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ACOUSTIC MODELS OF THE MAIN SOURCES OF NOISE OF MULTI-SPINDLE DRILLING WOODWORKING MACHINE

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Abstract: *The harmful factors in the woodworking industry include the increased noise level in the workplace. This is especially true when working with multi-spindle drilling woodworking machines. The main sources of noise include the constant speed of drills, the geometric shape of the motor and the body of the cutting units, as well as the pneumatic feed mechanism of the cutting units or workpieces.*

The article presents a theoretical justification of noise sources, which allowed us to obtain the equation of sound power of noise sources in general form. The acoustic calculation of the main noise sources was performed using the obtained formulas. The obtained acoustic models of the main noise sources showed that the practical calculation of sound pressure levels or sound power is actually reduced to determining the vibration rates on the natural vibration forms of the sources.

Keywords: *Noise, drill bit, sound pressure, multi-spindle woodworking machines*

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1. INTRODUCTION

Multi-spindle drilling woodworking machines include: a twelve-spindle vertical drilling machine Sv12 (Fig.1), an eight-spindle horizontal drilling machine Sv8 (Fig.2) and a horizontal and vertical drilling machine SGVP (Fig.3).

Fig. 1: A twelve-spindle vertical drilling machine

Fig. 2: An eight-spindle horizontal drilling machine Sv8

Fig. 3: A horizontal and vertical drilling machine SGVP

The key features of the layout, kinematics, largely determining the identification of the main noise sources, are:

- constant speed of the drill bits, as the spindle is directly connected to the motor through the coupling;
- the geometric shape of the motors and cutting unit bodies is close to cylindrical;
- feeding of cutting units or workpieces is performed using a pneumatic drive in the SGVP machine and a manual rack and a pinion drive in the Sv12 and Sv8 machines.

These features suggest that the main noise sources are the bodies of cutting units, drill bits and workpieces being processed.

2. THEORETICAL JUSTIFICATION OF THE NOISE SOURCES

The following are accepted as model noise sources:

- cylindrical emitters for the cutting unit bodies and drill bits;
- beams and plates of limited length for workpieces to be processed.

The initial dependence is the expression for sound pressure [1-5]

$$\mathrm{P} = \frac{\sqrt{2}i\omega\rho_0}{\sqrt{\pi kr}} \sum_{m_\mu=-\infty}^{+\infty} \frac{B_{m_\mu}(k\sin\beta)\, l^{ik_0 r}}{\cos\beta H^1_{m_\mu}(k_0\ R_1\cos\beta)} \cdot e^{i\left(m_\mu\varphi - \frac{2m_\mu+1}{4}\pi\right)} \quad (1)$$

where
ω is the circular oscillation frequency, rad/s;
ρ_0 is the air density, kg/m^3;
k is the wave number, 1/m;
r is the distance from the noise source to the workplace, m;

$$B_{m_\mu} = \frac{1}{2\pi^{1,5}} \int_0^l v(x)\, exp\left[-i\left(m_\mu\varphi + kx\sin\beta\right)\right] dx \quad (2)$$

B_{m_μ} is a function that depends on the oscillation velocity distribution on the noise source surface;
$H^l_{m_\perp}$ is a Hankel function of the first kind of the $m\mu^{th}$ order;
R_0 is the radius of the cutting unit and drill bit, m;
β is the radiation angle,
l_n is the length of the source, m.

We define the dependence of the oscillation velocities distribution as follows:

$$v(x) = \begin{cases} v_k(x) & if\ |x| \le l_0 \\ 0 & if\ |x| > l_0 \end{cases} \quad (3)$$

where
v_k is the source oscillation velocity, m/s.

Since all the above mentioned noise sources are made of steel and oscillate like a solid, then $m_\mu = 1$, and the expression for sound pressure is reduced to the following form:

$$\mathrm{P} = \frac{1{,}35\times 10^{-2} i v_k\, l_r}{r} \times \frac{exp\, i\left(kr - \frac{3\pi}{4}\right)}{H^{1'}_{m_\mu}(kR\cos\beta)}\, \varphi(\beta) \quad (4)$$

where

$$\varphi(\beta) = \frac{\sin\frac{kl_0}{2}\sin\beta}{\frac{kl_0}{2}\sin\beta}. \quad (5)$$

The geometric parameters (the length and radius) determine the nature of the sound emission. Therefore, the expressions for the sound pressure depending on the ratio kR_0, which allow us to refine the engineering calculations of the noise spectra for different frequency intervals, are stated below.

For $kR_0 = 0{,}02f_k \le 1$, we replace the derivative of the Hankel function with the asymptotic representation [1, 2]

$$H^{1'}(kR_0) = -i\frac{1}{\pi}\left(\frac{2}{kR\cos\beta}\right)^2 \quad (6)$$

Since the noise spectra are calculated for the location of the workplace relative to the sound source, the angle β is practically close to 0. In this case, the following expression is obtained for the sound pressure:

$$|P| = \frac{4\times 10^{-2} V_k l\, (R_0 f_k)^2}{r} \quad (7)$$

where
f_k is the natural oscillation frequencies, Hz.

