Design and analysis of shock and random vibration isolation system for a discrete model of submerged jet impingement cooling system

Ashkan Haji Hosseinloo, Fook Fah Yap and Liang Ying Lim

Abstract
High-powered embedded computing equipment using air transport rack (ATR) form-factors are playing an ever-increasing role in critical military applications in air, land and sea environments. High power and wattage of the electronics and processors require large heat dissipation, and thus more sophisticated and efficient thermal cooling systems such as loop heat pipes or jet impingement systems are demanded. However, these thermal solutions are more susceptible to harsh military environments and thus, for proper performance of thermal and electronic equipment, they need to be protected against shock and vibration inherent in harsh environments like those in military applications. In this paper, an isolated ATR chassis including two jet impingement chambers is modeled as a three-degrees-of-freedom system and its response to random vibration and shock has been studied. Both finite element and experimental modal analysis is utilized to characterize dynamics of the components of the jet impingement system. The response of the model is compared to that of the traditional single-degree-of-freedom model, and the isolation system is optimized in terms of its damping.

Keywords
Discrete model, jet impingement, modal analysis, random vibration, shock, vibration isolation

1. Introduction
As the trend of highly integrated electronics and simultaneous miniaturization escalates to include faster processors, more functions, and higher bandwidths, electronics continue to become more compact in response to size limitations and strict reliability requirements (Ohadi and Jianwei, 2004). The result is an increasing heat flux at both the component and circuit board levels. Limited power dissipation with some other pitfalls of convection and conduction cooling methods are pushing towards exotic techniques such as liquid-cooling and spray-cooling systems. These methods can provide higher heat dissipation power and can be used in rugged environments like in fighters or ground mobile vehicles if they are made rugged accordingly.

A jet impingement cooling system is a relatively new cooling technology that is nowadays being given serious consideration in high power electronics cooling systems. In this cooling system working fluid, which is usually water, is pumped into a chamber and is then pushed through tiny nozzles drilled into a so-called nozzle plate. On the other side of the nozzle plate, the high velocity water jets impinge on the heat sources such as circuit boards either directly or through an intermediate plate called an impingement plate that is mounted back to back to the heat sources. Thus, the heat is dissipated and water is drained to a heat rejection unit (HRU) such as a radiator to reject the heat from the water, and it is then pumped back to the chamber to repeat the closed loop. In this project, two impingement units are used to dissipate a total power of 2 kW from two circuit boards (1 kW each).
A CAD model of the impingement system is illustrated in Figure 1, and the thermal diagram of the cooling system is depicted in Figure 2. In the latter diagram the AC power is used to provide the required wattage for the heaters, and the data acquisition unit is used to capture the temperature of the thermocouples mounted in the impingement plates to assess the performance of the cooling system.

Two units of the cooling system shown in Figure 1 together with their attached heating sources are mounted in an ATR using wedge locks. The whole ATR chassis is subjected to severe random vibration and shock. The heat rejection unit is placed outside the chassis and is not subjected to shock and vibration. The goal in this project is to protect the cooling system rather than the heating sources (which could be printed circuit boards or some other heat sources) against shock and vibration. A common practice to ruggedize and protect sensitive devices against shock and vibration is either to stiffen them, which adds to the total mass and volume and is not always practical, or to isolate them from the base excitations. The traditional optimal design for vibration isolation from random vibration is based on a trade-off choice of damping and stiffness properties of mounts, and is focused primarily on optimizing the dynamic response of the entire enclosure or chassis, subject to limitations imposed on their rattle space (Veprik, 2003). However, some devices include lightly damped components that are highly responsive to a wide range of frequencies which are not considered in the traditional isolation system design. Consequently, traditionally designed isolation systems ignoring the responsive internal components are insufficient for maintaining a fail-safe vibration environment for the system.

