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Active & Passive Vibration Suppression For Space Structures

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Applicable Passive Treatments for LSS

Several test proven damping treatments apply

- * Viscoelastic shear damper
- * Rotational damper
- * Tube constrained layer damping
- * Plate constrained layer damping
- * Strut viscous damper

In this presentation, we contemplate the relative benefits of passive and active vibration suppression for Large Space Structures (LSS). The intent is to sketch the true ranges of applicability of these approaches using previously published technical results for this review. In part also, it is our hope to counter past incidences of overzealous advocacy of exclusive use of passive damping or exclusive use of active control and argue, instead, for the proper combination of both approaches.

First, let us consider the various methods of intrastructural damping treatment in use or being considered for use in LSS. Most of the listed damping techniques work by constraining a layer or annulus of viscoelastic material so that it is placed in a state of shear strain. Some devices use the resulting energy dissipation from shear-strain-rate to damp translational motions, whereas others, such as the rotational damper concept, employ an annulus of viscoelastic material to damp rotational motion. In addition there are essentially "add-on" damping treatments using a thin layer of viscoelastic material covered by a stiff "constraining layer" for the purpose of damping flexural vibrations in beams or plates. Finally, strut viscous damper concepts are well adapted to the damping of axial deformations of strut elements within built-up truss structures. These are all intrastructural damping concepts. There are also inertial damping concepts -e.g. the tuned-mass damper which we'll discuss in a moment.

Passive Damping for Vibration Suppression in LSS

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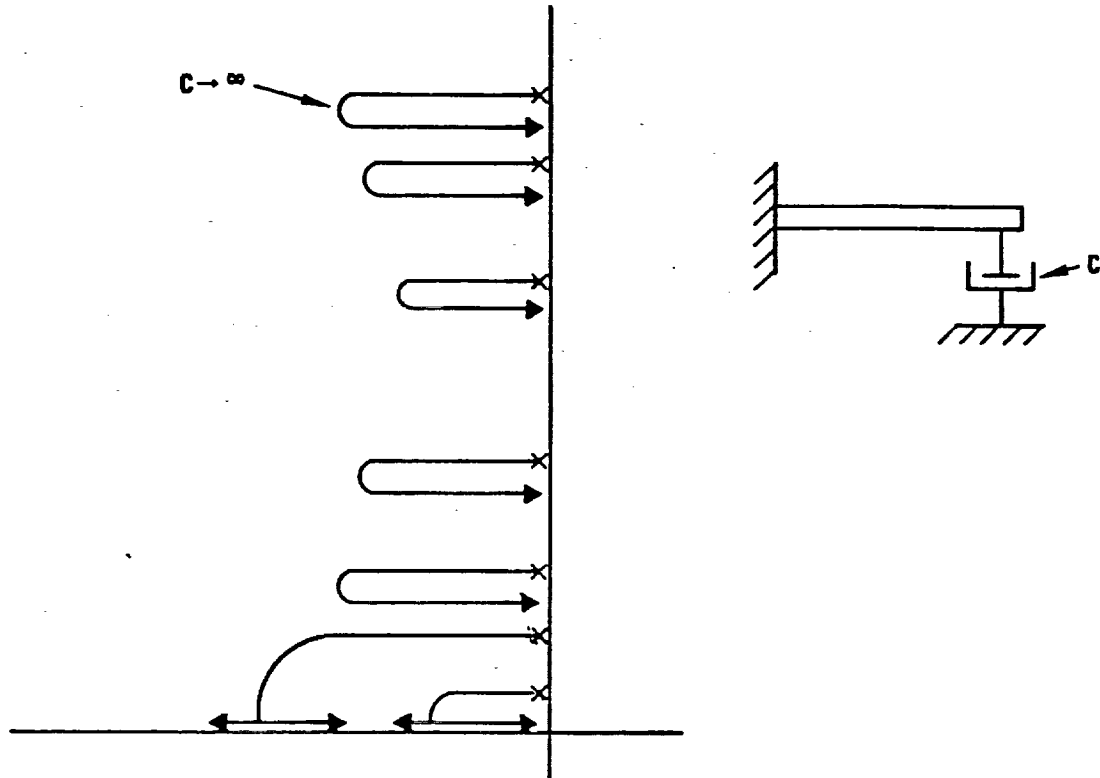
- * inherently stable
- * simpler to implement - no on-line processor necessary
- * reasonably weight efficient

-

- * inherent performance limitations
- * Viscoelastics are
 - are temperature sensitive
 - outgass
 - have low specific stiffness and strength
 - creep (dimensional stability issue)
- * tuned-mass dampers are effective only when the target mode frequency is well predicted
- * effective design requires good modelling information on performance - significant modes and modal strain energy maps.

Passive damping approaches offer many inherent advantages for LSS vibration suppression, i.e., these approaches are inherently stable, usually require no on-line processing or electronics and are reasonably weight-efficient. These advantages are presently well recognized and demonstrated. However, a sober assessment must recognize a number of engineering design and implementation issues that arise in LSS applications. First, there are inherent performance limitations to passive damping that we review presently. There are detailed design issues connected with the properties of viscoelastic materials -e.g. temperature dependence of the damping loss factor, outgassing, low specific stiffness and strength and viscoelastic creep which has a direct impact on dimensional stability performance of LSS. These negative factors are not necessarily irremediable - but the successful resolution of these issues in detailed design does contribute to the cost and complexity of final implementation.

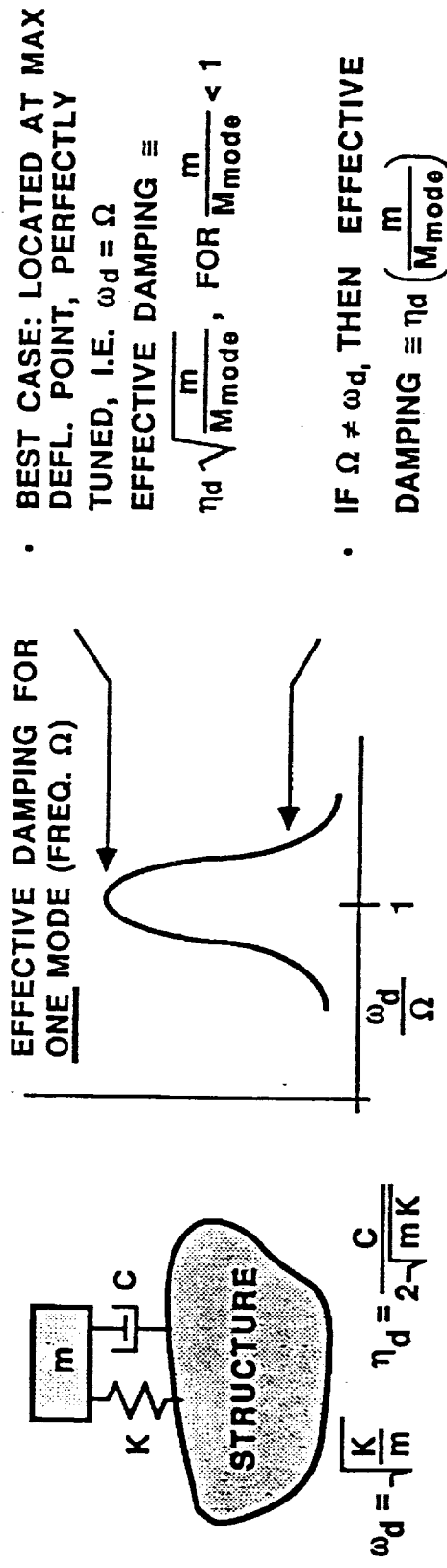
Also, the "bottom-line" performance (e.g. line-of-sight jitter, etc.) achieved by a given passive suppression system *does* often depend critically upon the accuracy of *a priori* structural dynamic modelling. For example, tuned-mass dampers are particularly effective only when the target mode frequency is well predicted. With regard to constrained-layer or truss member damping, effective design requires good-quality modelling information on the performance - significant modes and their strain energy maps. If in-mission changes or parameter errors cause significant departures from design-model dynamics, actual damping can be far less than that predicted or specified. Thus, while there is no issue with *stability robustness*, the issue of *performance robustness* remains.



- There is a maximum damping coefficient beyond which there is no further improvement
- For complex structures, sufficiently large damping coefficients decrease energy dissipation

Figure 3. Inherent Limitation to Passive (Semi-Passive) Damping

We illustrate the well-known performance limitations of intrastructural passive damping with the simple cantilevered beam example shown. The point is that the structural damping does not always increase with further increase in the end-mounted damper viscoelastic constant, C . In fact, there is a maximum value of C beyond which there is no further improvement in system damping. In the limit as C increases without bound, the system poles coalesce with zeros on the imaginary axes and there is no damping since the damper acts as a rigid constraint. This effect is due to the fact that spatially discrete dampers modify both the structural mode damping *and* the mode shapes. In consequence, it can sometimes happen that sufficiently large damping coefficients in discrete damper devices can actually decrease energy dissipation in critical regions of a complex multi-component structure.



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- M_{mode} TYPICALLY > 380 LBS., THEREFORE VERY LARGE m NEEDED TO GET 20% STRUCTURAL DAMPING.
- A DIFFERENT SET OF DAMPERS IS REQUIRED FOR EACH SIGNIFICANT MODE (MORE THAN 30).

