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THEORETICAL STUDY OF THE NOISE EMISSION OF THE FLEXIBLE CONNECTION OF THE ABRASIVE BELT-GRINDING WOODWORKING MACHINES

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***Abstract**: This paper presents the results of theoretical studies of the drive belting and grinding belts` acoustic characteristics as elements of flexible connections, that fundamentally distinguishes them from bobbin sanding and cylinder-grinding woodworking machines. It also has the significant differences in comparison with other sources in terms of stiffness and tension forces. The choice of the noise source models is justified by the characteristic features of the kinematics of the machine tools and the grinding technological process. The arrangement of the machines` acoustic system of chosen group suggests that the main sources of energy radiation are: grinding belts and belt drives of the work pieces and grinding units, ground work pieces, electric motors. It is compared the expected noise levels with sanitary standards for determining the excess and the sources that generate them. These data are the main information for the acoustic calculation and design of the noise reduction systems of the above sources at the design stage in such woodworking equipment, and it also determines the acoustic efficiency of the noise protection systems and consider to design these systems in accordance with the implementation of the sanitary noise standards at the machine operator workplaces.*

***Keywords:** acoustic models, octave levels, sound pressure, abrasive belt grinding, woodworking, sanitary standards*

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1. INTRODUCTION

It analyzed the working conditions at workplaces and determination of the risk exposure to production factors on operators of the belt-grinding woodworking machines. It showed factually that the acoustic characteristics significantly exceeded sanitary standards [1, 2]. The arrangement of the acoustic system of this group machines suggests that the main sources of the energy radiation are: grinding belts and belt drives of work pieces to be processed and grinding units. It should be noted that sanding belts and belt drives belong to the category of flexible connections, have significant differences in comparison with other sources in terms of stiffness and tension forces. The experimental studies of the noise characteristics have obtained that the sound pressure levels have shown an identical nature of its formation of the spectral composition and exceeded the standard values in the high-frequency field of the spectrum 1000-8000 Hz [3].

2. THEORETICAL STUDY OF THE NOISE SPECTRA OF BELT DRIVES AND SANDING BELTS

The belt drives and sanding belts are flexible link drives that move at an appropriate linear speed. Therefore, the calculation of the acoustic characteristics generated by them should be performed from the equations of free vibrations of a flexible connection [4]

$$\frac{\partial^2 y}{\partial t^2} = \frac{T_0}{m_0} \cdot \frac{\partial^2 y}{\partial z^2} + \frac{q}{m_0} \qquad (1)$$

where
T is voltage, N;
m_0 is the distributed mass, kg / m;
$\frac{q}{m_0}$ is the distributed load, n / m.

It determines the stress law in the steady-state vibration mode of the belt drive and the grinding belt.
The sheave is subject to a periodic disturbing moment from the side of the cutting forces $\Delta M = M_0 \sin(\omega t)$.

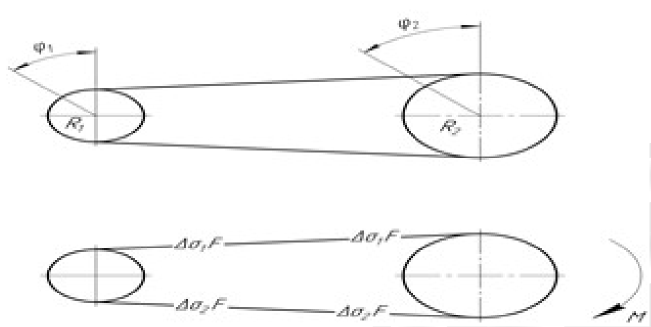

Fig. 1: The kinematic diagram of the sanding belts and belt drive

It is an elongation of the upper and lower branches of the elastic link:

$$\Delta l_1 = R_1\varphi_1 - R_2\varphi_2 = \alpha_1\Delta\sigma_1 \qquad \Delta l_2 = R_1\varphi_1 + R_2\varphi_2 = \alpha_2\Delta\sigma_2 \qquad (2)$$

where
l is the distance between the sheave axes, m;
R_1 and R_2 are the radii of the driving and driven sheaves, m;
φ is the rotation angle of the element relative to the position;
Δσ is the permissible stress in the leading and driven branches, n/mm²;
α is the angle of the belt wrap around the small sheave.

