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# Effect of Source Characteristics on Structure-borne Noise in an Isolation System over Mid and High Frequencies

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## Abstract

This article presents an analytical framework in terms of a generic multi-dimensional vibration isolation system with a rigid body source, over low, mid and high frequency regimes. Continuous system models are employed for the isolator, and finite plate models are used to describe the compliant receiver. The mobility synthesis method is adopted to predict the frequency domain behavior of a linear time-invariant system. Source characteristics of the vibration isolation system have been examined over a broad frequency regime. Our analysis indicates that the constant velocity source idealization is a good approximation over mid and high frequency regimes. And, the constant force source model works well at very low frequencies. Based on the aforementioned observations, we propose a new design principle that attempts to minimize the translational free source velocity with an objective of reducing vibration power transmitted to a receiver over mid and high frequency regimes. Finally, the proposed scheme is validated through experimental and computational studies using an inverted 'L' plate receiver and alternate isolators. Measured and predicted insertion loss spectra agree well since both show that structure-borne noise transmitted to a receiver and associated radiated sound are significantly reduced when the proposed design scheme is implemented.

## 1. Introduction

An accurate characterization of the dynamic behavior of practical vibration isolation systems, over a broad range of frequencies, is needed to develop reliable simulation models, suggest passive design strategies and propose active control concepts [1]. The isolation system literature is rich but the following issues are yet to be adequately addressed. First, significant interactions occur among the source(s), path(s) (isolator) and receiver(s) [2-4], and vibration transmission takes place in a multi-dimensional manner [3-5]. Longitudinal, shear and rotational motions of the isolator are involved [5], and inertial or wave effects emerge within the isolator as the frequency of excitation increases [6-8]. Second, the source is usually an unknown combination of ideal free-velocity or blocked force source along with its own mobility [6]. Consequently, the dynamic design principles of such systems, especially at higher frequencies, remain ill-understood. In this article, we propose an analytical framework in terms of a generic multi-dimensional system with focus on source characteristics and design principle(s) for reduced vibration over mid and high frequency regimes. Given the vast scope of the underlying problem, our analysis is limited to a linear time-invariant system, frequency domain approach and components of ideal geometry or boundaries. The resulting continuous system formulation will be used to describe the isolator and receiver.

For a vibration isolation system, two idealized source models have been assumed [3, 6-7]. The blocked force source behavior may be observed for a highly rigid foundation [2, 6]. Conversely, a massive vibrating engine that is attached to a thin flexible structure may be regarded as a free velocity source [2, 6]. However, the aforementioned source idealizations have yet to be analyzed in the context of a typical multi-dimensional system. Finally, a multi-dimensional rigid body source case has not been analytically investigated even though it is common to model an engine or motor as a rigid body. See reference [3] for a detailed literature review.

Overall, the source characterization issue requires further investigation. Accordingly, our problem is formulated in Figure 1. In this multi-dimensional system, the source is described by a rigid body and continuous system models are employed for the isolator. The compliant receiver is modeled in the analytical example via a finite plate with ideal boundary conditions. In the experimental example that will be discussed in Section 4, an inverted 'L' plate structure is considered as the receiver. Harmonic force and moment excitations are applied to the source.

