CONTRAVES OPTICAL TERMINAL – COARSE POINTING ASSEMBLY (CPA)

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ABSTRACT

The Contraves Coarse Pointing Assembly (CPA) is in essence a two axes pointing system. It comprises of a gimballed mirror driven by independent Elevation (ESU) & Azimuth (ASU) Scanning Units. Stability is the keyword for the design of the CPA, it must support the mirror without inducing any stresses mechanical or thermal. It must be impervious to micro vibrations originating from the host spacecraft and offer a stable optical platform when rotating the mirror i.e. a smooth well-damped motion. All of this is packaged in to a very restrictive mass budget of 2.5kg. This paper describes the key design drivers, the approaches that were taken in order to meet these and ultimately their effectiveness as measured during the development programme.

Keywords: Two axes, Stability, Damping, Low mass, CPA, Optical Terminal.

1. INTRODUCTION

The CPA has been developed by Contraves under the Inter-Satellite Link Front End project. ISLFE is co-funded by ESA as part of the ARTES-3 programme. The project is to develop the O-ISL terminal design to meet near term market opportunities.
Contraves has a family of three O-ISL terminals designed to suit the all currently foreseen requirements, the family consists of:

- OPTEL 80, the long range terminal capable of transmitting data at Gbit/s rates over distances of typically 80,000 km.
- OPTEL 25, the medium range terminal capable of transmitting data at Gbit/s rates over distances of typically 25,000 km.
- OPTEL 02, the short range ISL terminal capable of transmitting data at Gbit/s rates over distances of typically 2,000 km.

For the ISLFE project, based on near term market requirements a hybrid OPTEL 25 terminal was selected. The terminal is a hybrid in the sense that it uses the main elements of the OPTEL 25, the major exception being the CPA. This is taken from the LEO terminal which has been developed for constellations. So we now have a GEO OPTEL 25 terminal specifically configured for GEO crosslink applications.
The ISLFE terminal is made up of the following 3 major units:

- Optical Head (OH) Unit located outside the spacecraft nominally on the nadir panel.
- Laser Unit (LU) which can be located inside the spacecraft within the communications module (CM).
- Electronics Unit (EU) also located in the CM. For ease of accommodation the EU and LU are connected to one another via a harness.

The CPA is part of the Optical Head, the primary function of which is to provide the beam steering and realise a free-space optical link. The other major units within the Optical Head are the: Telescope, Optical Bench and the Fine Pointing Assembly.

The primary role of the CPA within this system is to point the terminal (beam) towards the partner satellite in addition it should compensate for low rate disturbances and drift.

2. DESIGN APPROACH

In order to establish an optimised overall optical terminal concept it was considered necessary to first establish an overall architecture that could meet the system level demands. This was done by holding a series of dedicated design workshops where all relevant skill groups were represented. This allowed top level system requirements and design targets to be negotiated and agreed, as a parallel activity to the preliminary concept definition. This approach enabled the allocation of budgets and the functional demands to be harmonised between subsystems and equipments. The most influential functional demands and design drivers established for the CPA were as follows:

- Provide a mechanism for controlling the azimuth and elevation orientation of a plane mirror (some concepts involved pointing the complete telescope).

- Achieve a very high mass and volumetric efficiency with minimum complexity.

- Provide high reliability over a long operating life.

- Ensure efficient operation with the FPA and minimise micro-vibration emissions.

The latter was considered a vital aspect of providing a sound system design. The concept chosen for acquisition and maintenance of the optical link is based on active compensation of on-board micro-vibrations (rather than passive isolation). The FPA is therefore designed to provide high bandwidth beam steering. This is achieved using flexure and voice coil technology, which means that the angular pointing range is quite, limited (i.e. ±7 milli-rads).

When in tracking mode, the FPA must avoid saturation with respect to angular range, acceleration and rate, and its design is optimised to negate the uncontrolled and cyclic micro-vibration inputs.

