Праці Одеського політехнічного університету, 2025. Вип. 1(71)

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

UDC 621.941

A. Orhiian,
A. Balaniuk, PhD, Assoc. Prof.,
A. Orgiyan, DSc., Prof.,
V. Kolesnik, PhD, Assoc. Prof.,
P. Prokopovych
Odessa Polytechnic National University, 1 Shevchenko Ave., Odesa, Ukraine, 65044; e-mail: balanuk.a.v@op.edu.ua

EXPERIMENTAL DETERMINATION OF THE INFLUENCE OF SPINDLE IMBALANCE WITH BORING BAR ON MACHINING ACCURACY

А. Оргіян, Г. Баланюк, О. Оргіян, В. Колеснік, П. Прокопович. **Експериментальне визначення впливу дисбалансу шпинделя із борштангою на точність обробки.** В роботі вивчено вплив параметрів пружної системи оздоблювально-розточувального верстата на статичні деформації, коливання та точність тонкого розточування. На основі експериментальних досліджень впливу дисбалансу шпинделя з борштангою на точність обробки було розроблено методику розрахунку відхилень від круглості, викликаних дією сил різання та відцентрових сил на пружні системи з анізотропною жорсткістю. Також вивчено статичні похибки та особливості деформації у замкнутій динамічній системі верстата. Повне відхилення від круглості, спричинене пружними деформаціями, обчислюється як сума статичних та динамічних складових. Наведено середні значення дисбалансу me, середні квадратичні відхилення χ та відхилення максимальних значень дисбалансу me_{max} до допустимих [me]. Встановлено вплив неврівноваженості на шорсткість обробленої поверхні. Оскільки збільшення відхилень від круглості викликається овалізацією отвору, то причиною порушення точності форми поперечного перерізу зі збільшенням поперечної сили є анізотропність радіальної жорсткості пружної системи верстата. При випробуваннях вузлів одного і того ж типорозміру в однакових умовах овалізація отворів не залишається постійною, тому анізотропність пружних властивостей пов’язана з індивідуальною якістю шпиндельного вузла, а саме, з точністю окремих деталей вузла та їх збіркою.
Ключові слова: шпиндель, місток, стик, підшипник, дисбаланс, відцентрова сила, жорсткість

A. Orgian, G. Balaniuk, A. Orgian, V. Kolesnik, P. Prokopovych. **Experimental determination of the influence of spindle imbalance with boring bar on machining accuracy.** The influence of the parameters of the elastic system of the finishing and boring machine on static deformations, oscillations and accuracy of fine boring is studied. Based on experimental studies of the effect of spindle imbalance with a boring rod on machining accuracy, a method was developed for calculating deviations from roundness caused by the action of cutting forces and centrifugal forces on elastic systems with anisotropic rigidity. Static errors and deformation features in the closed dynamic system of the machine were also studied. The total deviation from roundness caused by elastic deformations is calculated as the sum of static and dynamic components. The mean values of the imbalance me, the mean square deviations of χ and the deviation of the maximum values of the imbalance me_{max} to the permissible [me] are given. The effect of imbalance on the roughness of the treated surface has been established. Since the increase in deviations from roundness is caused by the ovalization of the hole, the reason for the violation of the accuracy of the cross-sectional shape with an increase in the transverse force is the anisotropy of the radial rigidity of the elastic system of the machine. Testing units of the same standard size under the same conditions, the ovalization of the holes does not remain constant, therefore, the anisotropy of elastic properties is associated with the individual quality of the spindle assembly, namely, with the accuracy of individual parts of the unit and their assembly.
Keywords: spindle, bridge, joint, bearing, imbalance, centrifugal force, stiffness

Introduction

The forces of unbalanced rotating masses (centrifugal forces) are an example of perturbing forces that produce forced vibrations. Every real shaft, due to inevitable manufacturing inaccuracies and inhomogeneity in the shaft material, has some static and dynamic unbalance. Centrifugal forces of unbalanced rotating masses can excite forced oscillations of two types: oscillations of the structure on which the rotating part is fixed and oscillations of the rotating part itself. In finishing boring machines, the most critical unit is the spindle head with the tool, the vibrations of which determine the accuracy of boring. With increasing speeds of the main cutting force, as well as the force of unbalanced masses, there are additional dynamic loads on the bearing systems of machine tools, on the spindle supports with rolling bearings, which increases the wear of elements of elastic systems and leads to loss of accuracy [1]. Studies of dynamic models of the spindle-boring bar subsystem allow to determine with great accuracy the characteristics of vibrations, their amplitudes and frequencies, as well as to perform their effective damping. By studying the process of balancing spindles with tools, it is possible to improve the accuracy of finish boring by simple means [2].

DOI: 10.15276/opu.1.71.2025.02

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)
Proceedings of Odessa Polytechnic University, Issue 1(71), 2025

