Frequency Variation for the Purpose of Vibration Isolation

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***Abstract* – This paper demonstrates a new idea for vibration isolation. This idea depends on using active embedded control system with proportional plus integral compensator to reduce the stiffness and corner frequency of the isolator in active state while keeping high rigidity in passive state. This type of isolation has proven the ability to reduce the frequency modes to about 50% of the natural frequencies in passive state. This document will concentrate on single axis isolation although the mentioned isolator can be used as one active leg (strut) in six-axis isolators (Stewart platforms).**

I INTRODUCTION

Vibration isolation is becoming more and more stringent as the mechatronic systems are advancing and developing in space and ground applications. In order to obtain high performance from the vibration isolator, the corner frequency should be as low as possible [1, 2]. This leads the researchers to reduce the frequency by the following ways:

- Increase the mass of the equipment needed to be isolated or to reduce the stiffness to have soft isolation mounts either in single or two stages [3].
- Using a hybrid, soft and stiff mounts, with velocity feedback using phase lag and phase lead compensators on the unity gain points [4].
- Using totally stiff mount isolator with proportional plus integral feedback compensator.

Increasing the mass or inertia is problematic specifically in space applications because of the limitations on the weights can be transported to space. On the other hand, reducing the mount stiffness leads to difficulties in stability of the systems under normal loads in ground applications and ground tests. This paper will present a stiff mount that can be used for the purpose of vibration isolation. This stiff mount can be softened in active ways by using a simple PI (proportional plus integral) compensator. This stiff mount isolator can be used as the active strut (leg) of six-axis Stewart platform interface with piezoelectric actuators to reduce the stiffness and corner frequency of the interface in active mode while keeping high stiffness and rigidity in passive mode [2]. The first part of this paper will present the main objectives of vibration isolation and some previous efforts in soft and stiff isolation mounts. The theory, design and control of the suggested isolator will be demonstrated in the next part. More concentration will be on simulation results for the system. Experimental results on single and multi degrees of freedom systems using the idea of frequency variation will be shown in the last part. Finally, conclusions and future work will be discussed and the end of the paper.

II OBJECTIVES AND BACKGROUND

Figure 1 shows a picture of the ULB piezoelectric Stewart platform; Fig.1(a) shows the complete Stewart platform where the connectors are the inputs to the actuators and the wires are the outputs of the sensors; Fig.1(b) shows the hexapod with the upper plate removed to show the details and configuration of the legs. The hexapod consists of two parallel plates connected to each other by six active legs. The legs are mounted in such a way to achieve the geometry of cubic configuration. Each active leg consists of a force sensor (B&K 8200), an amplified piezoelectric actuator (Cedrat Recherche APA50s) and two flexible joints as shown in Fig.2. In the ideal situation, the hexapod needs to be hinged using spherical joints, but to avoid the problem of friction and backlash, flexible tips are used instead of spherical joints. These flexible tips have the following properties: zero friction, zero backlash, high axial stiffness and relatively low bending stiffness. The bending stiffness of these joints makes a limitation for the active control authority, because it shifts the transmission zeros which decreases the closed-loop performance [5, 6].

