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November 1999

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A Discussion of Zero Spring Rate Mechanisms Used for the Active Isolation Mount Experiment

INTRODUCTION

In the summer of 1995 the Structural Dynamics Branch at NASA Langley Research Center set out to conceive a small, lightweight, low frequency isolation mount that could be used for spaceflight experiments. The Engineering Design Branch undertook the task of developing the isolation mount. This report describes the engineering process that led to three phases of a study entitled "Active Isolation Mounts" (AIM).

The problem with making a very low frequency (<10 Hz) isolation mount is that it requires a very soft spring. A conventional mechanical spring can become quite large to accommodate the large deflections at low frequencies for a given payload. To get around this problem, other more complicated systems have been developed, such as a pneumatic-magnetic suspension system, which uses compressed air as the soft spring. But, this system is impractical for spaceflight.

Another approach, used successfully in the laboratory for low frequency suspension systems, is a zero spring rate mechanism. It works by balancing both a positive and a negative stiffness so that the net result is a small positive stiffness. This approach was taken to develop hardware suitable for a payload in the 1 to 10 lbm range and simple enough for spaceflight.

REQUIREMENTS

At the outset of the program certain requirements were set as a guideline for the development. The requirements were:

1) Lower fundamental frequency of a simple spring mass system by a factor of ten
2) No local modes between 1 and 300 Hz
3) Minimize power, weight and volume
4) Use the simplest design possible
5) Design for downward scalability
6) Be able to handle some shear and moment loads
7) Payloads from 1 to 10 lbm
8) Be robust enough to handle the space environment including:
    - thermal variations
    - vacuum
    - launch loads
    - zero gravity
9) Ability to continuously vary corner frequency
10) Minimize damping to less than 20% at the lowest frequency

THEORY

The method chosen to reduce vibrations is called a zero spring rate mechanism[1] (See figure 1). Compressed arms act like a beam that is ready to buckle in the middle. This produces a negative stiffness related to the preload in the arm. Theoretically this negative stiffness can exactly compensate for the positive stiffness of a spring mass system. The result is a zero spring rate.

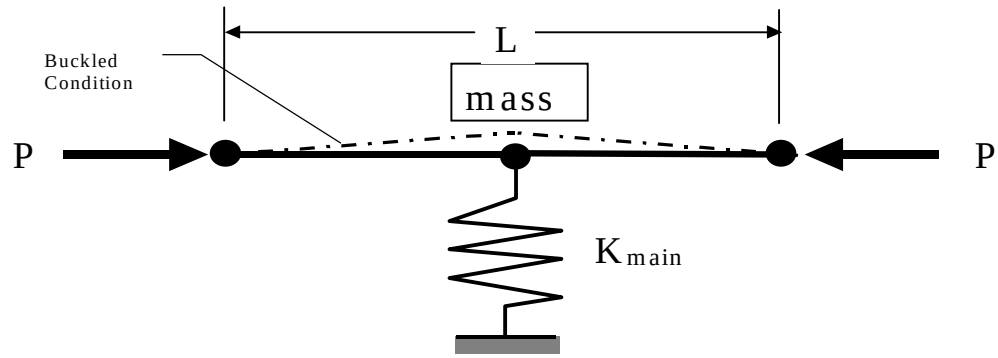

Figure 1: Schematic of a zero spring rate mechanism

The stiffness of the system theoretically (see reference 1) only depends on three factors: arm length, arm preload, and main spring stiffness when pinned conditions are assumed for the pivot points.

$$K_{Total} = K_{main} - P/L \tag{1}$$

where:
K_{total} = Total axial stiffness of the system
K_{main} = Stiffness of the vertical main spring
P = Preload in the arm
L = Length of the arm

The theoretical frequency of the system is given by:

$$\omega_n = \sqrt{\frac{K_{main} - P/L}{m}} \tag{2}$$

where:
ω_n = Fundamental frequency of the system (radians/second)
m = Mass of the system

Yet, while it is theoretically possible to decrease the stiffness of a system to zero, it is physically impossible. Figure 2 shows the fundamental frequency of the system as the preload in the compressed arms approaches a critical value. The closer the system gets to

zero stiffness the more unstable it becomes. This means that small disturbance forces or misalignments will destroy the delicate centered equilibrium.

