

Theoretical studies of vibrations and noise of grinding belt machines

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Abstract. The operation of belt grinding machines is usually accompanied by a significant level of noise and vibration, which adversely affect the health of operators and the accuracy of processing the final product. It is known that in the woodworking industry there is an increased risk of myocardial infarction due to significant noise in the workplace. The purpose of the article is a theoretical analysis of the occurrence of noise and vibrations of belt grinding machines and the construction of appropriate mathematical models. The influence of various noise sources in the machines under consideration was analyzed: electric motors, belt drives, grinding belts, axes of cutting units, and workpieces being ground. The solution of the vibration equations of the tape and the workpiece made it possible to obtain an analytical solution for the vibroacoustic characteristics. The obtained solutions are in satisfactory agreement with the experimental data and allow the design to develop engineering solutions that reduce the sound pressure level.

1 Introduction

The production noise that occurs during the processing of parts is a negative process that affects the health of the operator and reduces the accuracy of the final product. According to data [1], an increased risk of acute myocardial infarction is associated with the workplace, with increased incidence rates being found in the forestry and woodworking industries. Prolonged exposure to vibrations leads to an occupational disease that develops after 3–7 years of work in vibration conditions and affects peripheral vessels, the nervous and musculoskeletal systems [2]. Thus, the reduction of noise and vibration in the processing of parts is an important scientific problem and attracts the attention of researchers.

Vibration reduction of unbalanced rotor systems was investigated in [3]. The authors proposed to control the movement of the rotor by quantifying the active bearing force and phase when an active bearing was applied in a simple rotor model. The simulation was carried out on the basis of a multi-mass system with lumped parameters in the form of a system of ordinary differential equations.

The diagnostics of operation of rolling bearings of mine hoists is investigated in [4]. The authors propose a method for adaptive diagnostics of rolling bearing failures, which

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combines a variational modal decomposition model and a deep learning network model. The proposed method allows, with an accuracy of more than 92%, to assess their performance.

The vibroacoustic characteristics of a frequency-controlled pump were investigated in [5] to ensure stable operation of the water supply. Vibration signals are processed by smoothing, shaping and filtering based on the Fast Fourier Transform. A significant array of experimental data allowed the authors to identify the main features of the signal and develop a method for diagnosing the pump performance. Similar approach was used in [6].

The noise that adds up to the useful signal is also a negative component that makes it difficult to diagnose the units. To effectively combat noise, a method for diagnosing a rolling bearing failure is proposed in [7], based on signal decomposition and an optimization algorithm for the variational mode.

The vibroacoustic characteristics of a locally resonant plate loaded with liquid were studied in [8] to analyze the vibration and sound radiation of a structure. Analysis of the noise spectrum made it possible to identify areas of high sound pressure and develop measures to reduce noise. It has been shown that adding a small amount of damping to the resonator can reduce vibration and sound emission in the frequency range above or closer to the bandgap.

Vibration from the gear train of marine vessels is transmitted to the base of the ship through the transmission parts and the hull, which produces underwater noise and negatively affects the performance of the vessel. The analysis of vibration properties [9] and the determination of the optimal means of vibration protection allowed the authors to develop a low-noise gear train design.

Increased vibration and noise in agricultural machinery due to high power reduces productivity and adversely affects the health of the operator. To reduce the vibration of the grinder [10], the vibration characteristics of the spindle rotor were first analyzed using modal FEM simulation. Then, engineering solutions are proposed through the use of vibration-absorbing materials and noise-insulating coatings.

Vibration state monitoring, as well as a non-destructive testing method, has been proposed in [11] to detect cracks in teeth propagating in gears. The authors proposed to apply a methodology using time-domain analysis, frequency-domain analysis, and statistical control charts in the gears of a dynamic model of a helical-bevel gear system. This approach made it possible to diagnose cracks at an early stage. A similar smart diagnostic method was proposed in [12] to ensure the safety and stability of rotating equipment.

Modeling of noise generation processes in many articles is carried out using the hypothesis of a multi-mass system with viscous damping [13-15]. This approach makes it possible to obtain a system of linear ordinary differential equations, which is easily solved and gives a stable solution. However, the application of this approach is associated with errors, since the tape working bodies have a distributed mass, and not a concentrated one [16-19]. Article [20] shows that the application of the distributed mass approach leads to a more accurate statement of the problem and the solution is reduced to a partial differential equation.

