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CALCULATION OSCILLATIONS OF VARIOUS ELEMENTS OF THE ELASTIC SYSTEM OF THE CENTER-FREE GRINDING MACHINE SASL 5AD

During the design of operations of centerless intermittent grinding of surfaces there is a need to identify the natural frequencies of oscillations of the elements of the technological system of grinding. The method of calculation of rigidity, vibration resistance and forced oscillations of the elements of the circular grinding machine is offered in the article. Carrying out of experimental researches of rigidity of elastic system of the SASL 5AD grinding machine. We conducted preliminary experimental studies to measure the oscillations of various elements of the elastic system of the SASL 5AD grinding machine in the horizontal plane by piezoelectric sensors during grinding with continuous and discontinuous circles with different geometric parameters.
Keywords: *machine, system, oscillations, spindle, polished surface*

Introduction. Calculation of natural frequencies of oscillations is one of the tasks of dynamics of mechanical systems connected with definition of resonant modes of their work. A method for calculating the stiffness, vibration resistance and forced oscillations of grinding machines with a horizontal spindle is proposed. The elastic system of the machine was approximated by a mechanical model with 2 degrees of freedom. This technique allowed to make design changes at the design stage, which increased the rigidity of the machines by 70-90%, and also allows to predict changes in the design parameters of the elastic system of the machine in order to obtain regulated dynamic properties [1-5,7.8].

However, in the conditions of actual operation of the grinding machine, the change of dynamic characteristics is due to the change of the initial connection between the elements of the elastic system. Therefore, there is a need to determine the dynamic characteristics of elastic systems of different types of equipment, which provides for the introduction into the technological process of editing and balancing operations.

Material and results of the study. Experimental studies of the stiffness of the elastic system of the SASL 5AD grinder (the general view of which is shown in Fig. 1) in the horizontal plane by piezo acceleration sensors during grinding with continuous and discontinuous circles with different geometric parameters. The scheme of grinding of rings by an intermittent grinding wheel with the screw grooves inclined at an angle α to an axis is shown in fig.

Fig.1. Centerless grinding machine SASL5AD (general view)

Fig.2. Working area of the SASL5AD centerless grinding machine

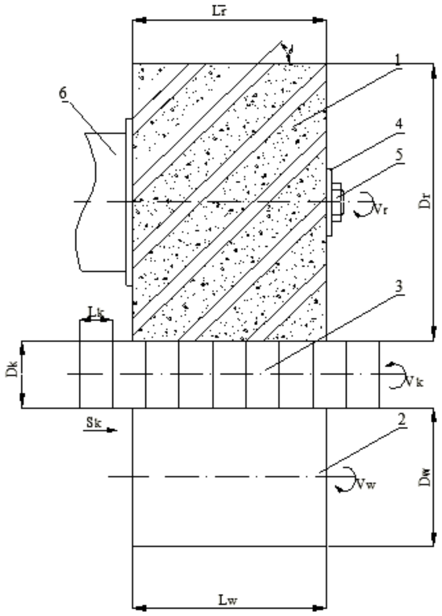

Fig. 3. Scheme of treatment of the base surface with an intermittent circle
1-grinding wheel with inclined grooves; 2- leading circle; 3 - processed rings; 4,5,6- elements of the machine spindle; Sk - supply; Dk is the outer diameter of the ring; Dr is the diameter of the grinding wheel, Dw is the diameter of the drive wheel

At the first stage of research, the calculated values of the natural frequencies of bending and torsional oscillations of the spindle of the machine SASL 5AD were obtained. The values of natural frequencies were determined using classical methods of oscillation theory [1, 2-7]. The spindle of the grinding wheel was replaced by a design scheme - a two-support cantilever beam with distributed mass (see Fig. 4), which consists of a single beam, because the spindle of the machine SASL 5AD has no pronounced diameter differences. At the ends of the beam were placed the concentrated masses of the grinding wheel and the drive pulley, which have the inertia of rotation. Hinged supports were placed in appropriate places. For the given calculation scheme, the differential equation of free bending oscillations has the form:

$$EIY^{IV} + m_0 \ddot{y} = 0 \quad , (1)$$

where E - modulus of elasticity;

I - equatorial moment of inertia of the spindle cross section;

m_0 - mass per unit length of the beam;
$y(x,t)$ - vertical movement.

The solution of equation (1) is written in the form:

$$y(x,t) = U(x)\sin(\omega t + \varphi'). \tag{2}$$

Substituting (2) in (1) we obtain the equation of forms of oscillations of the beam:

$$U^{IV} - \alpha^4 U = 0, \tag{3}$$

where, $\alpha^4 = \dfrac{\omega^2 m_0}{EI}$.

The solution of equation (3) is written in Krylov functions [4]:

$$U(x) = C_1 K_1(\alpha x) + C_2 K_2(\alpha x) + C_3 K_3(\alpha x) + C_4 K_4(\alpha x), \tag{4}$$

where C1-C4 - steel.

