Rationale for choosing the sound-absorbing materials for the operator`s cabins of the rail-grinding machine

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Abstract. The operation of the rail grinding machines that use noise and vibration-active parts in their structures inevitably leads to the increased levels of vibration and noise that adversely affect their operators. In this field, the specialists from different countries have carried out certain studies. However, the process of forming the vibro acoustic characteristics of the rail grinding machines has not been studied. It should be also noted that theoretical studies of the vibro acoustic dynamics of the rail grinding machines and methods for calculating the vibration and noise spectra are limited, which significantly complicates the substantiation of the engineering solutions to reduce the sound emission intensity of the dominant noise sources. Due to the negative impact of the elevated noise levels on rail grinding machine`s operators, it is necessary for the design stages with appropriate vibro acoustic characteristics that meet requirements of the sanitary standards.

The main practically and technologically possible way to align noise emission level is the choice of the sound-absorbing materials and the achievement of the sound insulation, based on the fulfillment of the sanitary noise standards. The article describes the rationale for choosing the sound-absorbing materials for the operator`s cabin of the rail-grinding machine due to the reduction of the air noise component, which is more common than the structural one. A criterion for minimizing acoustic characteristics inside the operator's cabin is determined to select the most rational engineering method of the sound absorption.

1. Introduction

The harmful effects of the vibration and high intensity noise on the labour’s health are well known [1-4]. It should be highlighted that vibration and noise data of various types of the technological machines are presented in works [5-7]. Moreover, the capability of vibro acoustic characteristics to the limiting spectra is not only one of the most important ergonomic indicators, but it is determined as technical condition of the engineering machines of various functional purposes and competitiveness [8-11].

A large number of the scientific works are devoted to the development of the theory and practice of dealing with noise and vibration of the road-building and technological machines [12-15]. However, the problem of reducing vibration and noise levels for different rail grinding machines has not been studied enough, which actually determines the relevance of the materials in this article.
2. **Object and methods of research**

The functional purpose of the soundproof operator’s cabin and enclosures is to reduce sound pressure levels to sanitary standards. In the overwhelming majority of cases, the operator’s cabins have a shape similar to a thin-walled rectangular parallelepiped exposed to the air and structural noise components. The calculation of the operator’s cabins under the influence of the structural noise component is carried out by the methods of energy balance with a significant vibration effect.

The air noise component is come out of external sources and the calculation of the operator’s cabin elements is done from the position of the required sound insulation value. To fulfill the sanitary and hygienic working conditions of the operators in the cabins, firstly, the air conditioners and fans are installed inside the operator’s cabins according to the parameters of the temperature requirements. In the vast majority of cases, these sources, which are classified as internal, create excess sound pressure levels in the cabins. As it is practically impossible to reduce the noise levels of such sources at the production enterprises, the most reasonable option for fulfilling sanitary standards in this case is a rational choice of sound-insulating materials.

For forming the minimization criterion, it is beyond argument that there is a requirement for maximum reduction of the noise levels at the calculation point in octave bands. However, this condition is necessary, but not sufficient, as the standard curve defines the acceptable noise levels in each of the bands. Therefore, as a necessary and sufficient criterion, it should be recognized one or another deviation degree of the calculated noise levels in octave bands of the spectrum from the normative ones.

![Figure 1](image)

**Figure 1.** Comparison of the standard and actual noise spectra: 1 – the standard curve; 2 – actual or calculated noise spectrum; \( \Delta_1 \ldots \Delta_3 \) – negative differences \( L_{ip} - L_{i,stand} \); \( \Delta_4 \ldots \Delta_9 \) – positive differences \( L_{ip} - L_{i,stand} \).

This deviation degree can be considered the sum of the differences between the calculated and standard noise levels for all nine octave bands of the audio range (from 31.5 Hz to 8000 Hz). At the same time, taking into account the different influence degree of sound radiation frequency on the human body (high frequencies are more unfavorable than low frequencies), it is logical to hypothesize that forming the optimization criterion for each octave band, a weight coefficient should be introduced, and the criterion should be chosen as a weighted sum, which value is necessary to minimize:

\[
K = \min_{\theta_i} \sum_{i=1}^{9} (L_{ip} - L_{i,stand}) \theta_i; L_{ip} > L_{i,stand}
\] (1.1)
where $L_{i\text{stand}}$ – standard and $L_{i\text{p}}$ – actual or calculated noise levels in $i$-th of nine octave bands of the spectrum; $\theta_i$ – the weight coefficient; $U$ – the vector of factors influencing the noise level which are controlled by designer; $W$ – the vector of constructive and technological transitions and physical constants.

