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# Application of Flexure Structures to Active and Adaptive Opto-Mechanical Mechanisms

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## ABSTRACT

*Active and adaptive structures, also commonly called "smart" structures, combine in one integrated system various functions such as load carrying and structural function, mechanical (cinematic) functions, sensing, control and actuating Originally developed for high accuracy opto-mechanical applications, CSEM's technology of flexure structures and flexible mechanisms is particularly suited to solve many structural and mechanical issues found in such active/adaptive mechanisms. The paper illustrates some recent flexure structures developments at CSEM and outlines the comprehensive know-how involved in this technology. This comprises in particular the elaboration of optimal design guidelines, related to the geometry, kinematics and dynamics issues (for instance, the minimization of spurious high frequency effects), the evaluation and predictability of all performance quantities relevant to the utilization of flexure structures in space (reliability, fatigue, static and dynamic modeling, etc.), material issues and manufacturing procedures.*

**Keywords:** flexure structures, smart structures, kinematic mechanisms, actuators

## 1. INTRODUCTION

Conventional kinematic linking between mechanical parts in relative motion, such as slides and rotating bearings, suffer from a number of limitations: friction, stiction, wear, backlash, machining accuracy of the surfaces in contact, fretting or even cold welding under vacuum conditions, where special lubrication or surface coating may be necessary.

Flexure technology is not affected by these effects. It relies on deformable structures which constitute flexible bridges between stiff structural parts. It can be applied with advantage to all mechanisms where the amplitude displacement range remains moderate, for instance in the field of high resolution actuators, of isostatic suspensions used to compensate thermal effects, of mechanical impedance adapters (lever motion transmission devices), or in the field of vibration control systems. In each of these cases it can provide the maximum accuracy and stability without friction, stiction, wear or backlash and with minimal hysteresis

Flexure technology allows for several functions to be integrated in a few monolithic structures, that can be substituted to complex kinematic assemblies. In addition, monolithic integration opens the way to.

- Simpler integration, guaranteed alignments
- Miniaturization, leading the way to integrated micro-mechanisms
- Higher reliability, particularly valuable for space mechanisms

The name FLEXTEC has been used at CSEM to refer to the technology related to the design and application of flexure structures in general, with the deformable elements being shaped as rods, membranes or thin blades, often integrated into monolithic structures. The CSEM FLEXTEC technology comprises the capability of conceiving and modeling flexure mechanisms of great geometrical complexity together with the design modules and technology solutions to a number of critical areas that have been identified during a long-time experience with high-precision mechanisms, such as.

- Frictionless motion transmission, in many cases with amplitude reduction or amplification
- Guiding accuracy and linearity
- Minimizing mechanism mass and volume
- Modal analysis and frequency response
- Reduction of unwanted dynamic effects by balancing and damping
- Manufacturing technology issues, in particular applicable to complex monolithic structures.
- Various technology issues relating to surface quality and treatments, as well as material fatigue.

Therefore FLEXTEC is a comprehensive and very mature flexure structure technology which emcompasses much more than the bare utilization of flexible elements in mechanisms.

Historically the main field of applications of flexure structures is found in precision mechanisms. An earlier example is found in the micropositioning device$^2$ shown in figure 1 In this case the motion time constant to be is of about one second, which limited the problem to geometrical and quasi-static effects.

