Journal of Sound and Vibration (2000) **238**(1), 19–30
doi:10.1006/jsvi.2000.3098, available online at http://www.idealibrary.com on IDEAL®

VIBRATION PROTECTION OF SENSITIVE ELECTRONIC EQUIPMENT FROM HARSH HARMONIC VIBRATION

A. M. VEPRIK AND V. I. BABITSKY

Department of Mechanical Engineering, Loughborough University, Leicestershire, LE11 3TU, England.
E-mail: *a.veprik@lboro.ac.uk*

(*Received* 1 *September* 1999, *and in final form* 17 *April* 2000)

Vibration protection of sensitive electronic equipment which operates in harsh environments often relies on resilient mounts. The traditional optimal design of vibration isolation from harmonic vibration is based on compromising damping and stiffness properties of mounts and is aimed, in general, at widening the frequency range over which the attenuation takes place, subject to limitations imposed on the rattlespace of the electronic box. Such a design typically incurs the use of heavily damped vibration isolators. Nevertheless, the reliability of the electronic instrumentation depends not only on the level of vibration experienced by the electronic box, but also, and primarily, on the vibration responses of the internal components that are often lightly damped and extremely responsive over a wide frequency range. The traditional approach, however, completely ignores the presence of such components. The heavily damped vibration isolators result in poor vibration isolation over the high-frequency span which typically contains resonant frequencies for critical internal components and, therefore, are insufficient for maintaining a fail-safe vibration environment for electronic equipment. The proposed design approach focuses primarily on the dynamic properties and responses of the critical internal components of an electronic device. In this instance, the heavy and rigid electronic box is thought of and utilised as the *first-level vibration isolation stage* (mechanical low-pass filter) relative to the sensitive internal components. The optimally chosen elastic and damping properties of the vibration isolators minimise the vibration experienced by the critical internal components, subject to restraints imposed on the peak deflections of the electronic box. The optimisation procedure relies on an analytical solution. The results of calculations are proven experimentally.

1. INTRODUCTION

Electronic devices which provide for control, guidance, reconnaissance and communication are vital components of modern aerospace, marine and vehicular applications. The sophisticated and highly reliable electronic packages must be able to operate over life cycles greater than 20 years, and to survive harsh environmental vibration.

Recent improvements in the quality and durability of low-cost commercial electronic equipment such as printed circuit boards (PCB), allows commercial off-the-shelf (COTS) electronics to be used in harsh environmental conditions.

PCBs which are mounted inside relatively rigid and heavy electronic boxes are the most susceptible to servere environmental disturbances. As a result of the need for greater ease of maintenance and installation, plug-in modules for PCB mounting are often used. Each board is connected to an electrical interface and held at the sides by edge guides, allowing for convenient maintenance and replacement. Since the PCBs are plate-wise lightly damped structures, external vibration transmitted directly from the rigidly mounted sub-chassis readily excites their resonant modes. Excessive deformations and accelerations of the PCBs results in damage to mounted components, soldered joints and electrical interfaces, as well as to the circuit board itself.

The accepted methods to make the existing equipment more rugged involve a combination of stiffening and damping elements [1, 2]. These methods aim to either reduce the resonant amplitudes by damping, or to avoid them altogether by ensuring that the relevant resonant frequencies are above the excitation frequency.

A typical solution was offered recently by the Thompson-CSF, CETIA as a standard optional accessory to its commercial-grade PCBs.† The Ruggedizer is a board level heat sink designed to add mechanical stiffening and protection. As the Ruggedizer covers the entire top face of the board, the vibration and shock characteristics of the latter are greatly improved. The "sandwich" structure of the Ruggedizer-equipped PCB reduces the effects of harsh shock and vibration by eliminating the most dangerous low-frequency resonance and thus reduces the risks of physical-failure fatigue.

However, an application of such solutions incurs high costs, long re-design time and makes maintenance work more difficult since the modified systems prevent use of COTS and standard components.

Recently, attempts have been made to use unmodified COTS equipment in combination with vibration protection of highly critical electronic boxes [3, 4].

The simplest form of vibration protection is vibration isolation. Low frequency and lightly damped vibration isolators provide the best attenuation of harmonic vibration over the widest possible frequency range. However, the use of such vibration isolators exposes an electronic box to the risk of excessive resonant deflections. The demands on the limited installation space, dynamic safety, and integrity of resilient mountings and flexible interfaces, necessitate close control of the peak deflections of the vibration isolated electronic device under harsh environmental conditions.

Traditionally, this is achieved by means of vibration isolators with essential stiffness and damping which attenuate harmonic vibration over the widest possible frequency range, and which keep the resonant deflections of the electronic box within safe limits [4–9].

However, the above approach completely ignores the presence of lightweight internal sensitive components (PCB in this instance) that are often undamped and extremely responsive over a wide frequency range.

The heavily damped vibration isolators (as used in traditional designs) have poor vibration isolation properties at high frequencies which typically contain the resonant frequencies for sensitive internal components. Such isolators might be insufficient, therefore, for maintaining a fail-safe vibration environment for electronic equipment.

