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SOUND TRANSMISSION THROUGH MOTOR VEHICLE STRUCTURES USING STATISTICAL ENERGY ANALYSIS

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1. INTRODUCTION

Noise and vibration transmission in motor vehicles is difficult to study due to the complexity of the vehicle structure and noise sources. Many noise sources can cause problems at higher frequencies, say above 250 Hz. At these frequencies Statistical Energy Analysis (SEA) can be used to study noise transmission. In this work SEA modelling is discussed. Ways of calculating power input to the vehicle are shown and measured and predicted sound pressure levels for the engine running are compared.

The aim of the work is to find a simple method of estimating the power input to the vehicle and subsequently predict the sound pressure level in the saloon.

2. OUTLINE THEORY

The motor vehicle used in this work is the same as was used by Steel [1]. The vehicle is subdivided into 83 subsystems plus 3 compartments associated with the boot, saloon and engine. The same structural model described by Steel [1] is used here but coupling to the compartments has been added. Energy levels are calculated for a given set of power inputs using the standard SEA energy balance equation [2],

$$\frac{[P]}{\omega} = [L][E] \tag{1}$$

where [E] is a matrix describing the vibrational energy in each subsystem, [P] is the power input matrix, [L] is the loss factor matrix and ω is radian frequency. Equation (1) is solved for a given set of power inputs to the vehicle structure to give the energy levels in each subsystem. The sound pressure level is then calculated from the equation [3],

$$L_e = L_p + 10\log V - 25.4 \tag{2}$$

where L_e is the energy level in dB re 10^{-12}J, L_p is the sound pressure level in dB re 2×10^{-5}N/m^2 and V is the compartment volume in m^3.

For a given running condition the set of vibrational power inputs to the structure have to be calculated. In this work the vehicle is assumed to be stationary with the engine running. To avoid modelling of the engine, its associated components and transmission through the subframe, the power transmitted at subframe mounting bolts and absorption of radiated noise from the engine are considered.

Power input at mounting bolts can be calculated using the equation [2],

$$P_{in} = \frac{1}{2}\left(\frac{a^2}{\omega^2}\right)\mathrm{Re}(Z) \quad (3)$$

where a^2 is the mean square acceleration, Z is the driving point impedance and *Re()* indicates the real part. If the impedance can be measured (or predicted) and the acceleration level at the mount is known then the power input can be calculated.

Sound radiated from the engine into the engine compartment will be incident on panels of the vehicle. Sound transmitted through the fire wall between the engine compartment and the saloon could be important. The power input to the fire wall from the engine compartment is equal to [3],

$$L_w = L_e + 10\log(\omega) + 10\log(\eta_{cp}) \quad (4)$$

where L_w is the sound power level in dB re 10^{-12}W, and η_{cp} is the coupling loss factor from the engine compartment to the fire wall. If the sound pressure in the engine compartment is measured at the fire wall then the sound power input can be calculated using Equation (4). The power input to the saloon due to direct, non-resonant, transmission from the engine compartment can be calculated in a similar manner [3].

3. MEASUREMENTS

Measurements were carried out on a motor vehicle constructed mainly from glass fibre reinforced plastic with a density and longitudinal wave speed of 1600 kg/m^3 and 2700 m/s respectively. The subsystems of the vehicle vary in surface area from 0.1 m^2 to 1.5 m^2 with thicknesses from 3 mm to 12 mm. The saloon has a volume of 1.1 m^3. The damping of the saloon was measured using the decay rate technique.

The driving point mobility *(Y=1/Z)* was measured at various mounting points from the subframe to the vehicle using an impedance hammer to measure the force and an accelerometer to measure the response of the structure. The signals were recorded on a real time narrow band frequency analyser, B&K type 2032. The results of ten impulses were averaged at each position. Fig. 1 shows the real part of two of the mobility's which were measured at the floor mount. This mount is positioned close to the joint between the floor and the fire wall.

The floor mobility is typically around -28 dB which is the same as the predicted value for an infinite plate 8 mm thick [2]. The floor thickness varies between 6 mm and 10 mm. At the joint between the floor and the fire wall the mobility is lower than the floor mobility at low frequencies, up to 1 kHz. At higher frequencies the two results are similar. The difficulty is in deciding which mobility to use when calculating the power input at the floor mounts below 1 kHz.

