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ANALYSIS OF LATHE VIBRATION INFLUENCE ON BLANK ROUGHNESS

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Abstract: *The effect of lathe vibrations to the roughness of machined surface was investigated in this article. The calculation schemes involve systems with one and two degrees of freedom representing the vibration of the blank as a rigid body hinged in the spindle and elastically supported in the tailstock of the lathe. Experimental measurements were performed on lathes of type 1K62 at different cutting speeds, feeds and depths of cut. Processing of the roughness measurements data confirmed the precision of the calculation model. Surface roughness parameters of the blank corresponded quite satisfactory to the corresponding data of the theoretical investigation. For research of influence of gyroscopic forces to the surface roughness was used a calculation model with two degrees of freedom. The results of experimental and theoretical investigations confirmed the theoretical hypothesis*.
Key words: *Lathe, vibration, calculation model, natural frequency, blank roughness.*

1. INTRODUCTION

In the turning operations, tool vibrations influence both product quality and productivity and may also have a negative influence on the working environment (Hakansson, 1999). The effect of lathe vibration to the roughness of machined surface is considered in this article. Dynamical phenomena's concerned with the vibration are caused from external factors on the strained system of the lathe. During the machining of material all disturbances finally lead to relative displacements of cutter and blank. It allows to link parameters of the surface roughness with the relative vibrodisplacements of the cutter and the blank (Gaponkin, 1995). On calculation of dynamics characteristics the real elastic system of the lathe was replaced by a system with finite degrees of freedom. Use of complicated calculation schemes can lead in case of insufficient accuracy of the underlying data to significant error of the calculation (Arshanski, 1998). Therefore simplified schemes were used which composite on the basis of experimental investigations.
The fundamental calculation scheme is a system with one degree of freedom representing the vibration of the blank as a rigid body hinged in the spindle and elastically supported in the tailstock of the lathe. Exact solution for continuous system having infinite number of degrees of freedom showed that the first natural frequency of the continuous model is keeping with the natural frequency of the accepted model. It makes possible to use this calculation model for analysis of the vibrations in case of the metal cutting.
Experimental measuring was performed on lathes of type 1K62 by different cutting speeds, feeds and depths of cut. The experimental results satisfactory coincided with the similar theoretical results in adequate frequency range. With growth of frequency number a divergence between the theoretical and the experimental results increased gradually. As a result the limits of use of the proposed mathematical model of the blank on lathe vibration were considered. After every cutting the surface roughness was measured by profilograph 'Surftronic 3+'. The processing of the roughness measurements data confirmed precision of the calculation model. Surface roughness parameters of the blank quite satisfactory agreed with the corresponding data of the theoretical investigation. For research of influence of gyroscopic forces to the surface roughness was used calculation model with two degrees of freedom. Until now in such formulation this problem has not been researched. The results of experimental and theoretical investigations confirmed the theoretical hypothesis. This knowledge gives the possibility to increase the accuracy of processing on different conditions of cutting. Calculation models with four degrees of freedom are assumed to use in future. Question of stability of the blank in the action of the moving cutting force was remained without investigation. It is necessary to derivate theoretical expressions enabling to determine the roughness more precisely. It gives the possibility to control and adjust the surface roughness of processing details. But for that is necessary to conduct a large number of experiments with good equipment.

2. THEORETICAL INVESTIGATION

Treatment of the dynamic calculation model started from a formulation of research problem. For selection of this model was used data received from workable experiments. In order to simplify the dynamic model were not taken into account factors which have a little effect on results of solution. Of course these models have unlimited area of use (Aryassov, 2003). In current work the dynamic models with one and two degrees of freedom are considered (Fig.1)

