Piezoelectric Stewart platform for general purpose active damping interface and precision control

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Abstract

This paper discloses a stiff active interface wherein a six degree of freedom Stewart platform, a standard hexapod with a cubic architecture, is used to actively increase the structural damping of flexible systems attached to it. It can also be used to rigidly connect arbitrary substructures while damping them. Each leg of the active interface consists of a linear piezo electric actuator, a collocated force sensor and flexible tips for the connections with the two end plates. By providing the legs with strain or elongation sensors, this active interface can also be used as an interface with infinite stiffness at low frequency (i.e. for machine tools), a 6 d.o.f. positioning and steering device for space applications as well as a microvibration isolator. The translation and rotation strokes of the interface are 90, 103 and 95 μm in the *x, y* and *z* directions respectively and 1300, 1150 and 700 μrad around the *x, y* and *z* directions respectively.

1. Introduction

Future astronometric missions will require improved angular resolution capabilities that are at least one order of magnitude better than the Hubble Space Telescope (0.05 μrad). This required angular resolution can only be achieved with either very large optics or interferometric devices, where the signals coming from several independent telescopes are combined to increase the global resolution. Space constraints such as weight or launcher size make interferometric devices attractive despite their increased complexity. To achieve the predicted resolution, the pointing error requirement of the individual telescopes is as low as a few nanoradians and their relative position must be preserved within a few nanometers. Usually, the optical path difference between the various sub-systems is monitored by a sophisticated laser metrology system and controlled by means of optical delay lines.

One concept for future space interferometers consists in mounting the various telescopes on a truss whose dimensions can be very large (i.e. 50-250m for IRSI-DARWIN). Because of the space constraints, this truss will be very flexible and subjected to a wide variety of static and dynamic perturbations (thermal loads, attitude control, reaction-wheels, cryo-coolers...). As the optical delay lines will compensate static and quasi-static perturbations, the main requirement on the supporting truss is rather stability than precision. This specification on the structural stability for scientific space missions has triggered extensive researches in the area of the active damping of flexible structures. These have led to numerous solutions, most of them based on the integration of SMART actuators and sensors in the structure itself. Several methods have been investigated by the present authors for the active damping of space structures:

- Replacing some bars of the truss by active struts [1].
- Integrating laminar piezo-electric patches [2]
- Using inertial actuators [3]
- Using active tendons [4]

In this paper, a stiff active damping interface is proposed. It can be used either as a support for payloads or to connect arbitrary substructures. It has the ability to introduce damping in the mechanical system attached to it while remaining stiff. The active interface consists of a six-degree of freedom Stewart platform, a standard hexapod with a cubic architecture. Each leg of the active interface is made of a linear piezo electric actuator, a collocated force sensor and flexible tips for the connection with the two end plates. The control architecture is based on six local/decentralized Integral Force Feedback controllers. By providing the legs with strain or elongation sensors, this active interface can also be used as an interface with infinite stiffness at low frequency (i.e. for machine tools), a 6 d.o.f. positioning and steering device for space applications as well as a microvibration isolator.

2. Assembly of the interface

The design of the proposed Stewart platform is based on the cubic configuration [5]. The dexterity and accuracy of the mechanism depends very much on the nominal geometry of the hexapod [6, 7]. The mobility of the platform is driven by the elongation of the legs [8, 9].

As far as the elongation of the legs in this application is very small, the kinematics configuration remains almost unchanged. Thus the Jacobian matrix relating the motion of the platform to the elongation of the legs remains constant and can be evaluated from the nominal configuration depending only on the length of the legs [10]. To achieve good performances in active vibration damping and precision pointing, several characteristics should be taken into account in the design of the Stewart platform:

- Uniformity of control capability in all directions.
- Uniform stiffness in all directions.
- Uniform cross-coupling amongst actuators.
- Simple kinematics and dynamic analysis.
- Simple mechanical design.
- Availability of collocated actuator/sensor pairs.

The cubic configuration was invented by the Intelligent Automation Inc. (IAI) to fulfill most of the above properties [5]. The nominal configuration is obtained by cutting a cube by two planes as indicated in Fig.1. The two planes constitute both the base and the mobile plates of the Stewart platform. The edges of the cube connecting the plates represent the six legs of the hexapod.