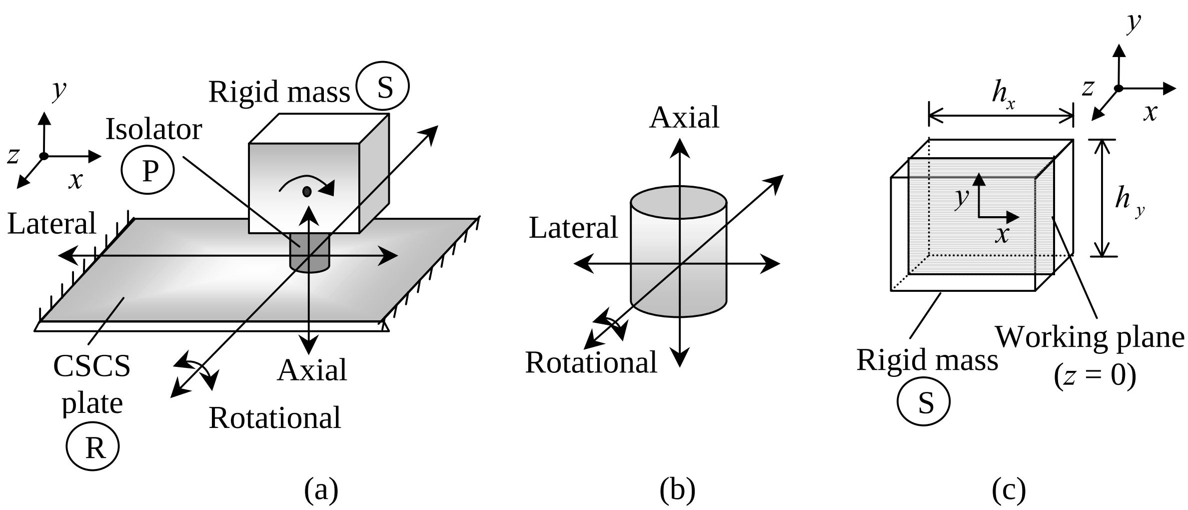


Figure 1. Configuration of the analytical vibration isolation system. (a) System with a finite plate receiver with mixed boundaries (CSCS); (b) a cylindrical isolator with vibration transmission components; (c) isolator location [ $x$, $y$, $z$ ] on the working plane of a cubic rigid body source: case 1 = [ 0, 0, 0 ]; case 2 = [ $-h_x/2$, 0, 0 ]. Here, S implies a simply supported edge and C denotes a clamped edge.

The scope of this study is limited to the analysis of a linear time-invariant (LTI) system with a single isolator (defined in terms of distributed parameter). Mobilities of each component are analytically or computationally obtained, and then the mobility synthesis method is employed to predict the harmonic response of the overall system [8]. Primary objectives of this study include: 1. Examine source characteristics of a multi-dimensional vibration isolation system; 2. Suggest an analytical procedure that can estimate free velocities of a multi-dimensional rigid body source given unknown vibration generation mechanism within the source; 3. Propose new design principle(s) in terms of the rigid body source parameters for improved isolation; 4. Validate the proposed principle by conducting experimental and computational studies on an inverted ‘L’ structure problem. Finally, we compare insertion loss spectra of vibration and structure-borne noise for an isolator as well as their placements.

## 2. Constant Velocity or Force Source Idealization

Vibration powers transmitted to a rectangular plate receiver with mixed boundaries are calculated to examine the source idealizations using the two isolator location cases of Figure 1(c). Harmonic excitation is applied at the mass center of a cubic rigid body with a mass of 1 kg and a length of 50 mm. Circular isolator is shown in Figure 1(b) along with vibration components transmitted through the path. The isolator is modeled using the Timoshenko beam theory (flexural motion) and the wave equation (longitudinal motion). Detailed mathematical treatment is given in a recent journal article we wrote [5]. The Young’s modulus, shear modulus and mass density of the rubber isolator used in this study are 16.2 MPa, 5 MPa and 1000 kg/m$^3$ respectively. Also, the circular cylindrical isolator has the length of 30 mm and the radius of 12 mm. Material properties of the receiver plate having a thickness of 5 mm, a length of 300 mm and a width of 100 mm are $6.688\times10^4$ MPa, 2723 kg/m$^3$ and 0.001 for Young’s modulus, mass density and loss factor respectively. Results of Figure 2 show that the constant velocity source ($\Pi_v$) closely approximates the exact model ($\Pi$) in terms of vibration power transmitted to the receiver especially at higher frequencies (beyond 200 Hz) for all cases. However, large discrepancies between $\Pi_f$ and $\Pi$ are seen in Figure 2. However, it is observed from Figure 2 that $\Pi_f$ is close to $\Pi$ at very low frequencies (20 Hz or less). The aforementioned observations imply that $\Pi$ is mainly dictated by the free velocity ($\mathbf{V}_{2,0}$) at higher frequencies. Refer to our article [3] for detailed analysis.

