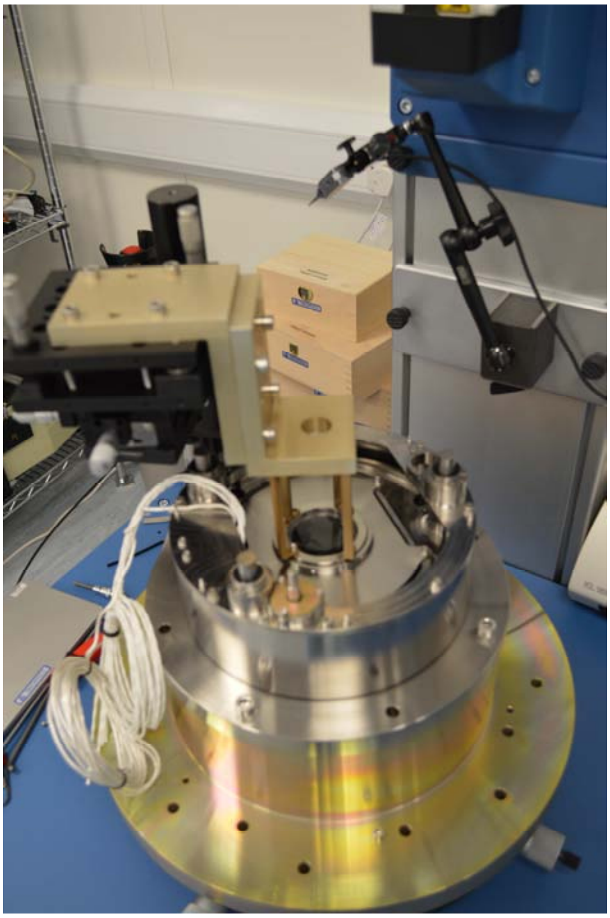

Figure 6. GSE Lens Installation with- Opticentric MOT300 with Focus Mechanism on Tip-Tilt Table

The micro-vibration "stinger test" verifies whether there is any gapping of the ball bushings or ball nut. The mechanism is subjected using a small micro-shaking voice coil (known as a "stinger") to 1, 2 and 3 N inputs in all three axes (micro-vibration amplitudes). At this level, if the mechanism is insufficiently preloaded it would see a precipitous drop in the first natural frequency, indicating a susceptibility to spacecraft micro-vibration.

4.1. The Failure of the Single Screw

The first article under test was the single screw configuration, and as expected it passed the functional and optical alignment tests with margin to spare.

During the vibration test, its behaviour was as predicted for the first two axes. During the Z axis sine test, however, where the first mode was excited to high levels for the first time, a 12g sine input was enough to exceed the design load.

In order to diagnose the reason for the frequency being lower than expected, the test was aborted, and the mechanism was subjected to micro-vibration "stinger" testing.

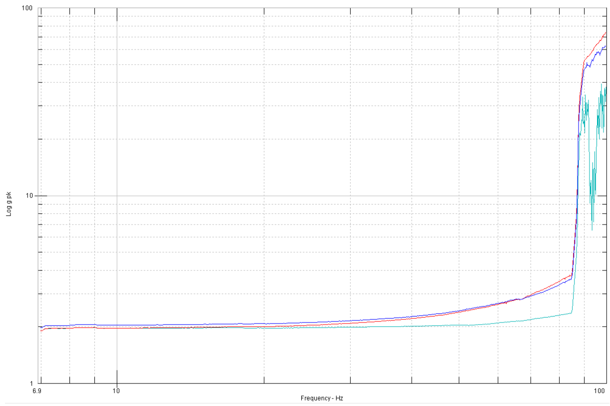

Figure 7. Plot of Z Axis (Optical Axis) Sine Response – Single Screw Variant Sees 82g, Abort!

