MACHINE BUILDING

МАШИНОБУДУВАННЯ

UDC 621.941

A. Balaniuk[1]**,** PhD, Assoc. Prof.,
A. Orgiyan[1]**,** DSc., Prof.,
H. Oborskyi[1]**,** DSc., Prof.,
V. Kolesnik[1]**,** PhD, Assoc. Prof.,
R. Matzey[2]**,** PhD, Assoc. Prof.
[1] National Polytechnic Odessa University, 1 Shevchenko Ave., Odesa, Ukraine, 65044; e-mail: balanuk.a.v@op.edu.ua
[2] Military Academy of Odessa, 10 Fountain road Str., Odesa, Ukraine, 65009

MEANS OF INCREASING THE DYNAMIC QUALITY OF FINISHING AND BORING MACHINES

Г. Баланюк, О. Оргіян, Г. Оборський, В. Колеснік, Р. Мацей. **Засоби підвищення динамічної якості оздоблювально-розточувальних верстатів.** У статті наведено конструктивні особливості ряду шпиндельних головок, що забезпечують високу точність прецизійних оздоблювально-розточувальних верстатів. Описано характеристики головок, пристрої шпиндельних опор, що підвищують загальну жорсткість та демпфування системи підшипників. Наведено основні конструкції шпиндельних головок, співвідношення конструктивних параметрів, а також аналіз їх конструкцій. На основі обробки осцилограм загасаючих коливань вивчено залежність логарифмічного декременту коливань від їхнього рівня. На основі експериментального дослідження характеристик оздоблювально-розточувальних головок визначено оптимальні значення попереднього осьового натягу підшипників. Встановлено, що із зростанням зусилля натягу декремент коливань змінюється немонотонно та має максимум. Така залежність визначається виникненням у динамічній системі шпиндельного вузла сил лінійного опору та сухого тертя. Оптимальні значення натягу забезпечують максимальну точність шпиндельного вузла, а збільшення температури не перевищує 4…6°. Встановлено, що створення порожнини на вільному кінці консолі підвищує динамічну якість шпиндельних вузлів із консольним інструментом. При цьому зменшуються коефіцієнти динамічних збурень у системі шпиндель-борштангу. У роботі визначено вплив довжини порожнини на коефіцієнт передачі обурень від передньої опори до різця.

Ключові слова: шпиндельна головка, підшипникова опора, декремент коливань, жорсткість, статичні та гармонічні коефіцієнти впливу, зусилля натягу

A. Balaniuk, A. Orgiyan, H. Oborskyi, V. Kolesnik, R. Matzey. **Means for increasing the dynamic quality of finishing and boring machines.** The article presents the design features of a number of spindle heads that ensure high accuracy of precision finishing and boring machines. The article describes the characteristics of the heads, spindle support devices, which increase the overall rigidity and damping of the spindle-bearing system. The basic designs of spindle heads, the relationships between design parameters are presented, and an analysis of their designs is also performed. Based on the processing of oscillograms of damped oscillations, the dependence of the logarithmic decrement of oscillations on their level was studied. Based on an experimental study of the characteristics of finishing and boring heads, the optimal values of the axial preload of the bearings were determined. It has been established that with increasing tension force, the oscillation decrement changes non-monotonically and has a maximum. This dependence is determined by the occurrence of linear resistance and dry friction forces in the dynamic system of the spindle assembly. Optimal tension values ensure maximum accuracy of the spindle assembly, and the temperature increase does not exceed 4…6°. We have found that creating a cavity at the free end of the console improves the dynamic quality of spindle assemblies with console tools. At the same time, the coefficients of dynamic disturbances in the spindle-boring bar system decrease. The work determines the influence of the cavity length on the coefficient of transfer of compensation from the front support to the cutter.

Keywords: spindle head, bearing support, vibration decrement, rigidity, static and harmonic influence coefficients, tension forces

Introduction

Reducing forced vibrations of machine tools, ensuring an increase in their dynamic quality, is based on research in two directions, namely, eliminating or weakening sources of mutual influences and increasing the stability coefficients of elastic systems. Balancing the rotors, increasing the accuracy of bearings and gears, and using vibration isolation products weaken external influences. Increasing the vibration resistance of a closed dynamic system of a machine tool can be achieved by increasing the rigidity and damping capacity of the elastic system, changing processing modes, choosing a tool material that creates favorable conditions for chip formation, and choosing a rational mutual orientation of the axes of rigidity and cutting force.

Analysis of last publications and problem statement

Works [1 – 4] explore ways to reduce vibrations using vibration dampers and devices for crushing drain chips (and articles on vibration dampers).

DOI: 10.15276/opu.1.69.2024.01

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

The works [5, 6, 7] provide solutions to the problem of modeling spindle units operating on rolling bearings. A method for complex analysis of their nonlinear tribomechanical properties at the development stage has been developed. The technique uses analytical models of beams, as well as elastic-strain, analytical and thermal models. A thermal model of high-speed spindle units for lathes and grinding machines has been developed. The model takes into account temperature estimates, thermal deformations, heat transfer from bearings, etc. [8, 9, 10].

Research [11] shows that the development of automated production is based on the criteria of physical systems that control the physical production process based on the results of virtual modeling using digital twins. The spindle units of such machines are called “intelligent spindles”. Their designs include tool condition, vibration, spindle damage, spindle balancing and durability. A dynamic model of a lathe, closed to the cutting process, is proposed. It should be noted that using the method of transition matrices with an analytical description of the deformation in the cutter-workpiece contact, the natural frequencies of the spindle were determined. This formulation of problems leads to clarification of the sources of vibrations of the closed system of the machine tool. Dynamic modeling of rolling bearings during high-speed milling was studied in [12, 13]. To describe dynamic interactions, a finite element model of the spindle assembly is used using the Timoshenko beam theory [12]. The vibration modes of the machine spindle assembly have been studied in publications [14, 15, 16]. It has been established that the highest level of vibrations of the spindle and the spindle unit body is achieved at the natural frequencies of the spindle and the combined frequencies of the “spindle-spindle unit body” system. A method for identifying spindle vibrations based on the results of measuring vibrations of the spindle assembly housing is substantiated. The calculated functions are the dynamic compliances of the partial subsystems of the spindle assembly.