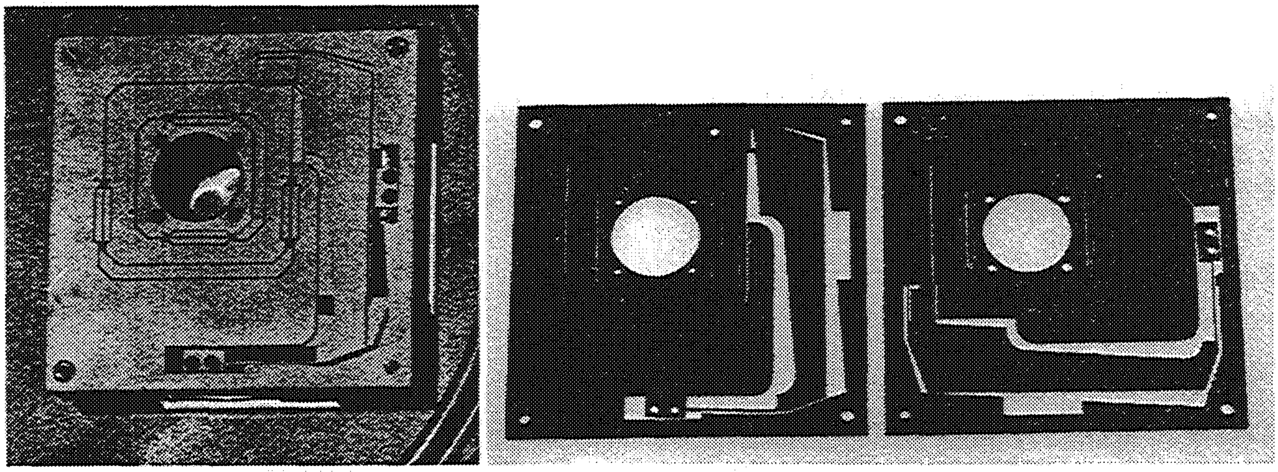


Figure 1 Micropositioning stage for an optical fiber system The mechanism is constituted by two monolithic components (each machined for a single plate such that it forms a system of multiple levers linked by flexure blades) which provide the guiding and reduction of the motion, as well as a complete uncoupling between the two main directions.

Today, however, the requirements for state-of-the-art flexure based mechanisms are more and more demanding in terms of dynamic bandwidth. While Figure 1 illustrates well the sophisticate geometry implemented in many FLEXTEC components, the manufacturing, lifetime, overall reliability aspects as well as considerations related to dynamic behavior take a larger and larger importance in state-of-the-art mechanisms and require innovative solutions even when geometrically simple flexure elements are implemented.

Consider for instance the classical compensated parallelogram which is frequently used for accurate linear guiding This simple geometry was developed at CSEM into a monolithic component (Figure 2) manufactured in aluminium alloy, suitable for a variety of designs that need the accurate linear guiding of an actuator or sensor.