Although the traditional design may fulfill the shock or vibration requirements in some applications (Allen et al., 2000; Yap et al., 2006; Harmoko et al., 2009; Oh et al., 2012; Baril et al., 2012), it does not always result in an optimum design. Therefore, the traditional trend of isolators’ design has been challenged in the last decade in certain areas such as printed circuit boards, cryogenic coolers and hard disk drives. Veprik and Babitsky (2000) added a secondary degree of freedom (d.f.) for a printed circuit board to the model of an isolated rigid electronic box and tried to optimize the isolator’s parameters for a harmonic excitation to minimize the PCB’s absolute acceleration or relative deflection, subject to an overall rattle space of the box. Later, Veprik (2003) used the same model to optimize the isolator’s properties this time for random vibration input. Veprik and his colleagues (2001) designed an isolation system for an infrared equipment to minimize the vibration-induced line-of-sight jitter which was caused by the relative deflection of the infrared sensor and the optic system. They added a second d.f. for the cold finger of the cryogenic cooler to the main single-d.f. model of the system to improve the model as a more realistic two-d.f. system. Later on, they used this two-d.f. model which included the sensitive cantilevered cold finger to design optimum compliant snubbers for impact protection (Veprik et al., 2008).
Jet impingement systems include dynamically sensitive components that could be very responsive to high frequency excitations. As mentioned earlier, the need for use of these cooling systems in harsh military environments is now increasing more than ever, hence calls for systematic shock and vibration protection analysis which is not available at the time. The authors previously designed an integrated linear-non-linear vibration isolation system for the ATR chassis mentioned above catering for base random vibration input in the presence and absence of g-loading (Hosseinloo and Yap, 2011). However, their model ignored the presence of internal components. In this article, the authors optimize a linear isolation system for shock and random vibration inputs considering the sensitive internal components of the impingement system.

2. Mathematical modeling

2.1. Traditional Single-degree-of-freedom model of the isolated system

A traditional single-d.f. model of the isolated system is shown in Figure 3, where the chassis and the included two cooling chambers and two micro pumps are modeled as a single rigid object with mass $m$. Isolator stiffness and damping are designated by $k_c$ and $c_c$.

Using random vibration theory (Newland, 1996) and transmissibility functions for an single-d.f. model, variance of absolute acceleration of the chassis and its relative deflection are found as, respectively:

$$\sigma_x^2 = \int_0^\infty S_{x_x}(j\omega)|T^{th}(j\omega)|^2 \, d\omega$$

$$= \int_0^\infty S_{x_x}(j\omega)\left|\frac{(\omega_c^2 + 2j\omega_c\zeta)\omega_c^2}{(\omega_c^2 - \omega^2 + 2j\omega_c\zeta)}\right|^2 \, d\omega$$

Figure 2. Thermal diagram of the jet impingement cooling system.

Figure 3. Single-degree-of-freedom model of the chassis with the cooling system.
and

$$\sigma_z^2 = \int_0^\infty \frac{S_b(j\omega)}{\omega^4} |T_{rel}(j\omega)|^2 d\omega = \int_0^\infty \frac{S_b(j\omega)}{\omega^2 - \omega^2 + 2j\omega \epsilon_\omega \zeta_c^2} d\omega$$

(2)

where $z$ is the chassis relative displacement ($z = x_c - x_b$), and $S_b$ is the base one-sided power spectral density. Conventional notation is used for natural undamped frequency ($\omega_c$) and damping ratio ($\zeta_c$) as below,

$$\omega_c = \sqrt{\frac{k}{m}} \text{ and } \zeta_c = \frac{c}{2\sqrt{km}}$$

(3)

The base excitation is Gaussian random excitation with zero mean value. Thus, as the system is linear, the response will also be Gaussian with zero mean that consequently allows us to use the $3\sigma$ rule to estimate the maximum vibration travel of the chassis as

$$z_{peak} = 3\sigma_z$$

(4)

Fatigue damage accumulation of sensitive components depends mainly on the transmitted absolute acceleration (Ho et al., 2003); hence, to assess and quantify the quality of vibration isolation a parameter called attenuation factor (AF) is introduced and defined as the root mean square (RMS) value of the absolute acceleration of the hard mounted chassis divided by that of the isolated chassis as follows:

$$AF = \frac{\sigma_{z_h}}{\sigma_{z_i}}$$

(5)

