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THEORETICAL STUDY OF THE VIBRATION EXCITATION AND NOISE GENERATION PROCESSES OF THE GRINDING WHEELS OF THREAD- AND SPLINE GRINDING MACHINES

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***Abstract**: The competitiveness of machine-building products is largely determined by the accuracy of mechanical processing of the manufactured parts and the state of the surface layer. The condition of the surface layer is carried out by finishing operations, such as grinding. The volume of grinding operations is from 25 to 60 % of various technological operations. Working conditions in the grinding areas are considered harmful and dangerous. [1]. The noise levels at the workplaces of these machines' operators exceed the standard values. The sound emission from the part during grinding is a particular feature of this process. A theoretical study of the noise generation processes on such machines was carried out to develop the recommendations.*

***Keywords:** Noise generation during grinding, vibrations, sound energy, noise level, sound radiation*

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1. INTRODUCTION

The kinematic features of the modern thread-grinding and spline-grinding machines are the low rotation frequencies of the workpieces and using hydrostatic bearings in high-frequency grinding heads, as well as stepless drives, which suggests that the main sources of sound energy radiation and excess of the sound pressure levels at the operator's workplaces over the sanitary standards are grinding wheels and workpieces being processed.

2. JUSTIFICATION OF THE NOISE SOURCE MODELS

The geometric configurations of such radiating elements allow to use two types of noise sources:
- a round plate fixed in the center for grinding wheels;
- a beam of limited length for both threaded parts and spline shafts to be processed.

Grinding wheels are cantilever-fitted round discs mounted on the spindle of the grinding head (Fig. 1)

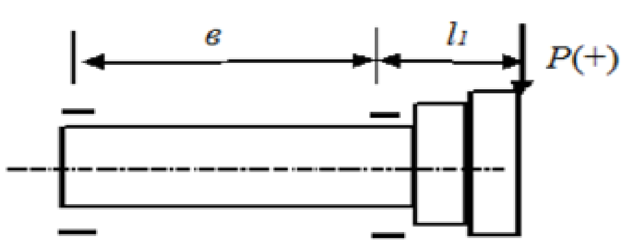

Fig. 1: Layout of the cutting unit

The sound pressure (***P***) and sound power (***N***) of such a source according to the research data is determined by the expressions:

$$P = \frac{R^2 \omega\, \rho_0 V_k}{2r} \quad \text{and} \quad N = \frac{\pi\, R^2 \rho_0\, C_0\, (k_0\, R)^2 V_k^2}{2} \tag{1}$$

where
- R is the radius of the circle, m
- $\rho_0 C_0$ is the air density (kg/m^3) and speed of sound in the air (m/s);
- ω circular oscillation frequency, rad/s;
- k_0 is the wave number, m^{-1};
- r is the distance from the noise source to the reference point, m;
- V_k is the oscillation speed of the circle, m/s.

Given the known physical and mechanical characteristics and data of paper [2] the expression levels of sound pressure and sound power were obtained:

$$L_p = 20\ lg \frac{V_k f_k}{2} + 40\ lgR + 126 \tag{2}$$

$$L_N = 20\ lg\ V_k\ f_k + 40\ lgR + 113 \tag{3}$$

where
f_k is the natural frequency of the noise source, Hz.

Two options for calculating the natural frequencies and velocities of the oscillations are considered in this section.

2.1. First calculation option

The first option takes into account the layout of the cutting unit according to Fig. 1. For such a scheme, it is advisable to use an approach based on the functions of A.N. Krylov [3]. Then, for the sections of the cantilever (cutting tool) and the inter-bearing part, the deflection expressions are defined as follows:

$$\begin{aligned} y_1 &= C_1K_1(\lambda x) + C_2K_2(\lambda x) + C_3K_3(\lambda x) + C_4K_4(\lambda x), \quad if \quad 0 \le x \ge b \\ y_2 &= C_1'K_1(\lambda x) + C_2'K_2(\lambda x) + C_3'K_3(\lambda x) + C_4'K_4(\lambda x), \quad if \quad b \le x \ge l_1 \end{aligned} \tag{4}$$

where

y_1 and **y_2** correspond to deflections in the direction of the cutting force component