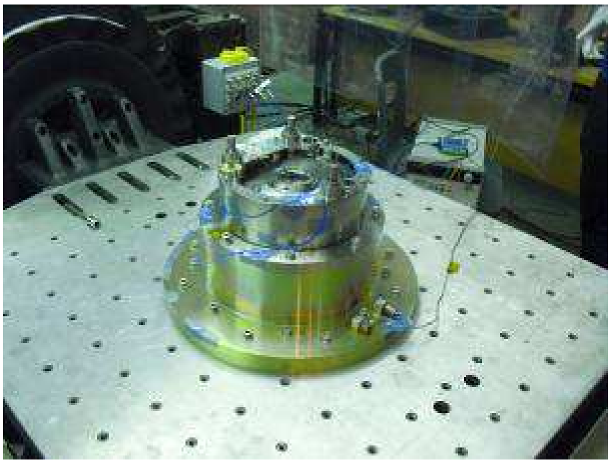

Figure 8. Vibration Test Configuration

As seen in Figure 9, the first modes in the sine vibration and micro-vibration tests are virtually identical, indicating that gapping or workmanship issues are not responsible for the lower than predicted first mode.

At this point the single screw test campaign was stopped due to the potential to exceed design load, and the multi-screw configuration was assembled.

4.2. The Multi-Screw Test – Tilt Issues

Upon assembly, the multi-screw configuration proved to be more robust than originally thought. The running torque of the mechanism at the input screw, predicted to be an order of magnitude greater than that of the single screw configuration, was only 50% greater, at approximately 48 mNm.

This trend continued with the functional test, as the minimum repeatable step increment was similar to that of the single screw, 2µm with an average error of 0.4µm.

The multi-screw configuration had its first soft failure on the optical alignment test. Although the lens decentration was within specification, the tilt exceeded the required 15 arc-seconds by 20%.

The nature of the soft failure combined with SSTL's systems engineering approach indicated that this result could possibly be tolerated. Cursory investigation into the root cause of the tilt showed quite clearly that the rotation of the driven slave ball screws was not consistently synchronized with that of the master ball screw. Moreover, this was most likely due to the ring gear oscillating eccentrically about the optical axis since