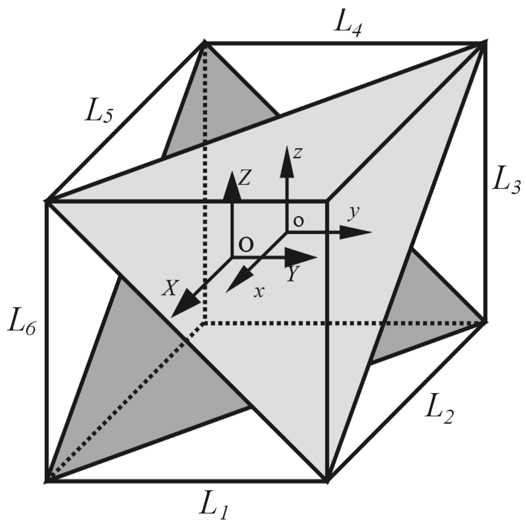

Figure 1: Cubic configuration of Stewart platform

Fig.2(a) shows a picture of the complete Stewart platform and Fig.2(b) shows the same but with removing the upper plate. The two plates are circular aluminum plates with a thickness of 20 mm and a diameter of 250 mm. The plates are connected to each other by six active legs; the legs are mounted in such a way to achieve the geometry of cubic configuration (as explained before). Each active leg consists of a force sensor (B&K 8200) and an amplified piezoelectric actuator (Cedrat Recherche APA50s) as shown in Fig.3. To avoid the problems of friction and backlash in the joints, flexible tips were used instead of spherical joints. These flexible tips have zero friction, zero backlash, high axial stiffness and relatively low bending stiffness. It will be shown that the existence of this bending stiffness makes a limitation for the control authority because it shifts the transmission zeros to a higher frequency, which will decrease the damping effect expected from each closed-loop pole.

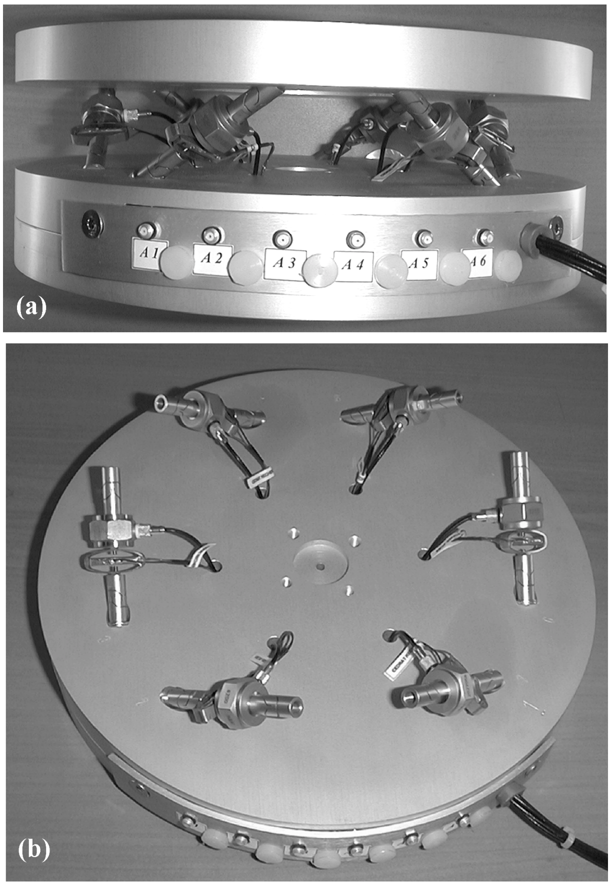

Figure 2: The Stewart platform (a): complete hexapod, (b): the hexapod with the upper plate removed

