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DEPC-EM 007**

**Critical review of
seismic vibration
isolation techniques**

**NMS Programme for Length
2002 - 2005**

S P REILLY and R K LEACH

SEPTEMBER 2006



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Critical review of seismic vibration isolation techniques

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S P Reilly and R K Leach
Engineering and Process Control Division

ABSTRACT

Vibration isolation is needed in many areas of metrology, including high accuracy measurements of force, mass and length. A literature review of vibration isolation systems has been undertaken and its findings are summarised in this report. The report provides an overview of basic vibration diagnostic and isolation concepts, and examines specific techniques and systems that have been purpose-designed or are commercially available. Particular emphasis has been placed on low frequency vibration isolation (less than 10 Hz) as higher frequency isolation and diagnostic systems are more readily available commercially.

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National Physical Laboratory
Hampton Road, Teddington, Middlesex, TW11 0LW

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1 INTRODUCTION

The National Physical Laboratory is developing traceable low force measurement systems and actuation systems as low force transfer artefacts. The technical details of this project can be found elsewhere (Leach *et al* 2005). Areas to which this work will be of particular interest include: forces exerted by scanning probe microscopes, forces required for biotechnology testing methods, frictional forces limiting the dynamic performance of various types of micro-electro-mechanical systems (MEMS) and the forces encountered by nano-indentors used for materials characterisation. The low force measurement system that is being developed will require a high degree of vibration isolation due to its inherent weight relative to the very small forces being measured (ten orders of magnitude difference). This report provides an overview of basic vibration diagnostic and isolation concepts and then examines specific techniques and systems, with particular emphasis on low frequency vibration isolation (< 10 Hz). The report names manufacturers products relevant to this guide, however, the appearance of a manufacturer in this report is not an endorsement of its products or services.

2 THE SEISMIC VIBRATION SPECTRUM

Different physical influences contribute to different frequency bands in the seismic vibration spectrum, a summary of which is shown in Table 1.

Table 1 Sources of seismic vibration and corresponding frequencies (Newell et al, 1997)

Frequency / mHz	Cause of vibration
< 50	Atmospheric pressure fluctuations
50 to 500	Ocean waves (60 mHz to 90 mHz fundamental ocean wave frequency)
> 500	Wind blown vegetation and human activity

Figure 1 shows measured vertical amplitude spectral densities for a vibrationally ‘noisy’ and a vibrationally ‘quiet’ area (Newell *et al*, 1997). Note that the spectrum below 0.1 Hz is limited by the seismometer’s internal noise. The solid curve represents the vibration spectrum on the campus of the University of Colorado, Boulder. The dashed curve is that from the

National Institute for Standards and Technology (NIST) site. The ‘quiet’ NIST laboratory is small, remote and separated from the main complex. In addition, all fans and machinery were turned off during the measurements at the NIST site.

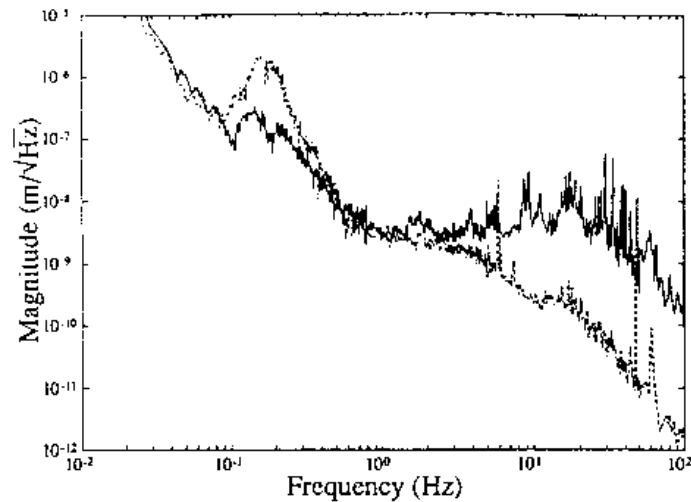


Figure 1 Measured vertical amplitude spectrum on the campus of the University of Colorado and NIST

Most of the increased vibration in the solid line above 10 Hz in Figure 1 can be attributed to human activity and machinery. The low frequency peak in the dashed line can be attributed to naturally occurring environmental effects such as high winds.

For very low frequency vibrations a gravitational wave detector in the form of a Michelson interferometer with 20 m arms, which will be discussed in a later section, was used to measure vibrations 1 km below sea level. A summary of the results is given in Table 2 (Araya, 2002).

Table 2 Possible sources of very low frequency vibration

Source	Period	Acceleration/ m s^{-2}
Earth's free seismic oscillation	$10^2 \sim 10^3$ s	$10^{-6} \sim 10^{-8}$
Core modes	$\sim 10^3$ s	$\sim 10^{-11}$
Core undertone	$10^3 \sim 10^4$ s	$\sim 10^{-11}$
Earth tides	$10^4 \sim 10^5$ s	$\sim 10^{-6}$
Post seismic movements	$1 \sim 10^3$ days	$10^{-6} \sim 10^{-8}$
Crustal movements	10^2 days	$10^{-7} \sim 10^{-9}$

3 DIAGNOSTIC TECHNIQUES

There are several ways in which vibration can be measured and the data subsequently presented. This section will describe some of the techniques for measuring mechanical vibration. Vibration can be measured and analysed in terms of displacement, velocity or acceleration. The displacement, x , of an object vibrating in a single direction with a single frequency is given by

$$x = A\sin(2\pi ft), \quad (1)$$

where A is the displacement amplitude and f is the frequency of the vibration. Differentiating this equation with respect to time, t , yields velocity, v , and differentiating velocity with respect to time yields acceleration, a , given by

$$v = 2\pi f A \cos(2\pi ft), \quad (2)$$

and

$$a = -(2\pi f)^2 A \sin(2\pi ft). \quad (3)$$

Generally, displacement, velocity and acceleration measurements are used to investigate different regions of the vibration spectrum - displacement is used at the lower frequencies and acceleration at the higher frequencies. Accelerometers are not used at low frequencies due to the relatively small inertial forces at low frequencies that will be below the noise threshold of an accelerometer. Therefore, at low frequencies displacement is likely to be measured rather than acceleration, whilst for high frequency, the displacements of the vibrations will be small and an accelerometer will be able to provide a better measurement. Generally, velocity measurements cover the mid-range between displacement and acceleration, typically 5 Hz to 2 kHz (Goldman 1999).

