HIGH ACCURACY FLEXURAL HINGE DEVELOPMENT

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ABSTRACT

This document provides a synthesis of the technical results obtained in the frame of the HAFHA (High Accuracy Flexural Hinge Assembly) development performed by SENER (in charge of design, development, manufacturing and testing at component and mechanism levels) with EADS Astrium as sub-contractor (in charge of doing an inventory of candidate applications among existing and emerging projects, establishing the requirements and perform system level testing) under ESA contract.

The purpose of this project has been to develop a competitive technology for a flexural pivot, usable in highly accurate and dynamic pointing/scanning mechanisms. Compared with other solutions (e.g. magnetic or ball bearing technologies) flexural hinges are the appropriate technology for guiding with accuracy a mobile payload over a limited angular ranges around one rotation axis.

1. INTRODUCTION

The technical challenge of developing a high accuracy flexural pivot is to design a device with an optimised blade configuration which creates only a small resistive torque, perfectly aligned with the rotation axis and which provides a high stiffness/load capability in all other axes.

Fig. 1. - Flanged pivots.

SENER has optimised the blades configuration, designing a monolithic component to overcome manufacturing/assembly issues, ensuring a satisfactory manufacturing repeatability and an acceptable recurring cost of the component.

The pivot comprises of a mobile and a fixed parts joined by blades which are bent when the pivot is rotating. The maximum rotation that is possible to reach due to this design is +/- 15.5 degrees, beyond this angle the parts touch each other and the movement is blocked. This mechanical end stop prevents from damaging the component by excessive mechanical rotation.

The pivot mechanical interfaces are a challenging issue, aiming to provide a small volume/lightweight component, an "easy to accommodate" unit and a composite device which can embed an absolute optical encoder or a rotary actuator (both being frameless).

In addition to this, based on these pivots SENER developed the HAFHA demonstrator, see Fig. 2, which is a one-axis mechanism, designed in order that the hinge meets the requirements derived from the FSM (Fine Steering Mechanism) architecture [2].

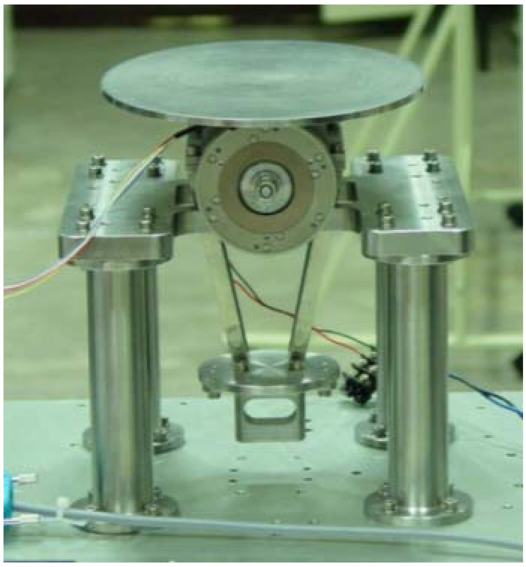

Fig. 2. - HAFHA Demonstrator.

2. DESIGN REQUIREMENTS

The major requirements for the HAFHA flexural pivot can be summarised as follows:

2.1 Angular motion range

- The pivot shall allow a range of at least +/-7 degrees, with infinite fatigue life duration.
- The flexural pivot design objective shall be to reach a range of +/-15 degrees.

2.2 Motion accuracy

The accuracy of the flexural pivot is defined in terms of centre-shift at maximum rotation angle:

- Centre shift at max rotation angle: 0.05 mm for FP

2.3 Stiffness

The stiffness values specified are assumed to be applicable for one flexural pivot:

- Rotational stiffness about X axis < 0.2 Nm/rad
- Axial stiffness along X axis >28 N/µm
- Axial stiffness along Y & Z axis >28 N/µm

2.4 Load capability

The flexural pivots shall be capable to withstand the following quasi-static loads at 0º:

- Axial static load > 500 N
- Radial static load > 500 N

2.5 Mass

The mass of the flexural pivot should be less than 50 g without account for the additional area required in the hinge for interfacing with encoder and/or motor.

2.6 Fatigue life

The pivots shall survive 365 000 FSM acquisitions (leading full range +/-7º actuations) during in orbit and 50 000 cycles during on ground AIT life.

