Low Frequency Vibration Isolation Technology for Microgravity Space Experiments

Carlos M. Grodsinsky and Gerald V. Brown
Lewis Research Center
Cleveland, Ohio

Prepared for the
12th Biennial Conference on Mechanical Vibration and Noise
sponsored by the American Society of Mechanical Engineers
Montreal, Canada, September 17–20, 1989
LOW FREQUENCY VIBRATION ISOLATION TECHNOLOGY FOR MICROGRAVITY SPACE EXPERIMENTS

Carlos M. Grodinsky and Gerald V. Brown
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135

SUMMARY

The dynamic acceleration environment observed on Space Shuttle flights to date and predicted for the Space Station has complicated the analysis of prior microgravity experiments and prompted concern for the viability of proposed space experiments requiring long-term, low-g environments. Isolation systems capable of providing significant improvements in this environment exist, but have not been demonstrated in flight configurations. This paper presents a summary of the theoretical evaluation for two one degree-of-freedom (DOF) active magnetic isolators and their predicted response to both direct and base excitations, that can be used to isolate acceleration sensitive microgravity space experiments.

NOMENCLATURE

$C$ capacitance
$c$ electro-magnet damping coefficient
$F_s$ isolator force
$g_0$ acceleration of the Earth at the surface
$I_{av}$ electro-magnet current (sensor bias)
$I_b$ magnetic-circuit current bias
$K$ passive stiffness coefficient
$k_a$ magnetic-circuit current amplifier stiffness
$k_{eq}$ magnetic-circuit Isolator stiffness
$k_g$ magnetic-circuit proportional gain
$k_i$ magnetic-circuit current stiffness
$k_p$ magnetic-circuit sensor amplifier gain
$k_r$ magnetic-circuit derivative gain
$k_\theta$ magnetic-circuit position stiffness
$m$ mass
$N$ number of ampere turns
$R$ resistance
$u$ position of base
$x$ position of payload
$\xi$ passive damping coefficient
$\tau$ time constant = RC
$\psi$ magnetic field strength
$\omega$ excitation frequency
$\omega_{ha}$ active system resonance frequency
$\omega_h$ system resonance frequency

INTRODUCTION

Interest in vibration isolation for microgravity experiments has increased within the microgravity science community as the flight program has progressed and the small, but significant levels of residual acceleration on the Space Shuttle (STS) have become more widely recognized and documented (Hamacher, 1986: Workshop Proceedings, 1986). These background accelerations result from several sources characteristic of the orbiting carrier and the orbital environment. Very low frequency (10^{-3} Hz to dc) accelerations due to drag, tidal effects, and gravity gradients contribute submicro-g levels. STS thruster activity can contribute 10^{-2} g/\theta0 accelerations with significant duration, but can be predicted and controlled. The most visible and troublesome contribution to most experiments is the moderate frequency (10^{-3} to 100 Hz) dynamic spectrum of accelerations having magnitudes in the range 10^{-5} to 10^{-2} g/\theta0. This dynamic background is due substantially...
to random excitations from manned activity on the orbiter. However, orbiter structure and flight systems also contribute observable intermittent and resonant accelerations to the background as the orbiter interacts with its dynamic mechanical and thermal environment. The evolution of the Space Station design has led to many discussions of the potential limitations on long term, low gravity experimentation in this environment. It is now obvious that most of the true microgravity experiments will require "protection" from this random, milli-g environment if valid and reproducible results are to be expected. Because of the presence of the transient disturbances have a frequency range from milli-Hz to 1 Hz. It is extremely difficult to design a passive isolation system with an effective frequency at which such a disturbance would act, in a random, milli-g environment if valid and reproducible experiments. On the other hand, when there are direct disturbances on the payload, a small value of stiffness is not desirable. Therefore, there is a trade off, and an optimal design would need to compensate for both direct excitations, but limited damping leads to potentially unstable systems in the random excitation environment. It is apparent that a passive isolation system would not suffice because of the requirement of an extremely low stiffness for the isolation of small disturbance frequencies. The second category is magnetic damping. The fully magnetically suspended system is evaluated using an attractive electromagnet, while the electromagnetically damped system is evaluated using a Lorentz magnet. These magnetic systems, specifically the attractive type, have been used for the suspension of rotating shafts for a number of years and the required negative feedback loops to control such systems have been discussed in numerous papers, giving the equivalent stiffness and damping coefficients for specific controllers. However, these studies have not been interested in the isolation of the suspended body from direct and base excitations. Thus, the response of such systems to these type of disturbances has not been documented. In this paper we evaluate the dynamic response to base and direct disturbances of both systems. A pictorial representation of both systems evaluated is shown in Fig. 1 where both systems can be represented by an isolator between a base support and the isolated payload. This isolator is simply an actuator which is driven in proportion to certain feedback signals depending on the definition of the load. Specifically, for the magnetic-circuit isolator, an attractive magnetic actuator was analyzed where both the stiffness and damping coefficients are derived from a relative position sensor and for the electromagnetic damping isolator a Lorentz actuator was analyzed where the damping coefficient was derived from an inertial sensor and the stiffness of the isolator is simply a constant spring stiffness of a passive spring. Presently, there are laboratory models of these theoretically represented one DOF isolation systems. However, the emphasis at NASA Lewis is to perform digital active control on dependent multidegrees of freedom. Currently, these systems are projected for test in a full six DOF free fall condition, provided by the NASA Lewis low gravity aircraft, in order to acquire the coupled response between all six DOF in a low gravity environment.