Computational and experimental results of studies of the kinematics and rigidity of angular contact ball bearings of high-speed spindle units are presented in [17, 18]. The elastic-strain characteristics of a spindle bearing at high rotation frequencies have been studied. It has been established that the characteristics of a ball bearing are influenced by the following factors: rotation speed, axial interference, ball material and ring temperature. An elastic-dynamic model of angular contact ball bearings has been developed.

A resonance method has been developed for diagnosing manufacturing and assembly defects in spindle units on rolling bearings that cause low-frequency vibrations. The dynamic behavior of a spindle assembly mounted on an elastic frame mounted on a massive base has been studied. The amplitudes of spindle oscillations during frame resonance at characteristic frequencies of the effects of manufacturing and assembling the spindle assembly of the grinding head are determined. Spectra of vibrations of the mandrel relative to the spindle sleeve are determined, which are used to calculate the accuracy indicators of spindle units.

Based on the above review of common spindle head designs, as well as a study of their dynamic quality, the purpose and objectives of the research are formulated in the work.

The purpose of work is to improve the dynamic quality of precision spindle heads.

To achieve this goal, the following tasks have been set:

1. Increase the vibration resistance of the spindle assembly of finishing and boring machines by increasing its damping capabilities.
2. Determine the optimal values of the axial tension force of the spindle rolling bearings.
3. Develop a calculation scheme to determine the coefficients of transmission of disturbances from the spindle supports to the tool.
4. Increase the dynamic quality of the spindle head when designing a cavity at the free end of the console.

Statement of the main material

The spindle head is the most critical component of a finishing and boring machine. The characteristics of the head determine the characteristics of the machine as a whole. The accuracy of the head directly affects the positioning accuracy, shape and surface roughness.

Leading enterprises and firms produce a range of normalized heads, on the basis of which special modifications and multi-spindle configurations are created.

The characteristics of the heads are:

1. Limits on the diameters of bored holes.
2. The highest number of spindle revolutions.

3. Overall and connecting dimensions: distance from the spindle to the base of the head; width and length of the head housing; flange diameter and spindle length, etc.

Heads of 4 standard sizes are assembled on angular contact ball bearings at the Pivden-Verstatmash subsidiary of VAT OZRSV.

Normalized heads of small standard sizes are produced by Heald, Gamba Fiorito, etc. These companies produce heads on ball and roller bearings in one standard size (Excello, Heald, Toyoda, etc.). The size of the head depends mainly on the diameter of the spindle.

In order to increase the vibration resistance of diamond boring heads, a number of spindle diameters of the heads of the AP16 – AP46 range (40, 55, 75, 120 mm) in the new range (УАР1П – УАР4П) are shifted upward to 55, 75, 100, 130 mm – without changing the speed characteristics: 5000, 3150, 2000, 1250 min^{-1}.

The Toyoda company (Japan) produces heads of 4 standard sizes with spindle diameters of 45, 55, 65, 85 mm and a maximum rotation speed of 3600…2000 min^{-1} (on ball bearings). The head of the 2nd standard size from Heald with bearings of type 36212 and 3621 in the front support is designed for a maximum rotation speed of 4000 min^{-1} and a diameter range of bored holes of 25…63 mm.

The limits of the diameters of boring holes are a conditional indicator, specified depending on the specific processing conditions: length of the boring bar, material being processed, allowance, requirements for processing quality, etc. There are no specific rules for choosing the spindle size depending on the processing diameter.

As can be seen from Table 1, for heads of similar sizes, the maximum diameters of bored holes vary within a very wide range.

Table 1

Technical characteristics of diamond boring heads

Factory, company	VAT OZRSV			Heald	Toyoda		De Valliere	Excello
Country	Ukraine			USA	Japan		France	USA
Head model	УАР16	УАР1П	УАР26	111	40G	50G	Machine AL-4	DB22
Spindle diameter, mm	40	55		50	45	55	40	41
Limits of diameters of bored holes, mm	8…32		20…65	25…63	10…100	15…150	6…150	9.5…152

The maximum rotation speed of the smallest diamond boring heads on angular contact ball bearings does not exceed 8000 min^{-1}. Heads on fluid friction supports have higher speed characteristics. The rotation speed of heads of the “TC” series design of accuracy class “A” on hydrostatic bearings reaches more than 8000 min^{-1}.

In diamond boring machines, spindle units on rolling bearings are most common. At the same time, in precision machine tool construction, spindle units on fluid friction supports, designed to perform particularly precise operations, most often with diamond tools, are more widely used.

The following types of bearings are used in the spindle units of diamond boring machines:

1. Ball bearings: radial; radial thrust; thrust-radial; special.

2. Roller bearings: double-row radial bearings with cylindrical rollers and a tapered bore; angular contact conical type; needle-shaped.

3. Fluid friction: hydrostatic, hydrodynamic.

In diamond boring heads on rolling bearings, the following basic design designs of supports are used:

1. Angular contact ball bearings (single row, double, special) with preload in both supports and axial fixation of the spindle mainly in the front support. When doubling angular contact bearings, a pinching moment occurs that prevents the spindle from bending. Installing two bearings in the front support improves the dynamic properties of the spindle assembly due to increased overall rigidity and

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

damping of the spindle-bearing system. Methods for installing double bearings: “duplex – O”, “duplex – X”, “tandem”.

