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Recent advancements in passive and active vibration control systems

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ABSTRACT

The control of environmental vibration is a prime consideration in designing sensitive electro-optical equipment. Pneumatic isolation systems with low natural frequencies are commonly used to shield the payload from random seismic vibration. In order to further optimize the isolation characterisics, it is desirable to decrease natural frequencies in all translational axes to well below 0.5 Hz, while at the same time achieving maximum damping at resonance. The limitations of conventional systems, however, are such that the lowest natural frequency achievable is 1 Hz. Typical resonance amplification is measured at approximately 10 to 20 dB.

The use of low-frequency pneumatic isolators in precision machinery with built-in XY-stages (i.e. microlithography, submicron inspection systems) highlights an additional problem. Low natural frequencies are equivalent to low spring stiffnesses. Inertial forces applied to mass-spring systems with low spring stiffnesses results in large dynamic deflections. This kind of motion has an impact on the stability and settling time of XY-stages mounted on the mass spring system. The positioning controller of the XY-stagemust overcome these disturbances which are induced by the reactingmass-spring system. The softer the isolation system, the larger the dynamic deflection of the mass-spring system will be. It would be ideal to have a system that would act like a perfectly soft suspension system for all floor induced motion, while simultaneouly creating an infinitely stiff system for all payload-induced forces. This can only be accomplished by using servo-control mechanisms with feedback and feed forward capabilities. This technology is introduced and analyzed in the body of this paper.

1. TERMINOLOGY OF ACTIVE AND PASSIVE VIBRATION CONTROL PRODUCTS

In order to clearly distinguish between the various alternative isolation systems and technologies, definitions of the terminology employed in this paper have been provided.

Vibration isolation systems may be categorized as active orpassive. The categorization is dependent on whether or not external power is required for the isolators to perform their functions. Active isolators are servomechanisms comprised of excitation and corresponse sensors, sensor signal processors, and actuators. The sensors provide signals proportional to dynamic excitation corresponse quantities. The signal processors modify and combine sensor signals to create a command signal. The actuators apply forces or induce motions in accordance with the command signal.

Depending upon the nature of the feedback and the actuation mechanisms, active isolation systems can generally be classified as electro-mechanical, pure fluidic (pneumatic or hydraulic), mechano-fluidic (mechanopneumatic or mechanohydraulic), electrofluidic (electropneumatic or electrohydraulic), electric-fluid (electrically conductive or magnetic fluids) or electrodynamic. Although any one of these systems may be suitable to solve a particular problem, the electrodynamic isolation system offers a number of advantages that make it more suitable for a wide range of seismic vibration problems.

In order to distinguish between the different categories, the following four group definitions are presented:

* Passive Isolation Systems
* Semi-Active Isolation Systems
* Active Isolation Systems
* Adaptive Isolation Systems

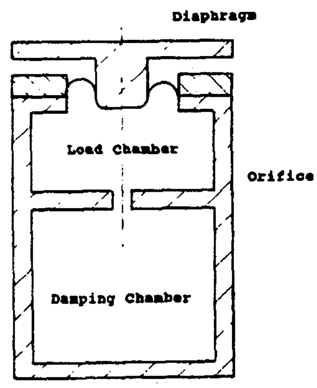


Fig. 1: Conventional Pneumatic Isolator

1.1. Passive isolators:

The two most essential components of a passive isolator are its load-supporting means (stiffness) and its energy dissipating means (damping). Typical passive isolators employ metallic springs, elastomers, wire cables, or pneumatic springs. They do not require any external power in order to function properly.

1.2. Semi Active isolators:

The term "semi-active" vibration isolators describes isolators which require external power. However, this external power is not used to change the dynamic properties of the isolator.

A perfect example of semi-active vibration isolators would be conventional pneumatic isolators (mechanopneumatic). These isolation systems are usually comprised of a piston-cylinder assembly with a rolling diaphragm as a seal. Two air chambers are connected through an orifice. A schematic diagram is shown in Fig.1.

Mechanical displacement feedback controls the flow of compressible gas to and from the actuator through the servovalve. The effect of displacement feedback actuation of the valve spool is to make the actuator output force a function of the time-integral of relative displacement - See Reference List

The integral displacement control operates in such a fashion that no static deflection results from mass loading. Transient deflections which occur in response to a sustained acceleration condition are initially reduced and eventually eliminated. Integral displacement control is designed to be effective only at extremely low frequencies so as not to materially effect the isolation system's natural frequency. Consequently, vibration isolation is provided essentially in accordance with the stiffness and damping characteristics of the passive pneumatic isolator.