Having taken a brief (but perhaps sobering) look at the pros and cons of passive vibration suppression, we pose the question of crucial interest here: With respect to robust performance and simplicity of implementation are active vibration control and passive damping really so distinct after all? (Or has the debate occurring over the recent past been largely a war of words?)

Let us explore this question by contrasting a passive approach with a corresponding active approach to inertial damping.

First, the passive approach considered here is the "tuned-mass" device illustrated in the Figure. Basically, this consists of a small mass (m) connected to the structure with an elastic element (with stiffness k) with viscoelastic material (the dashpot) in the load path to provide a large viscoelastic damping. This is a very simple and inherently stable damping augmentation device. On the other hand, although modal damping augmentation for the "targeted" structural mode can be substantial when the damper resonance (ω_d) is near the targeted mode frequency, damping augmentation is slight when there is frequency mismatch. Overall effectiveness depends on the ratio of the damper mass to the generalized mass of (M_{mode}) of the targeted mode (and in the system context of this particular diagram M_{mode} was typically several hundred pounds so that a large m would have been required to obtain the desired 20% damping). Thus, if there's modelling error resulting in significant "detuning", damping will be far less than predicted and one is stuck with the resulting performance loss. (Of course, a possible way around this problem is to build in an active electromechanical device capable of changing the damper stiffness, k , so as to "re-tune" the damper on-line, during the mission - but this refinement would negate most of the distinction between "passive" versus "active"!).

- PROOF MASS / VOICE COIL DESIGN
- GRAPHITE-EPOXY FLEXURES OBVIATE BEARING STICKTION PROBLEMS
- ULTRA-HIGH-BANDWIDTH CASING-MOUNTED ACCELEROMETER MEASURES STRUCTURAL MOTION
- PROOF-MASS MOUNTED ACCELEROMETER CLOSSES INTERNAL CONTROL LOOP RESULTING IN RELIABLY SHAPED FREQUENCY RESPONSE

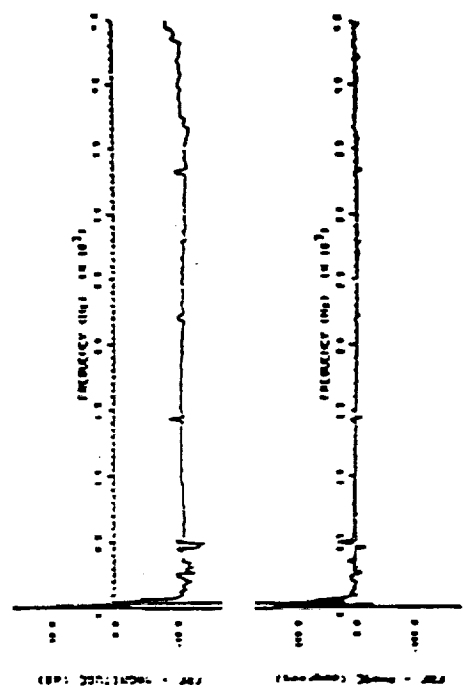
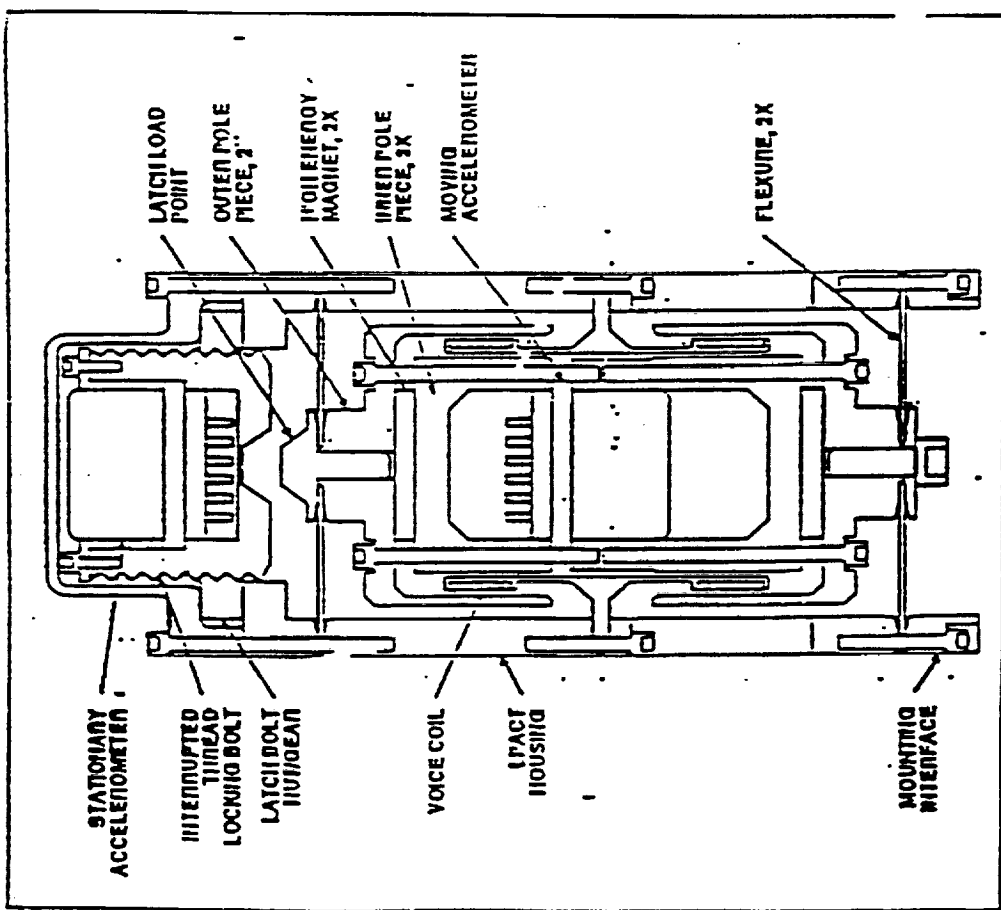
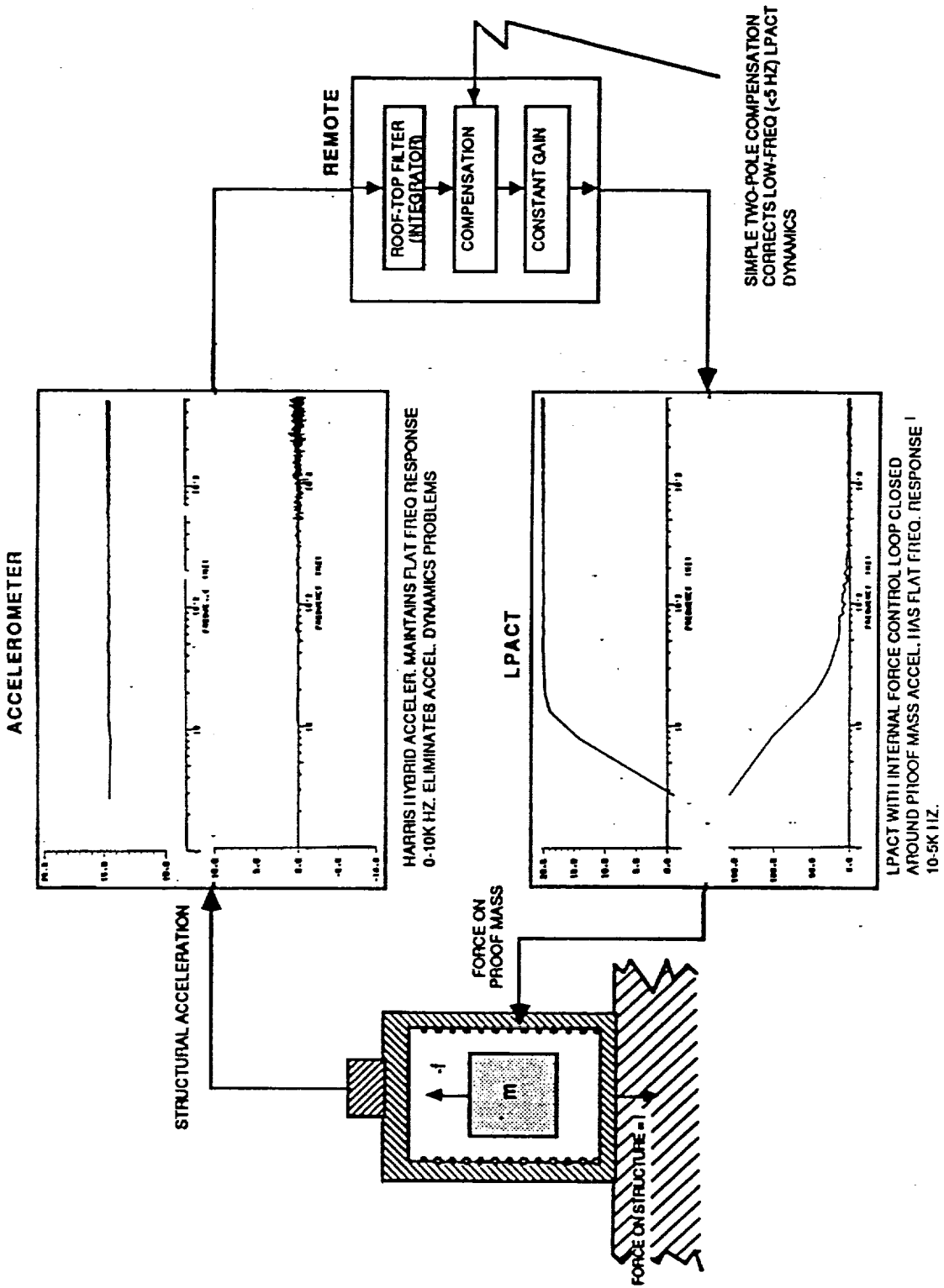