STANDARD OF THE PROBLEM

To categorize the disturbances which are present in the Space Shuttle and will be present in the Space Station, one can group these accelerations into three frequency ranges (Jones et al., 1987):

1. Quasi-Static External Disturbances
2. Low Frequency Vibration Sources
3. Medium-High Frequency Vibrations

The disturbances listed under the first category are aerodynamic drag, gravity gradient effects and photon pressure accelerations. These accelerations have frequency ranges less than a mHz and acceleration magnitudes about 10⁻⁶g. The second category would include excitations due to large flexible space structures, crew motion, spacecraft attitude control, and robotic arms. These disturbances range from mHz to about the 10 Hz frequency range. The third category would list disturbances due to on-board equipment such as pumps and motors having a dynamic range of about 10 Hz and above. (See appendix (Hamacher, 1986; Ruzicka, 1969; Hamacher et al., 1986).)

OVERVIEW

The active isolators described in this paper are effective at a frequency range of about a tenth of a Hz and above. This constraint does not arise from a technology limitation, but from practical limitations due to the fact that the strokes needed to isolate against the very low frequency range are not obtainable because of the volume constraints in the shuttle and in the future Space Station manned environment laboratory modules. For example, aerodynamic drag is a function of the atmospheric conditions during a specific mission, but an average magnitude of 10⁻⁶g will be used for the sake of argument. Therefore, the frequency at which such a disturbance would act, in a
solar pointing station, would be at the orbital frequency, which is about 90 min per orbit. Thus, the distance an object would travel under such an acceleration would be: 

\[ \frac{2\pi}{f} \cdot \text{height} = \frac{2\pi}{90} \cdot 27000 \approx 18 \text{ km} \]

Including initial conditions. Thus, realistically, an isolated payload would have to follow such movement of the spacecraft and be active in a much smaller region, which would depend on volume constraints of a payload in the shuttle or Space Station microgravity modules.

