24th INTERNATIONAL CONGRESS ON SOUND AND VIBRATION
23–27 July 2017, London

STRUCTURE DESIGN TO REDUCE WEIGHT AND VIBRATION BY USING STATISTICAL ENERGY ANALYSIS

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This paper describes a new way to reduce simultaneously the weight of a structure and the sound borne by it under real operating conditions. This can be accomplished by combining ex-perimental statistical energy analysis (ESEA) and a new method for reducing both vibration and weight. The general process of reducing vibration and noise involves preparing a numerical or experimental model of the vibration and noise characteristics of the system, identifying the external forces acting on the model during operation, and determining the necessary countermeasures by using the model and the input powers. Although it is difficult to implement this process by using conventional methods such as modal experimental analysis or the finite- or boundary-element method, ESEA allows for its successful implementation. The proposed process consists of four steps: 1) ESEA model construction, 2) identification of power inputs in real operation, 3) identification of ESEA parameters to change for reducing vibration in the target subsystem and reducing the total weight, and 4) determination of structural modifications to reduce the vibration and weight by realizing the desired ESEA parameters change. This paper proposes a new method for identifying the appropriate ESEA parameters in step 3). The proposed method and process are used to reduce the vibration of the power control unit in a hybrid electric vehicle. It is demonstrated that a countermeasure based on the proposed process reduces the vibration by 5 dB and the weight by 2 %.
Keywords: vibro-acoustics, statistical energy analysis, low vibration, weight saving, hybrid vehicle

1. Introduction

The quietness of automotive vehicles is an important issue, especially from a commercial perspective. High-frequency electromagnetic noise is now a problem because of recent advances in vehicle electrification, and countermeasures are being sought urgently. The traditional approach to reducing noise and vibration is to focus on natural frequencies and modes. That approach is very effective in cases in which the modal density of the product is low and the resonances are clear. However, in cases with numerous high-frequency natural modes, it is difficult to know which natural frequency is dominant and hence which countermeasures should be deployed. Methods such as statistical energy analysis (SEA) [1,2] and structural intensity (SI) [3], which are based on vibration energy propagation, are useful for solving such problems.

It is generally the case that countermeasures against noise and vibration require an increase in mass. However, in a vehicular context, such an approach is at odds with the need to save energy by reducing the mass. Against this background, we focus on the use of SEA and SI to predict vibrations and to design low-vibration structures. In Ref. 4, one of the present authors (T.Y.) proposed a process known as experimental SEA (ESEA) (4) for reducing structure-borne sound, and ESEA has been used to reduce the noise and vibration of various mechanical structures.

In this paper, we propose a new ESEA-based method for reducing both vibration and weight simultaneously by identifying the SEA loss factors (LFs) of a countermeasure. We begin by summarizing the ESEA process and explaining the proposed new methodology for reducing vibration and weight. We then use the proposed method in combination with ESEA to reduce the vibration and overall weight of the power control unit (PCU) of a hybrid electric vehicle by 5 dB and 2 %, respectively.

2. Structural Design Process Based on Experimental SEA

In this section, we summarize the process for reducing vibration and noise by using ESEA (4).

2.1 SEA model

In SEA, the system is regarded as an assembly of subsystems, the i-th of which has energy E_i. The external power input to this subsystem is denoted by P_i. By considering the power balance, we arrive at an SEA matrix equation of the form

$$\begin{Bmatrix} P_1 \\ \vdots \\ \vdots \\ P_N \end{Bmatrix} = \begin{bmatrix} \eta_1 + \sum_{i \neq 1}^{N} \eta_{1i} & -\eta_{21} & \cdots & -\eta_{N1} \\ -\eta_{12} & \ddots & & \vdots \\ \vdots & & \ddots & \vdots \\ -\eta_{1N} & \cdots & \cdots & \eta_N + \sum_{i \neq r}^{N} \eta_{Ni} \end{bmatrix} \begin{Bmatrix} E_1 \\ \vdots \\ \vdots \\ E_N \end{Bmatrix} \text{ or } \mathbf{P} = \omega \mathbf{L} \mathbf{E}. \qquad (1), (2)$$

Here, **P** is the vector of external power inputs, **E** is the vector of subsystem energies, and ω is the central angular frequency of the frequency band. Furthermore, **L** is the matrix of LFs, which is composed of the internal LF (ILF) η_i of each subsystem i and each coupling LF (CLF) η_{ij} between subsystems i and j. The CLFs are estimated theoretically in SEA (1,2), experimentally in ESEA (3,4), and numerically in SEA combined with finite-element modelling. Estimating the ILFs and CLFs constitutes constructing the SEA model.

