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INVESTIGATION OF CONSTRUCTION MACHINE CAB NOISE GENERATION

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Abstract

Sound fields of sound-proofed and vibration-isolated cabs are studied. Significant noise is found in the extremely low frequency range. These contributions are caused by cab air volume oscillations. Structure-borne noise contribution is also considered. The magnitude of a particular noise source contribution depends on the sound isolation of cab panels, on the acoustical tightness (seal) of the cab, on cab sound absorption, and on diffraction around cab elements. Results of experimental and theoretical studies of cab acoustical characteristics are presented.

INTRODUCTION

The cab that encloses the operator location of a construction machine is a complex system. It transfers sound due to reflection, sound absorption, resonances, diffraction, sound scattering, and other phenomena. The cab may be considered as an acoustical filter, as an acoustical barrier and as a sound radiator.

ANALYSIS OF CAB NOISE SPECTRA

Low-frequency tends to dominate the in-cab sound spectra of construction machines considered in this study (such as dozer tractors, wheel loaders, back-hoe loaders, and motor graders). Based on a large number of experiments, the average operator (or in-cab) SPL spectrum may be approximated by a straight line characterized by monotonic decay of by 5-6 dB per octave.

Characteristic peaks at frequencies 31,5 and 63 Hz have two different causes. The peak at 31,5 Hz is caused by forced vibrations of the internal combustion engine (i.c.e.) at the frequency coincides to the first harmonic of the i.c.e. shaft rotation (1st order) which is determined from the equation: $f = \frac{n}{60}$ (where n is the engine rpm). For a typical 6-cylinder diesel engine working in the range of 1700-2200 rpm, the engine first order corresponds to the frequency range $f = 29 - 37$ Hz. It is hypothesized that a flexible element of the cab (for example a large window or ceiling) starts pulsating from the forced influences and initiates the cab air volume oscillation. Sound pressure level variations in the range of 90-95 dB are considered at these frequencies. When a cab door is opened, additional air mass is added to the oscillating cab air mass. This causes reduction of sound pressure levels by approximately 10 dB. In this situation structure-borne noise mostly determines the total cab sound field.

Cab noise generation at 63 Hz mostly depends on structure-borne noise. As concluded from experimental studies, the structure-borne sound contribution in this frequency range is typically 3-5 dB higher than the air-borne noise contribution. Sound pressure levels at this frequency in different machine cabs are in the range 85-90 dB. Vibration is determined by the second harmonic of i.c.e. (about 58-75 Hz) which corresponds to the octave band with the geometrical mean frequency 63 Hz. These assumptions are confirmed by the presence of two peaks found at 1/3 octave band spectra around frequencies 25 Hz and 63 Hz (Fig. 1).

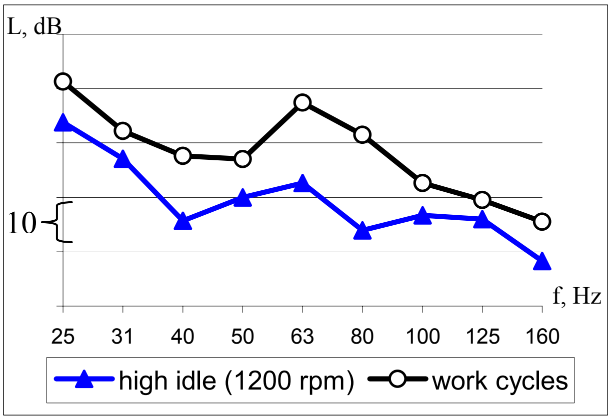

Figure 1 - Construction machine cab noise spectra (1/3 octave bands)

Formation of cab sound field in the frequency range 250-8000 Hz is mostly determined by the air-borne sound. For some machines structure-borne sound contribution at frequency 125 Hz is comparable. Dominant air-borne sound in different construction machines cabs can imply good vibration isolation of cabs. Primary features of cab noise generation are summarized in Table 1.

ACOUSTICAL PROPERTIES OF THE CAB

Characteristics of Cab Sound Field

The acoustics of construction machines is mostly determined by noise sources located outside the machine cab and by the forced vibrations of the cab elements. Each cab panel may be considered as a plane sound radiator that emits plane sound waves at small distances. Thus, the total cab sound field in any point within the cab may be presented as a superposition of a large number of plane waves equally distributed in the space.