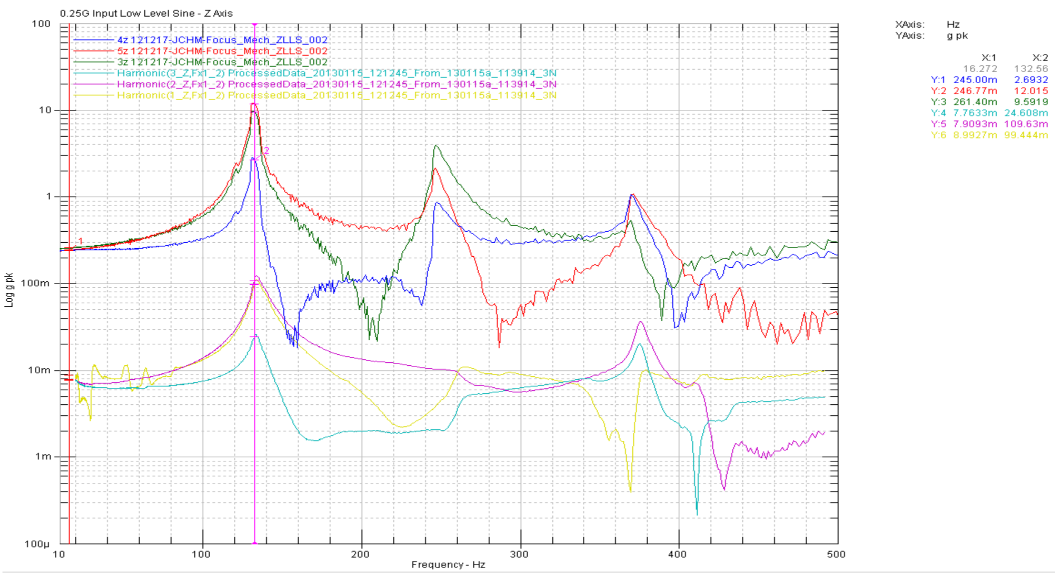

Figure 9. Low Level Sine Sweep and Stinger Responses – First Modes Coincide

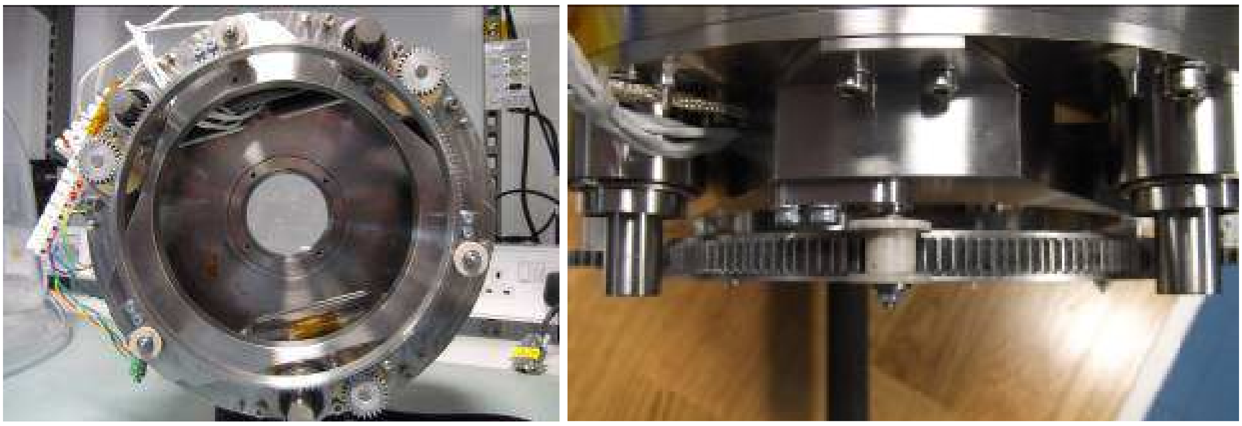

Figure 10. Guide Roller Module Installed on EM Focus Mechanism

it was not constrained concentrically. As a result, a retrofit of guide rollers would be introduced, but testing would continue with the multi-screw configuration as it was.

Vibration Testing was repeated, and the multi-screw variant passed with flying colours, with a first mode of greater than 450 Hz.

A life test was conducted of 800 cycles of 10mm in each direction for a total travel of 16000mm – a scatter factor of four on the expected mechanism life including ground cycles. The test was conducted at ambient pressure and -7.5°C. At the conclusion of the test, torque margin tests were conducted at -20°C and 50°C to verify the health of the mechanism. The results showed no degradation of the mechanism over the comparatively short life test.

4.3. Implementation of Guide Rollers

In order to restrain the ring gear and to ensure that it rotated about a fixed axis, three guide rollers were added to the multi-screw configuration (and a new gear with roller-interfacing shoulders was ordered). The rollers bear on a controlled reference surface of the ring gear to ensure the ring gear axis remains fixed, and hence the gear tooth engagement is consistent at each of the pinions at all times.

Following the implementation of this fix, the optical alignment testing was conducted again. As seen in Figure 11, the results were now well within the requirement of 15 arc-seconds, with the lens showing a maximum tilt from nominal of approximately 9 arc-seconds. Testing was repeated for the hot and cold cases, and similar results were obtained.

