MACHINE BUILDING

МАШИНОБУДУВАННЯ

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BALANCING SPINDLES WITH TOOLS FOR FINISHING AND BORING MACHINES

Г. Оборський, О. Оргіян, Г. Баланюк. **Балансування шпинделів з інструментами для чистових і розточувальних верстатів.** Оздоблювально-розточувальні верстати застосовують у серійному та масовому виробництві для обробки різноманітних за формою та деформативними властивостями деталей: гільз, шатунів, поршнів, фланців, муфт, а також корпусних деталей типу картерів, блоків, циліндрів та ін. На станині горизонтальних розточувальних верстатів встановлюють мости, на яких закріплюють шпиндельні вузли. Вирішальний вплив на точність обробки надають параметри шпиндельного вузла з інструментом та пристосування з деталлю. Застосування сучасних інструментальних матеріалів робить необхідним підвищення швидкохідності шпиндельних вузлів до 8...10 тис. об/хв під час обробки отворів діаметром 16...40 мм. У роботі досліджено анізотропію жорсткості пружної системи верстата, що призводить до відхилень від заданої глибини різання під дією сил, що обертаються разом з різцем. В експериментах вивчено вплив анізотропії радіальної податливості на похибки обробки. Отримано залежність визначення додаткової відцентрової сили, що забезпечує мінімум відхилень від круглості. Для забезпечення мінімуму відхилення від круглості розроблена методика дозволяє виконати всього три розточування. Зазначимо, що виконання вимог до форми поперечного перерізу за відсутності дисбалансу встановлюється після першого розточування. За розробленою методикою призначаються значення відцентрових сил, що задаються масою вантажу, закріпленого на фланці борштанги чи шківу. Виміряні значення відхилень від круглості та шорсткості поверхонь оброблених отворів підтверджують розрахункові оцінки. Встановлено, що лише для головок першого типорозміру (УАР-1) максимальна неврівноваженість близька до порядку величини до допустимої, тому балансування шпинделя слід проводити для головок, призначених для роботи на частотах обертання, близьких до максимальних.

Ключові слова: відцентрова сила, дисбаланс, шпиндель-борштанга, податливість, відхилення від круглості, шорсткість, деталь-пристосування, анізотропія

H. Oborskyi, A. Orgiyan, A. Balaniuk. **Balancing spindles with tools for finishing and boring machines.** Finishing and boring machines are used in serial and mass production for processing parts of various shapes and deformation properties, namely liners, connecting rods, pistons, flanges, couplings, as well as body parts such as crankcases, blocks, cylinders, etc. Bridges are installed on the bed of horizontal boring machines, on which spindle units are fixed. The decisive influence on the accuracy of processing is exerted by the parameters of the spindle assembly with the tool and the fixture with the part. The use of modern tool materials makes it necessary to increase the speed of spindle units up to 8...10 thousand rpm when machining holes with a diameter of 16...40 mm. The paper investigates the anisotropy of the rigidity of the elastic system of the machine tool, which leads to deviations from the specified depth of cut under the action of forces rotating together with the cutter. In experiments, the influence of the anisotropy of radial compliance on the processing errors was studied. The dependence of the determination of additional centrifugal force, which provides a minimum of deviations from roundness, is obtained. To ensure a minimum deviation from roundness, the developed technique allows you to perform only three borings. Note that the fulfillment of the requirements for the shape of the cross section in the absence of imbalance is established after the first boring. According to the developed method, the values of centrifugal forces are assigned, given by the mass of the load fixed on the flange of the boring bar or on the pulley. The measured values of deviations from the roundness and roughness of the surfaces of the machined holes confirm the calculated estimates. It has been established that only for heads of the first size (universal diamond-boring UDB-1) the maximum unbalance is close in order of magnitude to the allowable one. Therefore, spindle balancing should be carried out for heads designed to operate at speeds close to the maximum.

Keywords: centrifugal force, unbalance, spindle-boring bar, compliance, roundness deviation, roughness, fixture detail, anisotropy

Introduction

Finishing and boring machines (FBM) are high-performance semi-automatic machines designed for fine boring of holes, turning of external surfaces, turning grooves and cutting ends. These machines provide high accuracy of shape and location of machined surfaces.