Fig. 2 shows a typical transmissibility function of a mechano pneumatic system including maximum loop gain. The closed-loop-transfer-function for the relative displacement feedback loop is shown in Fig. 3. It shows that the relative displacement loop is only effective in a relatively narrow frequency band from DC to 0.5Hz.

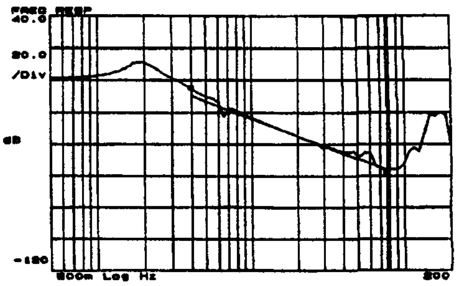


Fig. 2: Typical Vertical Transmissibility of Conventional Pneumatic Isolators

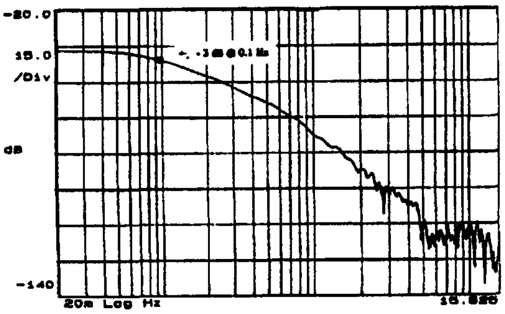


Fig. 3: Closed-Loop-Transfer Function of Mechanopneumatic Feedback Systems

1.3. Active vibration isolation systems:

Truly active isolators are capable of feedback and/or feedforward functions. They have sensors and actuators that permit changes to the isolators characteristics. Utilization of this concept provides the user with the full capability to design a resonance-freesystem. Within the limitations of today's sensor technology, systems have been built that isolate vibration down to 0.1 Hz.

Active systems can also interact with the dynamics of their payloads. They cannot be accurately characterized without taking into consideration the transfer function of the payload mass. Fig. 4 demonstrates the maximum performance characteristics (transmissibility) of an active isolation system with different payload structures. The ultimate objective of the system is to provide maximum vibration isolation for any and all payloads.

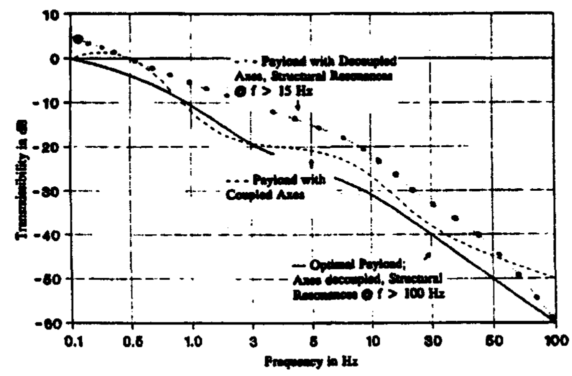


Fig. 4: Transmissibility Performance of an Active System with Different Payloads

1.4. Adaptive isolators:

As stated above, active isolation systems always interact with their payload mass. Their isolation and transient response profiles change in relation to changes in the mass structure and the disturbance profile. For example, a system that was initially intended to provide a high degree of high-frequency isolation might not be sufficient if its environment changes due to building modifications. In order to compensate for these variations, intelligence can be designed into an isolation system. By definingthe required performance characteristics, an adaptive system will independently calculate the ideal control algorithms. Furthermore, it will adapt to any changes either by modifying its entire control strategy or by modifying its fixed control system parameters.