The following two cases can be analyzed as spring-mass damper systems, where the spring and damper characteristics are actively controlled and translated into actuator response by a control law depending on the response characteristics desired. Using an attractive electromagnet actuator, one can produce forces in only one direction. Therefore, to achieve a push-pull configuration one needs to use two electromagnets acting on an armature. For these electromagnet actuators, the force produced by one magnet is proportional to the square of the current and inversely proportional to the square of the gap, making the system open loop unstable. Due to these nonlinear characteristics, a bias current linearization technique is utilized. In addition, nonlinearities also arise between magnetic flux and input coil current due to hysteresis and saturation. The force produced by a Lorentz actuator is a vector quantity equal to the current cross field, \( F = \frac{1}{2} \mu_0 n^2 A \beta \). To summarize the linearized control law for a one DOF magnetic-circuit isolator and the linear control law for a Lorentz electromagnetically damped one DOF system, one can give their transmissibilities and effectiveness in isolating against both base and direct disturbances. First, the response or transmissibilities of both systems will be generated for harmonic base excitations, using the active isolation system's differential equations of motion. These equations of motion were written using Newton's first and second laws. Where \( u \) is actually a time function so \( u = u(t) \) with the same implied for a directly applied force, such that in actuality \( F = F(t) \). Therefore, for a spring mass damper system, the equations of motion for base excitation become:

**Magnetic-circuit Isolator**

\[
-m \frac{d^2x}{dt^2} + k_eq(x - u) + c_eq \frac{dx}{dt} - Ku = 0
\]  

**Electromagnetic damping Isolator**

\[
-m \frac{d^2x}{dt^2} + c_eq \frac{dx}{dt} + Kx = Ku
\]

These systems look very similar to passive viscoelastic systems with the exception that, for all practical purposes, the stiffness and damping of both the magnetically suspended isolator and the electromagnetic damping case can be set at whatever coefficient is desired for an appropriate response to an excitation source. Therefore, these systems can be easily configured for an adaptive system where, by using sensed information from the disturbance environment, the control law could be changed to optimize the isolation of the payload. In using the magnetic-circuit actuator as an isolator, the stiffness and damping are not purely independent. However, this dependency is minimal and if certain control parameters are not violated, these isolation parameters can be assumed to be independent. To achieve a purely damped response independent of stiffness, be it active or passive stiffness, one would need to use a Lorentz actuator. In contrast, for the magnetic-circuit case, a certain amount of damping is needed in order to overcome open loop instability.

In defining the dynamic base motion equations for both systems, the stiffness and damping terms can be solved by using the appropriate control law needed for a stable negative feedback system. In summary, the stiffness coefficient for the magnetic-circuit becomes:
Thus, the transfer functions in terms of frequency response are vectors in the complex plane and the magnitudes of vibrations measured on the isolated payload resulting from a sinusoidal excitation \( \sin(\omega t) \) is the vector length of \( X/|U(j\omega)| \). This value is a scalar, since the phase angle is not used, and is called the transmissibility function of the system. The transmissibility is generally written as

\[ T = |X/|U(j\omega)|| \]

Therefore, the transmissibility functions for both systems of interest become:

**Magnetic-circuit Transmissibility Function**

\[ T = \frac{X}{U(j\omega)} = \sqrt{\frac{1 + \frac{2\xi \omega_n^2}{\omega_n^2}}{\left(1 - \frac{\omega_n^2}{\omega_n^2}\right) + \frac{2\xi \omega_n^2}{\omega_n^2}}} \]

**Electromagnetic Damping Transmissibility Function**

\[ T_{av} = \frac{X}{U(j\omega)} = \sqrt{\frac{1}{\left(1 - \frac{\omega_n^2}{\omega_n^2}\right) + \frac{2\xi \omega_n^2}{\omega_n^2}}} \]

