FLEXURE PIVOT FOR AEROSPACE MECHANISMS

Simon HENEIN, Peter SPANOUDAKIS, Serge DROZ, Leif. I. MYKLEBUST, Emmanuel ONILLON

Centre Suisse d'Electronique et de Microtechnique (CSEM), Jaquet-Droz 1, CH-2007 Neuchâtel, Switzerland.
simon.henein@csem.ch. +41 32 720 5111

ABSTRACT

A generic high precision flexible pivot dedicated to pointing and scanning space mechanisms is described. Typical applications are payload scientific instruments and inter-satellite communication systems.

The work started with a comprehensive analysis of the present and future market and mission needs. This survey was used to define requirements for a baseline pivot concept in terms of mobile mass, inertias, size, angular strokes, etc. A comprehensive solution catalogue was established and traded-off to select an original concept. This concept was designed in detail and manufactured by wire electro-discharge machining for characterization on special test benches. The main characteristic of the proposed design is its very small parasitic center translation : less than 2 µm off axis shift over a ±10° motion range (which is 65 times less than a classical cross spring pivot of the same size). Resolution better than 50 µrad was achieved over a ± 10° stroke. Fatigue tests show that the pivots can endure more than 1.4 million cycles of ± 7.5° stroke without failure.

1 INTRODUCTION

Flexible bearings are an alternative to conventional mechanical bearings [1]. While in the latter, motion is obtained by sliding or rolling between solid bodies with the resulting non-linearities, flexible bearings rely on the elastic properties of matter and are characterized by several advantages: high precision, no friction, no hysteresis, no wear and polluting debris, no need for lubrication, no jamming risks, no backlash, vacuum and cryogenic compatibility, and possible monolithic manufacturing.

A generic high precision flexible pivot dedicated to pointing and scanning space mechanisms has been developed to be used in payload instruments and optical inter-satellite communication assemblies.

The key advantages of the selected solution are:

- Minimized parasitic center shift : less than 2 µm for ±10° stroke. For comparison, a classical cross spring pivot with two 36.5 mm long blades (same outer dimensions as the butterfly pivot of Figure 1) would have center shift of 131 µm for a ± 10° stroke according to a formula derived by Wittrick [2].
- Planar design allowing monolithic manufacturing by wire electro-discharge machining (EDM).
- Folded architecture which leads to long angular strokes: 20 to 30 degrees.
- Embedded mechanical end stops, versatile mechanical interfaces.
- High radial, axial, transverse angular stiffnesses.

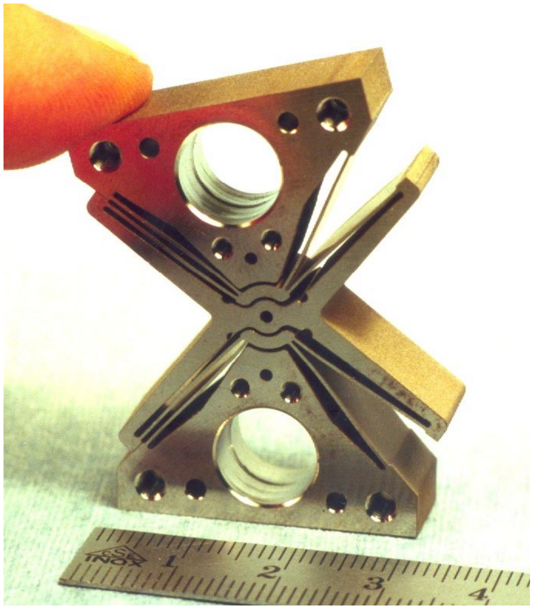

Figure 1: Picture of the monolithic "butterfly" flexure pivot. Angular stroke up to ± 15°. Parasitic translation (centre shift) below 2 µm for ± 10° strokes. Machined monolithically in Titanium.