2. A double-row radial roller bearing in the front support in combination with a pair of radial or angular contact bearings in the rear support, where the spindle is axially secured.

3. Radial contact tapered roller bearings (single or double) in the front and rear supports.

Most domestic unified and special heads are made according to one standard design. In the one shown in Fig. 1 diamond boring head of the subsidiary enterprise of VAT “OZRSV” “Pivden-Verstatmash” angular contact ball bearings are installed according to the “duplex – O” scheme.

The axial preload force is created by the spacer ring. The front and rear support bearings are tightened onto the spindle with a nut through a spacer sleeve. Axial fixation of the spindle is carried out in the front support by clamping the outer rings of the bearings in the housing between the thrust bushing and the cover. A through hole in the housing for bearings allows you to bore the housing in one installation and ensure high alignment accuracy. In some designs, the bearings are moved apart to accommodate springs to create an axial preload force.

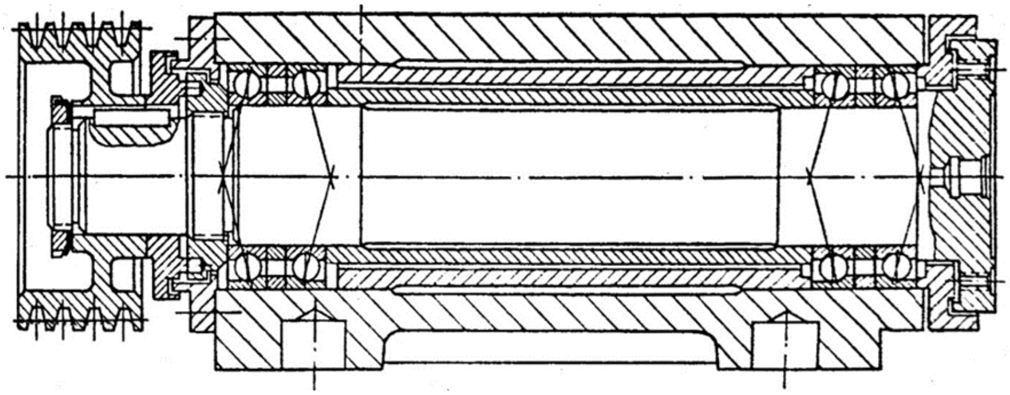

Fig. 1. Diamond boring head of finishing boring machine

Figure 2 shows a diamond boring head from Toyoda (Japan). The preload is created by a set of coil springs.

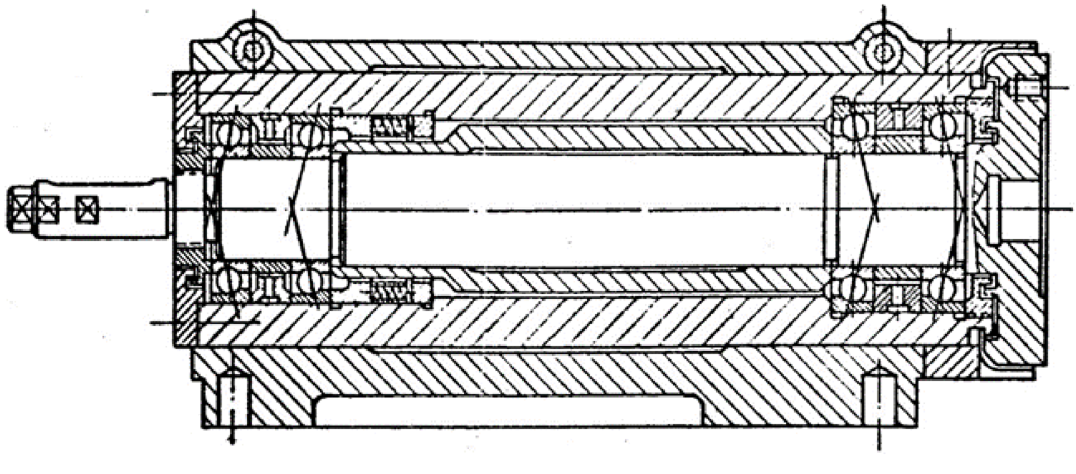

Fig. 2. Diamond boring head from Toyoda (Japan)

With rigid duplexing, the linear dimensions of the bushings must be maintained with an accuracy of 1…2 microns, while ensuring axial rigidity of the assembly in both directions. The distance between the bearings in this case can be calculated in such a way that the magnitude of the axial preload force remains constant as the temperature changes.

According to the spindle design, diamond boring heads can be divided into two groups (Fig. 3):

1) with a flange and a spacer between the bearings of the front and rear supports;

2) without a flange, with an increased diameter of the spindle in the span (similar to the spindles of internal grinding machines).

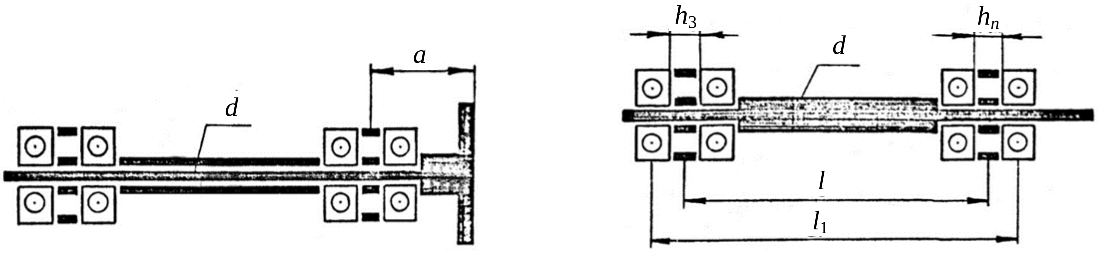

Fig. 3. Design diagrams of diamond boring heads: *a* – with a flange and a spacer sleeve; *b* – without flange

According to the number of bearings in the head: two-, three- and four-bearing.