2. ADVANCEMENTS IN SEMI-ACTIVE ISOLATION SYSTEMS

Presently, more than 90% of all seismic vibration problems aresolved by employing semi active pneumatic isolation systems. Conventional systems are usually comprised of a set of two chamberisolators. The two chambers are connected through an orifice. A rolling diaphragm provides almost hysteresis-free vertical motion, while at the same time sealing the pressurized air inside the isolator. The drawbacks of these systems are:

- vibration isolation is limited to natural frequencies greater than or equal to 1 Hz
- non-linear damping characteristics; damping is a function of isolator amplitude
- leveling feedback gain of mechanopneumatic systems is limited by the stability criteria of the P or PI feedback pressure control loop
- mechanical valve lever arms act like a mechanical linkage between floor and payload at certain amplitudes
- large deflections occur as a result of the payload-induced inertial forces

Theoretical and experimental analysis on alternative design concepts have resulted in the following advancements in semi-active isolation systems:

2.1. Improved diaphram design for ultra-low vertical natural frequencies:

It has been determined through extensive analysis on the various stiffness components in pneumatic springs that the only limitation to lowering the vertical natural frequency significantly below thetypical 1 Hz limit is the stiffness of the diaphram. Fig. 5.introduces a new design concept which compensates for the limitations of the diaphram. The conventional method is shown, on the left-hand side, using a convoluted diaphragm. On the right, then ewly developed design is introduced.

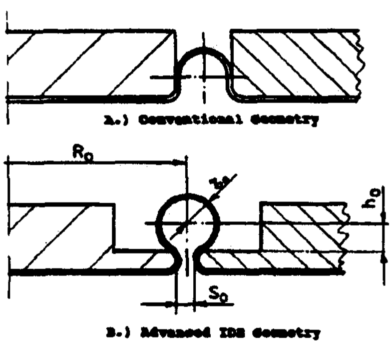


Fig. 5: New Piston-Diaphragm Geometry for IDE Pneumatic Isolators

Analyzing the theoretical stiffness function yields the equation:

$$k_{vertical} = ((ka^*p_{int}^*A^2{}_o)/V_o) - 2^*\pi^*R_o^*(h_o/s_o)^*p_{int} \qquad \text{Equation 1.}$$

Equation 1. clearly shows a negative stiffness element. This is related to a decrease in the effective piston area for downward piston movement - See Reference List

In comparison with the conventional design, thenew approach allows for lower natural frequencies. Fig. 6 demonstrates that using the same air volume, but different diaphragm designs, generates different natural frequencies. The lowest, practical natural frequency measurment was 0.62 Hz. Attempts to lower natural frequency beyond this level resulted in instability.

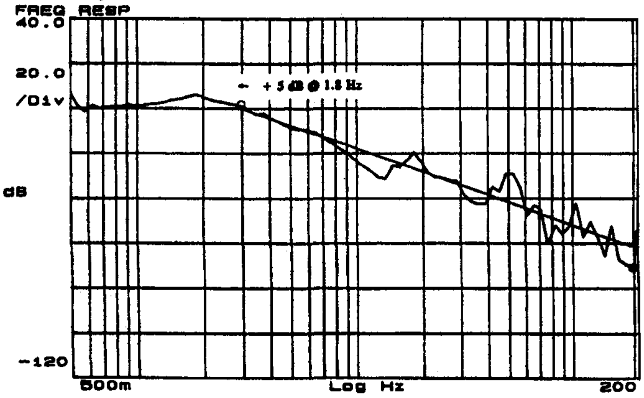


Fig. 6: Vertical Transmissibility of IDE Pneumatic Isolators

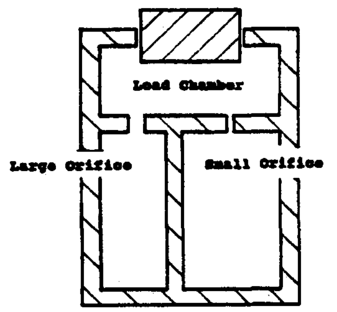


Fig. 7: "Three-Chamber-Design" of IDE Pneumatic Isolators

2.2. The "Three-Chamber-Damping" Concept:

Damping in pneumatic springs is usually referred to as orifice damping. The primary air chamber of the mount is connected to a secondary chamber (also referred to as the "Damping Chamber") through a flow restrictor (orifice). If air is compressed in the upper chamber, a pressure differential is created causing airflow through the orifice. The loss in kinetic energy is proportional to the pressure drop that the airflow experiences while being forced through the orifice.

It is obvious that pressure (energy) loss is related to the velocity of the air. Therefore, for a given orifice size, large isolator amplitudes result in large pressure losses, and conversely smaller amplitudes result in small pressure losses.

Most pneumatic systems are applied to equipment which require seismic amplitude isolation (0.1 to 10 micrometer). The systems must simultaneously react to the inertial forces of XY-stages, typically resulting in amplitudes of 2 mm. Therefore, it is impossible to design an ideal orifice for all amplitudes.