When the FPA detects the introduction of a significant bias it is the CPA that must provide corrective action. It is essential that this correction is applied in small steps and in a very smooth manner, i.e. very low transient velocities in order to avoid saturation of the FPA. So in principal the CPA can be operated in a quasi open loop mode driven by the drift detection from within the FPA, provided that it does not introduce transient perturbations during the bias correction movements. This implies a low bandwidth open loop system with virtually no jitter. This demand along with the others was agreed and formalised within a dedicated CPA requirement specification.
Details of the programme logic and the very demanding EBB schedule are outlined in chapter 6.

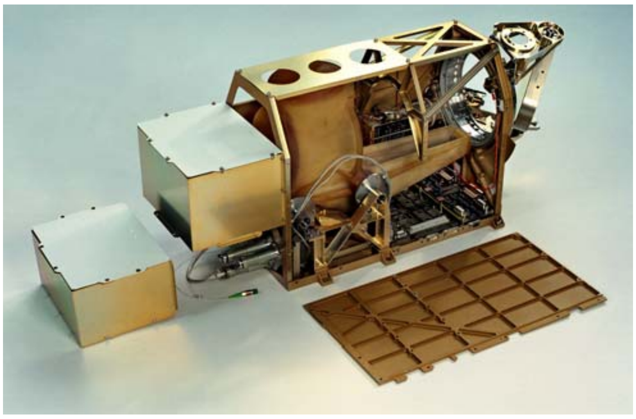

Figure 1. Photo of the Breadboard ISLFE OH

3. CPA REQUIREMENTS

After the terminal architecture (outlined above) was agreed upon, a dedicated CPA requirement specification was produced. The main points and design drivers are listed below for clarity.

- Accommodate an optical beam of ∅ 135mm
- Maximum mass 2.5kg (excluding mirror)
- First eigenfrequency to be above 160 Hz
- Support the mirror during launch, without the use of offload devices or launch locks
- Support the mirror during orbit without inducing any thermal or mechanical loading causing the optical performance of the mirror to be degraded
- Gimbal the mirror around 2 axes
  Azimuth +90° to –90°
  Elevation +5° to –5° (Mechanical)
- Actuation & Positioning shall be fully open loop
- Positional accuracy for each axis to be ±0.5 milli-rads
- Jitter:

$$Azimuth = \frac{170\mu rad}{JitterFrequencyHz}$$

$$Elevation = \frac{80\mu rad}{JitterFrequencyHz}$$

- Maximum power consumption to be below 5W
- On-orbit lifetime 15 years

4. CONCEPT DEVELOPMENT

4.1 Control Concept

Given that the design driver for the CPA control approach is compatible with the FPA, the adopted approach was to employ a very low bandwidth system that has minimal jitter and good damping. This is almost the complete opposite to conventional high precision pointing systems. A conventional approach could typically employ a stiff, high bandwidth servo system using a high resolution encoder for closed loop control and a three term controller to provide a stable damped system. A high gain would be needed to achieve the required pointing. Such a system would have a number of major drawbacks for this particular application, which include: -

- Considerable servo-jitter as the torque controller seeks the null position. While this may be low in amplitude it will have high frequency components making it difficult to achieve (or even predict) compliance with the flowed down jitter requirements.

- Servo dither will result in significant exported micro-vibrations; which would apply further demands on the FPA.

- Prolonged dither at specific angular positions could significantly degrade the bearings and their lubrication.

- A closed loop system would require the use of high resolution position transducers. This can be a major procurement issue for the azimuth drive axis, as the encoder size is driven by the need to accommodate the optical path through the mechanism.

Given the above issues, a far more suitable approach was to employ an open loop stepper motor drive system with the motors driven synchronously in micro-step mode. The natural low stiffness electro-magnet coupling between electrical field orientation and rotor position provides a low frequency drive system, minimising high frequency jitter. The other main advantages are the relatively simple drive electronics (no encoder or commutation). These motors are also inherently reliable (available with dual winding) and clean. This actuator and drive system approach also lends itself very well to the specific requirements of both axes.

The choice of this control and actuation approach requires that other elements of the design concept are implemented in a fashion that ensures a backlash free transmission of torque between motor rotor and inertial load with controlled stiffness, very low stiction, and the implementation of pure viscous damping to achieve a smooth controlled response at the mirror with minimal post micro-step oscillation. These characteristics have been achieved by the novel transmission systems, optimised bearing systems, all the drive system bearings have semi rigid preload to reduce stiction to an absolute minimum and passive damping using magnetic/eddy current elements.