The following plot illustrates the duty cycle:

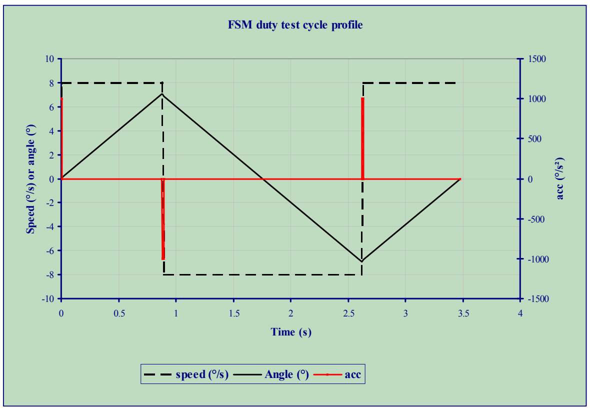

Fig. 3.- Fatigue duty cycle.

2.7 Strength Requirements

Axial & Lateral Sine		Axial & Lateral Random	
Hz	**Value**	**Hz**	**Value**
-18	11 mm	20-80	+3 dB/oct.
18-100	40 g	80-350	0.04 g2/Hz
50-80	6 g	350-2000	-3 dB/oct.
0-100	3.5 g	Overall Level = 6.06 grms	

Table 1.- Strength requirements.

2.8 Thermal Interface Requirements

Operating Temp. [º C]	Non-Operating Temp. [º C]
-40... +70	-50...+90

Table 2.- Thermal requirements.

3. PIVOT CONCEPT MAIN TRADE-OFFs

A general concept of flexural hinge is composed by three elements:

- internal ring
- external ring
- flexural blades for connecting both rings

The internal and external rings are two reference surfaces required mainly for I/F purposes. The flexural hinge has a well-defined rotation axis normally defined geometrically by the axis of the reference surfaces.

The flexural blades are the elements that connect the internal and external ring and provide the functional performances of the flexural hinge.

Several concepts were compared:

Fig. 4.- Three straight internal blades (TSIB)

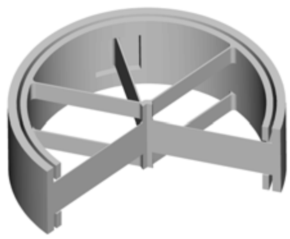

Fig. 5.- Concentric rings with blades in series (CRBS)

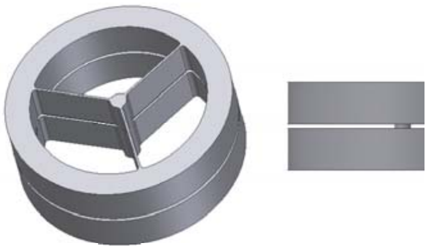

Fig. 6.- Adjacent rings with blades in series (ARBS)

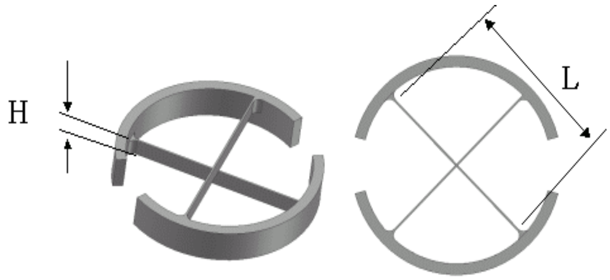

Fig.7.- Two blades- two shells concept (TBTS)

The trade-offs included stiffness, strength, centre-shift, fatigue, mass and manufacturing feasibility as main parameters. Finally, TBTS concept was the selected baseline for this FSM application as:

- A centre-shift of 0.05 mm is allowed
- High radial and axial stiffness
- Volume is small
- Optimisation of the blades size is possible.

SENER established a method to optimise the blades dimensioning. This optimisation method consisted on the preparation of a sequence of tables to connect the required performances (Stiffness, strength, fatigue life, centre shift) with the H/L ratio for different thicknesses.

From this method it could be noticed that the axial and radial stiffness for the TBTS concept were proportional to the factor H/L and to the thickness. If the thickness of the blade decreases the H/L ratio increases and if the thickness was optimised (minimum) to verify the rotational stiffness requirement, the radial and axial stiffness value increases.