Finally, the shock response of the chassis is obtained by solving the following dynamic equation:

$$\dddot{z} + 2\zeta_c \omega_c \dot{z} + \omega_c^2 z = \ddot{x}_b$$

(6)

### 2.2. Proposed three-degree-of-freedom model of the isolated system

The chassis is quite stiff and could be considered rigid for the frequency range of the excitation considered in this article, i.e. 20 Hz to 2000 Hz. There are two I-drive IEG micro pumps in the chassis that apply quite small harmonic forces to the chassis and are considered negligible. However, the three components of the impingement cooling system mentioned earlier could be flexible to the excitation normal to the impingement and nozzle plates. To check if the three components – namely, chamber housing, nozzle plate and impingement plate – have responsive modes to the random base vibration ranging from 20 Hz to 2000 Hz, modal analysis of the components is conducted in ANSYS. For the modal analysis both the chamber housing and the impingement plate have been fully constrained at their mounting holes. Table 1 shows the first six undamped natural frequencies of the chamber housing and impingement plate, and Figure 4 illustrates the corresponding mode shapes of these natural frequencies for the two components.

A rule of thumb in modal analysis states that all frequencies below one-and-a-half times the maximum excitation frequency should be considered for dynamic analysis. According to the modal analysis, the first natural frequency of the chamber housing is well above 3000 Hz and it can be considered rigid. The impingement plate is quite flexible in the excitation frequency range; however, this plate is mounted back to back to a thick copper heating plate (Figure 5) that both together can be considered rigid. According to the modal analysis of the impingement plate, if a heating source with relatively negligible flexural rigidity is used, the flexibility of the impingement plate should be considered.

The nozzle plate has 285 tiny nozzles with diameter of 0.5 mm which makes the element size in the finite element (FE) meshing very small, and hence increases the number of elements and the computational time significantly. To check if the nozzles have a considerable effect on the modal analysis of the plate, two quarter nozzle plates with symmetric boundary conditions have been modeled; one with and the other without nozzle holes, and their first few frequencies have been compared in Table 2. According to the table, nozzle holes have no significant effect on the modal analysis, and thus nozzle plate without nozzles is used for the modal analysis. The first six natural frequencies of the nozzle plate and the corresponding mode shapes are given by Table 1 and Figure 6, respectively.

According to Table 1 the nozzle plate has its first natural frequency below 3000 Hz, hence it should be considered in the dynamics of the problem. Since there is only one responsive mode for the excitation frequency range considered here, the nozzle plate

### Table 1. The first six natural frequencies of the three components of the cooling chamber.

<table>
<thead>
<tr>
<th>Mode number</th>
<th>Natural frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4395 7801 9364 12618 12991 15773</td>
</tr>
<tr>
<td>2</td>
<td>935 1583 2176 2643 2772 3773</td>
</tr>
<tr>
<td>3</td>
<td>2763 4983 5027 7854 9022 7942</td>
</tr>
<tr>
<td>4</td>
<td>4395 7801 9364 12618 12991 15773</td>
</tr>
<tr>
<td>5</td>
<td>935 1583 2176 2643 2772 3773</td>
</tr>
<tr>
<td>6</td>
<td>2763 4983 5027 7854 9022 7942</td>
</tr>
</tbody>
</table>
could be modeled as an additional single-d.f. system. There is another assumption made in this analysis regarding the water effect on the dynamics of the system. Because there is a constant water pressure difference between the two sides of the nozzle plate, it could reasonably be assumed that the water affects the plate by giving it a negligible small initial deflection and will also add some viscous damping to the system that is ignored in this article.

Having considered the assumptions, a three-d.f. model of the system could be represented as shown in Figure 7. Since the two nozzle plates are identical the three-d.f. model could be further reduced to a two-d.f. one to simplify the system dynamics.
In the proposed model the damping and effective mass of the nozzle plate are still unknown and they are found using an impact hammer test. The hammer test set-up is shown in Figure 8. The fixture is designed so as to resemble the actual boundary conditions. As only the first mode of the nozzle plate is of interest, the nozzle plate is hit at its center and the acceleration is read at the same point at the other side of the plate. Figure 9 shows the point accelerance frequency response function (FRF) of the nozzle plate. An embedded nonlinear least square curve fitting tool in MATLAB is used to curve fit the experimental accelerance of the plate with analytical accelerance equation.