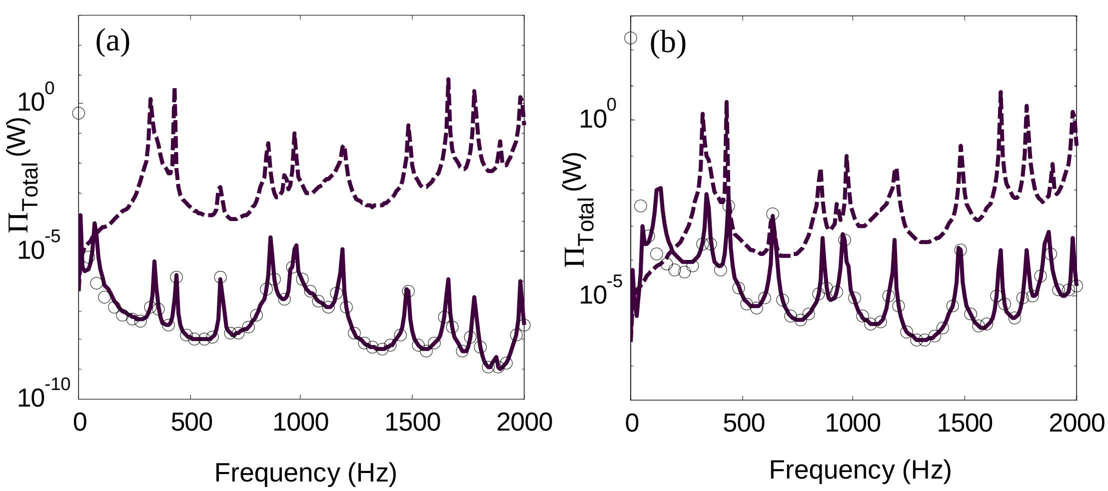


Figure 2. Total vibration power transmitted to a finite CSCS plate given moment excitation. (a) Case 1 of Figure 1(c); (b) Case 2 of Figure 1(c). Key:——— , exact; o , $\Pi_v$;--------, $\Pi_f$.

## 3. Design Principle by Minimization of Free Source Velocity

A design scheme that is based on source characteristics is proposed, with an objective of reducing the vibration power transmitted to a receiver. Recall the observations of Section 3 that illustrate that at high frequencies the transmitted power is mainly affected by the free source velocity ($\mathbf{V}_{2,0}$). Note that, given any source, $\mathbf{V}_{2,0}$ can be expressed only by the interfacial conditions between the source and isolator and the inertial property of source. Therefore, it is expected that an adjustment in the interfacial locations could alter power. For example, the rotational components ($\mathbf{w}_0$) of $\mathbf{V}_{2,0}$ are the same at all points within a rigid body source. However, the translational components ($\mathbf{v}_0$) of $\mathbf{V}_{2,0}$ depend on the locations within a rigid body. Therefore, one can locate the mount (on the source side) such that has only $\mathbf{w}_0$ with zero $\mathbf{v}_0$. The proposed design procedure is conceptually illustrated via Figure 3. Refer to our recent article [3] for details.