P_y and in the direction of the cutting force component **P_z**,

b and **l_1** on fig. 1,

C_1, C_2, C_3, C_4 are the integration constants

$K_1(\lambda x)$, $K_2(\lambda x)$, $K_3(\lambda x)$, $K_4(\lambda x)$ are the Krylov functions, defined as follows:

$$\begin{aligned} K_1(\lambda x) &= \frac{1}{2}(ch(\lambda x) + \cos(\lambda x)); \\ K_2(\lambda x) &= \frac{1}{2}(ch(\lambda x) + \sin(\lambda x)); \\ K_3(\lambda x) &= \frac{1}{2}(ch(\lambda x) - \cos(\lambda x)); \\ K_4(\lambda x) &= \frac{1}{2}(ch(\lambda x) - \sin(\lambda x)); \\ \lambda &= 2,5\, f_k^{0.5} \left(\frac{\rho F}{EY}\right)^{0.25} l_1 \end{aligned} \tag{5}$$

where

f_k is the natural frequency oscillations, Hz;

ρ is the density of the material, kg/m^3;

Y is the moment of inertia, m^4;

F is the cross – sectional area, m^2.

The integration constants are determined from the boundary conditions. The matching conditions, i.e., the equality of deflections, angles of rotation and bending moments for both sections of the cutting unit, as well as the jump of the transverse force equal in magnitude to the amplitude of the force action must be met at the point of application of the technological load.

In the first section: $y_1(b) = y_2(0) = 0$ so $C_1 = C_2 = 0$.

$$y_1(b) = y_2(b) \tag{6}$$

$$\left.\frac{\partial y_1}{\partial x}\right|_{x=b} = \left.\frac{\partial y_2}{\partial x}\right|_{x=b} \tag{7}$$

As the transverse forces in the end of the first and beginning of the second sections differ by the magnitude of the bearing reaction, taking into account the fact that the spindle is made of steel, and the moment of inertia, the following expression is obtained:

$$D_1^4 \left.\frac{\partial^3 y_1}{\partial x^3}\right|_{x=b} = (D_2^4 - d_2^4) - 9{,}7 \cdot 10^{-11} R \tag{8}$$

where

R is the reaction in the spindle bearing, N;

D_1 is the diameter of the grinding wheel, m;

D_2 and **d_2** are the outer and inner diameter of the spindle, m.

Bearing reactions are obtained using a traditional method and are determined for the bearings A and B:

$$R_A = P_p(t)\frac{\alpha+1}{\alpha} \tag{9}$$

$$R_B = P_p(t)\frac{1}{\alpha} \tag{10}$$

where

$\frac{b}{l_1}$; ***P(t)*** is the cutting force, N.

The expression for the deflection in this case will take the form:

$$y_1 = \frac{D_1^4}{D_2^4 - d_2^4}[C_2K_2(\lambda x) + C_4K_4(\lambda x)]\, \frac{P(t)\frac{\alpha+1}{\alpha} \cdot 10^{-10}}{\lambda^3(D_2^4 - d_2^4)} \tag{11}$$

The oscillation velocity in this case is determined by the dependence:

$$V_k = \frac{dy_1}{dt} \tag{12}$$

The resulting expression for **y_1** contains the constants **C_2**, **C_4**, and **R**, which can be determined from the conditions:

$$x = b, \quad y = 0,$$

$$x = l_1, \quad y = y = 0$$

As a result, we get a system of three equations:

$$\begin{aligned} &C_2K_2(\lambda x) + C_4K_4(\lambda x) = 0 \\ &\frac{J_1}{J_2}[C_2K_4(\lambda l_1) + C_4K_2(\lambda l_1)] + \frac{R}{\lambda^3 E J_2} K_2[\lambda(l_1 - b)] = 0 \end{aligned} \tag{13}$$