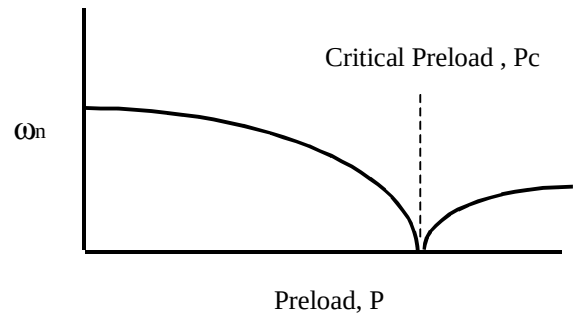

Figure 2: Effective frequency as a function of arm preload

In the real world many things affect the frequency. Thermal variations can change the length and force of the main spring and cause misalignments. Thermal variations can change the preload in the arms because the arms and the other components in the housing expand at different rates. Stress relaxation can change the preload in the arms non-uniformly and destroy the equilibrium. Forces that are too large will locally buckle the flexures at the pivot points in the arms and greatly reduce the preload. Varying dimensional tolerances can cause misalignments in the arms and uneven preloading.

DESIGN

AIM1

The Active Isolation Mount study evolved in three phases. The first phase, AIM1, was focused on making a small isolation mount that could be easily scaled downward (see figure 3). The main element to this structure was four arms arranged in a cruciform shape. These arms were preloaded and provided the negative stiffness to counteract the positive stiffness of the main spring. Each arm had a flexure at the hub and a flexure at the end to serve as pivot points. The flexures allowed for easy downward scalability and were simple to construct. A spread sheet was created to vary design parameters such as the length of the arms and the thickness of the flexures in the cruciform to obtain an optimal design for a given set of constraints. The arms had to be very stiff (100,000 lbf/in) because it was designed for the piezo actuator's short stroke (.001inch). If the arms were less stiff the piezo actuator would run out of control authority before the proper preload had been achieved. A differential adjustment screw was also needed to take up the slack in the system and provide a base preload in the arms of up to 500 lbf. The piezo actuator provided a fine adjustment of an additional 0–50 lbf. In practice however the differential adjustment screw proved to be of sufficiently fine resolution that the piezo was not required.