4.2 General

Given the size of the mirror required to accommodate a beam of ∅135mm and its mass budget of 1kg it can readily seen that the CPA mass budget of 2,5kg is very restrictive. This coupled with the first eigenfrequency requirement of 160Hz. Initial concept and trade off studies which included the requirement that no launch lock or offload devices were allowed, resulted in beryllium being selected as the baseline material structural material. It's excellent stiffness to mass ratio making it an ideal choice, the down side of Beryllium is of course it's toxic properties and cost. Due to the high procurement costs, a very detailed justification exercise was undertaken, for each beryllium part, which included looking at the gain in eigenfrequency (Hz) against the procurement cost in CHFr. This resulted in a material change from beryllium for two major items; firstly the gearbox housing is now manufactured from titanium and the mirror brackets in AlBeMet, which still results in a cost saving over beryllium.
The rotation of the azimuth & elevation axes presented two very different sets of problems, firstly the azimuth axis requires a 180° rotation and must allow a ∅135mm beam to pass through it, whilst conforming to the mass, pointing & jitter budgets. An example of the azimuth jitter budget for clarity is:
If the azimuth axis were to have a jitter frequency of 10 Hz the peaked stepped angular motion, including all step size contributions must be less than 17μrad. Standard bearings of this size would not only consume the entire mass budget, their stiction characteristics would also not allow the jitter requirement to be met.
The elevation axis has a very small angular displacement requirement of only ±5° full range, ball bearings do not generally lend themselves very well to such small rotations. It was therefore decided that a flexural element based system would be more suited to this axis.

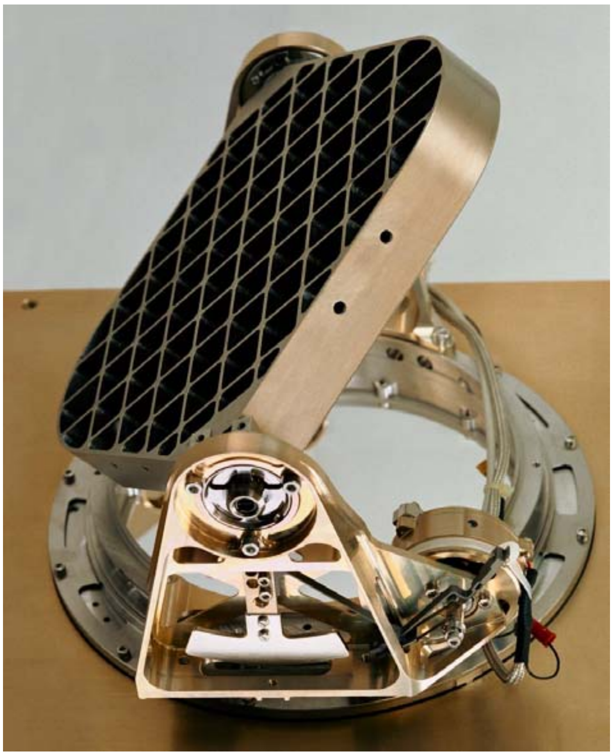

Figure 2. CPA

5. MECHANISM DESIGN

The CPA is made up of the following major elements:

- Mirror
- Elevation Axis Flex Pivots
- Elevation Drive System
- Mirror Support Structure
- Azimuth Bearing
- Azimuth Drive system
- Azimuth Cable Wrap

A brief description of these major elements are given in the paragraphs below:

5.1 Mirror

The mirror can be described as the mechanisms passenger, it is there looking for a nice smooth ride and does not want to be pushed, pinched or abused in any way. The mirror is made of beryllium, it is light weighted. The surface is gold coated, wave front error must be $\leq\lambda/25$rms. In order to maintain the mirror's optical properties it is essential that it is supported in a stress free manner, this is particularly important for the on-orbit thermal environment.