Additional stresses in the connection branches are determined from the relations:

$$\Delta l_1 = \frac{\Delta\sigma_1 l}{E} \qquad \Delta l_2 = \frac{\Delta\sigma_2 l}{E} \qquad (3)$$

where
l is the same as in formula (2), m;
E is modulus of the belt material elasticity, Pa;
Δσ is the same as in formula (2), n/mm².

From the condition of the sheave balance it follows that:

$$\begin{cases} J_1\ddot{\varphi}_1 + R_1F(\Delta\sigma_1 - \Delta\sigma_2) = 0 \\ J_2\ddot{\varphi}_2 + R_1F(\Delta\sigma_2 - \Delta\sigma_1) = M\sin\omega t \end{cases} \qquad (4)$$

where
J is the inertia moment of the work piece, m;
F is the cross-sectional area of the tape, m²;
R_1 and R_2 are the same as in formula (2), m;
Δσ is the same as in formula (2), n/mm².

In the operation, the motor torque is balanced by the cutting torque. The corresponding stresses in the branches are denoted by σ_{10} in the leading branch and σ_{20} in the main one. The branch elongations caused by additional stresses arising from sheave vibration are defined as

$$\Delta l_1 = \Delta\sigma_1 \cdot \left(\frac{l}{E} + \frac{3,3R_2}{E}(1 - l^{-2,6})\right) = \alpha_1\Delta\sigma_1$$
$$\Delta l_2 = \Delta\sigma_2 \cdot \left(\frac{l}{E} + \frac{3,3R_1}{E}(l^{-2,6} - 1)\right) = \alpha_2\Delta\sigma_2 \qquad (5)$$

where
l is the same as in formula (2), m;
E is modulus of the belt material elasticity, Pa;
R_1 and R_2 are the same as in formula (2);
Δσ is the same as in formula (2), n/mm²;
α is the angle of the belt wrap around the small sheave.

The periodic disturbing moment of cutting forces acts on the driven sheave

$$(K_c + f)(Q + G)R_1\sin(A_c nt + \varphi) \qquad (6)$$

where
n is the rotation frequency, rp/m.

In this case, the differential equations of the forced oscillations are obtained in the form:

$$\varphi_1'' + \frac{R_1^2E}{J_1}\alpha_3\varphi_2 = 0$$
$$\varphi_2'' - \frac{R_1R_2F}{J_2}\alpha_3\varphi_1 + \frac{R_2^2E}{J_2}\alpha_3\varphi_2 = (K_c + f)(Q + G)R_1\sin(A_c nt + \varphi) \qquad (7)$$

where
J_1 and J_2 are the inertia moments of the main and the driven, kg·m²;
F is the cross-sectional area of the tape, m²;
Kc is the content coefficient of the abrasive wheel;
$\alpha_3 = (\alpha_1 + \alpha_2)/\alpha_1\alpha_2$.

We find the solution of this system due to equations:

$$\varphi_1 = \varphi_{10}\, sin(0,1A_c nt + \varphi) \qquad \varphi_2 = \varphi_{20}\, sin(0,1A_c nt + \varphi) \qquad (8)$$

According to the system we get:

$$\varphi_{10} = \frac{0.3R_1R_2\alpha_3}{J_1J_2\left[\frac{v}{(A_cn)^2} - \left(\frac{R_1^2F}{J_1} + \frac{R_2^2F}{J_2}\right)\cdot\alpha_3\right]},$$
$$\varphi_{20} = \frac{0.3\left(\omega^2 - \frac{R_1^2\alpha_3}{J_1}\right)}{J_1J_2\omega^2\left[\frac{v}{(0,1A_cn)^2} - \left(\frac{R_1^2F}{J_1} + \frac{R_2^2F}{J_2}\right)\cdot\alpha_3\right]} \qquad (9)$$