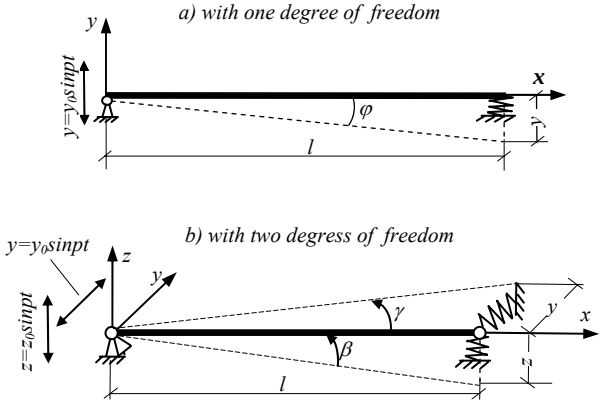

Fig.1. Calculation dynamical models.

2.1. Calculation scheme with one degree of freedom (Fig.1a).

On idling of the lathe the differential equation of forced vibrations induced by the foundation vibrations

$$J_o \ddot{\varphi} + k_y \varphi\, l^2 = M_0 \sin p\, t \qquad (1)$$

where y_0 is amplitude of the foundation vibrations; J_O is moment of inertia of the blank around the spindle; $p=2\pi f$, f is the frequency of the foundation vibrations in Hz; k_y is the spring constant of elastic support of the blank; $M_0 = m p^2 y_0 l/2$.

The proposed model is restricted with a solution determining steady-state forced vibrations

$$\varphi = \frac{M_0}{Jo(\omega^2 - p^2)} \sin p\,t \tag{2}$$

$\omega = \sqrt{k_y l^2 / J_0}$ is a natural frequency of the lathe system.

From (2) the velocity of the forced vibrations

$$v = \dot{y} = \dot{\varphi} \cdot l = \frac{p M_0}{J_0(\omega^2 - p^2)} \cos p\,t \tag{3}$$

The received function (3) is a relative velocity of the blank. The correctness of the accepted calculation model is controlled by means of comparison of the theoretical results with experimental results. In machining of the detail in the lathe the cutting force F (Fig.2) is not constant. It is determined by the action of many factors: change in thickness of the cut-off chips, change of mechanical properties of the blank material, wear and bluntness of the cutting instrument. The input coordinate of the lathe system is the cutting force F, which is a function of time and the output coordinate is the displacement of the cutter or the blank.

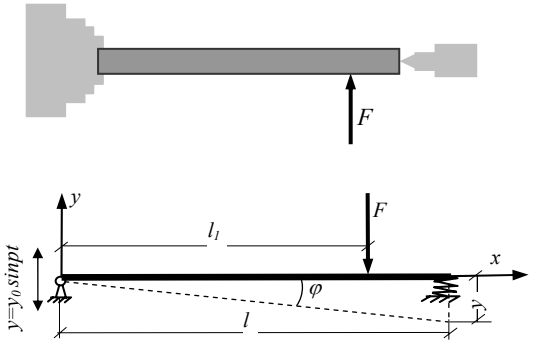

Fig. 2. Calculation scheme in cutting

The differential equation of forced vibrations in action of the cutting force F

$$J_0 \ddot{\varphi} + k_y \varphi l^2 = M_0 \sin p\,t + \left(F_r + F_a \cos \omega^* t\right) l_1 \tag{4}$$

where the cutting force F is represented as a sum of two items: the constant component F_r determined in practice by the simplified empirical formula [2] and the variable component $F_a \cos \omega^* t$. The amplitude value F_a of the variable component of the cutting force is connected with the roughness value and changed in a rather wide range.