5.2 Elevation Axis Flex Pivots

Due to the schedule restraints of the EBB programme it was decided to use standard commercial flex pivots on the EBB model's elevation axis. However as part of an in-house development programme specially designed pivots would be used on all subsequent models. One of the main aims of this programme was to produce an installation/application friendly pivot design. The problem with the standard pivot range is the time and effort (cost and compromise) that is required to engineer their fixation and protection in to the mechanism's design. The Contraves flex pivot uses a titanium flexible cross blade element welded in to a titanium housing. The flex pivot is screwed on the mirror then in turn screwed on to the mirror support bracket. The flex pivot is internally protected against radial and axial overload. The internal surfaces, which come in to contact in the event of an overload, are coated with a combination of Balinit A & Balinit C. The flex pivots have been designed to have an unlimited fatigue life over the ±5° range. The torque/rotation curve is also linear up to 5°
The physical properties of each flex pivots are as follows:

Axial stiffness	$1.23*10^6$N/m
Torsional stiffness	0.36Nm/rad
Radial stiffness	$0.69*10^6$Nm/rad
Bending	30.35Nm/rad
Mass	0.035kg

The main driver in determining the axial stiffness for the flex pivot was to increase the axial compliance for the mirror, during the on-orbit thermal loading. For the CPA EQM strain gauges are installed on the flex blade elements for health monitoring and to ensure that the mirror is mounted in to a stress free environment.

5.3 Elevation Drive System

The elevation drive system (see figure 3) comprises of four main elements, these are the type 21 stepper motor, a 0.020mm thick steel drive tape, driving arm and support arm.

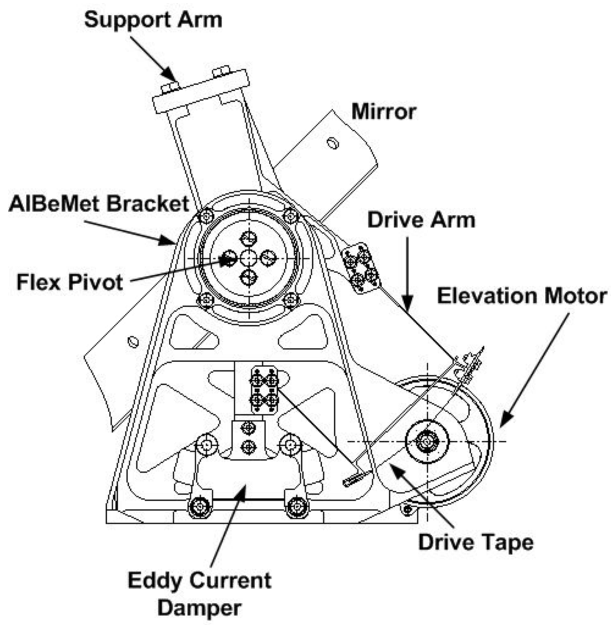

Figure 3. Elevation drive system

The stepper motor is driven in micro-stepping mode, using the maximum resolution, repeatable mirror steps of 5 μrad have been measured on the EBB. Positional reference is via a strain gauged soft end stop on the back of the motor, the end stop provides positional reference in both directions to better than 2μrad. The motor drives the mirror with a ratio of 27:1, the motor rotates the steel tape around a drum fixed on its axis. In order to maximise the drive ratio a tape drum ∅7.5mm is required which is slightly small for this tape thickness, bending stresses are greater than 700MPa. This level of stress is obviously not ideal when a long (15 years) fatigue life is required. In order to eliminate any possible stress raisers a 6 step manufacturing technique has been developed whereby we can achieve a 3μm radius on each corner of the tape, see figure 4, the target at the start of the programme was 5μm. Since the commencement of this programme we have also had contact with a tape manufacturer, who have manufactured a tape of 33μm thick which has successfully achieved 100 million revolutions around a ∅6.35mm drum for the computer industry.