The change in stresses in the tape branches is associated with the lengthening of the branches:

$$\Delta\sigma_{10} = \frac{R_1\varphi_{10} - R_2\varphi_{20}}{\alpha_1} \qquad \Delta\sigma_{20} = \frac{R_2\varphi_{20} - R_1\varphi_{10}}{\alpha_2} \qquad (10)$$

then the total stresses in the branches are determined as follows:

$$\sigma_1 = \sigma_{10} + \sigma_{10}\cdot sin\,\omega\, t \qquad \sigma_1 = \sigma_{20} + \sigma_{20}\cdot sin\,\omega\, t \qquad (11)$$

To determine the vibration velocities of the sanding belt branches as a moving flexible connection, we will use the Euler variables. From total derivatives to local ones, we get:

$$\frac{\partial y}{\partial t} = \frac{\partial y}{\partial t} + \frac{\partial y}{\partial t}\cdot\frac{\partial z}{\partial t} = \frac{\partial y}{\partial t} + v\cdot\frac{\partial y}{\partial z};$$
$$\frac{\partial^2 y}{\partial t^2} = \frac{\partial^2 y}{\partial t^2} + 2v\frac{\partial^2 y}{\partial z\partial t} + v^2\cdot\frac{\partial^2 y}{\partial z^2} \qquad (12)$$

Due to well-known vibrations` equation of a flexible connection for the case under consideration will take the form:

$$\frac{\partial^2 y}{\partial t^2} + 2v\frac{\partial^2 y}{\partial z\partial t} - \left(\frac{T}{m_0} - v^2\right)\frac{\partial^2 y}{\partial z^2} = 0 \qquad (13)$$

where
T is voltage, N;
m_0 is the distributed mass, kg / m;
υ is the linear speed of the belt, m/s.

The tension of the sanding belt branches changes in time and therefore the vibration equations for the driving and driven branches, respectively, take the form:

$$\frac{\partial^2 y}{\partial t^2} + 2v\frac{\partial^2 y}{\partial z \partial t} = \left(\frac{F\sigma_{10}}{m_0} + \frac{F\Delta\sigma_1}{m_0} sin(0,1A_c nt + \varphi - V^2)\right)\frac{\partial^2 y}{\partial z^2}$$
$$\frac{\partial^2 y}{\partial t^2} + 2v\frac{\partial^2 y}{\partial z \partial t} = \left(\frac{F\sigma_{20}}{m_0} + \frac{F\Delta\sigma_2}{m_0} sin(0,1A_c nt + \varphi - V^2)\right)\frac{\partial^2 y}{\partial z^2} \quad (14)$$

From this system, the $\left(\frac{\partial y}{\partial t}\right)$ vibration velocities of the belt drive on their own vibration modes are determined.

The natural vibration frequencies of the flexible connection are determined by the formula:

$$f_k = \frac{k}{2l}\sqrt{\frac{T}{\rho F}}\left(1 + \frac{\rho F v^2}{T}\right) \quad (15)$$

For the acoustic model of a linear source, the sound pressure ($P_{s.p.}$) and sound pressure levels (L_p) of the belt drive are determined:

$$P_{s.p.} = 6,7 \cdot \frac{v_k}{r}\left[khl\left(1 + \frac{\rho F v^2}{T}\right)\right] \cdot \left(\frac{T}{\rho F}\right)^{0,25}$$
$$L_p = 20\, lg\left(\frac{v_k}{r}\right) + 10\, lg\left(khl\left(1 + \frac{\rho F v^2}{T}\right)\right) + 5\, lg\left(\frac{T}{\rho F}\right) + 110, \quad (16)$$

where
k is a coefficient that determines the natural frequencies of the belt vibration;
h is the belt width, m;
l is the distance between the sheave axes, m;
r is a distance to the design point, m;
υ is the linear speed of the belt, m/s;
F is the cross-sectional tape area, m^2;
ρ is the density of the belt material, kg/ m^3;
T is voltage, N.

The difference in the calculations of the sound pressure levels of grinding belts when processing parts on grinding machines consists in a significant difference in the geometric configurations of the grinding belts and the processed products.