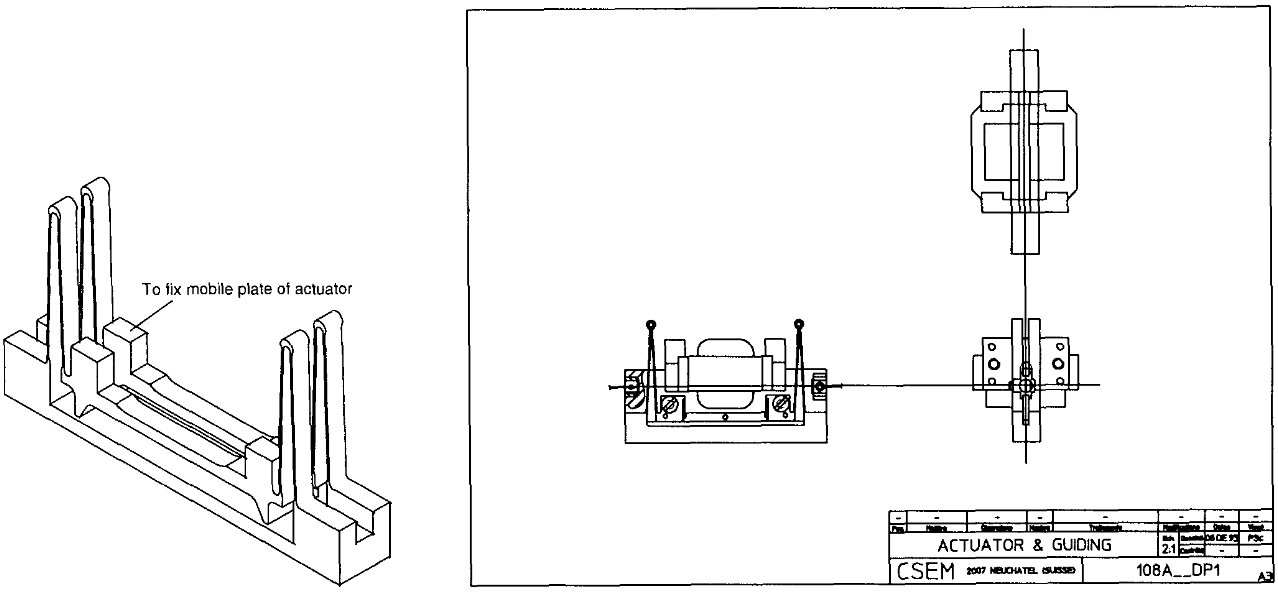


Figure 2 The monolithic "hairpin" compensated parallelogram. On the right side, illustration of the assembly with a linear electromagnetic motor.

The baseline model's size is about 4 cm and its motion range is ±1 mm but the concept is scaleable to smaller and large dimensions. The profile cutting is performed by electro-discharge machining (EDM): a particular EDM process was thereby selected in order to ensure the desired homogeneity of stiffness characteristics along the blades as well as the surface quality. The latter aspect is related also to the fatigue life, which is ultimately one of the very few real limitation of such flexure structures.

Another important aspect in the use of flexure structures concern spurious resonance modes and frequency response.

Compensated flexure parallelograms experience beside their fundamental mode, various higher order modes which will be excited at higher frequencies and can be quite undesirable for large frequency bandwidth applications. To reduce these unwanted dynamic effects, selective damping devices have been developed, which are located at the "hairpin" heads.

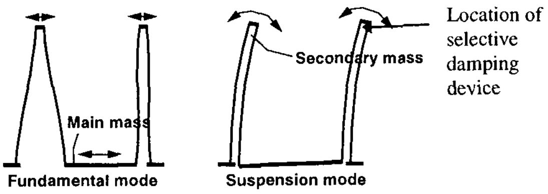


Figure 3 The fundamental mode and of one of the main disturbing modes.

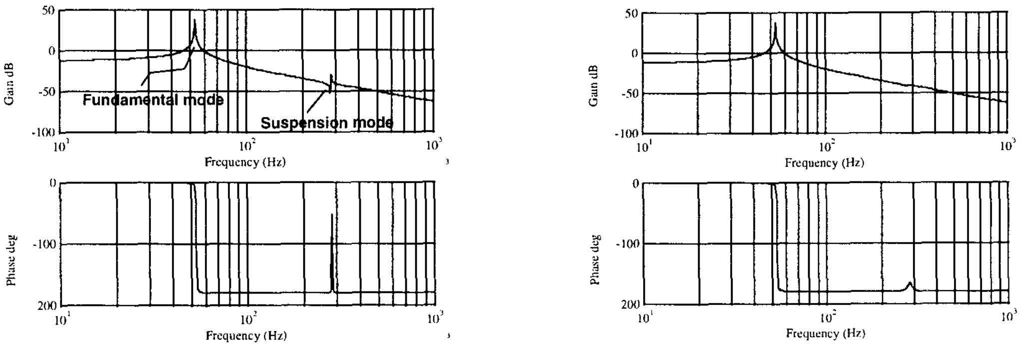


Figure 4 Left, frequency response function of the monolithic "hairpin" evaluated without damping. Right, frequency response function with selective damping of the suspension mode, which enlarges considerably the useful bandwidth of the system.

## 2. FLEXURE STRUCTURES AND ACTIVE MECHANISMS

Flexure structures find more and more applications as the key component of large bandwidth active and adaptive systems. These devices, also commonly called “smart” structures, combine in one integrated system various functions such as:

- load carrying and structural functions,
- mechanical (cinematic) functions,
- sensing,
- control,
- actuating (either in terms of forces or kinematic quantities)

Components that implement separately the sensing, control, and actuating functions have been developed in a large variety of situations applicable to closed-loop motion/force systems.

However, the key to overall performance predictability and the achievement of an high frequency bandwidth of the closed-loop controlled system is often found in the definition of the interfacing, load carrying and mechanical functions.

