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NOISE AND VIBRATION MINIMIZING IN MACHINES THREATED AS MULTI-SOURCES SYSTEM

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Abstract

Minimizing of noise at the operator's cage of engineering machines with complex structure requires the location and the identification of sources. The work is a direct continuation of the study on modelling of the acoustical structure of engineering machines that was presented at the WESTPRAC'94 and Inter Noise '97 Conferences, and is based on the extensive empirical material regarding vibration and noise in hydraulic excavators working in the most severe operating conditions (strip pits, quarries, etc.)

Introduction

The need to modify the model being the basis for a calculation algorithm [1,2,3], as proposed by the author, and further of a computer support system appeared when analyzing the possibilities to lower the vibration and noise in a non-typical working machine, such as Brawal 416 excavator. This excavator is equipped with two twin power units consisting of Diesel engines located asymmetrically in relation to the operator's cab and a mechanical energy accumulating unit consisting of a high-speed flywheel, pump, and a pressure tank. In relation to other machines subject to the author's research, the above mentioned excavator has a separate acoustical structure with the following features:

- There are two strong, independent sources with very similar frequency characteristics. Despite their similarity the sources are practically non-correlated. (Both motors have independent controls and can operate with a similar or different rotational speed).
- Existence of the energy accumulating unit leads to significantly smaller (practically negligible) changes of the characteristics of acoustical sources during a working cycle which created a need introduce various types of averaging. The accumulating unit itself is an additional source of noise.

Similarly as in other machines of this type the propagation of noise is practically impossible to describe in a theoretical manner and requires use of identification procedure, however the minimum number of main sources are 4 (2 vibration sources and two noise sources).

Model identyfication procedure

In the works [1,2] an identification procedure was applied with an assumption that noise at the operator's cab, measured and calculated with the use of simple acoustical formulae accounting for the insulating power of the partitions and shields, differs by certain unknown function (a vector in the frequency space) defining the path of propagation via the machine's structure and the influence of other factors (noise). This led to a vector equation in the domain of frequency which upon application of L operator enabled calculations to be conducted bands with freely defined width. In works [3,4], this course of procedure was generalized for two independent sources, while at the same time studying the possibility for accounting for unknown functions in an additive and multiplication way. Increase of the number of functions defined in the process of identification leads to a necessity of conducting an indirect measurement or of conducting a signal isolation test, which is difficult since the system is non-linear. In problems related to a system with more than two sources application of the above procedures leads to big errors. (Additive or multiplication definition of non-linear corrections with multiple, mutual multiplication of unknown functions is a too rough estimation).

Let us analyze the model presented in Fig. 1:

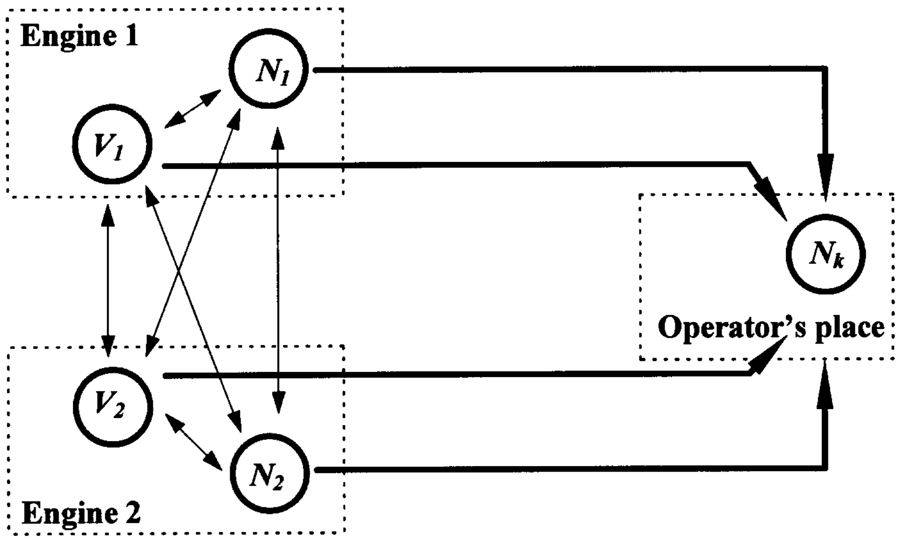

Fig. 1.