Analysis of recent publications and problem statement

At high main motion speeds, the unbalance force can increase the cutting force, disrupting workpiece shaping [3, 4]. In [5], modern designs of spindle assemblies for high-speed cutting are presented. The main elements of spindle assemblies are described, taking into account future design trends. Measuring systems, drives and sensors are investigated in order to improve the dynamic quality of machine tools. In [6], heat dissipation during spindle operation and methods of heat dissipation are studied. Heat dissipation heat dissipation paths were designed at the joints of bearings with the spindle housing in radial direction. To reduce heat transfer, the inner surface of the track wall was covered with a heat-insulating material. Therefore, heat was directly transferred from inside the spindle head to the external environment. The determining effect of heat dissipating tracks on the heat transfer efficiency was investigated. In [6, 7], some sources of unbalance of the spindle-tool system were studied experimentally, and then the quality of dynamic balancing for the whole system was presented. At idle speed, forces were studied at three test cases of machine tool vibration signal and its characteristics: spindle without tooling system, spindle with fully balanced tooling system and spindle with different unbalance of tooling system [8, 9]. To predict the dynamic interactions considering the assembly, modelling of the spindle head was performed using the tool holder and support connections. The connection parameters of the spindle toolholder were determined from the experimental platform based on model separation, and the parameters of the support connections were determined from Hertz contacts. Modal and harmonic response analysis was performed to design the spindle systems. The model is able to predict the corresponding frequencies and amplitudes of the spindle system. An important result of the spindle control system is the reduction of machining errors in real time [10, 11]. A dynamic model of the relationship between machining technology and cutting process has been developed for the control system. The transfer matrix method of multi-mass system was used. An elastic element is introduced in the model for tool to workpiece contact. The transfer matrix method is used to calculate the harmonic influence coefficients between the tool and workpiece. This technique is used to calculate the natural frequencies of vibration of the spindle with the workpiece and tool support [12, 13].

The development of calculation models of spindle heads allows to determine their frequencies, forms of unstable vibrations, as well as stiffness [14, 15]. Based on the use of balancing medium for automatic balancing of rotors, new results on the value of unbalance force were obtained [16].

Problems of static balancing of spindles with an unbalanced workpiece have been studied on the basis of random search for devices for moving corrective weights onto the spindle [17 – 20].

Purpose of the study

Experimental determination of roundness deviations caused by cutting force and centrifugal force on systems with anisotropic stiffness.

Objectives of the study: in order to achieve the objective, the following tasks were accomplished:

– determine the sources of anisotropy in the elastic system of the machine tool;

– determine the stiffness of the joint between the head and the bridge;

– determine the non-uniformity of radial stiffness over the angle of rotation caused by raceway deformations in the spindle support bearings;

– determine the effect of spindle-tool imbalance on machining quality.

Statement of the main material

There are machining conditions in which the workpiece-fixture ductility significantly exceeds the value corresponding to standard tests (e.g., axial ductility can exceed 0.01 μm/N). In these cases, the deformability of the workpiece-fixture system cannot be neglected. The statistical machining error is caused by the combined action of the cutting force and centrifugal force, with the workpiece-fixture subsystem being loaded by the cutting force alone. If the spindle-tool system is balanced, the static component of the roundness deviation is equal:

$$\Delta R_c = 0.5\sqrt{(\Pi_e^2 + \Pi_o^2 + \Pi_k^2)},$$

where: Π_e, Π_o, Π_k are the components of the static error [21], and in their calculation the value of the sum of the pliability of the spindle-tool and workpiece-fixture system is entered.

Determining the dynamic error is reduced to calculating the amplitude of forced oscillations of the cutter relative to the workpiece. The total deviation from roundness caused by elastic deformations is calculated as the sum of the static and dynamic components.

Праці Одеського політехнічного університету, 2025. Вип. 1(71) ISSN 2076-2429 (print) ISSN 2223-3814 (online)

In this paper, the allowable values of unbalance are determined and it is shown that the real values of unbalance on the spindles of UDB (universal-diamond boring head) do not exceed the allowable values. Table 1 presents the average unbalance values m_e, the mean square deviations χ and the ratios of the maximum unbalance values $m_{e\max}$ to the tolerance values $[m_e]$.

The effect of spindle-tool imbalance on machining quality is manifested in the increase of deviations from roundness of the hole (Table 2, steel 20, UDB-1, $t = 0.1$ mm, $S = 0.03$ mm/rev, $V = 350$ m/min.

Table 1

Average imbalance values

Spindle unit size	On the flange			On the pulley		
	m_e, g/cm	χ, g/cm	$\frac{m_{e\max}}{[m_e]}$	m_e, g/cm	χ, g/cm	$\frac{m_{e\max}}{[m_e]}$
UDB-1	12.5	6.4	0.8	8.4	5.6	0.6
UDB-2	15.8	9.1	0.2	9.2	5.7	0.2
UDB-3	17.5	11.2	0.1	16.8	7.6	0.1

Table 2

Effect of spindle-tool imbalance on machining quality

Centrifugal force, N	On the flange				On the pulley		
	0	80	120	250	130	180	250
Deviation from roundness, µm	0.5	0.5	0.7	1.8	0.6	1.0	1.8
Roughness, Ra, µm	0.12	0.16	0.18	0.16	0.32	0.32	0.32

It can be seen that the effect of unbalance on machining roughness is weakly expressed. The increase in non-roundness is caused primarily by the ovalization of the hole. Therefore, the reason for the violation of the accuracy of the hole shape when the transverse force increases is the anisotropy of the radial stiffness of the machine tool elastic system. When testing assemblies of the same size under the same conditions, the ovalization of the bore does not remain constant; therefore, the anisotropy of the elastic properties is related to the individual qualities of the spindle assembly, namely the accuracy of the individual parts of the assembly and their assembly.

Possible sources of anisotropy can be: the bridge, the joint between the head and the bridge, and the mating of the fixed bearing rings with the head housing. Bridges of finishing boring machines have the form of a portal. The bridge is represented by a frame with rigidly embedded struts, in which the axial moments of inertia of the cross-section of the transom and struts are calculated from the corresponding cross-sections of the bridge. Table 3 shows the values of bridge pliability in horizontal and vertical directions for some types of machines.