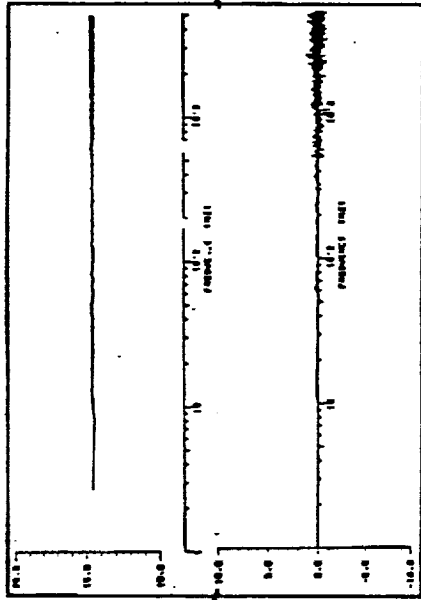
Fig. 5

LPACT Damping Unit Provides Reliable, High-Bandwidth Sensor/Actuator in a Single Package.

Now, consider an analogous *active* approach to vibration suppression using the Linear Precision Actuator (LPACT). The patented LPACT device (see Reference 1) is a bearingless voice coil proof-mass actuator which uses a proof-mass-mounted accelerometer to close a force control loop which serves to override nonlinearities and temperature-dependent effects. With this internal force compensation loop, the LPACT has flat frequency response from 3-10 Hz to at least 5 KHz. The LPACT design currently used in Harris test beds provides a maximum force of 5 pounds with 20 micropound resolution. Each LPACT has a casing-mounted accelerometer for implementation of vibration control feedback.

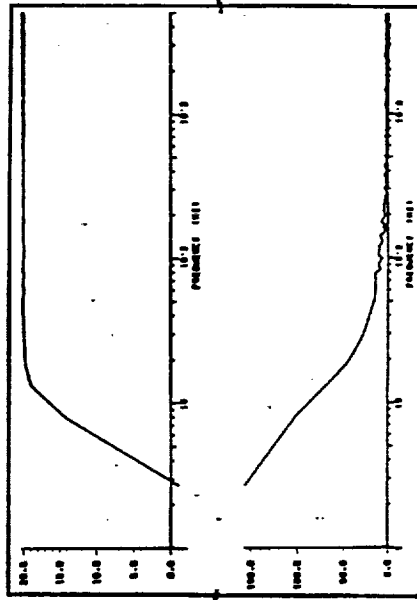


ACCELEROMETER



HARRIS HYBRID ACCELER. MAINTAINS FLAT FREQ RESPONSE
0-10K HZ. ELIMINATES ACCEL. DYNAMICS PROBLEMS

LFACT

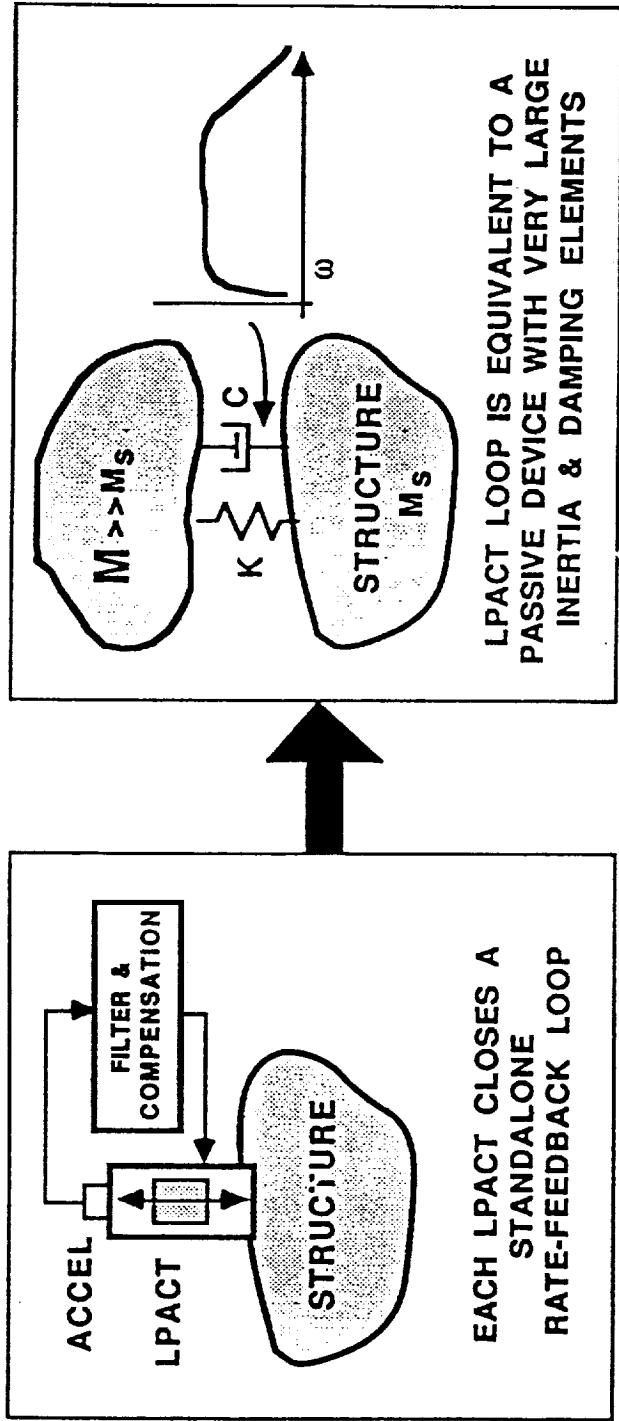


LFACT WITH INTERNAL FORCE CONTROL LOOP CLOSED
AROUND PROOF MASS ACCEL. HAS FLAT FREQ. RESPONSE 1
10-5K HZ.

SIMPLE TWO-POLE COMPENSATION
CORRECTS LOW-FREQ (<5 HZ) LFACT
DYNAMICS

The casing-mounted accelerometer is the new "Hybrid Accelerometer", an advanced acceleration sensor design providing flat frequency response from DC to at least 10 KHz.

The diagram illustrates that with the exceedingly high bandwidth and flat frequency response of the LPACT actuator and colocated Hybrid Accelerometer, it is now possible to implement a simple colocated rate feedback controller to provide broadband damping. Note that the LPACT with its Hybrid Accelerometer form one single compact "active damping unit."



- CONTROL IS EQUIVALENT TO A NETWORK OF PASSIVE DEVICES, BUT WE USE THE ELECTRICAL ENERGY OF THE LPACT TO EMULATE MUCH LARGER INERTIA & DAMPING THAN IS MECHANICALLY POSSIBLE.
- LARGE BROADBAND DAMPING IS REALIZED DESPITE THE SMALL ACTUAL MASS OF THE LPACT.

If as indicated on the left of the illustration, we use the LPACT to close a standalone feedback loop, then due to the high band width of the sensor/actuator hardware, the LPACT loop closely approximates a passive device – similar to the tuned mass damper – but with very large inertia and damping elements. As illustrated, the LPACT is equivalent to an inertially anchored damper with large viscoelastic damping and is thus able to provide very broadband damping (not just frequency-tuned damping) despite the small actual mass of the LPACT.

Thus, there presently does exist *active* control hardware that can emulate the inherently stable operation of passive vibration suppression but with the added flexibility to provide much larger effective inertia and damping than would be mechanically possible with passive devices.