The imbalance of the spindles during rotation leads to variable loads on the supports. These loads cause vibrations and lead to loss of accuracy of workpieces, premature tool wear, etc. Unbalance is especially undesirable in high-speed machines and mechanisms. It is known that the dynamic loads caused by the imbalance of a rotating rotor can many times exceed its gravity. It is known that dynamic loads can also occur when the center of gravity lies on the axis of rotation. The reasons for the im-

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balance can be inaccuracies in the manufacture and assembly of parts, the anisotropy of the rigidity of the spindles, etc. In mechanical engineering, static and dynamic balancing is carried out on special balancing machines.

Analysis of recent publications and problem statement

Disturbing forces act on the spindle of the boring machine, causing forced vibrations. These forces include the centrifugal forces of unbalanced rotating masses. Any real shaft, due to the inevitable manufacturing inaccuracies and inhomogeneities in the shaft material, has some imbalance [1]. In shafts rotating at a critical speed, forced oscillations of large amplitude occur [2, 3]. Two rotating forces act on the spindle: the component P_{yz} of the cutting force and the centrifugal force F_c. A non-rotating shear force from a belt drive is applied to the spindle pulley. Its influence on the distortion of the cross-sectional shape is significantly less than the influence of rotating forces [4].

It should be noted that a number of studies have studied the change in the vibration signal during idling due to the influence of the dynamic characteristics of the system during the excitation of internal parameters (changes in bearing tightness, violations of cooling conditions, inertial forces, etc.). The main factors characterizing changes in the dynamic quality of spindle assemblies are determined, namely: problems with the shaft (imbalance), misalignment with the tool, etc.); problems with bearings, non-linearity of bearings, etc. [5, 6]. The development of dynamic models of spindle assemblies of different configurations and algorithms for calculating their main dynamic characteristics (frequencies and forms of natural vibrations), as well as dynamic compliance, are described in [7, 8].

It has been determined that the processing accuracy is mainly determined by the dynamic state of the spindle assembly during cutting shaping. The imbalance of the spindle-tool-workpiece system increases significantly at high spindle speeds. In this case, the unbalance force can often exceed the cutting force and significantly affect the dynamic state of the workpiece [3, 4].

New results of automatic balancing of rotors with a balancing medium presented in [9, 10]. The rotors have cylindrical chambers partially filled with the working medium; at certain frequencies, the vibration resistance of rotating machine elements increases. The tasks of static balancing of spindle assemblies with an unbalanced workpiece are considered using the method of random search for automatic balancing by moving corrective masses on the spindle from the hydrostatic support [11, 12, 13].

High-speed spindle systems can generate considerable heat during operation and cause thermal distortion that affects the accuracy of the spindle. Heat dissipation is a common and effective method of removing generated heat. The temperature distribution of models with and without heat-conducting paths has been scientifically and experimentally studied [14]. Due to various errors in the process of special design, manufacturing and assembly, there is a varying degree of unbalanced residual content in the machine tool spindle system. Some types of sources of imbalance of the spindle-tool system, as well as the degree of quality of its dynamic balancing, were studied [15]. The design of the spindle system requires the modeling of dynamic characteristics taking into account the characteristics of the connection. To predict the dynamic behavior, a method of modeling a spindle system using a spindle tool holder and support joints was presented. Tymoshenko beams were used to describe the components of the spindle system. The presented modeling method is useful for evaluating the performance of the spindle system [16]. The work [17] developed a dynamic model of the interaction of the processing technological system with the cutting process. The model is presented in the form of a dynamic model of the elastic system “spindle assembly-workpiece/tool-cutting process-tool/workpiece”. The proposed approach is used to calculate the natural oscillation frequencies of the spindle with the part fixed in the chuck and supported on the tool.

Purpose and objectives of the study

Analyzing the results of studies on determining the dynamic imbalance of spindle assemblies it should be noted that many publications reflect the methods of balancing, taking into account the parameters of the elastic system spindle-tool. We also note the clearly insufficient number of publications on this issue for high-precision finishing and boring machines. Spindle imbalance in these machines is one of the sources of oscillations that generate errors in the shape of the machined surface.