This paper introduces a new design concept for vibration protection of electronic equipment which contains highly responsive internal components. In contrast to the traditional approach to vibration isolator design, the new design approach focuses primarily on the dynamic properties and responses of the sensitive internal elements. In this instance, the heavy‡ external electronic box is thought of and used as the *first-level vibration isolation stage* (mechanical low-pass filter) relative to the internal sensitive component.

The optimally chosen elastic and damping properties of the vibration isolators minimise the vibration transmitted to the critical internal components, subject to restraints imposed on the peak deflections of the entire electronic device.

†See http://www.cetia.com/product/whitepapers/prod_bds_whitepap_ruggedizer_000_0e.pdf .
‡As compared with the weight of sensitive internal components.

2. OPTIMAL DESIGN OF VIBRATION ISOLATOR

2.1. TRADITIONAL APPROACH

Figure 1 shows schematically a model of the electronic box which is mounted resiliently over the vibrating base and contains a critical, lightweight component (PCB in this instance). The electronic box is modelled as a primary s.d.o.f. system, where Ω_1, ζ_1 denote the natural frequency and loss factor. The PCB in a single-mode approximation [1] is modelled as a secondary s.d.o.f. system with a natural frequency and loss factor denoted as Ω_2, ζ_2. In Figure 1, $Y(t)$ is the motion of the base; X_1 and X_2 are the absolute deflections of the primary and secondary systems; $Z_1 = X_1 - Y$, $Z_2 = X_2 - X_1$ denote the relative deflections of the primary system to the base and of the secondary system to the primary system.

The traditional design of an isolator of harmonic vibration [5–9] is aimed at widening the frequency range of vibration isolation and maintenance of the resonant deflections of the protected device relative to the base within the pre-designed limits $\pm \Delta$. Since the attenuation starts from the frequency $\sqrt{2}\Omega_1$ (independent of the damping represented in the system), decreasing the natural frequency of the vibration isolator leads to a widening of the frequency range over which attenuation takes place. Therefore, in the traditional approach, the problem of optimal design takes the form:

$$\Omega_1 \to \min; \; |Z_1| \leqslant \Delta. \tag{1}$$

Critically, the mass of the sensitive internal element (PCB) is typically very small in comparison with the mass of the entire device. Therefore, the dynamic response of the internally mounted PCB does not affect the motion of the electronic box [7].

The equation of relative motion of the primary system takes the form:

$$\ddot{Z}_1 + 2\Omega_1\zeta_1\dot{Z}_1 + \Omega_1^2 Z_1 = -\ddot{Y}. \tag{2}$$

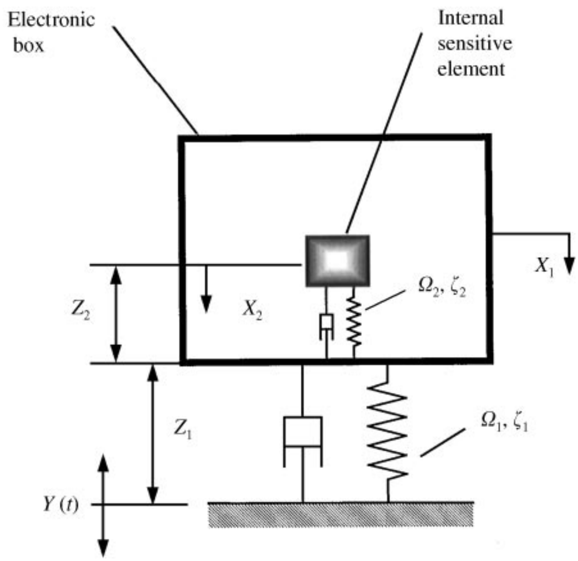

Figure 1. Dynamic model of vibration-isolated electronic device containing PCB.

For sinusoidal acceleration $\ddot{Y}(t) = \ddot{Y}_0 \sin \omega t$, where $\ddot{Y}_0$ is constant magnitude and ω is angular frequency, from equation (1), the magnitude value of relative deflection is:

$$|Z_1(\omega)| = \frac{\ddot{Y}_0}{\sqrt{(\Omega_1^2 - \omega^2)^2 + 4\omega^2 \Omega_1^2 \zeta_1^2}}. \tag{3}$$

From [7], for $\zeta_1 \leqslant 1/\sqrt{2}$, the maximum value of relative deflection which takes place at frequency

$$\omega_* = \Omega_1 \sqrt{1 - 2\zeta_1^2} \tag{4}$$

and equals

$$|Z_1(\omega)|_{max} = |Z_1(\omega_*)| = \frac{\ddot{Y}_0}{2\Omega_1^2 \zeta_1 \sqrt{1 - \zeta_1^2}}. \tag{5}$$

An application of restraint (1) to expression (5) yields the required value of the natural frequency:

$$\Omega_1 \geqslant \sqrt{\frac{\ddot{Y}_0}{2\Delta \zeta_1 \sqrt{1 - \zeta_1^2}}}, \tag{6}$$

which is minimal at the marginal value of the loss factor $\zeta_1 = 1/\sqrt{2}$. This value of the loss factor gives vibration attenuation over the widest possible frequency range and is recommended as the optimal value for different applications [8].

