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Daniela Doina Cioboată *Editor*

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Contents

Determination of Additional Braking Force for Hydraulic Cylinder Piston

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Abstract. The braking process of the hydraulic cylinder piston due to the displacement of the working fluid from the end gap between the side surfaces of the piston and the cover is considered. To describe the fluid flow, the cylindrical coordinate system is used; the flow is considered radial and laminar. To study the flow, the inertia of the flow, the compressibility of the fluid and the action of mass forces are neglected. As the mathematical model for the process of displacing the working fluid from the end gap between the side surfaces of the piston and the cover, the differential equation of the laminar axisymmetric motion of the incompressible viscous fluid is considered. By integrating the obtained equation of motion using Newton's law for fluid friction, the distribution for the velocity of the working fluid in the gap between the side surfaces of the piston and the cover is established. Based on the distribution of the fluid velocity in the gap, ordinary differential equation was obtained for the pressure distribution along the radius of the hydraulic cylinder piston. According to the established pressure distribution, the dependence was determined for calculating the additional braking force of the hydraulic cylinder piston. The analysis of the obtained dependence is carried out, the main parameters are established that determine the braking force due to the displacement of the working fluid from the end gap between the side surfaces of the piston and the cover of the hydraulic cylinder. The example of calculation is presented.

Keywords: Hydraulic cylinder · Braking force · Equation of motion · Flow velocity · Viscosity

1 Introduction

Hydraulic drives are widely used to move the working bodies of various machines. Hydraulic drives are especially widely used in automatic control systems of the working bodies of machines included in a closed technological cycle. There are automatic control systems for metal-cutting machines and automatic lines, robotic manipulators and presses, technological machines for metallurgical, food, light industries, etc. [1–5].

The widespread use of hydraulic drives in the considered areas is determined by their important advantages, which primarily include the ability to obtain large forces and torques with relatively small sizes of hydraulic motors, the smooth movement and

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stepless speed control in the large range, the low inertia, the ability to control modes processing during the movement of the working bodies, the simplicity of the implementation of rectilinear reciprocating movements and automatic control of the working bodies, the ease of protection against overloads and high operational reliability [6–10].

High layout properties of hydraulic systems, based on the constructive independence of the location of individual units, make it possible to create machines that are distinguished by high productivity, reliability and low material consumption. Machine tool construction belongs to those industries where hydraulic drives are traditionally used. Now in metal-cutting tools and forging equipment, the hydraulic drive is used to carry out both main and auxiliary movements, including automatic tracking movements of actuators, drive of working bodies, robotic manipulators, clamping, fixing and transport devices [11–15].

2 Literature Review

When designing hydraulic drives of systems, it is very important to assess the reliability, safety and quality of the system created on their basis. Such assessment can be given by studding of the dynamics of the hydraulic drive and technological equipment as whole, which is the final computational and design stage of creating equipment, automatic control systems with hydraulic drives. The main purpose of the dynamic research for hydraulic systems is to test the operability of the drive or control system based on one or another drive under typical external disturbing factors, as well as under given input (control) influences [16–20].

It is advisable to study the dynamics of hydraulic drives by means of mathematical modeling, which is based on the creation of a mathematical model, taking into account all the features of the drive and adequately reflecting its behavior in the dynamics of a system designed on its basis. The creation of a mathematical model and its research is based on a systematic approach to the description of all elements of a hydraulic scheme or drive, taking into account their dynamic characteristics, based on the methods of decomposition for complex interconnected elements of hydraulics, hydraulic devices and electrical equipment. The mathematical description of all elements is performed in the form of algebraic, logical, differential-integral equations and the representation of the latter in a form convenient for further research using software [21–25].

Despite the fact that the literature contains extensive material [26–30] on the mathematical description of working processes in hydraulic cylinders, there are a number of factors that affect their dynamic properties and, at the same time, are not fully understood. Among these factors is the additional braking force of the hydraulic cylinder piston, which arises due to the displacement of the working fluid from the end gap between the side surfaces of the piston and the cover.