The solution of the Eq. (4) was expressed in the form of the displacement of the blank end with regard to initial conditions y_0 and v_0

$$y = \varphi l = \left(y_0 - F_0 l_1 - E l_1 l\right)\cos \omega t + \left(\frac{v_0 - Dl\,p}{\omega}\right)\sin \omega t + \\ + Dl \sin pt + F_0 l_1 + E l l_1 \cos \omega^* t \tag{5}$$

whence the motion velocity

$$v = -\left(y_0 - F_0 l_1 - E l l_1\right)\omega \sin \omega t + \\ + \left(v_0 - Dl\,p\right)\cos \omega t + Dl\,p \cos pt - E l l_1 \omega^* \sin \omega^* t \tag{6}$$

$$D = \frac{M_0}{J_0(\omega^2 - p^2)}, \quad F_0 = \frac{F_r}{kl}, \quad E = \frac{F_a}{J_0(\omega^2 - \omega^{*2})} \tag{7}$$

In a similar way was obtained the solution with regard of resistance forces, which is not given here in view of the fact that it is very large.

2.2. Calculation scheme with two degrees of freedom (Fig.1b and Fig.3).

Such model makes possible to take into account an effect of gyroscopic forces resulting from the rotation of the blank.

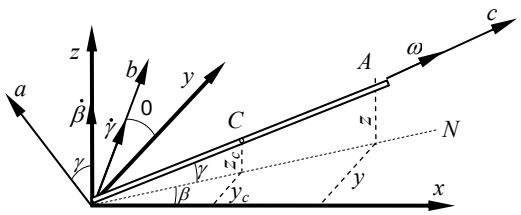

Fig.3. Gyroscope system with two degrees of freedom in cutting.

The differential equations of forced vibrations in action of the cutting force F can be represented by theorem of kinetic moment in the following form

$$J_0 \ddot{z} - A \omega_b \dot{y} + k_z z l^2 = M_z \sin pt$$

$$J_0 \ddot{y} + A \omega_b \dot{z} + k_y y l = M_y \cos pt + F_r l_1 + F_a l_1 \cos \varpi^* t \tag{8}$$

where ω_b is angular velocity of rotation of the blank, $My = m p^2 y_0 l/2$, $Mz = m p^2 z_0 l/2$, k_z and k_y are spring constants, A is moment of inertia of the blank with respect to axes of rotation. The general solution of the Eqs. (8) represents free vibrations

$$y_1 = a_1 \sin(p_1 t + \alpha_1) + a_2 \sin(p_2 t + \alpha_2), \\ z_1 = \mu_1 a_1 \sin(p_1 t + \alpha_1) + \mu_2 a_2 \sin(p_2 t + \alpha_2) \tag{9}$$

where a_1, a_2, α_1 and α_2 are constants of integrations to be determined from the initial conditions; μ_1 and μ_2 are rations of the amplitudes for corresponding two principal modes of vibrations; p_1 and p_2 are natural frequencies of vibrations with gyroscopic forces

$$p_{1,2} = \sqrt{\left(b^2 \pm \sqrt{b^2 - 4 J_0{}^2 l^4 k_y k_z}\right) \cdot 0.5 \cdot J_0},$$

where $b = J_0 l^2 \left(k_y + k_z\right) + A^2 \omega_b{}^2$ (10)

The analysis shows that when value of ω_b is increased then difference between the lower frequency p_1 and higher frequency p_2 increases as well (Fig.4).

It was found that for the first mode with higher frequency p_1 the ratio μ_1 was positive, i.e. their vibrations y_1 and z_1 are in phase or in so called direct precession. For lower frequency p_2 it was negative, i.e. their vibrations y_1 and z_1 are in opposite phase or in so called reverse precession.

That means in the first mode of vibration the point A of the blank axis moves on circle to the side of own rotation and in the second mode moves in opposite direction of rotation (Fig.4).

