A LOW COST HINGE FOR APPENDICES DEPLOYMENT: DESIGN, TEST AND APPLICATIONS

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Abstract

The concept of micro-satellites requires adapted mechanisms with good efficiency for low cost and weight. It is therefore very attractive first to integrate several functions in the same device, and additionally to develop generic technologies that may be reused for several missions.
Following this philosophy, this paper presents the works conducted together by METRAVIB RDS and the CNES, to develop a self actuating, guiding and locking hinge for appendices deployment.
After the presentation of the basic principle, we describe the development logic which consists in three steps: the preliminary design lies on rough models to handle the weight and size to torque and stiffness compromise; then, advanced numerical simulations are requested to complete the design justification vs operational requirements; finally, various tests, from ground laboratory conditions, on a dedicated test bench, up to microgravity fligths, brought the insights for validation of the models.
In the last section, we present the technical data sheet of a first product dedicated to Solar Array of CNES MicroSatellites family, together with its application to a Deployable Sensor Carrier developed for DEMETER mission. This latter demonstrate the generic character of this promising device currently under qualification.

1. Introduction

METRAVIB RDS company and CNES (French Space Agency) have been working for more than 6 years on technological simplification and technical comprehension of Carpentier Joint based hinge principle for space applications. The basic idea consists in fully avoid a dedicated guiding mechanism, involving any kind of sliding or even rolling contacts as a potential source of problem related to (vacuum) tribology.
Therefore, a specific design has been imagined to combined, simply by using elastic strips, the functions of driving, guiding and locking satellites deployable appendices.
As the main drawback of such a principle is the uncontrolled character of the deployment kinematic resulting in relatively hard ending locking shocks, it has been envisaged to develop a composite concept of strips, integrating damping elastomer. The proof of concept study has been successfully conducted[1],[2] but suffers of a limited applicability regarding the narrow temperature range where the efficiency of the viscoelastic layer may be insured.
Finally, the concept, without damping, has been foreseen, for its first application, as a good candidate for a robust and low cost hinge for MicroSatellites solar arrays. Then a second step of development have been launched, in order to get a industrial product, and is reported in its main phases in the present paper.
In the second section, we describe the basic principle of the proposed technology, stressing on its expected advantages. Then, in the two following sections, we detailed the work related to the design and modelling of such articulations, that represents, in no doubt, the key point of the success of the development. As it will be presented, taking advantage of the technological simplicity of the product required quite a deep effort in terms of modelling, in order to control the specification oriented design. This was especially justified by the generic character we were looking for. This task is presented in two parts: in section 3, simplified tools for preliminary design (to handle roughly the compromise between dimensions and driving torque and stiffness), and in section 4 advanced simulations, finally required to complete the design justification file (note that due to very little driving torque and stiffness, some justifications cannot be performed through ground tests because of gravity induced perturbations). Experimental set up and measurements (both ground and microgravity tests) are presented in section 5, together with correlation with theoretical results.
Finally, in section 6, we present in a few figures the technical performances of the industrial version of the MicroSatellite SA hinge as well as a dedicated deployable instrument carrier that has been built around it and therefore demonstrate the generic character of this development.

2. Basic principle

The starting idea below the concept tried to simplify, by reducing the number of mechanical parts and avoid sliding / moving interfaces, the traditional "spring" based articulations technology. In the past, Carpentier Joint principle, which consists in a (usually steel) strip

with a curved cross section, has already been used for its capacity to combine guiding (anisotropic stiffness), actuating (spring effect) and locking (high stiffness contrast due to post buckling behaviour).
In the present case, a particular arrangement of three Carpentier strips (see figure 1) has been selected to provide a quasi centre - pin guiding behaviour. Compared to a single Carpentier Joint, such a geometry provides a high stability versus twist during opening, and increases both the driving torque and stability after locking (holding torque and forces, and stiffness).
Both the size of the strips (that may be different from the centre to lateral ones), and their relative position in the plane of the bases provide the flexibility to adapt the design to various compromises versus twist stability.
On a technological point of view, as the material of the strips must be highly resistant versus large deformations and induced stresses (special steel) while the base plates must be light and compatible with interfaces (aluminium alloy), it is mandatory to manufacture the articulation in several parts. The assembling procedure consists in clipping and bonding the strips between two semi base plates, with such a design that stress concentration is limited when submitted to both mechanical and thermal loads (differential expansion in strips, bonding layer and base plates). A special care must be taken to insure the final interface positioning which quality is one of the major points of the specification.