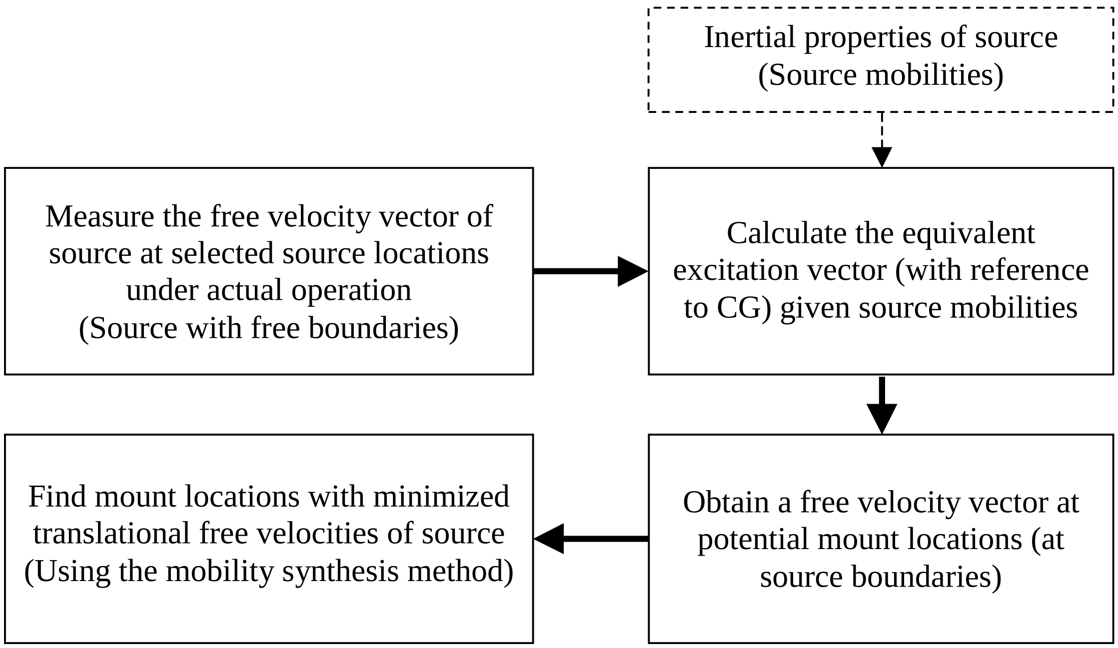


Figure 3. Proposed design scheme based on the minimization of the free source velocity.

## 4. Experimental System

An experimental system, as shown in Figure 4, is examined to investigate the source characteristics and vibration isolation schemes. An inverted 'L' plate receiver is employed to describe both in-plane and out-of plane motion transmissions to the receiver. Overall, an isolator of Figure 1(b) is experimentally studied. The dimensions of the source are 140, 64 and 47 mm and the mass is 1.2 kg. Material properties of each square receiver plate having a thickness of 1 mm, a length of 400 mm are $19.5\times10^4$ MPa, 7700 kg/m$^3$ and 0.001 for Young's modulus, mass density and loss factor respectively. The circular cylindrical isolator with a length of 30 mm and a radius of 12 mm is located at either the center or edge of mass to examine the source characteristics. However, the mount location on the receiver side is unchanged. Velocities and sound pressures at selected points on the receiver plates are examined, as summarized in Table 1. Measurement are conducted under the sine sweep excitation (up to 3 kHz). In-phase and 180º out-of-phase forces (in $y$ direction) are separately applied to the edges of the rigid source to simulate the force ($f_y$) and moment ($q_z$) excitations at $G$ respectively. Forces from two shakers and accelerations at the driving point locations are measured using two impedance heads. Plastic and steel stingers are used over low (up to 1 kHz) and high (1 – 3 kHz) frequency regimes respectively since the dynamic forces could not excite the system above 1 kHz with plastic stingers. The input forces with almost the same magnitudes and 180º (or 0º) phase difference are maintained throughout the experiments for moment excitation. Forces measured at the driving point locations are used for computational predictions.