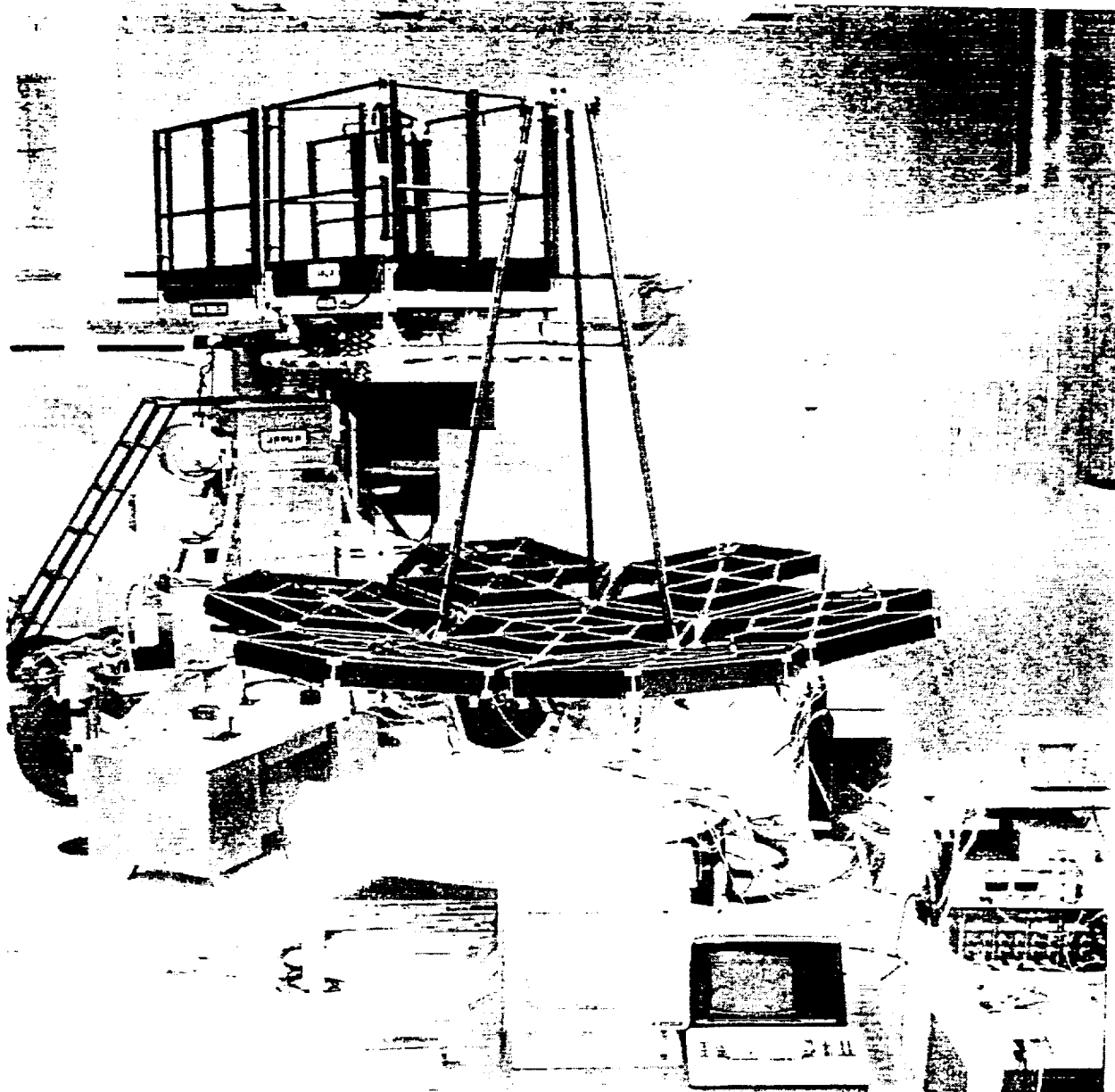
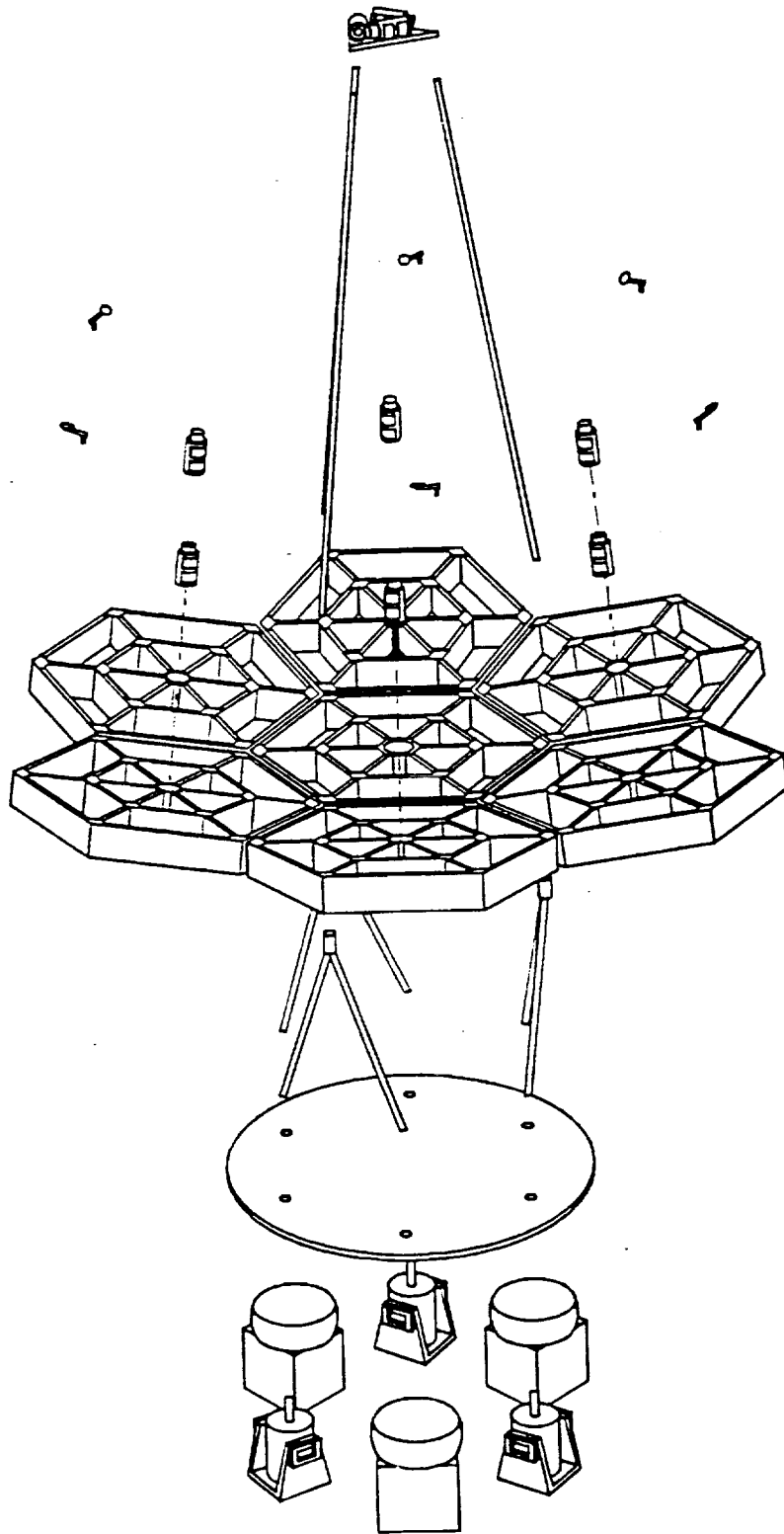


Figure 8. The Multi-Hex Prototype Experiment is the third in a series of 3 experiments designed and implemented at Harris.



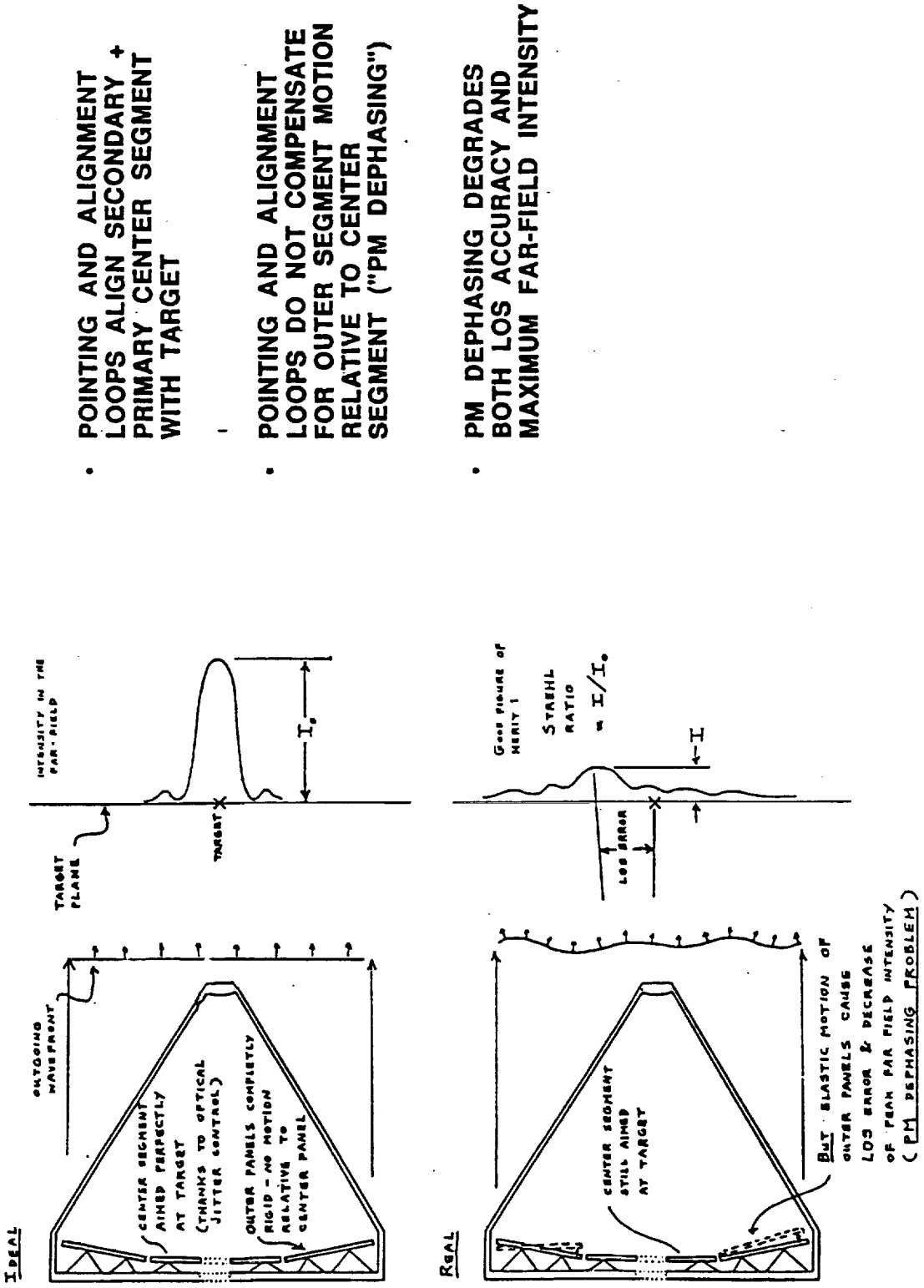
SC-7 (XPro)

Fig. 8.a
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The above performance benefits of LPACT sensor/actuator units have been experimentally demonstrated using the Multi-Hex Prototype Experiment (MHPE). The MHPE (Reference 2,3) is a vibration control testbed developed on Harris IR&D to study the vibration issues associated with generic Cassagrain configurations with large multi-segment primaries.

