

LATHE TOOL HOLDER WITH THE ORIENTED POSITION OF THE CENTER OF RIGIDITY

Assistant professor mag. eng. Vakulenko S.V.

Faculty of Mechanical of the National Technical University of Ukraine "Kyiv Polytechnic Institute", Kyiv, Ukraine

Abstract: A research proposes a new method for decreasing vibrations which occur when cutting metal by using the newly designed construction of tool holder with oriented center of rigidity. Newly designed tool holder likewise the partial oscillation system with a special oriented axes of stiffness makes dynamic influence on carriage's dynamic system vibrations with higher vibration stability. Elastic parameters of the proposed tool holder were selected based on theoretical stability analysis using Nyquist plots of developed mathematical model closed dynamic system "carriage - tool holder - cutting process" in the MATLAB Simulink.

KEYWORDS: VIBRATION, VIBRO STABILITY, TOOL HOLDER, LATHE WORK, THE DYNAMICS OF MACHINES, CENTER OF RIGIDITY

1. Introduction

Vibration resistance of the lathe system depends mainly on the elastic parameters of its elements, such as stiffness factors, their ratio and orientation of elastic axes relative to the cutting force direction. To ensure the lathe system vibration resistance its stiffness should be greater in the direction of normal to the machined surface or in the direction of cutting force, and in other directions the system stiffness should be lower not to cause the loss of the system stability due to the coordinate bond. A significant loss in vibro stability during lathe work with rising self-excited oscillations is caused by the presence of the negative influences of coordinate bond between the cutting tool elastic vibro displacement at the normal plane to the work surface, vector direction and the value of cutting force.

The lathe carriage elastic system as a complex dynamic system with many degrees of freedom has a lot of its own forms of oscillations on each of which displacement ellipses can be detached which principal axes directions do not coincide with the lathe general coordinate axes. Intense self-oscillations are performed at the frequency which corresponds to natural oscillations frequency of the dominant elastic system, namely, till oscillation frequency of that link that has the largest sizes of displacement ellipse. Given this fact, the carriage dynamic system mathematically can be represented as a one-mass system with two degrees of freedom. This simplification of multi-link system is allowed because oscillations at other frequencies during cutting are not dominant and have little coordinate bond impact on machining. A simplified one-mass mathematical model of the carriage potentially unstable system allows to conduct theoretical studies of instability according to Nyquist criterion, since this system Nyquist plot consisting of frequency characteristics of each of the two normal modes of oscillation crosses the negative real axis.

Solving problems of increasing machining vibration resistance is the use of tool holder with oriented center of rigidity which corrects the carriage dominant system displacement ellipse on its main oscillation frequency, since these dynamic links are partial and connected with each other. Urgent matter is to carry out theoretical and experimental studies of the influence of the tool holder elastic parameters with oriented center of rigidity on the lathe machining vibration resistance taking into account frequency characteristics of shape-generating lathe components, physical properties of the machined material and cutting tooling geometry.

2.1. Preconditions and means for resolving the problem

The highest vibration resistance during turning is achieved when the cutting force direction approaches the axis of the greatest stiffness of the lathe carriage elastic system, otherwise the coordinate bond negative impact on relative oscillation level of tooling and workpiece increases in the lathe dynamic system. The

system maximum stiffness direction at one given point of the tool top passes through the point of its center of rigidity, the main elastic deformations during the loading action in this direction are determined by the system stiffness principal axes directions. If direction of loading on the carriage system does not pass through the point of center of rigidity additional elastic rotation appears around this point in the system, so the stiffness in the direction perpendicular to the given maximum stiffness axis will be minimum. When developing a lathe design it is difficult to predict or calculate the carriages center of rigidity future position which is determined experimentally by means of the well-known "direction finding" method already for the lathe real design. If the actual position of the lathe carriages center of rigidity and of turning angle of principal given stiffness axis determined experimentally does not meet the above requirements, the lathe elastic system will be potentially unstable and will require additional structural and technological measures to ensure vibration resistance machining.