The fatigue response is connected with the stress level of the blade, so the lower the stress the longer life. When the concept with two blades was studied the stress value considered for calculating the MS against fatigue was the one due only to the bending moment.

SENER already used the “Flexural flexible monolithic unitary modules” concept in the design of the Two Axis Scan Mechanism (TASM) Flexural Pivots under ESA contract during the year 1996 [1].

The current flexural pivot design for the HAFHA project, the TBTS, is basically the TASM design scaled down.

4. DESIGN DESCRIPTION

The Flexural Pivot comprises:

- a fixed structural part
- a mobile structural part
- two perpendicularly crossed flexural blades
- I/F device between the pivot structural parts and:
    - the mobile payload, directly or via the hinge structure
    - the angular sensor
    - the fixed support

These I/Fs have been designed to:

- allow a robust mechanical mounting of the pivot into the mechanism
- provide direct interfaces to accommodate an optical encoder, leading to an integrated “pivot-encoder” component.

As illustrated in Fig. 8, specific mechanical interfaces are accommodated into the pivot:

- Threaded holes on the mobile and fixed shells of the pivot, to provide reliable attachment points of the pivot to the fixed and mobile part of the mechanism.
- A flange with three holes orientated at 120°, to attach the pivot to the fixed part of a motor (for mechanism actuation) or encoder (for pivot rotation angle fine measurement)

- Three threaded holes on the mobile interface ring (behind the flange), to attach the pivot to the motor or encoder mobile part.

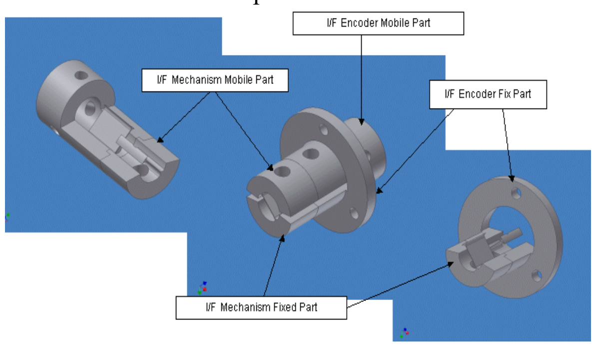

Fig. 8.- Pivot configuration.

Custom 455 and Ti 6Al 4V have been the materials used to manufacture the pivots.

The mobile part external diameter is 15.5 mm, the flange diameter is 30.5 mm, the total length is 31.2 mm for flanged pivots and 22.9 mm for the non-flanged pivots.

The thickness of the blades are 0.15 mm for Custom 455 and 0.20 mm for Ti 6Al 4V.

4.1 Description of the demonstrator

The pivots were integrated and tested into a demonstrator representative of one of the candidate applications. A one-axis motorized hinge, representative of Mirror Fine Steering Mechanism (FSM) for optical communication was selected, considering that:

- This application is not achievable with "off the shelf" standard pivots from the market, and it combines demanding performances and low recurring cost.

- The FSM mechanism architecture (motor / pivots / encoder) aiming at being generic, customizable for several applications.

TheFig. 9 shows the general assembly of this demonstrator. It features a total mobile mass of 2 kg including a counterweight to align the centre of gravity with the rotation axis.
The major components of the demonstrator are:

- two SENER HAFHA flex-pivots
- a Limited Angle Torque (LAT) Motor (Auxitrol) Peak Torque = 0.6 Nm
- an absolute optical encoder, featuring a +/- 15 arc sec accuracy and 20 bit resolution. (CODECHAMP)
- all mechanical parts (structure, connecting parts, dummy mirror) are made of stainless steel.

In addition to this, as the blade thickness required for +/- 7˚ was in the order of tenths, the design of the HAFHA demonstrator included a hold-down device, which was designed in order to increase the stiffness of the mechanism in launch configuration.

