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ACOUSTICAL EXCITATION OF BUILDINGS BY ROAD VEHICLE LOW FREQUENCY NOISE

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INTRODUCTION

The vibration response of buildings in the vicinity of roads, as felt by residents in terms of annoyance, is partly due to airborne excitation from the vehicle exhaust system and partly to solidborne excitation induced by the vehicle into the soil. The acoustic component is generally a low frequency -narrow band excitation on octave bands 63 Hz and 125 Hz. In this frequency range, the response of the building is a modal response, considering either the acoustical response of the rooms or the mechanical response of the panels.

MODELLING OF THE ROAD VEHICLE LOW FREQUENCY NOISE EXCITATION

The main part of vehicle low frequency noise is generated by the exhaust system, due to the high temperature gas flow coming from the cylinders. Thus the model involves (1) the engine itself with the cylinders, (2) the exhaust system, (3) the exhaust end radiation.

Engine

The model is limited to Diesel engines with 4 cycles comprising adiabatic compression, constant volume combustion, constant pressure combustion, adiabatic relaxation, finally constant volume release. The necessary simplifying hypothesis make it possible to compute the gas temperature at the outlet of the engine as a function of construction parameters (cylinder volume, number of cylinders etc ...) and working parameters (engine speed and load, power ...). From the outlet gas temperature, it is possible to evaluate the mean gas flow Qmoy ; the instantaneous gas flow Q(t) can be considered as a succession of sine pulses, with Qmoy as a mean over time. From a decomposition of Q(t) into Fourier series, one obtains an estimation of the fundamental frequency amplitude Q1, the frequency being :

$$f_1 = \frac{n\,N}{120}$$ with n engine speed (rpm), N number of cylinders

Exhaust piping
The model estimates the noise reduction of the flow source with simple exhaust piping configurations (upstream and downstream pipes connected to an expansion chamber, a resonator, or two chambers with internal coupling). The hypothesis imply linear theory, plane wave propagation, no radiation of the exhaust surface, infinite acoustic impedance of the source, and the radiation admittance of the exhaust end so that

$$1/Z_b = 0.695 + j\,\frac{1.18}{ka} \qquad \text{with } a = \text{radius of the exhaust pipe},\; k = \text{wave number}$$

Under these assumptions, the model computes the value of the input impedance of the piping (ZR) and of the noise reduction (NR), for the different exhaust configurations, which gives the noise pressure level at the output as a function of Q1. NR and ZR depend on frequency and temperature.

Exhaust end radiation
Due to the low frequency, the source is considered as a plane piston radiating in an infinite baffle (hemispheric radiation).

Results
The model estimates the acoustic power of the exhaust end for the fundamental frequency as a function of engine and exhaust characteristics. Figure 1 demonstrates the results for different engine powers and speeds for a Renault SG2 Diesel-powered light truck, with an internal coupling silencer. The data given by the model are of the same order as the measured levels, but there are discrepancies in the details.

MODELLING OF THE LOW FREQUENCY NOISE OUTSIDE PROPAGATION

Open field
In open field, the sound wave is submitted to geometric spreading, atmospheric absorption and ground effect. Only the geometric spreading (in 1/r for pressure p) is to be taken into account for the frequency band involved.

Street with parallel facades
The model considers the image source theory, which can be justified if the dimensions of the street are of the same order or higher than wave length (which is about 3-10 m). Under this assumption, any point of a facade receives the direct and reflected waves of order i so that

$$P_e = \sum_{i=0}^{\infty} \frac{A}{d_i} R^{i/2} e^{-j\frac{\omega}{c}d_i} + \frac{A}{d'_i} R^{i/2} e^{-j\frac{\omega}{c}d'_i}$$

where $\sqrt{R}$ is the reflexion coefficient for pressure, di (resp. d'i) the distance from image source i(resp. i') to the receiver, A a constant related to acoustic power of source. The time delay between two image sources, due to the speed of the vehicle, can be taken into account. Figure 2 demonstrates the value (dB) to be added to open field sound pressure level to obtain the level with two parallel facades, for a given geometry.