Main design parameters of the heads:

d – diameter of the spindle in the span between the supports;

d_1 – inner diameter of the first bearing of the front support;

l – distance between the centers of the supports;

l_1' – distance between the centers of the outer bearings;

h_n, h_3 is the distance between the ends of the double bearings in the front and rear supports;

a_1 is the length of the console, i.e. distance from the middle of the front support to the base end of the spindle flange;

$\frac{l}{d}, \frac{l_1'}{d}, \frac{a_1}{d}, \frac{h_n}{d}, \frac{h_3}{d}, \frac{l}{a_1}$ – constructive relations.

The main relationships identified during the analysis of the designs of serial diamond boring heads are presented in Table 2 and Table 3.

Table 2

Relationships between design parameters for two-, three- and four-support spindle units

d_1, mm	d, mm	l/d	l_1'/d	a_1/d	h_n/d; h_3/d	l/a_1
25…40	38…45	3…8	4…9	1.4…2	0.12…0.25	2…6
40…65	40…65	4.5…7	5…8	1.2…1.5	0.12…0.18	3.5…5
70…100	70…100	3…5.5	3.5…6	0.6…1.2	0.1…0.12	3.5…8
Up to 200		2…3	2.5…3.5	0.7	0.1	3…4.5

Table 3

Relationships between design parameters for four-bearing spindle assemblies with widely spaced bearings

d_1, mm	d, mm	l/d	l_1'/d	a_1/d	h_n/d; h_3/d	l/a_1	d_1, mm
35…50	25…35	9…13	1…17	3…4	2…3	1.5…2	3…3.5
60…70	50…70	4…5	5…7	1.6…2	0.6…1.8	1.1…1.7	2.5…4

By processing oscillograms of damped free vibrations in the elastic system of a machine tool, we can distinguish three main types of dependence of the logarithmic decrement of vibrations on their level:

1. δ decreases monotonically as the oscillations decay;
2. δ, as the oscillations decay, first increases and then decreases;
3. δ does not change (inelastic resistance forces are linear). The dependence of δ on A (oscillation amplitude) is one of the signs of nonlinearity of the elastic system of the machine tool. In the case of nonlinearity of the first type, the monotonic function δ(A) can be approximated by the quadratic dependence:

$$\delta = \pi\left(æ + \frac{3}{4}æ_1 A^2\right), \quad (1)$$

which corresponds to the equation of free vibrations of an elastic system of the form:

$$\ddot{v} + æ\dot{v} + æ_1\dot{v}^3 + v = 0\,. \quad (2)$$

Parameters $æ$ и $æ_1$ are determined according to experimental data (Fig. 4):

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

$$æ = \frac{1}{\pi}\lim_{A\to 0}\delta,\ \ æ_1 = \frac{2}{3\pi}\lim_{A\to 0}\frac{d^2\delta}{dA^2}.$$

Nonlinearity of the second type can arise, for example, during the operation of tightened joints, where, due to stagnation, the damping effect of dry friction forces appears only at sufficiently large values of *A*, and with its further increase it lags behind the damping effect of linear friction forces. Equation of free vibrations: $\ddot{v} + F(v,\dot{v}) + v = 0$,

where
$$F(v,\dot{v}) = \begin{cases} æ\dot{v} & at\,|v| \prec A_1, \\ æ\dot{v} + æ_2\,\mathrm{sign}\,\dot{v} & at\,|v| \succ A_1, \end{cases} \quad (3)$$

describes the combined action of linear and dry friction forces.

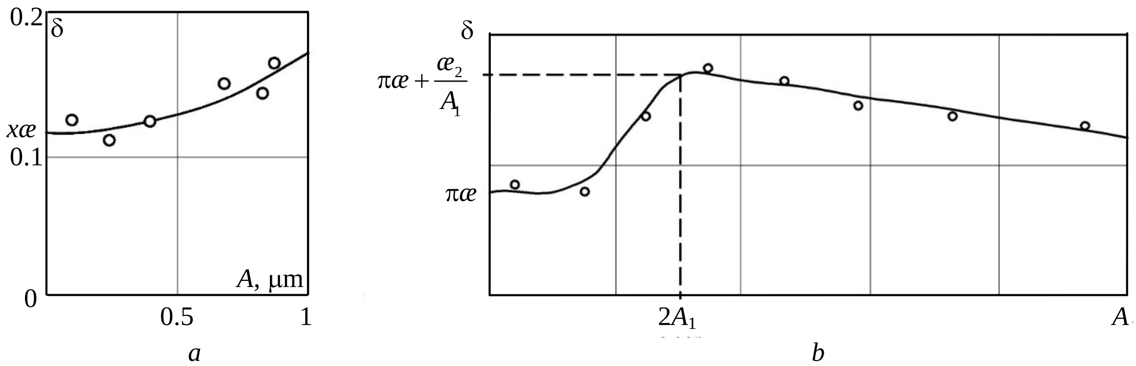

Fig. 4. Approximation of experimental data by calculated curves for monotonic (*a*) and non-monotonic (*b*) changes in the logarithmic decrement with damping of oscillations

To determine the parameters of nonlinear friction, we will use the technique of replacing the force $F(v,\dot{v})$ with an equivalent force of linear friction, expressing the decrement of oscillations $\delta = \Delta U / 2U_{\max}$ in terms of the energy ΔU dissipated during one cycle of oscillations and the maximum potential energy of the cycle $U_{\max}$. Assuming the quasi-harmonic nature of vibrations, we find from (3):

$$\delta = \begin{cases} \pi æ & at\ A \prec A_1, \\ \pi æ + \frac{4æ_2}{A}\left(1 - \frac{A_1}{A}\right) & at\ A \succ A_1. \end{cases} \quad (4)$$

The procedure for processing experimental data (Fig. 4b) is based on the fact that the function δ(*A*), given by expression (3), reaches its maximum value $\delta = \pi æ + \frac{æ_2}{A_1}$ at $A = 2A_1$.