In a particular solution of these equations Eqs. (8), depending on the disturbing force, the forced vibrations of the system are represented in the following form

$$z_2 = d \sin pt + d_1 \sin \varpi^* t, \\ y_2 = b \cos pt + b_1 \cos \varpi^* t + \frac{F_r l_1}{k_y l^2} \tag{11}$$

$$\text{where } b = \frac{M_z p A \omega_b - M_y\left(k_z l^2 - J_o p^2\right)}{p^2 A^2 \omega_b^2 - \left(k_y l^2 - J_0 p^2\right)\left(k_z l^2 - J_0 p^2\right)}$$

$$d = \frac{M_z\left(k_y l^2 - J_o p^2\right) - M_y p \omega_b}{\left(k_z l^2 - J_0 p^2\right)\left(k_y l^2 - J_0 p^2\right) - p^2 A^2 \omega_b^2} \qquad (12)$$

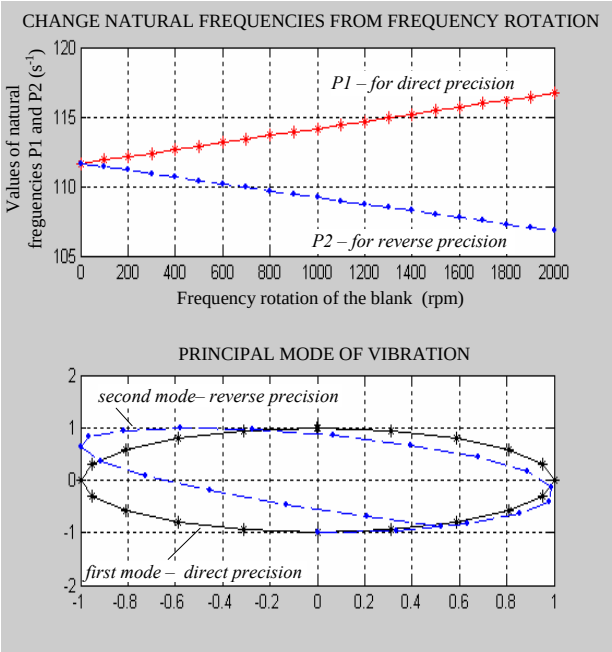

Fig. 4. Principal modes of vibration corresponding to two different natural frequencies with gyroscope forces.

$$b_1 = \frac{-F_a l_1\left(k_z l^2 - J_0 \omega^{*2}\right)}{A^2 \omega_b^2 \omega^{*2} - \left(k_y l^2 - J_0 \omega^{*2}\right)\left(k_z l^2 - J_0 \omega^{*2}\right)}$$

$$d_1 = \frac{-F_a l_1 A \omega_b \omega^*}{\left(k_z l^2 - J_0 \omega^{*2}\right)\left(k_y l^2 - J_0 \omega^{*2}\right) - A^2 \omega_b^2 \omega^{*2}} \qquad (13)$$

Summarizing the general solution Eqs. (9) and partial solution Eqs. (11) was received a general solution of the differential equations Eqs. (8) for displacements y and z of the blank end, from which velocities v_y and v_z can be determined easily.
Usually in studying of steady-state vibrations the components determining free damping vibrations drop. However, it is impossible in this instance, because operating conditions in the cutting due to roughness surface are changed.

3. EXPERIMENTAL ANALYSIS

The correctness of the accepted models was tested on the lathe 1K62.

3.1. Experimental test of the spring constant of the lathe

The exactness of calculations during the theoretical analysis depends both on the degree of fitness of the accepted calculation scheme of the real system and on exactness of finding of the mechanical characteristics of the lathe. One of these characteristics is the spring constant of the lathe. The spring constant of the lathe is determined by the static loading of its elements, connected with the workpiece and cutter. The direction and points of application of the force are selected according to typical situations of machining of the details on the specific lathe. For decreasing of influence of the resistance force on the test results the static rigidity was measured in the case of weak vibrations of the lathe excited by running of the electric motor and others mechanisms without load. The experiments were carried out in the case of two position of the load (Fig.5).