As illustrated in the photograph, the MHPE consists of a secondary mirror and support platform supported by a Gr/Ep tripod tower connected to the center segment of the primary reaction structure. The primary reaction structure is an array of seven Gr/Ep hexagonal box trusses. The array is approximately 4M across. A six member truss connects the seven-panel array to a circular baseplate (emulating a spacecraft bulkhead). The total static weight is supported by air-bag isolators and electrodynamic shakers are interfaced to the baseplate to provide disturbances emulating broadband spacecraft-generated disturbances. Line-of-Sight (LOS) jitter and panel-to-panel misalignments due to vibration are monitored by three complementary subsystems: (1) a pseudo-dephase-measurement system using a large number of accelerometers and on-line processing, (2) the Optical Performance Measurement Subsystem using laser interferometry to measure panel-to-panel misalignments and (3) an optical LOS scoring subsystem using a faceted secondary and optical flats distributed over the primary reaction structure.

IF ONLY THE PRIMARY MIRROR WERE RIGID, OUR PROBLEMS WOULD BE SOLVED! BUT...



- POINTING AND ALIGNMENT LOOPS ALIGN SECONDARY + PRIMARY CENTER SEGMENT WITH TARGET

- POINTING AND ALIGNMENT LOOPS DO NOT COMPENSATE FOR OUTER SEGMENT MOTION RELATIVE TO CENTER SEGMENT ("PM DEPHASING")

- PM DEPHASING DEGRADES BOTH LOS ACCURACY AND MAXIMUM FAR-FIELD INTENSITY

- 4 -
Fig. 9

The MHPE was designed to study a number of vibration control issues in large RF or optical systems, including both LOS jitter and "Primary Mirror (PM) dephasing". The PM dephasing issue illustrated here, arises because vibrational disturbances cause misalignments of the individual PM segments relative to one another. According to the laws of diffraction such "dephasing" of the PM segments can cause considerable reduction of the peak radiation intensity in the far field. Often, PM dephasing cannot be readily compensated by alignment elements in the system optical train and structural control of the PM assembly may be desired.

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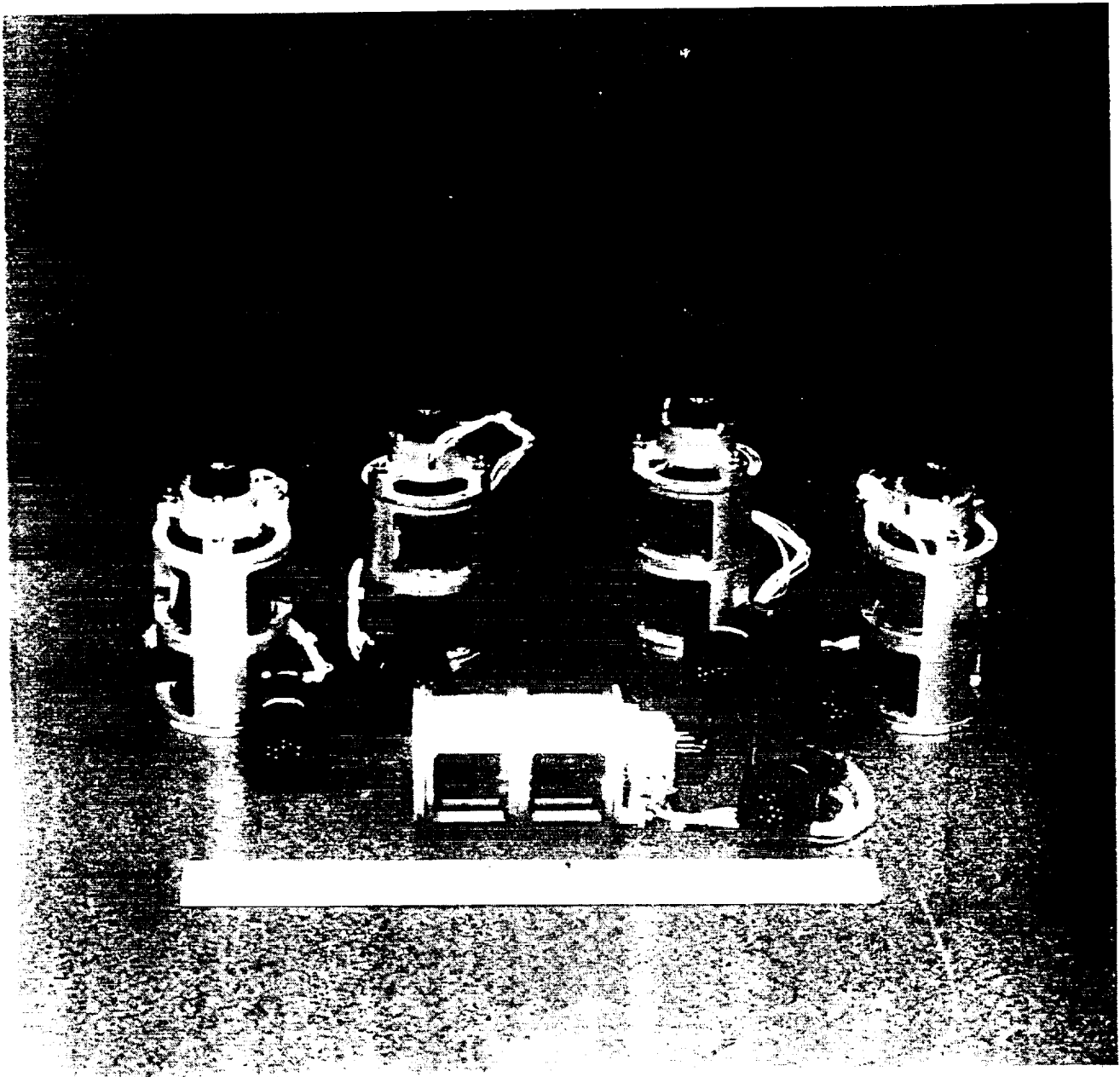


Figure 10. The Harris Linear Precision Actuator (LPACT) is a proof mass actuator developed for vibration suppression of flexible structures. The internal control loops of the LPACT yield a device transfer function which has high bandwidth and flat response over a large frequency range.

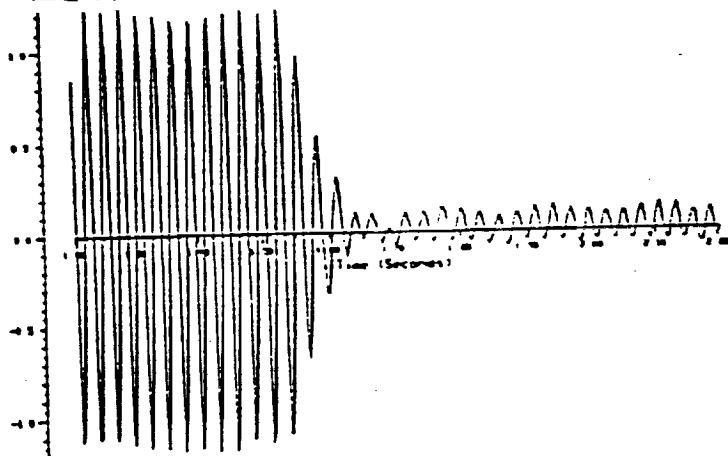
For active vibration control, the MHPE is instrumented with nine LPACT sensor/actuator units: three LPACTs on the secondary mirror platform to control the tower bending modes contributing to LOS jitter and six LPACTs mounted within the outer hex panels to control primary reaction structure panel dephasing. Both data acquisition and on-line control algorithm implementation are executed via the MCX-5 computer.