3. CALCULATION OF VIBRATIONS AND SOUND PRESSURE LEVELS OF SANDING BELTS

The calculation of vibrations and, accordingly, the sound pressure levels of the tapes is based on the equations of vibrations of a bent bar with a radius of curvature ρ_0

$$m_0\frac{\partial^2\omega}{\partial t^2} = \frac{\Delta N}{\partial S} - \frac{\Delta Q_y}{\rho_0} - Q_0\frac{\partial\Delta\varphi}{\partial S} + \Delta q_S$$
$$m_0\frac{\partial^2 u}{\partial t^2} = \frac{\partial\Delta Q_y}{\partial S} + \frac{\Delta N}{\rho_0} + N_0\frac{\partial\Delta\varphi}{\partial S} + \Delta q_y$$
$$\frac{\partial^2\omega}{\partial t\partial S} - \frac{1}{R}\frac{\partial u}{\partial t} = 0$$
$$\frac{\partial^2 u}{\partial t\partial S} + \frac{1}{R}\frac{\partial\omega}{\partial t} = \frac{\partial\Delta\varphi}{\partial t}$$
$$-EI_x\frac{\Delta\partial^2\Delta\varphi}{\partial S^2} = -\Delta Q_y$$
$$-\Delta Q = \frac{\partial M}{\partial S}; \quad \mu = EI\left(\frac{1}{\rho} - \frac{1}{\rho_0}\right) \quad (17)$$

where
ω and u are the displacement of the bar member;
E and **I** - modulus of elasticity (Pa) and moment of inertia, m^4;
Q is a shearing force, n;
M is a bending moment, n·m;
Δφ is the rotation angle of the element relative to the position;
N is a force impact, n;
R radius of the grinded product, m.

The differential equations of vibrations of a curved sanding belt are a special case, for flexible communication it is enough to accept that **E·I = 0**, and also $Q_{y0} = 0$; $N_0 = -P_{shl}F_s\, sin(0,1A_c nt + \varphi)$; $\Delta q_S = \Delta q_y = 0$; $\rho_0 = R$.

Then the oscillation equation of the sanding belt will take the form:

$$m_0\frac{\partial^4 u}{\partial S^2\partial t^2} - \frac{m_0}{R^2}\frac{\partial^2 u}{\partial t^2} - R\frac{P_{shl}\, sin(0,1A_c nt + \varphi)}{F_c} \cdot \left(\frac{\partial^4 u}{\partial S^4} + \frac{1}{R^2}\frac{\partial^2 u}{\partial S^2}\right) = 0 \quad (18)$$

The solution to the equation by the separation method of variables is given in the following form:

$$u = \sum_k f(t)\, sin\frac{kS}{R} \quad (19)$$

We perform the appropriate transformations and substitute the expressions for the derivatives into the equation

$$\frac{\partial^4 u}{\partial S^4} = \sum_k f(t)\left(\frac{k}{R}\right)^4 sin\frac{kS}{R};$$
$$\frac{\partial^2 u}{\partial S^2} = -\sum_k f(t)\left(\frac{k}{R}\right)^4 sin\frac{kS}{R} \quad (20)$$
$$\frac{\partial^2 u}{\partial t^2} = \sum_k f''(t)\, sin\frac{kS}{R}$$

And we obtain the following equation

$$f''(t) - \left(\frac{20}{A_C n}\right)^2 \sum_{k_b^*}\frac{P_{shl}\cdot k^2(k^2-1)}{F_s R\cdot m_0(k^2+1)} sin(0,1A_C nt + \varphi) = 0 \quad (21)$$

where
Fs is the area covered by the tape of the processed product, м^2;
K^*_b - a number of sanding belts;
m0 is the distributed mass, kg/m;
P_{shl} is the amplitude of the force action from the side of the grinding belt, N;
φ is the rotation angle of the element relative to the position;
n is the rotation frequency of the grinding belt drum, rp/m;
R is the radius of the grinded product, m.

From this equation, the values of the vibration velocities are determined by numerical methods and, in relation to a linear source, the sound pressure levels

$$L = 20\, lg\frac{v_k}{r} + 10\, lg\left(f_k h_b l_b^2\right) + 114 + 10\, lg\, k_b \quad (22)$$

where
h_b and l_b are the width and length of the tape, m.

The work pieces to be ground are cylindrical emitters and, therefore, the sound pressure levels are determined by the relationship

$$L = 20\, lg \frac{9{,}5 v_k (f_k 2\pi R l^2)^{0{,}5}}{2 \cdot 10^{-5} r} \qquad (23)$$

where
l is the length of the work piece, m.