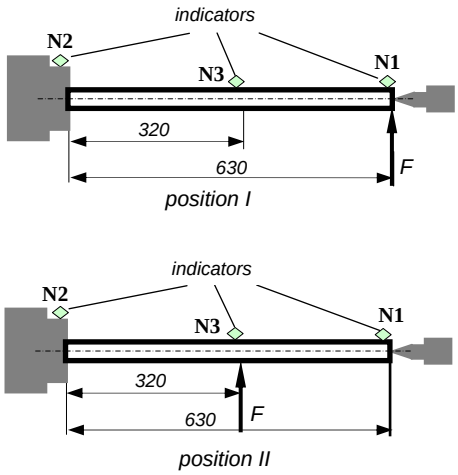

Fig. 5. Scheme for measuring of the vertical and horizontal rigidity of the lathe.

By means of the dynamometer the system was loaded gradually by step 100 N, whereas the displacements with accuracy of 0.002 μm were registered by the indicators in three points: on the tailstock (indicator 1), on the spindle (indicator 2) and on the blank (indicator 3).
In the Fig.6 are given the results of mathematical statistical analysis the experiment data in the form of correlative straight lines, where coefficient of direct regression is the unknown rigidity.

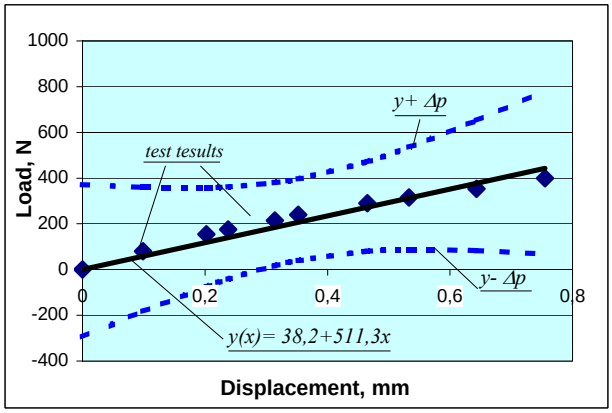

Fig. 6. Correlative function between the load and static displacement.

The coefficient of correlation was received close by unit that indicates of the linear correlative function between the load and displacement.

3.2. Experimental analysis of vibration on idling of the lathe

On the basis of the static experiment tests can be concluded, that it is possible to consider the blank as an ideal solid body hinged in the head-stock and elastically hinged in the back-stock. Therefore it was admissibility to accept the system with one degree of freedom for vibration analysis in two planes. (Fig.1 b).
As the basic measurement equipment was used vibration analyzer SigLab 20.22A with special software in MATLAB, designed for multi-channel investigations of vibroacoustic

signals in frequency band from 2 Hz to 50 kHz. As transducers piezoelectric accelerometers KISTLER 870B10 and KISTLER 8702B50 were used with sensitivity to 50 μv/g. In additions the pocket sized vibrometer was used –data collector PICOLOG CMVL 10 of firm SKF for measuring in frequency band from 30 Hz to 10 kHz.
The piezoelectric accelerometers were installed on the blank and onto the base of lathe.
The results of measuring of the vibrations in horizontal plane are given in Fig.7, where reference theoretical results of vibration velocity according to Eq. (3) are given too.

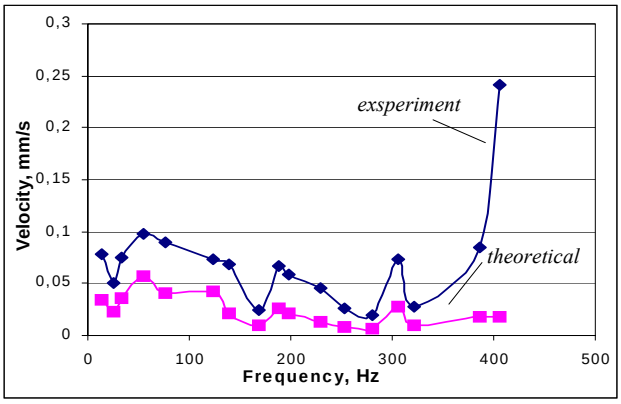

Fig.7. Experimental and theoretical results of the vibration in horizontal plane

The experimental results satisfactory coincided with the similar theoretical results in adequate frequency range. But with growth of frequency number a divergence between the theoretical and experimental results increased gradually. Thus correctness of accepted calculation model was confirmed for using over the sufficiently wide range of velocities of the lathe.