The characteristics of flexure structures (frictionless, no wear, etc.) are then particularly favorable in these respects and allow the design of integrated mechanisms implementing multiple functions:

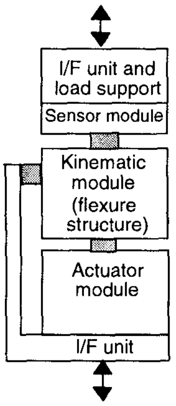


Figure 5 Schematic concept of a "smart structure" providing an actively controlled interface within a larger system

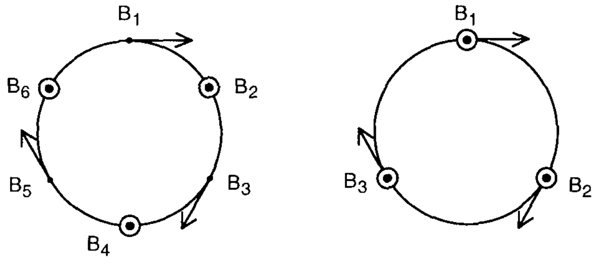
- Frictionless guiding, driven only by kinematic and stiffness constraints
- Uncoupling of degrees of freedom, which allows the making of isostatic structural systems, which are often the most direct way to a structural solutions and are also simpler to conceive and to predict.
- Motion amplification and reduction. In particular, the case occurs frequently in which the optimal ranges of the most suitable actuator, measuring devices and desired actuated motion at the interface, each taken individually, are widely different. The structural/mechanical components should then ideally be able to reconciliate these different ranges without loss of individual resolution or performance in the sensing, control and actuator stages. This implies the implementation of frictionless motion guiding/amplification/reduction stages as part of the intended system.

The following examples drawn from some recent projects will illustrate how these concepts are actually implemented in state-of-the-art active opto-mechanical mechanisms.

## 3. MECHANICAL ELASTIC ELEMENT FOR DAMPING AND ISOLATION (MEDI)

We describe here a flexure-based mechanical element designed as the main component of an isolation and damping interface between a spacecraft main structure and an optical terminal for inter-satellite laser beam communication.

For any solid in space, the position of the load interface is controlled by six independent degrees of freedom (DoF) which may be conveniently defined by one-dimensional translations of six attachment points $B_1$ to $B_6$ (see Figure 6A) or by two-dimensional translations of three attachment points $B_1$ to $B_3$ Figure 6B). Damping and isolation can then be achieved by transmitting separately the relative motion of the attachment point along the relevant direction.



A Control by six 1-DOF units B.Control by three 2-DOF units

Figure 6 Two possible configurations for 6-DoF load position definition

Consider the mechanism schematically represented in Figure 7: a monolithic flexure structure, essentially planar, is used to transmit a vertical vibration of the load interface through flexure hinges and a lever into a suitable isolator/damper unit The lever will generally allows adapting the optimum range of the isolator/damper unit to the actual amplitude of vibration. In order to avoid the hyperstatic overdetermination of the 6-DoF support (Figure 6), each single-degree-of-freedom mechanism must also provide the uncoupling of all other DoF's by assigning them a much lower stiffness than along its main direction. The flexure hinges that allow the function of the lever also provides the uncoupling of X-translation and Z-rotation. Additional flexure blades are obtained by reducing the thickness of the plate along the B-B junction, thus effectively uncoupling also the Z-translation and the X- and Y-rotations.

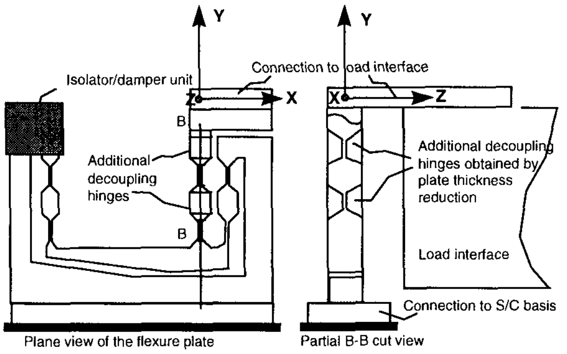


Figure 7 Basic concept for a single-degree-of-freedom isolation/damping element

For this particular project, the configuration of Figure 6B was preferred and implemented by a mechanical interface component combining stiffness and damping properties along two translation DoF's, in practice combining two single-DoF elements in one.