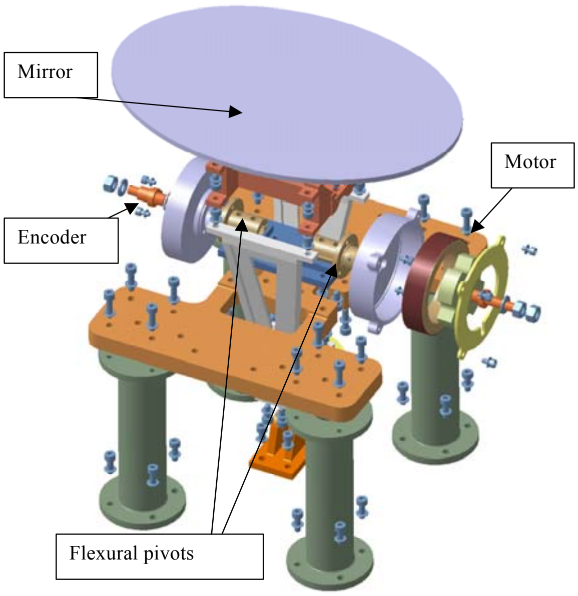

Fig. 9. Hafha demonstrator.

The encoder is a frameless design, its guiding being ensured by the flex pivot itself. These parts are joined to the pivot by intermediate connecting parts.

The scan unit housing is made of Aluminum. It is connected with the pivot by 3 screws; the pivot flange limits the axial displacement with respect to the pivot. The grated disk of the encoder is attached to the hinge by a connecting shaft, and three radial screws. The disk is pressed by the nuts and washers (see Fig. 10).

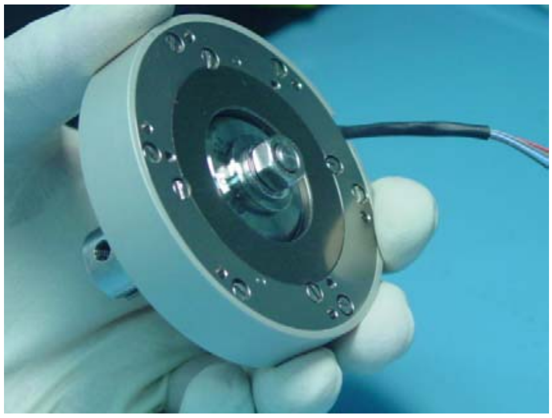

Fig. 10.- Encoder assembled with pivot.

5. QUALIFICATION TESTING

The test campaign has included functional tests at component and mechanism level, thermal cycling, vibration and life test

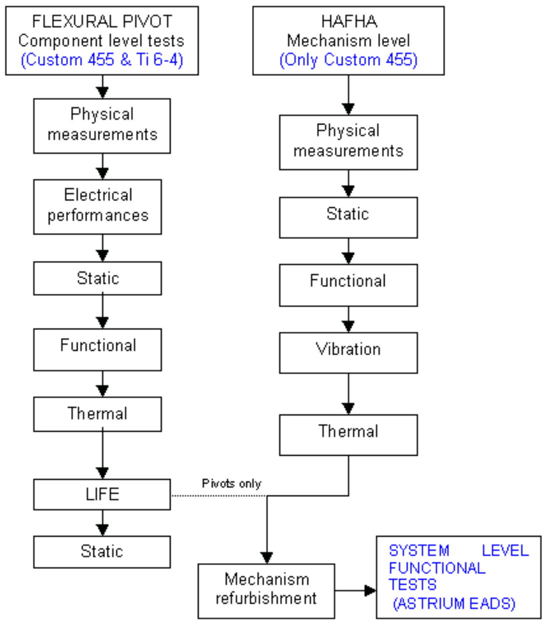

Fig. 11. - Test matrix.

The tested items are shown in table Table 3.

Type of test	Number of pivots to be tested			
	Flanged		Non-flanged	
	Custom 455	**Ti 6 Al 4V**	**Custom 455**	**Ti 6 Al 4V**
Hinge physical measurements	6	4	6	4
Hinge rotational stiffness	6	4	6	4
Hinge radial stiffness	2	2	2	2
Hinge axial stiffness	2	2	2	2
Hinge thermal cycling	6	4	6	4
Hinge centre-shift	3	2	3	2
HAFHA rotational stiffness	2 + 2			
HAFHA max. radial load			2 Destructive	
HAFHA functional	2 + 2			
HAFHA vibration	2			
HAFHA thermal tests (no vacuum)	2			
Hinge fatigue life	2 at ± 7° & 2 Destructive (1 at ± 10° & 1 at ± 15°)	2 at ± 7° & 2 Destructive (1 at ± 10° & 1 at ± 15°)	2 Destructive (1 at ± 10° & 1 at ± 15°)	2 Destructive (1 at ± 10° & 1 at ± 15°)
Hinge static	2	2		

Table 3. - Tested pivots.