It shows the frequency peaks appearing for such a configuration. They are very similar to the natural modal frequencies of the street. When the vehicle moves along the street, one obtains alternatively maximums and minimums of levels for one receiver point on the facades.

MODELLING OF THE SOUND TRANSMISSION INTO THE ROOM

Mass law

From the incident pressure p_e, the mass law gives a rough idea of the pressure to be obtained inside a room separated from the outside by a window, according to the characteristics of the window, with typical insertion loss of 10-20 dB.

Modal approach

An other way is to consider the window panel coupled to the acoustic cavity of the room with a modal approach. Is shows the importance of the coupling of window modes and room modes, on the basis of the works developped in references [1] and [2] where the walls are rigid. Nevertheless, the results are very sensitive to the exact values of coupling frequencies, and further work is needed to adjust the model to realistic situations. Figure 3 shows the comparison between mass law and modal approaches for a typical window/room coupling.

MODELLING OF THE ROOM PANELS VIBRATIONS

On the basis of an experiment for evaluating the room $\rightleftharpoons$ panels transfers in buildings exposed to traffic vibrations in the [20-200]Hz frequency domain[3], one can propose an empirical formula between sound pressure p_i at the corner of the room and acceleration γ_i at the middle of the panel. The formula is :

$$10 \log \left[\frac{p_i^2}{\gamma_i^2} \cdot \frac{\gamma_0^2}{p_0^2} \right] = 24 + 20 \log \rho_s \quad \text{with} \quad \begin{aligned} p_0 &= 2.10^{-5}\,Pa \\ \gamma_0 &= 6.9\ .10^{-4}\ m/s^2 \end{aligned}$$

ρ_s = surface mass of the panel.

Then we can estimate the vibration response of the room.

CONCLUSIONS

The vehicle $\rightarrow$ building analysis of low frequency noise induced vibrations makes clear the pure tone characteristics of excitation, related to engine/exhaust structure and working parameters, and the modal characteristics of response either outside or inside the building, related to geometry and building components. The existence of construçtive or destructive couplings between source and the different stages of response (street, window, room cavity, panels) is demonstrated. The model which has been presented is a first trial to evaluate the whole system towards a better understanding and control of low frequency noise generated vibrations.

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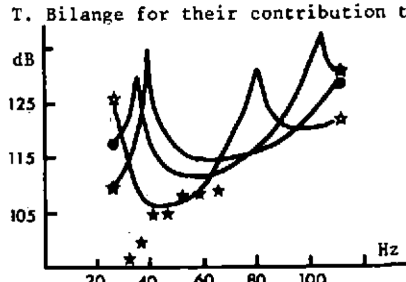

Figure 1 Sound power level (dB) of the fundamental component of the engine/exhaust noise of a vehicle. ——: computed (☆ no load, ● half load, ✱ maximum load); ★ : measured, no load.

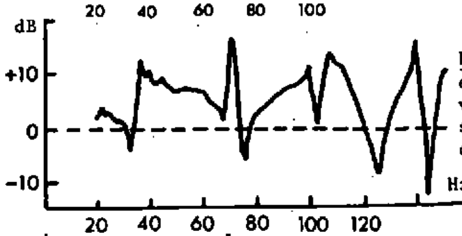

Figure 2 Computed reverberation effect (dB) of facades for a given source/receiver geometry; street with two parallel facades compared to open-field condition.

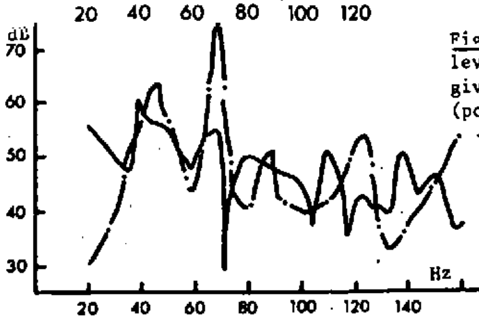

Figure 3 Computed sound pressure level (dB) inside a room for a given geometry; external source (power 95dB at any frequency); ——: mass law; —·—: modal approach; facade in a street with parallel facades.