3.1 VIBRATION SENSORS

Different types of sensors are available for vibration measurements that have different frequency ranges. The six most common commercially available sensors are proximity sensors, velocity pick-ups, piezoelectric accelerometers, capacitive accelerometers, laser vibrometers and seismometers.

3.1.1 Proximity sensor

A proximity sensor generates an electromagnetic field at a tip and the action of this tip moving relative to a metal target causes variation in the magnetic field, changing the output of the instrument. Proximity sensors are generally used in a frequency range from zero to a few hundred hertz. However, proximity sensors are generally unsuitable for environmental measurement due to the need for a reference signal, which can be problematic when the surrounding environment is subject to the same vibrations. Proximity sensors are most often used for vibration testing of mechanical machinery.

3.1.2 Velocity Pick-up

A velocity pick-up comprises a magnetic mass on a spring in a bath of oil with a coil surrounding the mass. As the pick-up is subject to vibration the magnetic mass oscillates relative to the coil, as shown in Figure 2.

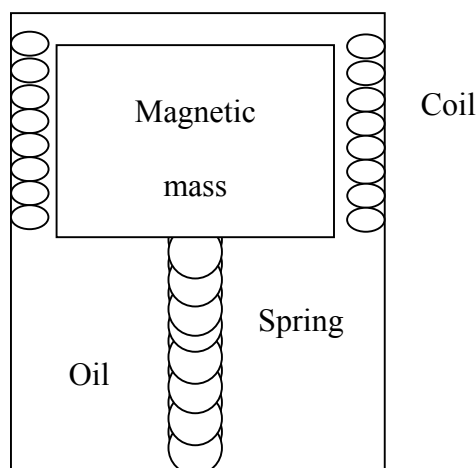


Figure 2 Schematic of seismic velocity pick-up

Velocity pick-ups are generally used to measure vibration in one degree of freedom. The resonant frequency of velocity pick-ups can be adjusted, the weaker the spring and the larger the mass, the lower the resonant frequency. Velocity pick-ups are filled with oil to dampen the resonance in the sensor, but this can over dampen the response. In addition, the frequency response of a velocity pick-up often decreases over 2 kHz and velocity pick-ups are often large enough to cause a disturbance to a small system. Nevertheless, velocity pick-ups are generally cheap in comparison with other vibration sensors.

3.1.3 Piezoelectric

The most common form of vibration sensor is the piezoelectric accelerometer, which is made of a very small mass mounted on a piezoelectric crystal. The crystal acts as a stiff spring and produces a charge under strain induced by the applied or ambient acceleration. The frequency response of a piezoelectric accelerometer is broad, often ranging from 10 Hz to 20 kHz. Piezoelectric accelerometers are widely available and commonplace in many laboratories that require vibration measurement. Piezoelectric accelerometers have emerged as the leading method of sensing higher (>10 Hz) frequency vibration and their wide spread use has contributed to a relatively low cost.

3.1.4 Capacitance

Capacitive accelerometers sense a change in electrical capacitance when subject to different accelerations. The sensing element consists of a parallel plate capacitor, which alters the peak voltage generated by an oscillator when the structure undergoes acceleration. The capacitor plates are attached to a diaphragm, which acts as a mass and bends under an applied acceleration. They are also relatively low cost.

3.1.5 Seismic

There are two types of seismic sensors: inertial seismometers, which measure ground motion relative to an inertial reference (a suspended mass), and strain-meters or extensometers which measure the motion of one point on the ground relative to another. Inertial seismometers are often variations on the velocity pick-up design. However, at very low frequencies it becomes increasingly difficult to maintain an inertial reference, and for the observation of low-order free oscillations of the Earth, tidal motions, and quasi-static deformations, strain-meters may outperform inertial seismometers. Strain-meters are conceptually simpler than inertial

seismometers although their technical realisation can be more difficult. They are also relatively low cost.

3.1.6 Laser vibrometry

Laser vibrometers are commonly used for high frequency, on-line testing of products during a manufacturing process. Laser vibrometers use a reflection from a vibrating surface mixed with a reference signal in an interferometer. A Bragg cell is usually employed to frequency shift the reference beam thus allowing the direction of displacement of the vibrating surface to be determined. However, laser vibrometers have a relatively high cost compared to other vibration sensors. In comparison with mechanical vibration transducers (accelerometers, strain gauges, *etc.*), laser vibrometers have the key advantage of being non-contact, with greater accuracy and without mechanical cross-coupling effects. However, as with proximeters they require a very stable reference and, therefore, may be unsuitable for environmental measurements.

Guidelines for conducting vibration measurements are given elsewhere (Meredith, 1995). A full description of the vibration analysis in preparation for the inauguration of a new facility is also available elsewhere (Bessason and Madhus, 2000). There have been several projects developing specialist equipment to measure low frequency seismic noise. Zhou *et al* (2004) developed a laser interferometer combined with a long-period folded pendulum. Ground tilt has been measured by using a high sensitivity rotational accelerometer (Luiten, 1997) and two dimensional tilt has been measured using a double-flexure two-axis tilt sensor with an optical sensor (Cheng *et al*, 2002). Table 3 summarises various properties of vibration sensors.