In fig. In Fig. 5 shows a diagram of the spindle assembly of the centerless grinding machine.

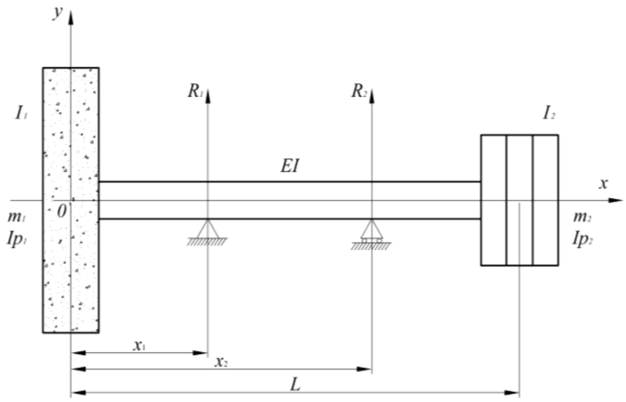

Fig. 4. The calculated scheme of the spindle assembly: X1 = 0,109m; X2 = 0.355m; L = 0.433m; EI = 6140kg · m2; m1 = 0.358 kg · sec2 / m; m2 = 0.08 kg · sec2 / m; Ip1 = 0.0018kg · m · sec2; Ip2 = 0.00012kg · m · sec2; I1 = 0.0009kg · m · sec2; I2 = 0.000077kg · m · sec2.

Substituting the obtained solutions to the boundary conditions at the ends of the beam:

$$\left.\begin{aligned} m_1\omega^2 U_1(0) - EIU_1'''(0) = 0 \\ EIU_1''(0) + \omega^2 (Ip_1 - I_1) U_1'(0) = 0 \end{aligned}\right\} x = 0;$$
$$\left.\begin{aligned} EIU_3'''(L) - m_2\omega^2 U_3(L) = 0 \\ EIU_3'''(L) + \omega^2 (Ip_2 - I_2) U_3'(L) = 0 \end{aligned}\right\} x = L, \tag{5}$$

where, I_1, I_2 - equatorial moments of inertia of the grinding wheel and drive pulley;
Ip_1, Ip_2 - polar moments of inertia of the grinding wheel and drive pulley;
m_1 - weight of the grinding wheel;
m_2 - weight of the drive pulley,
and adding the conditions of zero deflection on the supports:

$$U_1(x_1) = 0; U_2(x_2) = 0, \tag{6}$$

we obtain a homogeneous system of six equations with six unknowns: C1, C2, C3, C4, R1, R2, where R1, R2 are the amplitude values of the reaction on the 1st and 2nd supports, respectively.

For a non-trivial solution of this system, it is necessary that the determinant be equal to zero. From this condition, the method of selection of acceptable geometric and mass parameters of the spindle, the pulley circle (Fig. 4.), are the values of natural frequencies. The first two values are: $\omega_1 = 3580$ 1 / sec, $\omega_2 = 9270$ 1 / sec.

The calculation of two low natural frequencies of torsional oscillations is performed by a similar method. The calculation scheme does not change, but only its torsional and mass characteristics are taken into account, respectively. [2, 3, 4, 5, 7].

The differential equation of free torsional oscillations has the form:

$$GIp\frac{\partial^2\theta}{\partial x^2} - \theta_0\ddot{\theta} = 0, \tag{7}$$

where *Ip* - polar moment of inertia of the spindle cross section;
G - shear module;
θ_0 - moment of inertia of the mass per unit length of the rod;
θ - twisting angle.

The solution of equation (7) will take the form:

$$\theta(x,t) = \varphi(x)\sin(\omega_k t + \varphi''). \tag{8}$$

Substituting (8) into (7), we obtain the equation of the form of torsional oscillations:

$$\varphi^{II} + \beta^2\varphi(x) = 0, \tag{9}$$

where $\beta^2 = \dfrac{\omega_k^2\theta_0}{GIp}$.

The solution of equation (9) will take the form:

$$\varphi(x) = A\sin\beta x + B\cos\beta x, \tag{10}$$

where *AND* and B are constants.

Subjecting the obtained equation to the boundary conditions at the ends of the beam:

$$\begin{gathered} GIp\varphi'(0) + Ip_1\omega_k^2\varphi(0) = 0, x = 0 \\ GIp\varphi'(L) - Ip_2\omega_k^2\varphi(L) = 0, x = L \end{gathered}, \tag{11}$$

we obtain a system of two equations with two unknowns A and B. Equating the determinant of the system to zero, the selection method finds the values of the natural frequencies of free spindle oscillations: $\omega_{k1} = 8240$ and $\omega_{k2} = 27400$ rad / sec.

The value of the first natural frequency of the spindle bending oscillations mounted on rigid supports was determined using an experimental setup and was $\omega_{1Э} = 7770$ *I* /sec

The relative error of theoretical calculation and experimental determination of natural frequencies for bending oscillations is equal to:

$$\Delta_U = \frac{\omega_1 - \omega_{1Э}}{\omega_{1Э}} \cdot 100\% = 6{,}04\%.$$

The theoretically calculated value of natural frequencies corresponds to the results of experiments. The values of the first two natural frequencies of bending and torsional oscillations are obtained as a result of solving the system of differential equations.