Using three air chambers rather than two, and combining the three chambers by using two orifices with different effective areas can improve the performance of the system.

Fig. 7 introduces this new concept. A "typical load chamber" is located beneath the piston. Two "damping chambers" are used with differing volumes. Both chambers are connected with the "loadchamber" through an orifice.

Assuming small isolator amplitudes (strokes), the large orifice to the left does not act as a real flow restrictor. Hence, the upper and the lower chambers essentially create one large chamber which provides a low natural frequency. The "large chamber", however, is connected via the small orifice with the large right chamber. This chamber may now be tuned to generate optimal damping at amplitudes of 0.1 to 1 micrometer. When the isolator undergoes amplitudes as great as 1 mm, large amounts of airflow will be forced through the orifice. The left chamber, however, is blocked from any significant airflow. It virtually does not exist. However, the right chamber can make perfect use of its orifice. Optimal damping is achieved in both cases.

Fig. 8 shows the typical performance characteristics of a "three-chamber-isolation-system".

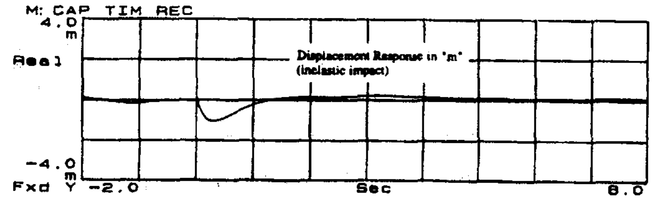


Fig 8: Step-Response Characteristics of IDE "Three-Chamber-Design" Pneumatic Isolators

2.3. Electronic leveling and feedback system performance:

The influence of mechano pneumatic leveling valves on the performance of the isolator is significant when working with seismic level, sub-micron vibrations. Mechanical "Stick-Slip"effects are introduced; the valve demonstrates highly non-linear behaviour and often drastically impairs the performance of the pneumatic isolator. Secondly, the valve spool must be consistently linked to the payload in order to make a relative reference, ultimately generating a higher overall stiffness.

A digital, electronic system has been developed. The system uses aneddy-current sensor to measure the relative position of the isolator piston (payload relative to the ground). The signal from the sensor, which is proportional to the piston, is amplified and fed into a small computer. The computer uses PID algorithms in order to calculate the appropriate output signal for a electro-pneumatic proportional valve. The valve uses this input signal togenerate a directly proportional output pressure. Non-linear sensitivity functions of the valve, the isolator and/or the sensor can be accommodated for in the computer. The software algorithms provide compensation and optimize the control algorithm parameters for optimal functioning and maximum system gain.

Fig. 9 shows the control system block diagram. A comparison of the closed-loop-system gain function for mechanopneumatic and electro-pneumatic systems is shown in Fig. 10. The left plot proves that the mechanopneumatic system maintains stability at maximum gain if the active band width does not exceed 0.5 Hz. Using the electro-pneumatic system, the bandwidth increases to 2 Hz (these closed-loop-transfer-functions have been taken at a disturbance amplitude of 0.5 mm).

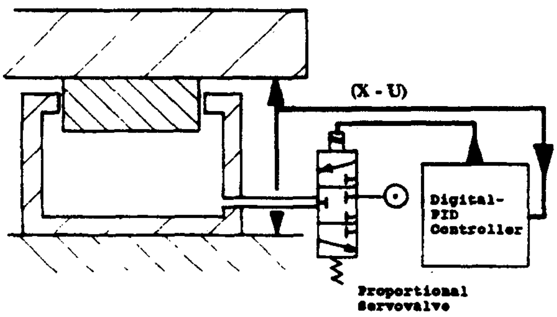


Fig. 9: Block Diagram of Electronic Leveling System Type IDE PID

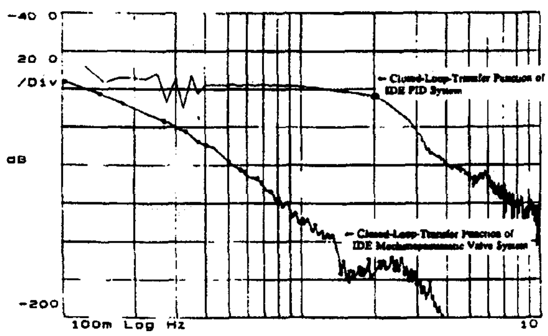


Fig 10: Closed-Loop-Transfer Functions of Mechanopneumatic and IDE PID Electropneumatic Leveling Systems