Table 2. Comparison between natural frequencies of the quarter nozzle plates with and without nozzles.

<table>
<thead>
<tr>
<th>Mode number</th>
<th>Natural frequency (Hz)</th>
<th>Discrepancy percentage %</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>With nozzles</td>
<td>Without nozzles</td>
</tr>
<tr>
<td>1</td>
<td>2752.5</td>
<td>2762.7</td>
</tr>
<tr>
<td>2</td>
<td>7759.8</td>
<td>7786.2</td>
</tr>
<tr>
<td>3</td>
<td>8945.4</td>
<td>8979.8</td>
</tr>
<tr>
<td>4</td>
<td>13306</td>
<td>13316</td>
</tr>
<tr>
<td>5</td>
<td>14433</td>
<td>14482</td>
</tr>
</tbody>
</table>

Figure 6. The first six mode shapes of the nozzle plate.
Figure 7. Three-degrees-of-freedom model of the isolated chassis and the cooling system.

Figure 8. Impact hammer test setup for modal identification of the nozzle plate.
of an single-d.f. model. The curve fitted parameters, namely the effective mass \( (m_n) \), natural frequency \( (\omega_n) \) and damping ratio \( (\zeta_n) \) of the first mode of the nozzle plate are found as 0.025 kg, 3003 Hz, and 3.7\%, respectively. The 8\% discrepancy between the natural frequencies obtained from FE and experimental analysis for the first mode is mainly due to the difference between the actual boundary condition and the simplified one used in FE modal analysis. With the above extracted effective mass, the nozzle plate mass ratio \( (\mu) \) will be about 0.0015.

After parameter identification through FE and experimental modal analysis, and modeling the system, dynamic equations of the system can be derived. Complex transmissibility of the absolute acceleration of the nozzle plate is derived as

\[
TR_{\omega_n}^\nu (s) = TR_{\omega_n}^\chi (s) = \frac{\dot{X}_n (s)}{X_b (s)} = \frac{X_n (s)}{s^2 X_b (s)} = \frac{(2\zeta_n \omega_n s + \omega_n^2)}{(s^2 + 2\zeta_n \omega_n s + \omega_n^2)} TR_{\chi b}^\nu (s),
\]

where \( TR_{\omega_n}^\nu (s) \) is complex transmissibility of the absolute acceleration of the nozzle plate, and

\[
TR_{\chi b}^\nu (s) = TR_{\chi b}^\chi (s) = \frac{\dot{X}_b (s)}{X_b (s)} = \frac{X_b (s)}{X_b (s)} = \frac{(s^2 + 2\zeta_n \omega_n s + \omega_n^2)(2\zeta_n \omega_n s + \omega_n^2)}{(s^2 + 2\zeta_n \omega_n s + \omega_n^2)[s^2 + (2\zeta_n \omega_n + 4\mu \zeta_n \omega_n) s + \omega_n^2 + 2\mu \omega_n^2] - 2\mu(2\zeta_n \omega_n s + \omega_n^2)^2}
\]

In the equations above, conventional notation has been used as follows:

\[
\omega_n^2 = \frac{k_n}{m_n}, \quad \omega_c^2 = \frac{k_c}{m_c}, \quad \zeta_c = \frac{c_c}{2\sqrt{k_n m_c}}, \quad \zeta_n = \frac{c_n}{2\sqrt{k_n m_n}}
\]

and \( \mu = \frac{m_n}{m_c} \)

Furthermore, the complex transmissibility functions of the chassis rattle space and the nozzle plate deflection can be obtained as

\[
TR_{\omega_n}^\nu (s) = TR_{\omega_n}^{\nu - \chi b} (s) = \frac{X_n (s) - X_b (s)}{X_b (s)} = \frac{X_n (s) - X_b (s)}{s^2 X_b (s)} = \frac{1}{s^2} \left( TR_{\omega_n}^\nu (s) - 1 \right)
\]