In a typical example, the base is subjected to harmonic vibration in the frequency range 20–500 Hz with a constant magnitude of acceleration, $\ddot{Y}_0 = 20\,\boldsymbol{g}$ (space-born specification in accordance with [10]). It is also assumed that the safe peak deflection of the electronic box relative to the base is $\Delta = 1$ mm.

Substitution of these values and $\zeta_1 = 1/\sqrt{2}$ into equation (6) yields $\Omega_1/2\pi \geqslant 70{\cdot}6$ Hz.

As previously mentioned, the reliability of the electronic device depends not only on the level of vibration experienced by the electronix box, but also, and primarily, on the vibration response of the *internal components*. To estimate this, the motion of the electronic box is now considered to be a vibration input to the internally mounted PCB.

Taking into account the low value of the loss factor ζ_2, the resonant amplitude of the absolute acceleration and relative deflection of the PCB is found to be

$$|\ddot{X}_2(\omega)|_{max} = |\ddot{X}_2\,(\Omega_2)| = |T_1(\Omega_2)|\,|T_2(\Omega_2)|\,\ddot{Y}_0, \tag{7}$$

$$|Z_2(\omega)|_{max} = |Z_2\,(\Omega_2)| = \frac{1}{\Omega_2^2}\,|T_1(\Omega_2)|\,|T_2^{rel}(\Omega_2)|\,\ddot{Y}_0, \tag{8}$$

where the magnitudes of absolute transmissibilities of the primary and secondary systems and relative transmissibility of the secondary system might be in the form

$$|T_1\,(\omega)| = \sqrt{\frac{\Omega_1^4 + 4\omega^2\Omega_1^2\zeta_1^2}{(\Omega_1^2 - \omega^2)^2 + 4\omega^2\Omega_1^2\zeta_1^2}}, \qquad |T_2(\omega)| = \sqrt{\frac{\Omega_2^4 + 4\omega^2\Omega_2^2\zeta_2^2}{(\Omega_2^2 - \omega^2)^2 + 4\omega^2\Omega_2^2\zeta_2^2}},$$

$$|T_2^{rel}(\omega)| = \frac{\omega^2}{\sqrt{(\Omega_2^2 - \omega^2)^2 + 4\omega^2\Omega_2^2\zeta_2^2}}. \tag{9}$$

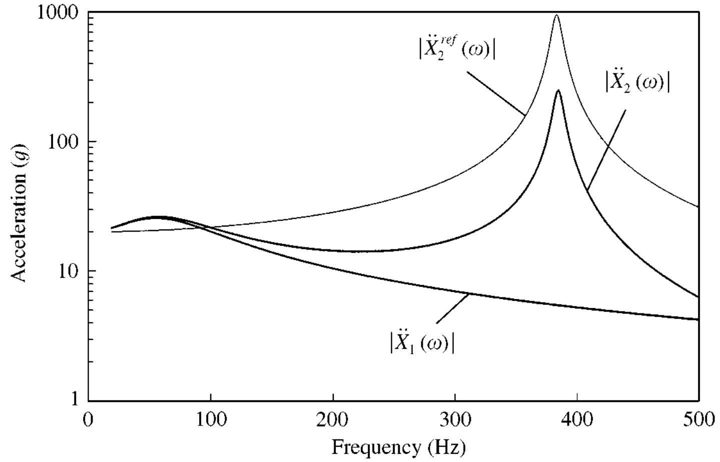

Figure 2. Absolute accelerations $|\ddot{X}_1(\omega)|$, $|\ddot{X}_2(\omega)|$, $|\ddot{X}_2^{ref}(\omega)|$ in traditional design.

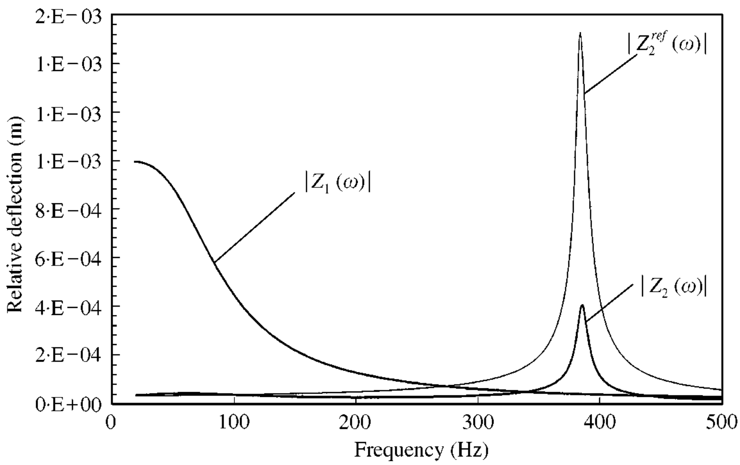

Figure 3. Relative deflections $|Z_1(\omega)|$, $|Z_2(\omega)|$, $|Z_2^{ref}(\omega)|$ in traditional design.