Table 3

Values of bridge pliability in horizontal and vertical directions

№ n/a	Machine model	Pliability of the bridge $K - 10^2$, µm/N		
		In the vertical direction, K_z	In the horizontal direction, K_y	Difference, $K_z - K_y$
1	Toyoda	0.25	0.12	0.13
2	2706	0.23	0.59	–0.036
3	2712	0.21	0.28	–0.07
4	2704B	0.036	0.053	–0.017
5	2706B	0.11	0.04	0.07
6	2712B	0.095	0.056	0.029

The difference $K_z - K_y$, which characterizes the anisotropy of the bridge, may increase compared to the data in the table when the loading force P and the measuring point are moved along the spindle axis, since torsional deformations of the transom and struts are taken into account. Let's analyse the displacements occurring in the tool cross-section when the spindle with the tool is loaded with radial force in this cross-section due to deformations in the junction of the head with the bridge. Fig. 1 shows the loading scheme of the joint at the vertical position of the transverse force Q.

The action of the force Q is reduced to loading the joint with the moment M_y, causing the joint to rotate with respect to the axis y orthogonal to the drawing plane by an angle φ_y, and to the vertical force Q applied at the centre of the joint. A view of the contact surface is shown in Fig. 2.

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)
Proceedings of Odessa Polytechnic University, Issue 1(71), 2025

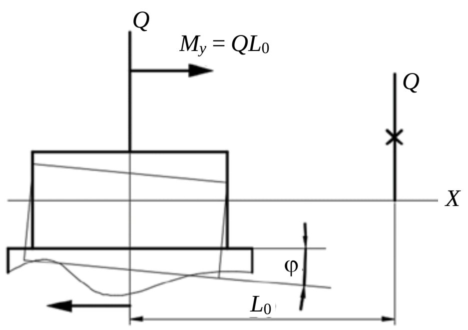

Fig. 1. Scheme of loading of the joint of the head housing with the bridge by vertical force Q

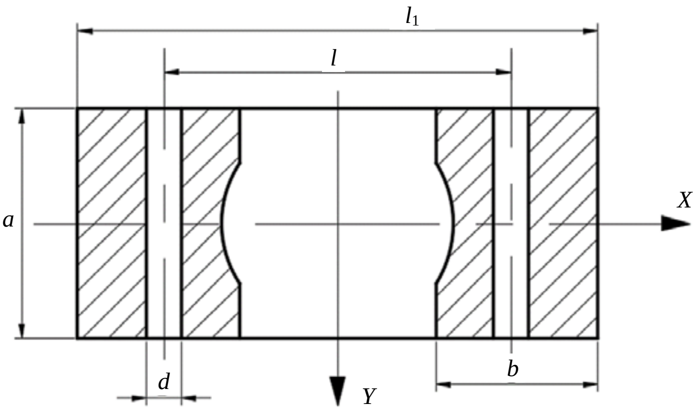

Fig. 2. Shape of the surface of the joint between the head housing and the bridge

The geometric characteristics of the joint are the moments of inertia J_y and J_x, and the area F. The moment $M_y = QL$ causes the joint rotation angle:

$$\varphi_y = \frac{k' M_y \mu_\varphi}{J_y},$$

and the force Q is the vertical displacement of the joint:

$$\delta_z = k'\sigma\mu',$$

where: $k' = \dfrac{cm}{(\sigma+\sigma_0)^{(1-m)}}$ – coefficient of contact pliability; c – coefficient depending on geometry of surfaces and type of material; m – exponent of degree; σ_0 – initial average pressure, N/cm^2; μ_φ, μ' – coefficients taking into account the influence of deviations from flatness on elastic displacements in flat joints.

At convexity of one of the mating surfaces μ_φ can vary within $1-13$, at other types of deviations from flatness $\mu_\varphi = 0.7...1.4$, μ' can vary from 1 to 5.

For mating parts made of cast iron with ground surfaces it is possible to take $c = 0.4$, $m = 0.5$. The value σ is equal to the ratio of Q to the joint area, and σ_0 is the ratio of the force P, with which the head body is bolted to the bridge, to the joint area. Taking into account $\sigma = \dfrac{Q}{F}$, $\sigma_0 = \dfrac{P}{F}$ and the values of c and m can be written:

$$\varphi_y = \frac{0.2\mu_\varphi\sqrt{F}}{J_y}\frac{QL_0}{\sqrt{Q+P}};$$

$$\delta_z = \frac{0.2\mu'}{\sqrt{F}}\frac{Q}{\sqrt{Q+P}}.$$

Full travel in any tool section:

$$\delta_B = \varphi_y L_x + \delta_z = \frac{0.2\sqrt{F}\cdot Q}{\sqrt{Q+P}}\left[\frac{\mu_\varphi L_0(L_0+l_x)}{J_y} + \frac{\mu'}{F}\right], \tag{1}$$

where: $l_x = L_x - L_0$ is the distance from the point of force application to the section where the displacement is determined.

When the force Q is horizontal, its action on the joint is equivalent to the action of the moments M_z and M_x and the central force Q (Fig. 3).

Праці Одеського політехнічного університету, 2025. Вип. 1(71)
ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

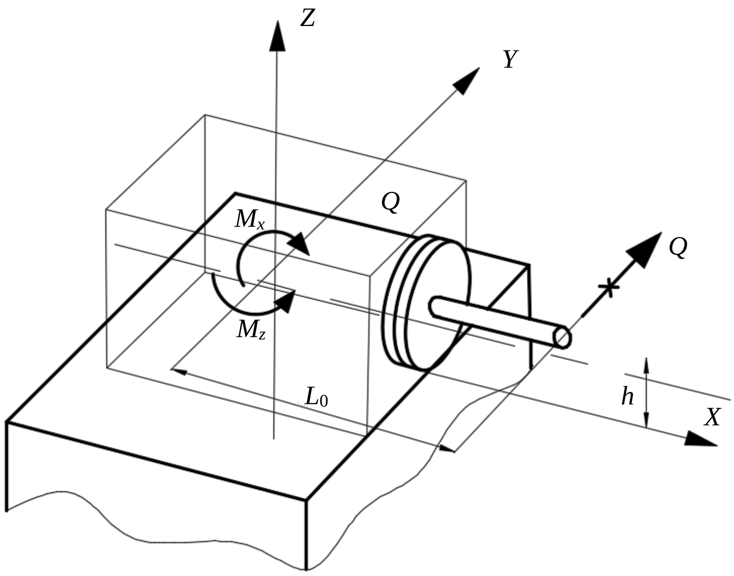

Fig. 3. Schematic of loading the joint with horizontal force Q

Let us determine the displacement at the point of application Q, caused by the moment M_z, namely $M_z = QL_0 = Rl$, where R is the resultant tangential stress at the back or front of the joint. Hence:

$$R = \frac{QL_0}{l}.$$

Average tangential stresses in the joint from the action of M_z:

$$\tau_M = \frac{2R}{F} = \frac{2QL_0}{lF}.$$

Angle of rotation θ_z, caused by M_z:

$$\theta_z = \frac{2k_\tau \tau_M}{l} = \frac{4k_\tau QL_0}{Fl^2},$$

where: k_τ – is the contact tangent ductility coefficient.

At contact of ground surfaces of ferrous metals $k_\tau = 0.025...0.012 \frac{\mu\text{mN}^{-1}}{\text{cm}^2}$, if normal pressure in the joint $\sigma = 50...150 \frac{\text{N}}{\text{cm}^2}$.

Tangential displacements in the joint from the central force $\delta_y = k_t \frac{Q}{F}$. The angle of rotation of the joint about the x-axis caused by the moment $M_x = Qh$, is determined by analogy with φ_y:

$$\theta_x = \frac{0.2\mu'_\varphi \sqrt{F}}{J_x} \frac{Qh}{\sqrt{Q+P}},$$

where: μ'_φ takes into account the type of non-planarity of the mating surfaces in the *YOZ* plane.

Total displacement at the point of force application Q when the force is horizontal:

$$\delta_{\text{hor}} = \theta_z L_0 + \delta_y + \theta_x h = \frac{k_\tau Q}{F}\left(1 + \frac{4L_0^2}{l^2}\right) + \frac{0.2\mu'_\varphi \sqrt{F}}{J_x} \frac{Qh^2}{\sqrt{Q+P}}. \quad (2)$$

Using formulas (1) and (2) it is possible to calculate the values of displacements in vertical and horizontal planes in any cross-section under the action of radial force Q and to estimate the value of

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)
Proceedings of Odessa Polytechnic University, Issue 1(71), 2025

hole ovality Δ_{ov}, resulting from the difference between δ_{ver} and δ_{hor} . Note that for the calculation it is necessary to choose correctly the coefficients μ_φ, μ'_φ and μ'.

Since the head body is joined to the bridge by small areas (see Fig. 2), the influence of convexity of one of the mating surfaces on the value of φ_y is excluded. Other types of non-planarity have much less influence on the deformations in the joint. In addition, careful machining of the mating surfaces must be taken into account. The hull bottom is lapped and the bridge surface is scraped. Therefore, the values of the coefficients μ_φ, μ'_φ and μ' may differ only slightly from 1.

The value of P *is* defined as the sum of the four forces acting in each M16 clamping bolt. For bolt M16 made of steel 35 $[P]=12$ kN. Calculations were further carried out at $P=40$ kN, which is slightly less than the permissible value.

Let's carry out the calculation for the head UDB-1, the dimensions of the joint of which are minimal, and the spindle speed is the highest compared to other sizes.

Initial data: $F=220$ cm^2; $J_y=47200$ cm^4; $J_x=3860$ cm^4. The dependences of the value of $2\Delta R$ on Q, P, L_0 are shown in Figs. 4, 5, 6.

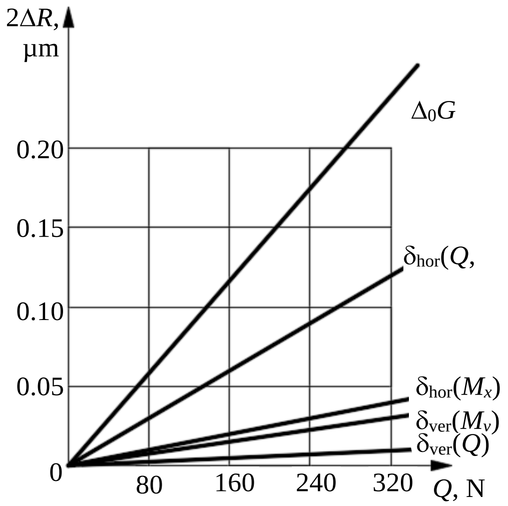

Fig. 4. Dependence of deviations from roundness of the hole when changing the value of radial force Q at $L_0=32$ cm, $P=40$ kN

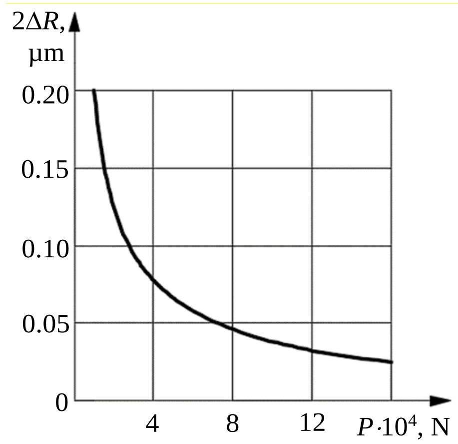

Fig. 5. Dependence of deviation from roundness of a hole at change of force P with which the head is pressed to the bridge of the machine at $Q=100$ N, $L_0=32$ cm

The ductility of the joint in the horizontal direction is greater than in the vertical direction. Therefore, for machines 2712B, 27068 (Table 2) the differences K_z-K_y for the bridge and the joint are obtained with different signs, and for machine 27048 with the same sign.