None of the sources is isolated and the measurement of both vibration and noise accounts for the influence (unknown and non-linear) of the remaining sources, and besides that the propagation paths are also non-linear. The mutual influence of the sources upon each other is different, which suggests an idea to simplify the model to the form:

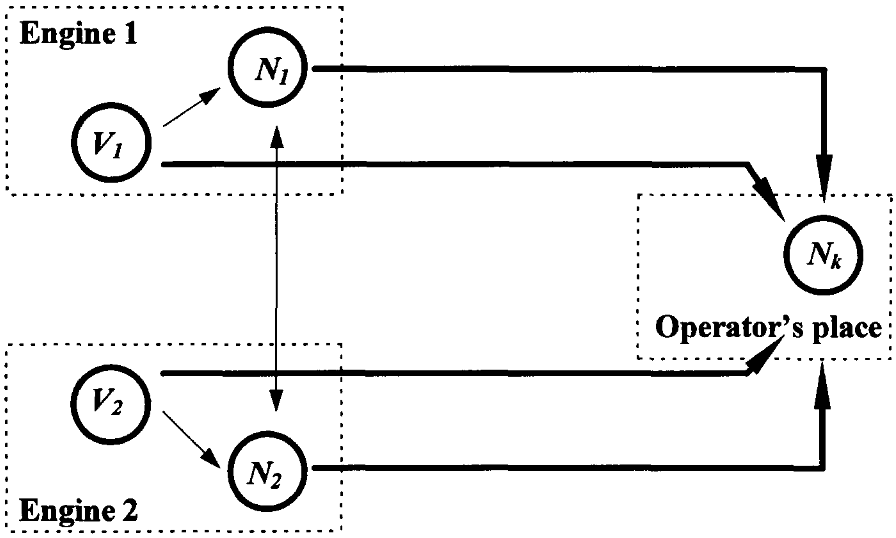

Fig. 2.

Since as we have mentioned, introduction to the system of equations defining the propagation path of „n" unknown functions makes the identification task difficult to execute, we have decided to have a two-stage procedure. In the first step the signals are separated precisely to the linear part on the basis of coherence method (Fig. 3).