Table 3 Summarising the properties of various types of vibration sensor

Sensor	Measurement	Frequency range /Hz	Cost
Proximeter	Displacement	0 – 300	Low
Velocity pick-ups	Velocity	10 – 2k	Low
Piezoelectric accelerometers	Acceleration	10 – 20k	Low
Capacitive accelerometers	Acceleration	0 – 1k	Low
Laser vibrometers	Velocity	0 – 1.5M	High
Seismometers	Velocity	0.01 – 50	Medium

4 PASSIVE VIBRATION ISOLATION

Passive vibration isolating systems, comprising springs and pendulums, can be applied to vertical and horizontal isolation. The resonant frequency of spring and pendulum systems is respectively given by

$$f_0 = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad (4)$$

and

$$f_0 = \frac{1}{2\pi} \sqrt{\frac{g}{l}} \quad (5)$$

where g is the local acceleration due to gravity, l is the length of the pendulum, k is the spring constant and m is the mass. The static restoring force of the spring is $k\delta l = mg$ making the resonant frequency in terms of the compression of a spring

$$f_0 = \frac{1}{2\pi} \sqrt{\frac{g}{\delta l}} \quad (6)$$

where δl is the spring extension. A low resonant frequency is important as attenuation takes place above the resonant frequency and to achieve this using a spring system a large static

extension of the spring is required. It is possible to improve a passive vibration isolation system by using multiple layers of isolation. Multistage passive isolation will not improve low frequency isolation but can have significant effects at high frequencies (Roberts *et al*, 1999). The transmission spectrum of a damped passive isolation system is shown in Figure 3 with a resonant frequency of 200 Hz and a damping factor of 0.01.

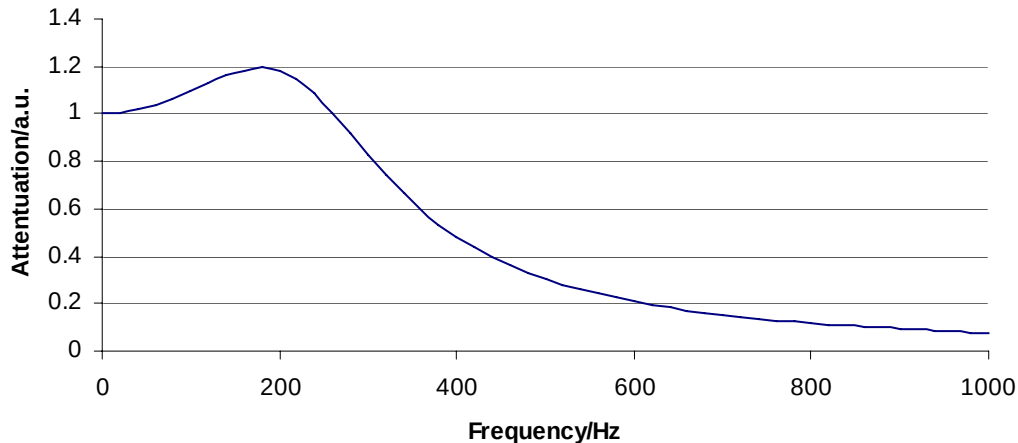


Figure 3 Transmission spectrum for a passive vibration isolation system with a resonance of 200 Hz and damping factor of 0.01

4.1 PASSIVE VIBRATION ISOLATION SYSTEM TYPES AND CHARACTERISTICS

The springs required in passive isolation systems can take several forms. They can be divided into three broad categories: pads, helical springs and air springs.

Pads are typically manufactured from fibreglass, felt, cork or neoprene. Pad systems are normally resonant at 5 Hz to 30 Hz but have a high damping rate at their natural frequency. However, pads can exhibit creep, the pad can stiffen with time and their natural frequency will slowly increase with time, as the spring constant increases (Meredith, 1995).

Springs, in their traditional helical form are available in many different sizes and load carrying capabilities. A spring system's resonant frequency is often between 1 Hz to 6 Hz and their damping rate is fairly low so amplification at resonance may be a problem and, therefore, require damping.

Air spring systems offer natural frequencies between 0.75 Hz and 4 Hz. Air springs have the lowest spring constant and are the most adaptable springs as the user can adjust the air pressure to suit the load carrying capacity or stiffness required. Air springs can be fitted with an active system combining a height sensor in a control loop with automated adjustment of air pressure. Integrated Dynamics Engineering, Inc. produce a commercially available passive pneumatic system with a resonance frequency of 0.8 Hz.

The inertia base is also an important part of any vibration isolation system. Typically made from steel or concrete and de-coupled from the floor by a passive vibrational isolating element, inertia bases or plinths provide a number of benefits for the user. For example, the base lowers the centre of gravity and minimises the tendency of the equipment to tilt on its isolators.

4.2 NEGATIVE K SYSTEMS

As noted in section 4 a very low resonant frequency is beneficial. A negative k system reduces the spring constant to vary the resonance of a system, by use of a counter spring. The action of a second spring works as a counter force which, when the springs act together, produces a very low spring constant, lowering the resonance. Negative k systems in an inverted pendulum format have been developed both in one dimension (Blair *et al*, 1994) and two dimensions (Winterflood and Blair, 1996) with resonances under 50 mHz. A commercial system is also available (Minus- k Technology); vertical-motion isolation is provided by a stiff spring that supports a weight load, combined with a negative-stiffness mechanism. The net vertical stiffness is made very low without affecting the static load-supporting capability of the spring. Figure 4 illustrates a negative k spring vertical isolation system.