Non-flanged pivots were designed to simplify the manufacturing. Both pivots were useful to carry out the tests and the final results were the same because the critical items, which are the blades, have the same dimensions. Flanged pivots were useful when the test require the motor and the encoder (Demonstrator rotational stiffness, functional tests and thermal test).

5.1 Component level tests

Physical Measurements

The following physical measurements were made

Pivot type	Custom 455	Ti 6Al 4V
Non flanged	22.3 g	12.0 g
Flanged	42.5 g	23.3 g

Table 4. - Physical measurements.

Rotational Stiffness

To perform this test, a pivot was mounted on the stiffness test tool, Fig. 12. A complete cycle between -15.5º and +15.5º was performed.

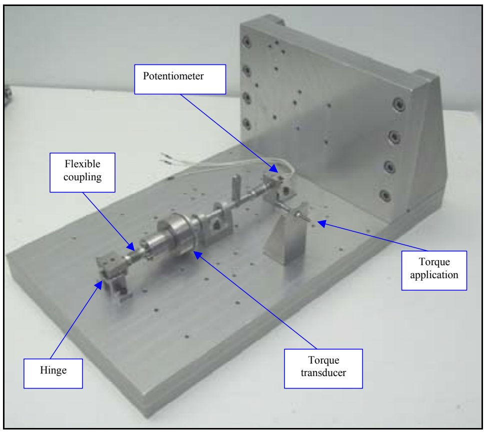

Fig. 12. - Rotational stiffness tooling.

Pivot type	Rotational stiffness Mean Value
Custom 455	0.19 (N·m/rad)
Ti 6Al 4V	0.21 (N·m/rad)

Table 5. - Rotational stiffness results summary.

Radial stiffness

The radial stiffness value per Custom 455 FP was 23.46 N/µm. and per Ti6Al4V FP was 23.76 N/µm.

Axial stiffness

The axial stiffness value per Custom 455 FP was 18.65 N/µm and per Ti6Al4V FP was 9.50 N/µm.

Functional Test.- Centre shift

The tooling was positioned on a 3D measuring machine which touched the mobile part of the FPs at several points for each rotated angle, then the circumference connecting all those points was calculated and the centre was determined w.r.t. the reference point (origin of coordinates).

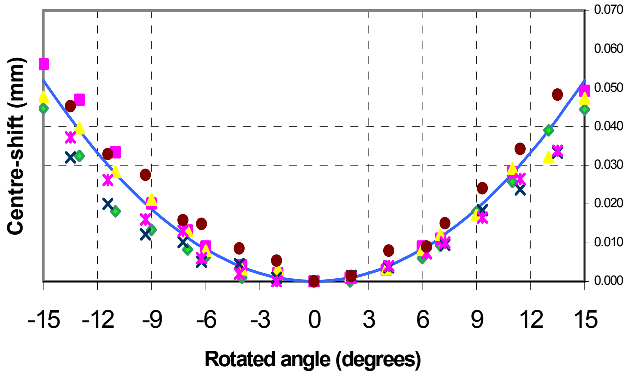

Fig. 13. - Custom 455 pivots centre shift.

At 7º, the mean values measured were 11.5 µm for Custom 455 pivots and 12.2 µm for Titanium that are good enough compared with 50 µm required.

Thermal cycling test

The objective of the test was to simulate the thermal environment on the pivots and to verify the pivot survival under extreme non-operating temperatures.

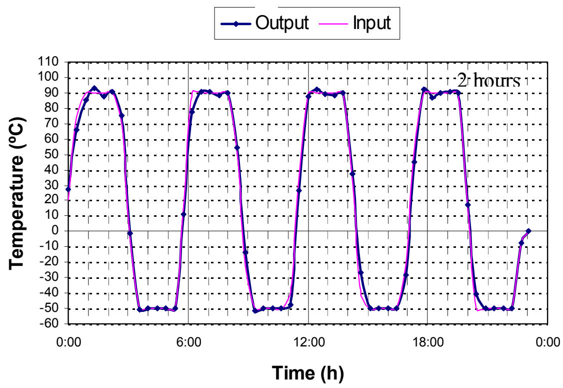

Fig. 14. - Thermal cycling.