In order to simplify the solution of the problem and estimate the degree of discrepancy of the obtained frequencies with the exact solution, consider the beam (spindle) as a three-mass system and solve for it the problem of finding the natural frequencies of bending oscillations.

The frequency equation for a three-mass system has the form:

$$p^3-(m_1\alpha_{11}+m_2\alpha_{22}+m_3\alpha_{33})p^2+ \\ +(m_1m_2f_{12}+m_2m_3f_{23}+m_1m_3f_{13})p-m_1m_2m_3f_{123}=0, \tag{12}$$

where $p=\dfrac{1}{\omega^{12}}$;

ω^{12} - natural frequency;
α_{ij} - coefficient of force on bending.

$$f_{ij}=\alpha_{ii}\alpha_{jj}-\alpha_{ij}^2;$$

$$f_{ikj}=f_{ij}\alpha_{kk}+f_{ik}\alpha_{jj}+f_{jk}\alpha_{ii}+2(\alpha_{ij}\alpha_{jk}\alpha_{ik}-\alpha_{ii}\alpha_{jj}\alpha_{kk}).$$

From equation (12) we can obtain the first underestimated value of the natural frequency according to the Dunkerley formula:

$$\omega^{12}=\frac{1}{\sum_{i=1}^{n}m_i\alpha_{ii}}, \tag{13}$$

where n - number of masses;
α_{ii} - find the following formulas:

$$\alpha_{11}=\frac{x_1^2x_2}{3EI};\alpha_{22}=\frac{(L-x_2)^2}{3EI}(L-x_1);\alpha_{33}=\frac{(x_3-x_1)^2(x_2-x_3)^2}{3EI(x_2-x_1)}. \tag{14}$$

After substitution we get:

$$\alpha_{11}=2{,}27\cdot10^{-7}\,\text{м/кг};\alpha_{22}=1{,}07\cdot10^{-7}\,\text{м/кг};\alpha_{33}=0{,}485\cdot10^{-7}\,\text{м/кг}.$$

Substituting the values of mi and α_{11} to formula (13) we obtain the value of the first natural frequency $\omega_1'=3080\text{I/c}$. The value of the natural frequency is found by formula (13), when replacing the spindle as a beam with a distributed mass on a beam with three concentrated masses greatly simplifies the calculations. The relative error of calculation when found an underestimated value of the frequency according to the Dunkerley formula, in comparison with the exact solution, is:

$$\Delta=\frac{3580-3080}{3580}\cdot100\%=14\%.$$

Conclusions. The described method of determining the natural frequency of the spindle of the centerless grinding machine is used to predict the parameters of the microrelief of the ground surfaces of rotation at the stage of technological design.

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РОЗРАХУНОК КОЛИВАНЬ РІЗНИХ ЕЛЕМЕНТІВ ПРУЖНОЇ СИСТЕМИ БЕЗЦЕНТРОВО- ШЛІФУВАЛЬНОГО АВТОМАТА SASL 5AD

Під час проектування операцій безцентрового переривчастого шліфування поверхонь виникає необхідність виявлення власних частот коливання елементів технологічної системи шліфування. В статті запропоновано методику розрахунку жорсткості, віброcтійкості та вимушених коливань елементів кругло-шліфувального верстата. Проведенні ексерементальні дослідження жорсткості пружної системи шліфувального верстата SASL 5AD. Нами проведено попередньо експериментальні дослідження по вимірюванню коливань різних елементів пружної системи шліфувального верстату SASL 5AD у горизонтальній площині давачами п'єзоприскорень під час шліфування суцільним та переривчатим кругами з різними геометричними параметрами.

Ключові слова: *верстат, система, коливання, шпиндель, шліфована поверхня*

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РАСЧЕТ КОЛЕБАНИЙ РАЗЛИЧНЫХ ЭЛЕМЕНТОВ УПРУГОЙ СИСТЕМЫ БЕСЦЕНТРОВОГО ШЛИФОВАЛЬНОГО СТАНКА SASL 5AD

При проектировании операций бесцентрового прерывистого шлифования поверхностей возникает необходимость выявления собственных частот колебания элементов технологической системы шлифования. В статье предложена методика расчета жесткости, виброустойчивости и вынужденных колебаний элементов кругло-шлифовального станка. Проведении ексерементальни исследования жесткости упругой системы шлифовального станка SASL 5AD. Нами проведено предварительно экспериментальные исследования по измерению колебаний различных элементов упругой системы шлифовального станка SASL 5AD в горизонтальной плоскости датчиками пьезоускорений во время шлифовки сплошным и прерывистой кругами с различными геометрическими параметрами.

Ключевые слова*: станок, система, колебания, шпиндель, шлифованная поверхность*