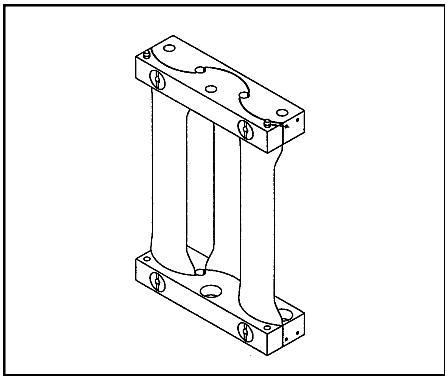

figure 1: Strips arrangement

3. Preliminary design

The design of an hinge is mainly governed by the specification of driving torque and the stiffness in open / locked situation. In the case of the presented technology, the calculation of the driving torque through numerical simulations (by finite elements) is of extreme complexity as we will see in the next section. Starting from blank, such an approach is clearly not appropriate to run parametric estimations of the driving torque for various geometry in order to set up a preliminary design. Facing that lack, an analytical model has been built to roughly estimate the driving torque, based on an elastic energy balance-sheet with strong hypothesis on the final deformed shape. The approach has been validated versus measurements on prototypes.
On the other hand, the stiffness in open situation is straight forward to obtain, as a linear elastic behaviour, from finite element modelling. Nevertheless, in order to save time for the design, a tentative of model reduction has been conducted: a finite element (FE) model of the hinge has been widely parameterised in terms of strips length, thickness, width, curvature and relative position to feed a large database of results through FE simulations; from this, well chosen simple interpolation laws can provide correct estimation of the stiffness. Again, the validity of the simplified model has been proven thanks to prototypes experiments.
An example of GUI window of this tool, developed under MATLAB® is given in figure 2.

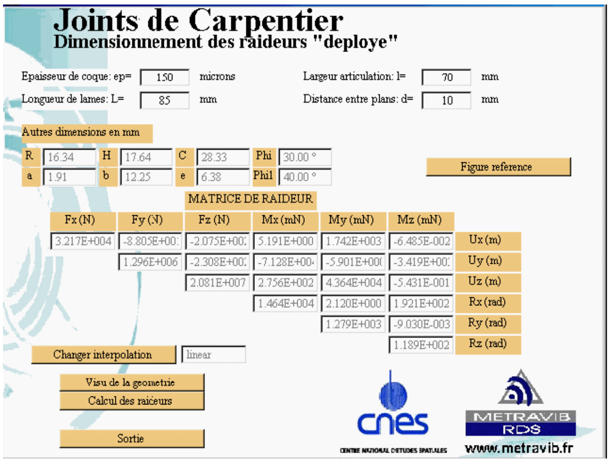

figure 2: Example of stiffness estimation issue from the model interpolation (simplified model in MATLAB® environment)

The use of this methodology may be successfully applied in two significantly different families of articulations, to predefine the overall dimension and strips thickness starting from required open stiffness and driving torque. Despite significant discrepancies on this latter parameter, the provisional design may be determined to enter the next step of optimisation that lies on more detailed (and costly) simulations.

4. Advanced simulations

In order to conduct the second step of the design, and to complete the full justification versus requirements, further simulations are requested. As explained before, the three major points to verify are:

1. the stiffness in open locked situation which governs the dynamic behaviour in orbital life, together with

2. the holding capacity, namely the torque (or lateral force) that the articulation can stand without unlocking, and
3. the available driving torque (and even stiffness) during deployment.

All this three points are studied thanks to dedicated FE simulations that were conducted on ANSYS® software (see an example of mesh in figure 3).