Typical values of the natural frequency and loss factor, which were obtained from experimentation (see PRACTICAL CASE) are $\Omega_2/2\pi = 386$ Hz, $\zeta_2 = 0{\cdot}011$.

By substituting the values of the parameters of the primary suspension $\zeta_1 = 1/\sqrt{2}$ and $\Omega_1/2\pi = 70{\cdot}6$ Hz, as obtained above, into equations (7)–(9) the magnitude of absolute acceleration of the internal element $|\ddot{X}_2(\Omega_2)| = 237\,\boldsymbol{g}$ and relative deflection of the internal component $|Z_2(\Omega_2)| = 0{\cdot}4$ mm are found.

Figure 2 shows the dependencies $|\ddot{X}_1(\omega)|$, $|\ddot{X}_2(\omega)|$ which are obtained in the example considered. In Figure 2, the dependence $|\ddot{X}_2^{ref}(\omega)|$ also shows the reference dynamic response of the PCB in the case of rigid mounting. From Figure 2, it can be seen that the traditionally designed vibration isolator reduces the resonant acceleration of the PCB by a factor of $\sim 3{\cdot}8$ compared to the case of rigid mounting.

Figure 3 shows the dependencies $|Z_1(\omega)|$, $|Z_2(\omega)|$ which are obtained in the example considered. In Figure 3, the dependence $|Z_2^{ref}(\omega)|$ also shows the reference relative deflection of the PCB in the case of rigid mounting. From Figure 3, the traditionally designed vibration isolator reduces the resonant relative deflection of the PCB by the same factor of $\sim 3{\cdot}8$ compared to the case of rigid mounting.

2.2. NEW APPROACH

In contrast to the traditional approach to the optimal design of vibration isolators (as described above), the new design approach focuses primarily on the dynamic properties of the sensitive internal element. In this instance, the external electronic box might be thought to be and utilized as the *first-level vibration isolation stage* (mechanical low-pass filter) relative to the internally mounted sensitive component. In this approach, the optimal problem is reformulated as follows:

To determine the natural frequency and loss factor Ω_1, ζ_1 of the primary system which minimises the resonance acceleration (relative deflection) of the secondary system, subject to the restraint, $|Z_1| \leqslant \Delta$.

Mathematically, this is expressed in the form

$$|\ddot{X}_2|_{max} \to \min \quad \text{or} \quad |Z_2|_{max} \to \min;\ |Z_1| \leqslant \Delta. \tag{10}$$

From equations (7) and (8), taking into account the small value of the loss factor ζ_2,

$$|\ddot{X}_2(\omega)|_{max} = |\ddot{X}_2(\Omega_2)| = \frac{1}{2\zeta_2}\sqrt{\frac{\Omega_1^4 + 4\Omega_2^2\Omega_1^2\zeta_1^2}{(\Omega_1^2 - \Omega_2^2)^2 + 4\Omega_2^2\Omega_1^2\zeta_1^2}}\,\ddot{Y}_0. \tag{11}$$

Similarly,

$$|Z_2(\omega)|_{max} = |Z_2(\Omega_2)| = \frac{1}{2\zeta_2\Omega_2^2}\sqrt{\frac{\Omega_1^4 + 4\Omega_2^2\Omega_1^2\zeta_1^2}{(\Omega_1^2 - \Omega_2^2)^2 + 4\Omega_2^2\Omega_1^2\zeta_1^2}}\,\ddot{Y}_0. \tag{12}$$

From (6),

$$\Omega_1 = \sqrt{\frac{\ddot{Y}_0}{2\Delta\zeta_1\sqrt{1-\zeta_1^2}}}. \tag{13}$$

Substitution of expression (12) into equation (10) or (11) eliminates Ω_1.

From equations (10)–(12), the optimal value of the loss factor ζ_1, which simultaneously minimizes $|\ddot{X}_2(\omega)|_{max}$ and $|Z_2(\omega)|_{max}$ at a given $\ddot{Y}_0$ and Δ, may be found numerically. Application of the values $\ddot{Y}_0 = 20\boldsymbol{g}$ and $\Delta = 1$ mm (see example above) yields

$$\zeta_1^{opt} = 0{\cdot}23.$$

Substituting this value into equation (12) yields

$$\frac{\Omega_1^{opt}}{2\pi} = 105{\cdot}7 \text{ Hz}.$$

Figure 4 shows the dependencies $|\ddot{X}_1(\omega)|$, $|\ddot{X}_2(\omega)|$ and $|\ddot{X}_2^{ref}(\omega)|$. Figure 5 shows the dependencies $|Z_1(\omega)|$, $|Z_2(\omega)|$ and $|Z_2^{ref}(\omega)|$.