Thus, the anisotropy of the fixed head body-bridge system depends on the ratio of the anisotropy of the bridge and the junction.

The non-uniformity of radial stiffness over the angle of rotation is also related to raceway deformations in the spindle support bearings and depends on the accuracy of the parts mating with the bearing.

Experimental studies of the radial stiffness of spindle assemblies mounted on the finishing and boring machines bridge have shown great variability in the shape of the radial stiffness measured by the spindle rotation angle. The measurements were performed on a large number of spindle assemblies on the same machine with a constant loading and measuring system. Since the changes in the experimental conditions were related only to individual features of each node, it is natural to assume a major role of mounting and precision parameters of spindle supports in the formation of inhomogeneities of radial stiffness. This is also evidenced by the fact of a significant change in the radial stiffness diagram after running-in of the head (Fig. 7).

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

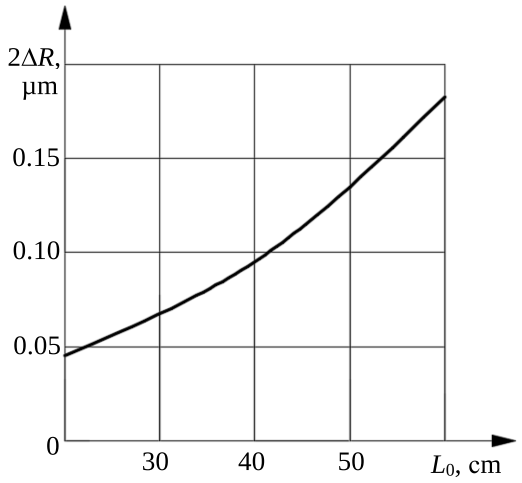

Fig. 6. Dependence of hole roundness deviation on the point of Q application along the mandrel axis at $Q = 100$ N, $P = 40$ kN

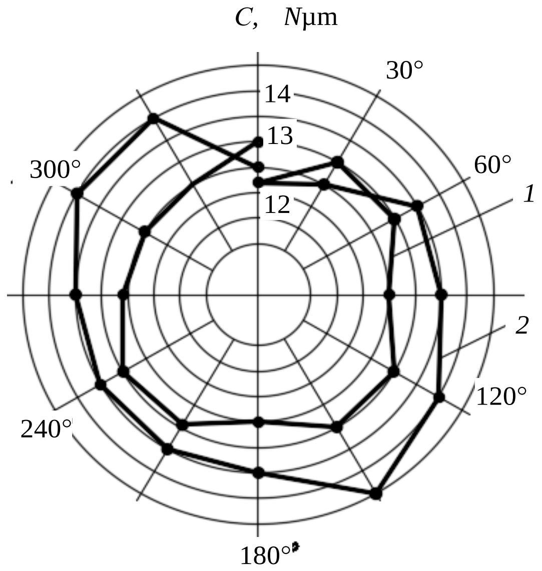

Fig. 7. Radial stiffness diagrams before running in (*1*) and after (*2*)

The results of measurements of pliability of assemblies with a control mandrel ($d = 30$ mm; $l = 100$ mm) are presented in Table 4.

Table 4

Results of measuring the pliability of assemblies with a control mandrel

Spindle unit size	Average value of radial pliability K, μm/N			Number of units tested
	K_{max}	K_{min}	ΔK	
UDB-1	0.092	0.076	0.016	5
UDB-2	0.067	0.052	0.015	6

The disadvantage of the applied measurement method is that the displacement reference point was a point on the bridge surface at the place of attachment of the lever with the indicator. Therefore, the measurements do not fully take into account the deformation of the bridge. More accurate measurements would be made if the displacement indicator was mounted on the machine bed.

The experimental data in Table 5 correspond to a certain mandrel length. The value of the pliability K and, accordingly, ΔK, is likely to change when the mandrel outreach is changed. Let us evaluate the nature of these changes. Consider the spindle as a two-axis rigid beam on linear-elastic supports. Reactions in rear A and front B supports:

$$R_{\mathrm{A}} = \frac{Qb}{a}; \; R_{\mathrm{B}} = \frac{Q(a+b)}{a}, \tag{3}$$

where: a is the inter-support distance, b is the distance from the front support to the section of the mandrel where the radial force Q *is* applied.

Displacement in support sections:

$$\delta_{z\mathrm{A}} = K_{z\mathrm{A}} R_{\mathrm{A}}; \; \delta_{z\mathrm{B}} = K_{z\mathrm{B}} R_{\mathrm{B}}, \tag{4}$$

where: $K_{z\mathrm{A}}$ and $K_{z\mathrm{B}}$ are pliability of supports in the direction of Z axis.

Movements at outreach *L:*

$$\delta_{zL} = \frac{(\delta_{z\mathrm{A}} + \delta_{z\mathrm{B}})(a+b)}{a}. \tag{5}$$

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)
Proceedings of Odessa Polytechnic University, Issue 1(71), 2025

Displacements at outreach L taking into account (3), (4) and (5):

$$K_{zL} = \frac{\delta_{zL}}{Q} = \left[K_{z\text{B}} + \frac{b}{a}(K_{z\text{B}} + K_{z\text{A}})\right]\left(1 + \frac{b}{f}\right).$$

A similar expression is obtained for K_{yL} , and $K_{y\text{A}} = K_{z\text{B}}$.
Then:

$$K_{zL} = K_{zL}\left[\left(1 + \frac{b}{a}\right)^2 + \frac{b}{a}\right] \text{ and } 2\Delta K_L = 2\Delta K_\text{B}\left[\left(1 + \frac{b}{a}\right)^2 + \frac{b}{a}\right], \tag{6}$$

where: $2\Delta K_\text{B} = K_{z\text{B}} - K_{y\text{B}}$.