Research results

An experimental study of the characteristics of finishing and boring heads depending on the force P_p of the preliminary axial preload of the bearings makes it possible to determine the optimal values of the tension P_p^{opt}. The value of P_p was set by the difference in the heights of the spacer rings and was measured by strain gauges. With increasing P_p, the radial compliance and amplitude of forced vibrations decrease, the temperature of the heads increases, and the vibration decrement changes non-monotonically, reaching a maximum at P_p values depending on the spindle diameter.

The extreme properties of the dependence of the logarithmic decrement of oscillations δ on P_p are explained by the occurrence of linear comparison and dry friction forces in the dynamic system of the spindle assembly. It is assumed that the dry friction forces arising on the contact surface of elements that do not perform specified movements are proportional:

$P_p : A_1 = b_1 P_p,\ æ_2 = b_2 P_p$, (in accordance with 3 and 4).

Damped oscillations correspond to a phase trajectory (Fig. 5), crossing a stagnation band with a width of $2A_1$. The closeness of the values of δ for small and large tension forces shows that the linear resistance forces weakly depend on P_p. Therefore, from relation (4) we find:

$$\delta = \pi æ + \frac{4b_2 P_p}{A}\left(1 - \frac{b_1 P_p}{A}\right), \ \frac{d\delta}{dP_p} \approx \frac{4b_2}{A}\left(1 - \frac{2b_1 P_p}{A}\right).$$

$$P_p^{\text{opt}} = \frac{A}{2b_1}, \ \delta_{\max} = \pi æ + \frac{b_2}{b_1}. \quad (5)$$

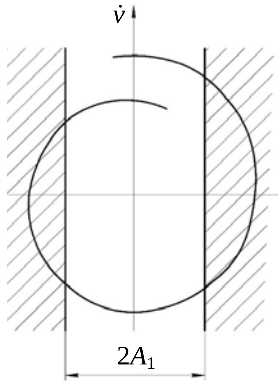

Fig. 5. Phase trajectory of damped oscillations: $2A_1$ – stagnation band width

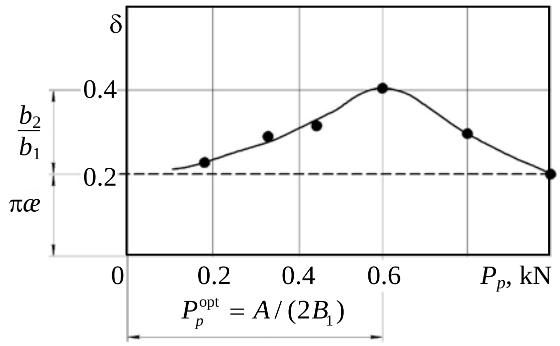

Fig. 6. Dependence of the logarithmic decrement of oscillations of the spindle with a boring bar on the axial preload force: • – experiment with a spindle diameter of 40 mm

Using these relationships, the parameters of the inelastic resistance forces æ, b_1, b_2 are determined from experimental data (Fig. 6). Let us present the data obtained for bearings 46100. For spindle diameter d = 40...50 mm, P_p^{opt} = 0.6 kN was found, for d = 75 mm P_p^{opt} =0.8 kN and for d = 100...120 mm P_p^{opt} = 1.2 kN.

From relations (5) it follows that the value P_p^{opt} depends not only on the design parameters and quality of manufacturing of the spindle assembly parts, which affect the coefficient b_1, but also on the level of disturbances under given processing conditions, which primarily affect the oscillation amplitude A.

At optimal tension values, the radial rigidity is sufficient to achieve maximum accuracy of the spindle assembly, and the temperature increase does not exceed 4...6°. The increase in system rigidity with increasing P_p above the optimal value turns out to be insignificant. This allows us to recommend set values P_p^{opt} for use when assembling finishing and boring heads.

Dynamic calculation of finishing and boring heads showed that impacts with rotational speed in the front support affect tool movements more strongly than impacts in the rear support, and high-frequency impacts in the front and rear supports cause vibrations with almost identical amplitudes (Fig. 7). It follows that the requirements for accuracy and tension in both supports must be the same.

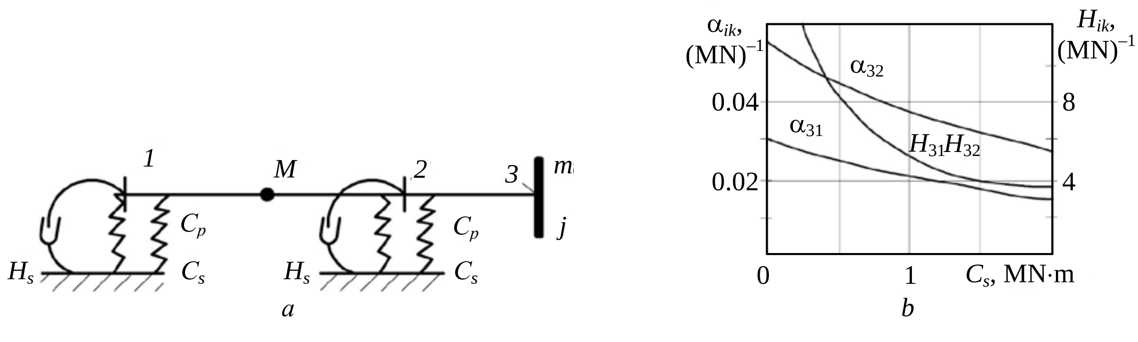

Fig. 7. Design diagram (*a*) and results (*b*) of calculating the coefficients of transmission of disturbances from the spindle supports to the tool: M, m – reduced masses of the span and cantilever parts of the system; j is the reduced moment of inertia of the cantilever part; α_{ik} – static influence coefficients; H_{ik} – resonant values of the transmission coefficients of harmonic moment disturbances