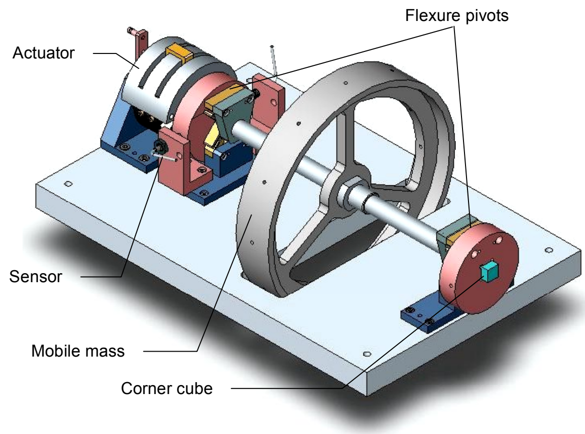

Figure 2: Pair of "butterfly" flexible pivots mounted on a test bench composed of a flywheel (mobile payload), a voice coil actuator, a pair of position sensors and a corner cube mirror used to measure the rotation centre shift.

Proc.10th European Space Mechanisms and Tribology Symposium, San Sebastian, Spain, 24-26 Sept. 2003, ESA SP-524, pp.285-288.

2 MARKET SURVEY

A literature review [3 4 5] was performed to obtain a database of pointing mechanisms by selecting relevant papers published in mechanism symposiums. Published papers discussed experiments that have already flown, technology developments and future trends in pointing mechanism requirements. The applications reviewed ranged from Scientific Instruments, Earth Observation Payloads to Satellite Communication Systems. The papers reviewed were not limited to flexure elements such that a comparison could be made between ball bearing and flexure bearing applications over a given angular range.

The review of the published material showed that there is a wide range of applications that utilise flexure pivots. When the angular rotation is limited, the friction-free flexible guiding element is clearly the designer's choice over ball bearings. It is interesting to note that a number of applications were based on custom pivot developments because standard off-the-shelf versions could not meet stringent space application requirements. The typical failure of industrial flexure pivots is their failure during vibration testing. A common failure is the weld which sometimes attaches flexible blades to rigid parts. In all the applications covered, a voice-coil or electromagnetic actuators is used.

A statistical approach was taken to analyse the data to define a future pivot specification with respect to the mechanism payload mass and a typical angular range. The total number of applications considered for the analysis was 16 flexure pivot mechanisms.

2.1 Payload mass

The payload mass varies from 12 g to 45 kg. The mean payload mass over this entire range is 8.5 kg, including three relatively large ones. A histogram of the payload range places the majority of the applications (75%) with a mobile mass of less than 5 kg. Within the 5 kg category range 75% have a mobile mass of 1 kg or less. The "butterfly" pivot was optimised to support a 1.2 Kg payload mass which covers more than 60 % of all the considered applications.

2.2 Pivot stroke

The average angular range of all the flexure pivot applications is ± 6°. The "butterfly" pivot was optimised to withstand 6 million ± 7.5° stroke cycles without failure, which covers more than 63 % of the applications considered in the study.

2.3 Pointing accuracy

Among the surveyed applications, the average required pointing accuracy is ± 90 µrad, with a large standard deviation. The most demanding application required ± 0.03 µrad and the lowest ± 500 µrad. A histogram of the mechanism pointing accuracy places the majority of the applications (80%) with a sensor accuracy better than ± 100 µrad. The "butterfly pivot" sensor and actuator were selected to have a pointing accuracy of ±50 µrad which covers more than 90 % of the surveyed applications.

3 DESIGN OF THE "BUTTERFLY" PIVOT

A wide solutions catalogue was established and traded - off. The selected solution is the "butterfly" flexure pivot (Figure 3). This structure is composed of four elementary pivots linked in series, which have their four centers of rotation collocated. A first pair of flexible blades guides the intermediate block 1 with respect to the fixed base, giving it one degree of freedom corresponding to a rotation about the axis O. A second pair of flexible blades guides the intermediate block 2 with respect to 1 about the same axis O. A third (respectively fourth) pair of blade does the same for the intermediate block 3 with respect to 2 (respectively the mobile block with respect to block 3). Since all 8 blades are identical, the angular stiffness of each elementary pivot is the same. Thus, when an angle α is imposed to the mobile block the four pivots share the stroke equally (i.e. block 2 rotates by $\alpha/4$ with respect to the fixed block, block 3 by $\alpha/2$ and block 3 by $3\alpha/4$).