Summarizing the results of studies of the dynamic quality of spindle assemblies with boring bars, as well as the part-fixture system, the purpose and objectives of this work are formulated.

The purpose of the work is to develop a methodology for determining the imbalance of the spindle with the tool, taking into account the accuracy of the machined surface, as well as the permissible values of the imbalance for different spindle heads.

Objectives of the study:

1. Determine the dependence of the amplitudes of the harmonics of the oscillations of the spindles on the magnitude of the centrifugal force F_c;

2. To study the influence of the anisotropy of radial compliance on the processing error;

3. Determine the magnitude and direction of the centrifugal force based on the results of successive borings, taking into account deviations from the roundness and roughness of the machined surface.

Main part

The influence of the rotating transverse force $\bar{Q} = \bar{P} + F_c$ on tool oscillations and deviations from roundness was investigated in experiments with idle rotation of the FBM spindle and at parting.

An idea of the effect of unbalance on the distortion of the cross-sectional shape of the machined hole is given by the round diagrams obtained by boring samples from steel 20X at $t = 0.15$ mm, $S = 0.03$ mm/rev and $V = 470$ m/min. With coinciding directions of action of the forces F_c and P_y, the oval is oriented in the same way as the elastic system stiffness diagram and with opposite directions of these forces, the orientation and magnitude of the ovality change.

In the experiments, the mass of unbalanced weights, their position, and the spindle speed were varied. Vibration spectra of the tool for values $F_c = 0 \ldots 200$ N were obtained experimentally using a vibrometer [18] on the stand. At the stand, we checked the indicators of dynamic quality when testing machines. The main criterion for dynamic quality indicators is the accuracy of processing, which is most affected by the influence of the variability of system parameters. Determination of changes in process parameters requires an increase in measurement accuracy. With the modulation of oscillations characteristic of non-stationary systems, side harmonics close in frequency appear in their spectra [19]. Therefore, during the experiments, spectrum analyzers were used. To determine the variability of the rigidity of the spindle assembly with a mandrel, a device was used, fixed on the bridge of the machine.

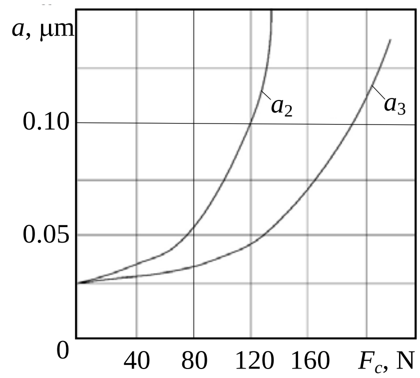

Fig. 1. Dependence of the amplitudes of harmonics of oscillations of the spindle of the UDB2 head on the centrifugal force applied at a distance of 65 mm from the spindle flange

With an increase in F_c, only the amplitudes of the a_2 and a_3 harmonics increase, corresponding to the doubled and tripled rotation frequency (Fig. 1). Dependences a_2 and a_3 on F_c are nonlinear and can differ significantly for individual instances of boring heads of the same size. The intensity of the increase in the amplitudes of harmonics increases when the application point F_c moves along the boring bar from the flange to the cutter. In accordance with the relationship $a_2 > a_3$, low frequency distortions of the cross-sectional shape of bored holes appear mainly in the form of ovality.

Calculations and experiments have established that the sources of anisotropy in the compliance of the FBM tool subsystem are the rolling bearings and their connection with the head body. The contribution of bridge deformations to the anisotropy of compliance is 2...5 %, and the deformation of the head-bridge joint is 1…2 %. The anisotropy of the compliance of the machine subsystem is also small, but can significantly increase depending on the shape and size of the part, on the conditions of its basing on the fixture.