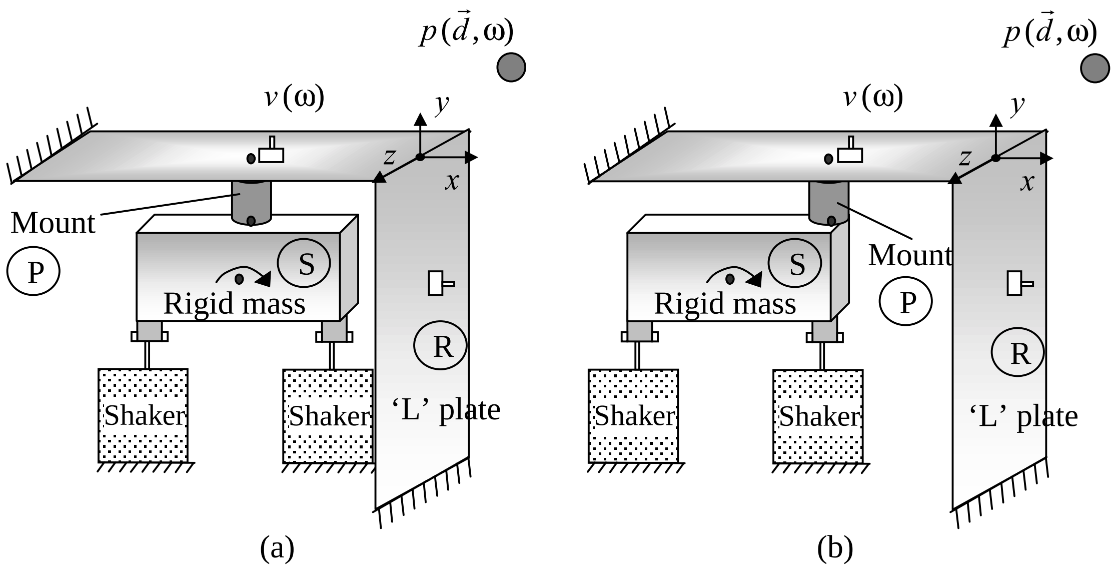


Figure 4. Experimental system with an inverted 'L' plate receiver, as excited by harmonic forces and moment. (a) System with rotational free velocity only; (b) system with translational and rotational free velocities.

Table 1. Locations of vibro-acoustic sensors

| Location No. | Measure | Coordinates (m) | 'L' structure location |
|---|---|---|---|
| 1 | Sound pressure | $x = 0.14$, $y = 0.14$, $z = 0$ | Acoustic free field |
| 2 | Sound pressure | $x = -0.18$, $y = 0.46$, $z = 0$ | Acoustic free field |
| 3 | Velocity | $x = -0.15$, $y = 0$, $z = 0$ | Horizontal plate |
| 4 | Velocity | $x = 0$, $y = -0.15$, $z = 0$ | Vertical plate |

The mobilities of the inverted 'L' plate structure are obtained by using a commercial finite element (FEA) IDEAS [9] code. Further, interfacial forces and moments between the isolator and receiver are calculated by synthesizing the mobilities of the inverted 'L' plate, source and isolator. Then, the plate velocity distribution from FEA calculation is provided to a commercial boundary element method (BEM) SYSNOISE [10] code to predict the sound radiation. Individual sound fields generated by each plate for interfacial forces and moments are superimposed to determine the resultant sound pressure. Note that direct radiation from either source or isolator is not included in such calculations. Overall, sound pressure and velocity amplitudes at locations of Table 1 are obtained using the FEA and BEM methods. Vibration power ($\Pi_{\mathrm{TR}}$) transmitted to the 'L' plate and the power ($\Pi_{\mathrm{RAD}}$) radiated to the acoustic medium from the receiver are also predicted.

# 5. Validation of Proposed Design Principle

The design principle that suggests isolator locations with zero translational free velocity is examined using the experimental system of Figure 1. Only the rotational free velocity of the source should exist for the moment excitation case when a mount is located at the center of the mass source. And, both translational and rotational free velocities occur when the isolator is placed at the edge of the rigid body source. As mentioned previously, the mount location on the receiver side is unchanged through the process. Insertion losses (*IL*) for sound pressure (*p*), vibration velocity (*v*) and acoustic power ($\Pi_{RAD}$) are calculated where

$$IL_{p_i} = 10\log_{10}\left(\frac{\Psi^2_{p_i,A}}{\Psi^2_{p_i,B}}\right), \text{ dB}; \qquad IL_{v_j} = 10\log_{10}\left(\frac{\Psi^2_{v_j,A}}{\Psi^2_{v_j,B}}\right), \text{ dB}; \tag{1a-b}$$

$$IL_{\Pi_{RAD}} = 10\log_{10}\left(\frac{\Pi_{RAD,A}}{\Pi_{RAD,B}}\right), \text{ dB}. \tag{1c}$$