In this case, it is obvious to take into account only those spectral components, the levels of which exceed the corresponding levels of the standard curve. The figure 1 shows such a noise spectrum being superimposed on the target curve. Minimization of the criterion is attracted to the actual noise levels to the normative in the $\Delta_i > 0$.

To calculate the weight coefficients $\theta_i$, the following principle is proposed (see Figure 2):
- for middle band of the spectrum, namely for a frequency $500$ Hz we take $\theta_5 = 1$;
- a straight line parallel to the ordinate c-c axis, we will consider the axis of symmetry, relative to which the normative curve $L_{i\text{stand}}$ is mapped to the curve $L_{i\text{p}}$. The ratio:

$$\theta_i = \frac{L_{i\text{p}}}{L_{i\text{stand}}}$$

will be considered a weighting factor for the $i$-th band of the spectrum.

The $\theta_i$ values are pointed out in Table 1. The weighting coefficients adopted in this way enhance the influence of high-frequency components and weaken the influence of low-frequency ones on the value of the criterion.

Not to dispute the certain conventionality of criterion (1.2), nevertheless, it must be admitted that to a certain extent it objectively assesses the degree of proximity between the calculated and standard values of the noise levels in octave bands.

**Figure 2.** For determination of the weight coefficients: "a" - standard curve (levels $L_{i\text{stand}}$); "b" - "anti-standard" curve obtained by symmetric mapping of the curve "a" relative to the c-c axis (levels $L_{i\text{p}}$)
Table 1. The weight coefficients for the terms of the optimization criterion

| № Bands | Geometric average frequency in octave bands | $L_{i,stand}$ | $L_{i,p}$ | $\theta_i$ |
|----------|---------------------------------|---------------|------------|----------|

3. Analysis of the parameters and controlled variables

According to the research data, the sound pressure levels inside a rectangular volume are determined by the formula:

$$ P = \rho_0 c_0^2 q_0 \exp(-i\omega t) \sum N \frac{\psi(P)\psi(N)\omega\Delta P}{(2\pi f)^2-(2\pi f N)^2}, \quad (1.3) $$

where $f_N$ is the natural frequencies of the internal air volume, which are found by the formula:

$$ f_N = \frac{c_0}{2} \left( \frac{n_x}{l_x} \right)^2 + \left( \frac{n_y}{l_y} \right)^2 + \left( \frac{n_z}{l_z} \right)^2 \right)^{0.5}. \quad (1.4) $$

The noise level generated by the source inside the hood is determined by the formula:

$$ L = 20 \log \frac{P}{P_0}. \quad (1.5) $$

where $P_0$ is the threshold sound pressure, $2\cdot10^{-5}$ Pa.

In practice, the effective sound pressure $\bar{P}^2$, which is usually measured and formed:

$$ \bar{P}^2 = \frac{\rho_0 c_0^2 q_0^2}{2v^2} \sum N \frac{\omega^2}{(2\omega N \delta_N)^2+(\omega N^2-\omega^2)^2}, \quad (1.6) $$

where $\delta_N$ is the attenuation factor.

The attenuation factor is determined by the following expression:

$$ \delta_N = \frac{c_0}{2v} A_S a_S, \quad (1.7) $$

where $A_S = 2(l_x l_y + l_y l_z + l_z l_x)$ is the area of the inner surface of the hood; $a_S$ – absorption coefficient. The absorption coefficient is:

$$ a_S = 1 - \xi^2 \quad (1.8) $$

where $\xi$ is the sound reflection coefficient.

The sound reflection coefficient has the formula:

$$ \xi = \frac{i(a+a^{-1}) \text{sh} a'h}{2\text{ch} a'h - i(a-a^{-1}) \text{sh} a'h}, \quad (1.9) $$
where \( \alpha = \frac{\rho a}{\rho_0 a_1} \), \( a_1' = \frac{2m f}{c} \left( \sin^2 \theta - \frac{c_2^2}{c^2} \right)^{0.5} \); \( a = \frac{2m f}{c} \cos \theta \); \( c \) is the speed of sound in the cabin material; \( \theta \) is the angle of incidence of the sound wave on the cabin wall.

Substituting (1.8) into (1.7) and making calculations, we obtain the expression for the absorption coefficient:

\[
\alpha_s = \frac{4(e h_1^2 a_1' + a^2 s h_1^2 a_1' h \cos^2 \theta)}{4(e h_1^2 a_1' + a^2 s h_1^2 a_1' h)}. \tag{1.10}
\]

where \( h \) is the cabin wall thickness.