The purpose of this paper is to study the braking process of the hydraulic cylinder piston by displacing the working fluid from the end gap between the side surfaces of the piston and the cover, determining the dependence for calculating the additional braking force of the hydraulic cylinder piston.

3 Study the Braking Process of the Hydraulic Cylinder Piston

In Figs. 1, 2 show typical designs of the single-rod and double-rod hydraulic cylinders, respectively. In these designs, the positioning of the hydraulic cylinder piston is carried out on rigid stops, which are the inner surfaces of the side covers.

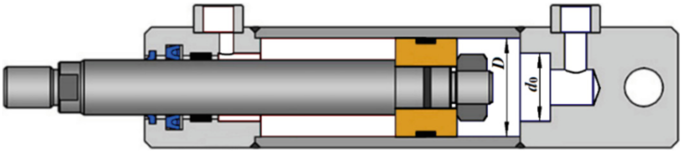

Fig. 1. The single-rod hydraulic cylinder.

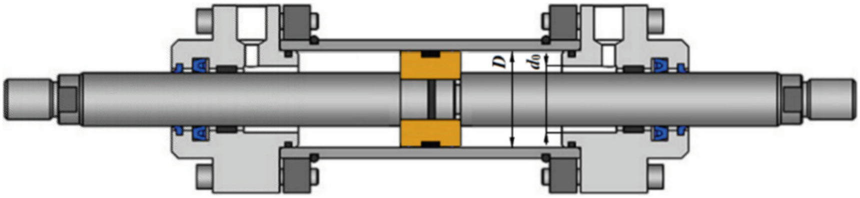

Fig. 2. The double-rod hydraulic cylinder.

Before the piston reaches the extreme position (for example, the extreme right one in Figs. 1, 2), the process of its braking occurs due to the displacement of the working fluid from the end gap between the side surfaces of the piston and the cover. The design scheme of the fluid movement in the variable end gap is shown in Fig. 3.

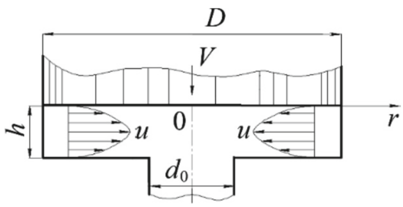

Fig. 3. The design scheme of the fluid movement in the variable end gap.

The diagram shows: V – velocity of the hydraulic cylinder piston; u – velocity of the working fluid in the end gap between the side surfaces of the piston and the cover; D – piston diameter; d_0 – diameter of the channel in the hydraulic cylinder cover; h – height of the variable end gap; r – radial coordinate.

It is quite appropriate to assume that the flow is axisymmetric and, therefore, to use the cylindrical coordinate system, placing the z axis along the piston axis (Fig. 4). We consider the flow to be laminar. We assume that the flow is quasi-radial, i.e., the tangential component of the velocity u_φ is equal to 0, and the axial component of the v locity u_z is much less than the radial u_r. We also neglect the flow inertia, fluid compressibility and the action of mass forces. Thus, the velocity of the working fluid $u = u_r$ in the end gap u is the function of the coordinates z and r.

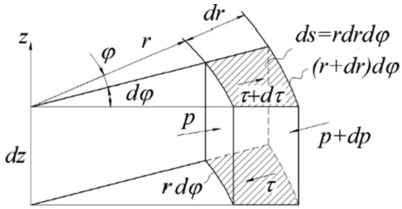

Fig. 4. The elementary volume of the fluid in the end gap.

By virtue of the assumptions made for the elementary volume of the liquid (see Fig. 4), the differential equation of the laminar axisymmetric motion of the incompressible viscous fluid is valid

$$(\tau + d\tau)ds = (p + dp)(r + dr)d\varphi dz + prd\varphi dz, \tag{1}$$

where p – hydrostatic pressure; τ – shear stress; $ds = rd\varphi dr$ – elementary area on which the shear stress acts; dp, $d\tau$, dz, dr, $d\varphi$ – increments of variables and coordinates.