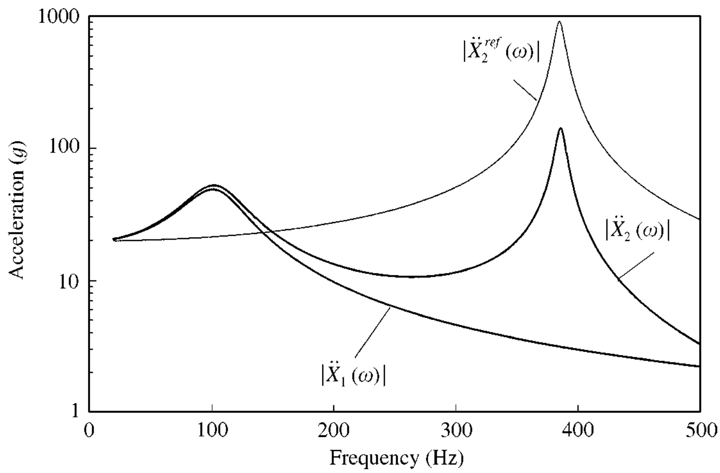

Figure 4. Absolute accelerations $|\ddot{X}_1(\omega)|$, $|\ddot{X}_2(\omega)|$, $|\ddot{X}_2^{ref}(\omega)|$ in new design.

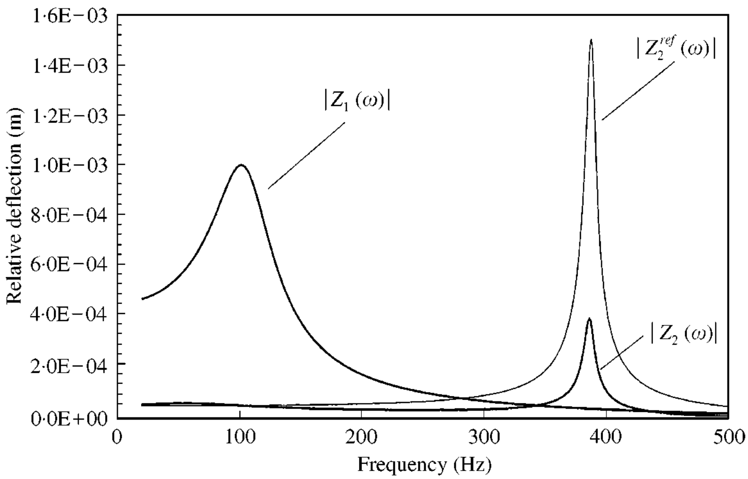

Figure 5. Relative deflections $|Z_1(\omega)|$, $|Z_2(\omega)|$, $|Z_2^{ref}(\omega)|$ in new design.

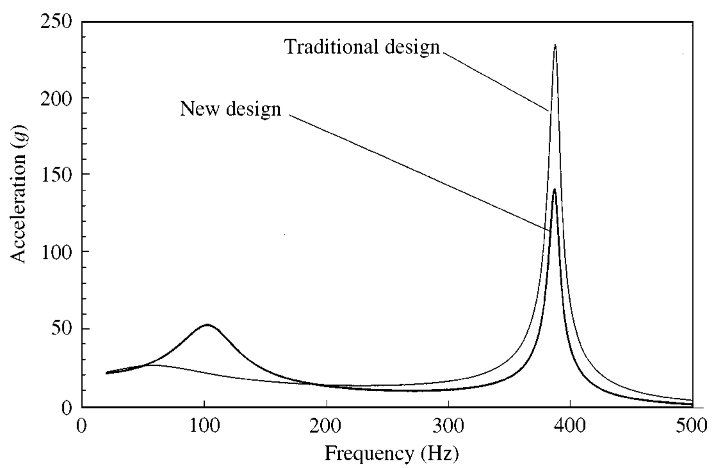

Figure 6. Comparison of absolute accelerations $|\ddot{X}_2(\omega)|$ in traditional and new designs.

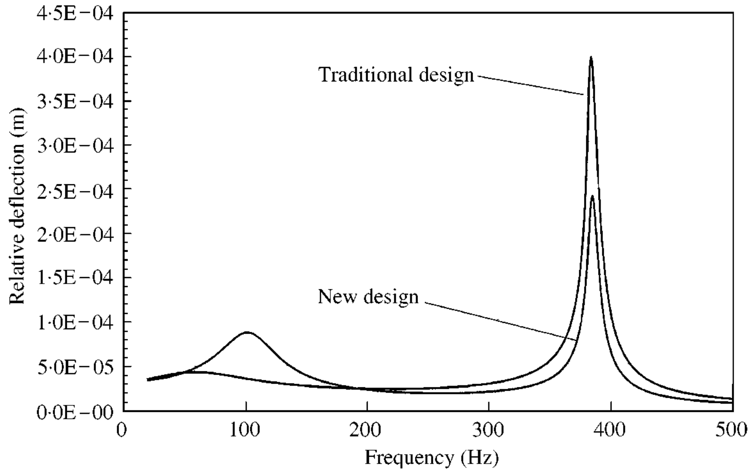

Figure 7. Comparison of relative deflections $|Z_2(\omega)|$ in traditional and new designs.