By plotting these transmissibilities, one can see the effect of changing the stiffness or damping of either system. As can be seen from the stiffness coefficient for the magnetic suspension case, using the attractive magnetic-circuit actuator, the natural frequency of the system is frequency dependent on the excitation source. However, for small disturbance frequencies of 100 Hz and below, this dependence is negligible and \( \omega_n \) is assumed constant. The transmissibility curve for the first case is illustrated in Fig. 3. Figure 3 demonstrates the effect of increasing the damping coefficient of the magnetic-circuit isolator system. As can be seen by the curves in Fig. 3, increasing the velocity feedback gain, \( K_r \), the system can become overly damped, which gives rise to the damped response at resonance and less isolation at excitation frequencies above \( \omega_n \) times \( \omega_n \) than would be achievable with a less damped system. The effect of increasing the position gain would shift the natural frequency of the system to the right because of the increase in equivalent stiffness of the system, while the opposite would result by decreasing the position gain \( K_p \). The subsequent electromagnetic damping case is illustrated in Fig. 4. Figure 4 shows the effect of increasing the damping coefficient of the Lorentz electromagnetic damping system. The curves show the response of the system to increasing the velocity feedback term and thus, increasing the damped response of the system. The advantage of active damping feedback, derived from an inertial reference, is that it can remove the resonant response, broadening and smoothing the transition between the low frequency and high frequency regions, while reducing both the transmission and the response, particularly in the low frequency range of interest. The effects of such a system for large values of velocity feedback gain can be understood by considering that it is equivalent to having a passive damper attached between the isolated mass and a virtual inertial reference. As the friction in the damper is increased, the isolated mass becomes more and more tightly coupled to the (motionless) ideal inertial reference. Unlike the passive

\[ k_{eq} = k_b + \frac{k_i k_d k_p \left[ k_f \left(1 - R \tau \omega^2\right) + \left(k_g + k_r\right)(R + \tau R) \right]}{\left(1 - R \tau \omega^2\right)^2 + \left(R + \tau R\right)^2 \omega^2} \]

and for the electromagnetic isolator, because the mass is being statically supported by a passive spring, the stiffness is simply \( K \). Summarizing the damping coefficients for both isolators, the magnetic-circuit damping coefficient becomes:

\[ c = -\omega_n N_{avv} \text{ where: } I_{avv} = \frac{E_{avv}}{R} \]
damper, the stronger the coupling, the better the isolation. This arises because the velocity proportional gain is determined from the integration of an inertial sensor signal. This type of response is not seen in the pure suspension case because the velocity term was determined from the derivative of a relative position sensor giving rise to the response shown in Fig. 3.

\[
\xi = \frac{K_g K_p (K_g + K_F)}{(1 - RC \omega^2)^2 + (RC + \tau_1)^2 \omega^2}
\]

\[
u_n^2 = K_0 + \frac{K_i K_p (K_g (1 - RC \omega^2) + (K_g + K_F + K_p RC + \tau_1) RC \omega^2)}{(1 - RC \omega^2)^2 + (RC + \tau_1)^2 \omega^2}
\]

As explained previously, these curves all demonstrate system response to base excited harmonic motions. However, disturbances may also be generated directly on the payload itself. The sensitivity of the isolated payload to a disturbing force will be characterized by a term called the isolated payload mobility. The mobility of the payload is the vector magnitude of \(X(s)/F(s)\). This parameter measures the amplitude of the payload deflection per unit of force amplitude. The equations of motion for both systems, for direct disturbance only are:

Magnetic-circuit Equation of Motion

\[
M \frac{d^2 x}{dt^2} = F(t) - K_{eq} \frac{dx}{dt} - C_{eq} x
\]
In order to evaluate the effectiveness of these active systems, the ratio of $X(s)/F(s)$ active to $X(s)/F(s)$ for an equivalent passive system will be used. This ratio will be called the mobility effectiveness $X_f(s)$. Therefore, if $X_f(s)$ is unity, the active system does nothing. If $X_f(s)$ is zero, no motion of the payload results from a finite applied force. If $X_f(s)$ is greater than unity, then the active system amplifies the effect of the applied force, increasing the payload motion. The equations for the effectiveness function for both cases, in terms of frequency response, where the vector length of $X_f(s)$ is $|X_f(j\omega)|$, become:

**Magnetic Circuit Effectiveness**

$$X_f(j\omega) = \frac{\omega_n}{\omega_{na}} \sqrt{\frac{1 - \left(\frac{\omega}{\omega_n}\right)^2}{1 - \left(\frac{\omega}{\omega_{na}}\right)^2} + \frac{2\xi_1}{\omega_n} \frac{\omega}{\omega_{na}}}$$  \hspace{1cm} (16)

Where: for small excitation frequencies $\omega_{na} = \omega_n$; $\xi_1$ = damping coefficient of passive spring ($A$ value of 0.05 was used for $\xi_1$.)

**Electromagnetic Damping Effectiveness**

Active: $\omega_n = \sqrt{\frac{c_{eq}}{M} + \frac{K}{M} = 2\xi\omega_n}$

$$X_f(j\omega) = \sqrt{\frac{1 - \left(\frac{\omega}{\omega_n}\right)^2}{1 - \left(\frac{\omega}{\omega_{na}}\right)^2} + \frac{2\xi_1}{\omega_n} \frac{\omega}{\omega_{na}}}$$  \hspace{1cm} (17)

Where:

$$\xi = \frac{2\xi_1}{\omega_n} \cdot \omega_n = \sqrt{\frac{c}{M} \cdot \xi = \frac{1}{2} G_v \sqrt{\frac{1}{KM}}}$$

In order to evaluate the effectiveness of these active systems, the ratio of $X(s)/F(s)$ active to $X(s)/F(s)$ for an equivalent passive system will be used. This ratio will be called the mobility effectiveness $X_f(s)$. Therefore, if $X_f(s)$ is unity, the active system does nothing. If $X_f(s)$ is zero, no motion of the payload results from a finite applied force. If $X_f(s)$ is greater than unity, then the active system amplifies the effect of the applied force, increasing the payload motion. The equations for the effectiveness function for both cases, in terms of frequency response, where the vector length of $X_f(s)$ is $|X_f(j\omega)|$, become:

**Magnetic Circuit Effectiveness**

$$X_f(j\omega) = \frac{\omega_n}{\omega_{na}} \sqrt{\frac{1 - \left(\frac{\omega}{\omega_n}\right)^2}{1 - \left(\frac{\omega}{\omega_{na}}\right)^2} + \frac{2\xi_1}{\omega_n} \frac{\omega}{\omega_{na}}}$$  \hspace{1cm} (16)

Where: for small excitation frequencies $\omega_{na} = \omega_n$; $\xi_1$ = damping coefficient of passive spring ($A$ value of 0.05 was used for $\xi_1$.)

**Electromagnetic Damping Effectiveness**

Active: $\omega_n = \sqrt{\frac{c_{eq}}{M} + \frac{K}{M} = 2\xi\omega_n}$

$$X_f(j\omega) = \sqrt{\frac{1 - \left(\frac{\omega}{\omega_n}\right)^2}{1 - \left(\frac{\omega}{\omega_{na}}\right)^2} + \frac{2\xi_1}{\omega_n} \frac{\omega}{\omega_{na}}}$$  \hspace{1cm} (17)

Where:

$$\xi = \frac{2\xi_1}{\omega_n} \cdot \omega_n = \sqrt{\frac{c}{M} \cdot \xi = \frac{1}{2} G_v \sqrt{\frac{1}{KM}}}$$
$\xi_1$ = damping coefficient of passive spring (A value of 0.05 was used for $\xi_1$.)

These effectiveness functions are plotted in Figs. 6 and 7. Figures 6 and 7 present the effectiveness of the active feedback, force actuated vibration isolation systems as compared to a passive system with a critical damping coefficient of 0.05, which is typical of passive systems of the type utilized with low frequency system resonances.