Some pivots were subjected to four complete cycles between the extreme in-orbit non-operational temperatures: -50ºC to + 90ºC (see Fig. 14). No variation in the mechanical and functional performances of the pivots was found after this test.

5.2 Mechanism level tests (Demonstrator)

Physical measurements

The mass of the main parts which compose the Demonstrator (without the locking mechanism) are listed in the following table:

Mobile parts mass	2048.92 g
Fixed parts mass	4855.08 g

Table 6. - Demonstrator mass.

Maximum radial load

The radial load that the hinge withstands at 0 degrees was determined. This load depends on the height and the thickness of the blade.

The test set-up and the test procedure were similar to the radial stiffness test. The only difference is that now the locking device was included to avoid the mechanism rotation.

The maximum load supported by both hinges was **5995 N**, so one pivot supports **2997.5** N according to the way the load was applied.

Functional tests

The main objectives are to characterise the system in open loop and to analyse the closed-loop performance.

Open loop

Natural Frequency and Bode Diagram

The following Fig. shows Amplifier voltage (V), Motor current (A), Angular position (deg).

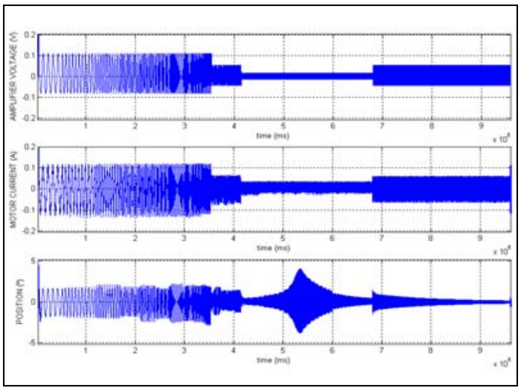

Fig. 15. - Excitation signals and response

These are corresponding to the frequency sweep performed to gather the data necessary to analyse the frequency response of the system in closed-loop

The bode plot obtained considering the input as the demanded current and the output as the angular position is shown in Fig. 16, where the natural frequency of the system is located at 1.14 Hz that is lower than the max. required (1.6 Hz at the X axis of the mechanism).

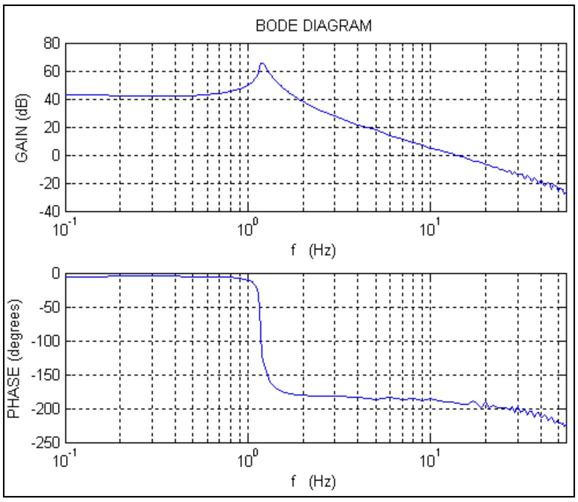

Fig. 16. - Bode diagram open loop of the system : demanded current vs. Angular position (deg/A).

Phase Margin and Gain Margin procedure

From the bode diagram (see Fig. 17), the following stability margin were obtained:

- Phase Margin: 52 deg at 10.7 Hz
- Gain Margin: 16.7 dB at 46.8 Hz

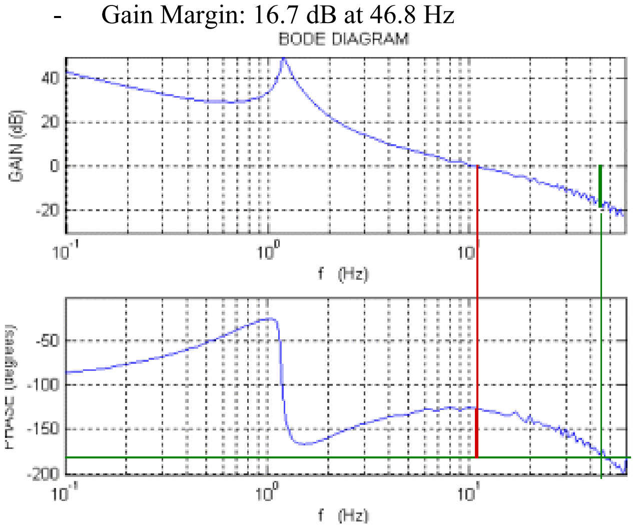

Fig. 17. - Controller and system Bode diagram.