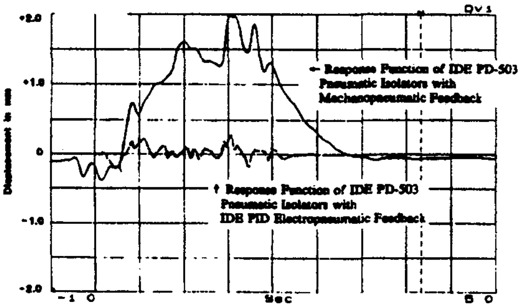


Fig 11: Response Function of IDE PID Electropneumatic Leveling System during a Shift of XY-stage (mounted on top of 1500 kg payload) with constant velocity profile

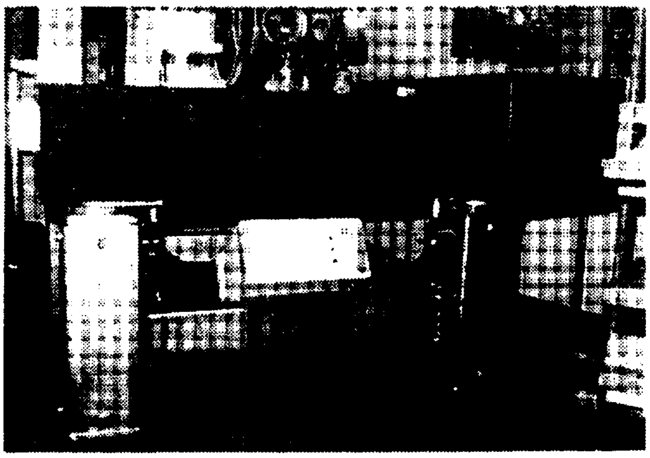


Fig. 11 offers the results of tests on a 2500 kg payload, used with four isolators (type IDE PD-503) and the PID electro pneumatic system. Dropping a 500 kg mass from approx. 100 mm onto the isolators has the described effect in the time domain. The difference is depicted by the dotted line, which describes the performance of the mechanopneumatic system. Fig. 12 shows the application of the PID leveling system in conjunction with a high-speed Coordinate-Measuring-Machine.

3. RECENT ADVANCEMENTS IN ACTIVE VIBRATION CONTROL SYSTEMS

3.1. Present State-of-the-art in Active Systems:

Active vibration isolation systems have been in existence since the early 1960's. The technology was re-introduced to seismic level vibration isolation problems in the early 1980's. It is currently being used for applications in numerous research facilities and laboratories around the world, using analog electronics or modern DSP based systems. These systems are integrated into passive pneumatic springs or their elastomer counterparts, with accelerometers or velocity sensors attached.

Fig. 13 shows the principal of an active system with a sensor mounted on the payload. The purpose of the control loop is to minimize the absolute payload motion. Independent of the source of the forces, a reacting force is applied to the payload. It is equivalent to the product of a gain term times the measured signal. Depending on the type of feedback (absolute displacement, velocity or acceleration) different system characteristics are produced. Assuming absolute acceleration feedback, the payload mass, which is a sum of the real payload mass and a virtual mass, can be increasedby increasing the feedback gain. Using absolute velocity feedback, a damping equivalent term is introduced into the equation.

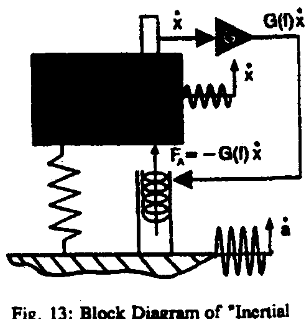


Fig. 13: Block Diagram of "Inertial Reference" Active Vibration Isolation System

However, in contrast to conventional damping, which is proportional to relative velocity between the payload and the base, active electronic damping is only proportional to the absolute motion of the payload. This is commonly referred to as the "Skyhook" concept. Displacement feedback adds a stiffness proportional term. However, as in the case of the velocity proportional feedback, the "electronic-stiffness" component is tied to "outer space" by its link to absolute motion. Other configurations of closed-loop vibration isolation systems use feed forward techniques. "Feedforward" can include sensing of floormotion and inducing this signal into a feedforward loop. The benefit of this approach is that the effect of the passive isolator, which is similiar to that of a low-pass-filter, can be overcome. The floor vibration signal can be used in an unfiltered manner, thus broadening the dynamic response capabilities of he active system. Other feedforward techniques include the measurement of payload-inherent signals, which avoid control system instabilities due to structural resonances. These signals are used without the phase shift that the structure can add. Finally, other signals can be used, including information from XY-stage-drivers. These approaches provide the potential for improved performance of an active isolation system. Their implementation, however, is often complicated due to their structure-specifictransfer functions.