The review of the literature shows that the analysis of noise and vibration is widely used in the problems of diagnostics and monitoring of various parts and assemblies. The analysis of the time-frequency signal makes it possible to judge the performance of the structure. At the same time, a promising direction is the analytical representation of the vibroacoustic response. The purpose of this article is a theoretical study of the noise and vibration of belt grinding machines.

2 Materials and methods

The following schemes of the kinematics of machine tools and the technological process of grinding are considered (Figure 1).

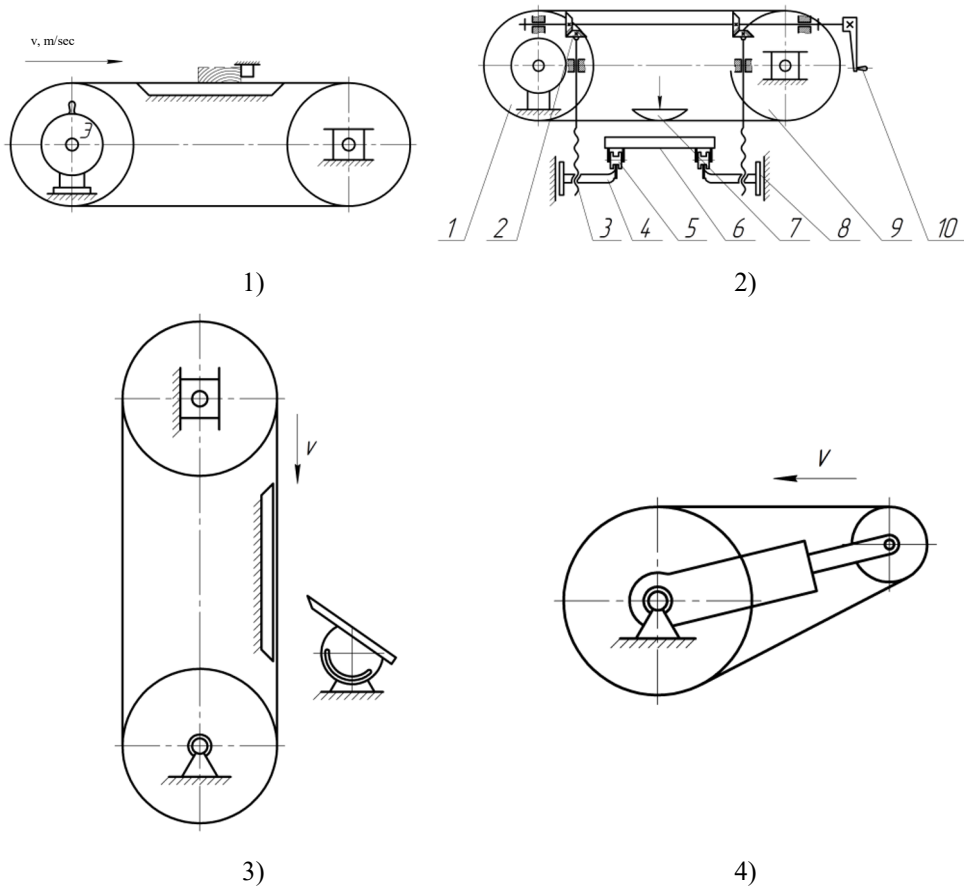


Fig. 1. Kinematic diagrams of belt grinding machines: 1) horizontal belt grinding machine with a fixed belt support; 2) a horizontal belt grinder with a movable table and a contact clamp of the tape: 1 - the drive pulley of the cutting unit; 2, 3 and 10 - the mechanism for moving the table 6; 4 – table support brackets 6; 5 - running wheels of the table; 7 - iron clamp sanding belt 9; 8 - guides for moving the table; 3) vertical belt sander with fixed belt support 4) free belt sander.

The sources of noise in the machines under consideration are: motor rotors and bearings, belt drives, grinding belts, workpieces being ground.