2.2 ESEA-based structural design process for reducing vibration

An advantage of ESEA is that it allows us to identify the inputs during actual operation and to clarify the propagation paths. Also, it is feasible to integrate SEA with structural design because the ILFs (the damping in each subsystem) and CLFs (the ease of transmission of vibrations between subsystems) have physical meanings that direct us to the appropriate countermeasures. Our proposed ESEA-based structural design process that developed from the aforementioned considerations is summarized as follows; the details of each step are given in the following subsections.

Step 1) Construct an ESEA model for the target system

Step 2) Identify the external operational power input to each subsystem

Step 3) Specify the LFs to be changed for vibration reduction

Step 4) Design the structure appropriate to the desired LFs

2.2.1 Step 1) Constructing an ESEA model

In ESEA modelling, the target mechanical system is subdivided virtually into multiple subsystems. The equations for evaluating the CLFs and ILFs from data obtained from a hammering test are given by

$$\eta_{ij} = \frac{E_{ji}/P_i}{\omega E_{ii}/P_i \cdot E_{jj}/P_j} \text{ and } \eta_i = \frac{1-\omega \sum_{j \neq i}^{n} \left(\eta_{ij} E_{ii}/P_i - \eta_{ji} E_{ji}/P_i \right)}{\omega E_{ii}/P_i}. \tag{3), (4}$$

Here, E_{ij} is the energy of subsystem i under power input P_j to subsystem j.

2.2.2 Step 2) Identifying the external power input

It is very important to identify the external power inputs to the subsystems during real operation because the resulting countermeasures are strongly dependent on them. The SEA methodology allows the operational power input **P** to be identified easily from Eq. (1) or (2) by measuring only the subsystem energy **E** if the ESEA model **L** is constructed.

2.2.3 Step 3) Specifying the loss factors

The SEA methodology makes it easy to understand which SEA parameters affect the responses and how they do so. It also allows low-cost calculation by having fewer degrees of freedom in comparison with the conventional methods. By using approaches such as sensitivity analysis and the perturbation method, the SEA model and the external power inputs yield the sensitivities of the subsystem energies to each LF.

An LF η_n can be expressed in terms of a perturbation α_n by using the deterministic (expected) value $\overline{\eta}_n$ of the LF without perturbations:

$$\eta_n = \overline{\eta}_n \left(1 + \alpha_n\right). \tag{5}$$

If we assume that the external power input **P** in Eq. (2) remains unchanged when the structure is modified, Eq. (5) indicates that the LF matrix **L** and the subsystem energy matrix **E** in Eq. (2) can be approximated with a first-order Taylor expansion. Thus, the sensitivity of the subsystem energy matrix **E** to each LF can be derived as

$$\frac{\partial \mathbf{E}}{\partial a_n} = -\overline{\mathbf{L}}^{-1} \frac{\partial \mathbf{L}}{\partial a_n} \overline{\mathbf{E}}. \tag{6}$$

The sensitivities of the subsystem energies indicate which LFs exert the greatest influence on the system and which should therefore be modified to obtain the most efficient low-vibration structural design.

2.2.4 Step 4) Structural design for desired loss factors

Based on the LFs identified in step 3), we seek actual countermeasures corresponding to the LF changes. An ILF increment equates to increased damping such as the attachment of damping sheets. A CLF is modified by changing the subsystem configuration or the coupling conditions. For example, the CLF between two plate subsystems is formulated theoretically as

$$\eta_{ij} = \frac{2L_{ij}\tau_{ij}}{\pi S_i} \sqrt{\frac{h_i}{\omega}} \sqrt[4]{\frac{E_i}{12\rho_i (1-\nu_i^2)}}, \tag{7}$$

where L_{ij} is the coupling length between subsystems i and j, τ_{ij} is the average energy transmissibility, h_i, S_i, E_i, ρ_i, and ν_i are the plate thickness, surface area, Young's modulus, material density, and Poisson's ratio of subsystem i, respectively. Equation (7) suggests that reducing the material density would increase the CLF, whereas using a material with a small Young's modulus would decrease the CLF. In this paper, we use Eq. (7) to propose a method for reducing both vibration and weight.