Fig. 14 shows an example of a control system with normal payload-motion feedback capabilities and floor-motion feed forward techniques. Typical transmissibility functions for this system (Integrated Dynamics type TC-1) with feedback and/or feedforward performance is depicted in Fig. 15. This system is used for the isolation of a "Field-Emmission-Microscope (FEM)". Fig. 16a shows the FEM on the isolated platform.

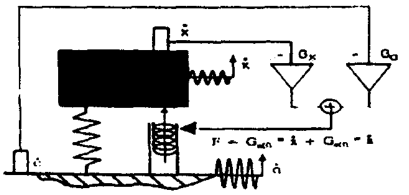


Fig 14: Block Diagram of Active Vibration Isolation System Type IDE TC-1

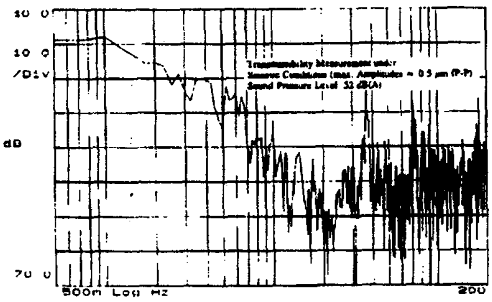
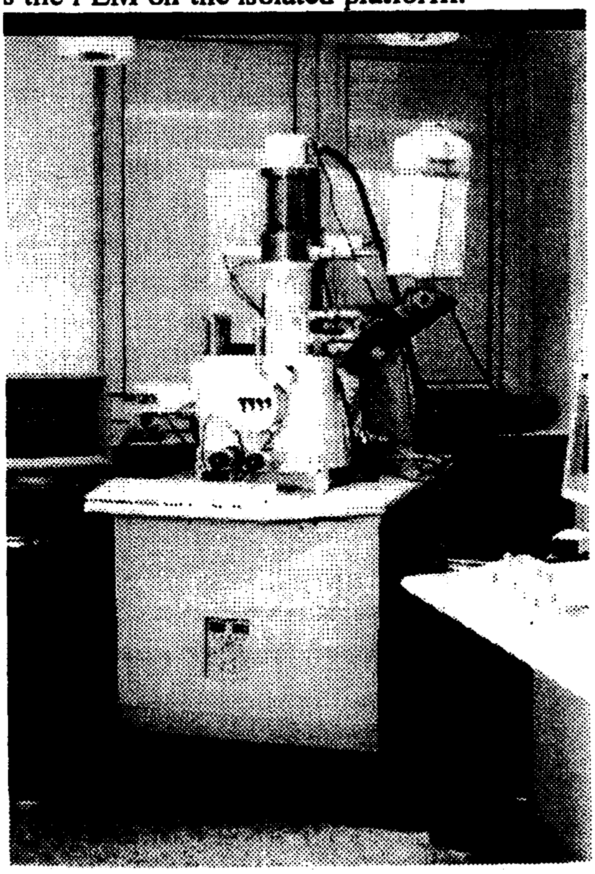


Fig 15 Vertical Transmissibility Performance of IDF TC 1 Active System in comparison with Field-Emission-Microscope



Active vibration isolation systems are currently being successfully used to solve actual on-site industrial vibration problems. Their performance capabilities are widely recognized and accepted. Installations have been successfully completed in a wide range of application areas.