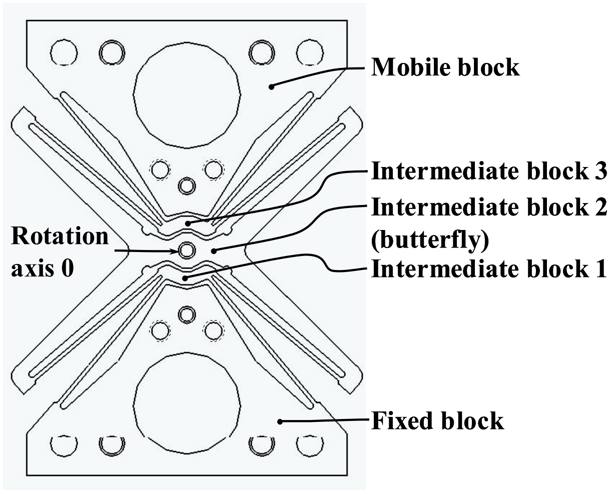

Figure 3: The "butterfly" pivot structure

3.1 Parasitic center shift compensation

In a planar analysis, the movement of the mobile block of flexure pivots can be described as a pure rotation of the block about the ideal rotation axis of the pivot, followed by a translation. The translation vector is called *parasitic center shift* because it corresponds to the displacement of a point of the mobile block which would be located on the ideal rotation axis when the flexure pivot is in nominal position.

A key advantage of the butterfly pivot arrangement is the very low parasitic center shift which accompanies the rotation. This results from two effects :

a) Distribution of the total stroke over four stages.

The magnitude of the parasitic translation which accompanies the rotation of flexure pivots increases non-linearly in a progressive manner with angular stroke. This means that a flexure pivot rotating by an angle α, produces a parasitic center shift with a magnitude greater than the sum of the magnitudes of the parasitic center shifts of two pivots (identical to the former) rotating by $\alpha/2$. Distributing the rotation over four stages is therefore a mean of reducing the total center shift.

b) Center shift compensation between stages

Attaching four elementary flexure hinges is series leads to a total parasitic center shift which is the vectorial sum of the shifts of each stage (not the sum of their magnitudes). The first pair of pivots (linking the fixed block to block 1, then 2) are arranged in a manner that make their two respective parasitic shift vectors compensate one another efficiently. The same is true for the second pair of pivots.

3.2 Load capabilities and eigen-modes

The "butterfly" pivot was designed to withstand several thousands of ± 10° stroke cycles and to make the "butterfly" internal eigen-mode frequency as high as possible (320 Hz). The selected material is Titanium (TiAl6V4). The overall size of the pivot is 43x30x10 mm^3. The blade dimensions are 0.35x10x15 mm^3. The calculated maximum stress in the blades, including a 1.2 stress concentration factor due to the small filet radius at the tips is 376 MPa for a ± 10° stroke.

The load capabilities of a hinge composed of a pair of pivots (Figure 2) were calculated using the finite element method. The admissible stress used for these calculation is 260 MPa. The calculated axial load capability is 180 N . The calculated radial load capabilities is 430 N.

The angular stiffness of the hinge assembly is 2 Nm/rad. This leads to an eigen frequency of the flywheel of 2.7 Hz. The axial eigen mode is 128 Hz. The radial eigen modes are above 320 Hz.

4 EXPERIMENTAL TESTS

A total of 15 "butterfly" flexure pivots were manufactured by wire electro-discharge machining in titanium (TiAl6V4). The blade surface roughness is better than Ra 0.2 µm.