3. New Method for Reducing Both Vibration and Weight

In this section, we introduce our new method for reducing both vibration and weight, and we incorporate it into step 3) from Section 2.2.3. The main feature is the consideration of subsystems whose energy could be increased. The sensitivities of the subsystem energies to be reduced or increased are considered simultaneously, and then the CLF that reduces both vibration and weight is extracted to be changed. We focus on the material density and Young's modulus in Eq. (7), and we assume that reducing these values corresponds to reducing the weight.

3.1 Development of the method

The sensitivities of the subsystem energies to be either reduced or increased can be obtained according to Section 2.2.3 with Eq. (6). We begin by considering a subsystem whose energy is to be reduced. A positive sensitivity requires a reduction in CLF, which from Eq. (7) means either increasing the density or decreasing the Young's modulus; the opposite is required to increase the subsystem energy. As such, reducing both vibration and weight can be classified as follows.

(α) Only reducing subsystem energy
A positive sensitivity requires a smaller Young's modulus. A negative sensitivity requires a lower density.

(β) Only increasing subsystem energy
A positive sensitivity requires a lower density. A negative sensitivity requires a smaller Young's modulus.

(γ) Reducing some subsystem energies by (α) and increasing some others by (β)
The four combinations that facilitate simultaneously reduced and increased subsystem energies are listed in Table 1.

- Sets A and B
  The sensitivities for reducing and increasing the subsystem energies have the same sign; hence, there is no solution that reduces both vibration and weight.
- Sets C and D
  The sensitivities now have opposite signs; hence, a solution exists. For set C, the solution is to decrease the Young's modulus (i.e. the stiffness), whereas for set D it is to decrease the density (i.e. the mass).

In this paper, we apply the proposed method with ESEA based on set D to reduce both the vibration and weight of an automotive PCU.

Table 1: Sensitivities of subsystem energies and countermeasure to reduce vibration and weight

Set	Sensitivity of subsystem energy to be reduced	Sensitivity of subsystem energy to be increased	Countermeasure to reduce vibration and weight
A	+	+	No solution
B	−	−	No solution
C	+	−	Stiffness decrement
D	−	+	Mass decrement

4. Application to Automotive Power Control Unit

In this section, we verify the method by using it to reduce the vibration and weight of the PCU of a hybrid vehicle.

4.1 Target PCU

The target PCU consists of a lower cover, an inverter case, and a converter case on which there is a reactor that is the only source of vibrations. The driving-current frequency of the reactor can be arbitrary set; the frequency is set constant by each vehicle model.

4.2 Step 1) Constructing the ESEA model

We subdivide the target PCU into 10 virtual subsystems as shown by the thick lines in Fig. 1; the thin lines represent the subsystem network and the subsystems are numbered as shown. Subsystem 1 is the upper surface of the inverter case. Subsystems 2 and 3 are the right- and left-hand halves, respectively, of the converter case. Subsystems 4, 5, and 6 are the parts for mounting the PCU on the vehicle body. Subsystems 7–10 are the side surfaces of the inverter case. Here, we target the energy of subsystem 1 for reduction.

We set the frequency range for the ESEA as 100 Hz to 20 kHz in 1/3 octave bands. All measurements were carried out with the four edges of the PCU mounted on rubber blocks.

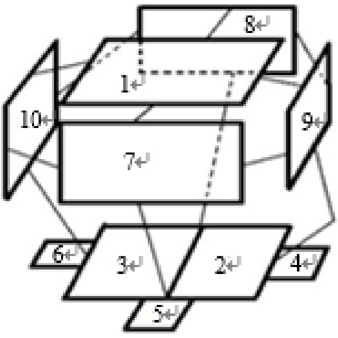


Figure 1: Subdivision and subsystem network of the test PCU with 10 subsystems

We constructed the ESEA model according to Section 2.2.1 and Ref. 4, and we used accelerometers to access the out-of-plane vibration of each subsystem. We used four measurement points each to evaluate the energies of subsystems 1, 2, and 3, three for subsystem 3, one each for subsystems 4, 5, and 6, and five each for subsystems 7–10. An impulse hammer was used to excite vibrations in each subsystem, and the force and acceleration near the input point were measured to evaluate the input power.

To assess the accuracy of the constructed ESEA model, Fig. 2 shows a comparison between the predicted and measured energies of subsystems 2 and 9 for an impulsive power input to subsystem 2; all energy values are normalized with respect to the input power. From this, we find that the model is accurate to within 3 dB between 800 Hz and 16 kHz.