CONCLUDING REMARKS

In conclusion, it is apparent that the active magnetic systems described here have advantages over passive isolators due to their ability to isolate against the low frequency regime of the orbital carriers, as well as their ability to implement an adaptive control to isolate against both the direct and base excitations which will be present in all pressurized modules. Therefore, the optimal isolation of a microgravity science payload will eventually need an adaptive digitally controlled system in order to optimize isolation coefficients to most effectively prevent disturbances from perturbing the isolated payload. In order to lower the corner frequencies of such an active system one would need to use actuators with larger and larger strokes. However, this would be impossible due to the volume constraints present in space flight vehicles. Thus, such an isolated payload will have to follow these very low steady-state accelerations such as aerodynamic drag and gravity gradient effects. In order to achieve the microgravity requirements imposed on the Space Station facility (Fig. 8), for any significant length of time, microgravity vibration isolation will have to become a systems engineered solution as well as an experiment specific concern. Thus, these requirements for acceleration sensitive microgravity space experiments will dictate multi-stage isolation concepts which will combine both passive and active systems where the control of the center of gravity of the Space Station will be closed around microgravity steady-state accelerations.

In order to lower active system one
larger and larger
impossible due to
the volume constraints present
space flight vehicles. Thus, such an isolated payload will have to follow these very low steady-state accelerations such as aerodynamic drag and gravity gradient effects. In order to achieve the microgravity requirements imposed on the Space Station facility (Fig. 8), for any significant length of time, microgravity vibration isolation will have to become a systems engineered solution as well as an experiment specific concern. Thus, these requirements for acceleration sensitive microgravity space experiments will dictate multi-stage isolation concepts which will combine both passive and active systems where the control of the center of gravity of the Space Station will be closed around microgravity steady-state accelerations.

![Figure 6](image-url)
REFERENCES


APPENDIX

The range of accelerations which have been observed on several IST missions or estimated for the accessible orbit are summarized below (Hamacher, 1986; Ruzicka, 1969; Hamacher, et al., 1986).

Quasi-Steady or "DC" Acceleration Disturbances

<table>
<thead>
<tr>
<th>$g/g_0$</th>
<th>Frequency, $Hz$</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>10^-7</td>
<td>0 to 10^-3</td>
<td>Aerodynamic drag</td>
</tr>
<tr>
<td>10^-8</td>
<td>0 to 10^-3</td>
<td>Light pressure</td>
</tr>
<tr>
<td>10^-7</td>
<td>0 to 10^-3</td>
<td>Gravity gradient</td>
</tr>
</tbody>
</table>

Periodic Acceleration Disturbances

<table>
<thead>
<tr>
<th>$g/g_0$</th>
<th>Frequency, $Hz$</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>2x10^-2</td>
<td>9</td>
<td>Thruster fire (orbital)</td>
</tr>
<tr>
<td>2x10^-3</td>
<td>5 to 20</td>
<td>Crew motion</td>
</tr>
<tr>
<td>2x10^-4</td>
<td>17</td>
<td>Ku band antenna</td>
</tr>
</tbody>
</table>

Nonperiodic Acceleration Disturbances

<table>
<thead>
<tr>
<th>$g/g_0$</th>
<th>Frequency, $Hz$</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>10^-4</td>
<td>1</td>
<td>Thruster fire (attitude)</td>
</tr>
<tr>
<td>10^-4</td>
<td>1</td>
<td>Crew push off</td>
</tr>
</tbody>
</table>

ORIGINAL PAGE IS OF POOR QUALITY
The dynamic acceleration environment observed on Space Shuttle flights to date and predicted for the Space Station has complicated the analysis of prior microgravity experiments and prompted concern for the viability of proposed space experiments requiring long-term, low-g environments. Isolation systems capable of providing significant improvements in this environment exist, but have not been demonstrated in flight configurations. This paper presents a summary of the theoretical evaluation for two one degree-of-freedom (DOF) active magnetic isolators and their predicted response to both direct and base excitations, that can be used to isolate acceleration sensitive microgravity space experiments.