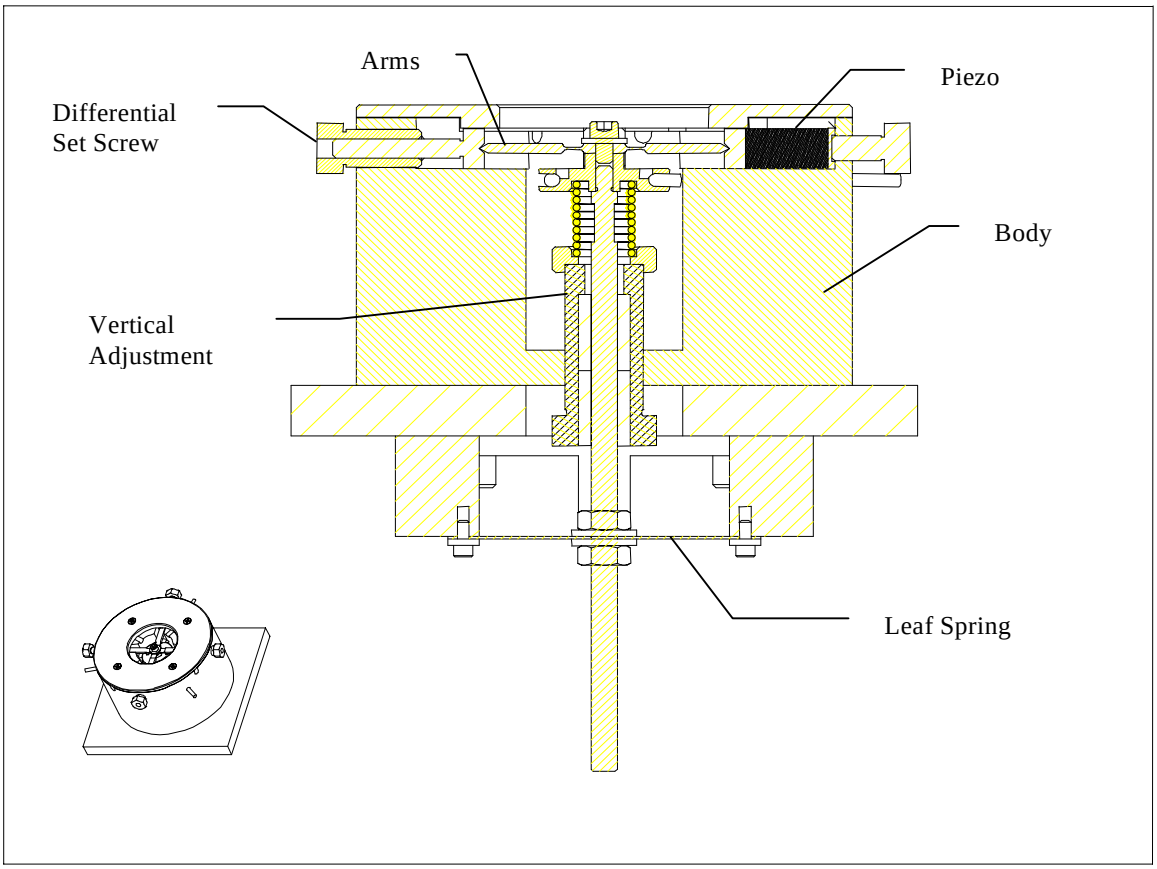

Figure 3: Active Isolation Mount 1 & 2 - cruciform configuration

Piezo Actuators - The 10mm x 10mm x 20mm piezo electric actuators, which produced 500 lbf in the clamped condition, were initially chosen because they were relatively small and were available. It was later discovered that the piezos did not provide enough force so it was necessary to switch piezos, which were almost the same size but produced up to 770 lbf.

Differential Adjustment Screw - A differential set screw was chosen to take up the slack and as a course adjustment because of its high resolution and high load capability. A regular micropositioning screw from a micrometer only has a pitch of 80 threads/inch, which equals .0125 inch/turn. This was not enough resolution, and the extremely fine thread pitch could not handle the high loads required. The differential set screw has an order of magnitude better resolution .0013 inch/turn (equivalent thread pitch of 756 threads/inch) and it has a very high load capability.

Compression Spring - A compression spring was chosen because it was readily available and different spring stiffnesses could be easily interchanged. Extension springs were rejected because they were too long.

Strain Gages - Two strain gage bridges were placed on each of the four arms. One bridge was placed in the middle of the arm to monitor the preload. The second bridge was

placed on the outer flexure to monitor the bending. The bending had to be maintained below 50,000 psi to preserve the fatigue life of the flexure. Monitoring the bending strain also turned out to be a useful way to know when the arms were co-linear, because bending stress went to zero.

AIM2

The goal of the second phase, AIM2, was to reduce the initial frequency of the system so the adjusted frequency would be lower. Changing the arm design reduced the initial frequency. First, the outer flexure of the arms were changed to a knife-edges (see figure 3). Secondly, the thickness of the remaining flexure was reduced from .040 inch to .020 inch. These two changes greatly reduced the initial vertical stiffness of the system from 2200 lbf/in to 500 lbf/in. Correspondingly the preload per arm was reduced from 550 lbf in AIM1 to 125 lbf in AIM2.

A leaf spring was also employed (see figure 4). The reason for this was twofold. First, the spring in conjunction with the arms allowed the mount to take some amount of shear load and moments. The second reason for employing the spring was because of its low aspect ratio. Since it is much thinner than a compression spring, the whole device could be made much shorter. In practice, the leaf spring was difficult to adjust vertically. The problem was the vertical adjustment was made directly on the center shaft. Therefore any jitter from the manual adjustment was transferred to the compression arms and made the system buckle prematurely. In the previous design (see figure 5), the load path went through the compression spring and was more forgiving. The previous compression spring was used for most tests, because it was easier to adjust.