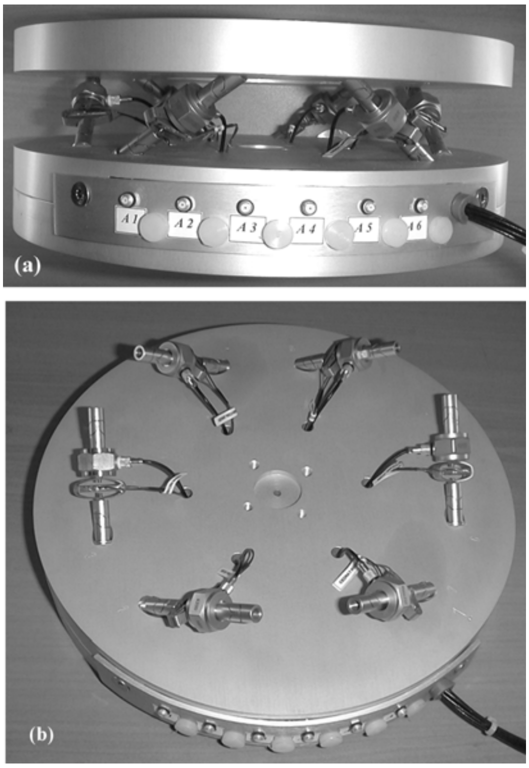

Figure 1
ULB stiff Stewart platform

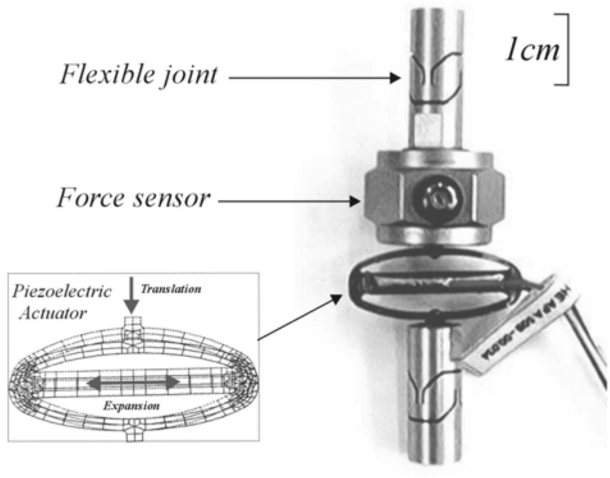

Figure 2
Active leg of the ULB stiff Stewart platform

This interface has been used so far for the purposes of active damping and precision pointing only because of high stiffness and rigidity in the legs.

Several previous works has been invented to reduce the frequency of the isolation system using piezoelectric actuators. A single-axis vibration isolation system (Quiet pier) has been invented by the Technical Manufacturing Corporation (TMC) to solve the problem of the high corner frequency when using a piezoelectric actuator [4]. This system (in Fig.3) consists of a piezoelectric actuator represented by its extension $\boldsymbol{\delta}$ and stiffness $\boldsymbol{k}$, a payload mass $\boldsymbol{m_1}$ and an intermediate passive mount. The intermediate mount consists of a mass $\boldsymbol{M}$ and an elastomer with a stiffness $\boldsymbol{k_1}$ and a damping factor $\boldsymbol{c_1}$. The isolator frequency formed by the stiffness of the actuator $\boldsymbol{k}$ and the intermediate mass $\boldsymbol{M}$ is equal to 1000 Hz. The passive elastomer (represented by the spring $\boldsymbol{k_1}$ and the dashpot $\boldsymbol{c_1}$)

forms a new resonance with the payload mass $\boldsymbol{m_1}$ equals to 20 Hz. The two stiffness values, $\boldsymbol{k}$ and $\boldsymbol{k_1}$ are in series; this results in having the corner frequency of the system corresponding to the lower stiffness $\boldsymbol{k_1}$. A geophone velocity sensor is installed at the intermediate mass $\boldsymbol{M}$. The active control strategy is based on feeding the signal of the geophone back to the piezoelectric actuator after being properly filtered and amplified. This inertial feedback leads to quiet the intermediate mass $\boldsymbol{M}$ which results in isolating the motion $\boldsymbol{x_{c1}}$ of the payload mass $\boldsymbol{m_1}$ from the seismic disturbance $\boldsymbol{x_d}$.

