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NOISE PATH IDENTIFICATION USING POWER FLOW ANALYSIS

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ABSTRACT
Transfer path analysis using vibroacoustic reciprocity has generally been used for structure-borne noise path analysis in a car noise engineering. This methods are useful in solving particular problem but do not quantify the effectiveness of vibration isolation of each isolator of a vehicle. To quantify the effectiveness of vibration isolation, the vibrational power flow has been used for a simple isolation system. It is often difficult to apply the vibrational power flow technique to the complex isolation system like a car. In this paper, a simple equation is derived for calculation of the vibrational power flow of an isolation system with multiple isolators such as a car. According to the results, the main contributor of eighteen isolators is the rear roll mount of an engine. The reduced structure-borne noise level is about 5dBA.

1 - INTRODUCTION
Compliant vibration isolators are often used to reduce the transmission of unwanted vibration from an engine to seating structures. However, despite isolation, vibration transmission problems in automotive engineering sometimes still remain particularly in complex installations with more stringent requirement for vibration and noise levels, combined with a trend to lighter car bodies and more power engine. Especially, in a competitive automotive market there is obviously the need to reduce the noise level in the compartment to the lowest possible level within a given cost constraint. Such a procedure would sensibly involve some sort of structure-borne transmission path analysis to assess the relative noise contributions. This can be done at present using reciprocity technique [1]. These methods are useful in solving particular problem but do not quantify the effectiveness of vibration isolation of each isolator of a vehicle. Pinnington [2] did a good job in explaining the vibrational power flow technique as a method to quantify the effectiveness of vibration isolator and it application in a mechanical system. In this paper, a simple equation is derived for calculation of the vibrational power for the isolator system with multiple isolators like car. It is also successfully applied to identifying the major isolator contributing to the increase of structure-borne noise in a passenger car. Eighteen isolators of a test vehicle are considered as vibration transmission paths which are the engine mount, the gearbox mount, the exhaust mount, etc. By the evaluation of the vibrational power flow of each isolator, it is found that the real roll mount of the engine is a major path and its reduced stiffness results in the reduction of the booming noise up to 5dBA around 1800 rpm.

2 - THEORY OF VIBRATIONAL POWER FLOW
Power is the rate at which work is done and is given by the relationship [2]

$$P_i = F_i V_i \tag{1}$$

where F_i and V_i are the instantaneous values of force and velocity at a point. When vibrational power flows through an isolator it is necessary to consider it as an intensity and therefore with the force F_i determined as a stress. With a vibrating structure the time averaged vibrational flow of power is more important than the instantaneous value and when force and velocity are harmonic this is given by

$$P = \frac{1}{T} \int_0^T F_i V_i dt \tag{2}$$

A force with a harmonic time dependence $F_i = \tilde{F} e^{j\omega t}$ acting at a point on a structure of mobility $\tilde{M} = |M| e^{j\phi}$ causes a velocity of $V_i = \tilde{V} e^{j\omega t}$ at that point, where $\tilde{F}$ and $\tilde{V}$ are complex. The time averaged vibrational power is given by $P = \frac{1}{2} |F| |V| \cos\phi$, or

$$P = \frac{1}{2} \mathrm{Re} \left\{ \tilde{F}^* \tilde{V} \right\} = \frac{1}{2} \mathrm{Re} \left\{ \tilde{F} \tilde{V}^* \right\} \tag{3}$$

where ϕ is the relative phase angle and * denote the complex conjugate. When the acceleration on a structure is used instead of a velocity, the time averaged power, Eq. (3) is rewritten as follows [3]:

$$P_i \cong \frac{1}{2\omega} \mathrm{Im} \left\{ \tilde{M}_{sr,ii} \tilde{a}_{s,i} \tilde{a}^*_{r,i} \right\} \tag{4}$$

In principle this method is the most useful as only the isolator properties need be known, and $\tilde{M}_{rs,ii} \cong \frac{K_i}{\omega^2} (1 + j\eta_i)$ for a isolator of stiffness K_i. To find the total power from n isolator the contribution of each isolator calculated from Eq. (4) would have to be summed.