Belt drives and grinding belts are flexible-coupled gears that move at the appropriate linear speed. Therefore, the calculation of the acoustic characteristics created by them should be performed from these positions.

The equation of free vibrations of a flexible connection

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

where $\frac{2}{m_0}$ is the distributed load.

The pulley is subjected to a periodic perturbing moment from the cutting forces $\Delta M = M_0 \sin(\omega t)$.

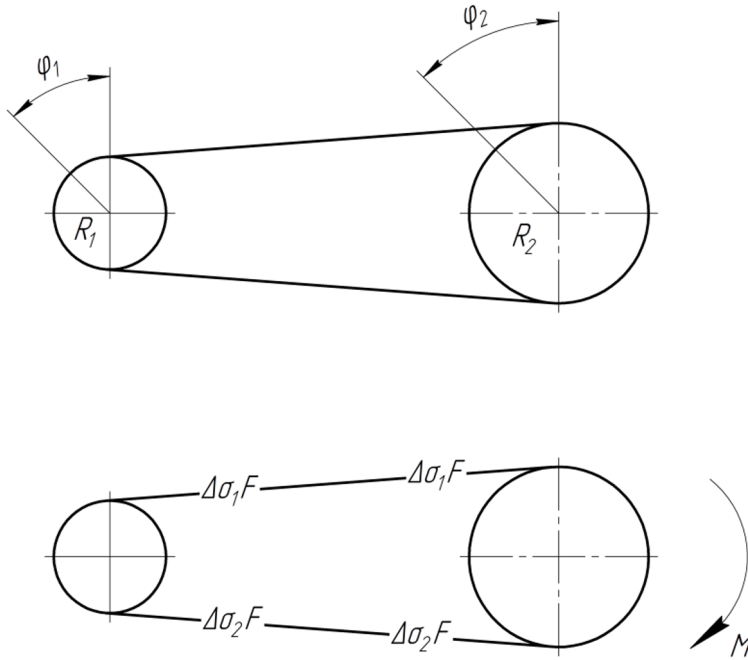


Fig. 1. Scheme of forces and moments of a belt drive

Elongation of the upper and lower branches of the elastic connection:

$$\begin{aligned}\Delta l_1 &= R_1 \varphi_1 - R_2 \varphi_2 = \alpha_1 \sigma_1, \\ \Delta l_2 &= -R_1 \varphi_1 + R_2 \varphi_2 = \alpha_2 \sigma_2.\end{aligned}\quad (2)$$

Additional voltages in the communication branches are determined from the relations

$$\begin{aligned}\Delta l_1 &= \frac{\Delta \sigma_1 l}{E}, \\ \Delta l_2 &= \frac{\Delta \sigma_2 l}{E}.\end{aligned}\quad (3)$$

From the balance condition of the pulleys, it follows that:

$$\begin{cases} J_1 \ddot{\varphi}_1 + R_1 F (\Delta \sigma_1 - \Delta \sigma_2) = 0 \\ J_2 \ddot{\varphi}_2 + R_2 F (\sigma_2 - \Delta \sigma_1) = M \sin \omega t \end{cases}\quad (4)$$

In the steady state of operation, the torque of the motor is balanced by the torque of the cutting forces. We denote the corresponding voltages in the branches: σ_{10} - in the leading branch and σ_{20} - in the driven one. The extensions of the branches caused by additional stresses arising from the vibration of the pulleys are defined as:

$$\begin{aligned}\Delta l_1 &= \Delta \sigma_1 \cdot \left(\frac{l}{E} + \frac{3.3 R_2}{E} (1 - l^{-2.6}) \right) = \alpha_1 \Delta \sigma_1 \\ \Delta l_2 &= \Delta \sigma_2 \cdot \left(\frac{l}{E} + \frac{3.3 R_1}{E} (l^{-2.6} - 1) \right) = \alpha_2 \Delta \sigma_2,\end{aligned}\quad (5)$$

where l is the distance between the axes of the pulleys, m; E is the modulus of elasticity of the belt material, Pa; R_1 and R_2 are the radii of the driving and driven pulleys, m.

The driven pulley is affected by a periodic perturbing moment of cutting forces $(K_c + f)(Q + G)R_1 \sin(0.1A_c n t + \varphi)$, where n is the rotational speed, rpm.