From Figure 4, the resonant magnitude of absolute acceleration of the internal element is $|\ddot{X}_2(\Omega_2)| = 142\,\boldsymbol{g}$. From Figure 4, it can be seen that the newly designed vibration isolator reduces the resonant acceleration of the PCB by a factor of 6·4 compared to the case of rigid mounting. This means a 67% improvement (in terms of absolute accelerations) in the vibration protection of the PCB compared to the traditional design.

From Figure 5, the magnitude of relative deflection of the internal element is $|Z_2(\Omega_2)| = 0{\cdot}23$ mm. From the same Figure it can be seen that the newly designed vibration isolator reduces the resonant acceleration of the PCB by a factor of 6·4 compared to the case of rigid mounting. This means a 67% improvement (in terms of relative deflections) in the vibration protection of the PCB compared to the traditional design.

Figures 6 and 7 compare the magnitudes of absolute accelerations and relative deflections of the internal element which are obtained by the traditional and new designs.

3. PRACTICAL CASE: VIBRATION PROTECTION OF PCB

3.1. ESTIMATION OF THE PCB PARAMETERS

A simple PCB which carries the flatpack chip is chosen to demonstrate the principles of the practical design and attainable performance of a vibration protection system. The above PCB is mounted rigidly over the fixture which is attached to the electrodynamic shaker (vibration test system V550/PA550L, Ling Dynamic Systems Ltd.) with its plane perpendicular to the direction of the motion (see the schematics of experimental rig in Figure 8).

The miniature accelerometer (B&K, Type 4393) is bonded to the central flatpack chip. This accelerometer contributes an additional mass of 2·4 **g** and is considered as a part of the system which has an overall effective mass of 30 g. The reference accelerometer (B&K, Type 4393) is mounted over the fixture. Both accelerometers are connected to the charge conditioners (B&K, Type 2635). The data analysis is performed by means of the portable vibration analyser (Signal Calc Ace, Data Physics Corporation).

First, the absolute transmissibility of the PCB is measured. For this purpose a swept-sine vibration with magnitude of acceleration 20 **g** in the frequency range of 20–500 Hz (sweep rate 5 Hz/s) excites the PCB; the module of absolute transmissibility is measured.

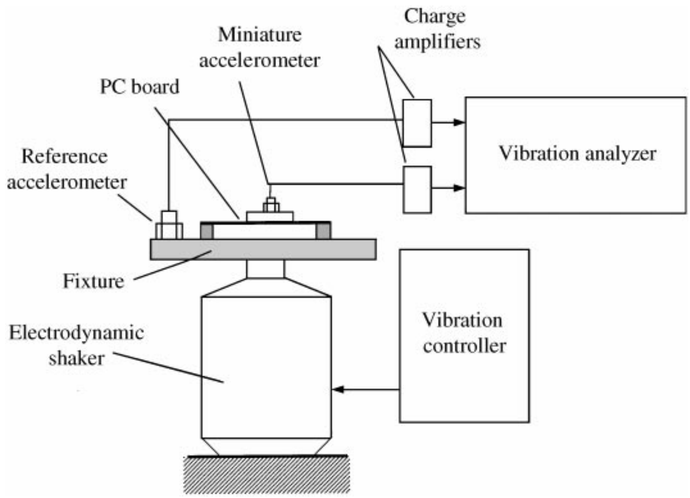

Figure 8. Experimental rig for identification the dynamic properties of PCB.

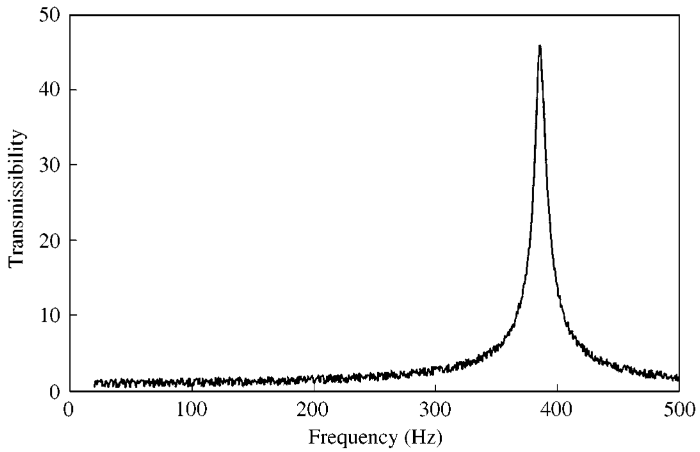

Figure 9. Experimental transmissibility of PCB.

Figure 9 shows the graph of the module of absolute transmissibility against frequency. From curve fitting, the natural frequency is found as $\Omega_2 = 386$ Hz and loss factor as $\xi_2 = 0{\cdot}011$.