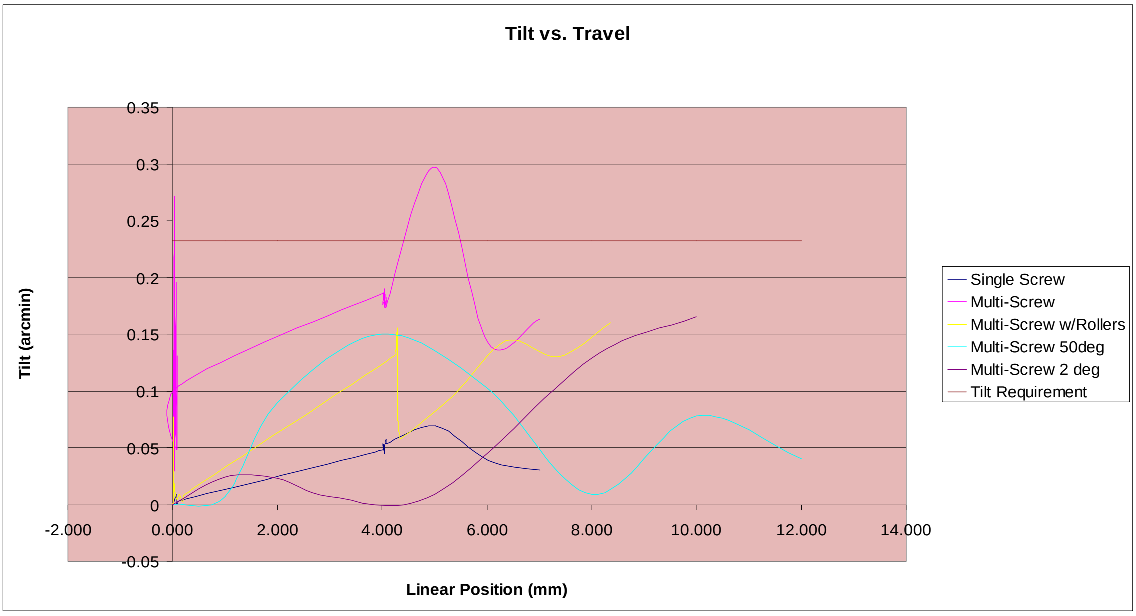

Figure 11. Focusing Lens Tilt vs. Elapsed Distance

5. IMPLICATIONS FOR FLIGHT MODEL

After undergoing a longer than expected EM test campaign, the mechanism performed either better or equal to its initial performance with respect to alignment, resolution, and torque margin. Despite misgivings about its robustness, the multi-screw proved to be a resilient mechanism capable of withstanding much greater loads than expected on S1. The gear transmission synchronizing the ball screws is the critical element in ensuring that the most difficult requirement, the tilt restriction, is met. In particular, the quality of the ball screw/ball nut, the most expensive COTS mechanical component, was instrumental in reducing running friction and compensating for misalignments in the mechanism.

Efforts made to stiffen the mechanism to increase its resistance to external and internal disturbances were made during the design of the EM, initially with a view to making the single screw a viable solution. These efforts were relatively fruitless – minimizing internal disturbances and moment loads proved far more effective.

With these conclusions in mind, the multi-screw mechanism has been put into production for flight models with small design changes, put in place to further improve the gear set:

- Anti-Backlash Pinions – as the running torque of the mechanism is a relatively low 48mNm, a relatively low spring force on the anti-backlash pinions can function effectively and decrease the transmission error between driven pinion and slave pinion.
- Ring Gear/Pinion Module – the module of the gear set was reduced from 0.8 to 0.5 with similar geometry. This increases the gear contact ratio and decreases the likelihood that a single flank gear error causes tilt in the focus lens.

In addition, the flight stepper motor has been wired in cold redundant fashion. It can be driven with either set of windings, or both sets can be combined if boost torque is necessary. This fits in with the rest of the design philosophy for the mechanism in that mechanics are considered to have a reliability of 1 (hence 1 housing, 1 gear etc), but the electronics have redundancy (motors and encoders).

6. CONCLUSIONS

Despite several issues arising during test, these have been mitigated and the multi-screw focus mechanism is a viable, cost-effective solution for a demanding requirement. Through careful component selection and applying high performing parts where absolutely necessary, high performance has been married with a cost-effective COTS methodology.