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

Increasing the dynamic quality of spindle assemblies with a cantilever tool is also achieved by creating a cavity at the free end of the cantilever. This method is based on the extreme properties of the first natural frequency of bending vibrations of a console with a cavity, determined by the fact that the weakening of sections near the free end significantly reduces the reduced mass, weakly affecting the rigidity, and vice versa near the pinching.

Let us represent the first angular natural frequency of bending vibrations in the form:

$$\omega_1 = \sqrt{\frac{12EI_1}{\mu_1 l_1^4}} \cdot \Psi(\varsigma, \lambda), \tag{6}$$

where I_1 and μ – axial moment of inertia of the section and linear mass in the unweakened part of the console, $\varsigma = d_0 / d_1$, $\lambda = l_0 / l_1$. The values of the function $\Psi(\varsigma, \lambda)$ are calculated using the Rayleigh method. When the diameter of the cavity changes within the range of (0.3...0.9)*d*, its length, corresponding to the maximum of the first natural frequency, is 0.5...0.6 of the console length.

The presence of a cavity leads to a decrease in the transmission coefficients of disturbances in the system of the spindle assembly with the tool. When drawing up the calculation diagram of the elastic system, a reduction was made to three masses concentrated in the middle of the spindle span, at the center of gravity of the flange and at the cutter (Fig. 8, *a*).

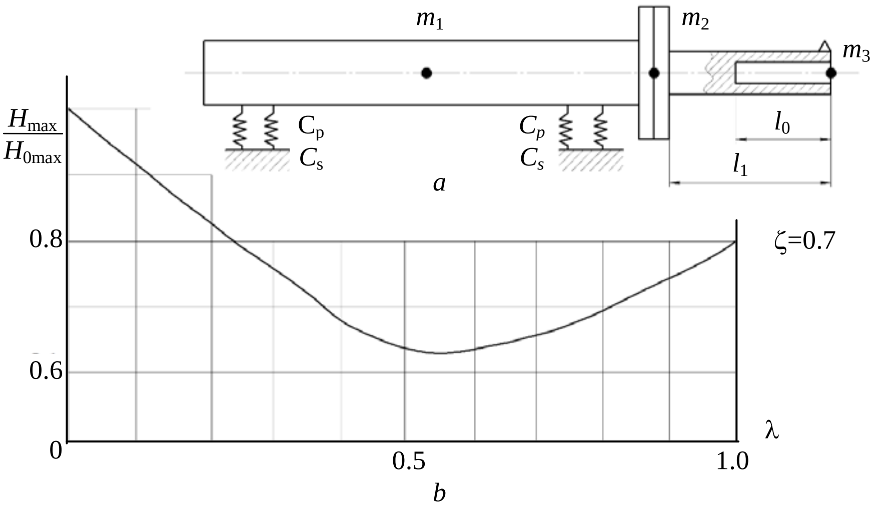

Fig. 8. The influence of the cavity length on the coefficient of transmission of disturbances from the front support to the cutter

The coefficients *H* of the transmission of disturbances from the front support of the spindle to the cutter are calculated depending on the length of the cavity l_0. Fig. 8 *b* shows the calculation results for the spindle head AP-IB in the form of the ratio of the resonance values H_{max} and H_{0max}, the latter of which corresponds to a boring bar without a cavity. The minimum disturbance transfer coefficient is achieved at a cavity length value close to that generating the maximum natural frequency of bending vibrations.

Conclusions

Analysis of head designs allows us to draw the following conclusions.

1. The vast majority of serial heads have a spindle and flange. Mostly special heads are produced without flanges, including those with remote spindles. Double bearings are equipped with spacer rings, the height of which for adjacent bearings is chosen structurally from 3 to 10 mm.

In heads with widely spaced bearings, d_1 is often 5…15 mm larger than *d*; the distance between the ends of the double support bearings ranges from 20 to 100 mm. The maximum distance h_n between the dual bearings of the front support should not exceed $1/3l$. The minimum length of the console a_1 is determined structurally and is not of decisive importance, since the lengths l_1 of the boring bars must be many times greater than the length of the console, and the ratio of the distance *l* between the supports to the reach from the front support to the cutter l_1 is within 1-2.

2. Spindle units of finishing boring machines are the most critical design elements that ensure processing accuracy. Calculations of bending movements of cantilever boring bars without taking into account the resistance to rotation of the spindle bearings lead to significant errors. The problem of optimizing the damping parameters of spindle units is considered, taking into account the rotational rigidity of the spindle supports. The rolling bearings of the spindle units are non-stationary elements of the elastic system. Variability of contact conditions in rolling bearings arises due to inaccuracies in the manufacture of bearing rings and rolling elements, as well as the impact of the latter with cages. As a result of interactions, parametric disturbances and forcing forces arise that move the support sections. The excited narrow-band random oscillations have a spectrum that depends on the periods of rotation of the bearing elements.

Studies of bearing units have proven the need to optimize the preload value of rolling bearings P_p. It has been established that the amount of interference affects the main performance characteristics of spindle units, and its optimal value is determined by a cumulative assessment of rigidity, vibrations, heating, processing accuracy and durability of bearings.