Closed loop

Closed-Loop performance

Different profiles reproducing the defined system manoeuvres were defined as the command angle

- A 0. 1º Slew Step
- A 1º Slew Step
- A 7º Slew Step
- An alternative ramp at 0.6º/s
- An alternative ramp at 8º/s
- Sinusoidal scanning up to 0.6º/s

The operation limits considered were: Angular range: +/- 7º, Max speed: 8º/s, Max. acceleration: 1000º/s^2.

Results	Settling time (s)	Accuracy (deg)	Stability over 25ms (deg)	Stability over 30s (deg) [1σ]	Dead Time (s)	Overshoot (%)
0.1° Slew Step	0.1	0.0004	0	0.000213	0.01	19.5
1° Slew Step	0.19	0.0004	0	0.000165	0.015	10
7° Slew	0.84	0.0004	0	0.000176	0.018	1.4
0.6°/s Alternative Ramp	-	0.0025	0.0003	-	0.007	-
8°/s Alternative Ramp	-	0.0044	0.0004	-	0.007	-
Sinusoidal scanning	-	0.004	0.0002	0.00095	-	-

Table 7. Close loop behaviour results

Vibration tests

The objective of the test was to simulate a dynamic environment more severe than the one expected during the launch.

The main results of the vibration tests were:

- The X-axis fundamental frequency of the Demonstrator was found at 415 Hz , Y-axis at 490 Hz and the Z-axis at 517 Hz.
- The maximum accelerations measured during X-axis test were 20g in sine and 47g in random at the mirror dummy. During Y-axis test were 20g in sine and 48.5g in random at the mirror dummy and during Z-axis qualification vibration were 20g in sine and 47.5g in random at the mirror dummy.

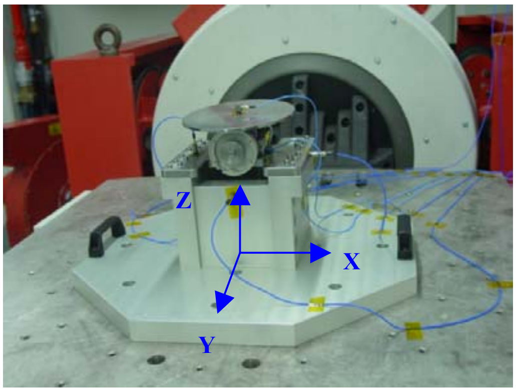

Fig. 18. - Y-axis vibration test.

Thermal tests

In order to verify the pointing performances of the mechanism at high and low operating temperatures, four tests have been performed:

- Functional Test Profile at ambient before the thermal cycling profile (before the thermal test)
- Functional Test Profile at +80 ºC
- Functional Test Profile at –40 ºC
- Functional Test Profile at ambient temperature after the thermal cycling profile.

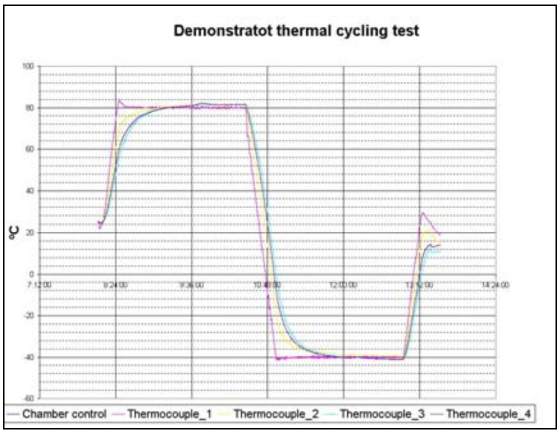

Fig. 19. - Thermal cycle for functional tests.

The dynamic behaviour of the mechanism in these tests is practically identical to that shown in the performance tests carried out in closed loop tests.

Life tests

The tests were developed pivot by pivot using a motor and an adjustable four-bar linkage, designed to rotate +/-7º, +/-10º and +/-15º. A torque transducer was included in order to know the moment the pivot breaks being the human inspection not mandatory.

Fig. 20. - Life test configuration.