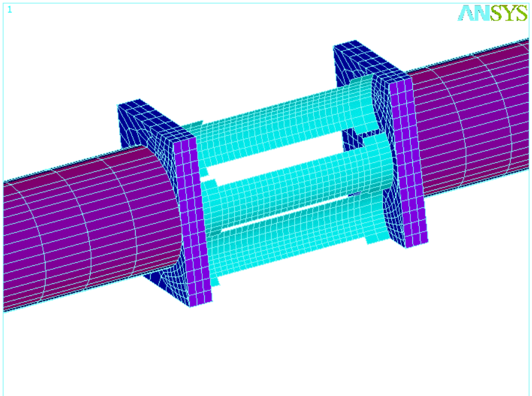

figure 3: Example of FE mesh (articulation + arms)

4.1 Open stiffness

As already mentioned, the verification of the open stiffness is relatively straight forward as it requires a linear elastic analysis. Nevertheless we may point out the difficulty of providing "elementary" stiffness, direction by direction. Indeed, as the articulation is fully 3D, its behaviour may not be strictly reduced to a centre pin hinge one. In other words, coupling will exist between translations and rotations. For instance, to get the lateral stiffness from a static analysis, looking at displacement induced by a unitary force, we face the problem of a dependency upon the location of the point where the force is applied.

When looking at vibration modes of a system made of the articulation loaded with a mass, the distance between this mass and the articulation base plate (or equivalently the relative distribution of the full inertia matrix terms) is therefore of prime importance. As the requirements are expressed in terms of minimum resonance frequencies under a given inertia matrix, the verification calculations were settled in that sense. An example of the deformed shape of the first mode is given in figure 4.

In case of an almost pure "axis - inertia loading", i.e. with a concentrated mass mounted on a relatively long light arm (which represents most of the practical applications), it is nevertheless possible to reduce the stiffness to a single axial term which is obviously important as an insight to verify compliance for various applications (especially in the context of the development of a generic component).

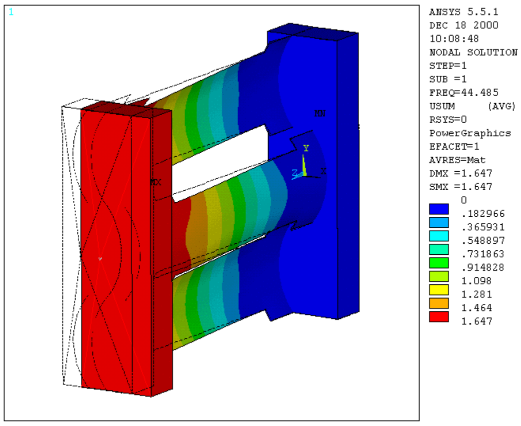

figure 4: First modal frequency calculation

4.2 Holding load capacity(unlocking limit)

The holding load capacity is also determined with the same FE model as used in the previous section, but here by a buckling analysis (critical load).

Again, the remark about the influence of the load application point remains valid.

Additionally, we have stated that the critical load is different in both directions on the hinge (negative angle), due to the unsymmetrical design.

It is important to indicate that the result of these simulations have two applications:

1. First, it provides the holding capacity to verify compliance with orbital life condition (no unlocking);
2. Secondly, it gives an input parameter for global kinematics models, in order to handle successive unlockings (if any) during deployments.

If, in the first case, conservative values must be provided (taking into account security margins, especially versus materials characteristics and geometrical inaccuracies), the data required for the second application must be given with the best accuracy as possible. Ground tests on a dedicated laboratory set up (see next section) have been extensively used to validate the model.

4.3 Deployment simulations

Last but not least, as the main operating function, it is necessary to predict the deployment kinematics. The objective is to verify that, in orbital situation (zero gravity), the driving torque is sufficient, all along opening path, to deploy the structure and to be sure that no interference arises with surrounding elements. Regarding the very low torque and stiffness, ground tests are unfeasible due to gravity induced perturbations, which emphasises the interest of such simulations.