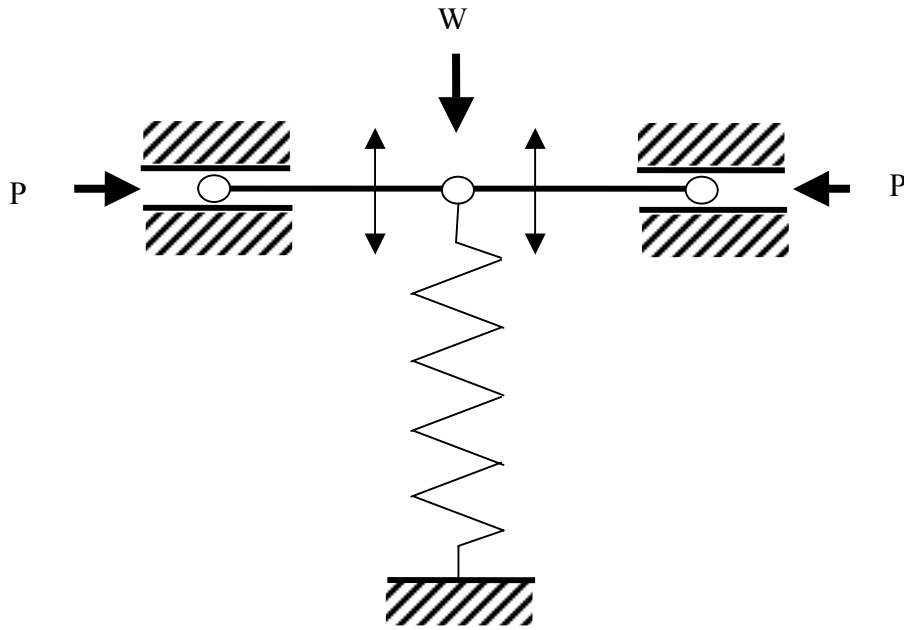


Figure 4 Negative k spring vertical isolation system. W is the weight on the system and P is the compression on the pivots

The negative k system shown in Figure 4 uses a conventional spring connected to a negative spring mechanism consisting of two bars hinged at the spring, supported at their outer ends on pivots, and loaded in compression by forces, P . The spring is compressed by weight, W , to the operating position of the isolator, as shown in Figure 4. The stiffness of the isolator is $K = K_S - K_N$ where K_S is the spring stiffness and K_N is the negative stiffness, which is a function of the length of the bars and the load, P . The isolator stiffness can be made to approach zero while the spring supports the weight, W . Therefore, the system requires tuning to the required payload.

A horizontal motion isolation system consisting of two beam column isolators is shown in Figure 5. Each isolator behaves like two beam columns loaded axially by a weight load, W . Without the load the beam columns have horizontal stiffness, K_S . With the weight load the lateral bending stiffness is reduced by the ‘beam column’ effect. The beam column effect is equivalent to a horizontal spring combined with a negative-spring mechanism so that the horizontal stiffness is $K = K_S - K_N$, and K_N is the magnitude of the beam column effect. Horizontal stiffness can be made to approach zero by loading the beam columns to approach their critical buckling load.

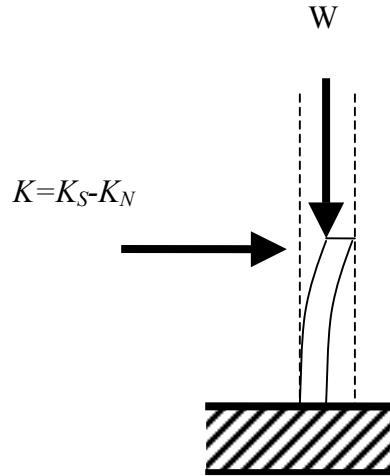


Figure 5 Horizontal axis vibration isolation systems

The Minus-k system has passive tilt suppression comprised of a tilt flexure with damping that provides isolation against tilt in either direction as shown in Figure 6. The Minus-k uses three isolators stacked in series: a tilt-motion isolator on top of a horizontal motion isolator on top of a vertical motion isolator. This gives vibration isolation in all six degrees of freedom.

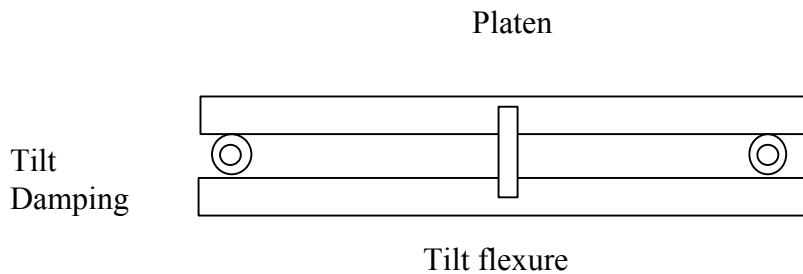


Figure 6 Tilt flexure vibration isolation mechanism

4.3 DAMPING

Vibration damping is a key element in vibration suppression. All physical systems have some inherent damping, but the level of damping can be augmented to increase energy dissipation in particular vibration modes by the use of, for example, dashpots. In this way, the response of a structure driven at a resonant frequency can be greatly decreased. This in turn can significantly reduce overall motion or acceleration of the structure. However, velocity-

dependent damping, which is generally known as viscous damping, does not affect the low frequency performance of the system (Beranek, 1988). A variety of damping techniques are available including dashpots and eddy current dampers.

