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EFFECT OF ORIENTATION OF PRINCIPAL AXISES OF STIFFNESS OF AN ELASTIC-SYSTEM OF THE TOOLHOLDER ON A STABILITY OF TURNING PROCESS

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ВЛИЯНИЕ ОРИЕНТАЦИИ ГЛАВНЫХ ОСЕЙ ЖЕСТКОСТИ УПРУГОЙ СИСТЕМЫ ИНСТРУМЕНТА НА УСТОЙЧИВОСТЬ ПРИ ТОКАРНОЙ ОБРАБОТКЕ

Purpose. Research of dynamic quality coefficients of the cantilevered toolholders and definition of directions of heightening of the conditions of chatter stability at turning.
Design/methodology/approach. In a paper it is shown, that essential effect on a level of relative oscillations of the cutter and a workpiece at turning is related with an angle of rotation of principal axises of a stiffness of an elastic-system of the toolholder and selection of the proportions of stiffness of elements of system. Proposed the theory, which defines conditions of raise of a chatter stability of machining by the nonrigid toolholders and consider a rule of orientation of the principal axises of stiffness of a toolholder in an elastic subsystem of cutter-saddle concerning to a cutting force direction.
Findings. As a result of mathematical simulation the complex mathematical model of the closed dynamic system of a lathe is developed. This model allows to evaluate influence of parameters of the nonrigid toolholders on accuracy of the form of machined surface.
Originality/value. It is shown, that the cutter-saddle elastic-system have the most chatter stability at machining if the angle of rotation of principal axises of a stiffness of this system is equated to half of angle of a cutting force direction. On the basis of the results of theoretical and experimental researches the recommendations for designing and a effective using of nonrigid toolholders at turning are given.
Key words: nonrigid toolholder, oriented axis's of stiffness, turning, chatter stability

Introduction

Feature of turning by the nonrigid cantilevered cutters, such as toolholders, various holders and boring bars, is easiness of beginning of vibrations at cutting that reduces accuracy, quality and productivity of handling of details, restricts technological possibilities of machine tools.At turning by the nonrigid cantilevered toolholders the struggle with needless oscillations is especially important. It is caused by essential effect of such equipment on operation of all technological system of the machine tool. Sampling of rational parameters of the nonrigid tooling by providing of necessary static and dynamic characteristics allows to raise a turning chatter stability essentially.

The tooling can have high enough stiffness, or to be nonrigid. In the first case the oscillation which originate at cutting, are defined by all multiple-loop elastic-system of the machine tool, in the second case – by stiffness and tooling oscillations, and chatter stability loss originates basically on the mode shape of its oscillations. Depending on a worked stock, geometry of the cutter, conditions of cutting and etc the chip forming can be stable with formation of a continuous chips, or unstable with formation of a chips of an incipient fracture or element. Cutting force in the first case is relatively constant, and in second case - periodically varies.

Assurance of necessary vibrational stability of machine tools by using of the nonrigid toolholders, can be carried out as follows: - heightening of static stiffness of the tooling, self-resonant frequency and vibration damping by the rational selection of its design data; - correcting of orientation of the principal axises of stiffness of an elastic-system of the cutter in relation to a cutting force direction; - sampling of a rational proportion of stiffness of the toolholders along the principal axises of stiffness ; - assurance of conditions at which the increase of cutting force cause a pulled position of the cutter from a workpiece; - using of dampers and dynamic vibration suppressor of oscillations.

Research objective

Purpose of this analysis is research of dynamic quality coefficients of the cantilevered toolholders and definition of directions of heightening of the conditions of chatter stability at turning.

Basic maintenance and results of research

One of principal causes of origin of auto-oscillations at cutting is change of the cutting force and area as a result of a relative oscillating motion of the cutter and a workpiece taking into account presence of co-ordinate link between process of cutting and traffic on different co-ordinates of an elastic-system of the machine tool [1, 2].