Let us consider the influence of the anisotropy of radial compliance on the machining error. Assuming that the compliance diagrams of the tool and workpiece subsystems are elliptical, we represent the radial compliance of each subsystem as a function of the spindle angle ψ:

$$K_i(\psi) = K_i + \frac{1}{2}\Delta K_i \sin 2\psi,$$
$$K_g(\psi) = K_g + \frac{1}{2}\Delta K_g \sin 2(\psi + \psi_0), \tag{1}$$

where K_i, K_g are the average values of subsystem compliance, ΔK_i, ΔK_g are the differences of maximum compliance values, φ_0 is the angle between the directions in which the subsystem compliance is maximum. The tool subsystem is loaded in the direction of the normal to the machined surface by force $P_y + F_c\cos\alpha$ (α is the angle between the directions P_y and F_c), and the part subsystem – by the

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force P_y. Therefore, the cross-sectional radius of the machined hole is determined by the expression $R(\psi) = R_0 + \left[K_i + \frac{1}{2}\Delta K_i \sin 2\psi \right](P_y + F_c \cos\alpha) + \left[K_g + \frac{1}{2}\Delta K_g \sin 2(\psi + \psi_0) \right] P_y$, from which you can find the deviation from roundness ΔR_o:

$$\Delta R_0 = \sqrt{\left[(P_y + F_c \cos\alpha)\Delta K_И + P_y K_g \cos 2\psi_0 \right]^2 + (P_y \Delta K_g \sin 2\psi_0)^2}. \quad (2)$$

The resulting dependence allows, during the debugging of the machine, to minimize deviations from roundness by selecting a suitable centrifugal force. Perform no more than three borings in the order indicated in Table 1.

The magnitude and direction of the centrifugal force are assigned according to the results of successive borings, for the description of which the following designations are accepted:

$\Delta R_{(i)}$ – roundness deviation after *i*-th boring,

$[\Delta R]$ – allowable deviation value,

ψ_p – the angle between the major axes of the ovals after the first and subsequent boring.

Compliance with the requirements for the shape of the cross section in the absence of imbalance (Scheme 1, *a*) is established by the results of the first boring, if both subsystems are isotropic ($\Delta K_i = \Delta K_g = 0$), and if $\psi_p = 90°$ at $\Delta K_i = \Delta K_g$. Having found the excess of permissible deviations from roundness (1, *b*), the second boring is carried out, setting $F_c = -P_y$. If the anisotropy of the tool subsystem is large ($\Delta K_i >> \Delta K_g$), then the shape of the hole is corrected during the second boring (2, *a*).

If $\Delta K_i = \Delta K_g$ and $\psi_p = 45°$, then the minimum deviation from roundness is achieved at the set value $F_c = -P_y$ (2, *b*), and further correction of the shape is impossible. If the anisotropy of the subsystems is the same ($\Delta K_i = \Delta K_g$ и $\psi_p = 0°$), then the shape correction is achieved at $F_c = -2P_y$ (2, *c*). In other cases, corresponding to a large anisotropy of the subsystem of the part ($\Delta K_g >> \Delta K_i$), to assign F_c, the third boring should be performed.

Small changes in hole ovality after the second boring compared to the first one (2, *d* and 2, *e*) may occur at $\psi_p = 45°$, when after the third boring the ovality changes slightly (3, *a* and 3, *b*), as well as at $\psi_p = 0°$ (3, *e*), or $\psi_p = 90°$ (3, *d*) when the third boring fully corrects the cross-sectional shape.

Note: thin lines in diagrams (2, *a*) – (3, *b*) show the contours of the hole after the first boring:] – the selection of F_c is completed.

When correcting the cross-sectional shape by creating an imbalance on the boring bar when boring samples of bronze BrOCS 6-6-3, the imbalance me=140 g.cm at a spindle speed of 1800 rpm creates a centrifugal force $F_c = 50$ N. The action of this force reduces the deviations from roundness by about 4 times.

Relation (2) is also used to determine the allowable values of the imbalance of the spindle with the tool according to the standard values of the allowable deviations from roundness ($[\Delta R] = 1.2$ µm for UDB1, 1.6 µm for UDB2 and 2 µm for UDB3).