3 - REDUCTION OF NOISE IN COMPARTMENT

A test vehicle is equipped with a 2.0L 4-cylinder engine and automatic transmission. Aim of this paper is to determine and improve the contribution of different transmission paths in the structure energy flow from engine vibration to the car's interior acoustic pressure for a low frequency booming problem. The structure transmission paths under investigation can be divided in two main groups. The first group includes the direct transmission ways from the engine to the body. This yields transmission through the engine, the gearbox and the exhaust mounts. The indirect transmission can also be very important. Through the driveshaft the wheels are excited by the engine. Over the suspension triangles and shock absorbers the vibration energy from the wheels is then transmitted to the body. By a baseline test, this vehicle has the interior noise booming in the back of car around 1800 rpm and 2400rpm. It can be clearly felt when driving the car on the public road or on the rolling road in the laboratory. The frequency is the 2^{nd} order component of engine speed. At the first status of vehicle, the influence of boundary conditions on noise levels and frequency content of the interior noise and vibration has been evaluated in order to determine whether the interior noise is airborne or structure borne. This evaluation shows that the interior booming noise problem is not an airborne problem, but caused by structure-borne phenomena. Therefore, a structure borne transmission path analysis is carried out in order to the phenomena causing this interior noise booming. The time averaged vibrational power flows through 18 isolators in a test car are measured using Eq.(1,2) and the complex dynamic stiffness is also measured on the test rig which is particularly designed to measure the stiffness of automobile isolators. The isolator has not a resonance up to 120Hz. Fig. 1 shows the measured vibrational power flow through vibration isolators 1,2,3 and 4, which denote the engine mount, the front roll mount and the real roll mount of the engine and the gearbox mounts respectively. The solid line denotes the positive vibrational power flow and the dot line denotes the negative vibrational power flow.

The vibrational power flows through these four isolators are greatly higher than that through other fourteen isolators as shown in Fig. 2. Fig. 2 explains the contribution of power flows through isolators around 1800 rpm. According to this result, it is concluded that two roll mounts of the engine are important contributors to the booming noise around 1800rpm. The rear roll mount of the engine together with the engine mount of the engine is also contributor around 2400rpm. The gearbox mount plays a role as a sink of power flow because the vibrational power flow is negative. From Fig. 2 it can also be understand that the front bush mount of center member of the isolator 6 is also important a vibration transmission path. Therefore, in order to reduce the interior booming noise around 1800 rpm, the rear roll mount of the engine or the front bush mount of center member should be modified.

In this vehicle, there is some constrain for modification of front bush mount. Thus the complex dynamic stiffness of the rear roll mount is reduced up 15%. This modification yields the reduction of interior noise level up to 5dBA around 1800 rpm as shown in Fig. 3.

4 - CONCLUSION

Identification of the structure-borne transmission path of energy flow from the vibration of the engine to the acoustic pressure in the compartment is useful for reduction of interior noise. The estimation of the vibrational power flow through each transmission path is a useful tool to evaluate the effectiveness of

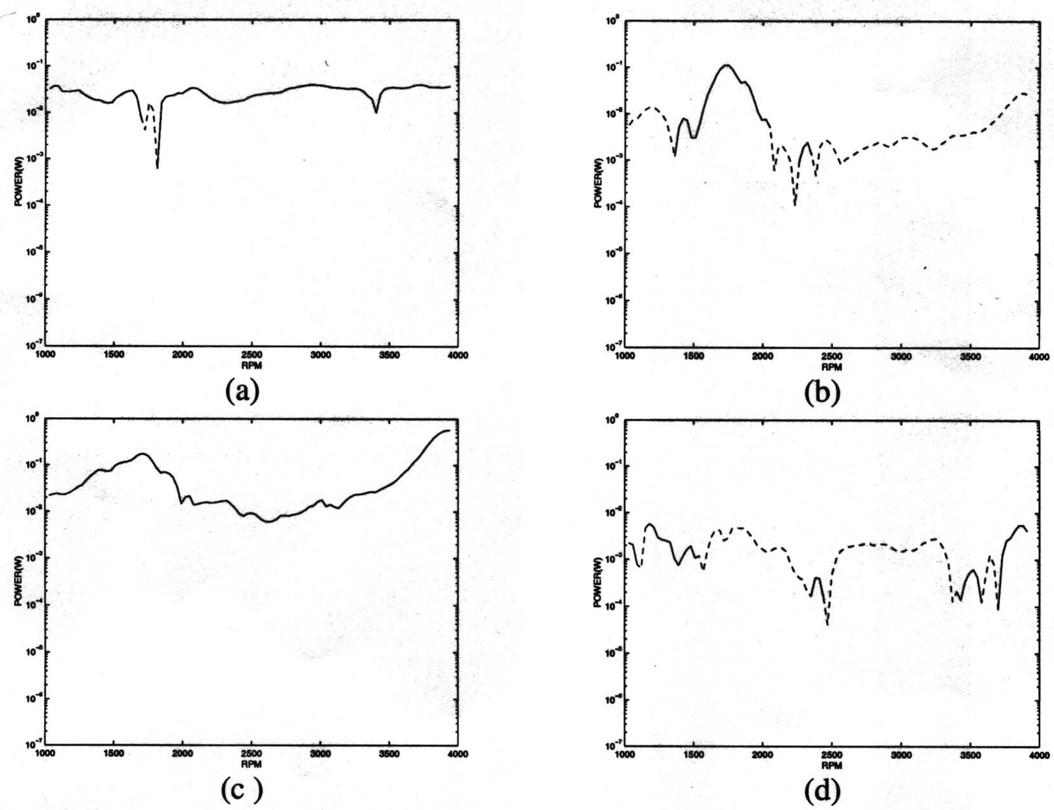

Figure 1: Measurement of vibrational power flow through each mount (a) the engine mount (b) the front roll mount (c) the rear roll mount (d) the gear box mount; the positive vibrational power flow; the negative vibrational power flow.