(10)

and

\[
TR_{\chi b}^\nu (s) = TR_{\chi b}^{\nu - \chi} (s) = \frac{X_b (s) - X_n (s)}{X_n (s)} = \frac{X_n (s) - X_b (s)}{X_b (s)} = \frac{1}{s^2} \left( TR_{\omega_n}^\nu (s) - TR_{\omega_n}^\chi (s) \right)
\]

(11)

Figure 9. Experimental and fitted theoretical single-degree-of-freedom accelerance curves.
According to the last two equations the chassis vibration travel and the nozzle plate deflection are designated as \( z_1 \) and \( z_2 \). Having derived the transmissibility functions, RMS absolute acceleration of the nozzle plate can be calculated as

\[
\sigma_{x_n} = \sqrt{\int_0^{+\infty} S_{x_n}(\omega) d\omega} = \sqrt{\int_0^{+\infty} |TR_{x_n}(j\omega)|^2 S_{x_n}(\omega) d\omega}
\]

(12)

The maximum nozzle plate deflection and chassis vibration travel can also be evaluated using the 3\( \sigma \) rule as follows, respectively:

\[
z_{\text{peak}}^2 = 3\sigma_{z_2} = \int_0^{+\infty} S_{z_2}(\omega) d\omega = \int_0^{+\infty} |TR_{x_n}(j\omega)|^2 S_{x_n}(\omega) d\omega
\]

(13)

and

\[
z_{\text{peak}}^1 = 3\sigma_{z_1} = \int_0^{+\infty} S_{z_1}(\omega) d\omega = \int_0^{+\infty} |TR_{x_n}(j\omega)|^2 S_{x_n}(\omega) d\omega
\]

(14)

For a meaningful definition, attenuation factor for the three-d.f. system is defined as

\[
AF = \frac{\sigma_{z_1}|_{\omega = 0}^{+\infty}}{\sigma_{x_n}}
\]

(15)

After defining and formulating the attenuation factor and relative deflections, the optimization problem should be defined and formulated as well; the problem is to find \( \omega_c \) and \( \zeta_c \) such that \( \ddot{x}_n \) and \( z_2 \) are minimized, subject to a maximum rattle space: \( z_1 \leq \Delta_{\text{max}} \) where \( \Delta_{\text{max}} \) is the maximum allowed vibration travel of the chassis.

Regarding the problem definition, one should note that:

(i) Minimization of \( \ddot{x}_n \) is important for fatigue and catastrophic failure of the nozzle plate. Acceleration of the nozzle plate may also affect the impingement mechanism.

(ii) Minimization of \( z_2 \) is important for thermal performance of the cooling system. \( z_2 \) is a measure of change in impingement height (the clearance between the nozzle plate and the impingement surface), which directly affects the cooling performance.

(iii) Rattle space and vibration travel constraint is important because the system should be finally confined within a specified volume.

In addition to random base excitation, response of the system to base shock excitation should also be analyzed. As mentioned earlier, since the two d.f. pertaining to the two nozzle plates are identical, dynamic equations of motion can be reduced from three to two, and be formulated in the matrix form as follows:

\[
\ddot{z} + C \dot{z} + K z = r \ddot{v}_b
\]

(16)

where

\[
z = [z_1 \ z_2]^T, K = \begin{bmatrix}
\omega_c^2 + 2\mu\omega_n^2 & -2\mu\omega_n^2 \\
-\omega_n^2 & -\omega_n^2
\end{bmatrix},
\]

\[
r = [-1 \ -1]^T, \text{ and } C = \begin{bmatrix}
2\zeta_c\omega_n + 4\mu\zeta_n\omega_n & -4\mu\zeta_n\omega_n \\
-2\zeta_n\omega_n & 2\zeta_n\omega_n
\end{bmatrix}
\]

(17)

In the above equation, \( z_1 \) is deflection of the nozzle plate relative to the base (\( z_3 = x_n - x_b \)).