3. Experimental studies of the characteristics of spindle heads make it possible to determine the optimal values of the axial tension P_p^{opt}. The value P_p was set by the difference in the heights of the spacer rings and was measured tensometrically. With P_p increasing, the radial compliance and amplitudes of forced vibrations decrease, the temperature of the heads increases, and the vibration decrement changes non-monotonically, reaching a maximum at $P_{\text{н}}$ values of depending on the spindle diameter.

With a spindle diameter d = 50 mm, $P_p^{\text{opt}} = 0.6\ \text{kN}$, and with d = 120 mm, $P_p^{\text{opt}} = 1.2\ \text{kN}$. At optimal tension values, the radial rigidity is sufficient to achieve maximum accuracy of the spindle assembly, and the temperature increase does not exceed 5…6°.

4. The experimental properties of the dependence of the logarithmic decrement of oscillations δ on P_p are determined by the forces of linear resistance and dry friction arising in the dynamic system of the spindle assembly. It has been established that the forces of linear friction weakly depend on P_p, and the logarithmic decrement of oscillations δ at P_p^{opt} reaches values of 0.35…0.4.

5. It has been established that impacts with rotational speed in the front support affect the vibrations of the cutter more than disturbances in the rear support. High-frequency disturbances in the front and rear supports cause vibrations with almost identical amplitudes. Therefore, the requirements for accuracy and tension in both supports must be the same.

6. Based on the determination of the static and harmonic transmission coefficients of disturbances from the spindle supports to the tool, the possibility of increasing the dynamic quality of spindle units by creating a cavity at the free end of the boring bar has been proven. The presence of a cavity (very convenient for installing vibration dampers) leads to a decrease in the transmission coefficients of disturbances from the spindle supports to the tool. The minimum transmission coefficient occurs at the value of the cavity length at which the maximum natural frequency of bending vibrations is excited.

Література

1. Improvement of the Dynamic Quality of Cantilever Boring Bars for Fine Boring / Oborskyi G., Orgiyan A., Ivanov V., Balaniuk A., Pavlenko I., Trojanowska J. *Machines.* 2023. 11. 7. DOI: https://doi.org/10.3390/machines11010007.
2. Improvement of the physical and mechanical properties of the cutting tool by applying wear-resistant coatings based on Ti, Al, Si, and N. J. / Hovorun T., Khaniukov K., Varakin V., Pererva V., Vorobiov S., Burlaka A., Khvostenko R. *Eng. Sci.* 2021. 8. P. 13–23.
3. Auleley M., Thomas O., Giraud-Audine C., Mahé H. Enhancement of a dynamic vibration absorber by means of an electromagnetic shunt. *J. Intell. Mater. Syst. Struct.* 2020. 32. P. 331–354.
4. Thomas O., Ducarne J., Deü J-F. Performance of piezoelectric shunts for vibration reduction. *Smart Materials and Structures*. 2012. 21(1). P. 1–16.
5. Weber F., Distl H., Fischer S., Braun C. MR Damper Controlled Vibration Absorber for Enhanced Mitigation of Harmonic Vibrations. *Actuators.* 2016. 5. 27. DOI: https://doi.org/10.3390/act5040027.
6. Zverev I., Eun I.U., Lee C. Simulation of Spindle Units Running on Rolling Bearings. *Int. Journ. Adv. Manufac. Tech.* 2003. 21. P. 889–895. DOI: https://doi.org/10.1007/s00170-002-1411-2.

7. Kim J., Zverv I., Lee K. Model of Rotation Accuracy of High-Speed Spindles on Ball Bearings. *Engineering*. 2010. Vol. 2 No. 7. P. 477–484. DOI: 10.4236/eng.2010.27063.
8. Kim J., Zverv I., Lee K. Thermal Model of High-Speed Spindle Units. *Intelligent Information Management*. 2010. Vol. 2 No. 5. P. 306–315. DOI: 10.4236/iim.2010.25036.
9. Hongqi L., Yung C. S. Analysis of Bearing Con-figuration Effects on High Speed Spindles Using an Inte-grated Dynamic Thermo Mechanical Spindle Model. *International Journal of Machine Tools and Manufacture*. 2004. Vol. 44. P. 347–364.
10. S.C. Jeng and W.C. Kwan. Bearing Load Analysis and Control of Motorized High Speed Spindle. *International Journal of Machine Tools and Manufacture*. 2005. Vol. 45, No. 12-13. P. 1487–1493.
11. 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.
12. Hentati T., Barkallah M., Bouaziz S. Dynamic modeling of spindle-rolling bearings systems in peripheral milling operations. *J. Vibroeng.* 2016. 18. 1444–1458.
13. Dynamic Modeling for Chatter Analysis in Micro-Milling by Integrating Effects of Centrifugal Force, Gyroscopic Moment, and Tool Runout / Liu X., Liu D., Du C., Li Y., Wang C., Fu Z. *Micromachines.* 2024. 15. 244. DOI: https://doi.org/10.3390/mi15020244.
14. Данильченко Ю.М., Петришин А.И. Моделирование форм колебаний механической колебательной системы «шпиндельный узел» металлорежущего станка. *Вісник НТУУ «КПІ». Серія ашинобудування*. 2012. № 66. С. 46–50.
15. Данильченко Ю.М., Петришин А.И. Идентификация колебаний шпинделя по результатам измерения вибраций корпуса шпиндельного узла. *Вісник НТУУ «КПІ». Серія машинобудування.* 2014. №2 (71). С. 147–152.
16. An experimental study on the rotational accuracy of variable preload spindle-bearing system / Hu G., Gao W., Chen Y., Zhang D., Tian Y., Qi X., Zhang H. *Adv. Mech. Eng*. 2018. 10. 1–14.
17. Зверев И.А., Данильченко Ю.М. Кинематика и жесткость радиально-упорных шарикоподшипников. *Вісник НТУУ «КПІ». Серія машинобудування.* 2013. No3 (69). С. 5–12.
18. Зверев И.А., Данильченко Ю.М. Комплексное моделирование при проектировании шпиндельных узлов на опорах качения. *Вісник СевНТУ: зб. наук. пр. Серія: Машиноприладобудування та транспорт*. 2014. Вип. 150/2014. С. 75–80.