vibration isolation of isolators. In this paper, a test vehicle equipped with a 2.0L 4-cylinder engine and automatic transmission is used. The vibrational power flow through eighteen isolators of the test vehicle has been measured. From results, it concluded that the vibrational power from the engine is mainly transmitted to the car structure through the rear roll mount of the engine. The dynamic stiffness of the rear roll mount is reduced by 15%. This modification yields the reduction of interior noise level up to 5dBA around 1800 rpm.

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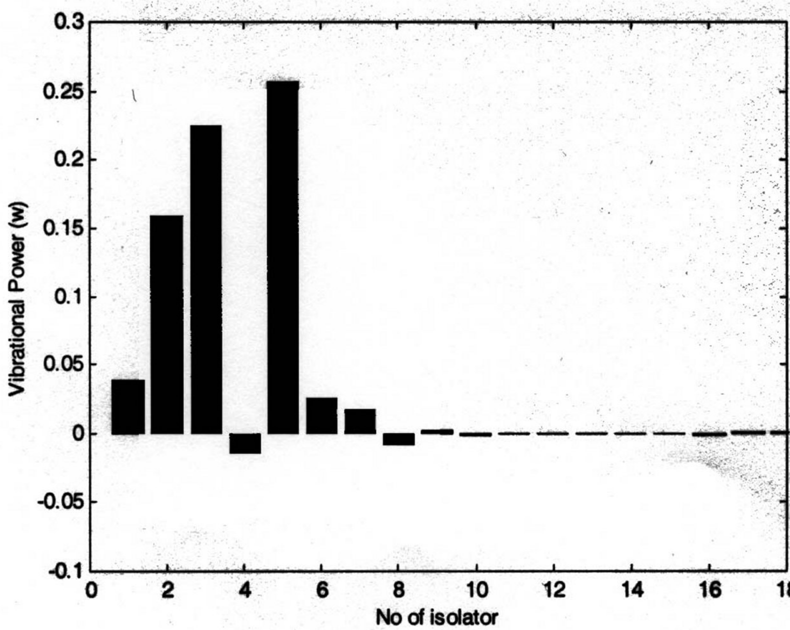

Figure 2: Comparison between the vibrational power flow through eighteen isolators of the test vehicle.

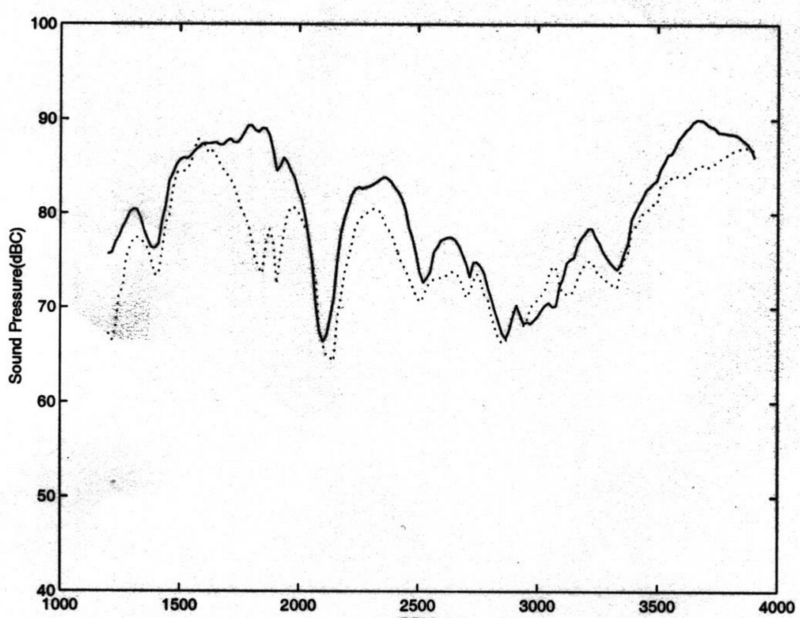

Figure 3: Illustration of the booming noise reduction by the modification of the rear roll mount of the engine; continuous curve: original; dashed curve: modification.