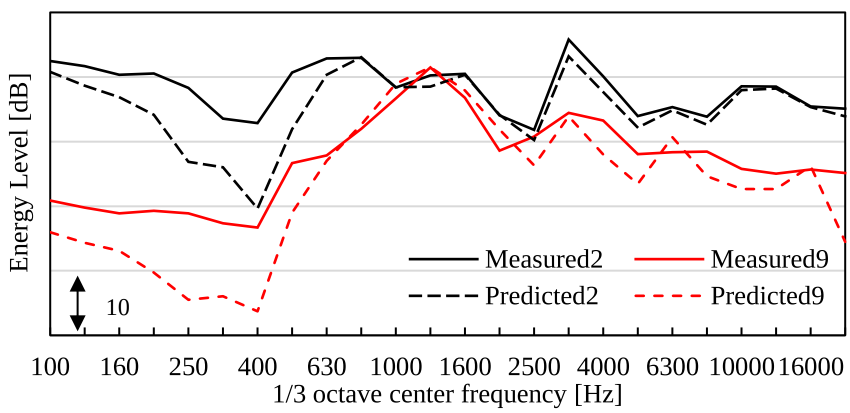


Figure 2: Comparison of energy levels of subsystems 2 (black) and 9 (red) under power input to subsystem 2: measurements (solid), ESEA predictions (dashed)

4.3 Step 2) Identifying the external power input

Based on Section 2.2.2 and Ref. 4, we identified the input power in only the case of the reactor operating at a driving-current frequency of 12.5 kHz. The subsystem energies were measured at the same points used to construct the ESEA model. The identified power inputs are shown in Fig. 3, where Pin1–Pin3 indicate the power inputs to subsystems 1–3, respectively. The input power to subsystem 2 (Pin2) dominates, with the largest input value at the 12.5 kHz band. This agrees with the facts that the reactor is installed on subsystem 2 and the driving-current frequency is 12.5 kHz.

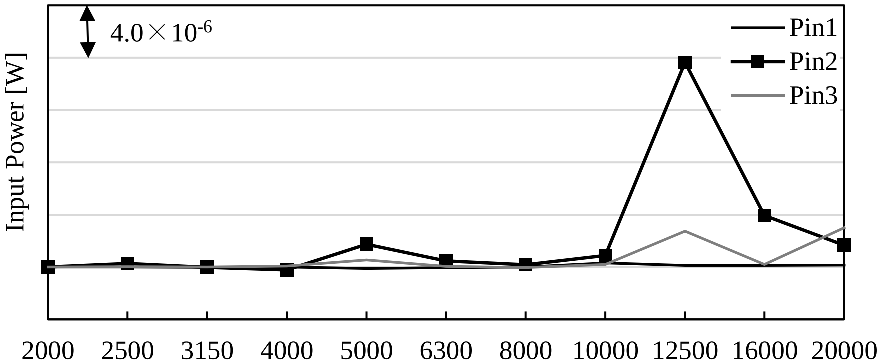
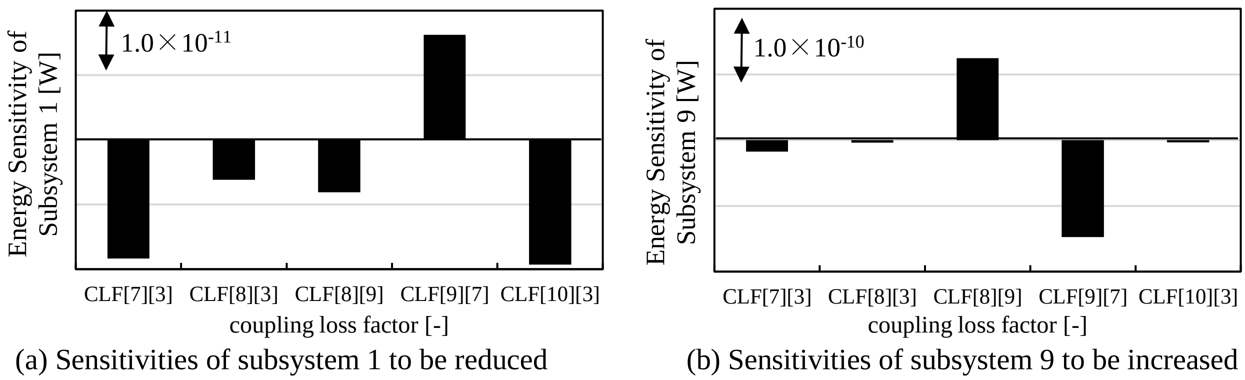


Figure 3: Identified power input during operation

4.4 Step 3) Specifying the loss factors

We combine the new method proposed in Section 3 with the ESEA process to reduce both vibration and weight. As discussed above, subsystem 1 is targeted for energy reduction and subsystem 9 is targeted for energy increase.