4.1 Closed loop position control

The positioning performances were measured on a special test bed (Figure 2) including a contactless actuator (Figure 4), contactless angular position sensors (Vibrometer eddy current transducer position sensor DMS 110 with TQ401 used in differential mode against a cylindrical eccentric target), a flywheel representing a dummy payload, mechanical end stops, and a corner cube optical mirror to measure the parasitic center shift.

The pivot assembly is controlled using a discrete state space representation of the mechanism and a pole placement algorithm (poles are chosen to be the poles of a 3 Hz Bessel filter) plus a feed-forward loop.

The positioning repeatability is better that 52 µrad over the ± 10° range. The rise time to perform a 20° step (0.35 rad) is less than 0.25 seconds. The maximum speed reached during this steps is 2.8 rad/s, the maximum acceleration is 35 rad/s^2.

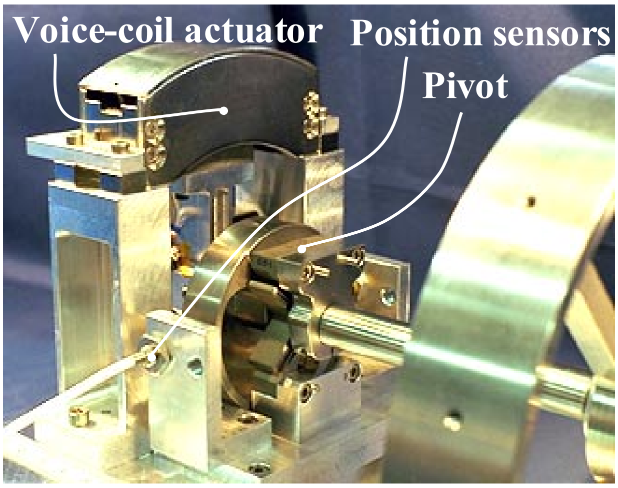

Figure 4: BEI rotary voice coil actuator used in differential mode to drive the hinge assembly.

4.2 Parasitic center shift

The parasitic centre shift was measured using a corner cube reflector mounted on the rotation axis and a Newport / MicroControle “straightness” collimator model LDAXIS, with LDS1000 electronics. The parasitic centre shift is smaller than 2 µm over a ± 10° motion range of the hinge assembly.

4.3 Fatigue life cycling

The fatigue tests were performed by Mecanex SA (Nyon, Switzerland) on their special test bed composed of a DC motor actuating 5 flexure pivots simultaneously at 3 Hz. The angular stroke is imposed on the pivots via flexible bellows which load them with a pure torque. A preload spring applies a slight, well controlled, traction force onto the pivots which ensures that when blade failure happens, the mobile part of the pivot hits the dedicated electrical contact.

The Stress versus Number of cycles (SN or Wöhler) curve obtained from the pivots fatigue tests fit very precisely SN curves obtained on 42 CSEM standard test specimens machined in the same alloy, with the same EDM process. This validates the analytical and finite element method calculations made to calculate the stress in the "butterfly" flexure with respect to angular stroke. Hence, the same theoretical model can safely be used to design similar pivots with other specifications (smaller of larger size, greater or shorter admissible strokes etc.)

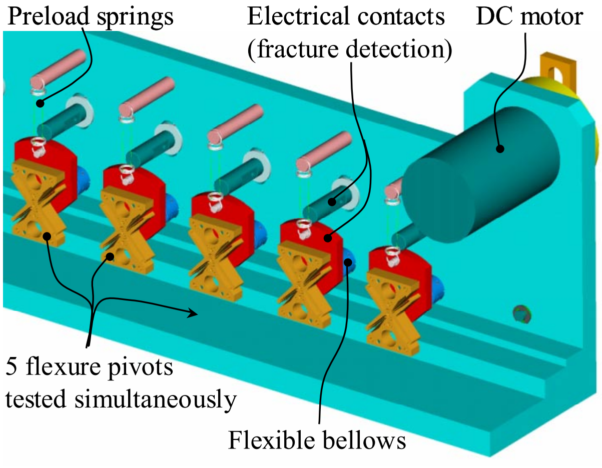

Figure 5: Fatigue test bed

A total of 9 flexure pivots where tested. Table 1 presents the fatigue results.