3.2. EXPERIMENTAL STUDY OF VIBRATION ISOLATION

To put the theory into practice, the vibration isolation system was modelled for the above mentioned PCB. The optimal parameters of the primary suspension $\Omega_1^{opt}/2\pi = 105{\cdot}7$ Hz and $\zeta_1^{opt} = 0{\cdot}23$ have been estimated in the preceding section. Although the test rig does not truly represent an actual electronic box in a dimensional sense nor the PCB system fastening conditions, it has comparable properties of masses, natural frequencies and damping ratios.

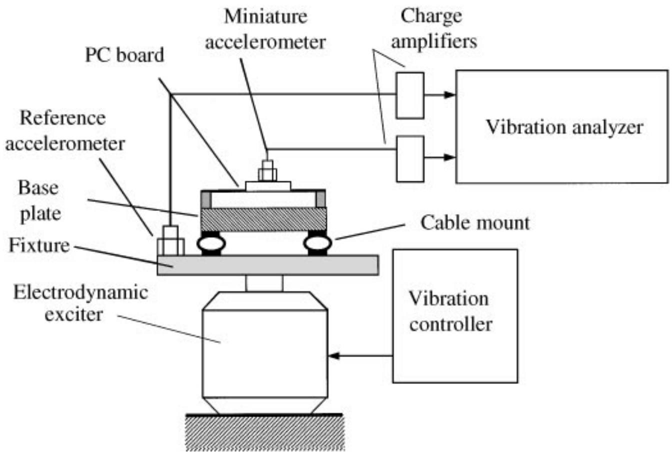

Figure 10. Experimental rig for studying the dynamic properties of compound system.

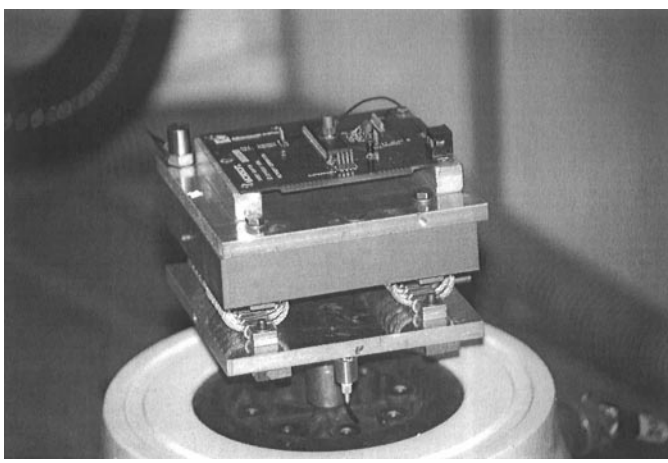

Figure 11. Detailed layout of the compound system.

Standard commercially available cable mounts (Enidine,† Shock Tech,‡ Barry Controls,§ Aeroflex International [11]) provide the desired loss factor and natural frequency. Such cable mounts are especially designed to withstand severe environmental conditions while demonstrating the persistence of parameters over a wide temperature range compared to the polymer isolators. Since the quality of the vibration protection system depends strongly on the parameters of the primary suspension, this feature is, probably, the most critical for the choice of an appropriate vibration isolator.

Figure 10 shows the experimental rig, which is quite similar to that used above for the estimation of the dynamic properties of the PCB. The PCB is secured to a larger base plate, suspended over the fixture by the two Shock Tech cable mounts. Figure 11 shows the detailed layout of experimental rig.

† See http://www.enidine.com/Wire%20Rope/WireRope_1.html#WireRopePDF.
‡ See http://www.shocktech.com/products.htm.
§ See http://www.bpg-inc.com/barry.htm.

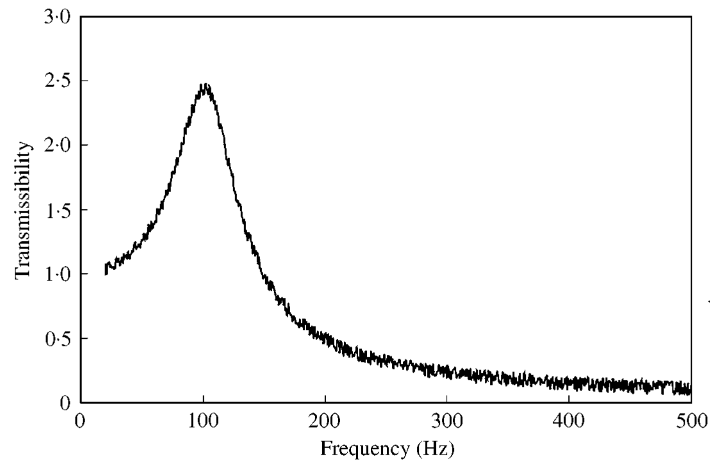

Figure 12. Experimental absolute transmissibility of the primary system.