Having derived the equations of motion of the two systems, the response of the single-d.f. and three-d.f. models to shock and random vibration inputs will be presented in the following section.

3. Results and discussion

The random vibration excitation considered in this article is an ergodic and stationary random vibration with power spectral density (PSD) shown in Figure 10(a). The vertical axis of the figure is in decibels and in units of (g\(^2\)/Hz). The flat part of the profile corresponds to a PSD level of 0.50 g\(^2\)/Hz. The corresponding overall RMS value of the base acceleration is about 31grms. The shock excitation considered here is an end-terminating sawtooth shock with duration of 11 ms and maximum acceleration of 20 g (Figure 10(b)).

Figure 11 shows dependency of the attenuation factor on damping ratio for a few isolation frequencies. According to the figure, attenuation factor decreases with an increase of the isolation frequency for any isolation damping ratio; however, there is an optimum damping ratio that maximizes the attenuation factor at the nozzle plate for a given isolation frequency. The optimum damping ratio varies from 0.17 to 0.31 for isolation frequency range of [20, 50] Hz. Although the optimum damping ratios for a given isolation frequency are almost the same for the two models, the
For a given isolation frequency, as the isolation damping increases maximum isolator deflection (chassis vibration travel or rattle space) decreases; however, this value for any given damping ratio is maximized at an isolation frequency of about 22 Hz. This is because of the resonance of the isolation system at the beginning of the base excitation frequency; lower excitation frequencies result in higher displacements. Figure 12 shows how vibration travel of the chassis varies with

![Figure 10](image1.png)  
**Figure 10.** (a) Power spectral density profile of the base random acceleration; (b) end-terminating sawtooth shock profile of the base.

![Figure 11](image2.png)  
**Figure 11.** Dependency of attenuation factor on isolation damping ratio at different isolation natural frequencies.

For a given isolation frequency, as the isolation damping increases maximum isolator deflection (chassis vibration travel or rattle space) decreases; however, this value for any given damping ratio is maximized at an isolation frequency of about 22 Hz. This is because of the resonance of the isolation system at the beginning of the base excitation frequency; lower excitation frequencies result in higher displacements. Figure 12 shows how vibration travel of the chassis varies with
isolation frequency at a given damping ratio. According to the figure, there is not much difference between the responses of the two models in terms of the chassis vibration travel. The reason is that the nozzle plate mass ratio is so small ($\mu = 0.0015$) that the nozzle plate’s dynamics will not have a significant effect on the dynamics of the rest of the system.

A pitfall of the single-d.f. model is that it is incapable of giving any information about the nozzle plate deflection; however, in view of the three-d.f. model this information can be evaluated. Nozzle plate deflection is important as it affects the impingement height that consequently affects the thermal performance of the impingement system. It is desirable that the variation in impingement height is minimal. Figure 13 depicts how maximum deflection of the nozzle plate (change in the impingement height) varies with isolation damping ratio for different isolation frequencies. According to the figure, for a given isolation frequency there is an optimum damping ratio that minimizes the change in impingement height. This optimum damping value is the same damping that maximized the attenuation factor shown in Figure 11. This is because the frequency response function of the nozzle plate’s deflection is the same as that of the plate’s acceleration divided by the frequency squared. Because the power spectral density is a product of the input power spectral density and square of the FRF magnitude, PSD of the nozzle plate’s deflection will be the same as that of the plate’s acceleration divided by a frequency to power of four. This consequently results in the same optimum damping minimizing the plate’s acceleration and deflection.

Figure 14 depicts the shock response spectrum of the normalized absolute acceleration of the nozzle plate for the two models. Acceleration is normalized by the maximum input shock, i.e. 20 g. According to the figure there is not much difference between the two models’ responses for the shock input, and that is because frequency contents of the shock are not close enough to the nozzle plate’s first natural frequency. Figure 15 shows normalized maximum absolute acceleration of the nozzle plate as a function of isolation damping ratio at different isolation frequencies for the two models. For the single-d.f. model in Figures 14 and 15 the acceleration of the chassis is taken as the acceleration of the nozzle plate. According to Figure 15 there is an optimum damping ratio for isolation natural frequencies below 40 Hz that minimizes the maximum acceleration transmitted to the nozzle plate. This optimum damping ratio is about 0.25 for isolation frequencies ranging from 10 Hz to 40 Hz.