SENER pivots, at the end of the life tests it has been verified that the hinges can rotate:

Material	Type	±7º	±10º	±15º
		Number of cycles	Number of cycles	Number of cycles
Custom 455	Flanged	>1.000.000	925.258	56.680
	Non flanged	>1.000.000	445.260	63.225
Ti6Al4V	Flanged	>1.000.000	353.117	68.471
	Non flanged	>1.000.000	151.842	75.228

Table 8. - Life test results.

For the Custom 455, which is a ferritic material, it can be ensured that over 1.000.000 cycles and below ±9º the failure line remains horizontal, so 1.000.000 cycles value is taken as reference as infinite fatigue life. For Ti6Al4V, as it is not a ferric material, the infinite fatigue life borderline is not so clear, nevertheless in order not to enlarge the test duration, if the FP exceeds 1.000.000 cycles it is considered infinite life.

Fig. 21. - Broken pivots after fatigue test.

6. CONCLUSIONS

- Target requirements have been verified. A very low value of the rotational stiffness (0.19 N·m/rad for Custom 455) has been achieved although this value is opposite to reach a high stiffness/load capability in all other axes. A low value of rotational stiffness is very important as the number of possible applications is increased allowing either a higher frequency response or a lower power consumption of the pointing/scanning mechanism when it operates in a control loop.
- The axial stiffness is close to specified values for Custom 455 (about 84 %) while the radial stiffness is more distant (about 67 %). As there is dependence between rotational and axial/radial stiffness, it must be considered by the space scanning mechanisms to allow the tuning of the rotational and axial/radial stiffness.
- The selected design has been verified against the worst environmental conditions without any degradation of its performances.
- The dynamic correlation with the structural analysis model was successfully proved.
- The system operating range (from +7º to –7º) was accomplished; the system natural frequency is 1.14 Hz.
- Static gain has been obtained along the whole operating range.
- The system open loop bode diagram and the transfer function has been obtained and the results are as it was expected. No significant differences were found.
- The controller has been tuned based on the results of the open loop analysis in order to accomplish the performance specifications. The measured performance parameters in the time domain largely accomplish all the requirements.
- The controller robustness has been demonstrated as virtually the same values of accuracy and stability are obtained varying the controller nominal gain within 50% - 150 %
- The system closed-loop bode diagram and the transfer function has been obtained and the results are as it was expected. No significant differences were found to the theoretical model.
- For life cycles, the results corresponds with expected values for Custom 455 and Ti6Al4V except for +/- 10º but to establish a general conclusion a wide test campaign can be needed as only two pivots of each type were tested and this amount is not enough to get a general rule.

To sum up, it can be concluded that the developed pivot and the associated demonstrator cover the requirements and can be a valid alternative for a wide range of space pointing / scanning applications, as it has been verified by the system level tests where the performances were in complete adequacy to the OISL (Optical Inter-Satellite Link) mission requirements.

With respect to this SENER makes an acknowledgement to EADS Astrium (Mr. T. Blais) for its collaboration in defining the requirements and for the system level test performance.

Material	Blade dimensions (mm)			Mass (g)		Krot (Nm / rad)	Krad (N/μm)	Kaxial (N/μm)	Non operating temp (ºC)	Fatigue 7º	Fatigue 15º	Centre -shift 7º (mm)	Buckling radial load (N)	Tensile radial load (N)	Tensile axial load (N)
	L	H	T	F	NF.										
Custom 455	6.0	9.7	0.15	42.5	22.3	0.19	23.46	18.65	90 / -50	>1. e6	> 65.000	0.011	398.5	2997.5	553.89
Ti6Al4V	6.6	8.2	0.2	23.3	12.0	0.21	23.76	9.50	90 / -50	>1. e6	> 15.000	0.012	375	633.87	412.16
Verification method				Test		Test	Test	Test	Test	Test	Test	Test	Analysis	Analys /Test	Analysis

Table 9.- HAFHA performances summary.

7. REFERENCES

[1] “Scan Mechanism for Master Limb Sounding Instrument”, 8th European Space Mechanisms & Tribology Symposium, Toulouse, France, Proceedings ESA SP-438, September 1999

[2] “Fine Steering Mechanism for New Generation Optical Communication Terminal” 8th European Space Mechanisms & Tribology Symposium, Toulouse, France, Proceedings ESA SP-438, September 1999