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

$$\begin{aligned} \varphi_1'' + \frac{R_1^2 E}{J_1} \alpha_3 \varphi_1 - \frac{R_1 R_2 F}{J_1} \alpha_3 \varphi_2 &= 0 \\ \varphi_2'' - \frac{R_1 R_2 F}{J_2} \alpha_3 \varphi_1 + \frac{R_2^2 E}{J_2} \alpha_3 \varphi_2 &= (K_c + f)(Q + G)R_1 \sin(0.1A_c n t + \varphi), \end{aligned} \quad (6)$$

where J_1 and J_2 are the moments of inertia of the master and slave, $\text{kg} \times \text{m}^2$; F is the cross-sectional area of the tape, m^2 ;

3 Results

The resulting equation describes the forced oscillations of the grinding machine belt under a periodic perturbing action of the cutting force. We are looking for the solution of this system of equations in the form:

$$\begin{aligned} \varphi_1 &= \varphi_{10} \sin(0.1A_c n t + \varphi) \\ \varphi_2 &= \varphi_{20} \sin(0.1A_c n t + \varphi) \end{aligned} \quad (7)$$

From the system we obtain

$$\nu = 10^{-2} (A_c n)^2 \quad (8)$$

$$\begin{aligned} \varphi_{10} &= \frac{0.3 R_1 R_2 \alpha_3}{J_1 J_2 \left[\frac{\nu}{(A_c n)^2} - \left(\frac{R_1^2 F}{J_1} + \frac{R_2^2 F}{J_2} \right) \cdot \alpha_3 \right]}, \\ \varphi_{20} &= \frac{0.3 \left(\omega^2 - \frac{R_1^2 \alpha_3}{J_1} \right)}{J_1 J_2 \omega^2 \left[\frac{\nu}{(A_c n)^2} - \left(\frac{R_1^2 F}{J_1} + \frac{R_2^2 F}{J_2} \right) \cdot \alpha_3 \right]}. \end{aligned} \quad (9)$$

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

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

$$\Delta \sigma_{20} = \frac{R_2 \varphi_{20} - R_1 \varphi_{10}}{\alpha_2} \quad (11)$$

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

$$\begin{aligned} \sigma_1 &= \sigma_{10} + \sigma_{10} \cdot \sin \omega t \\ \sigma_2 &= \sigma_{20} + \sigma_{20} \cdot \sin \omega t. \end{aligned} \quad (12)$$

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

$$\begin{aligned}\frac{\partial y}{\partial t} &= \frac{\partial y}{\partial t} + \frac{\partial y}{\partial z} \cdot \frac{\partial z}{\partial t} = \frac{\partial y}{\partial t} + v \cdot \frac{\partial y}{\partial z}; \\ \frac{\partial^2 y}{\partial t^2} &= \frac{\partial^2 y}{\partial t^2} + 2v \frac{\partial^2 y}{\partial z \partial t} + v^2 \cdot \frac{\partial^2 y}{\partial z^2},\end{aligned}\quad (13)$$

where v is the linear speed of the belt, m/s.

Then the well-known equation of flexible connection oscillations for the case under consideration will take the form:

$$\frac{\partial^2 y}{\partial t^2} + 2v \frac{\partial^2 y}{\partial z \partial t} - \left(\frac{T}{m_0} - v^2 \right) \frac{\partial^2 y}{\partial z^2} = 0, \quad (14)$$

where m_0 is the distributed mass, kg/m; T is belt tension force, N.

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

$$\begin{aligned}\frac{\partial^2 y}{\partial t^2} + 2v \frac{\partial^2 y}{\partial z \partial t} &= \left(\frac{F\sigma_{10}}{m_0} + \frac{F\Delta\sigma_1}{m_0} \sin(0.1A_c n t + \varphi) \right) \frac{\partial^2 y}{\partial z^2}, \\ \frac{\partial^2 y}{\partial t^2} + 2v \frac{\partial^2 y}{\partial z \partial t} &= \left(\frac{F\sigma_{20}}{m_0} + \frac{F\Delta\sigma_2}{m_0} \sin(0.1A_c n t + \varphi) \right) \frac{\partial^2 y}{\partial z^2}.\end{aligned}\quad (15)$$

From this system, the oscillation speeds of the belt drive are determined on their own modes of oscillation.