In calculating the allowable value $[F_c]_p$ of the centrifugal force applied near the cutter, one should enter the value $K_R[\Delta R]$, where $K_R = \sqrt{1 - (\Delta R / [\Delta R])^2}$ coefficient that takes into account deviations from roundness ΔR_1, caused by the kinematic error of the machine tool and high-frequency oscillations of the boring bar. The permissible value of the centrifugal force is determined by the expression:

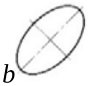
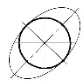
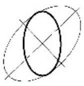
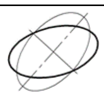
$$[F_c]_p = \frac{P_y}{\cos\alpha} \left\{ \sqrt{\frac{K_R^2 [\Delta R]^2}{P_y^2 (\Delta K_i)^2} - \left(\frac{\Delta K_g}{\Delta K_i} \right)^2 \sin^2 2\psi_0} - \frac{\Delta K_g}{\Delta K_i} \cos 2\psi_0 - 1 \right\}, \quad (3)$$

whence at $\alpha = \psi_0 = 0$ we find the minimum value $[F_c]_{p,\min} = K_R [\Delta R] / P_y \Delta K_i - \Delta K_g / \Delta K_i - 1$.

The value of $[F_c]_p$ depends on the length l_1 of the boring bar, because with a change in the distance to the spindle flange, the characteristic of the anisotropy of compliance (Fig. 2) also changes. Calculation of elastic displacements for finishing boring heads UDB1, UDB2 and UDB3 leads to the ratio $[F_c]_f = (1+0.003\, l_1)\, [F_c]_p$, which determines the allowable value of the centrifugal force applied to the flange of the boring bar.

Table 1

Selection of the centrifugal force value based on the results of boring

Boring number	The magnitude of the centrifugal force when boring	Hole shape after boring: scheme	Hole shape after boring: $\Delta R_{(i)}$	Hole shape after boring: Ψ_p	Appointment of centrifugal force based on the results of boring
1	$F_c = 0$	*a*	$\Delta R_{(1)} \prec [\Delta R]$	–	$F_c = 0]$
		 *b*	$\Delta R_{(1)} \succ [\Delta R]$	–	$F_c = -P_y$
2	$F_c = -P_y$	 *a*	$\Delta R_{(2)} \prec [\Delta R]$	–	$F_c = -P_y]$
		 *b*	$\Delta R_{(2)} \approx 0.7\Delta R_{(1)}$	15…30°	$F_c = -P_y]$
		 *c*	$\Delta R_{(2)} \approx 0.5\Delta R_{(1)}$	~0°	$F_c = -2P_y]$
		 *d*	$\Delta R_{(2)} \approx 0.9\Delta R_{(1)}$	~0°	$F_c = -(4...5)P_y$
		 *e*	$\Delta R_{(2)} \approx 1.1\Delta R_{(1)}$	~0°	$F_c = (4...5)P_y$
3	$F_c = -(4…5)P_y$	 *a*	$\Delta R_{(3)} \approx \Delta R_{(2)}$	15…30°	$F_c = -P_y]$
	$F_c = (4…5)P_y$	 *b*	$\Delta R_{(3)} \approx \Delta R_{(1)}$	15…30°	$F_c = P_y]$
	$F_c = -(4…5)P_y$	 *c*	$\Delta R_{(3)} \prec [\Delta R]$	–	$F_c = -(4...5)P_y]$
	$F_c = (4…5)P_y$	 *d*	$\Delta R_{(3)} \prec [\Delta R]$	–	$F_c = (4...5)P_y$

Discussion of results

The calculated values $[F_c]_f$ (Fig. 3) are compared with the results of tests by cutting heads of three sizes (Table 2).

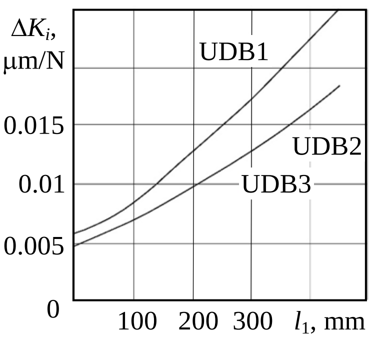


Fig. 2. Dependence of the anisotropy of the radial compliance of the tool subsystem on overhang