4.4 INTERNAL RESONANCES

A limitation to high frequency vibration isolation systems is caused by internal resonances of the object being isolated or the isolation structure itself (Beranek, 1988). The typical resonances of the objects or the isolation structures are around a few hundred hertz, for example, a typical Newport table has an internal resonance of 160 Hz (Roberts *et al*, 1999).

5 ACTIVE VIBRATION ISOLATION

Active vibration isolation can generally extend the low frequency isolation of a system. Active systems use signals acquired by sensitive vibration detectors, then apply an opposing signal to actuators that produce the counter-force to compensate for the vibration. An advantage of active vibration isolation over passive isolation is that through optimising the control loop, it is possible to make the resonance of the isolation system negligible, and tuned to the mass of the system to be isolated (Sandercock, 1987). A disadvantage of active systems is that feedback can often excite resonant modes in the system.

5.1 HEXAPODS

Single degree of freedom isolation systems are of little practical use as the vibrational noise can be reintroduced through the other degrees of freedom. Therefore, to be effective an active vibration isolation system should operate in six degrees of freedom. Hexapod structures (Rahman *et al*, 1998) are capable of movement in all six degrees of freedom and can be classified into two basic categories. The first category is the hard active mount, where a stiff actuator, such as a piezoelectric actuator, is used either alone or in series with a soft spring. The second is a soft active mount, which uses a soft actuator, typically a voice coil, in parallel with a soft spring. Figure 6 shows a typical hexapod and a summary of hexapod designs is given in Appendix A.

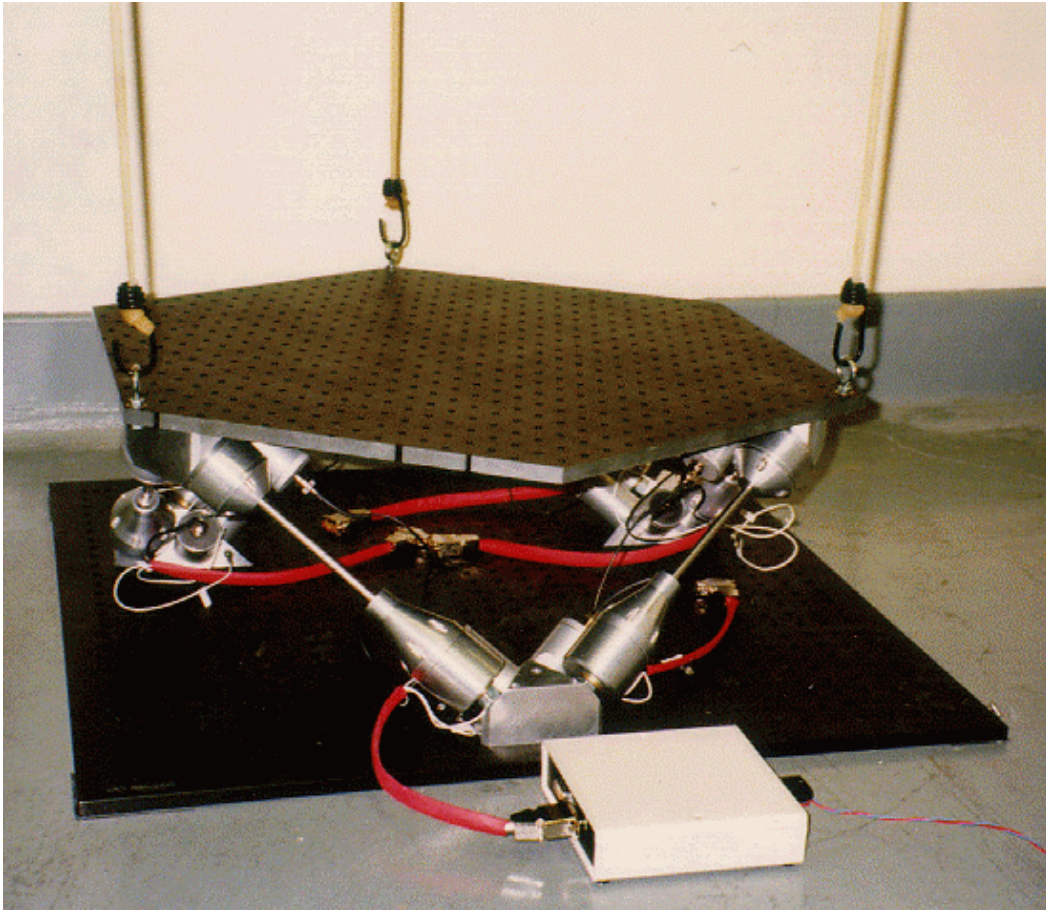


Figure 7 A typical hexapod structure (courtesy of Cornell University)

5.1.1 Hard active mount systems

A hexapod system produced by Intelligent Automation, Inc. uses a magneto restrictive material known as Terfenol-D as a stiff actuator (Geng and Hayes, 1994, Geng *et al*, 1995). Load cells and accelerometers are the feedback sensors and can be used alone or in combination to provide adaptive control. The Intelligent Automation struts have not been designed for any passive isolation capability and use an internal spring to off-load the payload mass from the actuator. The Intelligent Automation system has an active vibration isolation bandwidth of approximately 10 Hz to 200 Hz and an attenuation of approximately 30 dB at peak resonance.

A hexapod has been designed by Draper Labs (Henderson, 1996) specifically for a large payload (a 102 inch diameter Cassegrain telescope). As with the Intelligent Automation system, the control bandwidth requirement is 10 Hz to 200 Hz and no passive vibration

isolation capability is included. The Draper design struts use piezoceramics for their actuators with load cells providing force feedback. A 20 dB performance improvement in vibration isolation is achieved across the 10 Hz to 200 Hz frequency band.