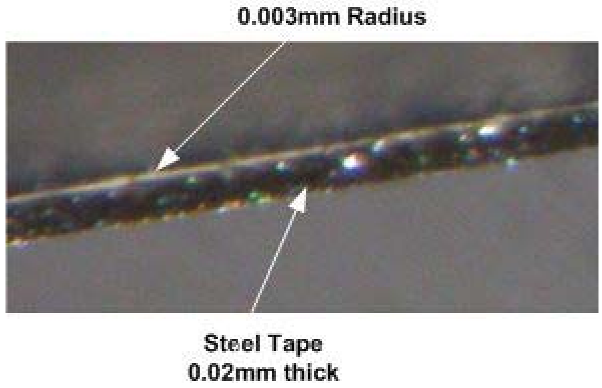

Figure 4. View on edge of Steel Tape

Figure 4 is a x180 magnification of the steel tape edge. The tape is connected to the drive arm, the drive arm is a flex element, which is now made from 0.4mm thick titanium, (the original drive arm as discussed in the problem below was 0.75mm thick see figure 6). The drive arm stiffness is sized is to attenuate all motor disturbances above 7Hz. The natural frequency of the drive system is designed to be less than 7Hz. The drive arm is connected to the support arm, which is in turn fixed to the mirror. In order to attenuate the incoming micro vibrations, which could cause mirror rotations up to 1 mill-rad an eddy current damper has been engineered in to the support arm. The damper comprises of magnets bonded to the inside of a vacoflux fork assembly, a high purity copper slug is bolted to the end of the support arm. The slug travels through the fork assembly, which is bolted to the mirror, support bracket. In order to meet the required performance it was calculated that a damping ratio of 0.4 would be required. During the EBB testing a damping ratio of 0.35 was measured. There was also no measurable stick slip above 2µrad.

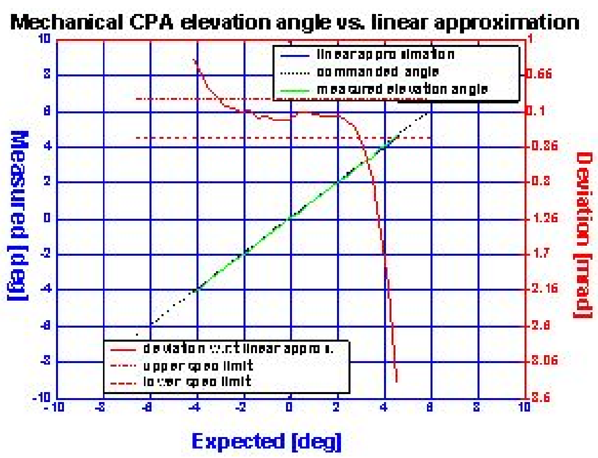

Figure 5. Non linearity Characterisation

There were however some problems encountered during the EBB testing, namely the non linearity of the elevation drive system, the effect of which can be seen in figure 5. Although this non linearity could quite easily be corrected with the control software, it was unexpected and it's cause unknown. Investigations were performed in order to determine it's cause, these took on many forms including investigating the effect of the gravity vector. Finally the reason was found to be local deformation of the drive arm, see figure 6.

This was resulting in a change of the effective drive arm radius causing lost motion in the positive rotation sense and decreasing the arm length in the negative direction, increased motion.

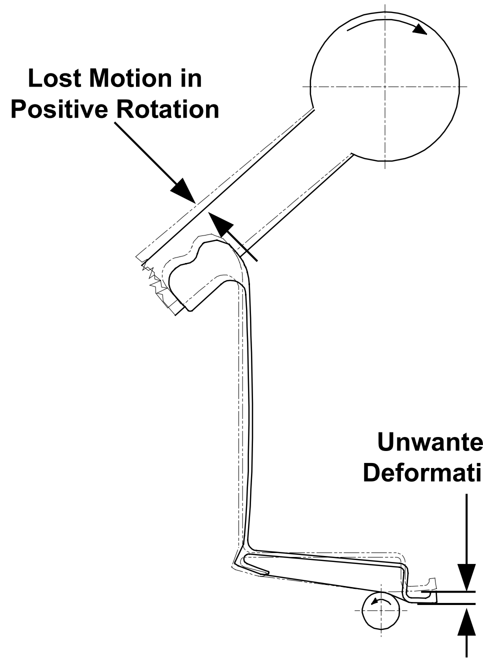

Figure 6. Local Deformation of Elevation Arm

In order to eliminate this phenomenon the design was modified, whereby a second leg has been introduced making the design more symmetrical and eliminating the local deflection problem. The downside of the modification is that the new arm is very thin, only 0.4mm, see figure 3.