Neglecting the terms of the equation of a grater order of smallness, we further simplify expression (1)

$$rd\tau drd\varphi = pdzdrd\varphi + rdpdzd\varphi. \tag{2}$$

We transform (2) and obtain the following differential equation

$$\frac{d\tau}{dz} = \frac{p}{r} + \frac{dp}{dr}. \tag{3}$$

We integrate (3) along the z coordinate in the range from zero to the size of the gap h, taking into account that $dp/dz = 0$,

$$\frac{\tau|_{z=h} - \tau|_{z=0}}{h} = \frac{p}{r} + \frac{dp}{dr}. \tag{4}$$

To calculate the shear stress on the piston and the cover, we obtain the velocity distribution over the gap. For this, we transform expression (2) taking into account the Newton's law of fluid friction

$$\tau = \rho\nu\frac{du}{dz}, \tag{5}$$

where ρ, ν – density and kinematic viscosity of the working fluid.

Then from (3)

$$\rho\nu\frac{d^2u}{dz^2} = \left(\frac{p}{r} + \frac{dp}{dr}\right). \tag{6}$$

Since $dp/dz = 0$, we have

$$\frac{d^2u}{dz^2} = A, \tag{7}$$

where A – parameter that does not depend on the z coordinate.

Therefore, the velocity distribution is a parabola, the equation of which can be represented as

$$u = -6u_0\left(1 - \frac{z}{h}\right)\frac{z}{h}, \tag{8}$$

where u_0 – average velocity over the height of the gap, and the sign "-" indicates the direction of the velocity to the coordinate center.

Without taking into account the compressibility, the flow rate of the working fluid through the annular slot $2\pi rh$ with the area is equal to the flow rate displaced by the piston in the area $\pi\left(D^2/4 - r^2\right)$, therefore

$$u_0 = \frac{V\pi\left(\frac{D^2}{4} - r^2\right)}{2\pi rh} = \frac{V}{2h}\left(\frac{D^2}{4r} - r\right). \tag{9}$$

Then

$$u = -\frac{3V}{h}\left(\frac{D^2}{4r} - r\right)\left(1 - \frac{z}{h}\right)\frac{z}{h}. \tag{10}$$

According to (5), we define

$$\tau|_{z=0} = \rho\nu\frac{du}{dz}|_{z=0} = -\frac{3V\rho\nu}{h^2}\left(\frac{D^2}{4r} - r\right), \tag{11}$$

$$\tau|_{z=h} = \rho\nu\frac{du}{dz}|_{z=h} = \frac{3V\rho\nu}{h^2}\left(\frac{D^2}{4r} - r\right). \tag{12}$$

Substituting (10), (11) into (4), we obtain the ordinary differential equation of the first order for the pressure distribution along the piston radius

$$\frac{dp}{dr} = \frac{6V\rho\nu}{h^2}\left(\frac{D^2}{4r} - r\right) - \frac{p}{r}, \tag{13}$$

which we integrate with the initial condition $p = 0$ at $r = d_0/2$.

We get the following distribution

$$p = \frac{6V\rho\nu}{h^2}\left(\frac{D^2}{4}\left(1 - \frac{d_0}{2r}\right) - \frac{r^2}{3}\left(1 - \frac{d_0^3}{8r^3}\right)\right). \tag{14}$$

For the additional braking force for the hydraulic cylinder piston, the integral expression is valid

$$F_b = \int_0^{2\pi} d\varphi \int_{d_0/2}^{D/2} prdr. \tag{15}$$

We substitute (13) into (14) and integrate, after which we obtain

$$F_b = k_d \frac{\rho \nu V D^4}{h^3}, \tag{16}$$

where k_d – dimensionless coefficient determined by the ratio of diameters d_0/D

$$k_d = \frac{3\pi}{4}\left(\frac{1}{2}\left(1-\frac{d_0}{D}\right)^2 - \frac{1}{3}\left(\frac{1}{4}\left(1-\frac{d_0^4}{D^4}\right) - \frac{d_0^3}{D^3}\left(1-\frac{d_0}{D}\right)\right)\right). \tag{17}$$

The dependence $k_d(d_0/D)$ is shown in Fig. 5. In Fig. 6 shows the example of calculating the additional braking force for the hydraulic cylinder piston due to the displacement of the working fluid from the end gap between the side surfaces of the piston and the cover for the following parameters: $D = 100$ mm; $d_0 = 50$ mm; $\rho = 900$ kg/m^3; $\nu = 30$ cSt.