Another hexapod with piezoceramics was built by the Harris Corporation known as the “Active Isolation Fitting” (Hyland *et al*, 1996). The Harris Corporation approach uses both base and payload accelerometers for control. A servo loop is closed around each sensor, with the base accelerometer loop providing feed-forward cancellation of incoming disturbances and the payload accelerometer loop inertially stabilising the payload. Implemented on different testbeds, performance of up to 30 dB attenuation over a 10 Hz to 200 Hz bandwidth was accomplished.

A further hard active mount system is the ‘Ultra-Quiet Platform’ (UQP) constructed by CSA Engineering for the Space Test Research Vehicle (STRV-2) (Anderson *et al*, 1996). The UQP has stiff electromagnetic actuators providing the force necessary for control with geophones used as vibration sensors. The UQP’s control system consists of a single input, single output loop closed around each strut. The UQP has a slightly lower control bandwidth than the previously described hard active mount systems of 3 Hz to 100 Hz and attenuation of 20 dB is achieved over this bandwidth.

5.1.2 Soft active mount systems

Honeywell have developed a hexapod system, the ‘Vibration Isolation and Suppression System’ (VISS), which flew on the STRV-2 (Davis *et al*, 1996). The VISS comprises a voice coil used as the soft actuator and accelerometers as sensors. The VISS control bandwidth is approximately 0.2 Hz to 40 Hz and it is able to achieve attenuation of greater than 20 dB over the control bandwidth. Also, because a voice coil is used as the actuator, the maximum possible displacement is more than an order of magnitude greater than any of the hard active mount systems.

Another soft active mount system is a hexapod built by the Jet Propulsion Laboratory (JPL) (Spanos *et al*, 1995). The JPL hexapod was developed primarily to improve the expected vibration environment on future space borne interferometers with a target control bandwidth of 1 Hz to 100 Hz. As with the VISS, the actuators selected were voice coils.

5.2 COMMERCIALY AVAILABLE ACTIVE ISOLATION SYSTEMS

Several active vibration isolation systems are available commercially, for example, Halcyonics produce a number of active vibration isolation tables that provide isolation in all six degrees of freedom. The Halcyonics system uses eight piezoelectric accelerometers that measure vertical and horizontal vibration with eight corresponding electrodynamic actuators. Halcyonics specify a vibration range beginning at 0.6 Hz, up to 40 dB at 10 Hz with correctional forces of 8 N and 4 N in the vertical and horizontal axis respectively. It is also possible to stack the active vibration isolation stages on top of each other to increase the isolation although this brings diminishing returns with each stack. Newport's Elite™ 3 Series Active Isolators provide a stable platform support, engineered specifically for high-resolution wafer production, metrology and microscopy applications. Piezoelectric actuators are designed into the isolator to support stiff work surfaces and actively cancel low frequency floor vibrations. The Elite 3's specification includes three degrees of freedom active vibration isolation, 10 dB attenuation at 0.5 Hz and both horizontal and vertical isolation. IDE's hybrid ETC series uses all metal springs for passive vibration isolation with a low resonant frequency - optimal isolation starts at 0.7 Hz and isolates in all six degrees of freedom. It offers 15 dB of isolation at 1 Hz and 30 dB at 10 Hz, (where passive isolation dominates). IDE's PTC series uses piezoelectric actuators that provide displacement of up to 30 µm. The PTC offers 15 dB at 1 Hz and 40 dB at 10 Hz; the system has a resonant frequency of 12 Hz, but active isolation from 0.1 Hz to 100 Hz. A summary of the performance of some active vibration isolation systems is given in Appendix B.

6 VIBRATION ISOLATION RESEARCH PROJECTS

The following section reviews various research projects that require a high degree of vibration isolation particularly at low frequencies. The techniques applied and performance of the vibration isolation systems will be discussed.

6.1 GRAVITATIONAL WAVE DETECTORS

Gravitational wave detectors require a displacement resolution of around 10^{-18} m and, therefore, seismic vibration isolation is key to the detector's performance. There are several

gravitational wave detectors undergoing construction around the world and these are summarised in Table 4. The most accurate gravitational wave detectors are large scale Michelson interferometers the arms of which range in size from 300 m to 5 km (Takahashi *et al*, 2002b).

Table 4 Interferometric gravitational wave detectors

Project	Countries	Baseline/km	Scheduled observation
LIGO	USA	4	Now online
VIRGO	France/Italy	3	2006
GEO	Germany/UK	0.6	Now online
TAMA	Japan	0.3	Now online

6.1.1 TAMA300

TAMA300, a 300 m Michelson interferometer in Japan, requires long-term dimensional stability and the ability to operate in an ultrahigh vacuum of 10^{-6} Pa. The target vibration level is a horizontal displacement noise of 5×10^{-20} m/ $\sqrt{\text{Hz}}$, however, the ground vibration level at Mitaka, Japan (the site of TAMA300) is 5×10^{-12} m/ $\sqrt{\text{Hz}}$. Therefore, isolation of more than one part in 10^8 is required. This was achieved with a double pendulum and multilayer stack system. Each mirror is hung in a suspension system with a double pendulum and uses eddy current damping that stands on a vibration isolated breadboard. The vibration isolation in the suspension system was measured at a part in 10^5 at the required frequency of 300 Hz. A further part in 10^3 of vibration isolation was supplied by a rubber and metal layered stack that supports the breadboard. A schematic of the TAMA300 system is shown in Figure 8.