5.4 MIRROR SUPPORT STRUCTURE

The mirror support brackets are constructed in AlBeMet, the azimuth shaft and housing are manufactured in beryllium. Beryllium offered the only real solution to meeting the low mass & high eigenfrequency requirements, it also has the advantage of being thermally matched to the mirror. Although as discussed in the flex pivot description it was also decided to design in further axial compliance via the flex pivots. Other material such as titanium were considered for such items as the brackets, as they presented significant cost savings but this presented serious thermal loading problems for the elevation motor on the on-orbit case. The brackets also provide structural support other equipment such as the beacon assembly.

Figure 7. AlBeMet Brackets on Beryllium Shaft

The photograph in figure 7 was taken during the EQM assembly programme.

5.4 AZIMUTH BEARING
The azimuth bearing allows rotation about the azimuth axis, it also allows the optical beam to pass along its axis of rotation to the telescope. One of the critical factors for this bearing, besides, mass and reacting the launch loads without any offload device, was the Delta Friction Torque. The difference between the static friction and the running friction as this would have a large and adverse effect on the jitter performance.

With a view to minimising this DFT the following parameters were investigated:

- Reducing ball compliment
- Use harder balls- i.e. Ceramic
- Reduce conformity between ball and race
- Reduce ball spin, by changing contact angle
- Optimise ball separation system

The final selection for this application was an angular contact, face to face, ∅200mm bearing pair. The bearing is "semi-soft" preloaded to 500N using a titanium membrane to clamp the bearing pair. The balls are ceramic with a free rotating ball separator system. The bearing is lubricated with Penzanne 2001, as are all of the CPA bearings. Bearing lubrication and torque characterisation were performed by ESTL. To prevent "excessive" gapping of the bearing under launch loads a system of snubbers are positioned around the titanium flange, these are adjusted on assembly and limit the axial movement to a maximum of 0.02mm.

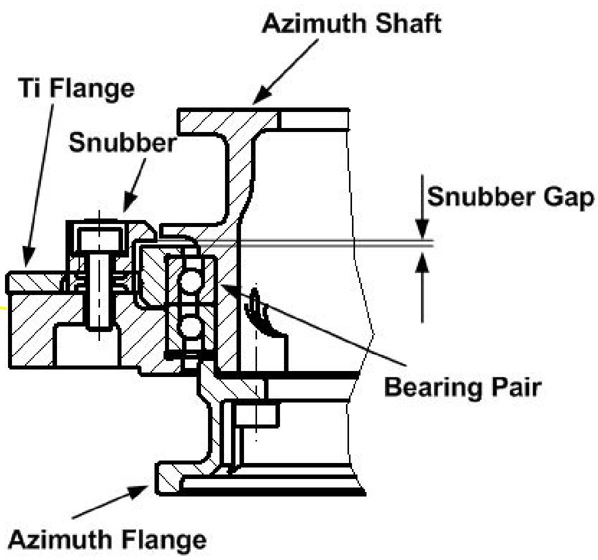

Figure 8. Azimuth Bearing Installation

5.5 AZIMUTH DRIVE SYSTEM

The azimuth drive system comprises of 3 main elements, these are type 21 stepper motor, eddy current damper and gearbox. The type 21 stepper motor is exactly the same as that used on the elevation axis. A type 23 motor was used for the EBB, but as the programme demonstrated just how large our margins were we decided to change to the type 21. This offered some mass saving and commonality of parts. The azimuth drive system has been designed with maximum stiffness in mind. This is to prevent to the build up of strain energy, which will cause sudden, erratic unwanted movements. The eddy current damper, which is attached to the motor shaft, was slightly changed from the EBB version. It was in fact turned inside out, that is to say the magnets attached to the vacoflux ring are now rotating with the motor shaft, this was implemented to maintain the inertia for the rotating parts. The high purity copper ring is attached to the motor housing via 3 stand-offs.