Maximum changes of the square of a stratum of the metal which is cut off are take place at a result of cutter motion in the normal direction to a cutting surface. Veering of principal axises of rigidity of an elastic-system a cutter-saddle of the machine tool concerning to a cutting force direction it is possible to diminish energy of autoexcitation of auto-oscillations and to ensure vibrational proof process of cutting including also machining by the nonrigid toolholders.

For the majority of real dynamic systems of lathes the lateral oscillations in a plane *yOz* for coordinate system of cutter are dominating. For the mathematical description of an elastic-system of the machine tool in relative motion of the cutter and workpiece in its technological system the cutter-saddle subsystem and a spindle-chuck-detail subsystem are chosen. Models of each of these subsystems are considered as system with the oriented axes of stiffness and integrated into the one complex mathematical model.

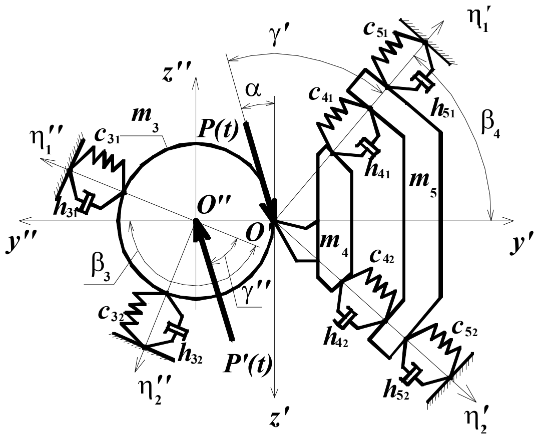
The analytical model of an elastic subsystem a cutter-saddle (fig. 1,a) is presented in the form of two concentrated reduced mass of the toolholder m_4 and a saddle m_5 which are linked among themselves and baseline of the machine tool by links with the elastic and dissipative properties oriented under an angle β_4 to frame $y'O'z'$. On mass m_4 under an angle α concerning a tangential plane of the surface of workpiece the force of cutting $P(t)$ acts. It is supposed, the masses are moving only in directions of principal axises of co-ordinates $O'\eta'_1$ and $O'\eta'_2$. As co-ordinates of moving for mass m_4 are taken co-ordinates η_{4_1} and η_{4_2}, and for mass m_5 co-ordinates η_{5_1} and η_{5_1} are taken. In a subsystem the total reduced coefficients of damping and rigidity h_{41}, h_{42} and c_{41}, c_{42}, and also h_{51}, h_{52} and c_{51}, c_{52} accordingly the toolholder *(m_4)* and a saddle *(m_5)* in directions of principal axises of co-ordinates $O'\eta'_1$ and $O'\eta'_2$ are considered.

The analytical model of an elastic subsystem a spindle-chuck-detail (fig. 1,a) is presented in the form of the concentrated reduced mass m_3, linked with baseline of the machine tool by links with the elastic and dissipative properties oriented under an angle β_3 to frame $y''O''z''$. In a subsystem total reduced coefficients of damping and rigidity h_{31}, h_{32} and c_{31}, c_{32} in principal co-ordinate system $\eta''_1 O'' \eta''_2$ are considered. It is supposed, the masses are moving only in directions of principal axises of co-ordinates $O''\eta''_1$ and $O''\eta''_2$.

On mass m_3 the force of cutting $P'(t)$ which is pair to $P(t)$ acts. The mathematical model of a subsystem is presented as one-mass system with two degree of freedoms, builted in the principal co-ordinate system $\eta''_1 O'' \eta''_2$.

The main drive of the basic model of lathe consist of a motor, belt drive and spindle unit. The dynamic model of the main drive is presented in the form of the two-mass system which moment of inertia, namely, mechanical systems of motor and spindle unit, is considered as subsystems with stiffness links and one degree of freedom, connected by elastic noninertia link which models the belt drive.