Angular stroke	Number of parts tested	Average number of cycles before failure
± 14°	2	42'000
± 10°	3	190'000
± 7.5°	2	more than 1.4 10^6

Table 1:Results of the fatigue tests performed on the butterfly flexure pivots

4.4 Launch vibrations

A flexure hinge assembly was mounted with as dummy mass of 1.2 Kg representing the payload. A flexible bellow was used to block only one degree of freedom: the natural rotation of the assembly. The 5 other degrees of freedom were left free.

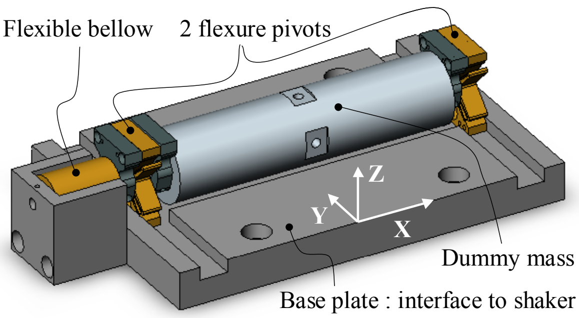

Figure 6:Vibration test bench

This test was meant to estimate the vibration survivability limits of this type assembly without launch clamping. The system has been vibrated with a typical random vibration spectrum. It withstood up to 3 grms acceleration in the Z axis. At a higher level, some of the blades buckled. Since the typical vibration levels reached during launching phase are much higher, it shows that proper of the payload launch clamping is required to protect such an assembly

5 CONCLUSION

The search for a long stroke flexure pivot with a reduced parasitic center shift lead to an original design composed of a four stage structure where the errors of pairs of stages compensate one another. The presence of three internal degrees of freedom could be drawbacks for on-ground applications (possible excitation of these modes by dynamic or static loading). Nevertheless, in the particular type of applications identified in the market survey, low motion speeds and accelerations are required and the static loads are low since the devices work in micro-gravity. In addition to the center shift compensation, this planar structure has the advantage of being manufactured monolithically which suppresses many of the assembly problems of the state of the art flexure pivots like Lucas Free-Flex or Bendix (US), C-Flex Bearing Co. (US) or B.E System Celtic Pivots (F).

For these reasons, the authors believe that the "butterfly" pivot is an interesting alternative to the classical flexure pivots especially when pure rotation over long angular strokes is required.

ACKNOWLEDGMENTS

This work was funded by ESA, we thank them for their support. Part of the work presented in this paper was done by Mecanex SA (Nyon, CH), we thank N. Hamze for this collaboration. The flexure pivots were machined by Mecartex SA (Losone, CH), we thank Stefano Bottinelli for this work.

REFERENCES

[1] S. Henein, *Conception des guidages flexibles*, Collection Meta, Presses Polytechniques et Universitaires Romandes, Lausanne, 2001.

[2] W. H. Wittrick. The properties of crossed spring pivots, and the influence of the point at which the strips cross. *The aeronautical Quarterly II (4), 272-292*, 1951.

[3] Robert M. Warden. Cryogenic Scan Mirror Mechanism For SIRTF/MIPS. *32nd Aerospace Mechanisms Symposium, NASA/CP-1998-207191*, May 13-15, 1998.

[4] Aage Skullestal. Pointing Mechanism for Optical Communication. *8th European Space Mechanisms and Tribology Symposium, ESA SP-438,* Toulouse France, 29 September - 1 October 1999.

[5] R. Hofferbert. A Cold Focal Plane Chopper For Herschel-Pacs -- Critical Components and Reliability. *9th European Space Mechanisms and Tribology Symposium, ESA SP-480,* Liège, Belgium, September 19-21, 2001.