As an example, Figure 1 shows the carriage stiffness diagram determined experimentally with the set lathe 1K62 tool holder which helps to identify directions of given maximum and minimum stiffnesses characterized by the angle $\beta = -60^\circ$ relative to axis y . Since machining on only one side is structurally factored into the test lathe, namely radial feed is performed in the negative direction of axis y , the carriage elastic system during cutting is potentially unstable and prone to increased amplitudes of self-oscillations. The problem of the lathe turning low vibration resistance also occurs on the lathes, for example ПАБ-350, which in technological capabilities allow machining on both sides when the spindle rotating direction is changed. Results of theoretical determination of the given elastic parameters of ПАБ-350 [1] lathe shape-generating units have found that the lathe carriage elastic system has directed stiffness that allows for high vibration resistance of machining of only one side.

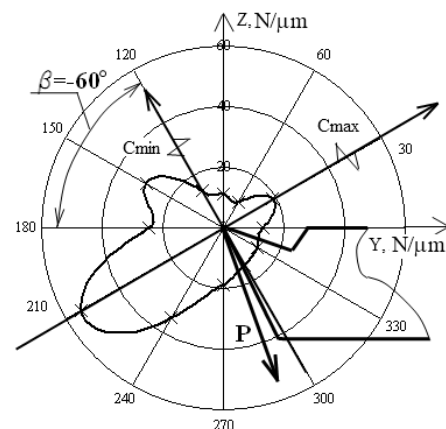


Fig. 1. Circular diagram of the carriage group stiffness of 1K62 lathe

When designing tool holder with oriented center of rigidity for a lathe with defined elastic parameters of the carriage the question of choosing rational elastic and damping parameters of the tool holder comes up the use of which has increased vibration resistance of machining. Theoretical studies of the impact of changing orientation of tool holder principal axes of stiffness on the dynamic characteristics of the carriage - tool holder system are presented in the work [2] using the developed mathematical model. The issue of determining the efficiency of additional tool holder in order to increase the lathe turning vibration resistance with a potentially unstable dynamic system of the lathe carriage during cutting needs to create a general mathematical model of the lathe dynamic system, its modelling and determination of vibration resistance reserve of machining at different elastic parameters of tool holder.

2. Solution of the examined problem

The structure of the closed dynamic machining system in the general form is shown in Fig. 2 and consists of the equivalent elastic system (EES) "carriage - tool holder" and a "workpiece" which have negative feedback through the cutting process. Determination of the stability reserve of the lathe closed dynamic system during cutting is performed using Nyquist frequency method which requires construction of open-loop dynamic system Nyquist plot. According to Fig. 2 the lathe open-loop dynamic system has input in the form of changing cutting layer thickness of the machined workpiece Δy and output as general relative dynamic elastic displacement of the workpiece and tooling under the influence of cutting force.

Calculation model of the elastic system carriage - tool holder and the workpiece shall be presented in the plane perpendicular to the workpiece axis yOz in the form of lumped given masses of the tool holder m_1 , the carriage m_2 and the workpiece m_3 which are connected with each other and by the lathe base with the links with elastic and dissipative properties (Fig. 3). Each mass is considered as a subsystem with oriented axes of stiffness, namely, for the mass m_1 a turning angle of principal axes of stiffness relative to an arbitrary coordinate system yOz shall be defined as β , for mass m_2 - δ , for mass m_3 - γ respectively. Under the influence of the cutting force dynamic part $P(t)$ which is applied to the cutter top at the angle α to the axis z , the mass m_1 performs oscillatory motions in the directions of the principal axes of coordinates η_1 and η_2 , and the mass m_2 in the directions of the principal axes η_5 and η_6 . For elastic and dissipative links reduced total damping coefficients h_1 , h_2 and h_5 , h_6 and stiffness coefficients c_1 , c_2 and c_5 , c_6 respectively of the tool holder (m_1) and the carriage (m_2) shall be taken into account. The cutting force also influences the workpiece (m_3) subsystem in the opposite direction due to which the workpiece makes oscillatory motions in the directions of the main coordinate axes η_3 and η_4 taking into account the given parameters of the spindle-holder-workpiece system of stiffness c_3 , c_4 and damping h_3 , h_4 .