The natural frequencies of the oscillations are obtained from the determinant

$$\begin{vmatrix} ch\lambda x + \sin\lambda x & ch\lambda x - \sin\lambda x & 0 \\ \frac{J_1}{J_2}(ch\lambda l_1 - \sin\lambda l_1) & \frac{J_1}{J_2}(ch\lambda l_1 + \sin\lambda l_1) & \frac{ch\lambda(l_1-b)+\sin\lambda(l_1-b)}{\lambda^3 E J_2} \\ \frac{J_1}{J_2}(ch\lambda l_1 - \cos\lambda l_1) & \frac{J_1}{J_2}(ch\lambda l_1 + \cos\lambda l_1) & \frac{ch\lambda(l_1-b)+\sin\lambda(l_1-b)}{\lambda^3 E J_2} \end{vmatrix} = 0 \tag{14}$$

According to the standards of cutting modes [6], the cutting force is determined using the formula:

$$P_p = \frac{N_p}{V_p} \qquad (15)$$

where
Np – is the cutting power,

$V_p = \frac{\pi R n}{30}$ – is the cutting speed, m/s; n is the rotation speed, rpm,

$$N = C_N V_3^{k_1} S_p^{k_2} d^{k_3} b^{k_4} \cdot 10^3 \cos(0{,}1n\, K_3 + \varphi) \qquad (16)$$

where
V_3 – is the workpiece speed of rotation;
S_p – is the radial flow;
d – is the circle diameter;
b – is the circle width;
K_3 – is the coefficient of a circle grit;
$k_2 \ldots k_4$ and ***C_N*** – coefficients are specified in the tables of cutting conditions standards [4], $\varphi = \mathrm{arctg}\frac{V_3}{V_p}$

The scheme corresponds to the grinding of the thread with a multi-thread circle of a cantilever-fixed part. In this case, the cutting power is defined as

$$N = C_N V_3^{k_1} t^{k_2} S_{bp}^{k_3} d^{k_4} \cdot 10^3 \cos(0{,}1n\, K_3 + \varphi) \qquad (17)$$

where
t – is the cutting depth, mm;
S_{bp} – is the movement of the grinding wheel in the direction of its axis (mm) for one revolution of the workpiece.

2.2. Second calculation option

The second method (simplified) is based on the fact that sound energy inter-bearing part of the spindle is emitted in the internal air volume of the wheelhead body [5,6] and due to its high insulation has no effect on the formation of the sound field at the workplaces of the rolling and spline grinding machine operators. In this case, the radiation of the grinding wheel itself is taken into account, the oscillation velocity of which is determined from the differential equation

$$m\frac{d^2Y}{dt^2} + \frac{2m\delta_0}{T}\frac{dy}{dt} + C_y = P(t) \qquad (18)$$

where
m – is the circle mass, kg;
T – is the oscillation period, s;
δ_0 – is the logarithmic decrement of oscillations, equal to 0.32 for grinding mandrels according to the data of paper [3];
C – is the system stiffness: $C = \frac{3EJ_2}{l_1^3}$;
E – is the elastic modulus, Pa;
I – is the moment of inertia, m⁴.

The natural frequency of the grinding wheel is modified as:

$$f_h = \frac{k}{2R}\sqrt{\frac{E}{\rho b}} \qquad (19)$$

where
k – is the coefficient defining the natural frequency of a circle;
E – is the modulus of elasticity, PA;
ρ – is the density of the circle material, kg/m³.