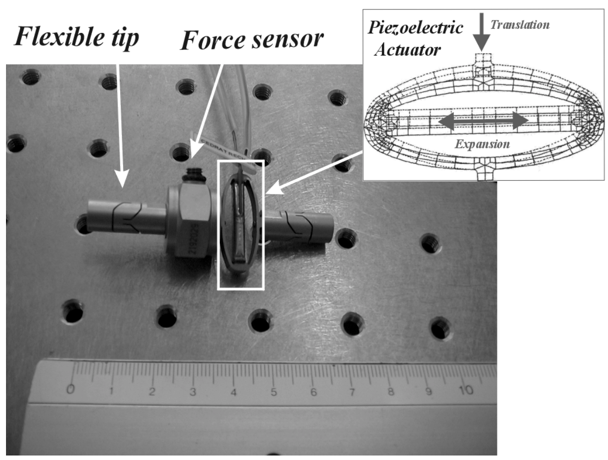

Figure 3: The active leg assembly of Stewart platform

3. Pointing performance

In the proposed Stewart platform, the total length of the legs is 135mm, including the thickness of the two end plates. The Jacobian matrix J relating the extension of the piezo actuator q to the global degree of freedom χ is:

$$\dot{q} = J\dot{\chi} \tag{1}$$

where $\dot{\chi} = (\mathrm{v}^T, \omega^T)^T$, v and ω being respectively the translation and rotation velocities of the top plate. In this case the Jacobian matrix is:

$$J = \begin{pmatrix} 0.408 & 0.707 & 0.577 & 0 & -0.064 & 0.078 \\ 0.408 & -0.707 & 0.577 & 0 & -0.064 & -0.078 \\ -0.816 & 0 & 0.577 & 0.055 & 0.032 & 0.078 \\ 0.408 & 0.707 & 0.577 & 0.055 & 0.032 & -0.078 \\ 0.408 & -0.707 & 0.577 & -0.055 & 0.032 & 0.078 \\ -0.816 & 0 & 0.577 & -0.055 & 0.032 & -0.078 \end{pmatrix}$$

Neglecting the bending stiffness of the flexible tips in the legs, we find that the general stiffness matrix of the hexapod $K = J^T K_l J$, where K_l=0.87 10^6N/m is the axial stiffness of each leg. The global stiffness of the proposed Stewart platform is

$$K = 10^6 \begin{pmatrix} 1.74 & 0 & 0 & 0 & -0.068 & 0 \\ 0 & 1.74 & 0 & 0.068 & 0 & 0 \\ 0 & 0 & 1.74 & 0 & 0 & 0 \\ 0 & 0.068 & 0 & 0.0106 & 0 & 0 \\ -0.068 & 0 & 0 & 0 & 0.0106 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0.0319 \end{pmatrix}$$

The maximum stroke of the actuator is 55 μm (±27.5 μm) but this motion is magnified by the mechanism of the Stewart platform to give the motion of the mobile plate. Table 1 shows the maximum pure translations and rotations in the different degrees of freedom. The total stroke of the piezo actuator is 55 μm, for symmetric operations, the maximum stroke of the actuator s = ±27.5 μm. δq_i is the elongation in the i^{th} leg given in μm and DOFs are the maximum pure translations (in μm) and rotations (in μrad) travelled by the center of the upper plate.

DOFs	δq_1	δq_2	δq_3	δq_4	δq_5	δq_6
x_{pure} = 33.7	s/2	s/2	-s	s/2	S/2	-s
y_{pure} = 38.9	s	-s	0	s	-s	0
z_{pure} = 47.5	s	s	s	s	s	s
θ^x_{pure} = 498	0	0	s	s	-s	-s
θ^y_{pure} = 431	-s	-s	s/2	s/2	s/2	s/2
θ^z_{pure} = 350	s	-s	s	-s	s	-s

Table 1: Maximum pure translations (in μm) and rotations (in μrad) traveled by the moving plate and the corresponding leg configurations (s = ±27.5 μm)

The previous table shows the motions for the half-stroke of the actuator, which means that the full translations along x, y and z (in μm) and the rotations around them (in μrad), respectively, are (67, 79, 95, 996, 862, 700).