The system can implement both centralized, MIMO, control algorithms and/or decentralized control designs and a variety of designs have been tested and included in live demonstrations of active vibration control provided to Harris visitors over the last two years. Here we show data (References 4,5) on the decentralized rate-feedback control design discussed above.

```

/ID 3 response to steady state sine disturbance with single controller
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/CHANNEL DESC L2 velocity estimate
/RANGE 0 2711E-02 /SCALE 0 10000E+01

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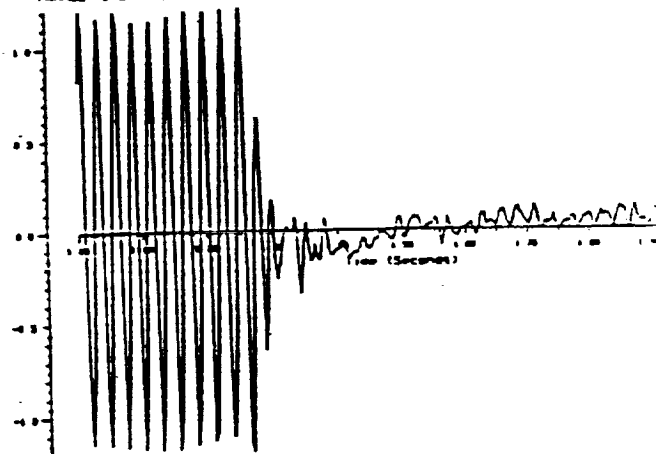
(a) controller on t=1.56 sec

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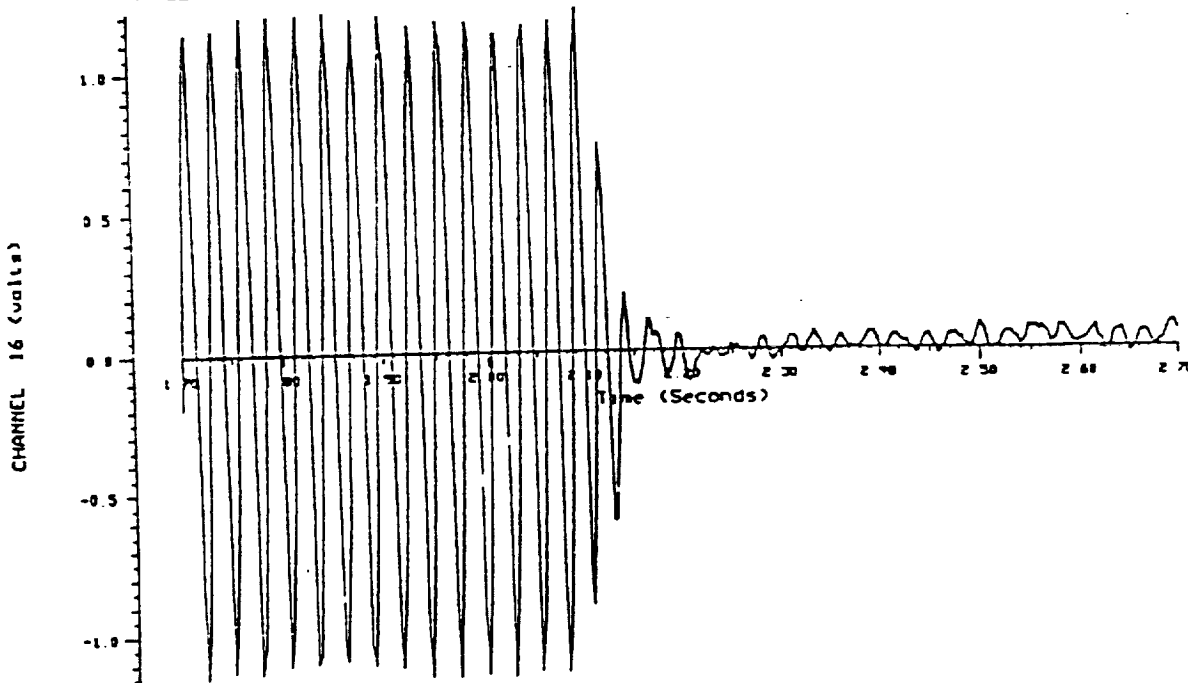
(b) controller turn on t=1.3 secs

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/CHANNEL DESC L2 velocity estimate
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(c) both controllers on t=2.2 secs

Figure 11. The MHPE hybrid control heirarchy features both performance and fault tolerance.

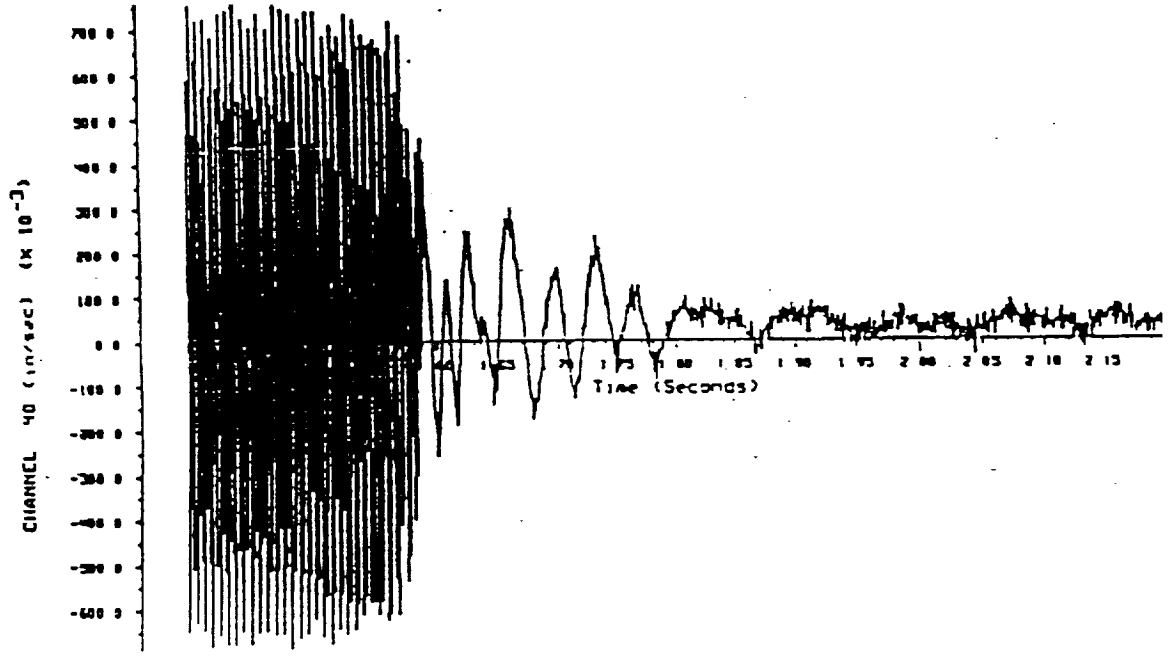
The decentralized design is a hybrid design consisting of a high bandwidth (1000 Hz) analog control for damping of very high frequency modes and a lower bandwidth high gain digital control for enhanced suppression of the lower frequency modes. Overall, an order of magnitude suppression of LOS jitter and rms dephasing is obtained for broadband disturbances. To illustrate this capability for visitors in our live demonstrations we show open and closed-loop performance for a medley of modes -using sinusoidal disturbances at modal frequencies in order to make the vibrations palpable to the human senses. The demonstration sequence starts with lower frequency modes, which can be felt by touching the MHPE panels and concludes with high frequency modes which can be clearly heard.

Here, for example, we show via one of the accelerometer measurements, the open and closed-loop vibration for a 35 Hz mode involving large panel-to-panel misalignment. The bottom plot shows the complete hybrid controller. Here the mode is excited sinusoidally with the disturbance maintained throughout the test period. Up to $t = 2.2$ sec., the control is turned off and open-loop vibration is observed. When, at $t = 2.2$ sec., the controller is turned on, the vibration level quickly drops by approximately an order of magnitude.