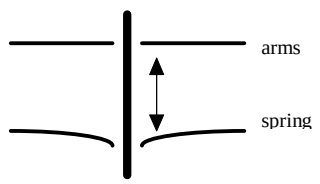

Figure 4: AIM2 design - direct vertical adjustment in center

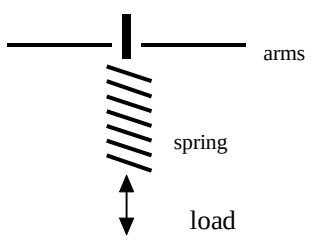

Figure 5: AIM 1 design - vertical adjustment through the spring

For future designs (see figure 6), a leaf spring could still be employed in one of two ways. The vertical adjustment could be made automatically with a piezo-screw, thus eliminating the jitter of the manual adjustment. Or, a manual adjustment could still be made if the leaf spring was adjusted at the edges instead of in the middle. This would make the load path go through the spring and the arms would be less sensitive to the jitter. Yet, either design would add complexity to the system.

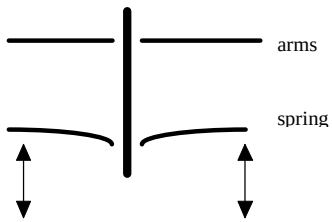

Figure 6: Vertical adjustment at the outer edge

Using a linear bearing to add shear and moment capability was also considered. This idea was rejected because it was thought the friction in the bearing would produce too much damping. Also, the linear bearing could not be scaled down to a very small package because the smallest readily available linear ball bearing requires a 1/8 inch shaft size.

AIM3

The goals of the third phase, AIM3, were to reduce the initial frequency further, try a four-bar linkage for lateral stability, and try an offset in the arms to create a stable dead zone (see figure 7).

Reducing the initial frequency was accomplished by changing all pivot points from flexures to knife-edges. This eliminated the vertical stiffness contributed by the flexures so the vertical stiffness was only dependent on the compression spring. The vertical stiffness was lowered by more than an order of magnitude from approximately 500 lbf/in to less than 10 lbf/in. The only limitation was the size of the spring that would fit into the device and how much payload it must counterbalance.

The four-bar linkage was employed for several reasons. The main reason was because the kinematic nature of the four-bar linkage constrained the center to only vertical motion in the plane of the device. Rotation out of the plane of the device was eliminated because the knife-edges contact at a line and not a point so the arms cannot pivot out but only up and down. Unfortunately the arms can slide out because they are only retained by friction. But, the double arms allow more variations in preloading the system. The preload in the upper and lower arms can be varied independently to find the best combination.