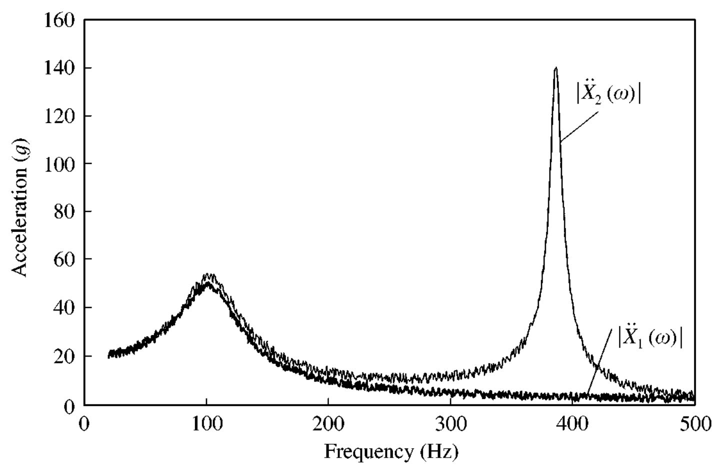

Figure 13. Accelerations of the primary and secondary systems.

A proper choice of cable mounts enables the desired characteristics of the primary system to be obtained. The swept-sine test is carried out (magnitude or harmonic acceleration is 20 $\boldsymbol{g}$ in the frequency range 20–500 Hz; sweep rate 5 Hz/s) and the module of absolute transmissibility of the primary system is estimated (see Figure 12). The procedure of curve fitting gives a natural frequency of 105 Hz and loss factor of 0·22, which are close to the desired values. From the same swept-sine test the absolute accelerations of the primary and the secondary systems, along with the relative deflection of the primary and secondary systems, are estimated. Figure 13 shows the magnitudes of the absolute acceleration of the primary and of the secondary systems against the frequency (compare with Figure 4). Figure 14 shows the peak deflection of the primary and secondary systems against the frequency (compare with Figure 5). The experimental results are in fair agreement with analytical solution.

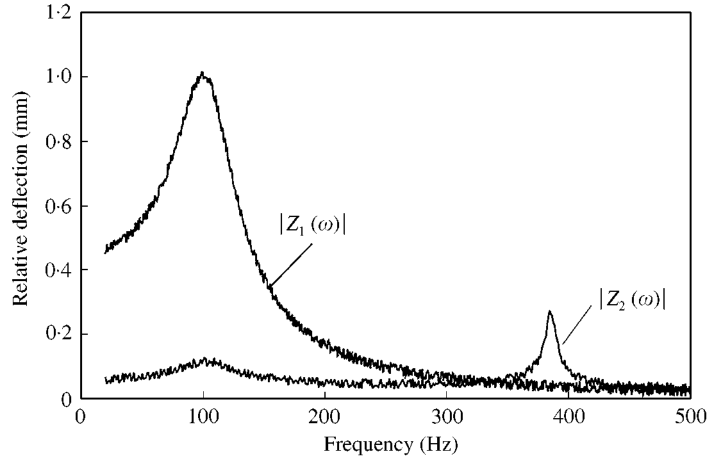

Figure 14. Relative deflections of the primary system.

4. CONCLUSIONS

The problem of vibration protection of the critical components of electronic equipment, which operates in harsh environmental conditions, is addressed. The solution proposed utilises the resiliently mounted electronic box as the *first-level vibration isolation stage* (mechanical low-pass filter) relative to the sensitive internal components. It was shown analytically and experimentally that the protection from harmonic vibration might be improved by about 67% compared to traditional design.

REFERENCES

1. D. S. STEINBERG 1988 *Vibration Analysis for Electronic Equipment*. New York: John Wiley & Sons, Inc.
2. C. V. STAHLE 1989 *Proceedings of Damping*'89, *FL*, *Vol.* AAB. Pioneering damping in space at General Electric Astro Space Division.
3. J. V. DENTE, J. L. MCCUTCHEON and A. K. JAIN 1995 *ShockTech Reports*. Vibration qualification of commercial computers for use in military tactical environments. http:///www.shocktech.com/reports.html.
4. S. S. SEGUIN, H. LEKUCH *COTS Journal*. Isolating COTS equipment from shock and vibration. http://www.shocktech.com/shockVibCusAss.htm.
5. J. W. DALLY 1990 *Packaging of Electronic Systems* (*A mechanical Engineering Approach*). New York: McGraw-Hill Publishing Company.
6. M. Z. KOLOVSKY 1999 *Nonlinear Dynamics of Active and Passive Systems of Vibration Protection*. Berlin: Springer-Verlag.
7. C. M. HARRIS editor 1996 *Shock and Vibration Handbook*. New York, London: McGraw-Hill.
8. K. V. FROLOV editor 1981 *Vibration in Engineering*, *Reference*, *Vol.* 6. Mashinostroenie. Moscow (in Russian).
9. E. SEVIN, W. D. PILKEY 1971 *Optimum Shock and Vibration Isolation*. US Department of Defence: The Shock and Vibration Information Centre.
10. *Military Standard*, *Environmental Test Methods*, *MIL-STD*-810*E* 1988 *Department of Defence*, *Washington*, *DC*.
11. E. JASSON 1999 *COTS Journal*. Shock and vibration isolators for COTS equipment. http://www.rtcgroup.com/cotsjournal/cotsjulyaugustp27.html.