Level detection of influence of dynamic characteristics of the saddle-feed drive system on process of shaping a cylindrical surface conducted on an example of a longitudinal turning of short cylindrical workpiece (Fig. 1,b). The model consists of the concentrated reduced mass of a saddle m_5 and toolholder m_4 with a cutter which are connected among themselves by links with elastic and dissipative properties and total reduced coefficients of stiffness $c_{x_{45}}, c_{y_{45}}$ and dampings $h_{x_{45}}, h_{y_{45}}$ accordingly in directions of axes *Ox* and *Oy*. The saddle receive a moving from the longitudinal feed drive through a lead screw t_{XB}. The mechanical system of a revolving part of the feed drive consists of the motor with a reduced moment of inertia J_6 which through toothed belt drive U_p with coefficients of torsional stiffness and damping c_{56} and h_{56} is connected with a lead screw t_{XB}.



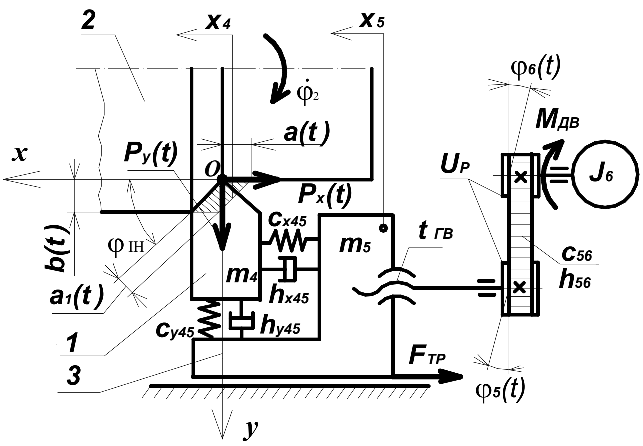


Fig. 1. The model of an elastic-system of a lathe in a plane yOz where the co-ordinate system y'O'z' belongs to a subsystem a cutter-saddle, and co-ordinate system y''O''z''- belongs to a subsystem a spindle-chuck-detail (*a*) and the model of an elastic-system a saddle-feed drive (*b*): 1 - toolholder with a cutter, 2 - workpiece, 3 - saddle

The co-ordinates which define a system translational motion, are for mass m_4 co-ordinate x_4, and for mass m_5 co-ordinate x_5. Generalized co-ordinates for a rotational part of the feed drive are an angle of rotation of lead screw $\varphi_5(t)$ and an angle $\varphi_6(t)$ of rotation of a rotor of motor J_6, reduced to a lead screw by U_p. System is loaded by rotational moment M*(t)*, axial $P_x(t)$ and radial $P_y(t)$ components of cutting force *P(t)*, and also generalized frictional force F_{TP}. It is supposed, that masses m_4 and m_5 moving only in an axis direction *Ox*. Reduction link is the angle of rotation of lead screw $\varphi_5(t)$ of the feed drive.

Cutting force *P (t)*, in the models considered above, is presented by formula:

$$P(t)= K_{уд}\ a(t)\cdot b(t), \tag{1}$$

where *a(t)* and *b(t)* – current values of width and depth of a layer of metal which vary in a time depending on intensity of relative oscillations of the cutter and a workpiece at turning process, $K_{уд}$ - specific cutting force.

Current value of the cutting width of *a(t)* is represented in form:

$$a(t)=\frac{\dot{x}_4(t)}{\dot{\varphi}_2(t)}\cdot 2\pi, \tag{2}$$

where $\dot{x}_4(t)$ - feed rate of the toolholder (m_4), $\dot{\varphi}_2(t)$ - workpiece rotational speed. Current value of depth of cut is defined by components:

$$b(t)=b_0+\Delta_y(t)+e\cdot\sin\dot{\varphi}_2(t), \tag{3}$$

were b_0 – setting depth of cut; *e* – workpiece blank eccentricity; $\Delta_y(t)$ – the component caused by relative elastic deformations of subsystems a cutter-saddle and a spindle-chuck-detail along axis *O'y'* under the loading by cutting force of *P(t).*