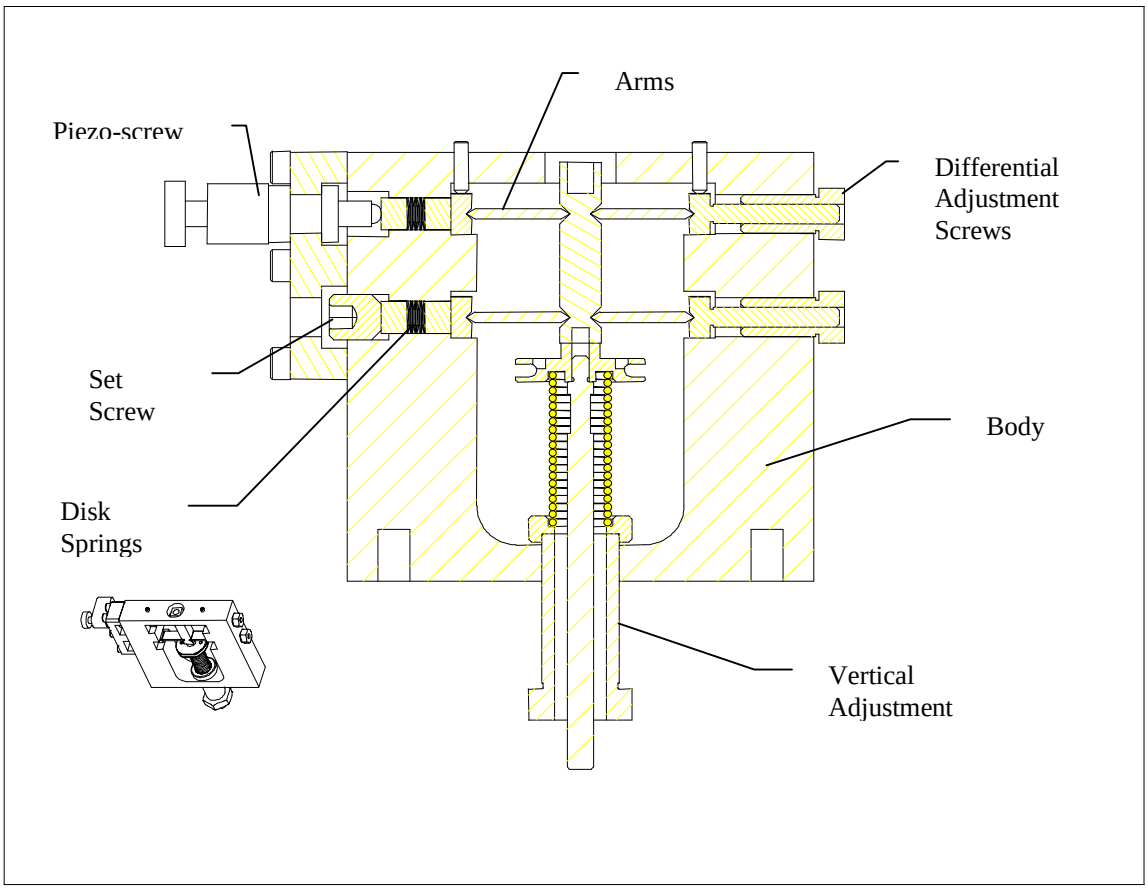

Figure 7: Active Isolation Mount 3 (AIM3) - four bar linkage configuration

The arms can be tipped up and down slightly (see figure 8) to cause a stable region in the center of the vertical stroke. The arms will seek an equilibrium in which they are all compressed equally. Moving the center up or down vertically will cause one pair of arms to be compressed more than the other creating a restoring force. The system will only become unstable when both pairs of arms have been pushed far enough that they are going the same direction. It was hoped this condition would allow for a more stable device and one that is easier to set, allowing for the lowest possible frequency.

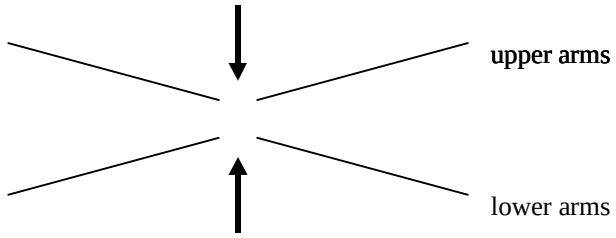

Figure 8: Exaggerated four bar linkage offset

The body of the device was made of a vertical slab of steel for stiffness. A vertical adjustment screw allowed the device to be centered vertically. A compression spring provided the initial spring stiffness of the device. Like AIM1 & 2 a rod through the vertical adjustment screw and compression spring allowed a mass to be hung from the device. A hole in the top of the body allowed a mass to be attached to the center hub for supporting a load from underneath. Four double knife-edge arms provided two sets of four bar linkages that constrain the device to vertical motion. The arms can be loaded in a number of ways to reduce the fundamental frequency. Differential adjustment screws on one side of the device provide fine adjustment of the preload in the arms. On the other side of the device, either a piezo-screw, which provides extremely fine adjustment, or a course adjustment set screw can be used to preload the arms.