Features of cab sound field

Table 1

Main cab sound features in particular frequency ranges		
31,5 Hz	63-125 Hz	250-8000 Hz
Forced oscillations of the cab air volume (cab door is closed) $f = \frac{n}{60}$, Hz	Structure-borne sound contribution is dominant (at 63 Hz) or compared with the air-borne noise (at 125 Hz) $f = \frac{(2,4)n}{60}$, Hz	Air-borne noise contribution is dominant, sound propagates from outside through the cab panels

It is assumed that the sound field in construction machine cab is fairly uniform and quasi-diffuse. It has been confirmed by the number of experiments that an oscillation of the cab air volume at the frequency of the i.c.e. rotation (n/60) occurs without influence of wave phenomena at frequencies well below the cab cavity modes. These oscillations are likely caused by pulsations of the most flexible cab panels due to the forced excitation. Cab cavity modes may resonate and affect the sound field when excited by engine orders or during machine operation. The first acoustic mode of a typical construction machine cab is about 100 Hz. Experimental results confirm that the sound field in the frequency range 250-8000 Hz is fairly uniform and the assumption of its diffuse character is valid [1], provided the sources of excitation are outside the cab.

Sound Absorption of Cabs

Sound wave traveling in a cab are reflected and absorbed by cab panels and boundaries. Sound absorbing materials transform sound energy into heat and reduce the reflected energy. The average coefficient of sound absorption is a frequency-dependent quantity characteristic of the sound absorption and determined as follows:

$$\bar{\alpha}_{cab} = \frac{1}{S_{cab}} \sum_{i=1}^{n} \alpha_i^f S_i , \qquad (1)$$

where:

S_{cab} is the cab area, m^2;

α_i^f is the frequency dependent coefficient of sound absorption of *i*-th surface of the cab characterized by the area S_i, m^2;

n is the number of surfaces characterized by different sound absorption α_i^f.

The average coefficient $\bar{\alpha}_{cab}$ may be estimated using the Eq. (1), but usually experiments based on reverberation time are carried out in order to estimate the sound absorption coefficient using Sabine or Norris-Eyring equations. Cab sound absorption coefficients obtained from measurements of modern construction machines are typically 0,1-0,2 in the low frequency range. The coefficient constantly increases in the medium frequency range and reaches approximately 0,25 in the high frequency range.

Sound Diffraction around the Cab

Sound traveling around an obstacle (cab panel) when a noise source is located outside the cab is called diffraction. The sound wave is incident upon a cab front panel travels around the cab and diffracts at the cab's edges. A sound "shadow" is formed behind the cab in the area where sound energy decreases. As a rule, for construction machines, two cab side panels and a back panel (relative to the dominant noise sources) occur in the shadow zone. In some cases (Fig. 2) the cab may be considered as an acoustical obstacle (barrier). The cab is a wide barrier where double diffraction (at side panels and at the back panel) will occur.

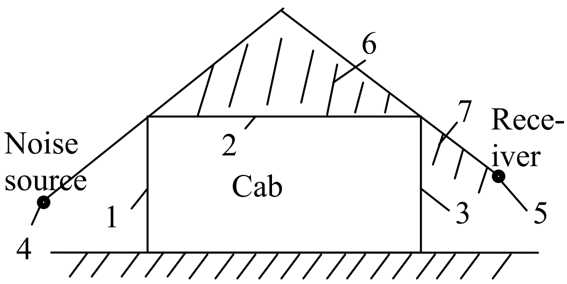

Figure 2 - Calculation scheme of sound diffraction at cab panels:
1 – front panel, 2 – ceiling and side panels, 3 – back panel, 4 – noise source,
5 – receiver position, 6, 7 – sound shadow.

Sound attenuation is obtained at frequencies where the cab size is comparable or exceeds the sound wave length. Sound energy is reduced beyond cab panels as the distance from the noise source increases. The level of attenuation depends not only on the panel size and sound wave length but also on the properties of the boundaries where sound wave diffracts. Most of cabs have rigid exterior panels where sound reflects. Sound reflection at a hard surface occurs without change of the phase, thus two parts of the sound wave are characterized by the same phase and their total amplitude increases in comparisons with the amplitude of a single incident wave.

Sound attenuation also depends on the noise source's characteristics. If the noise source is a point source (for instance, an exhaust pipe) it emits spherical waves and sound attenuation is higher than from a plane noise source (for instance, an engine enclosure). Experimental results of sound attenuation around a hydraulic excavator cab are presented in the Table 2. Analysis of data presented in Table 2 shows that in the low frequency range (63 Hz) sound attenuation is almost negligible. In the medium frequency range (125-250 Hz) it is still rather low. When the cab size exceeds sound wave length (frequencies higher than 500 Hz) the attenuation level increases but does not change significantly in the frequency range from 500 Hz to 4000 Hz. It is noted that the magnitude of sound attenuation at particular cab panels influences the average sound isolation of cab panels, which may reach 5-14 dB in the frequency range 125-4000 Hz depending on particular panel location.