3.3. Measuring of vibration in case of rotation of the blank

Similar experiment was conducted also in case of the rotating blank. Tests were carried out with different frequencies of rotation of the spindle. In contrast to the previous test one of the piezoindicators was installed not on the blank but on the tailstock. It slightly distorted results of measuring but the general picture remained unaltered. It was confirmed by measuring by vibrometer PICOLOG which was in contact with the surface of the rotation blank. The test results in horizontal plane and corresponding theoretical results of vibration velocity according to Eq. (3) and Eqs. (9-12) with gyroscopic forces are given in Fig.8. One can see that the theoretical results which were received taking into account gyroscopic forces noticeable agreed with the experimental results.

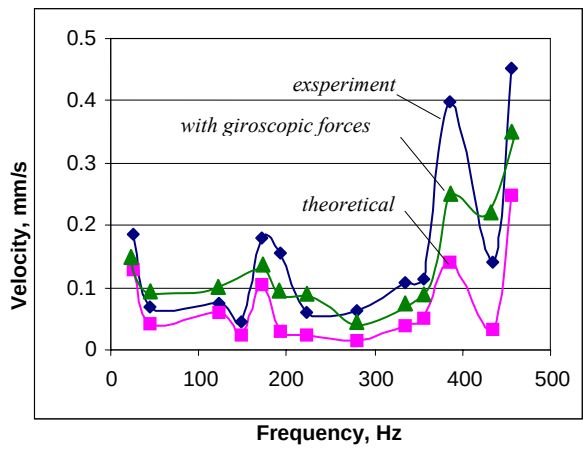

Fig.8. Experimental and theoretical results of vibration with gyroscopic forces

3.4. Measuring of vibration in cutting

Experimental measuring was performed with different cutting speeds, feeds and depths of cut. Test results and results of the calculation by Eqs. (6-7) and Eqs. (11-13) taking into account gyroscopic forces are given in Fig.9. After every cutting the surface roughness was measured by profilograph 'Surftronic 3+'. The amplitude value F_a of the variable component of the cutting force in Eq. (4) was taken according to the experimental value of the roughness. The processing of the roughness measurements data confirmed precision of the calculation model. Surface roughness parameters of the blank quite satisfactory agreed with the corresponding data of the theoretical investigation. From comparison of the theoretical results with experimental results one can see that they satisfactory coincide. As well in the previous case the results of calculation with gyroscopic forces by Eqs. (11-13) coincided well with the experimental results.

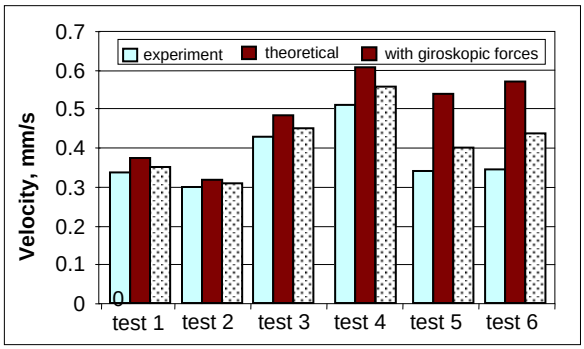

Fig.9 Comparative analysis of experimental and theoretical results

4. CONCLUSION

The processing of the roughness measurements data confirmed precision of the calculation model. Surface roughness parameters of the blank quite satisfactory agreed with the corresponding data of the theoretical investigation. For research of influence of gyroscopic forces to the surface roughness was used calculation model with two degrees of freedom. The results of experimental and theoretical investigations confirmed the theoretical hypothesis especially when calculation model with two degrees of freedom was used.

5. REFERENCES

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