References

1. Oborskyi, G., Orgiyan, A., Ivanov, V., Balaniuk, A., Pavlenko, I., & Trojanowska, J. (2023). Improvement of the Dynamic Quality of Cantilever Boring Bars for Fine Boring. *Machines,* 11, 7. DOI: https://doi.org/10.3390/machines11010007.
2. Hovorun, T., Khaniukov, K., Varakin, V., Pererva, V., Vorobiov, S., Burlaka, A., & Khvostenko, R. (2021). Improvement of the physical and mechanical properties of the cutting tool by applying wear-resistant coatings based on Ti, Al, Si, and N. *J. Eng. Sci.,* 8, 13–23.
3. Auleley, M., Thomas, O., Giraud-Audine, C., & Mahé, H. (2020). Enhancement of a dynamic vibration absorber by means of an electromagnetic shunt. *J. Intell. Mater. Syst. Struct.*, 32, 331–354.
4. Thomas, O., Ducarne, J., & Deu¨, J-F. (2012). Performance of piezoelectric shunts for vibration reduction. *Smart Materials and Structures*, 21(1), 1–16.
5. Weber, F., Distl, H., Fischer, S., & Braun, C. (2016). MR Damper Controlled Vibration Absorber for Enhanced Mitigation of Harmonic Vibrations. *Actuators,* 5, 27. DOI: https://doi.org/10.3390/act5040027.
6. Zverev, I., Eun, I.U., & Lee, C. (2003). Simulation of Spindle Units Running on Rolling Bearings. *Int. Journ. Adv. Manufac. Tech.*, 21, 889–895. DOI: https://doi.org/10.1007/s00170-002-1411-2.
7. Kim, J., Zverv, I., & Lee, K. (2010). Model of Rotation Accuracy of High-Speed Spindles on Ball Bearings. *Engineering*, 2, 7, 477–484. DOI: 10.4236/eng.2010.27063.
8. Kim, J., Zverv, I., & Lee, K. (2010). Thermal Model of High-Speed Spindle Units. *Intelligent Information Management*, 2, 5, 306–315. DOI: 10.4236/iim.2010.25036.
9. Hongqi, L., &Yung, C. S. (2004). Analysis of Bearing Con-figuration Effects on High Speed Spindles Using an Inte-grated Dynamic Thermo Mechanical Spindle Model. *International Journal of Machine Tools and Manufacture*, 44, 347–364.
10. S.C. Jeng, & W.C. Kwan. (2005).,Bearing Load Analysis and Control of Motorized High Speed Spindle. *International Journal of Machine Tools and Manufacture*, 45, 12-13, 1487–1493.
11. Danylchenko, Y., Storchak, M., Danylchenko, M., & Petryshyn, A. (2023). Cutting Process Consideration in Dynamic Models of Machine Tool Spindle Units. *Machines*, 11, 582. DOI: https://doi.org/10.3390/machines11060582.

12. Hentati, T., Barkallah, M., & Bouaziz, S. (2016). Dynamic modeling of spindle-rolling bearings systems in peripheral milling operations. *J. Vibroeng*, 18, 1444–1458.
13. Liu, X., Liu, D., Du, C., Li, Y., Wang, C., & Fu, Z. (2024). Dynamic Modeling for Chatter Analysis in Micro-Milling by Integrating Effects of Centrifugal Force, Gyroscopic Moment, and Tool Runout. *Micromachines*, 15, 244. DOI: https://doi.org/10.3390/mi15020244.
14. Danylchenko, Yu.M., & Petryshyn, A.I. (2012). Modeling of the oscillating forms of the mechanical oscillating system "spindle node" of the metal-cutting machine. *Bulletin of NTUU "KPI". A series of railway construction*, 66, 46–50.
15. Danylchenko, Yu.M., & Petryshyn, A.I. (2014). Identification of the oscillating spindle based on the results of measuring the vibrations of the spindle unit housing. *Bulletin of NTUU "KPI". Mechanical engineering series*, 2 (71), 147–152.
16. Hu, G., Gao, W., Chen, Y., Zhang, D., Tian, Y., Qi, X., & Zhang, H. (2018). An experimental study on the rotational accuracy of variable preload spindle-bearing system. *Adv. Mech. Eng.* 10, 1–14.
17. Zverev, I.A., & Danilchenko, Yu.M. (2013). Kinematics and stiffness of angular-contact ball bearings. *Newsletter of NTUU "KPI". Machine-building series*, 3 (69), 5–12.
18. Zverev, I.A., & Danilchenko, Yu.M. (2014). Complex modeling in the design of spindle units on rolling bearings. *News of SevNTU: zb. Sci. etc. Series: Machinery equipment and transport.* 150/2014, 75–80.

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
Оборський Геннадій Олександрович; Gennadiy Oborskyi, ORCID: https://orcid.org/0000-0002-5682-4768
Колеснік Василь Михайлович; Vasyl Kolesnik**,** ORCID**:** https://orcid.org/0000-0002-6573-2961
Мацей Руслан Афанасійович; Ruslan Matzey, ORCID: https://orcid.org/0000-0001-6393-8678

Received March 23, 2024

Accepted May 10, 2024