1 6 1 9 ## 4. ADAPTIVE VIBRATION CONTROL

4.1. Features of an ideal vibration isolation system:

Is mentioned above, most of the active isolation systems on the market today rely on analog electronics. Although typical analog systems are far less expensive than their digital counterparts, there are some significant disadvantages in using them, such as:

- adjustment and calibration is payload specific and requires engineering time for every installation
- changes in the payload require additional adjustments and calibration procedures
- attainment of optimal performance is dependent on the experience of the servo engineer responsible for the installation
- coupling horizontal and vertical axes limit the achievable gain of the active system
- adding feedforward techniques demands new PC-board layouts for specific applications
- service and repair functions must be performed by a servo engineer
- performance characteristics must be flexible in order to accommodate the needs of different applications (tradingmaximum high-frequency isolation vs. low-frequency isolation,and/or structural damping vs. low-frequency isolation)

Generally speaking, once an analog active system is installed, no changes are allowed to the payload. Any changes to the environment of the control system will result in instability or a loss of performance. Avoiding this problem requires a certain level of intelligence inside the servo controller in order to adapt to any changes.

There are additional factors which must be present in order to build a truly "state-of-the-art" active isolation system:

- EMI generation by electromagnetic motors must be kept at a minimum (i.e. impact on E-beam equipment)
- heat generation of force motors must be controllable
- mechnical design must be user-friendly for easy enduser setup
- there must be a balance of low-frequency positioning loops and high-frequency electromagnetic motors in order to optimize XY-stage servo settling time and throughput

Active systems for vibration isolation control should be evaluated beyond the traditional transmissibility performance. Some of the necessary performance parameters for a modern seismic isolation system are:

- adaptive control system (self—tuning for changes in the payload and/or the disturbance profile)
- dynamic range (vibration amplitude): 90 dB - (i.e. calibrated for 2 mm -> 0.05 micrometer)
- active bandwidth : DC to 100 Hz (while allowing greater than or equal to 10 structural resonances from DC to 100 Hz)
- error rejection capability of 30 dB or more for 2 Hz to 100 Hz
- EMI generation : 0.1 milli Gauss
- acceleration settling time : 20 msec
- frequency response of positioning loop : -3 dB at 2 Hz
- relative positioning accuracy: 5 micrometer
- controlled degrees of freedom : 6 (feedback)
- feedforward channels: 6 (floor motion in 6 D.o.F)
- maximum force output capability: 200 N per axis

4.2. The control system concept and its "brain":

Fig. 17 shows the block diagram of Integrated Dynamics' answer to the above requirements. In order to provide the necessary dynamic range, a "floating-point" DSP is used as a central CPU (Texas-Instruments TMS 320C30). A theoretical dynamic range of 108 dB can be calculated using a multi-channel multiplexing unit in conjunction with a high-speed, 18 bit A/D converter. A total of twelve (12) D/A-converters with similiar resolutions and speeds and 16 digital I/O-ports complete the control system hardware.

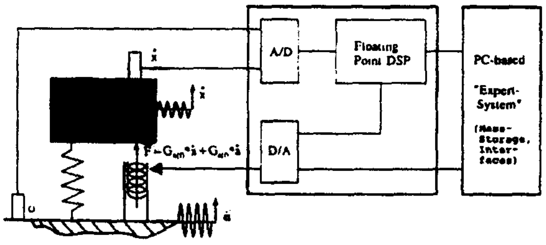


Fig 17: Block Diagram of an Adaptive Vibration Isolation System

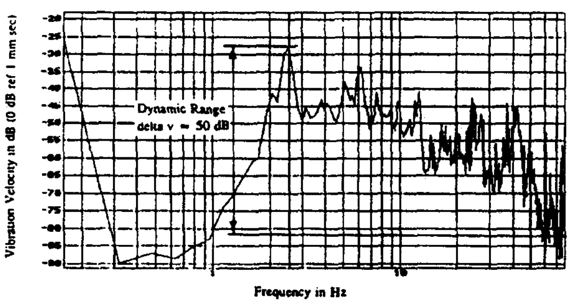


Fig 19: Typical Measurement of Floor Vibration Velocity Spectrum

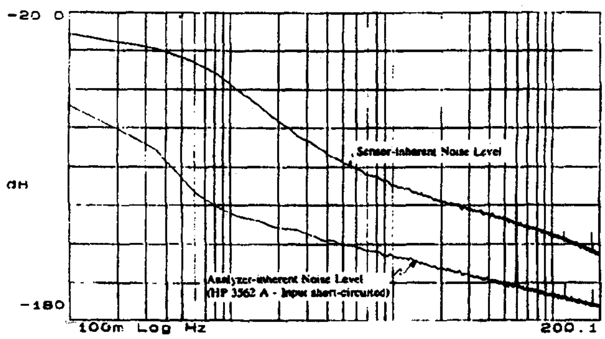


Fig 18: Inherent Noise Measurement of IDE Inertial Sensors (0 dB ref 1 µm RMS)

In addition to the DSP control system, a PC is used to interface with the system. The PC serves several functions. Primarily, it contains the software to provide intelligence to the active system. Additionally, the PC contains algorithms that replace the servo engineer and allow the system to adapt to any change in the payload. Furthermore, it adds mass-storage functions and user interfacing, as well as modem capability for remote control and final storage for all significant changes which effect the system.