The original MEDI concept is schematically illustrated in Figure 8 below Its main component is a flexure structure by which the two in-plane displacements at the load interface are.

- uncoupled with respect to each other and with respect to the other DoF's,
- transmitted and amplified to two linear electromagnetic actuators, which act as passive dampers.

Also the support of the two linear actuators consist of a compensated flexure parallelogram (see Figure 2) so that the entire mechanism is free from friction or wear effects of any kind.

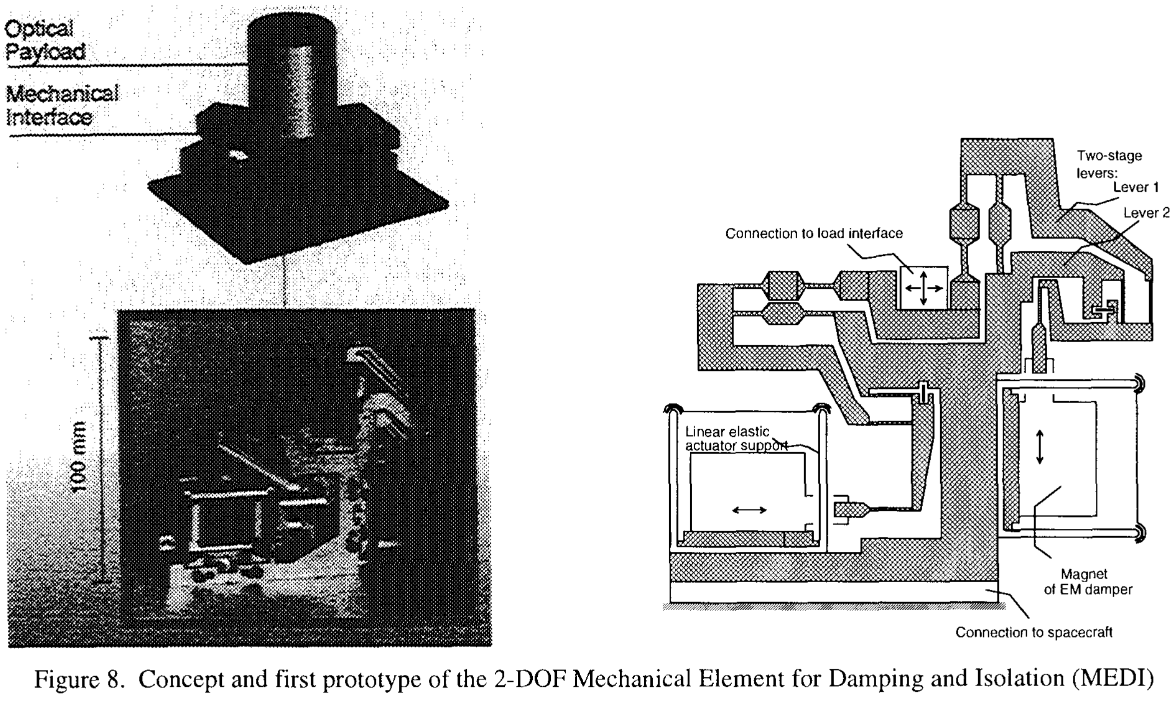


Figure 8. Concept and first prototype of the 2-DOF Mechanical Element for Damping and Isolation (MEDI)

The performance objective was a reduction of 20 dB in the vibration amplitude of the main modes of the optical terminal. This was achievable through mere passive damping by the electromagnetic actuators, although the MEDI concept is actually also very suitable for active control. A typical frequency response function for the 6-DoF system is given in Figure 9 here below.