From (6) we can see that the ductility changes according to the parabolic law when changing the mandrel outreach, and the argument includes the mandrel length. With the help of (6) and experimentally obtained value $2\Delta K \quad (L = 90)$ (Table 4) it is possible to obtain values $2\Delta K_\text{B}$ and then the character of anisotropy variation along the mandrel length will be given. Fig. 8 shows the characteristic of anisotropy of heads as a function of mandrel length L.

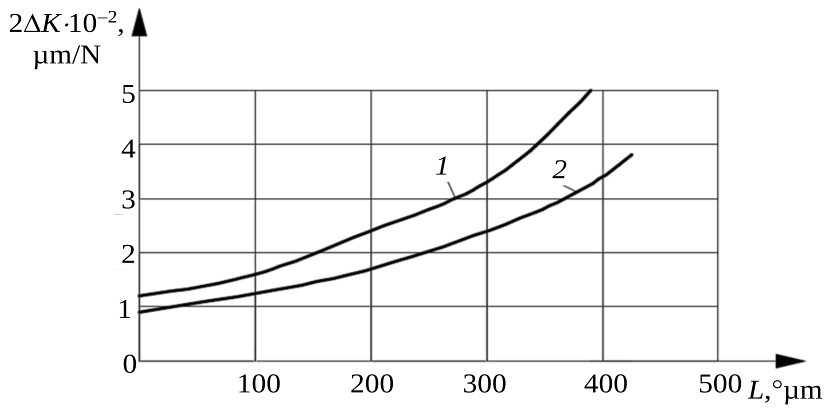

Fig. 8. Variation of anisotropy of radial pliability along the mandrel length: *1* – UDB-1, UDB-2; *2* – UDB-3

he anisotropy of the part subsystem is determined by the elastic properties of the part, the table fixture and their connections. The part can have a wide variety of configurations. The fixture is rigid enough, it has no moving contacts and its anisotropy will be lower than in the tool subsystem. The deformation of the fixture of the "vertical post" type under the action of a force directed along the Y-axis can be considered as a transverse bending of the console, the moment of inertia of the cross-section J_x of which is determined by the horizontal cross-section of the fixture. In the Z-axis , the pliability of the fixture is determined by the compressive strain. The relationship K_z and K_y is expressed by the formula:

$$\frac{K_z}{K_y} = \frac{J_x}{Fh^2},$$

where: F is the area of the average horizontal section of the fixture, h is the height of the axis of the specimen being bored above the table mirror.

Estimated fixture stiffness for machine tool 2712A: $C_y = 6.7 \cdot 10^4$ N/μm. This value is significant and the anisotropy of the workpiece subsystem obviously has little dependence on the elastic properties of this type of fixture. Therefore, it can be said that the anisotropy of the part subsystem is determined by the shape and dimensions of the part, its connection to the fixture, and the moving contact between the table and the bed. Figure 9 shows the variation of maximum deviations from roundness of machined holes when changing the length of boring bars mounted on balanced and unbalanced spindles.

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

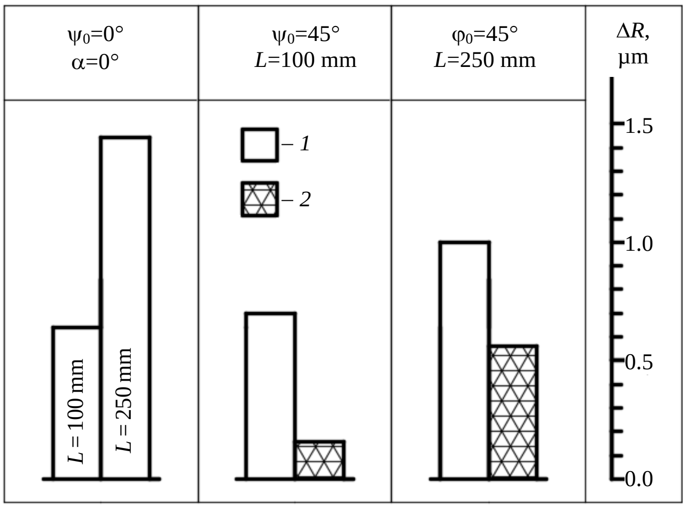

Fig. 9. Maximum hole unroundness *ΔR*: *1* – without balancing, *2* – after balancing

Conclusions

Unbalanced spindles have little effect on surface roughness. The main cause of hole shape accuracy violation with increasing shear force is anisotropy of radial stiffness of the machine tool elastic system. Anisotropy of elastic properties is related to the accuracy of individual parts of the assembly and their mounting. It has been established that the sources of anisotropy can be: the bridge, the joint of the head with the bridge, as well as the mating of fixed bearing rings with the head housing.

The dependence of the hole ovality on the radial force Q at the head-to-bridge clamping force P from 10 kN to 120 kN has been determined. With the increase of *P* within the specified limits from 4 kN to 120 kN the deviations from roundness decrease from 0.2 μm to 0.03 μm. Thus, the anisotropy of the fixed hull-bridge system depends on the anisotropy ratio of the bridge and the joint.

Comparing the contents of Table 3 with the calculated data on bridge and joint anisotropy, we conclude that the anisotropy of the spindle subsystem is determined by the rolling supports and their connection to the head housing: the contribution of bridge deformations to the pliability anisotropy is on average $2...5\%$, and that of joint deformations $1...2\%$. Obviously, as the accuracy of spindle assemblies increases, the balance of anisotropy will change towards an increase in the contribution of the bridge and joint. Therefore, for machine tools of accuracy classes A and C, the anisotropy of elastic properties created by bridge and joint deformations can be significant.

Maximum deviations from the roundness of the holes at the outreach of the boring bar from 100 mm to 250 mm, without balancing varies from 0.6 µm to 1.0 µm, and after balancing is in the range from 0.1 µm to 0.6 µm.