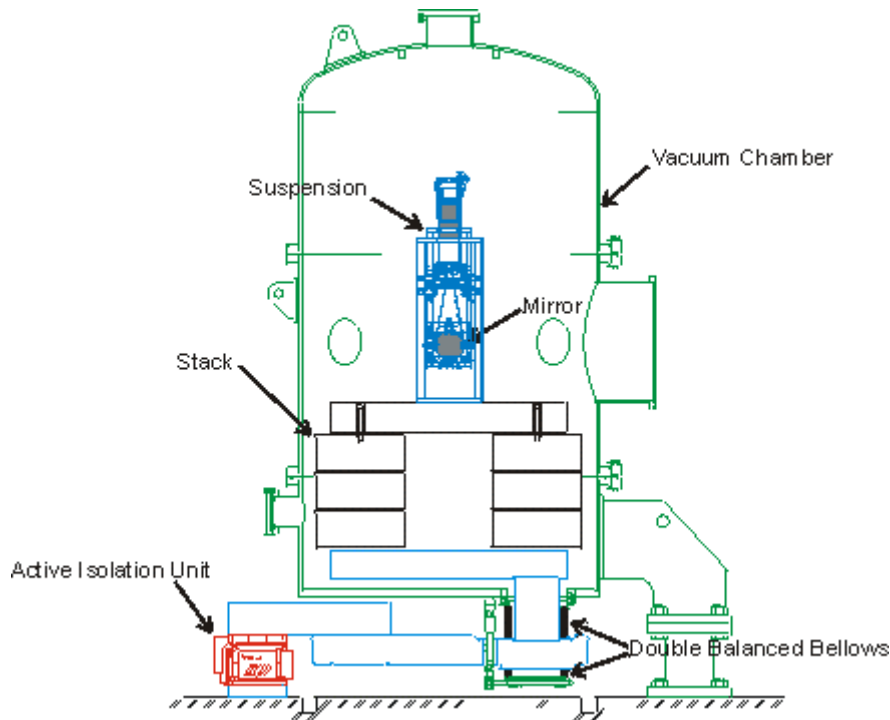


Figure 8 Cross section of the vibration isolation system of TAMA300 (courtesy of Takashi *et al*, 2002)

The spring constants and masses of the metal and rubber stacks on the TAMA300 were chosen so that the fundamental frequency was 3 Hz. This is just above the fundamental frequency of the double pendulum system at 2 Hz.

Long-term stability in dimension and angle must be maintained to extremely high precision for the interferometer to remain locked, as a result pneumatic actuators were introduced to compensate for the tendency to drift (Takahashi *et al*, 2002a). An active isolation system was installed below the baseplate outside of each vacuum chamber. Tokkyokiki Corporation's $\alpha 2$ product was used as the active vibration isolation system. The $\alpha 2$ consists of four compact units with two dimensional pneumatic actuators. The $\alpha 2$ has three control loops: an absolute vibration feedback loop from 1 Hz to 20 Hz, a ground vibration feed-forward loop from 0.5 Hz to 50 Hz and position feedback loop from zero to 0.1 Hz. The signals from the acceleration sensors on the suspended parts of the pneumatic actuators are processed and fed back to the pneumatic actuators to control all six degrees of freedom. The signals from acceleration sensors on the ground are fed forward to the pneumatic actuators for additional control of three degrees of freedom; x , y and z . The measured transfer functions showed a fundamental passive resonance at 9 Hz for x and y , and at 5 Hz for z . Filters in the digital signal processor suppressed these resonances. The measured isolation with active control was 10 dB to 20 dB at frequencies greater than 1 Hz for all six degrees of freedom. Before the $\alpha 2$

vibration isolation system was implemented it was tested extensively. A Michelson interferometer, built specifically to measure the vibration levels (Araya *et al*, 1993), confirmed vibration isolation over a part in 10^8 at 300 Hz.

6.1.2 LIGO

The Laser Interferometer Gravitational wave Observatory (LIGO) (Raab, 1995) consists of a pair of 4 km long Michelson interferometers at either end of the USA (see Figure 9).



Figure 9 Photograph of the LIGO gravity wave detector (courtesy of LIGO)

The original low frequency sensitivity of LIGO was limited to 10 Hz to 50 Hz by seismic noise. A team at NIST (Newell *et al*, 1997) became involved in designing an isolation system that would allow the LIGO to operate close to 1 Hz with vibration isolation of one part in 10^6 in the horizontal axis and greater than 10^6 in the vertical axis. As in the TAMA300 project, a three-stage isolation system was designed comprising both active and passive components including a double pendulum and stack system. However, there are differences from the TAMA300 system within the active system that supports the ultra-high vacuum chamber. An active vibration isolation system was developed capable of operating in all six degrees of freedom and consisted of three stages. The active isolation is provided by servo control loops, using signals from low frequency displacement sensors to feed back to non-contacting force transducers. These quasi-independent single input, single output feedback loops are based on

a passive system that has resonant frequencies below 2 Hz. The platform to be isolated is an equilateral triangle and relative motion is detected by a two-axis seismometer made specifically for the project. The horizontal and vertical resonant frequencies of the seismometer are 0.8 Hz and 1 Hz respectively.

Tests made on the first stage of the three stage system using two Teledyne S13 seismometers showed vertical vibration isolation of one part in 100 between 2 Hz and 20 Hz and horizontal vibration was also reduced by one part in 100. A further measurement was taken by injecting the system with white noise then monitoring the response on the platform with a seismometer. Again a vibration isolation of over one part in 100 was demonstrated above 0.5 Hz in horizontal motion and 1 Hz in vertical motion.