Then the equation will take the form

$$\frac{d^2Y}{dt^2} + 5\cdot 10^{-2}\frac{k}{R}\sqrt{\frac{E}{\rho h}}\frac{dy}{dt} + 0{,}75\frac{l^2E}{\rho l_1^3 h}\, y = \frac{0{,}32\, P_p}{\rho R^2 h}\cos(0{,}1n\, K_3\, t + \varphi) \qquad (20)$$

From this equation, a partial solution is found with respect to the modulus of the oscillation velocity

$$|V_{kr}| = \frac{3{,}2\cdot 10^{-5} P n K_3}{\rho R^2 h}\sum\frac{\sin(0{,}1\, n\, K_3\, t + \varphi)}{\sqrt{\left(75\frac{R^2\, E}{\rho R^2 h} - n^2 K_3^2\right)^2 + 25\cdot 10^{-2}\frac{R^2 E n^2 K_3^2}{\rho l_1^3 h}}} \qquad (21)$$

where
h – is the thickness of the grinding wheel, m.

The general solution is obtained taking into account the assumptions that:

1. $0{,}05\frac{k}{R}\sqrt{\frac{E}{\rho h}} \ll 0{,}75\frac{R^2\, E}{\rho R^2 h}$
2. the deflection of the cantilever part consists of the elastic displacements of the spindle bearing deformation [70]

$$Y_1 = \frac{P}{j_A}\left(\frac{\lambda+1}{\lambda}\right)^2 + \frac{P}{j_B}\frac{1}{\lambda^2} \qquad (22)$$

where
j_A and ***j_E*** – are the stiffness of the front and rear bearings, respectively, n/ m;
$\lambda = \frac{b}{l_1}$ as well as the deflection of the cutting unit as an elastic beam

$$Y_2 = \frac{P\, l_1^3\, b}{3E_1 J_1} + \frac{P\, l_1^3}{3E_2 J_2} \qquad (23)$$

where
I_1 and ***I_2*** – are the moments of inertia of the inter-piston canti lever sections of the spindle, m⁴.

Then the equation of free oscillations and the general solution with respect to the modulus of the oscillation velocity are defined as

$$\frac{d^2y}{dt^2} + 0{,}75\frac{R^2E}{\rho l_1^3 h}\, y = 0 \qquad (24)$$

$$|V_{k_0}| = \left[\frac{P}{j_A}\left(\frac{\lambda+1}{\lambda}\right)^2 + \frac{P}{j_B}\frac{1}{\lambda^2} + \frac{P\, l_1^3\, b}{3E_1J_1} + \frac{P\, l_1^3}{3E_2J_2}\right]\cdot 0{,}9R\sqrt{\frac{E}{\rho l_1^3 h}}\sin 0{,}9R\sqrt{\frac{E}{\rho l_1^3 h}}\, t \qquad (25)$$

In this case, the oscillation velocity is determined by the expression:

$$|V_k| = \left[\frac{P}{j_A}\left(\frac{\lambda+1}{\lambda}\right)^2 + \frac{P}{j_B}\,\frac{1}{\lambda^2} + \frac{P\,l_1^3\,b}{3E_1\mathcal{J}_1} + \frac{P\,l_1^3}{3E_2\mathcal{J}_2}\right] 0{,}9R\sqrt{\frac{E}{\rho l_1^3 h}}\sin 0{,}9R\sqrt{\frac{E}{\rho l_1^3 h}}t \;+ \frac{3{,}2\cdot 10^{-5}PnK_3}{\rho R^2 h} * \sum \frac{\sin(0{,}1\,n\,K_3\,t+\varphi)}{\sqrt{\left(75\frac{R^2\,E}{\rho R^2 h} - n^2K_3^2\right)^2 + 25\cdot 10^{-2}\frac{R^2 E n^2 K_3^2}{\rho l_1^3 h}}} \tag{26}$$

The oscillation velocity of a cantilever-fixed part is determined by a similar expression, taking into account the radius of the part (R_g), the length of the part (bg), the cantilever part (l_{1g}) and the elastic modulus.

3. CONCLUSION

The obtained dependences allow us to theoretically determine the oscillation velocities of grinding wheels and cantilevered workpieces at their natural oscillation frequencies, use in the formula of sound pressure levels or sound power, and actually determine the levels of the spectral noise components. The obtained theoretical values can be used in the calculation of the operator's workplace noise and for development of the noise reducing structures.

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