Following Section 2.2.3, the energy sensitivities of subsystems 1 and 9 were calculated and are presented in Fig. 4; the sensitivities are largest in the 12.5 kHz band. For the CLF from subsystem 7 to subsystem 3 (CLF[7][3], η_{73}), both sensitivities shown in Fig. 4 are negative, from which we judge there to be no solution (Table 1, Set B). For CLF[8][9] (η_{89}) and CLF[9][7] (η_{97}), the sensitivities shown in Fig. 4 have different signs. Reducing the mass of subsystem 8 corresponds to CLF[8][9] (η_{89}) (Table 1, Set D) and reducing the stiffness of subsystem 9 corresponds to CLF[9][7] (η_{97}) (Table 1, Set C).



(a) Sensitivities of subsystem 1 to be reduced (b) Sensitivities of subsystem 9 to be increased

Figure 4: Energy sensitivities of subsystems 1 and 9 for several CLFs at the 12.5 kHz in 1/3 octave band

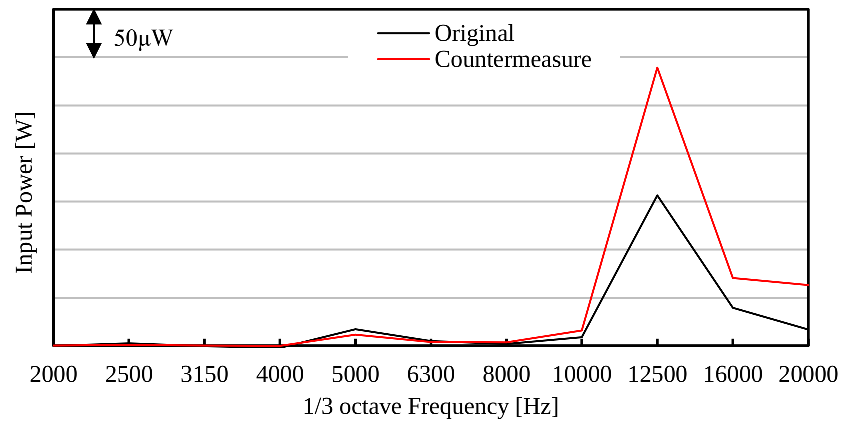
4.5 Step 4) Designing structure appropriate to desired loss factors

Based on the guidelines from the previous section, we focused on CLF[8][9] (η_{89}) by reducing the mass of subsystem 8. We did this by removing some ribs from the subsystem, thereby reducing the mass by 2 % at part. By comparing the frequency response functions of subsystem 8 before and after rib removal (except for the lowest resonance), we found the associated change in rigidity to be negligibly small.

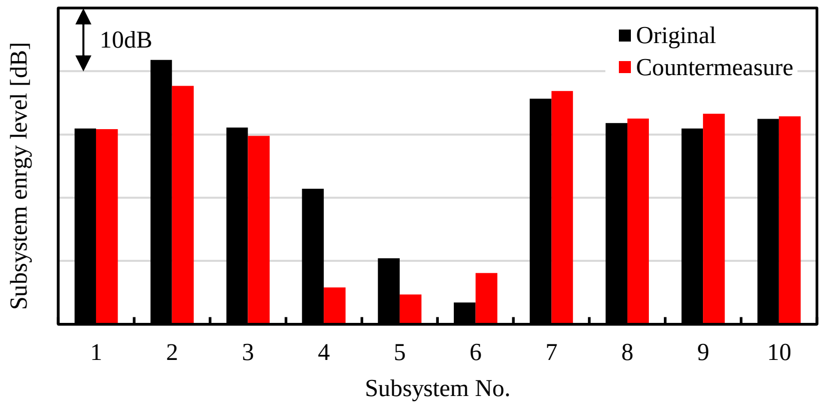
4.6 Confirmation of the countermeasure in laboratory tests

We conducted a series of laboratory tests on the PCU to measure the input power, subsystem energy, and CLF[8][9] (η_{89}) before and after removing ribs from subsystem 8. From Section 2.2.3, we assume that the input power remains constant, but in practice it depends on the structural design. Figure 5(a) shows the input power into subsystem 2 before and after the modification, where it can be seen that the input power at 12.5 kHz nearly doubled.