Considering time lag and possible interruption of cutting process from the equation (1) we will gain dependence for current value of cutting force:

$$P(\tau)=\begin{cases} K_{рез}\cdot a(t)\cdot b(t)\cdot\left[1-e^{-\frac{t-t_0}{T_p}}\right] & \text{if } a(t)\cdot b(t)>0; \\ 0 & \text{if } a(t)\cdot b(t)\le 0, \end{cases} \tag{4}$$

where T_p – constant of the chip formation process*;* *t* – current time; t_0 - initial time at the moment of incut in a workpiece $(t \ge t_0)$. Having substituted values of *a(t)* and *b(t)* from (2) and (3) in the formula (4) the expression for current value of cutting force *P(t)*, which value depends on relative position of the cutter and workpiece, is determined.

For definition of influence of the relative oscillations of the cutter and a workpiece on accuracy of the form of machined surface is used the special function. Thus, as a result of mathematical simulation the complex mathematical model of the closed dynamic system of a lathe is developed. This model allows to evaluate influence of parameters of the nonrigid toolholders on accuracy of the form of machined surface.

The static characteristic of a cutter-saddle subsystem for the scheme (fig. 1) is given by formula:

$$K_{ПС}=y_4/P=\cos(\beta_4-\alpha)\cdot\sin\beta_4/c_{\eta_2}-\sin(\beta_4-\alpha)\cdot\cos\beta_4/c_{\eta_1}. \tag{5}$$

If to install a range of values of reduced stiffness coefficients of an elastic cutter-saddle subsystem c_{η_1} and c_{η_2} in a direction of principal axises of co-ordinates $O\eta_1$ and $O\eta_2$, then by means of expression (4) there is a possibility to define the angle of rotation β_4 of the principal axises of stiffness of a subsystem at which it will have the minimum flexibility in a direction of axis *Oy '* and to ensure conditions for stable machining.

Calculated values of static characteristic of a cutter-saddle subsystem $K_{ПС}$ are illustrated on fig. 2. At calculations $K_{ПС}$ different proportions of stiffness coefficients c_{η_1} and c_{η_2} were considered, the direction of principal axises of co-ordinates $O\eta_1$ and $O\eta_2$ over the range of angle $\beta = 0\ldots120^0$ varied and a cutting force direction *P* over the range of angle $\alpha = 15^0\ldots 30^0$ varied too.

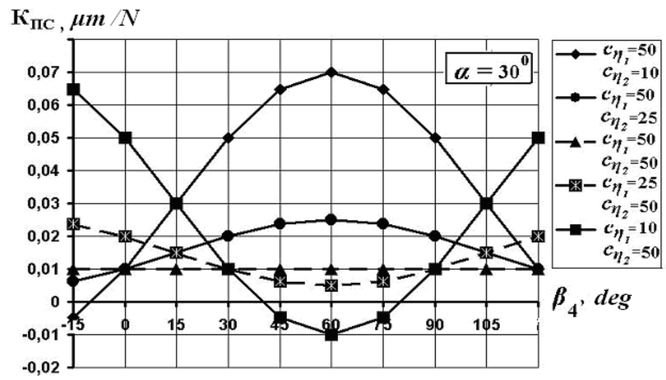


Fig. 2. Graphs of change of static characteristic of a cutter-saddle subsystem $K_{ПС}$ depending on veering of the direction of principal axises of stiffness β_4 and proportions between stiffness coefficients c_{η_1} and c_{η_2} (N/μm) at a cutting force direction under an angle $\alpha = 30^0$

The analysis of results of calculations by formula (4), which is displayed on fig. 2, draw a conclusions: - the essential influence on value of static characteristic of a cutter-saddle subsystem $K_{ПС}$ has an angle of rotation β_4 of the principal axises of stiffness of this subsystem and selection of the proportion of stiffness coefficients along principal axises; - the influence of change of proportions of stiffness coefficients c_{η_1} and c_{η_2} on value $K_{ПС}$ will be minimum provided that an angle of rotation β_4 of the principal axises of stiffness will be equal to half of angle α which defines a cutting force P direction, namely $\beta_4 = \alpha/2$.