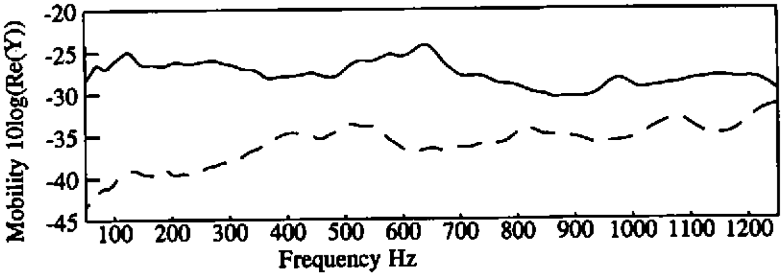

Fig. 1. Measured mobility (real part) at the floor mount. ____ , floor; - - - , joint.

Fig. 2 shows the measured power input to the car with the engine running at 3000 rpm. Acceleration levels were measured at the mount points. The sound pressure level was measured at the fire wall, in the engine compartment.

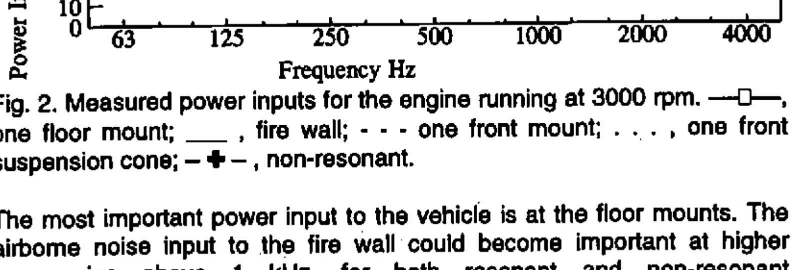

Fig. 2. Measured power inputs for the engine running at 3000 rpm. —□—, one floor mount; ___ , fire wall; - - - one front mount; . . . . , one front suspension cone; – ✚ – , non-resonant.

The most important power input to the vehicle is at the floor mounts. The airborne noise input to the fire wall could become important at higher frequencies above 1 kHz, for both resonant and non-resonant transmission.

A set of experiments were carried out to allow comparison of measured and predicted sound pressure levels in the saloon. To simplify the experiment only two variables were measured. For a given engine speed the sound pressure level at different positions in the saloon and acceleration levels at the floor mounts were measured. The procedure was repeated ten times for each engine running speed. the average floor mount acceleration level was then used to calculate the power input to the vehicle which was subsequently used in the SEA model to predict the sound pressure level in the saloon. A range of results could be expected depending on which impedance is used for the mount point when calculating the power input.

Fig. 3. shows the measured and predicted sound pressure levels in the saloon for an engine running speed of 3000 rpm with the vehicle stationary. Two predictions are shown. The dashed curve used the mobility at the joint (given in Fig. 1) and the solid curve used the floor mobility. Good agreement between measured and predicted results is shown. Similar agreement is found for measurements at other running speeds.

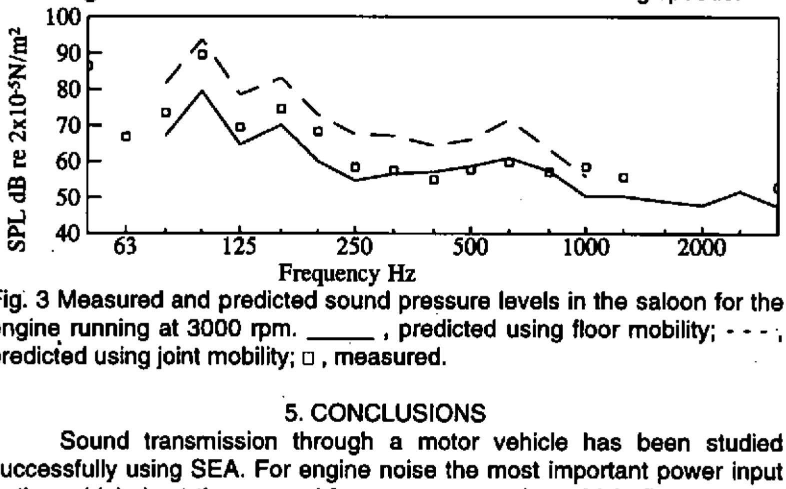

Fig. 3 Measured and predicted sound pressure levels in the saloon for the engine running at 3000 rpm. _____, predicted using floor mobility; - - -, predicted using joint mobility; □, measured.

5. CONCLUSIONS

Sound transmission through a motor vehicle has been studied successfully using SEA. For engine noise the most important power input to the vehicle is at the rear subframe mounts to the vehicle floor. A good estimate of the sound pressure level in the vehicle saloon can be obtained by only considering the power input at the floor mounts in this case.

6. REFERENCES.

[1] J A Steel, SAE P-291, 951330, 758-790 (1995).

[2] R H Lyon, Statistical Energy Analysis of Dynamical Systems: Theory and Applications. MIT Press (1975)

[3] R J M Craik, Sound transmission through buildings using statistical energy analysis. (Gower, UK, 1996)