Compared to the previously presented models, this point was much more difficult to handle on both theoretical and experimental sides.
On a theoretical point of view, two models have been set up, one to capture the behaviour of the articulation itself (local model LM), one to describe the deployment of the complete systems (several articulations, several arms; global model GM). This latter is still under development (correlation with tests), and the present paper focuses on the first one.
While, for the LM, there is no alternative to the use of FE simulations, regarding the complexity of the problem (full 3D geometry, highly non linear elasticity), the choice of a kinematics oriented method was done for the GM. Then, it is anticipated that this (lighter) approach will allow large parametric studies about various possible flight conditions or system configurations (mass, inertia and length of arms).
The key point of the methodology obviously lies upon the extraction, from the LM, of the required parameters for the GM, namely the articulation model reduction. The adopted hypothesis is a behaviour of centre pin hinge ("pivot"), which seems valid regarding the relative length of the deployable structures versus articulations dimensions.

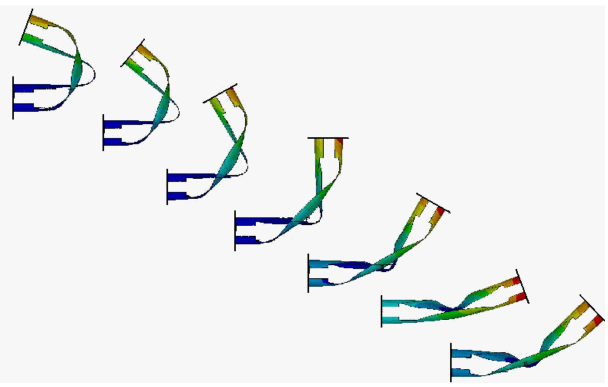

figure 5: Large strain deformed shapes

A non linear finite element model, neglecting inertia effects for the moment i.e. quasi static, has been developed under ANSYS® for the single hinge, very difficult to fix due to the strong mechanical instabilities (post buckling). First the choice of the formulation of the shell element, together with the mesh, had to be optimised through several iterations. Then several numerical schemes (solver method) have been tried. Despite these numerical precautions, it appears that the convergence of the calculation is impossible to achieve all along the opening path without coming back to the physic and setting time dependant boundary conditions to "help" passing hard points which corresponds to successive buckling related instabilities. Moreover, some instabilities may be passed in different paths (bifurcations), which corresponds to qualitative observations one can have when manipulating prototypes. Again, the way of applying the load (i.e. the length of the arm connected to the articulation) appears to be of extreme importance, as for the previous models.
Finally, the procedure consists in imposing the movement (angle of rotation of the arm), from the deployed (open) situation toward the closed one, and then going back to the open configuration to simulate the deployment. The driving torque is calculated for every 10° positions, providing the first parameter required by the GM, to describe the actuating function.
Figure 5 presents successive deformed shapes during opening, and correlation with measurements will be given in the next section (see figure 8).
The second part of the work consist then in extracting the stiffness, at different positions, in order to provide to the GM the second set of parameters, related to the guiding function.
This has been completed through the following procedure:

1. At a given position (angle), the deformed mesh is passed to another model, as the "mean" state (see figure 6-A);
2. A mass (known characteristics) is then added to one of the base plate while the other is clamped (see figure 6-B);
3. A linear modal analysis is conducted to get the first 6 modes (see figure 6-C);
4. Equivalent stiffness are then deduced.

The results from these calculations have been correlated with measurements as well (see next section, figure 9).

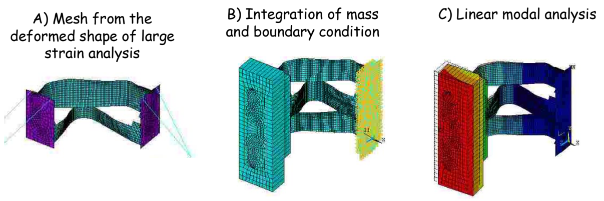

figure 6: Procedure to estimate stiffness during opening

From the data provided by this first model, correlated to elementary tests, the CNES is in the process of doing a full kinematic model of deployment, with a pivot reduction of the hinge.
The risk lies in the very low stiffness of the articulation during deployment. Under deployment inertia, large deformations are generated and characteristics may be modified (stiffness and torque) in such a way that the reduction of the hinge model could not be valid any more. In such a case, the only solution would be a dynamic large strain FEM based simulation as the global model. This way is currently envisaged in parallel.