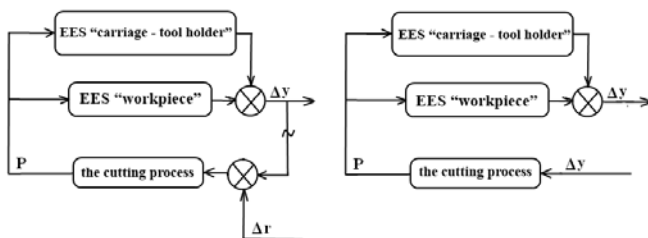


Fig. 2. Block diagram of the lathe closed and open dynamic system during cutting

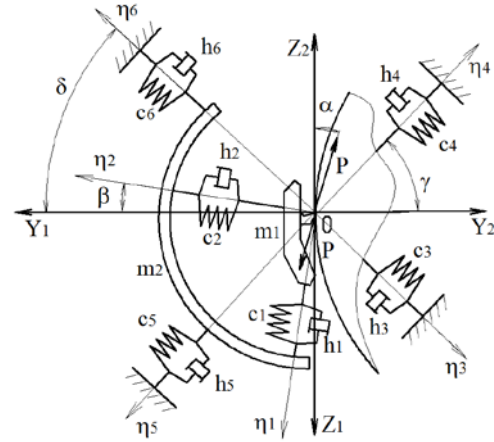


Fig. 3. Calculation diagram of the lathe elastic system in the plane yOz with two coordinate systems

Equation of motion in the directions of the main coordinates for the dynamic two-mass subsystem carriage - tool holder with four degrees of freedom shall be presented in the form of the second order differential equations system obtained from Lagrange equation of the second kind for mechanical system:

$$\begin{cases} m_1 \ddot{\eta}_1 + h_1 [\dot{\eta}_1 - \dot{\eta}_5 \cos(\varphi) - \dot{\eta}_6 \sin(\varphi)] + c_1 [\eta_1 - \eta_5 \cos(\varphi) - \eta_6 \sin(\varphi)] = P(t) \cos(\alpha - \beta) \\ m_1 \ddot{\eta}_2 + h_2 [\dot{\eta}_2 - \dot{\eta}_6 \cos(\varphi) + \dot{\eta}_5 \sin(\varphi)] + c_2 [\eta_2 - \eta_6 \cos(\varphi) + \eta_5 \sin(\varphi)] = P(t) \sin(\alpha - \beta) \\ m_2 \ddot{\eta}_5 + h_5 [\dot{\eta}_5 - \dot{\eta}_1 \cos(\varphi) + \dot{\eta}_2 \sin(\varphi)] - h_1 \cos(\varphi) [\dot{\eta}_1 - \dot{\eta}_5 \cos(\varphi) - \dot{\eta}_6 \sin(\varphi)] + h_5 \dot{\eta}_5 + c_5 \sin(\varphi) [\eta_2 - \eta_6 \cos(\varphi) + \eta_5 \sin(\varphi)] - c_1 \cos(\varphi) [\eta_1 - \eta_5 \cos(\varphi) - \eta_6 \sin(\varphi)] + c_5 \eta_5 = 0 \\ m_2 \ddot{\eta}_6 + h_6 [\dot{\eta}_6 - \dot{\eta}_2 \cos(\varphi) + \dot{\eta}_1 \sin(\varphi)] - h_2 \sin(\varphi) [\dot{\eta}_1 - \dot{\eta}_5 \cos(\varphi) - \dot{\eta}_6 \sin(\varphi)] + h_6 \dot{\eta}_6 - c_2 \cos(\varphi) [\eta_2 - \eta_6 \cos(\varphi) + \eta_5 \sin(\varphi)] - c_1 \sin(\varphi) [\eta_1 - \eta_5 \cos(\varphi) - \eta_6 \sin(\varphi)] + c_6 \eta_6 = 0 \\ \varphi = \beta - \delta \end{cases} \quad (1).$$

Equation of motion in the directions of the main coordinates of the workpiece one-mass system shall be also presented in the form of differential equations system:

$$\begin{cases} m_3 \ddot{\eta}_3 + h_3 \dot{\eta}_3 + c_3 \eta_3 = -P(t) \cos(\gamma + \alpha) \\ m_3 \ddot{\eta}_4 + h_4 \dot{\eta}_4 + c_4 \eta_4 = P(t) \sin(\gamma + \alpha) \end{cases} \quad (2).$$

The result of calculations of the system (1) and the system (2) is to determine vibrational motion trajectory of the tool holder m_1 in the directions of the main coordinates η_1 and η_2 , the carriage m_2 in the directions of the coordinates η_5 and η_6 and the workpiece m_3 in the directions of the coordinates η_3 and η_4 . The relationship between the generalized coordinate y_1 with the main coordinates η_1 and η_2 for the tool holder m_1 has the form: $y_1 = \eta_1 \sin(\beta) + \eta_2 \cos(\beta)$, and between the coordinate y_2 and the main coordinates η_3 and η_4 for the workpiece: $y_2 = \eta_3 \sin(\gamma) + \eta_4 \cos(\gamma)$.