4.3. The sensor technology:

The absolute payload velocity is sensed. An analysis of a typical vibration spectra of building floors in semiconductor and research facilities has proven that velocity measurements demand the smallest amplitude dynamics. Both acceleration and displacement can vary by more than 60 dB on the floor. If the transfer function is added, (typically 40 dB) at least 100 dB will be necessary. This does not allow for any payload induced motion.

Fig. 18 shows the sensor's noise profile in RMS displacement vs.frequency, for the system's velocity sensor. A typical floor measurement with this sensor is shown in Fig. 19. In comparison with accelerometers, this sensor is insensitive to mechanical shocks. Unlike displacement sensors, it is almost in sensitive to temperature variations, and does not require any calibration ormechanical alignment.

4.4. The periphery:

The force motors are linear electromagnetic with a maximum force capability of 50 N each. A total of four motors in each axis add sup to 200 N (44 lbs). The motor design is contactless and incorporates EMI shielding techniques. Pneumatic springs are used to carry the payload and lower the payload impedance. They are of a "three-chamber-design" for optimal performance and maximum linearity. The leveling function is conducted by electro pneumatic servovalves. The control function for these valves, however, is coordinated and provided by the DSP system.

4.5. The software strategy:

The major improvements to this system lie in the capabilities of the software package. As stated above, the inclusion of a closed-loop-control function did not present the challenge to this development effort. The challenge was in ensuring the system's ability to adapt to changes in the payload. This is the only approach which ensures maximum performance for all payloads, under all conditions. Therefore, the software is divided into two separate packages: 1 .The closed-loop-control system and 2. the algorithm providing adaptive, but off-line measurements and parameter correctingfunctions.

4.6. Performance characteristics:

Fig. 20 shows the theoretical transmissibility performance of this configuration compared with a conventional pneumatic system. Fig.21 offers insight into the dynamic response behaviour. The plotshows a typical response in the time domain to an in elasticvelocity impact ("sandbag drop").

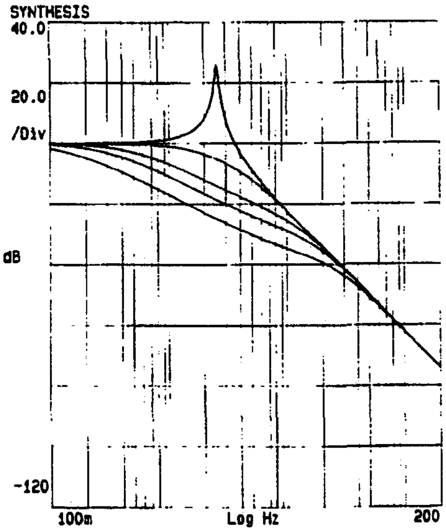


Fig 20 Theoretical Transmissibility Functions of an Adaptive Vibration Isolation System with Varying Gain

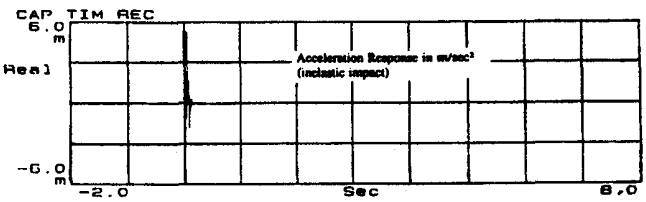


Fig 21: Step-Response Characteristics of an Adaptive Vibration Isolation System

5. Conclusions

The inherent problems of the classic "active vibration isolation system" prompted the development of a more efficient and effective alternative solution. The result is a new concept in vibration isolation called "Adaptive Systems". These systems are able to adapt to changes in the payload or in the disturbance profile, ultimately resulting in superior performance of the effected equipment.

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