For testing of response of mathematical model of the closed dynamic system of a lathe on change of orientation of principal axises of stiffness of a cutter-saddle subsystem in coordinate system $\eta_1' O' \eta_2'$, simulation of process of turning for values of an angle $\beta_4 = 0^0$, 15^0, 45^0, 60^0 is spent. Calculations are executed at a proportion of toolholder stiffness c_{41}/c_{42} = 20/50 N/μm, stiffness coefficients of saddle c_{51}=75 N/μm and c_{52}=50 N/μm and a cutting force direction $\alpha = 30^0$. All parameters of an elastic-system of the machine tool did not vary at change of an angle β_4.

The analysis of calculated trajectories of a cutting point in co-ordinate system $y'O\,z'$ at different angles β_4 has displayed, that only at accomplishment of a condition $\beta_4 = \alpha/2$ the system is most resistant to external loading effects, which imitated the cutting force.

On the basis of the analysis of frequency characteristics of a transfer function of a cutter-saddle elastic subsystem it is proved, that only at angle $\beta_4 = \alpha/2$, change of the proportions of toolholder stiffness from c_{41}/c_{42} =2 to c_{41}/c_{42} = 1/2 practically does not influence increase in amplitudes of oscillation of masses of a subsystem at frequencies of natural oscillations. This testify the realization of conditions for vibrational proof machining by the nonrigid toolholders. Thus, it is possible to recommend a condition $\beta = \alpha/2$ as one of the basic conditions of chatter stability of turning by the nonrigid toolholders. Besides, the maximum calculated values of the cutting width are gained under a condition $\beta = \alpha/2$ and at a proportion of toolholder stiffness in directions of principal axises of stiffness $c_{min}/c_{max} \approx 0{,}7$.

Special toolholder is produced for testing of efficiency of the conditions of chatter stability of turning by nonrigid toolholder with the oriented rigidity [3] in which design the specified conditions are included.

At machining by the given toolholder it is attained increases of the cutting width not less than in 1,4 times in comparison with the base toolholder of the machine tool (Fig. 3).

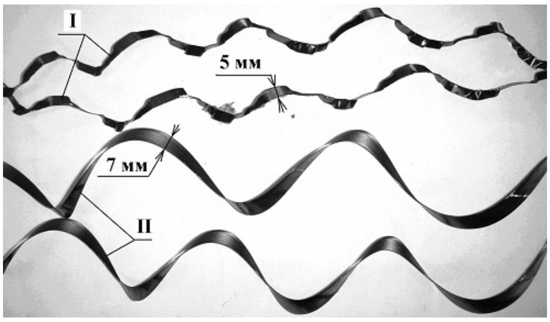


Fig. 3. The examples of a chips which are the results of machining on a lathe model 1А616: I - базовым резцедержателем станка при режимах резания V = 200 м/мин; S = 0,1 мм/об; t = 5 мм; II – резцедержателем с ориентированной жесткостью при режимах резания V = 200 м/мин S = 0,1 мм/об; t = 7 мм

As an example of the practical use of the results of executed researches the modernizing of toolholders of a saddle of a lathe model ПАБ-130 (Kyiv machine tool plant) can confirm the efficiency of the offered conditions. Calculation of polar chart of flexibility of the cantilevered toolholders in plane YOZ is executed by a finite element method (Fig. 4,a). Calculation