Література

1. Oborskyi H., Orgiyan A., Balaniuk A. Balancing spindles with tools for finishing and boring machines. *Proceedings of Odessa Polytechnic University*. 2023. Issue 1(67). P. 5–14. DOI: 10.15276/opu.1.67.2023.01.
2. Залога В. О., Криворучко Д. В., Шаповал Ю. В., Дрофа К. А. Динамічне управління коливаннями при точінні. *Mechanics and Advanced Technologies*. 2017. No79. С. 100–107.
3. Залога В. А., Зинченко Р. Н., Шаповал Ю. В. Обработка деталей малых диаметров точением с высокой частотой вращения шпинделя. *Сучасні технології в машинобудуванні: збірник наукових праць*. 2014. Вип. 9. С. 50–62.
4. E. Abele, Y. Altintas, C. Brecher. Machine tool spindle units. *CIRP Annals Volume*. 2010. 59, Issue 2. P. 781–802. DOI: https://doi.org/10.1016/j.cirp.2010.05.002.
5. A heat dissipation enhancing method for the high-speed spindle based on heat conductive paths / Li Y, Yu M, Bai Y, Hou Z, Zhang H, Wu W. *Advances in Mechanical Engineering*. 2023. 15(4). DOI: 10.1177/16878132231167675.

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)
Proceedings of Odessa Polytechnic University, Issue 1(71), 2025

6. Wu Y., Zhang L. Basic Theory and Method of Spindle Dynamic Balance. In: *Intelligent Motorized Spindle Technology. Springer Tracts in Mechanical Engineering*. Singapore: Springer, 2020. DOI: https://doi.org/10.1007/978-981-15-3328-0_4.
7. Shen C., Wang G., Wang S., Liu G. The Imbalance Source of Spindle-Tool System and Influence to Machine Vibration Characteristics. In: *2011 Second International Conference on Digital Manufacturing & Automation*. Zhangjiajie, China, 2011. P. 1288–1291. DOI: 10.1109/ICDMA.2011.317.
8. Данильченко Ю. М., Петришин А. І. Моделювання форм коливань механічної коливної системи «шпиндельний вузол-основа». *Надійність інструменту та оптимізація технологічних систем*. 2012. No 30. С. 309–316.
9. Данильченко Ю. М., Петришин А. І. Динамічний аналіз механічної коливної системи «шпиндельний вузол-основа». *Надійність інструменту та оптимізація технологічних систем*. 2011. No 28. С. 169–174.
10. Dynamic characteristics of “tool-workpiece” elastic system in the low stiffness parts milling process / Danylchenko Y., Petryshyn A., Repinskyi S., Bandura V., Kalimoldayev M., Gromaszek K., Imanbek B. In: *Mechatronic Systems 2: Applications in Material Handling Processes and Robotics*. London, UK : Routledge, 2021. P. 225–236.
11. Данильченко Ю. М., Петришин А. І. Идентификация колебаний шпинделя по результатам измерения вибраций корпуса шпиндельного узла. *Вісник НТУУ «КПІ», сер. Машинобудування*. 2014. No 71. С. 147–152.
12. Danylchenko Y., Storchak M., Danylchenko M., Petryshyn A. Cutting Process Consideration in Dynamic Models of Machine Tool Spindle Units. *Machines*. 2023. 11, 582. DOI: https://doi.org/10.3390/machines11060582.
13. Сахно Є.Ю. Волик В.С. Механічна обробка незрівноважених деталей на токарному верстаті з гідростатичнимі опорами. *Вісник двигунобудування*. 2006. No2. С. 129–133.
14. Струтинський В. Б., Федориненко Д. Ю. *Статистична динаміка шпиндельних вузлів на гідростатичних опорах*: монографія. Ніжин : ТОВ «Видавництво «Аспект-Поліграф», 2011. 464 с.
15. Шаповал Ю. В., Криворучко Д. В. Стенд для исследования процесса точения с высокими частотами вращения шпинделя. *Журнал інженерних наук*. 2014. Т. 1, No 3. С. 11–18.
16. Драч І. В., Ройзман В. П. Автоматичне балансування обертових тіл рідиною: *монографія. Хмельницкий : ХНУ*, 2018. 189 с.
17. Ройзман В.П., Драч І.В. Теоретичне дослідження процесу автоматичного балансування роторів з вертикальною віссю обертання рідкими робочими тілами (випадки ідеальної та в’язкої рідин). *Вібрації в техніці та технологіях*. 2015. No3 (79). С. 50–58.
18. Кальченко В.І., Сахно Є.Ю., Федориненко Д.Ю. Шляхи вдосконалення процесу та пристроїв балансування роторів. *Вісник Чернігівського технологічного інституту*. 1996. N1. С. 111–118.
19. Dynamics prediction of spindle system using joint models of spindle tool holder and bearings / Xu C., Zhang J., Yu D., Wu Z., Feng P. *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*. 2015. 229(17). P. 3084–3095. DOI: 10.1177/ 0954406215569588.
20. Rauscher C. Grundlagen der Spektrumanalyse. Rohde & Schwarz, 2011.
21. Orgiyan A., Oborskyi G., Balaniuk,A., Tonkonogyi V., Dasic P. Development of Calculation of Statistical and Dynamic Errors upon Fine Boring with Console Boring Bars. *Lecture Notes in Mechanical Engineering*. 2021. P. 588–597. DOI: https://doi.org/10.1007/978-3-030-68014-5_57.