```

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/CHANNEL DESC VELOCITY ESTIMATOR LS PRI
/RANGE 0 24414E+02 /SCALE 0 10000E+01

```



controller turned on at t=1.58 secs

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/! velocity estimator /MEAN /

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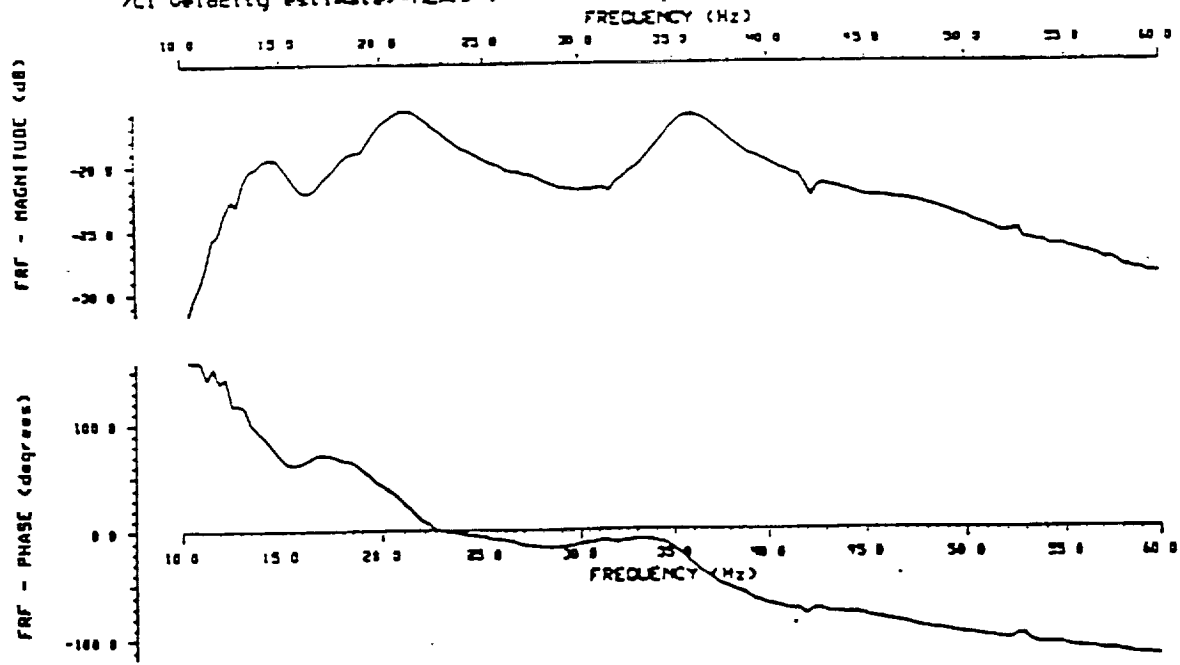


Figure 12. Colocated analog control loops performance illustrated in both time and frequency domain.

Similar results are obtained for the other performance-significant modes. In addition, we demonstrate high levels of active damping even for very high frequency modes (up to approximately 900 Hz). For example, the top plot shows open- versus closed-loop results when a 411 Hz mode is excited (vibration in this mode is clearly audible). When the control is turned on at $t = 1.58$ sec., the vibration amplitude again drops to a substantially lower level. Similar attenuation is observed for the other high frequency modes - up to approximately 900 Hz where the control feedback gain begins to toll-off.

Such results demonstrate simple decentralized control that implements "semi-active" damping, and show an order of magnitude improvement in dephasing with rugged bolt-on hardware. Again, an important point is that active control has matured to produce active hardware permitting control that is at least as effective and as reliable as passive damping over frequencies below 1 KHz. Added benefits include the scope to achieve even better performance with more sophisticated control strategies and the capability to revise these strategies as needed.

Further MHPE experiments have combined active control with passive constrained-layer damping. Although these activities are the subject of a separate report, we should note that the active and passive components are clearly complementary, the active control providing large attenuation from 10 to 900 Hz and the passive damping providing suppression of the multitude of very closely spaced modes near 1 KHz and above.

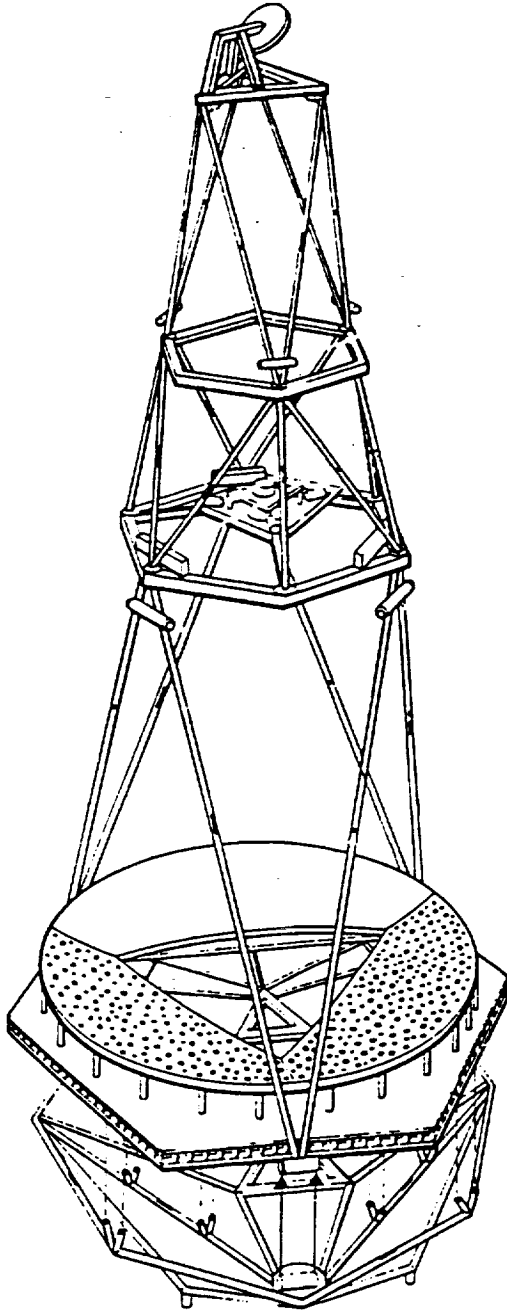
Passive Damping and Active Control

With active control, passive/active system requires

- less power
- fewer actuators
- increased robustness
- simpler control algorithms

Passive Damping must be designed in early in design process

Now that active control technology has matured to the point that its risk and complexity have been greatly reduced, it's time to consider an overall approach combining active control and passive damping. Individually, these technologies are not panaceas but the most cost-effective route is the proper *orchestration* of both. As indicated in the chart, the combination of active and passive technologies offers many synergistic advantages. In particular a combined active/passive vibration suppression system may require less power, less instrumentation, less complicated control algorithms while offering more robust performance.



- **Base disturbances broadband over 5–1000 Hz**
- **Control objective: reduce vibration 40–60 dB relative to open loop response over frequency band from 5 to 500 Hz**

Figure 14. HALO Optical Structure

The mutually reinforcing benefits of passive stiffness augmentation, passive or "semi-passive" damping augmentation and active control are illustrated by some results obtained approximately five years ago (see Ref. 6). This example involves an experimental configuration for the HALO (High Altitude Large Optics) structure, which is a graphite/epoxy truss with ellipsoidal optics, and we postulate the use of HALO as a test-bed for various vibration control methodologies. To this end, the basic scheme features the use of electrodynamic shakers to provide