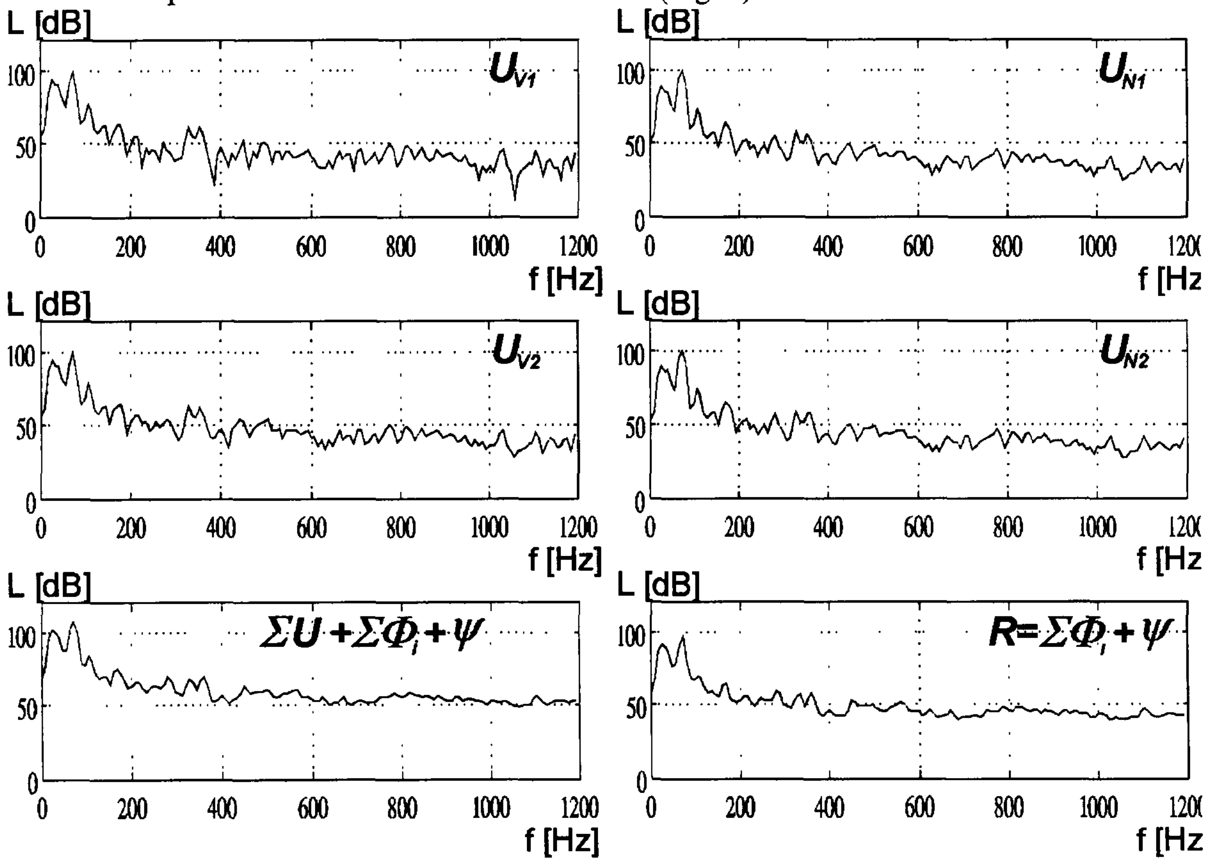

Fig. 3.

In the second, the non-linear part, which in our case denotes the sum of propagation of noise via the machine's structure and the mutual influence of individual sources upon each other, "splits" into individual propagation paths defining the unknown functions Φ, in a system of equations:

$$
\begin{aligned}
N_1^* \cdot H_{N1} &= (N_1+N_2 \cdot H_{N21}+V_1 \cdot H_{V11}) \cdot H_{N1} = Y_{N1} \\
N_2^* \cdot H_{N2} &= (N_2+N_1 \cdot H_{N12}+V_2 \cdot H_{V22}) \cdot H_{N2} = Y_{N2} \\
V_1 \cdot H_{V1} &= Y_{V1} \\
V_2 \cdot H_{V2} &= Y_{V2} \\
\sum Y_{ij} + \sum \Phi_{ij} + \Psi &= X
\end{aligned}
\qquad (1)
$$

where:

- N_i - Fourier transform of noise source,
- V_i - Fourier transform of vibration source,
- H_{Ni} - calculated transmitance between source andoperator place,
- H_{Nij}-H_{Vij} - real existing transmittances between measerment points,
- Y_{ij} - partial noise in the operators cage,
- Φ_i - nonlinear parts of noise propagation,
- X - measured noise in the operators cage.

the identification criterion is as follows:

$$
\begin{aligned}
N_1 \cdot H_{N1} + N_1 \cdot H_{N21} \cdot H_{N2} &= U_{N1} + \Phi_1 \\
N_2 \cdot H_{N2} + N_2 \cdot H_{N12} \cdot H_{N1} &= U_{N2} + \Phi_2 \\
V_1 \cdot H_{V1} + V_1 \cdot H_{V11} \cdot H_{N1} &= U_{V1} + \Phi_3 \\
V_2 \cdot H_{V2} + V_2 \cdot H_{V22} \cdot H_{N2} &= U_{V2} + \Phi_4 \\
\\
N_1 + N_2 \cdot H_{N21} + V_1 \cdot H_{V11} &= M_{N1} \\
N_2 + N_1 \cdot H_{N12} + V_2 \cdot H_{V22} &= M_{N2} \\
V_1 &= M_{V1} \\
V_2 &= M_{V2} \\
\sum \Phi_i &= R - \Psi
\end{aligned}
\qquad (2)
$$

where:

- U_{ij} - spectrum of the separated parts of noise,
- M_{ij} - spectrum of measured noise and vibrations near the souces.