It should be noted that in dynamics the velocity of the piston movement does not remain constant, but decreases, therefore, the value of the braking force also decreases.

The obtained dependence for the additional braking force (15) can be used to study the dynamics of the hydraulic cylinder piston, the dynamic characteristics of the hydraulic drive as a whole, as well as to make a decision on the need to install additional braking devices for the output link of the hydraulic drive.

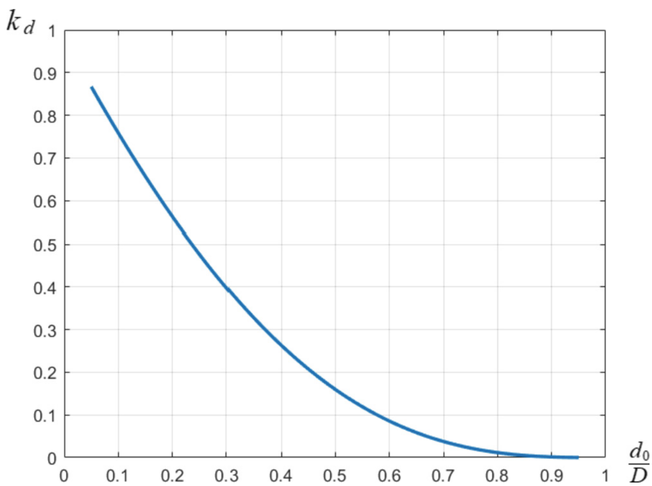

Fig. 5. The dependence for the coefficient k_d.

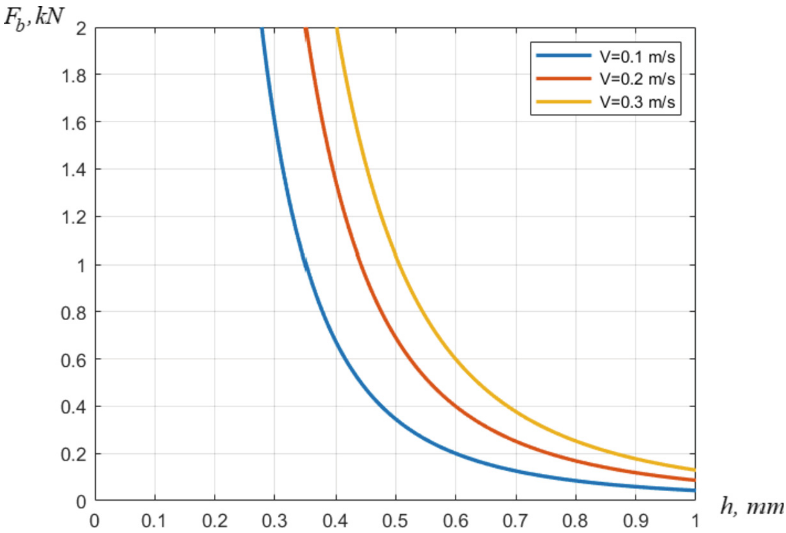

Fig. 6. To determine the velocity distribution in a flat slot.

4 Conclusions

Thus, the braking process of the hydraulic cylinder piston due to the displacement of the working fluid from the end gap between the side surfaces of the piston and the cover is considered. To describe the fluid flow, the cylindrical coordinate system was used; the flow is considered radial and laminar. To study the flow, the inertia of the flow, the compressibility of the fluid and the action of mass forces are neglected.

As the mathematical model for the process of displacing the working fluid from the end gap between the side surfaces of the piston and the cover, the differential equation of the laminar axisymmetric motion of the incompressible viscous fluid is considered. By integrating the obtained equation of motion using Newton's law for fluid friction, the distribution for the velocity of the working fluid in the gap between the side surfaces of the piston and the cover is established. Based on the distribution of the fluid velocity in the gap, ordinary differential equation was obtained for the pressure distribution along the radius of the hydraulic cylinder piston.