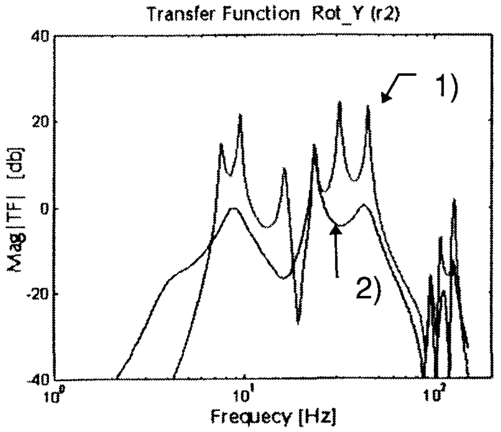


Figure 9 Response simulation: transmissibility of the terminal Y-rotation with respect to base Y-translation for 1) rigid interface, 2) MEDI interface

## 4. TWO-AXIS TILT FINE POINTING ASSEMBLY (FPA) MECHANISM

In the field of inter-satellite optical communications, a pivotless tip-tilt mechanism implementing the frictionless support of three linear actuators with a 3-rod suspension allows a fine pointing mirror to combine the high static deflection range with high frequency bandwidth and step response performance required by the acquisition and tracking functions.

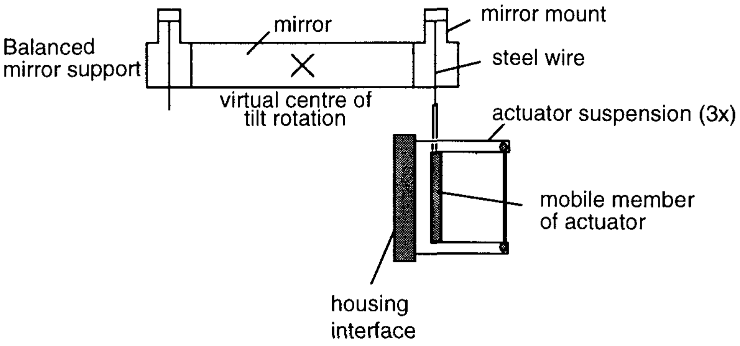


Figure 10 Principle of mirror suspension

The main technical specifications for this mechanism were two-axis deflection of an optical beam of 10 mm in diameter over a optical steering angle of +/-100 mrad, with a resolution above 1/20'000 and a small signal bandwidth of above 1 kHz. The FPA design was aimed at a small compact module (300 g / 300 $cm^3$)

Three linear actuators with ±1 mm range are placed in a star configuration and drive the mirror frame through intermediate steel wire flexible rods. The center of rotation (range: ±100 mrad optical) of the mirror and its frame coincide with their center of gravity. The three actuators are of the mobile magnet type and are suspended with the monolithic "hairpin" flexure parallelogram as already illustrated in Figure 2 In this case the issue of stray resonances, one of which is notably coupled with the main direction of movement, is particularly critical. A thin beam, squeezed by washers made of a special "rubber-like" material between the heads of the two 'hairpin' pairs, creates friction opposing the rotation of the hairpin tips which is characteristic for the higher order resonance mode shapes. All non parallel movements of the two 'hairpin' tip pairs are then inhibited. This device constitutes a selective damper and does not impair the fundamental actuator motion.

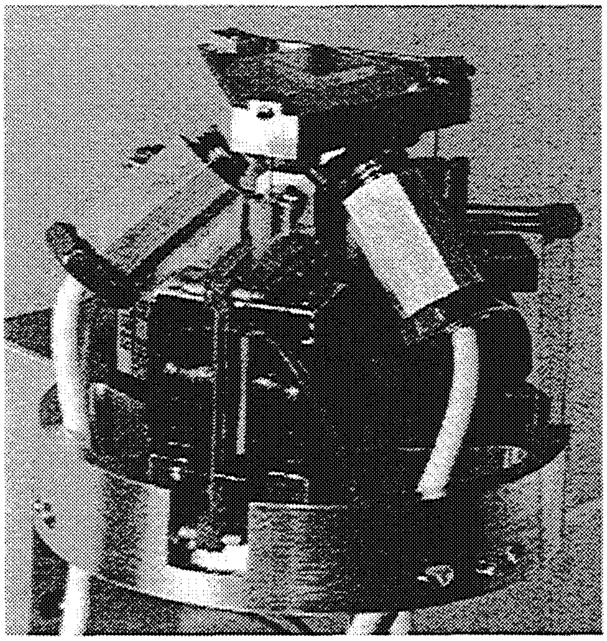


Figure 11 Photograph of two-axis tilt fine-pointing assembly prototype

The actuators are powered through linear amplifiers in push-pull configuration. The available voltage reserve for dynamic movements is a decade larger than the static voltage needed for full mirror deflection. The total static power dissipation of the three linear actuators is less than 0.35 W at full deflection.