Here, $p_i$ and $v_j$ are sound pressure at acoustic field point *i* and velocity at receiver structure location *j* respectively.

The $IL_p$ and $IL_v$ spectra are obtained from both experimental and computational studies but only the computed results are used for $IL_{\Pi_{RAD}}$. Experimental and computation results at the response locations of Table 1 are shown in Figure 5. Spectra are given at the center frequencies of the 1/3 octave band. Resulting vibration and acoustic measures can not be normalized with respect to their excitation forces since two different input forces are used. However, it is observed that measured forces from the shaker stingers to a mass source do not vary much given different system configurations. Both computational and experimental results of Figure 5 show that all vibration and acoustic measures are significantly reduced when the polypropylene isolator is moved from the edge to the mass center. However, the experiment results of $IL_p$ do not exhibit as much reduction as the ones computed beyond 500 Hz. This is because the actual sound radiated from the receiver is lower than shaker noise beyond 500 Hz, especially when the isolator is located at the center of source. See Figure 5(b) where the background noise from shakers is also shown with mean-square sound pressure ($\Psi^2_p$). Note that measured $\Psi^2_p$ of the system shows almost the same level as $\Psi^2_p$ of shakers as shown in Figures 5(b). Further, note that $IL_v$ spectra, that are not contaminated by shaker noise, are much higher than $IL_p$ beyond 500 Hz as shown in Figure 5(e-f). Both computational and experimental results support the proposed design principle, especially at higher $\omega$. Further, the proposed scheme also reduces transmission at lower $\omega$ even though the actual source behavior deviates slightly from the constant velocity assumption. Overall, reasonable agreements between computed and experimental results are observed even though some measurements are contaminated by the shaker noise. Refer to our article [3] for further results on other isolators.