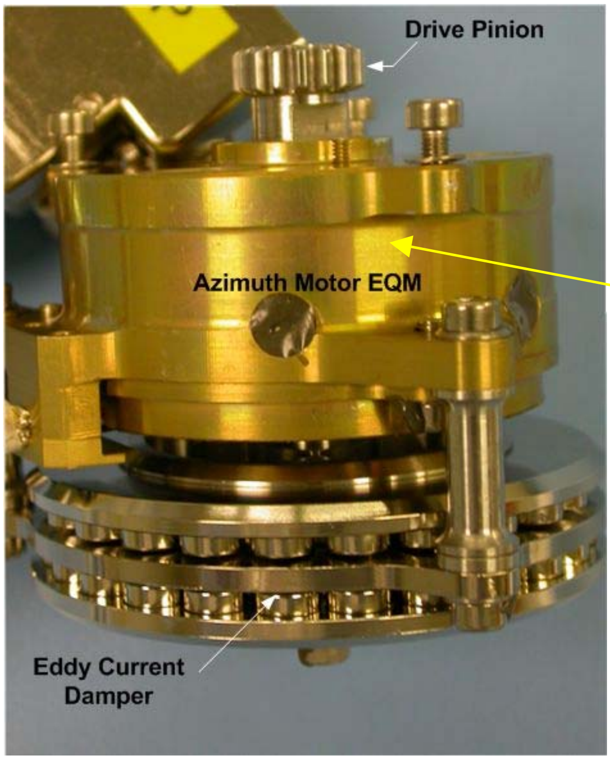

Figure 9 Azimuth Damper

Figure 9 shows the final EQM version of the azimuth damper, the photograph was taken during the EQM assembly programme.

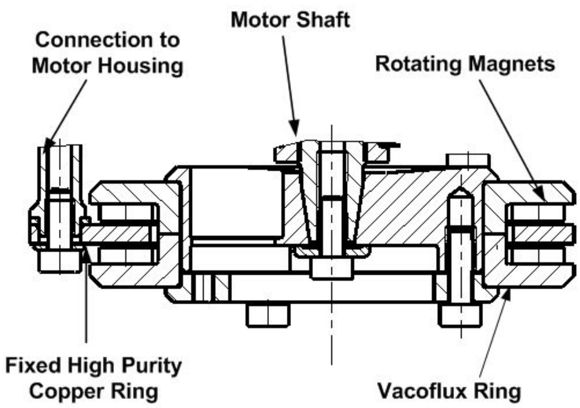

Figure 10. Cross Section of Azimuth Damper

The motor is again used in a micro-stepping mode. It has two end stops fixed to the optical head which again limit the travel by providing a soft stop to avoid inducing shock loads in to the drive system and strain gauges provide an accurate reference point for the step counting system.

For the gears a module of 0.8 was selected; the gears are steel driving, bronze for the first stage and steel driving aluminium for the second stage.

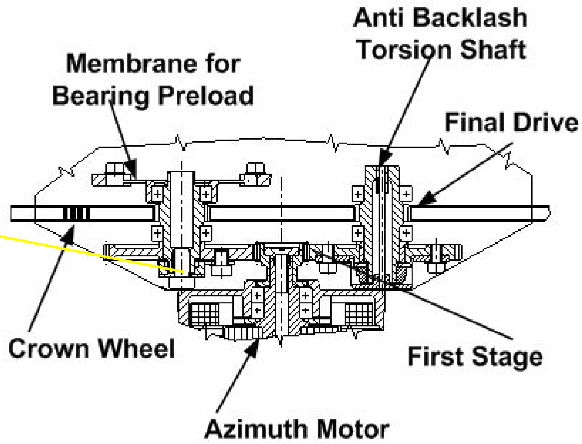

Figure 11. Cross Section of Gearbox

The gears are lubricated with penzanne 2001. As is also the case with the motors anti creep barrier has been applied to the relevant surfaces to prevent oil migration.
The gearbox is a 2 stage reduction system the first being 3.125:1 the second 16:1 giving an overall ration of 50:1.
Backlash is eliminated by the use of a torsion shaft (spring) fixed on the centre of one of the gears. See figure 11.
The following parameters were measured during the EBB test programme:
Natural frequency of mirror drive system 98Hz the design target was 100Hz
Damping ratio 0.45 the design target was >0.4
Minimum commanded and achieved step size was 2μrad. During measurements to quantify the jitter performance the following characteristics were identified; no identifiable stick slip, the stiction dead band is in the order of 1μrad, this is the oscillation about the commanded angle position.