5. Tests

In this section, we present first the elementary ground tests that have been performed to validate the advanced simulation previously detailed, and then microgravity tests (in Zero g flights) conducted on full deployable structures that will be used in the near future to validate the global kinematic model.

5.1 Ground tests

These tests were conducted in parallel with the advanced modelling previously presented to insure their validity which is mandatory regarding their complexity, especially for deployment.

We will not give a detailed description of elementary tests related to open stiffness and holding torque as they do not present any particularity and the results are extremely consistent with calculations. The first test (open stiffness) consist in a modal analysis of a system {hinge + inertia arm}, in a vertical position to avoid gravity effect or unlocking. The second test (holding torque) is also very simple and consist in loading, with increasing masses, the same system placed now horizontally, up to unlocking.

Let's now focus on the most interesting part of the experimental work related to the deployment. For driving torque measurement, it appeared that friction and gravity effects introduce too much perturbations due to low lateral stiffness. This justified the development of a dedicated test bench able to reproduce the "in plane" kinematics of a single hinge system.

This bench (see figure 7) is composed of:

1. A vertical rotating axis on which the hinge is mounted, driven by a motor to control the angular position for torque measurements;
2. An arm (adjustable inertia) mounted on the hinge, which extremity may slide on an air cushion table (no friction, gravity compensation) for deployments;
3. Angular position sensors (each 10 degrees) for deployments kinematic measurements;
4. A force sensor to estimate driving torque when rotating about central axis.

A direct exploitation of this experimental set up consists in measuring the driving torque versus opening angle. This was done on several prototypes, and with mounting / unmounting between measurements, to estimate the variability of the results (a special care regarding the highly unstable behaviour). The results are given in figure 8 where the thin curves are the different measurements, exhibiting the relative reproducibility at least in the more stable regions, i.e. far from open situation (where the main buckling arises, and where the torque strongly decreases from high value of holding capacity up to much smaller driving torque). In particular, the characteristic allure of the plots, with a "hole" about 100°, is relatively stable from one test to another.

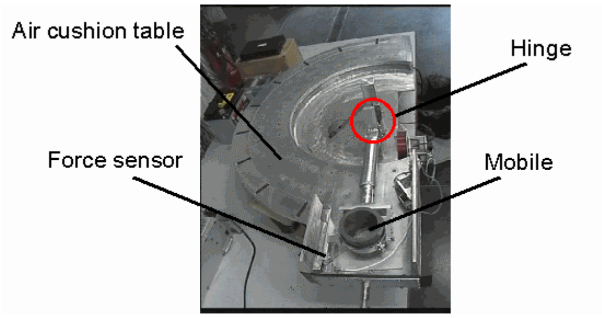

figure 7: Dedicated deployment bench

Also plotted in figure 8, for correlation, one can see the thick curve obtained from the previously presented model. To get this response curve, the model has been set up to describe very accurately the conditions of the test (length of the arm, positioning of the base plate). In conclusion, and especially taking into account the complexity of the behaviour, the correlation may be considered of very good quality for both absolute values of torque and allure of angle dependency.

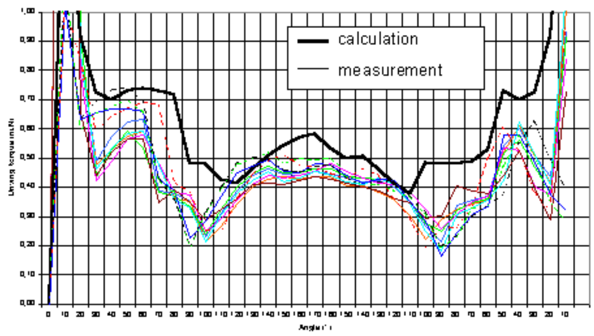

figure 8: Comparison between measured and calculated torques

Thanks to a dedicated experimental set up reproducing the condition of the model for stiffness estimation during deployment (see figure 6), this other part of the numerical simulation has been validated as well. The difficulty came from the practical arrangement (elastic cable suspension to maintain the hinge in the proper mean position), first not to introduce additional stiffness and secondly to stabilise partially open positions. Figure 9 presents the obtained results in a rough manner (so that a direct comparison is possible between calculations and measurements) i.e. in terms of resonant frequencies of the first modes that could be actually identified. Again, regarding the difficulties encountered for both experiments and simulations, the quality of the results is quite good, and for sure satisfying for a practical help to design.