The monolithic frame also integrates target surfaces for the Eddy current sensors. These are inclined at 45° with respect to the direction of movement of the corresponding actuator to accommodate for their reduced detection range. Nonetheless these sensors achieve a resolution of 1:100'000 for a reduced bandwidth, within an overall bandwidth of 20 kHz. The whole mechanism weighs 300 grams, and fits into a cylindrical volume of Ø 65x65 mm.

The complete fine pointing mechanism has been tested in open loop to identify the system parameters. As three mechanical oscillators are mounted in parallel, sharing a common oscillating mass, the global resonance modes are separated into one axial and two rotational modes at different frequencies. With a controller design adapted to the identified system parameters, the small signal step response (0.1 mrad optical) in closed loop shows an equivalent time constant of 130 μsec, corresponding to a 1.2 kHz bandwidth.

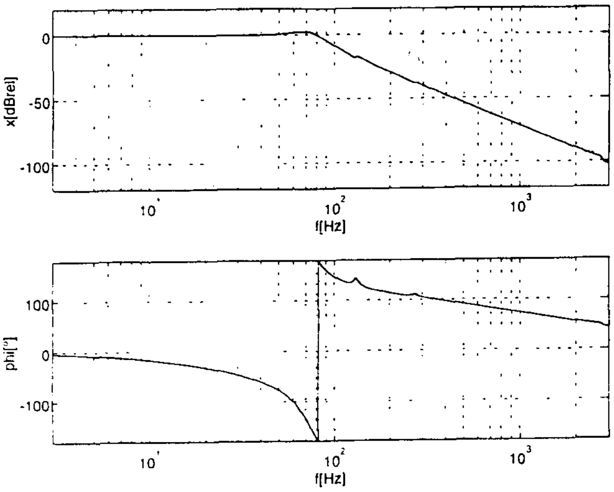


Figure 12 Measured open-loop frequency response

## 5. ACTIVE COMA-FREE SECONDARY MIRROR

This project concerned an active secondary mirror (M2) unit to achieve active high pointing stability of large spaceborn telescopes. Current demands on spacecraft structural and pointing stability requirements are increasing to the level that conventional spacecraft positioning techniques must be reviewed. Typical space science observation missions operating down to the near UV wavelength region will require line of sight stability in the milli-arcsecond accuracy or even below. New trends try to incorporate high frequency stabilisation elements at positions where the moving parts have low inertia, rather than attempting fully stabilize the spacecraft to this high pointing accuracy. The project considered a one meter space telescope dedicated for planetary research, with an actuated secondary telescope mirror (M2) of 100 mm in diameter as active pointing element.

The main technical requirements for the M2 mechanism were: the coma-free rotation around a point situated some 200 mm behind the secondary mirror apex, the tip/tilt resolution (better than 1:20'000) implied to sense displacements in the nanometer region, and a minimum bandwidth for vibration rejection of more than 100 Hz.