The dynamic characteristics of the cutting force $P(t)$ in the above subsystems models are presented in the form [3]: $T_p \frac{dP(t)}{dt} + P(t) = K \cdot (y_1(t) + y_2(t))$, where $y_1(t)$ and $y_2(t)$ - the current values of the coordinates of the relative oscillations of the tooling and the workpiece along axis y during cutting that define cutting layer thickness of the workpiece machined, K - specific cutting force, equal $K = 10C_p t^{x-1} S^y V^n k \frac{t}{\tan \varphi}$, where φ - angle in the cutter plane, T_p - fixed value of the chip formation process, a_0 - slice thickness $a_0 = S \sin \varphi$, ξ - chips shrinkage (for steel $\xi = 3$).

3. Results and discussion

As an example of modelling we shall consider experimentally determined elastic and damping parameters of the carriage systems, basic tool holder and a workpiece of lathe 1K62 dynamic system carriage - tool holder of which during cutting is potentially unstable. Specifications as for principal coordinate directions: given carriage stiffness $c_5=80$ N/ μ m, $c_6=47$ N/ μ m and given mass $m_2=15$ kg, spindle-holder-workpiece system stiffness $c_3=c_4=8$ N/ μ m and mass $m_3=4$ kg, base tool holder stiffness $c_1=c_3=20$ N/ μ m and mass $m_3=4$ kg, or tool holder with oriented center of rigidity $c_1=20$ N/ μ m, $c_6=5$ N/ μ m given mass of which is the same $m_1 = 1.8$ kg. Damping coefficients for each subsystem in the given direction are determined by the formula: $h=2m\omega\delta_0$ kg/s where m – given mass, ω – natural oscillation frequency, δ_0 – oscillations damping logarithmic decrement ($\delta_0=0.31$). Turning angle of stiffness principal axes of the lathe carriage elastic system $\delta=-60^\circ$. Also, cutting force determined main parameters, workpiece material - structural steel, tool - hard alloy T15K6, cutter geometry: $\varphi=60^\circ$, $\gamma=10^\circ$, $\lambda=0^\circ$, $\alpha=5^\circ$, $Tp=3.6 \cdot 10^{-4}$, feed $S=0.39$ mm/rev, cutting angle approximate value $\alpha_p=15^\circ$.

As the cutting rate is proportional to layer thickness of the workpiece machined surface, then using a Nyquist frequency criterion, we can determine the maximum cutting depth t_p which is sufficient if preserving the lathe dynamic system stability during cutting. For metal-cutting lathe systems stability margin on the amplitude $L \geq 8-12$ dB, and the phase $\gamma \geq 30^\circ$ [3]. Fig. 4 shows machining modelling results and limit cutting depth t_p determination for each angle δ value of turning carriage stiffness system main axes and rationally selected value of the axes orientation angle of tool holder stiffness system β where maximum turning vibration resistance is observed, and four different variants of values of the tool holder stiffness parameters C_1 and C_2 in the principal coordinates directions.

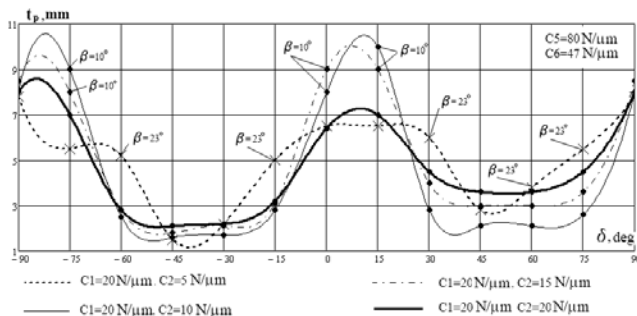


Fig.4. Diagram determining limit cutting depth using tool holder with oriented center of rigidity

According to the obtained data of the conducted theoretical analysis on the determination of rational elastic and dynamic parameters of the lathe tool holder with oriented center of rigidity and experimental study of the effectiveness of its use to ensure the highest machining performance the author has made prototype tool holder [4]. The lathe tool holder design was developed according to the design technique tested and proposed by the author [5] which allows to provide necessary elastic and dynamic parameters of tool holder accurately enough depending on the position coordinates of its center of rigidity relative to the cutter top. Experimental studies were carried out on the lathe 1K62 that has the problem of low vibration resistance of a workpiece due to existing irrational orientation of maximum given stiffness with respect to the cutting force direction. Fig. 5 shows a pilot study stand to determine the cutter vibration level in the directions of the coordinate y and z during machining using a proposed special tool holder compared with a base one.