According to the established pressure distribution, the dependence was determined for calculating the additional braking force of the hydraulic cylinder piston. The analysis of the obtained dependence is carried out, the main parameters are established that determine the braking force due to the displacement of the working fluid from the end gap between the side surfaces of the piston and the cover of the hydraulic cylinder. The example of calculation is presented.

The obtained dependence for the additional braking force can be used to study the dynamics of the hydraulic cylinder piston, the dynamic characteristics of the hydraulic drive as a whole, as well as to make a decision on the need to install additional braking devices for the output link of the hydraulic drive.

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Modular Spindle Tooling of the Machining Center with Increased Resource

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Abstract. The design of spindle heads of multi-axis metal-cutting machines with driven bevel gears is considered. The kinematic diagrams of the main movement drive for multioperational machines equipped with spindle heads of different technological capabilities have been constructed. Three-dimensional modeling of horizontal, vertical and angular spindle heads in the integrated computer-aided design system KOMPAS-3D is carried out. 3D models of complex housing parts of heads using an extended range of graphic primitives and new versions of the geometric and parametric cores for the system have been built. The new functional of the specialized applied application of the KOMPAS system is used in the procedure for express-building of high-precision models of bevel gear rims with geometrically correct tooth surfaces. The rendering of the spindle head structures in the Artisan rendering module is performed. A variant of solving the problem of increasing the two-stage drive mechanisms service life of spindle heads according to the criterion of effective bevel gears lubrication is proposed. An analytical apparatus for partitioning the total gear ratio of a two-stage drive for cases of using gears with straight and circular teeth is presented. A derivative analytical dependence, which approximates the basic expression reflecting the ratio of the gear ratios of the two-stage drive mechanism is introduced. Variants of a possible combination of hardness of bevel gear teeth on the first and second stages of the spindle head drive mechanism are considered.

Keywords: Spindle head · 3D modeling · Bevel gear · Transmission resource · Gear ratios

1 Introduction

In the process of creating new designs of multioperational machine tools (MMT), special influence is paid to the design of spindle heads and spindle assemblies, which are the main structural elements of any milling, drilling and boring machine. These particular characteristics that mainly affect the choice of a machine tool and are always the main ones, since the quality of the finished product depends on the reliability and accuracy of these elements.

One of the main trends in the growth of machining efficiency is the increase in the technological capabilities of machine tools. In this regard, various spindle heads for

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MMT of the drilling, milling and boring group such as machining centers are used [1–3]. It is here that the principle of modular construction of machine tools based on unified and specialized designs of spindle heads is most fully implemented. As a result, there is an effect of increasing the technological possibilities of using the machine tool [4, 5] and increasing the variety of part machining, without changing the basic (bearing system, drives) of the machine structure.

A lot of research and development is currently underway on additional modular tooling for modern horizontal machining centers. The design of such machines is based on the idea associated with the possibility of changing the working position of the tool blocks in space. As an example, we consider removable angular spindle heads (ASH) with a variable angle of the spindle part with a working tool, which are attached to the spindle unit of the machine [6]. The consequence of this is an increase in the number of differently located surfaces of the part that can be processed from one setup. Taking into account the frequent change in the range of processed parts, the variety of their geometric shapes, the development of new spindle heads designs is an urgent task.

2 Literature View

The issues of shaping spindle unit's development for MMT are presented in works [7–9].

The work [7, 8] considers the analysis of the functioning of a multi-axis milling machine with various technological equipment. The authors' approach to the procedure for revealing the influence of the machine tool layout with various configurations of the rotation axes of the spindle assemblies and rotary tables with workpieces mounted on them is interesting. In the developed databases, various alternative options for the layouts and configurations of the coordinate axes are presented. Databases were structured in the form of separate sections: a) spindle heads; b) instrumental blocks; c) rotary tables, etc. Along with the design component, a complex of analytical models and experimental data arrays was formed. At the same time, the issues of complex 3D modeling of spindle heads using unified components and specialized applications are not touched upon in this work.