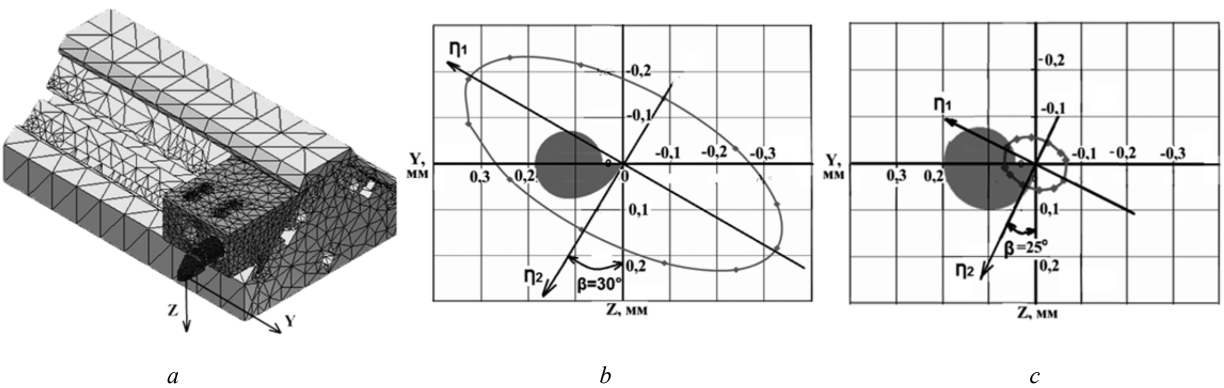


Fig. 4. Model of the instrumental system (*a*) and polar charts of flexibility of the cantilevered toolholder of a base model of lathe - C_{min} = 2,70 N/μm, C_{max} = 6,25 N/μm; C_{min}/C_{max} = 0,43 (*b*) and modificated model - C_{min} = 14,2 N/μm, C_{max} = 20 N/μm, C_{min}/C_{max} = 0,71 (*c*)

of a base model (fig. 4,b) has illustrated, that strains in a point of application of cutting force are primarily defined by strains of a cantilevered part of a toolholder. For a stability improvement at cutting it is offered to change the shape of a cross-section of a cantilevered part of a toolholder (fig. 4,c), that allows to raise stiffness of a cantilevered part of the toolholder several times with taking into account commended proportion $c_{min}/c_{max} \approx 0{,}7$.

Summary

The analysis of results of simulation of shaping process at turning lets the possibility to do following outputs:

- the considerable effect on the level of relative oscillations of the toolholder and workpiece at cutting has an angle of turn of principal axises of stiffness of a cutter-saddle subsystem and proportion of stiffness of a subsystem along principal axises of co-ordinates;
- the cutter-saddle elastic-system have the most chatter stability at machining if the angle β_4 is equated to half of angle of a cutting force *P* direction, namely $\beta_4 = \alpha/2$;
- for increase the limit of the cutting width *b* at machining by the nonrigid toolholders the proportion of stiffness of the toolholder in directions of principal axises of co-ordinates $C_{min}/C_{max} = 0{,}7$ is recommended, under a condition $\beta_4 = \alpha/2$.

Thus, on the basis of executed theoretical and experimental researches the recommendations for projection and an effective application of the cantilevered toolholders are offered for turning processes.

Анотація. *В статті наведені результати теоретичних та експериментальних досліджень, які визначають умови підвищення віброcтійкості токарної обробки нежорстким консольним інструментальним оснащенням, що полягають у врахуванні впливу орієнтації та забезпеченні відповідного положення головних осей жорсткості інструментального оснащення в пружній системі інструменту відносно напрямку дії сили різання та рекомендованому співвідношенні жорсткостей оснащення за цими осями.*

Ключові слова: *токарна обробка, нежорстке інструментальне оснащення, орієнтована жорсткість, віброcтійкість*

Аннотация. *В статье приведены результаты теоретических и экспериментальных исследований, которые определяют условия повышения виброустойчивости токарной обработки нежесткой консольной инструментальной оснасткой, которые состоят в учете влияния ориентации и обеспечении соответствующего положения главных осей жесткости инструментальной оснастки в упругой системе инструмента относительно направления действия силы резания и рекомендованного соотношения жесткостей оснастки вдоль этих осей.*

Ключевые слова: *токарная обработка, нежесткая консольной инструментальная оснастка, ориентированная жесткость, виброустойчивость*

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