The signal to noise ratio of commercial power electronics for piezo actuators is about 80dB. As the position noise is linearly proportional to the electrical noise, the resolution of piezoelectric actuator is about 0.01% of its stroke. In present case, the piezo noise for a 55μm stroke actuator should be 5.5$\mathrm{nm_{rms}}$. As the pointing commands in the hexapod are amplified and transferred into motion of the upper plate, the noise is also amplified. To find the RMS values of the noise on the platform, consider that

$$\delta x_i = \sqrt{\sum_j \left(J_{ij}^{-1}\right)^2 \delta q_j^2} \tag{2}$$

where δx_i is the amplified noise in the i^{th} direction of motion of the platform, δq_j is the noise produced by the j^{th} leg and J_{ij} is the Jacobian value relating the previous two. Table 2 gives the resolution of the platform.

DOFs	Resolution
x_{noise}	4.5 $\mathrm{nm_{rms}}$
y_{noise}	5.2 $\mathrm{nm_{rms}}$
z_{noise}	3.9 $\mathrm{nm_{rms}}$
θ^x_{noise}	66.4 $\mathrm{nrad_{rms}}$
θ^y_{noise}	57.5 $\mathrm{nrad_{rms}}$
θ^z_{noise}	28.7 $\mathrm{nrad_{rms}}$

Table 2: Resolution of the six degrees of freedom of the platform

4. Active damping results

In order to evaluate the damping performances of the interface, the hexapod is connected to a flexible payload, a 150 cm long steel truss structure (Fig.4). A decentralized integral force feedback control scheme was applied to the experimental system. The six independent controllers have been implemented on a DSP board. The six loops have been closed separately and, although the control loops were independent, the feedback gains used in all the loops are identical. Fig.5 presents some experimental results. The time response shows the signal from one of the force sensors located in the legs; the truss is subjected to an impulse at middle height, without then with control. The frequency responses (with and without control) are obtained between a perturbation signal applied to the piezo actuator of one the leg and its collocated force sensor.

One can see that fairly high damping ratios can be achieved for the low frequency modes (4-5Hz) but also significant damping in the high frequency modes (40-90Hz). Unfortunately, the results on the torsion mode have been disappointing, probably due to the flexion stiffness in the flexible tips.

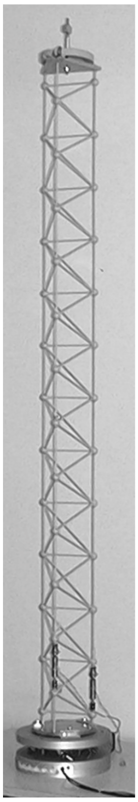

Figure 4: Experimental setup

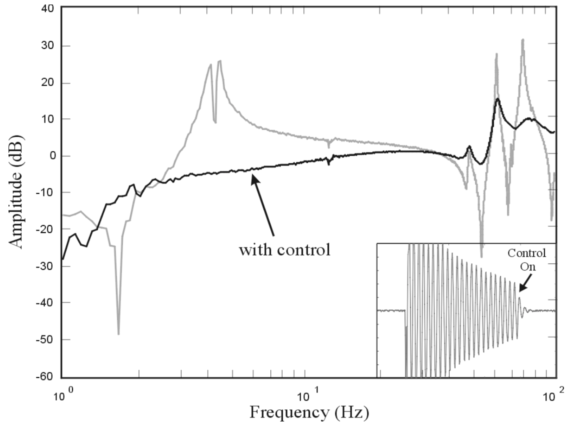

Figure 5: Experimental time response and frequency response function of the truss mounted on the active interface.

5. Conclusions

This paper describes a new generation of hard mount Stewart platform. Some terrestrial and spatial applications were discussed with deeper overview on the possibility of using such a device in the precise pointing and vibration suppression of the highly sensitive spatial equipment. The second part of this paper shows the design, assembly and configuration of the hexapod explaining the principle and advantages of the cubic configuration. The kinematics and pointing performance of the interface was discussed in the third section. Analytical and numerical analyses were established to show how the different degrees of freedom of the mobile plate are related to the elongation of the six legs of the hexapod. Part four concentrated on the dynamics and the theoretical relations of the applied control technique. Eventually, some experimental results were shown in the last section explaining the performance of the device in active vibration damping.

Acknowledgment

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