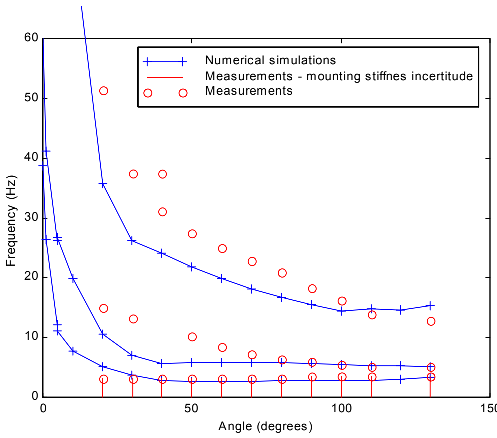

figure 9: Comparison between measured and calculated stiffness* vs opening angle (* here, resonance frequencies in conditions described in figure 6)

5.2 Microgravity tests

In the intention of kinematics predictions and of correlating the global model, deployments have been carried out in a micro gravity campaign on Zero-G flights.
A specific bench (see figure 10) has been developed:

- to release easily the dummy and relocks it quickly (practically, the time between two parabolic flights is limited);
- to realise a parametric study over arms inertia, with possibility for 2 arms - 2 hinges systems.

During deployments, base forces and accelerations are measured. The kinematic is recorded by a numerical camera.

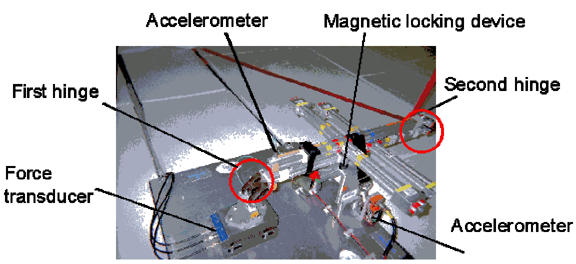

figure 10: Microgravity test bench

After first parabolas, it appeared that residual micro gravity (about 0,05g) were still impacting significantly the kinematic. Therefore, deployment with smallest inertia as possible have been proceeded to correlate the simulations without perturbation.

For further exploitation still under progress (correlation with global model), video images have been translated in graphs with angles versus time.
As the example shown in figure 11, which is a configuration with 2 arms representative of the application presented in the next section (IMSC mast), it was extremely interesting to observe the detailed opening kinematic that may exhibit several unlockings before reaching the ending open position.

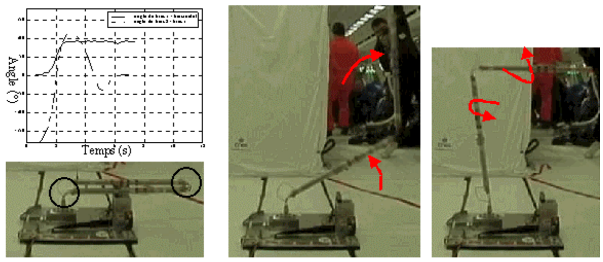

figure 11: Micro-gravity two hinges / two arms deployment

6. Technical data sheet

In this section, we give the detailed technical figures for the two industrial products currently under qualification.
At first, the generic version of the Carpentier Joint Hinge (CJH) was developed for a "recurrent" use to deploy micro-satellite solar arrays (SA). In this case, we underline the reliability of the hinges due to the vital function of solar panels.
The second application is in fact a first specific usage of the generic CJH and consists in a deployable arm for CNES DEMETER micro-satellite, which function is to move magnetic sensors away from the satellite (see figure 12). This demonstrates, with the using of the same hinge, the generic character of the technology.