Aplications and Conclusions

The automatic procedure for selecting the acoustical characteristics of the sought sound-insulating shield or casing consists in finding earlier [1,2] the optimum for the given decrease of noise in the cabin. This is selected from an extensive database located in the computer's memory with the observance of the possibility of examining the influence that individual source has on the whole system.

An additional benefit of the proposed method is the possibility to estimate (in relative scale) the influence that vibration has on the level of noise in the operator's cab. This estimation was confirmed empirically. Change of the characteristics of the power unit's suspension, leading to decrease of the level of vibration by 7 dB, led to the decrease of the noise at operator's cab by 3 dB and to a similar decrease of the external acoustical field.

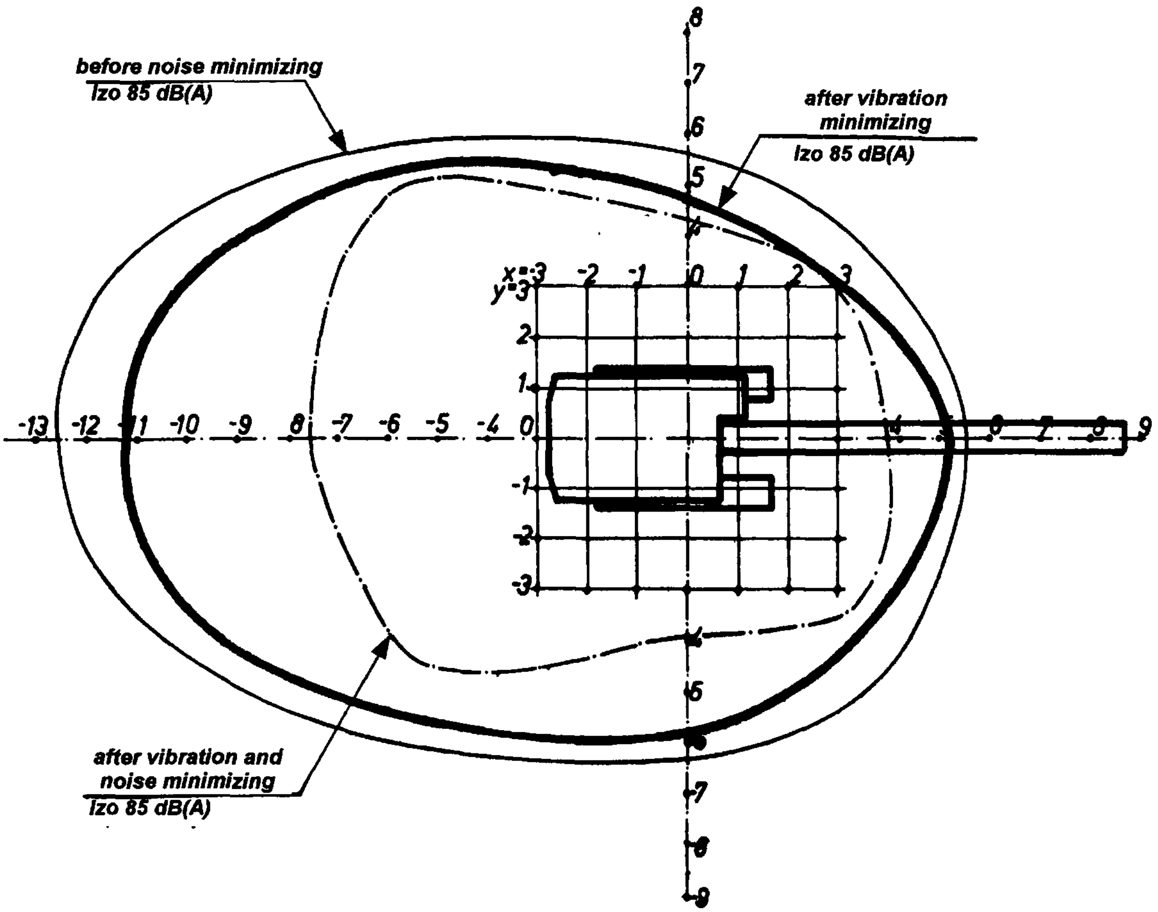

Fig. 4.

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This work is now (08.1997) in preparing, the full theory and some aplications will be presented in Conference