Sound pressure levels in this case

$$L = 20\lg v_k + 20\lg\frac{l}{r} + 40\lg R\, f_k + 66 \quad (8)$$

For the ratio $kR_0 = 0{,}02f_k > 1$, the asymptotic representation of the Hankel function is given as

$$H^{1'}(kR_0) = -i\sqrt{\frac{2}{\pi k_0 R}}\exp i\left(k_0 R - \frac{3\pi}{4}\right) \quad (9)$$

In this case, the sound pressure and sound pressure levels are determined using the following expressions:

$$|P| = 24\frac{v_k l_0\sqrt{R\, f_k}}{r} \quad (10)$$

$$L = 20\lg v_k + 20\lg\frac{l}{r} + 10\lg R\, f_k + 122 \quad (11)$$

The sound power of the sources is defined as

$$W = \pi r^2 |P|\cdot|q| \quad (12)$$

where

$q = i\frac{\frac{\partial P}{\partial(kr)}}{\rho_0\, C_0}$ is the real velocity of air vibrations, m/s;

for the ratio $kR_u < 1$ $\quad q_r = \frac{8\cdot 10^{-5} f_k^2}{r} V_k l_u R_u^2$,

for the ratio $kR_u > 1$ $\quad q_r = \frac{5{,}4\cdot 10^{-2} V_k l_u\sqrt{f_k R_u}}{r}$ (13)

Then the expression for the sources sound power and sound power levels is obtained in the following form:

for $kR_u < 1$

$$W = 10^{-5} R_u^4 f_k^6 l_u^2\ ;$$
$$L_w = 10\, lgV_k l_u + 40lgR_u + 60lgf_k + 20lgl_u + 70$$

for $kR_u > 1$

$$W = 3{,}8(V_k l_u)R_u f_k;$$
$$L_w = 20\, lgV_k l_u + 10lgR_u l_u + 123$$

(14)

3. ACOUSTIC CALCULATION OF THE MAIN NOISE SOURCES

Both the drill bits and cutting unit bodies, are cantilever fitted parts according to the methods of fixing them. Then their natural oscillation frequencies are defined as follows:
drill bits as a steel rod

$$f_k = 0,1\left(\frac{2k-1}{e}\right)^2 \sqrt{\frac{E}{\rho}(D^2 - d^2)} \quad (15)$$

where
D and ***d*** are the outer and inner diameter of the cutting unit, m;
k is the coefficient that determines the natural oscillation frequencies.

For beam-type workpieces, the expressions for sound pressure levels, sound power, and sound power levels are determined by the following dependencies

k h <1

$$P = \frac{1{,}3\cdot 10^{-2} V_k l_u S_u f_k^2}{2}$$

$$L_p = 20\, lgV_k l_u S_u + 40 lgf_k - 20 lgr + 62$$

$$W = 10^{-6}(V_k l_u)^2 f_k^4 S_u'^2$$

$$L_w = 20\, lgV_k l_u S_u + 40 lgf_k + 60 \quad (16)$$

k h >1

$$P = \frac{13{,}5\, V_k l_u (S_u f_k)^{0,5}}{2}$$

$$L_p = 20\, lgV_k l_u + 10 lgS_u f_k + 117$$

$$W = 2,3(V_k l_u)^2 f_k\, (S_u')^{0,5}$$

$$L_w = 20\, lgV_k l_u + 10 lgf_k + lgS_u + 124 \quad (17)$$

where
S_u is the source surface area, m^2;
h is the smaller size of the workpiece cross-section, m^2.

For beam-type workpieces mounted on machine tables, the natural oscillation frequencies are defined as [6]

$$f_k = \frac{1}{2\pi}\sqrt{\left(\frac{\pi k}{l}\right)^4 \frac{EJ}{\rho F} + \frac{j_p}{\rho F}} \quad (18)$$

where
F is the cross-sectional area, m^2;
J_p is the technological system stiffness 'workpiece – machine table', n/m.

For workpieces of the plate type, the sound power is determined according to the data of [7] as follows:

at $f_{mn} < f_{kp}$

$$W = \frac{\rho_0 C_0 k_0^2\, P^2 K_{cb}}{4\pi (10 m_{pl})^2} \quad (19)$$

where
f_{mn} is the natural oscillation frequencies of the plate, Hz;
f_k is the critical frequency, defined as [8]

$$fk = \frac{C_0^2 \sqrt{3(1-\mu^2)}}{\pi C_{pl} h_{pl}} \quad (20)$$

where
μ is the Poisson's ratio;
C_{pl} is the speed of wave propagation in the plate, m/s;
h_{pl} is the plate thickness, m;
P is the cutting force, n;
m_{pl} is the distributed mass, kg/m^2;
k_{c3} is the number of drills;
f_{mn} are the natural frequencies of plate oscillations, Hz;

$$f_{mn} = \frac{\pi}{2} h_{pl} \sqrt{\frac{\mathrm{E}}{12\rho(1-\mu^2)}}\left(\frac{m^2}{l_1^2} + \frac{n^2}{l_2^2}\right) \quad (21)$$

for ***E*** is the modulus of elasticity, PA;
P is the density, kg/m^3;
m and ***n*** are the coefficients that determine the natural frequencies of plate vibrations, m.
for $f_{mn} \geq f_{kp}$

$$W = \frac{P^2 K_{cb}}{16\sqrt{m_{pl} EJ}} \frac{\eta_{rad}}{\eta_{rad} + \eta} \quad (22)$$

where

$$\eta_{rad} = \frac{\rho_0 C_0}{2\pi f_{mn} m_{pl}\sqrt{1 - \left(\frac{f_{kp}}{f_{mn}}\right)^2}}$$ is the radiation coefficient;

η is the effective coefficient of the oscillation energy loss.

4. CONCLUSION

The obtained dependences take into account the geometric and physical and mechanical properties of the above mentioned noise sources. Practical calculations of the sound pressure levels or sound power are actually reduced to determining the oscillation velocities on the natural forms of the source oscillations.

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