Figure 4 shows the open-loop FRF between the voltage input to the actuator and the velocity measured by the geophone. The gain (magnitude) of the open-loop transfer function climbs at 40 dB/decade then levels at 40 dB when it reaches to 4.5 Hz at the resonance frequency of the geophone. The geophone acts as a second order high-pass filter that cuts the signals off below 4.5 Hz. The high frequency attenuation is achieved by locating a low-pass filter at 300 Hz (before the resonance of the piezoelectric actuator). A lag compensator is placed near the low frequency unity gain point (at 0.2 Hz), and a lead compensator is placed near the high frequency unity gain point (at 350 Hz). The advantage of adding this lag-lead compensation is to reduce the amplifications (overshoots) that appear at the unity gain points when the loop is closed. Figure 5 shows the transmissibility FRF between the seismic disturbance displacement $\boldsymbol{x_d}$ and the sensitive payload displacement $\boldsymbol{x_{c1}}$. The overshoots caused by inertial feedback can be seen clearly on the two unity gain points of the closed-loop FRF. Using a phase lag compensator near the low unity gain frequency reduced the overshoot which means better transient response and lower settling time. Similarly, using a phase lead compensator near the upper unity gain frequency could increase the phase margin which improves the stability conditions of the system. One can see clearly that despite using a hard piezoelectric actuator, the passive vibration isolation occurs here near the low frequency resonance of the passive mount (20 Hz). Moreover, the closed-loop active vibration isolation occurs much lower than that (at 0.2 Hz) leading the system to have a high isolation performance for a wide band of disturbances [2, 4].

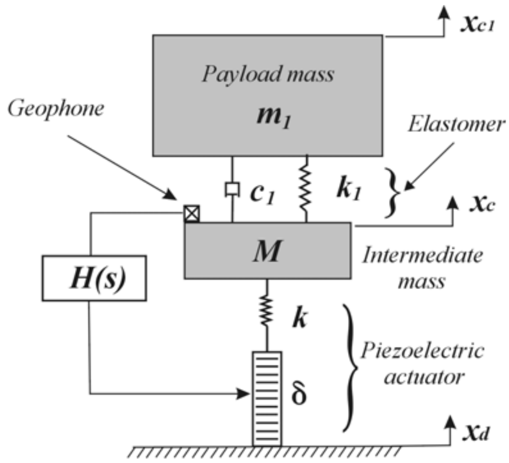

Figure 3
TMC single axis isolator using intermediate soft mount

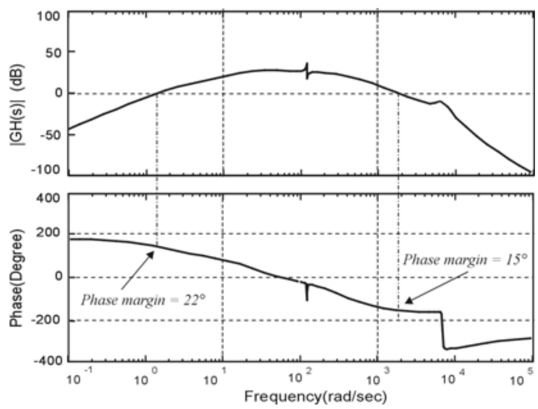

Figure 4
Open-loop FRF of TMC single axis isolator

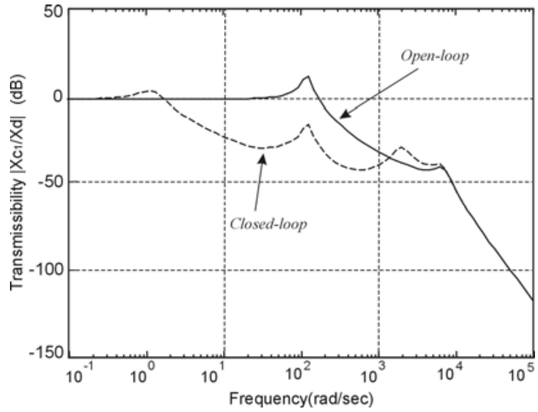

Figure 5
Transmissibility FRF of TMC single axis isolator

III SINGLE AXIS PIEZOELECTRIC ISOLATOR USING PI CONTROLLER

Consider the schematic drawing shown in Fig.6. This figure represents a vibration isolation interface with the disturbance source (mass ***m***), the sensitive payload (mass ***M***), a force sensor ***F*** and a piezoelectric actuator represented by its stiffness ***k*** and extension ***δ***. Proportional plus Integral (PI) feedback controller is used here to reduce the corner frequency and improve the response. This system can be used as an active strut for the previously mentioned Stewart platform. As an application for the frequency reduction, one can imagine adaptive structures that can change their resonance frequency instantaneously to avoid being excited when the excitation frequency approaches a resonance.