Piezo-screw - The combination of a soft compression spring and knife edge pivot points reduced the preload in the arms to a point where it became feasible to use a piezo-screw actuator instead of a piezo stack actuator. The piezo-screw has several advantages over a piezo stack. It is has a much longer stroke, 1-2 inches as opposed to a piezo stack, which only has a stroke of .001 inch. Also, the piezo-screw only has to be powered when it moves. Friction locks the screw in place when the actuator is unpowered. The drawback of a piezo-screw is low force output. It is presently limited to only 5 lbf. In the earlier designs (AIM1& 2) the force would have had to be multiplied by a mechanical lever arm which would have added complexity.

Disc Springs - The disc springs between the v-groove blocks and the coarse adjustment screw or the piezo-screw were used to reduce the effective spring rate of the arms. The arms are laterally stiff (approximately 500,000 lbf/in) because they are made of stainless steel and they have no flexures to reduce the cross section. The eight 3/16 inch diameter disc springs in series reduce the effective lateral stiffness of the arms to approximately 500 lbf/in and make the preload easier to control.

TESTING

The systems were tested by hanging payloads from the devices ranging in weight from 3.3 to 10.3 lbm. A calibrated impact hammer provided the disturbance force and an LVDT provided the displacement response. Preliminary frequency data was obtained from an oscilloscope hooked to the LVDT. Formal test runs were collected by a data acquisition system and post processed.

The following table is a summary of the tests performed on the various configurations of the isolation mount. The “initial frequency” represents the system in its unadjusted state. The “lowest frequency” was the adjusted frequency just before the arms buckled. The lowest frequency was an important parameter because the system can only attenuate disturbances above this frequency. The ratio of initial frequency to lowest frequency was important because it gave an indication of the bandwidth of the system. The greater the ratio the more the system can be adjusted to avoid a specific frequency.

Table 1: Summary of Active Isolation Mount Tests

Description	Payload Mass (lbm)	Vertical Stiffness (lbf/in)	Initial Freq. (Hz)	Lowest Freq. (Hz)	% Critical Preload (%)	Initial/ Lowest (Hz/Hz)
Formal Tests						
AIM1 run4	10.3	1417	36.7	5.7	98%	6.44
AIM1 run5	5.3	1674	55.6	11.8	95%	4.71
AIM1 run6	5.3	2218	64	10.1	98%	6.34
AIM1 run7	5.3	2190	63.6	8.4	98%	7.57
AIM1 run9	10.3	1982	43.4	7.1	97%	6.11
AIM1 Average			*52.7*	*8.6*	*97%*	*6.23*
AIM2 Stiff Spring	10.3	451	20.7	4.5	95%	4.60
AIM2 Stiff Spring	10.3	442	20.5	7.3	87%	2.81
AIM2 Soft Spring	10.3	189	13.4	4.5	89%	2.98
AIM2 Leaf Spring	10.3	668	25.2	12	77%	2.10
AIM2 Average			*20.0*	*7.1*	*87%*	*3.12*
AIM3 Run1 3lb Load Above	3.3	272	28	17	64%	1.65
AIM3 Run2 10lb Load Below	10.3	272	17.5	7.5	78%	2.33
AIM3 Run3 10lb Load Below	10.3	272	17.5	7.3	79%	2.40
AIM3 Average			*21.0*	*10.6*	*74%*	*2.13*

AIM1 performed well. The initial frequency of the system was reduced by an average factor of 6.2. An average of 97% of the critical preload was achieved. The drawback of this device was instability. The low frequency could only be held for a matter of seconds in some cases. There were also long-term creep effects. The frequency would rise slowly over a course of hours probably due to stress relaxation.

AIM2 succeeded in reducing the initial frequency and the lowest frequency. It was also more stable than AIM1, but the percent reduction in frequency was not as great as AIM1. The frequency was reduced only by an average of 3.1 times. The leaf spring did not perform well due to difficulty in the vertical adjustment.

AIM3 succeeded in lowering the initial frequency even further for a 10 lbm load, but the four bar arrangement failed to create a stable zone. The very low vertical stiffness made it difficult to adjust the vertical alignment. Therefore the overall reduction in frequency was only an average of 2.1 times.