The obtained dependences allow to determine the levels by calculation and take into account both the layout solution of the machine drive and the load characteristics. Comparison of expected noise levels with sanitary norms makes it possible to determine the excesses and the sources that form them. Actually, this information is the main one for choosing engineering solutions to ensure sanitary noise standards.

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

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

$$f_k = \frac{k}{2l} \sqrt{\frac{T}{\rho F} \left(1 + \frac{\rho F v^2}{T} \right)}, \quad (16)$$

where ρ is the density of the belt material, kg/m³.

For the acoustic model of a linear source, the sound pressure (P_{sp}) and the sound pressure level (L_p) of the belt drive are determined:

$$\begin{aligned}P_{sp} &= 6.7 \cdot \frac{v_k}{r} \left[khl \left(1 + \frac{\rho F v^2}{T} \right) \right] \cdot \left(\frac{T}{\rho F} \right)^{0.25}, \\ L_p &= 20 \times \lg \left(\frac{v_k}{r} \right) + 10 \times \lg \left(khl \left(1 + \frac{\rho F v^2}{T} \right) \right) + 5 \times \lg \left(\frac{T}{\rho F} \right) + 110,\end{aligned}\quad (17)$$

where k is the coefficient that determines the natural frequencies of the belt; h is the width of the belt, m; r is the distance to the calculated point, m.

4 Discussion

The obtained dependences make it possible to determine the sound pressure levels in the process of processing on belt grinding machines by calculation. The analysis of the equations obtained allows the design to take into account the layout of the machine drive and load characteristics, and apply noise reduction measures. Comparison of noise levels obtained by calculation with standard values allows determining noise discomfort already at the design stage and identifying sources generating noise. The information obtained is the basis for making decisions on ensuring sanitary noise standards.

The resulting equation can be simplified with the condition that the working body refers to narrow belt grinding machines. Naturally, a workpiece in the form of a rectangular rod located on the machine table was adopted as a design scheme (the reduced plane of such a system is designated $j_{np}y$). Then, the differential equation of bending vibrations has the form:

$$m_0 \frac{\partial^2 y}{\partial t^2} + EJ \frac{\partial^4 y}{\partial x^4} + j_{np}y = (Q + G) \sin(0.1A_c n t + \varphi) \delta(x - x_0), \quad (18)$$

where m_0 is the distributed mass of the workpiece being ground, kg/m; E is the modulus of elasticity of wood, Pa; J is the moment of inertia of the workpiece, m; P is the amplitude of the force impact from the side of the grinding belt, N; k_s is the coefficient of content of the abrasive wheel, n is the frequency of rotation of the drum of the grinding belt, rpm; $P = Q + G_x$ - pressing force on the tape, taking into account the weight of the part, N; $\delta(x - x_0)$ is the delta function applied to the x_0 coordinate.

Since the grinding belt is pressed against the workpiece with a flat iron, when calculating the vibroacoustic characteristics, it can be assumed that the load application coordinate does not change during the grinding time.

Comparison of the results obtained in this study with the results obtained by the authors [20, 21] shows that taking into account the effects of the distributed mass and modeling the impact force in the form of the Dirac delta function makes it possible to obtain a more accurate solution. Such a solution takes into account the inertial forces of the belt and vibrations of the workpiece and contains dynamic indicators in the equations for sound pressure (17).

5 Conclusion

The theoretical results of the study of vibration and noise of belt grinding machines during the processing of various workpieces are obtained. The solution is based on the assumption that the working body (grinding belt) has a distributed mass and is exposed to a periodic perturbing force from cutting the workpiece. The resulting system of partial differential equations is solved by the Fourier method. From the oscillation equations obtained, the oscillation velocities of the belt drive on natural oscillation modes and the corresponding natural oscillation frequencies are determined. From the equations for vibration velocities, expressions for sound pressure and noise levels are obtained.

Comparison of theoretical and experimental results showed a satisfactory agreement within 12%. In the high-frequency region of the noise spectrum, there is a discrepancy in the intensity of sound radiation in the range from 5 to 8 dB per octave.

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