This schematic drawing represents the active leg of Stewart platform shown in Fig.2 where the active feedback controller is applied by acquiring the signal measured by the force sensor and feeding it back to the piezoelectric actuator after being filtered and compensated with PI compensator. The governing equation of motion for the system in Laplace transform is:

$$Ms^2 x_c = -ms^2 x_d = k(x_d - x_a) = F$$

And

$$\delta = x_c - x_a$$

The open-loop FRF between the extension of the piezoelectric stack in the piezoelectric actuator ***δ*** and the output of the force sensor ***F*** reads:

$$\frac{F}{\delta} = k\frac{Mms^2}{Mms^2 + k(M+m)}$$

Applying a force feedback control strategy using a proportional plus integral compensator, the control law reads:

$$\delta = \frac{g}{ks}(1 + as)F$$

Here ***ga*** is the proportional gain and ***g*** is the integral gain. The root locus for the closed-loop poles of this system is shown in Fig.7; it shows that increasing the loop gain decreases the frequency of the closed-loop poles. If the proportional term is used alone, the poles will move on the imaginary axis towards the origin but this means the risk of destabilizing the system at any instant. The use of the integral controller here pushes these poles deeper to the left half plane increasing the stability.

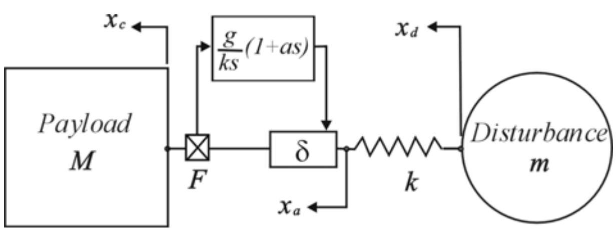

Figure 6
Single axis piezoelectric isolator with PI feedback

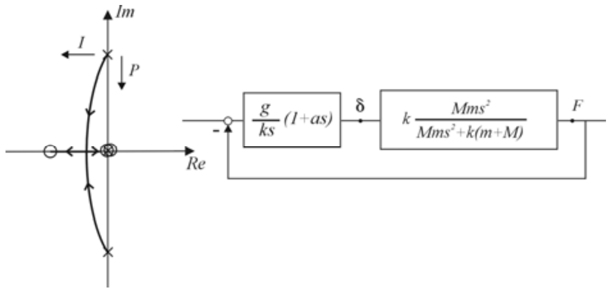

Figure 7
Root locus of single axis piezoelectric isolator with PI feedback

From the analytical calculation, the intermediate displacement x_a is:

$$x_a = \frac{sx_c + g(as+1)x_d}{s + g(as+1)}$$

From the foregoing equations, one can calculate the transmissibility FRF between the disturbance displacement and the payload displacement, which is equal to:

$$\frac{x_c}{x_d} = \frac{1}{s^2[(1+ga)/\omega_n^2] + s[g/\omega_n^2] + 1}$$

Where ω_n is the natural frequency of the system. This implies that the corner frequency ω_c of the system is determined by the proportional gain of the compensator:

$$\frac{1}{\omega_c^2} = \frac{1+ga}{\omega_n^2}$$

The damping of the system is determined by the gain ***g*** of the compensator

$$\frac{g}{\omega_n^2} = \frac{2\xi}{\omega_c}$$

If ω_n is much larger than ω_c then

$$\frac{\omega_n^2}{\omega_c^2} \approx ga$$

$$\frac{ga}{k} = \frac{1}{M\omega_c^2} = \frac{1}{k^*}$$

Here ($1/k^*$) is the closed-loop flexibility of the system and is proportional to the gain. From the foregoing analysis, one can see that the closed-loop stiffness of the system is inversely proportional to the control gain; in other words, if one increases the proportional gain, the stiffness is reduced.