References

1. Oborskyi, H., Orgiyan, A., & Balaniuk, A. (2023). Balancing spindles with tools for finishing and boring machines. *Proceedings of Odessa Polytechnic University, 1*(67), 5–14. DOI: 10.15276/opu. 1.67.2023.01.
2. Zaloha, V. O., Kryvoruchko, D. V., Shapoval, Y. V., & Drofa, K. A. (2017). Dynamic control of vibrations during turning. *Mechanics and Advanced Technologies*, (79), 100–107.
3. Zaloha, V. A., Zinchenko, R. N., & Shapoval, Y. V. (2014). Machining small diameter parts by turning with high spindle speed. *Modern Technologies in Mechanical Engineering: Collection of Scientific Works*, (9), 50–62.
4. Abele, E., Altintas, Y., & Brecher, C. (2010). Machine tool spindle units. *CIRP Annals Volume, 59*(2), 781–802. https://doi.org/10.1016/j.cirp.2010.05.002.
5. Li, Y., Yu, M., Bai, Y., Hou, Z., Zhang, H., & Wu, W. (2023). A heat dissipation enhancing method for the high-speed spindle based on heat conductive paths. *Advances in Mechanical Engineering*, *15*(4). https://doi.org/10.1177/16878132231167675.

ISSN 2076-2429 (print)
ISSN 2223-3814 (online)

6. Wu, Y., & Zhang, L. (2020). Basic theory and method of spindle dynamic balance. In *Intelligent Motorized Spindle Technology* (pp. 71–99). Springer Tracts in Mechanical Engineering. Springer, Singapore. https://doi.org/10.1007/978-981-15-3328-0_4.
7. Shen, C., Wang, G., Wang, S., & Liu, G. (2011, August). The imbalance source of spindle-tool system and influence to machine vibration characteristics. In *2011 Second International Conference on Digital Manufacturing & Automation* (pp. 1288–1291). IEEE. https://doi.org/10.1109/ICDMA.2011.317
8. Danylchenko, Y. M., & Petryshyn, A. I. (2012). Modeling of oscillation forms of the mechanical oscillatory system “spindle unit-base”. *Reliability of Tools and Optimization of Technological Systems*, (30), 309–316.
9. Danylchenko, Y. M., & Petryshyn, A. I. (2011). Dynamic analysis of the mechanical oscillatory system “spindle unit-base”. *Reliability of Tools and Optimization of Technological Systems*, (28), 169–174.
10. Danylchenko, Y., Petryshyn, A., Repinskyi, S., Bandura, V., Kalimoldayev, M., Gromaszek, K., & Imanbek, B. (2021). Dynamic characteristics of “tool-workpiece” elastic system in the low stiffness parts milling process. In *Mechatronic Systems 2: Applications in Material Handling Processes and Robotics* (pp. 225–236). Routledge.
11. Danylchenko, Y. M., & Petryshyn, A. I. (2014). Identification of spindle vibrations based on measurement results of spindle unit body vibrations. *Bulletin of NTUU “KPI”, Series Machine Building*, (71), 147–152.
12. Danylchenko, Y., Storchak, M., Danylchenko, M., & Petryshyn, A. (2023). Cutting process consideration in dynamic models of machine tool spindle units. *Machines*, *11*(6), 582. https://doi.org/10.3390/machines11060582.
13. Sakhno, Y. Y., & Volyk, V. S. (2006). Mechanical processing of unbalanced parts on a lathe with hydrostatic supports. *Bulletin of Engine Building*, (2), 129–133.
14. Strutynskyi, V. B., & Fedorynenko, D. Y. (2011). *Statistical dynamics of spindle units on hydrostatic supports*: Monograph. Aspect-Polygraph Publishing House.
15. Shapoval, Y. V., & Kryvoruchko, D. V. (2014). Test bench for studying the turning process with high spindle rotation frequencies. *Journal of Engineering Sciences*, *1*(3), 11–18.
16. Drach, I. V., & Roizman, V. P. (2018). *Automatic balancing of rotating bodies by liquid*: Monograph. KhNU.
17. Roizman, V. P., & Drach, I. V. (2015). Theoretical study of the automatic balancing process of rotors with a vertical axis of rotation by liquid working bodies (cases of ideal and viscous liquids). *Vibrations in Engineering and Technologies*, *3*(79), 50–58.
18. Kalchenko, V. I., Sakhno, Y. Y., & Fedorynenko, D. Y. (1996). Ways to improve the process and devices for balancing rotors. *Bulletin of Chernihiv Technological Institute*, (1), 111–118.
19. Xu, C., Zhang, J., Yu, D., Wu, Z., & Feng, P. (2015). Dynamics prediction of spindle system using joint models of spindle tool holder and bearings. *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, *229*(17), 3084–3095. https://doi.org/10.1177/0954406215569588.
20. Rauscher, C. (2011). *Grundlagen der Spektrumanalyse* [Fundamentals of Spectral Analysis]. Rohde & Schwarz.
21. Orgiyan, A., Oborskyi, G., Balaniuk, A., Tonkonogyi, V., & Dasic, P. (2021). Development of calculation of statistical and dynamic errors upon fine boring with console boring bars. *Lecture Notes in Mechanical Engineering*, 588–597. https://doi.org/10.1007/978-3-030-68014-5_57.

Оргіян Андрій Олександрович; Andrii Orhiian, ORCID: https://orcid.org/0009-0002-5514-3239
Баланюк Ганна Василівна; Anna Balaniuk, ORCID: https://orcid.org/0000-0003-1628-0273
Оргіян Олександр Андрійович; Alexander Orgiyan, ORCID: https://orcid.org/0000-0002-1698-402X
Колеснік Василь Михайлович; Vasyl Kolesnik, ORCID: https://orcid.org/0000-0002-6573-2961
Прокопович Павло Ігорович; Pavlo Prokopovych, ORCID: https://orcid.org/0009-0003-8007-7680

Received February 06, 2025

Accepted March 17, 2025