Sound attenuation at outer cab panels of the hydraulic excavator

Table 2

Panel	Sound attenuation, dB, in octave frequency bands, Hz						
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**
top	–	5	7	8	8	8	8
left	–	5	8	10	10	10	11
back	–	8	10	12	13	14	14

In an elaborated theoretical-empirical prediction method [2] diffraction is taken into account by a diffraction coefficient. In order to simplify the estimation procedure this coefficient is used to increase or decrease the particular sound isolation of a respective cab panel. Thus, the average sound isolation of a cab panel in the elaborated prediction method is estimated by the following:

$$\overline{SI}_{cabi} = SI_{cabi} + t_{difi}\,, \tag{2}$$

where

SI_{cabi} is the sound isolation of the i-th cab panel, obtained from the measurements, dB;

t_{difi} is the the *i-th* cab panel sound isolation correction (depends on the location of the panel relative to a noise source), dB.

For instance, if we estimate sound propagation through the back cab panel of a hydraulic excavator at frequency *f*= 1000 Hz, and measured sound isolation is $SI^{1000}_{front\_panel}$ = 40 dB, then $t^{1000}_{dif} = 13dB$, so, SI$_{front_panel}$*=40+13=53 dB.*

Thus, it is important to take into account diffraction phenomena features in cab noise prediction models.

Sound Isolation of Cab Elements

Sound isolation is a numerical characteristic of the insulating properties of a wall. A wide range of experiments was conducted in order to study sound isolation of cab panels of modern construction machines (excavators, loaders, dozer tractors, etc.) The average sound isolation of hydraulic excavator cab panels obtained from experiments is presented in Fig. 3. As follows from the figure there is no significant reduction of sound isolation in low frequencies. It may be explained by presence of its peak values at frequencies lower than 50 Hz. An increase of sound isolation with increasing frequency is found for curves (2) and (3). For the lowest curve (1), significant reduction of sound isolation is shown for frequencies higher than 2000 Hz. This may be explained by presence of openings and apertures at this cab panel. As follows for experimental research of a large number of construction machines most of modern cab frame panels have high sound isolation that reaches 20-30 dB in the frequency range 63-2000 Hz.

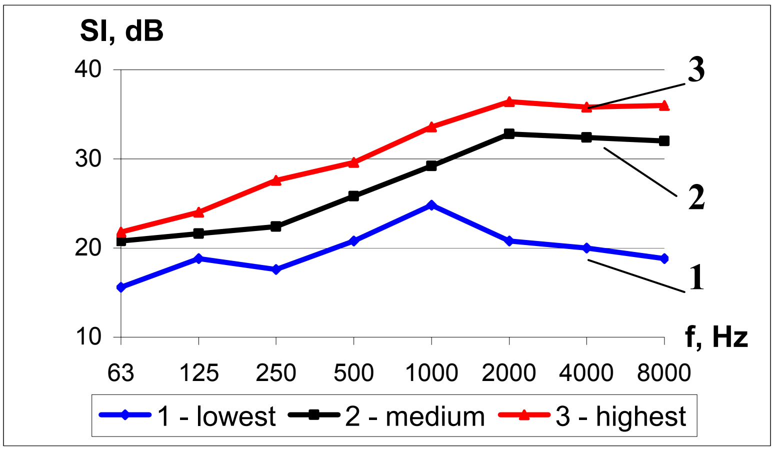

Figure 3 - Average sound isolation of hydraulic excavator cab panels

Further increase of sound isolation may be obtained by more careful sealing of openings, by vibration damping and in some cases by increase of surface mass.

SEPARATION OF STRUCTURE-BORNE NOISE CONTRIBUTION

Structure-borne noise is oscillations transmitted from a vibration source to cab panels through surrounding structures. In other words, vibrations transmitted through shielding constructions impact cab noise.

General features of cab noise generation process are presented in Fig. 4. A cab sound field is formed by three main ways. The first way is caused by vibration excited by mounts and bearings of the internal combustion engine (i.c.e.) and by other vibration sources, which is transmitted through the vehicle frame to the cab vibration isolators. Then it excites vibrations of the cab panels, in turn emitting structure-borne noise into the cab. The second way is caused by the air-borne sound propagation from the exterior noise sources through the cab panels. The third way is caused by reflected sound formed within the cab by the air-borne and structure-borne noise contributions when sound waves traveling within the cab reflect at the cab panels. Thus, three components (air-borne, structure-borne and reflected) sound shall be considered to analyze the total cab sound field. The intensity of structure-borne noise in the cab depends on the vibration level at the engine mounts, on efficiency of vibration isolators of the i.c.e. and the cab, on vibration attenuation level at the vehicle frame and on damping characteristics of the cab panels.