6.1 Generic (SA) CJ Hinge

Compatible environmental conditions:

Temperatures	-75°C / +105 °C;
Out-gassing	CVCM < 0.1%; TML < 1%;
Radiation	15 krads

Design:

Dimensions	70 x 20 x 110 mm^3
Mass	85g

Performances:

Angular positioning accuracy*	< 1°
Driving torque	> 0.15 N.m
Stiffness in open situation	10^3 N.m/rad
Minimum holding torque	4.5 N.m

* This includes both manufacturing dispersions, effects of the launching environmental conditions (in particular vibrations) and effects of numerous opening / closing operations required for the qualification versus reliability. Much better accuracy may be obtained.

6.2 Deployable instrument carrier (IMSC mast for CNES DEMETER mission)

Compatible environmental conditions (with MLI):

Temperatures	-75°C / +105 °C;
Out-gassing	CVCM < 0.1%; TML < 1%;
Radiation	15 krads

Design:

Dimensions (close)	450 x 185 x 85 mm^3
Deployed length	1200 mm
Mass w/o instrument	1 kg
Mass of instrument	1.2 kg

Performances:

Angular positioning accuracy	1°
First resonance (open)	4.5 Hz
Deployment kinematic	< 20 s

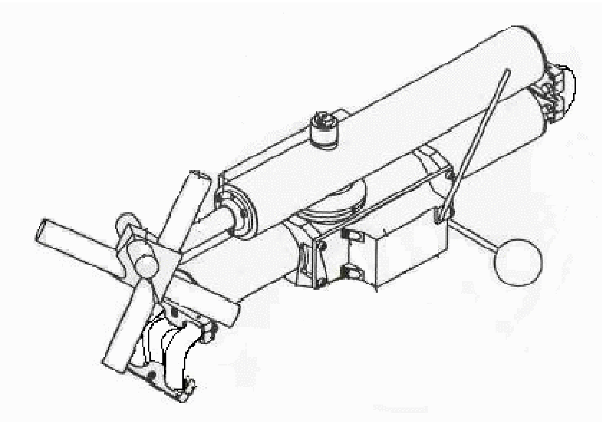

figure 12 : Specific DEMETER deployable instrument carrier

7. Conclusion

Without friction or contact, this new articulations constitute a low risk technology. The device is autonomous with an accurate positioning (angular and distance), a zero working play and a low cost.

Despite this simplicity and attractive character, the design control of such large strain spring strips is really difficult as shown.

However, thanks to both calculation and tests development, the whole characteristics have been checked and every model tuned. Moreover, tools have been elaborated which may be re-used for different designs.

After this component design period, we have to master the global mechanism (SA, IMSC mast or other) deployment kinematic; soon by the means of the present global model or later with a full dynamic large strain FEM simulation.

At present, qualification tests (vibration, thermal, and functional) are defined for every micro satellite equipments and the hinge are about to pass them.

Every manufactured hinge will also suffer acceptance tests which confirm main characteristics. Those are partially completed on the specific air cushion bench (figure 7) to define the driving torque an the stiffness in open situation.

In conclusion, the successful results obtained in this development allow to offer a "ready to use" industrial form of this new family of low cost hinge, for all deployment applications where kinematic control is not mandatory.

Future works are currently planned to both improve the quality of the numerical prediction (in particular some progress are needed to get a better view of the reliability margins, when submitted to combinations of static and dynamic stresses and also versus the number of admissible deployments), and to introduce, as a come back to the initial CNES / METRAVIB RDS development, damping solutions to limit the ending (locking) shocks.

Acknowledgement:
This work has been conducted on behalf of CNES (DEMETER programme).

References:

[1] A. Donzier & J. Sicre, **Self actuating damped hinge**, *7th European Space Mechanisms and Tribology Symposium, ESA/NIVR/CNES/ESTEC, Noordwijk, The Netherlands, 1997*

[2] French patent 96.14416, **Joint d'articulation automoteur, auto-verrouillant et amortissant et articulation équipée de tels joints**, European extension 97 947 079.6 and U.S. extension 09/297,095