DESIGN CONSIDERATIONS

Active Vertical Adjustment

In practice, the vertical centering of a device must be constantly adjusted. One case in point is an informal test on AIM3. A preload was set and a frequency of 8.8 Hz was measured. Without any adjustment, one minute later it was 9.6 Hz. Three hours later it

was 10.6 Hz. One day later it was 13.2 Hz. Finally the vertical height was readjusted and the resulting frequency was 6.2 Hz. The system was very sensitive to the vertical adjustment. Even as the preload was increased from zero to the critical preload, the vertical adjustment had to be constantly monitored and reset. The problem intensified as the preload approached the critical point. An active adjustment with a control loop driven by the vertical position is recommended for this adjustment. The actuator could be a piezo-screw.

Three Arm Configuration

Although only a two arm and four arm configuration were used for this series of test, a three arm configuration offers some advantages (see figure 9). The system can be laterally supported in each direction with the minimum number of arms. Also one actuator can provide equal forces to each arm, because of the geometry of a three arm system. One negative aspect of pushing from only one side is that for every .001 inch the arm is compressed the center will move .0005 inch laterally. With high enough preloads this could cause alignment problems.

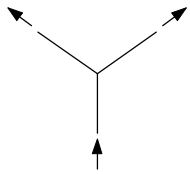

Figure 9: Schematic of a three arm configuration

Pivot Design

Two designs were used for the pivot points. At first, a flexure was chosen because it was simple and could be scaled down. Yet a flexure also provided vertical stiffness. This stiffness increased the initial corner frequency of the system but it also helped to center the arms vertically. This vertical self-centering made the device easier to adjust and thus compensated for the increased initial corner frequency by allowing the device to be adjusted closer to the critical preload.

The knife-edges were chosen in the later design because it provided no vertical stiffness and allowed the device to have a lower initial corner frequency. It also eliminated the concerns of fatiguing the flexure material or bending the flexure too far because of a large vertical stroke. Unfortunately the lack of vertical stiffness also caused the system to be more unstable and more difficult to control the closer the preload in the arms approached the critical preload.

Increasing Vertical Stroke

The vertical stroke of the system can be increased in two ways. The first way is to make the arms less stiff. The arms will expand more at the end of the stroke, as they become unloaded. But, reducing the stiffness in the arms can introduce local vibration modes in

the arms. Also, a piezo stack cannot be used to actuate the arms because of the larger displacements needed.

The second way to increase vertical stroke is to lengthen the arms. Now, for any given angle of the arms the stroke will be increased because of the geometry of the system. One drawback is that it makes the device larger because the arms are longer. A second drawback is that the critical preload in the system increases because it is related to the length of the arms. The body of the device must now be stiffer to account for the higher loads and the actuator must be able to produce higher preloads.

Decreasing Corner Frequency

Operating at a low corner frequency will help any isolation system because all disturbances above this frequency will be attenuated. One way to reduce the corner frequency is to start with a lower initial corner frequency by using a softer main spring. But, the softer springs can be quite large for a given payload thus increasing the size of the entire system.

Another way to reduce the corner frequency is to operate closer to the critical preload. The problem with this is that the system becomes very unstable. Even a small disturbance can destroy the equilibrium of the system. Also, low frequencies produce large amplitudes for a given disturbance and may exceed the stroke of the system even for very small disturbances. Large amplitudes are a problem with all low frequency systems.

CONCLUSION

The Active Isolation Mount study demonstrated devices that could reduce the initial corner frequency by a factor of six for brief periods and a factor of two for extended periods. The designs were relatively simple and minimized weight, volume, and power. They could be scaled down and they were made of spaceflight compatible materials. All designs offered the ability to continuously vary the fundamental frequency. Yet, the goal of reducing the frequency by an order of magnitude was not achieved because the systems were too unstable near the critical preload. Any disturbance would destroy the delicate equilibrium and make the arms buckle prematurely. There was a trade between performance and stability.

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1. S. E. Woodard and J. M. Housner: *Nonlinear Behavior of a Passive Zero-Spring-Rate Suspension System*. Journal of Guidance, Control, and Dynamics, Volume 14, Number 1, January-February 1991, Pages 84-89.

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