broadband force excitations to the base of the bottom truss structure and to the secondary mirror platform. In particular three independent base disturbances are postulated having flat power spectral density over 5-1000 Hz. The overall vibration suppression objective is to reduce rms line-of-sight (LOS) and wave front (WF) errors by approximately 60 dB relative to the open loop. An iterative design process led to the selection of vibration control hardware consisting of a number of colocated accelerometer/voice coil actuator units and noncolocated linear DC motor actuators and internal alignment optical sensors.

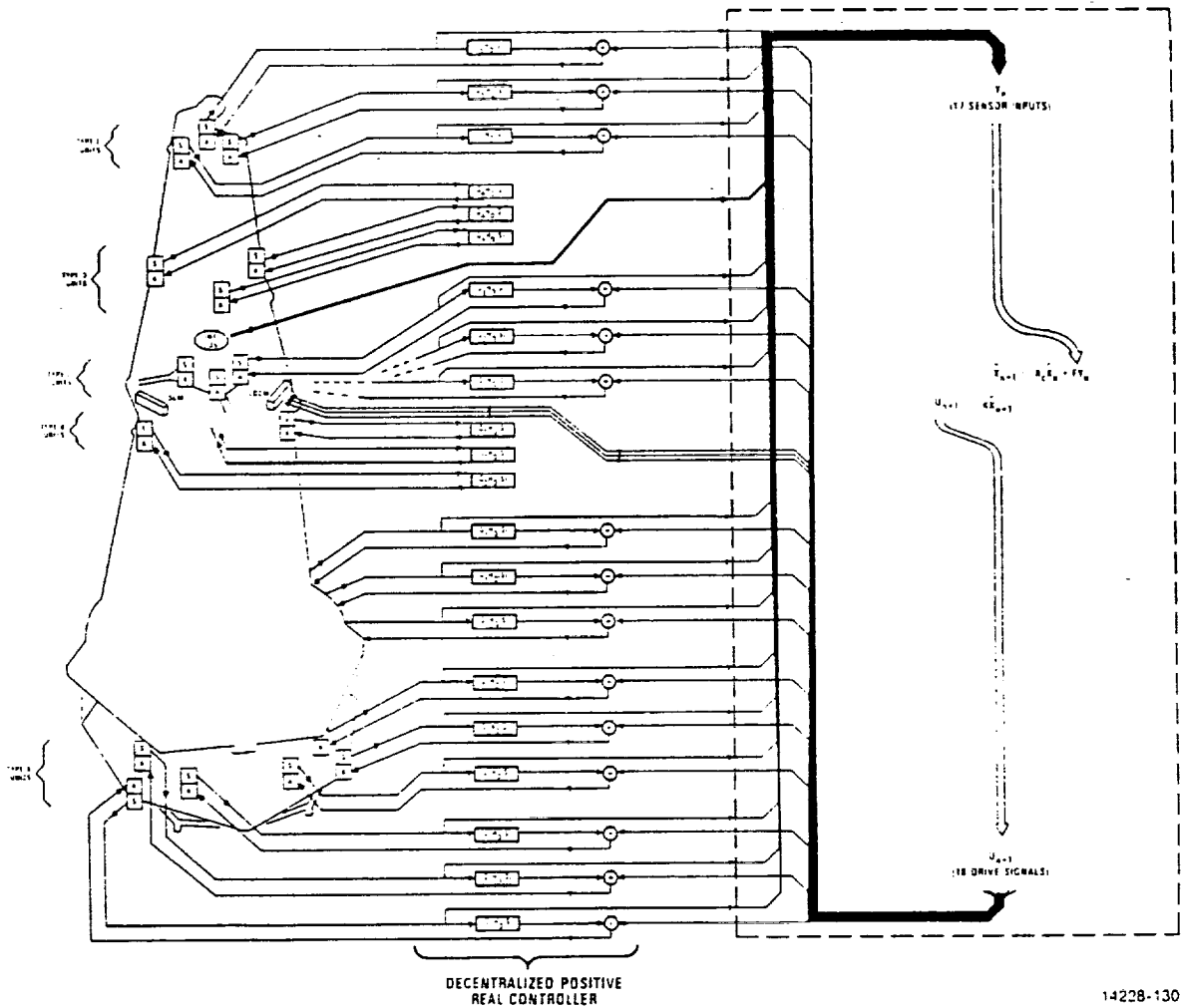


Fig. 15

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In the design studies for the HALO modal, we traded off various levels of advanced materials usage, semi-passive damping and active control. As indicated in the diagram, the control system has a two-level architecture consisting of:

1. 21 independent decentralized positive-real controllers (DPRC's) imposing local feedback between voice-coils and colocated accelerometers.
2. A Centralized Coordinating Dynamic Compensator (CCDC) which provides simultaneous coordination of many noncolocated sensors and actuators.

The DPRC's represent a semi-passive damping approach similar to the LPACT rate feedback loops discussed above for the MHPE. The CCDC is the centralized "active" control component.

With this two-level control architecture, we compared cases involving the original Gr/Ep structure with a structure wherein the main components are composed of a Metal Matrix Composite (MMC) offering a four-fold increase in the stiffness of Gr/Ep.

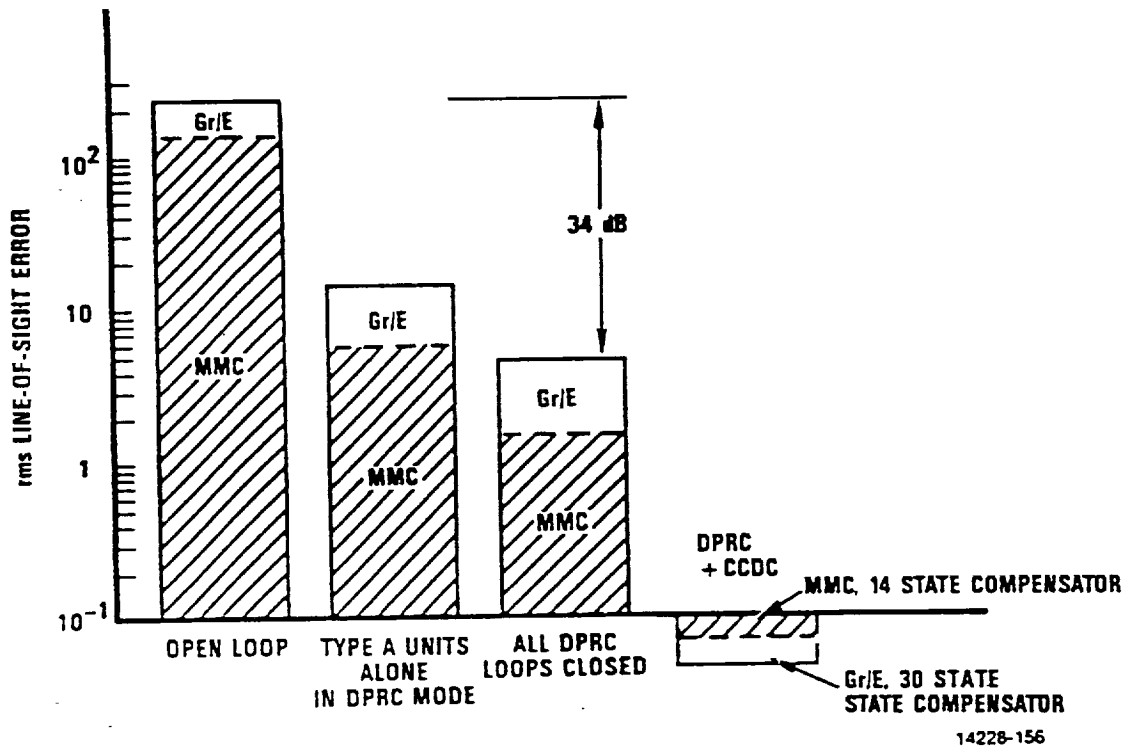


Figure 16. Summary of HALO Controller Performance Results: Line-of-Sight Error

This Figure summarizes LOS jitter performance results for both Gr/Ep and MMC structures. Specifically, for both material selections we show rms LOS errors for the open-loop, for the semi-passive controllers alone and finally, for the complete control including the centralized active control design. The increased stiffness of the MMC structure gives only modest performance improvement in the open-loop. However, it is evident that increased stiffness combined with semi-passive vibration suppression and centralized active control gives performance improvement well beyond what might be expected of each design measure individually. The final performance, being more than the sum of its parts, indicates the synergistic benefits of combining passive and active suppression techniques.

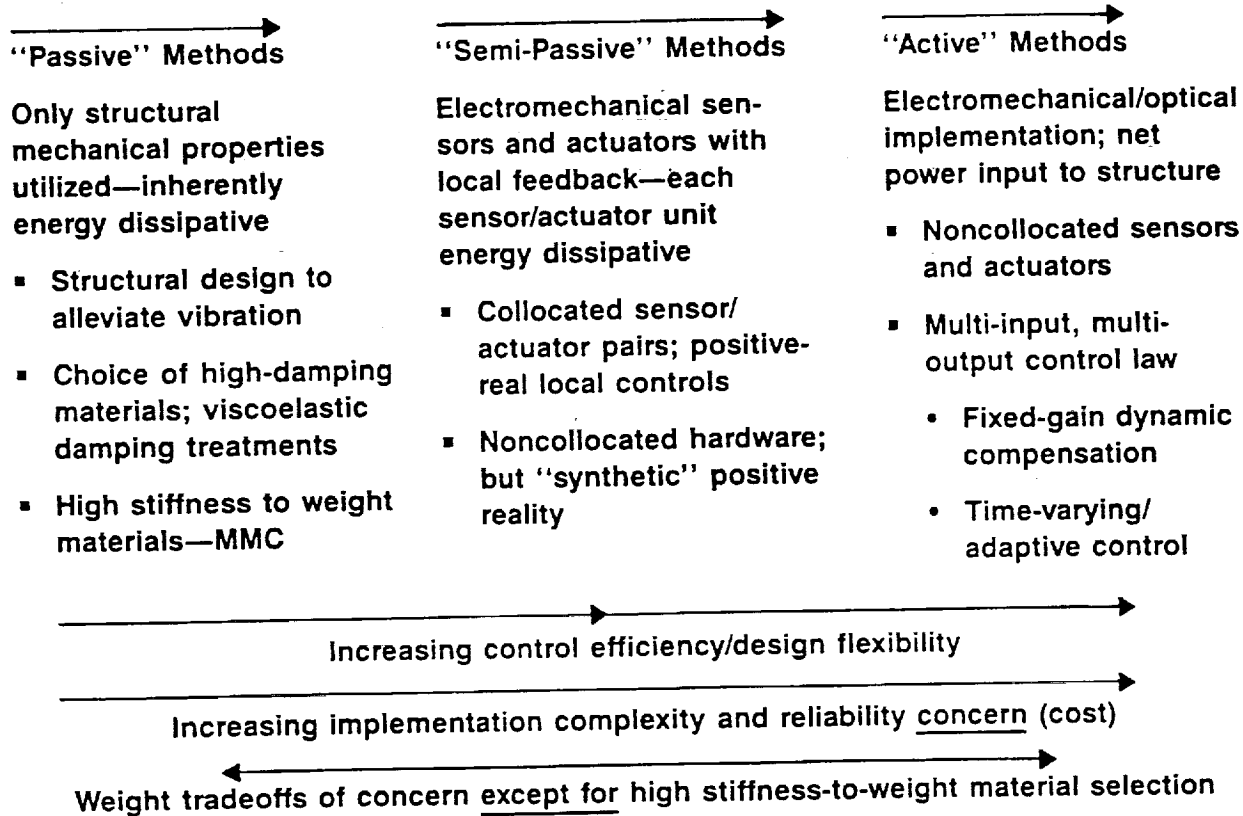


Fig. 17

In summary, we have examined the distinction between “passive” and “active” approaches to vibration suppression for LSS and have found that the distinction is not as sharp as might be thought at first. The relative simplicity, reliability, and cost-effectiveness touted for passive measures are vitiated by “hidden costs” bound up with detailed engineering implementation issues and inherent performance limitations. At the same time, reliability and robustness issues often cited against active control as risk factors are greatly mitigated by recent advances in active vibration control hardware. Accordingly, we see not a sharp “passive versus active” dichotomy, but as illustrated in this chart, a *continuum* of vibration suppression measures offering mutually supporting capabilities. The challenge for LSS vibration suppression is the proper *orchestration* of this spectrum of methods, (via system-level design) to reap the *synergistic* benefits of combined advanced materials, passive damping and active control.

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