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EARTH-MOVING MACHINE CAB ENCLOSED SOUND FIELD ACTIVE CONTROL SIMULATION

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INTRODUCTION

Experimental tests carried out inside an earth-moving machine cab enlightened that the noise level at the operator's ear position was higher than 80 dB. The relevant spectrum contained dominant contributions at the first and eighth harmonics of the engine fundamental frequency, due to the engine firing and the cooling system, respectively. Referring to typical machine working conditions and depending on the requested power, the engine rotational speed varies from 1500 r.p.m. to 2500 r.p.m. with a consequent engine fundamental frequency shift from 25 Hz to 40 Hz. The dominant contributions to the global noise level turn out to be always within 400 Hz. In order to have a complete characterisation of the acoustic field in the volume generally occupied by the operator's head, sound pressure maps have been obtained on three vertical parallel planes at a reciprocal distance of 10 cm. Moreover a commercially-available active noise device working as to a single-channel feedforward scheme has been used within the 0÷500 Hz frequency range. A first series of experiments has been carried out using the best secondary source positioning on account of practical requirements as implementation facilities and cost minimisation [1]. The results showed noise reductions up to 5÷6 dB(A) at the operator's ear position. However the spatial extent of the controlled area and the noise reduction amount strongly depend on the setting of several control parameters like signal amplifications and error microphone position.

The present paper reports a first series of results concerning the simulation of the cab enclosed acoustic field. The relevant active noise control achieved by means of the total time-averaged acoustic potential energy minimisation strategy has been emphasized.

CAB ENCLOSED SOUND FIELD SIMULATION

The machine cab volume has been modelled by means of an FE fluid mesh (468 nodes and 320 solid linear brick and wedge elements, pre-processing phase by I-DEAS software), its boundary dimensions defined referring to the structure real ones (Fig.1). Moreover a BE mesh has been directly generated by SYSNOISE software taking the envelope of this original 3-D fluid mesh. In such a way a BEM coupled analysis has been carried out in order to take into account the physical properties of each structural element (kind of material and correspondent thickness) and compute the displacement boundary conditions to be used with the FEM uncoupled analysis. The cab iron parts, windows and vertical centre rod have been modelled and two spherical sources with dominant levels at 60 Hz and 120 Hz the first, and 288 Hz the second (as experimentally measured), placed below the cab base at a distance of 50 cm, have been assumed the only excitations to the structure. This condition simulated the engine and cooling system noise emissions at 1800 r.p.m. engine rotational speed. Also the noise field reflection by the floor has been simulated. Finally the three measurement planes on which experimental results were previously collected have been discretized by three field point meshes.

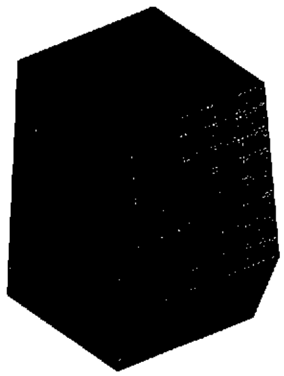

Fig.1 - FE fluid mesh.

Referring to this kind of boundary conditions, the sound pressure values on the three field point planes have been computed by means of the BEM indirect coupled analysis. Table 1 reports the comparison between simulated and experimentally measured pressure levels at the operator's ear position.

Hz	BEM simulated level (dB)	Measured level (dB)
60	85.9	87.2
120	74.6	73.8
288	66.0	78.1

Tab.1: simulated and experimental pressure levels at the operator's ear position.

Notwithstanding the evident simplification in the definition of sources and structural boundary conditions, the simulated values well match the experimentally measured ones, except at the frequency of 288 Hz.

ACTIVE CONTROL SIMULATION

Control strategy

The control strategy described in this application is based on the concept of representing both primary and secondary sound fields by means of the modal superposition approach [2]. Particularly, the primary and secondary sound fields can be expressed as:

$$p_p(\omega) = \sum_{i=1}^{i=n} a_i^p(\omega)\Phi_i$$

and

$$p_s(\omega) = \sum_{i=1}^{i=n} a_i^s(\omega)\Phi_i$$

being a_i^p and a_i^s the modal participation factors of mode i in the primary and secondary field modal expansions, respectively, n the number of modes and Φ_i the i-th acoustic normal mode. The control and reduction of the primary field amplitude is achieved by means of a combination of this field with the secondary one characterised by an on purpose selected source amplitude that minimises the total time-averaged acoustic potential energy [2]. Particularly the optimised secondary source amplitude has been computed by means of FORTRAN programs on purpose developed taking into account the relevant modal participation factors in the primary and secondary field modal expansions within the frequency range of interest [3].

Application

The cab 3-D FE fluid mesh model above described has been used for a FEM uncoupled analysis in order to simulate the effect of the active noise control. According to the maximum mesh element length and the characteristics of the fluid (air), the analysis frequency range has been considered 0 Hz to 300 Hz. As the boundary conditions are concerned, in order to take into account the cab structural characteristics, the displacement boundary conditions computed by means of the BEM coupled analysis and translated in velocity ones have been assigned to the element envelope faces. Eleven acoustic modes have been computed to represent the primary and secondary acoustic fields inside the cab by means of the modal superposition approach. A very good match has been achieved between the results of the FEM and BEM computation procedures (Table 2). The pressure value at 288 Hz turns out to be again less exact in comparison to the others due to the closeness of this frequency to a cab acoustic mode resonance and, as for the BEM procedure, to the upper frequency limit for this analysis (283.3 Hz) that ensures to have at least six elements per wavelength.

The active noise control simulation has been carried out assuming as secondary source a spherical one positioned as the loudspeaker in the experimental tests. The results are represented in Table 2.

Hz	FEM results (dB) no control	FEM results (dB) control on
60	85.7	72.9
120	74.9	69.1
288	67.5	66.8

Tab.2: simulated (no control) and controlled pressure levels at the operator's ear position.

Noise level reductions of about 13 dB, 6 dB and 1 dB at the operator's ear position can be argued relevant to the three excitation frequencies. Moreover in order to consider cab wall and volume effects, different positions both of the secondary source and operator's ear location have been tested. A 2 dB maximum noise level reduction increase has been computed.

CONCLUSIONS

A correct modelling of an earth-moving machine cab and relevant boundary conditions by means of the BEM code allowed the achievement of a reasonable match between the computed inner noise field pressure values and those obtained by experimental measurements. As far as the effectiveness of the active noise control is concerned, a global control strategy requiring that both primary and secondary sound fields are represented by means of the modal superposition approach has been applied. The results obtained by FEM analysis are satisfactory with pressure noise level reductions at the operator's position up to 13 dB at the dominant frequency.

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