Just like the random vibration input, the maximum isolator’s deflection is analyzed for the shock input. The isolator’s deflection is important as it gives information about meeting the rattle space requirement and about the possibility of bottoming out if it is above the maximum allowed stroke of the isolator. Figure 16 shows the shock response spectra of the isolator’s deflection.
for the two single-d.f. and three-d.f. models. According to the figure, there is not much difference between the responses of the two models to the shock excitation and the reason lies again in the fact that the nozzle plate mass ratio to the rest of the system is too small to affect the dynamics of the chassis.

As mentioned earlier, the three-d.f. model enables us to look at maximum deflection of the nozzle plate while

**Figure 13.** Maximum deflection of the nozzle plate versus isolation damping ratio at different isolation frequencies.

**Figure 14.** Normalized absolute acceleration shock response spectrum of the nozzle plate for the single-degree-of-freedom and the three-degrees-of-freedom models.
the single-d.f. model is incapable of giving this information. Figure 17 shows maximum deflection of the nozzle plate as a function of isolation damping ratio at different isolation natural frequencies for the three-d.f. model. According to the figure there is an optimum damping ratio for isolation natural frequencies lower than 40 Hz that minimizes the nozzle plate’s maximum deflection. Similar to the random excitation input, this
optimum damping ratio is equal to the damping ratio that minimizes the nozzle plate’s maximum acceleration when subjected to the shock, i.e. a damping ratio of 0.25.

4. Conclusion

In this article random vibration and shock protection of submerged jet impingement cooling systems have been investigated and compared with traditional ruggedization methods. Unlike traditional methods the current research aims at reducing the transmitted vibration to internal critical components of the impingement system. FE and experimental modal analysis are conducted to characterize different components of the cooling system; hence, a three-d.f. model of the isolated system is proposed that has an addition of two d.f. for the nozzle plates as compared to the traditional single-d.f. model. The problem is defined as to find optimum isolator properties to minimize the nozzle plates’ acceleration and deflection while the whole chassis containing the impingement system is confined within an allowed rattle space.

The results show that there is a considerable difference between the calculated acceleration of the nozzle plate using the single-d.f. and three-d.f. models. Furthermore, the three-d.f. model enables us to calculate the nozzle plate’s deflection, which is important for the impingement cooling system performance. According to the results, for random base excitation there is an optimum damping ratio minimizing both the nozzle plate acceleration and deflection that varies from 0.17 to 0.31 for an undamped isolation frequency range of [20, 50] Hz. Furthermore, there is an optimum isolation damping ratio of 0.25 for isolation frequencies below 40 Hz that minimizes the nozzle plate acceleration and deflection when the system is subjected to base sawtooth shock excitation. This value of damping is also an optimum damping for random base excitation if the isolation frequency is 30 Hz. Therefore, an isolator with isolation frequency of 30 Hz and damping ratio of 0.25 could be an optimum isolator for the jet impingement cooling system considered here, subjected to the harsh environment of shock and random base vibration. With such an isolator there will be an attenuation factor of about 15 at the nozzle plate, and chassis vibration travel of less than 2.5 mm when subjected to random base vibration, and normalized transmitted shock of 0.75 (i.e. equal to a transmitted shock of 15 g in this case) and chassis vibration travel of 3.6 mm when subjected to shock input.

Acknowledgments

The authors would like to gratefully acknowledge the financial and technical support of their funding bodies.

Figure 17. Maximum deflection of the nozzle plate versus isolation damping ratio at different isolation natural frequencies.
Funding
This work was supported by the Defense Science Organization of Singapore and Temasek Laboratories at NTU (grant number TL/DSOCL/0083/02).

References