For fixing conditions corresponding to a hinged-supported work piece, the natural frequencies are determined by the well-known formula

$$f_k = \frac{\pi}{2}\left(\frac{k}{l}\right)^2 \cdot \sqrt{\frac{EI}{\rho F}} \qquad (24)$$

To consider the numerical values of the $\frac{E}{\rho}$ ratios, the dependences for calculating the natural frequencies of vibrations and noise levels will take the form:

$$f_k = 3{,}7 \cdot 10^4 R \left(\frac{k}{l}\right)^2 \qquad (25)$$

$$L = 20\, lg \frac{v_k}{r} Rk + 128 \qquad (26)$$

The vibration rates are determined from the equation

$$\frac{d^2 y}{dt^2} + 8 \cdot 10^{11} \left(\frac{k}{l}\right)^4 R^2 y = \frac{5 \cdot 10^{-2}}{\rho R^2 l} \sum_{k_b} \sum_{k} sin \frac{\pi k x_i}{l} \cdot sin(0{,}1 A_C n t + \varphi) \qquad (27)$$

where
x_i are the coordinates of the grinding belts location on the work piece being processed, m.

The solution of the equation with respect to the maximum values of the vibration velocities is determined by the dependence

$$v_k = \frac{5 \cdot 10^{-3} P A_C n}{\rho R^2 l} \sum_{k_b} \sum_{k} \frac{\left[8 \cdot 10^{11} \left(\frac{k}{l}\right)^4 R^2 - (0{,}1 A_C n)^2\right] sin \frac{\pi k x_i}{l}}{\left[8 \cdot 10^{11} \left(\frac{k}{l}\right)^4 R^2 - (0{,}1 A_C n)^2\right]^2 + 6{,}4 \cdot 10^{23} \left(\frac{k}{l}\right)^8 R^4 \eta^2} \qquad (28)$$

The sound pressure levels are found using the formula

$$L = 20\, lg \frac{P A_C n}{\rho R l r} \sum_{k_b} \sum_{k} \frac{\left[8 \cdot 10^{11} \left(\frac{k}{l}\right)^4 R^2 - (0{,}1 A_C n)^2\right] sin \frac{\pi k x_i}{l}}{\left[8 \cdot 10^{11} \left(\frac{k}{l}\right)^4 R^2 - (0{,}1 A_C n)^2\right]^2 + 6{,}4 \cdot 10^{23} \left(\frac{k}{l}\right)^8 R^4 \eta^2} \qquad (29)$$

The sound pressure levels or sound power of several sources emitting sound simultaneously is determined by the principle of energy summation

$$L_\Sigma = 10\, lg \sum_{1}^{k_4} 10^{0{,}1 L_i} \qquad (30)$$

where
k_4 is the number of sources;
L_i is the sound pressure levels of the ***i-th*** source, dB.

The obtained dependencies give calculating the levels and consider both the layout of the machine drive and the load characteristics. The comparison of the expected noise levels with sanitary standards determines the excess and the sources that generate them. Consequently, this information is the main one for choosing engineering solutions to ensure sanitary noise standards.

These data make it possible to significantly clarify the formation of the noise characteristics and theoretically substantiate rational options for vibration damping of dominant noise sources, and, factually, to ensure noise reduction in the source itself to the maximum permissible values.

4. CONCLUSION

The obtained results enable, theoretically, at the design stage of such machines to determine the expected sound pressure levels generated by grinding work pieces and especially important is the excess over the maximum permissible values in the corresponding octaves. It is these data that make it possible to theoretically substantiate noise and vibration protection systems in accordance with the implementation of the sanitary standards.

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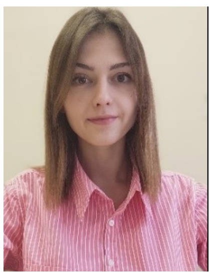

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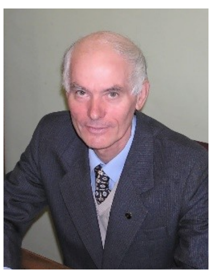

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