5.6 AZIMUTH CABLE WRAP

As the azimuth axis has only a limited rotation requirement ±90° a cable wrap (flex print) system has been employed to allow this rotation. The wrap is an Omega type (shape), which radially requires very little space, as it folds back over itself there are also no sliding surfaces therefore no risk of generating debris. See figure 12. The cable wrap is custom made, it has a total of 35 power and signal tracks each of which is insulated with two layers of Kapton.

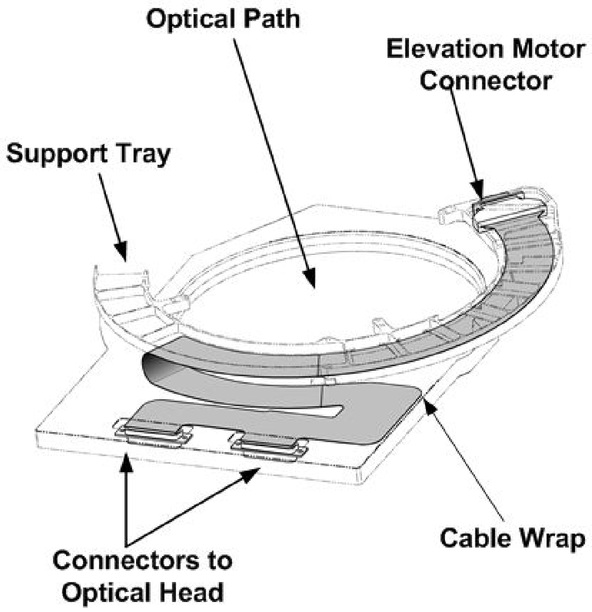

Figure 12. Omega Cable Wrap

The cable wrap is supported on a tray running around the circumference of the azimuth axis. The cable is only 0.37mm thick and a mass of 650 grammes including connectors. Its resistance torque is less than 100Nmm.

6. DEVELOPMENT APPROACH

The CPA programme consists of 2 models. A breadboard model EBB and a qualification model EQM. The development logic was to produce the EBB in a very short time scale. The schedule for EBB was to design, manufacture and assemble the CPA within a 9 month period, this programme was actually achieved. This very short programme was driven by two main requirements:

- The need to demonstrate the proof of this novel design concept as early as possible

- A fully functional CPA was required at system level in order that other terminal sub-systems could be developed and tested within a reasonable time.

The EBB is fully functional, but as one would expect with such a heavily compressed schedule there are a number of compromises in this model. The main build standard compromises (compared to the proposed flight standard) are:
The use of aluminium as a structural material in place of beryllium and AlBeMet.
Aluminium mirror in place of the beryllium one.
Non standard motor bearings, Conrad type bearings were used in place of the longer lead-time angular contact pairs.
Non standard bearing lubrication system
No cable wrap.
The EBB is currently in stalled in the breadboard OH and is being used for development testing.
The EQM is a fully flight representative model, which will undergo a full qualification test programme including an accelerated life test upon the completion of all of the environmental tests.

7. CONCLUSION

From the testing performed to date on the EBB and the feedback already obtained from the EQM programme it can be said that novel approach adopted in this CPA design has been a success. It is true that the EQM must still undergo it's full qualification programme, the deign analysis has been completed so it is hoped that this test programme will provide results of interest and nothing else.

8. REFERENCES

[1] G.C. Baister, Dr. Ch. Haupt, S. Matthews, T. Dreischer, R. Pender and A. Herren; The ISLFE terminal development project - results from the engineering breadboard phase

9. ACKNOWLEDGEMENTS

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