Let us consider the structure-borne sound contribution in comparison with the air-borne contribution using the example of a hydraulic excavator presented in the Table 3. Separation of these two components is obtained by using a theoretical-empirical prediction method based on the measured acoustical characteristics of noise sources and on further theoretical estimation of noise sources contributions [2].

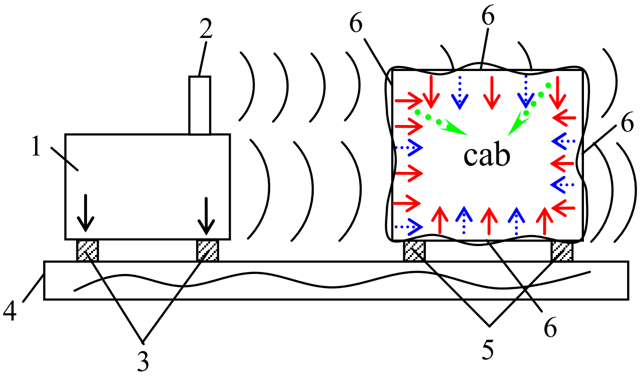

Figure 4 - Scheme of cab noise generation: 1 – internal combustion engine (i.c.e.), 2 – i.c.e. exhaust, 3 – i.c.e. vibration isolators, 4 – vehicle frame, 5 – cab vibration isolators, 6 – shielding (frame) panels of the cab,
→ vibration in i.c.e. mounts, ~ vibration in cab shielding panels,
sound radiation by cab elements due to vibration effect,
sound propagation into the cab from outer noise sources,
))) noise sources emissions towards the cab panels,
reflected sound in the cab.

Based on the analysis of structure-borne noise contributions to the total cab sound field of a hydraulic excavator, it is assumed that main contribution is caused by the i.c.e. and not by hydraulic system. This assumption is confirmed by measurements of structure-borne noise at two modes of excavator operation: when hydraulic system is turned on and when it is turned off. Experimental results in these two cases were almost similar, confirming the hypothesis of domination of the i.c.e. vibration effect.

Comparison of structure-borne and air-borne sound contribution into the total cab noise of a hydraulic excavator

Table 3

Descriptor	Sound pressure level, dB, in the frequency range, Hz			Contribution, dBA
	63	**125**	**250**	
Air-borne sound	82	72	69	66
Structure-borne sound	84	72	–	60

Analysis of experimental results obtained at different types of construction machines shows that the contribution of structure-borne noise in the 63 Hz octave band is dominant (3-5 dB higher) compared to the air-borne noise. The structure-borne noise contribution in the 125 Hz octave band is almost comparable to the air-borne sound (just 1-2 dB higher). The structure-borne noise effect at higher frequencies is not considered. This circumstance is proved by a series of measurements of efficiency of vibration isolators of cabs (Fig. 5), which is found to be 20-30 dB in the frequency range 63-1000 Hz.

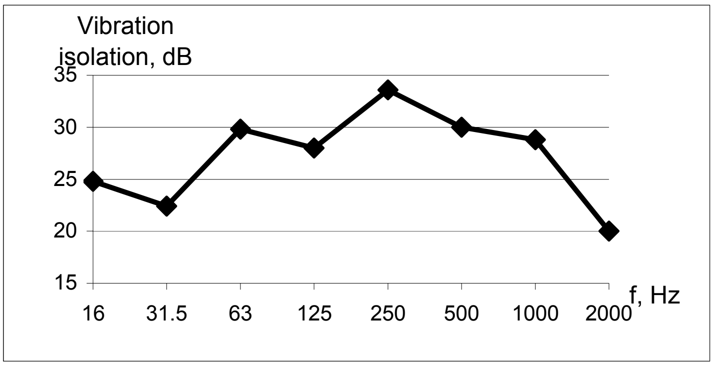

Figure 5 - Average efficiency of vibration isolators of a hydraulic excavator cab

CONCLUSION

Sound radiation, diffraction, vibration in cab, air-borne and structure-borne noise paths all affect the acoustic field inside a cab. The sound field sound field may be considered as quasi-diffuse starting from 250 Hz. The very low frequency booming (in the 31.5 Hz octave band) is formed by means of forced vibrations of the air mass. Structure-borne contribution typically dominates cab noise spectrum at frequency 63 Hz. Air-borne noise sources located outside the cab dominates the higher frequency spectrum, and originates from machine components such as the i.c.e. exhaust, i.c.e. body, hydraulic system, etc. Sound absorption within a cab plays a major role. The average coefficient of sound absorption in modern cabs is between 0,15 and 0,25. Sound isolation of cab panels is not uniform. It is shown that structure-borne sound influences cab noise mostly in the low frequencies.

REFERENCES

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