IV SIMULATION RESULTS

The system shown in Fig.6 has been simulated using Matlab software. The simulation was based on the previous analysis of the system taking the mass ***m*** as 1.1 kg, the mass ***M*** as 1.7 kg and the stiffness of the piezoelectric actuator ***k*** as $1*10^7$ N/m. Figure 8 depicts the root locus prediction that the poles should follow when the control loop is closed. The root locus shows that the poles will remain in the left hand side of the s-plane which means that the system is unconditionally stable. On the other hand, the loop of the plot is not moving in a circular shape which means that when we increase the gain of the controller the distance

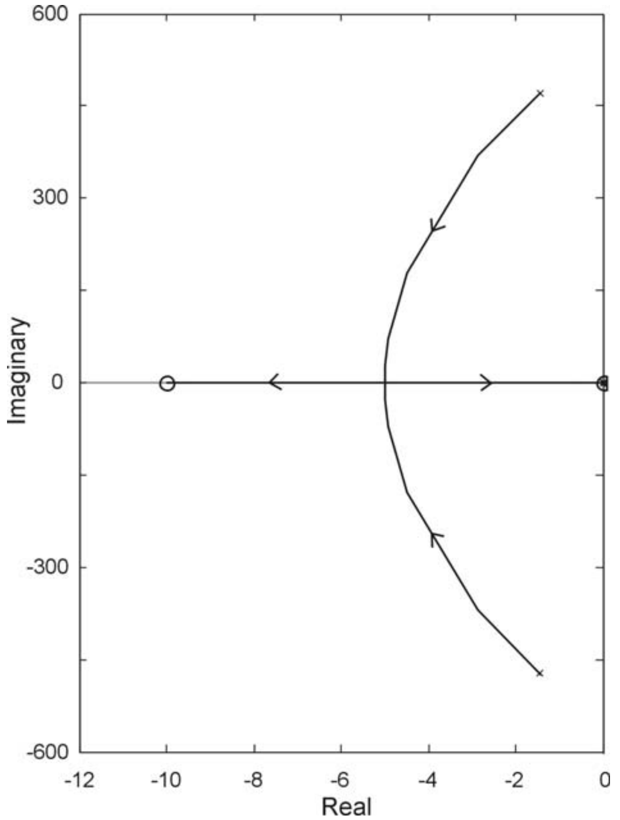

Figure 8
Root locus prediction

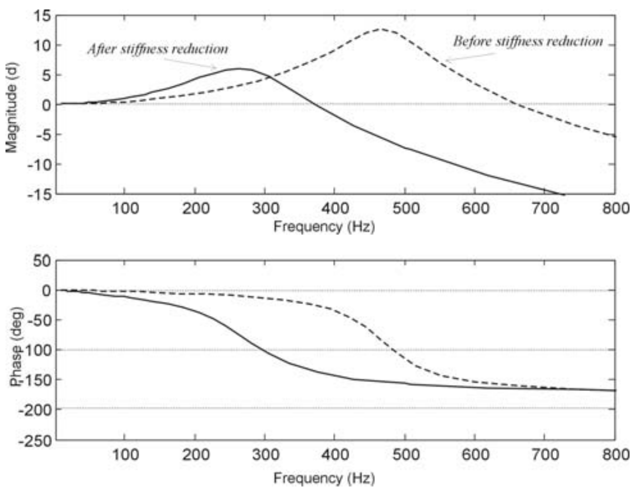

Figure 9
Predicted transmissibility from simulation results

between the pole and the origin will be shorter leading to slow down the poles or to reduce the frequency of the corresponding mode. Theoretically, the poles will move till reaching critical damping but in real time work this is impossible as will be shown in the experimental verification part. Figure 9 shows the transmissibility FRF (**x_c / x_d**) before and after stiffness reduction using PI controller.