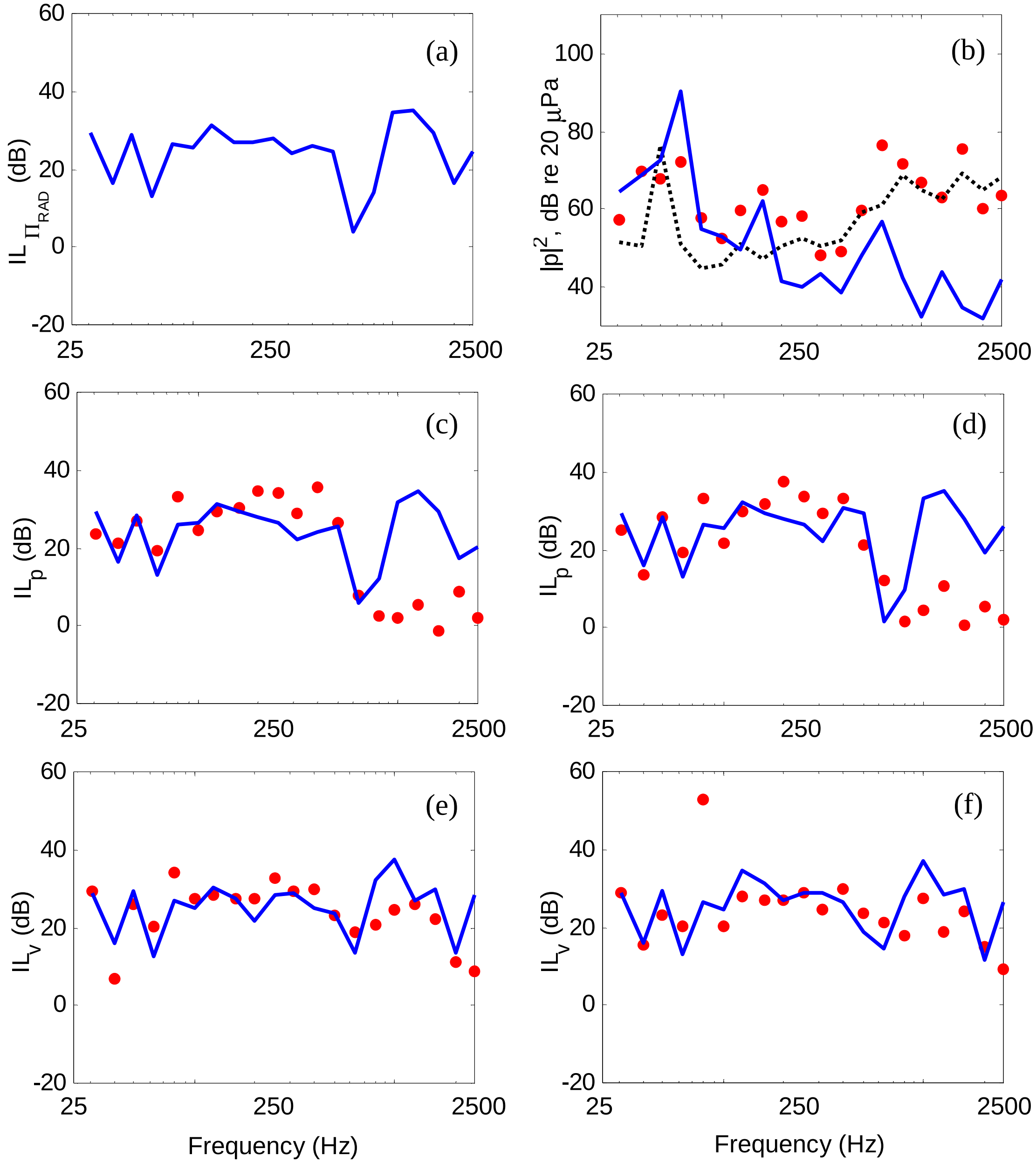


Figure 5. Vibration isolation measures on mount location for a polypropylene isolator given moment excitation. (a) Insertion loss ( $IL_{\Pi_{RAD}}$ ) of acoustic power radiated from the 'L' plate receiver; (b) absolute sound pressure level at location 1 of Table 1 with mount configuration of Figure 4(a); (c) insertion loss ( $IL_p$ ) of sound pressure at field location 1; (d) $IL_p$ at location 2; (e) insertion loss ( $IL_v$ ) of velocity at plate location 3; (f) $IL_v$ at location 4. Key: ———, calculated; ●, measured; ............, background noise from shakers. Results are given in terms of 1/3 octave band center frequencies from 31.5 to 2500 Hz. Only the mean values within each bandwidth are plotted here.

# 6. Conclusion

Based on a new analytical framework, important source characteristics of a multi-dimensional vibration isolation system have been examined over a broad frequency regime. Our analysis illustrates that the constant velocity source idealization is a good approximation over mid and high frequency regimes. A new design principle has been proposed for reducing vibration and structure-borne noise transmitted to a receiver structure over mid and high frequency regimes. This strategy eliminates the translational free velocities of a rigid body source of dimension 6. The proposed design scheme has been validated through experimental and computational studies using an 'L' plate receiver. Measured and predicted insertion loss spectra show that vibration and structure-borne noise transmitted to a receiver is indeed significantly reduced when the chosen isolator location minimizes the translational free velocity of the source. Future work is required to extend the proposed design principle to "real life" systems such as ground vehicles. Further analysis must also incorporate flexibility within the source structure. Also, a proper interpretation of competing measures of isolation performance requires a more critical examination in the context of design work. Related experimental studies are also needed to further clarify the high frequency issues.

# Acknowledgements

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