Fig.5. The stand to study the cutter vibration level using lathe tool holder with oriented center of rigidity when machining on the lathe 1K62

Vibration studies were conducted at the maximum allowable cutting modes for roughing on this lathe, namely: workpiece material – steel 40, aluminum; cutter material - T15K6, blank diameter $D=120$ mm; spindle rotation speed $n=160$ r/min; cutting speed $v=60$ m/min; cutting depth $t=5$ mm; feed $S=0.3$ mm/rev; cutting force value $P=4$ kN. The study results are presented as AFC diagrams of the cutter vibration acceleration amplitude relative level during machining using a base tool holder of Fig. 6 and a proposed tool holder with oriented center of rigidity of Fig.7. Comparing experimental characteristics obtained it can be seen that using a base elastic tool holder during machining self-oscillations on the coordinate y on the lathe carriage dominant potentially unstable system frequency $f=530$ Hz can be observed. The use of the proposed tool holder under the same cutting conditions almost twice reduced the cutter vibration acceleration relative level which confirms the theoretical data of Fig. 4. Thus the use of lathe tool holder with oriented center of rigidity can enlarge machining productivity by increasing cutting modes at vibration level acceptable value.

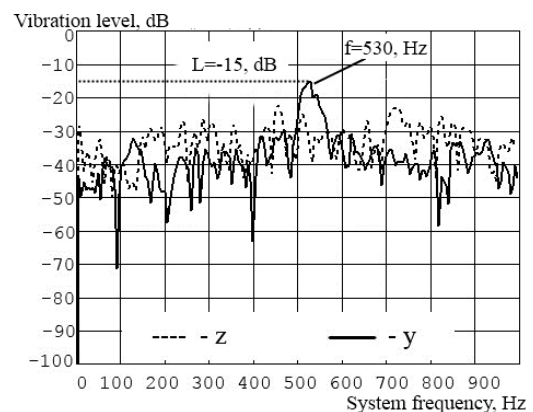


Fig.6. The cutter vibration acceleration level AFC during machining using a regular tool holder

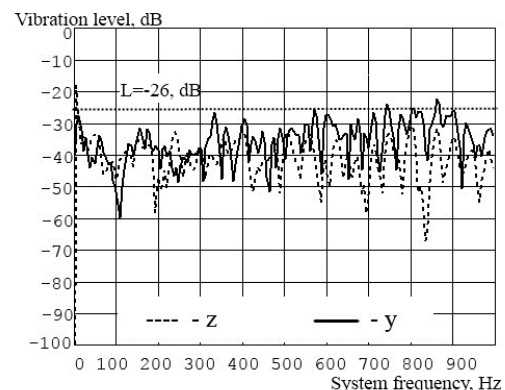


Fig.7. The cutter vibration acceleration level AFC during machining using tool holder with oriented center of rigidity

4. Conclusion

Analysis of the obtained data of mathematical modelling and the cutter vibration acceleration experimental AFC of the studies conducted to determine the maximum allowable vibration-free cutting modes depending on the elastic parameters of the carriage system and tool holder allows to make the following conclusions:

- the use of tool holders with appropriately oriented stiffness allows to reduce self-oscillations amplitudes during machining on the lathe with a potentially unstable elastic carriage system, due to the tool holder system dynamic impact on the carriage dominant system oscillations decreasing coordinate bond negative effect;
- according to the results of theoretical calculations and experimental studies conducted for carriage of lathe 1K62 with a negative orientation angle of axes of stiffness $\delta = -60^\circ$ the use of tool holder with oriented center of rigidity and given elastic parameters $C_{max} = 20 \text{ N}/\mu\text{m}$ and $C_{min} = 5 \text{ N}/\mu\text{m}$, $\beta = 23^\circ$ allows to increase the maximum cutting depth 1.7 times compared to using a base elastic tool holder.

So, theoretical and experimental studies conducted allow recommending the developed mathematical model of the lathe dynamic system to determine appropriate elastic and dynamic parameters of tool holder with oriented center of rigidity in order to increase machining vibration resistance.

5. References

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