V EXPERIMENTAL VERIFICATION

In this part, we discuss, experimentally, using the proportional plus integral (PI) controller to reduce the stiffness of the structure as discussed theoretically in the previous section.

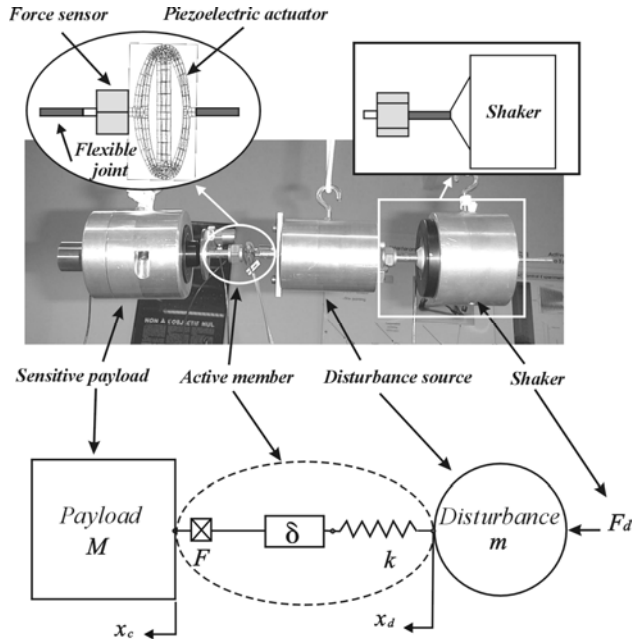

Figure 10
Experimental setup of single axis piezoelectric isolator

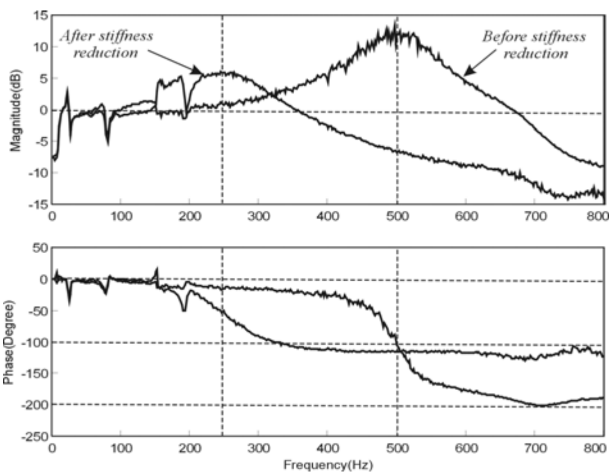

Figure 11
Experimental transmissibility (x_c / x_d)

Consider the experimental set-up shown in Fig.10. It consists of two masses connected to each other by an active member; the active member consists of a piezoelectric actuator and a force sensor. Using an external shaker, the system is excited with a random signal ranging from 1 to 800 Hz and the transmissibility FRF between the displacement of the disturbance source body and that of the payload mass is measured (Fig.11). The resonance of the system is found at 500 Hz. A feedback system with a PI control law is applied to the system and the same FRF measured again. Figure 11 shows the two measured FRFs: the open-loop (before stiffness reduction) and the closed-loop (after stiffness reduction). The natural frequency of the system has been reduced 50%; from 500 Hz to 250 Hz. The maximum reduction has been obtained by increasing the gain of the proportional part of the compensator, but this leads to the risk of walking along the imaginary axis which can lead to instability if the surrounding conditions change slightly. Thus, there is a need to increase the integral gain too at the same time to increase the stability margin of the system.

In the same context, another experiment has been done. The same control technique has been applied to the truss structure shown in Fig.12. The truss contains two active struts like the one shown in Fig.2 replacing two passive members. These two struts are used for the purpose of adding active damping to the system.