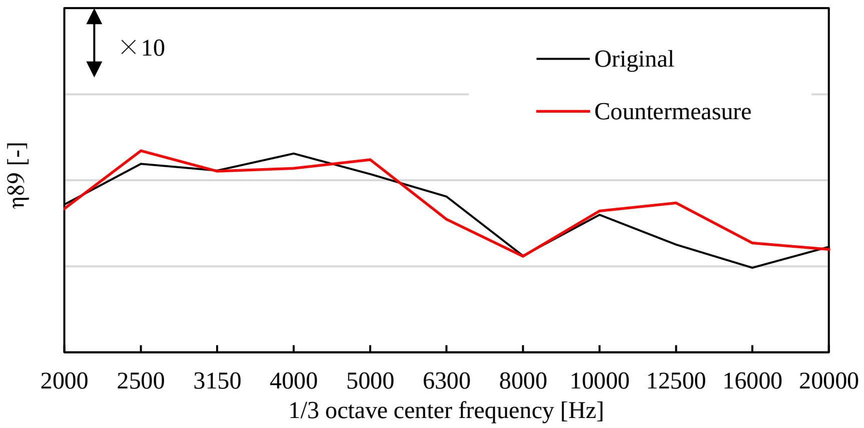
Figure 5(b) allows comparison of the energies of all subsystems at the targeted frequency of 12.5 kHz. The energy of subsystem 1 (which we seek to reduce) can be reduced by around 0.1 dB, whereas the energy of subsystem 9 (which we seek to increase) can be increased by around 2.3 dB.



(a) Input power to subsystem 2



(b) Energies of subsystems 1 to 10 at the 12.5 kHz in 1/3 octave band



(c) Coupling loss factor CLF[8][9] (η_{89})

Figure 5: Comparisons with and without countermeasure: laboratory tests

Figure 5(c) allows comparison of the CLF[8][9] (η_{89}), in which there is a relatively large increase at 12.5 kHz as desired. We note the existence of frequency-dependent trade-off relationships before and after modification; however, the difference is largest at 12.5 kHz. From this, we conclude that the proposed method succeeded in reducing the vibrations of the targeted subsystem while reducing the overall weight by 2 %.

4.7 Confirmation of the countermeasure in bench tests

The vibrations of the PCU before and after modification were also measured in bench tests. The measurement points were the same as those in the laboratory tests, but the driving current was formed instead by combining currents with frequencies of 2.5 kHz and 9.55 kHz.

Figure 6 allows comparison of the energy of subsystem 1 before and after modification. The energy was reduced by around 0.6 dB at the frequency of 12.5 kHz targeted in the laboratory tests. Moreover, the vibration energies at 3.15 kHz and 5 kHz were reduced by around 3 dB, whereas that at 10 kHz was reduced by around 0.2 dB. From this, we conclude that the countermeasure derived by the proposed method also worked well in the bench tests.

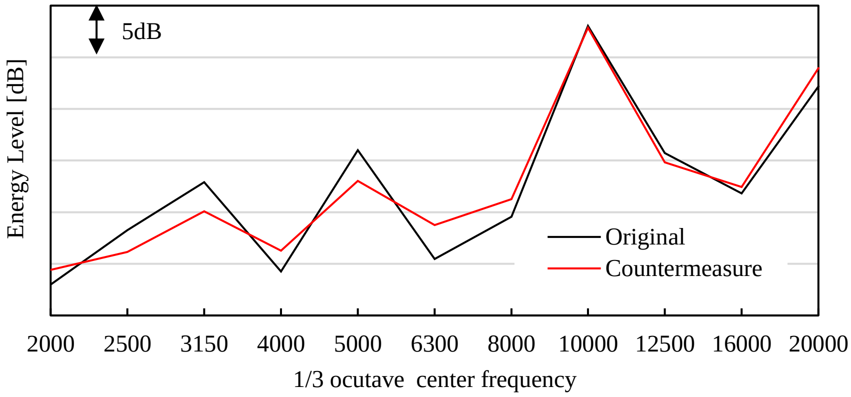


Figure 6: Comparison of energy of subsystem 1 with and without countermeasure: bench tests

5. Conclusions

This paper has proposed a method for reducing structural vibration and weight simultaneously by using the ESEA process. We conclude with the following specific remarks.

(1) We summarized a structural design process for reducing structure-borne sound, and extended it to reduce both vibration and weight simultaneously.

(2) We applied the proposed method to the PCU of a hybrid automotive vehicle and succeeded in reducing its vibration and weight. From this, we verified that the proposed method works well.

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