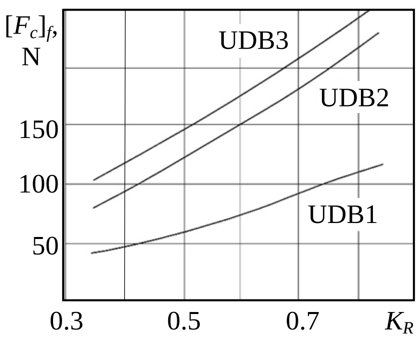


Fig. 3. Calculated permissible values of centrifugal force on the flange

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Table 2

Influence of unbalance on boring accuracy

Head size	F_c, N		ΔR, µm	R_a, µm
	On the flange	On the pulley		
UDB1	0	–	0.5	0.15
	80	–	0.5	0.14
	120	–	0.7	0.23
	250	–	1.8	0.22
	-	120	0.6	0.17
	-	180	1.0	0.16
	-	250	1.8	0.19
UDB2	0	–	0.7	0.22
	80	–	0.8	0.22
	200	-	1.4	0.24
UDB3	0	–	0.6	0.23
	120	–	0.9	0.24
	250	–	1.8	0.23

The boring of samples from steel 20X was carried out at $t = 0.1$ mm, $S = 0.03$mm/rev, $V = 220$m/min. The value of F_c was set by the mass of the load fixed on the flange of the boring bar or on the pulley. The measured values of deviations from roundness and surface roughness of the machined holes confirm the calculated estimates.

Mass measurements of the accuracy of spindle assemblies show that under constant machining conditions, the level of vibrations and the associated waviness of the cross section of the machined surface can change quite significantly. In the experiments described above, a small level of fluctuations ($\alpha \approx 0.1$ µm) was provided, which corresponds to the calculated value of $K_R = 0.8$. At $\alpha = 0.5 \ldots 0.8$µm, we obtain $K_R = 0.5$ and correspondingly lower values of $[F_c]_f$. It was found that the influence on the processing errors of centrifugal forces applied on the pulley and on the flange is approximately the same. Similar results were obtained in the processing of bronze.

Permissible unbalance values $[me]$ at $K_R = 0.5$ are compared in Table 3 with the results of measurements of the unbalance of spindle assemblies performed on a machine for dynamic balancing of modes 9.01-30. Only for heads of the first standard size the maximum unbalance is close in order of magnitude to the permissible one. Therefore, preliminary balancing of the spindle without a tool should be carried out only for UDB1 heads designed to operate at speeds close to the maximum.

Table 3

Comparison of maximum and permissible imbalance values

Head size	Number of heads tested	n, rpm	$me_{\max}/[me]$	
			On the flange	On a pulley
UDB1	22	4000	0.7	0.6
UDB2	26	2500	0.2	0.1
UDB3	12	1500	0.07	0.05

Since the unbalance of the spindles does not exceed the allowable values, the choice of the magnitude and direction of the centrifugal force should be performed in accordance with Table 1, depending on the parameters of the spindle with the tool and the fixture with the workpiece.

Conclusion

1. A technique has been developed for determining the magnitude of the imbalance of the spindles of finishing and boring machines and compared with its permissible value for the line of spindle heads.

2. Based on the results of only three consecutive borings, this technique allows, when debugging the machine, to minimize deviations from roundness by selecting the appropriate centrifugal force.

3. The results of the first boring determine the shape of the cross section when both subsystems of the spindle-boring bar and the part-fixture are isotropic. When the specified deviations from roundness are exceeded, a second boring is performed, setting $F_c = P_y$. At the same time, the cross-sectional

shape is corrected if the anisotropy of the tool system is greater than the anisotropy of the part subsystem; if the anisotropy of the subsystems is the same, then the correction of the form is achieved at $F_c = -2P_y$. If the anisotropy of the part subsystem is greater than that of the tool, the third boring is performed for assignment F_c (Table 1).

4. In the calculation of the allowable value $[F_c]_p$ of the centrifugal force applied near the cutter, a coefficient should be introduced that takes into account deviations from roundness caused by the kinematic error of the machine tool and high-frequency oscillations of the boring bar.

5. The calculation of elastic displacements for different spindle heads leads to the relation $[F_c]_f = (1 + 0.003l_1)[F_c]_p$, which determines the allowable value of the centrifugal force applied to the flange of the headstock.

6. Preliminary balancing of the spindle without a tool should be carried out only for heads operating at close to maximum speeds (Table 3).

7. The parameters of the spindle with the tool and the fixture with the part completely determine the magnitude and direction of the imbalance.

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