6.2 VIBRATION ISOLATION FOR LARGE AIRBORNE SYSTEMS

The United States Airforce airborne laser program requires a laser to be aligned and operational whilst the aircraft is airborne. Effects such as beam pointing accuracy and jitter are affected by vibration inside an aircraft fuselage. Passive isolation systems for optical benches on an aircraft must cope with floor vibrations up to three orders of magnitude larger than ground based systems and the level of very low frequency vibration due to the fuselage flexing and the aircraft manoeuvring. The proposed system by CSA Engineering (Kienholz, 2000) is to combine a pneumatic spring passive system with an active system capable of dealing with the very low frequency movements. Air springs provide vertical vibration attenuation down to around 2 Hz, horizontal restraint is provided by the flexural stiffness of the air spring itself or by pendulum action of the mechanism joining the air spring and payload. The active vibration isolation system required electromagnetic voice coils, as the actuators had to be capable of relatively long travel due to the large degree of pitch, roll and yaw occurring during aircraft flight. Also the system had to deal with a substantial payload, 500 kg to 1000 kg. As the disturbances from the aircraft moving are generally confined to below 10 Hz, active isolation is confined to below this frequency and above 10 Hz a passive isolation system attenuates the vibrations. This hybrid system of semi-passive pneumatic and mechanical isolation can compensate for displacements of the order of 30 mm.

6.3. “G-LIMIT” – SPACE STATION VIBRATION ISOLATION SYSTEM

The International Space Station (ISS) was envisaged as a unique state-of-the-art microgravity science laboratory. However, it was predicted that acceleration levels would not meet the required specification (Wharton, 2002), therefore, extra low frequency vibration isolation will be required in the payload/rack locations to compensate for the insufficiently quiescent micro gravity, due to the space station manoeuvring. At frequencies below 0.01 Hz the isolation system must transmit the very low frequency quasi-static accelerations with enough travel to prevent the isolated elements from bumping into the vehicle chassis, therefore, a large separation is required between the isolated platform and the cargo hold. At frequencies between 0.01 Hz and 10 Hz the amount of attenuation must increase one order of magnitude per decade increase in frequency, therefore, the isolation system must measure and cancel 10^{-6} g variations over periods of tens of seconds (where g is the local acceleration due to gravity). The g-LIMIT (Glovebox Integrated Microgravity Isolation Technology) (Wharton, 2001) is a micro gravity vibration isolation system built by NASA for science payloads on the ISS. The g-LIMIT has two main sub-systems: the inertially isolated assembly, which the experiment is attached to, and the base assembly, which is rigidly attached to the ISS rack support. Control forces are applied to the isolated assembly using non-contact Lorentz force actuators each with a peak force magnitude of approximately 3 N per axis. Accelerometers are used to measure the inertial motion of the isolated platform. The g-LIMIT is also a relative position sensor used to measure the relative motion between the isolated assembly and the base mount. There is a maximum displacement of 10 mm between the isolator assembly to allow ‘rattle free’ operation. G-LIMIT accomplishes inertial isolation using a high gain, high bandwidth acceleration feedback control system applying six independent forces to the isolation platform. A low frequency outer loop position feedback control system is used to maintain centering of the isolated platform.

7 SUMMARY

This report provides an overview of some vibration diagnostic and isolation techniques then reviews several seismic isolation techniques and systems that have been developed for diverse applications such as gravity wave detection and airborne lasers.

Low frequency vibrations (less than 10 Hz) are predominantly naturally occurring and isolation against these environmental effects normally requires the development of specialist equipment. Higher frequency isolation and diagnostic systems are more readily available commercially, this is primarily due to the relative ease at which high frequency vibration can be detected in comparison to low frequency vibration.

Passive vibration isolation in the form of pads or air springs supporting a table is often used as the first step in vibration isolation, active systems can then be used to extend the vibration attenuation range that is greater over a resonant frequency of the passive system.

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APPENDIX A HEXAPOD DESIGNS FOR ACTIVE VIBRATION ISOLATION

Hexapod	Actuator	Actuation Stroke /mm	Feedback Sensors	Passive Damping	Active Bandwidth Requirement /Hz
Intelligent Automation	Magnetorestrictive (HAM)	± 0.127	Load cell, accelerometer	None	10-200
Draper	Piezo-ceramic (HAM)	± 0.025	Load cell	None	10-200
Harris/AIF	Piezo-ceramic (HAM)	± 0.025	Base and payload accelerometer	None	10-200
CSA	Electro-magnetic (HAM)	± 0.020	Payload geophone	Elastomer	3-100
Honeywell	Voice-coil (SAM)	± 2	Payload accelerometer	Fluid	0.2-40
JPL	Voice-coil (SAM)	± 0.5	Load cell	Eddy current	1-100
JPL/UW	Voice-coil (SAM)	± 5	Load cell	Elastomer	1-100
Hood/UW	Voice-coil (SAM)	± 5	3-axis load cell, payload geophone	External suspension	0.3-30

APPENDIX B COMMERCIALLY AVAILABLE VIBRATION ISOLATION SYSTEMS

System	Isolation and frequency range	Comments	Contact (at the time of writing)
Halcyonics	>0.6 Hz, 40 dB attenuation at 10 Hz	Active system	www.halcyonics.com
Minus-K	>0.7 Hz 23 dB attenuation at 10 Hz	Passive system	www.minusk.com
Newport elite 3	>0.5 Hz 30-35 dB attenuation at 10 Hz	Active system better in vertical rather than horizontal isolation	www.csaengineering.com
JRS mod 2	>0.6 Hz, 40 dB attenuation at 10 Hz	Active system	www.jrs-si.ch
IDE ETC	>0.7 Hz 30 dB attenuation at 10 Hz	Active and passive system	http://www.ideworld.com/index1.html
IDE PTC	>0.1 Hz 40 dB attenuation at 10 Hz	Active and passive system	