The signals of the two force sensors, in the two active struts of the truss, have been filtered using the (PI) compensator in a Digital Signal Processor (DSP) and fed back to the piezoelectric actuators. The two control loops have been closed independently, forming a decentralized controller. Again, by increasing the proportional gain, the stiffness of the active struts in the structure has been reduced significantly. Figure 13 shows the first two modes of the FRF between the voltage input to one of the actuators and the force output from the collocated force sensor. The open-loop FRF (before stiffness reduction) shows that the two modes are located at 8.8 and 10.5 Hz. Using this control technique, they have been moved to 2.6 and 5 Hz, respectively. A potential application of this is the adaptation of structural resonances to a narrow band disturbance of variable frequency.

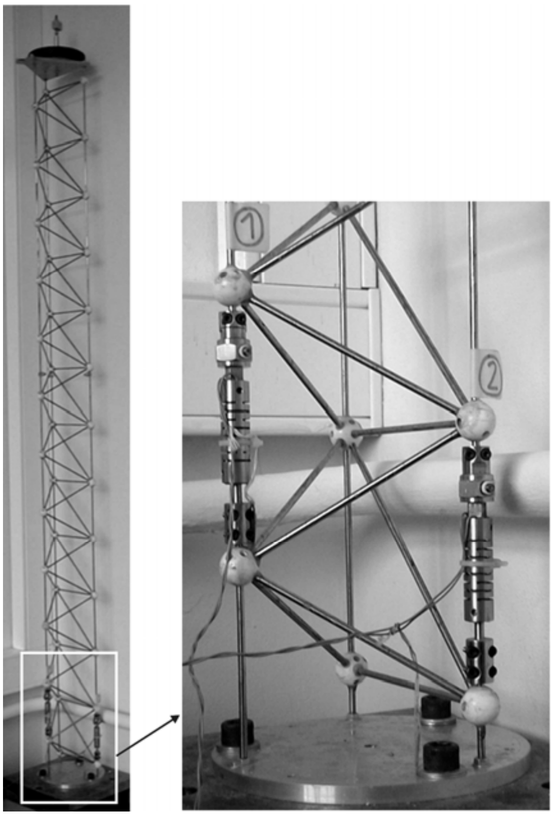

Figure 12
Passive mechanical truss with two active legs

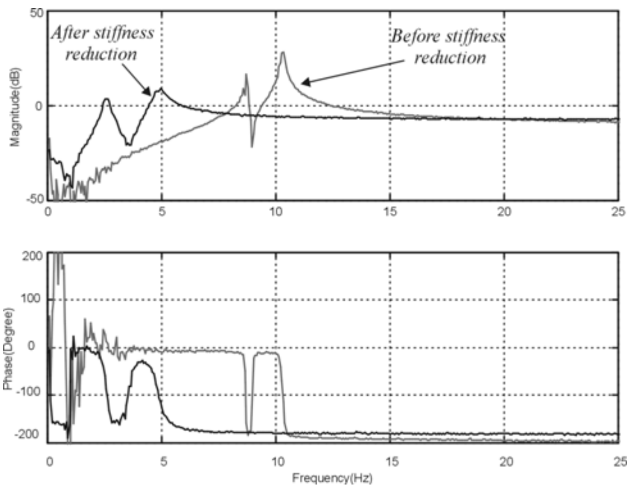

Figure 13
Experimental FRF between extension of the actuator and force in the leg

VI CONCLUSIONS

The analytical, simulation and experimental result discussed in this paper prove clearly that frequency variation is a reasonable method that can be used in vibration isolation by reducing the corner frequency of the system. Another application for this method is to simultaneously escape from excitations by rapid changing. This is possible by using rapid piezoelectric actuators that are stiff enough to stand under loads and can be softened for the purpose of vibration isolation. A future work to be done on this method is to use it in a multi function six degrees of freedom interface (Stewart platform). Simulation proved that the corner frequency can be reduced to low levels but there was a difficulty to reach that level experimentally because of the misalignment of the experimental setup that causes coupling with other exciting modes limiting the robustness and performance of the system.

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