The formula (1.8) is valid for the case of the oblique incidence of a sound wave on the cabin wall. The reflection coefficient of sound in the case of normal incidence on the wall is determined by the expression:

\[
\xi = \frac{a - a^{-1}}{a + a^{-1} - 2 i \tan \alpha h}. \tag{1.11}
\]

Then the sound absorption coefficient is obtained by the formula:

\[
\alpha_s = \frac{4 e c t g^2 a h + a^2 \sin^2 \varphi}{4 e c t g^2 a h + a^2}, \tag{1.12}
\]

where \( \varphi = a c r t g^2 c t g a h \).

For more exact calculations, the absorption coefficient can be calculated by the formula [192]:

\[
\alpha_s = \frac{0.162 v}{t_p}, \tag{1.13}
\]

where \( t_p \) is the sound reverberation time in the volume, which is easily found experimentally.

The dependence of the performance of a noise source (in this case, an air conditioner or a fan) is set in the following form:

\[
Q = S_d \theta_{di}, \tag{1.14}
\]

where \( S_d \) is a source surface area, \( m^2 \); \( \theta_{di} \) is a body vibration velocity, \( m / s \).

Substituting the formulae (1.6), (1.13) into the noise level formula (1.5) and considering the air density \( \rho_0 = 1.23 \text{ kg/m}^3 \) and the sound speed in air \( c_0 = 343 \text{ m/s} \), we obtain the formula for the noise level created by a point source inside the cabin:

\[
L = 20 \log \frac{S_d \theta_{di}}{V} + 10 \log \sum_{k \in N} \left( \frac{2n/\pi}{f_j - f_j^2} \right)^2 + 200. \tag{1.15}
\]

The formula (1.14) can be used to calculate both the general noise level and the sound pressure levels in octave and three-octave bands. This requires knowing the bandwidth \( \Delta f_j \) of the spectrum analyzer that will be used for calculations. Then the sound pressure level in the \( j \)-th band will be equal to:

\[
L = 20 \log \frac{S_d \theta_{di}}{V} + 10 \log \sum_{j \in N} \left( \frac{2n/\pi}{f_j - f_j^2} \right)^2 + 200.
\]

where \( N_j \) is the number of natural frequencies of the air volume that get into the \( j \)-th band; \( f_j \) is the geometric mean frequency of the corresponding filter band pass.

The noise levels in octave frequency bands depend on a number of parameters which are:

- **Physical constants**: \( c_0 \) is the sound speed in air, \( c_u \) is the speed of range of the flexural wave in the glazing elements, \( \rho_0 \) is the air density, \( \rho \) is the density of the glass.
- The engine’s vibration levels.
- The values of these physical quantities are stored in the database.

- **Geometric variable**: the dimensions of the enclosure \( (l_1 \times l_2 \times l_3) \) and the dimensions of the glazing \( (l_4 \times l_5) \), \( A \) is the internal surface area, \( r \) is the distance from the noise source to the design point.
Parameters which values are characteristic for each octave band and must be calculated along with solving problem by using parameters of the first two groups: $N_1$ is the number of natural frequencies of the air volume oscillations; $m$ and $n$ are constant numbers that determine the modulus of oscillation of the glazing element;

**Controlled variables**, their defined values must provide an extreme criterion and must be found in the design input. These include:

1) The thickness of the glazing elements with definition limits $h_1 \leq h \leq h_2$. This variable is continuous.

2) The absorption coefficients $\alpha_s$ for $N_2$ facing materials in the form of $N_2 \times 9$ matrix, which gives $\alpha_s$ values for each of the $N_2$ materials in nine octave bands. The matrix is located in the database. So, this variable is discrete and is set by a set of $N_2$ values in each octave band.

Consequently, with regard to the found extreme value $h$, choose such a facing material from $N_2$ investigated, which would provide the minimum values for the $M_2$ component over all octave bands:

$$M_2 = 10 \log 10 \sum_{N} \sum_{n=1}^{∞} \sum_{m=1}^{∞} \frac{1}{13.7f_Aa_s+(f_n-f_m)^2}$$

In connection with the discreteness of the $\alpha_s$ value, the optimum of the $M_2$ component is found as a result of enumerating 37 specified $\alpha_s$ values in the matrix in each octave band.

To base on the results obtained, we draw up the expressions of the air levels and structured noise for all walls of all components in octave bands and form the objective function (1.1) in at least three similar options for using covering materials.

**4. Conclusion**

By this means, the reduction of the air noise component, which is more common than the structural one, comes down to two engineering methods:

1. It is rational choice of the sound-absorbing linings on the internal surfaces of the corresponding elements of the cabins.

2. It ensures the required soundproofing of the conforming units of the operator’s cabin.

To select the most rational option for combining the above two methods, the criterion for minimizing the acoustic characteristics inside the operator`s cabin was determined.

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