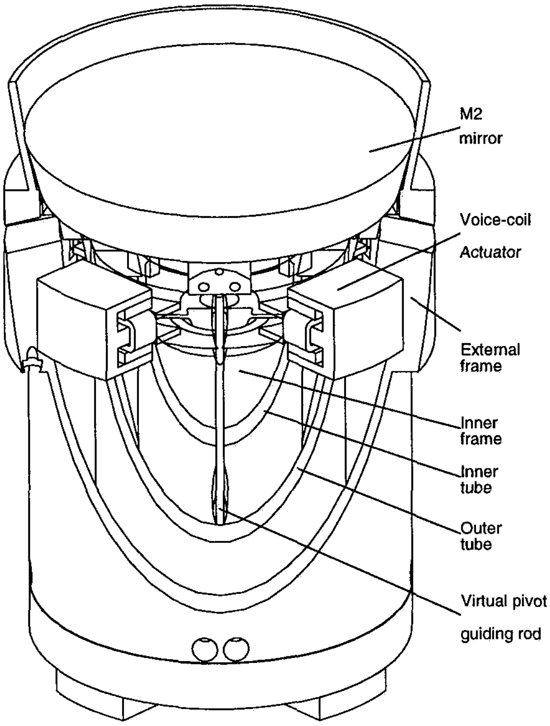


Figure 13 3-D view of the M2 mechanism

The concept developed by CSEM (figs. 13 and 14) is based on two coaxial cylinders connected by transverse (flexible) membranes, acting as a double reduction lever for the transmission and inversion of two-axis transverse motion from an actuator ring of mass $\mu$ to the M2 mobile mirror stage. By suitable choice of the reaction mass $\mu$, the transverse reaction induced by any commanded M2 motion on the telescope structure is compensated by the inverse motion of the actuator ring. The lever reduction factor is equal to 10, corresponding to a mass $\mu$ of the order of 50 grams, for a M2 mobile stage mass of 500 grams. The ring is moved by four push-pull voice coil actuators, two per axis in order to include redundancy, within a transverse range of $\pm 500$ $\mu$m corresponding to $\pm 50$ $\mu$m at mirror stage level.

In order to ensure tip-tilt rotation, the M2 mirror stage is further guided by four concentric rods pointing to the virtual center of the coma-free rotation point. Very high axial rotation stiffness is obtained by attachment to the lever cylinders without other structure. This combination has the advantage of maintaining the suspended mass to a minimum and the overall system can be contained into a very reduced volume (see Figure 14).

The actuators are of voice coil type, and the sensor are of the Eddy current proximity type, with capacitive-type sensors as a backup solution.

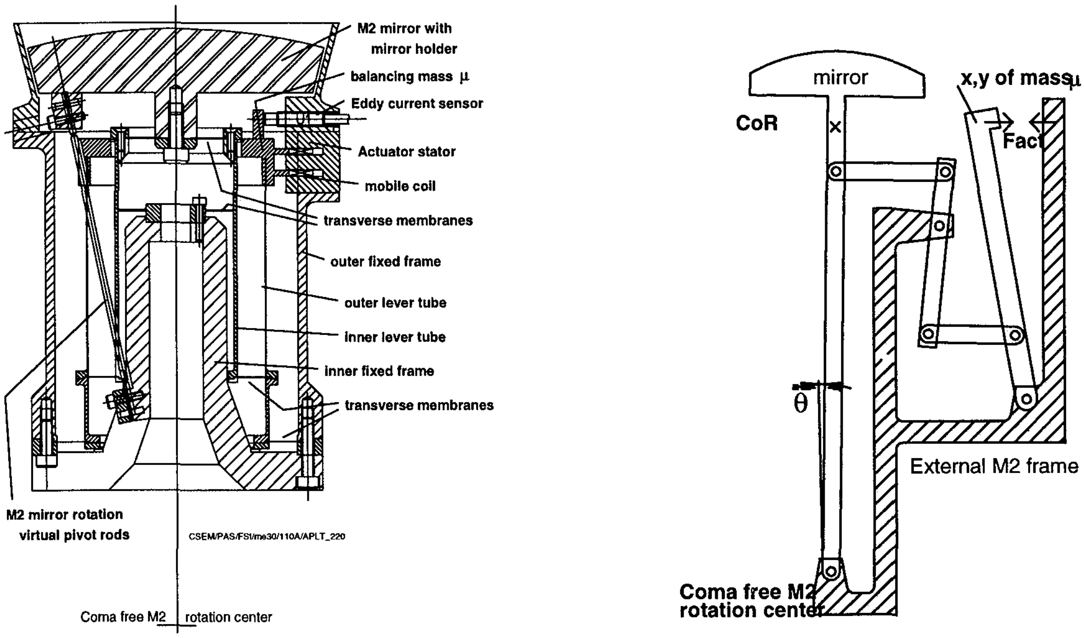


Figure 14 Basic flexure kinematics of the coma-free secondary mirror unit: note that the rotation point is far outside the envelope of the system. Right, the corresponding kinematic theoretical mechanism, which if based on stiff members and hinges would require a much large volume

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