In [9], 2D and 3D modeling of a horizontal spindle assembly mounted on two duplex bearings on the front support and a single bearing on the rear support was carried out. An iterative procedure for calculating the stiffness of a spindle presented in the form of a beam element using the FEM toolkit has been implemented [10–12]. At the same time, the issues of modeling the vertical and angular heads of machining centers within the framework of creating a three-dimensional models section of spindle units were not touched upon in the work.

High efficiency and reliability of spindle heads depends on the resource and uninterrupted operation of drive gears [13, 14]. Among the various types of gears, bevel gears intended for transmit motion with a change in direction should be distinguished. This is especially important for machining centers with multi-axis machining capabilities. Therefore, the issues of increasing the durability, load capacity of the gear wheels of the main motion drive by improving design solutions are one of the dominant areas of research and development.

When designing gears at the initial stages of research [13], the search for an analytical apparatus for describing the profile of the lateral surface of the bevel gears teeth is carried out. This description will be the basis for obtaining optimized contact conditions over a wide range of gear transmission operation.

In the fundamental work [14], a new design of gears with increased load capacity for contact stresses is proposed. The design component is based on material-physical ratios (in accordance with the accepted normative calculation ANSI/AGMA 2003-B97) and allows you to assess the effect of the material on the subsurface layer. The proposed computational model makes it possible to assess the level of risk of side surface breakage and pitting corrosion based on load Tooth Contact Analysis. As for the geometric relationships, it should be noted that the new design of gear wheels is characterized by a slight decrease in the normal modulus and the total overlap ratio with an increased average pitch diameter.

Indicators of the load capacity and resource of gear transmissions depend not only on the improvement of their design, but also on the lubrication system. Under the conditions of the most common oil-bath lubrication system at speeds up to 12.5 m/s, a suspension of oil particles that uniformly cover the surfaces of the gear wheels ensure their continuous lubrication. The level of gear wheel immersion h_o an oil reservoir $3m \leq h_0 \leq 0.25d_{2l}$ is also regulated: depending on the gearing module m and the diameter of the low-speed stage d_{2l} [15]. For bevel gears, a significant difference in the diameters of the driven wheels of the two-stage transmission is possible. Moreover, the smaller of these wheels is not directly immersed in the oil reservoir, which worsens the contact conditions of the teeth.

The procedure of three-dimensional representation of the designed mechanism as an integrating link in the scale of the life cycle of creating a new gear mechanism is considered. According to [16], one of the main goals is to create a working 3D model for the analysis of the contact geometry, visual identification and assessment of the relationship of the set of the tooth geometric parameters. Important parameters include the parameters of the groove profile, which are largely determined by the shape of the longitudinal generatrix of the gear and wheel axoids. The design toolkit should include a method for creating a three-dimensional model of a gear in the corresponding integrated CAD systems, such as KOMPAS-3D [17–19].

Based on the analysis of the considered problems of creating an extended section of shaping spindle heads and increasing the resource of their 2-stage gear drive, we form the purpose of the research:

To carry out 3D modeling of the spindle head structures of metal-cutting machines with an increased resource of gear transmissions.

To achieve this purpose, it is necessary to solve the following tasks:

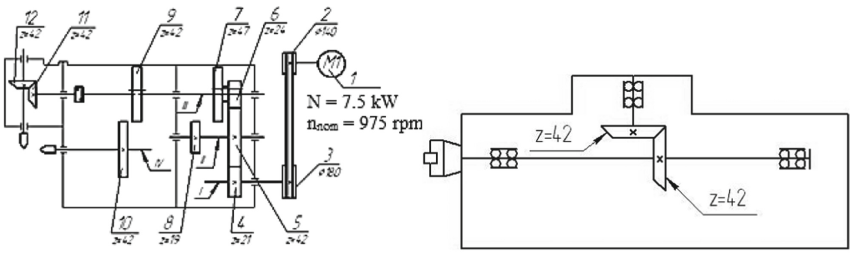
1. To develop a complex of three-dimensional models of MMT spindle heads in the CAD KOMPAS-3D environment using a specialized application program.

2. Research and create a modernized design of a gear drive with an increased resource.

3 Research Methodology. 3D Modeling of Spindle Heads

To increase the technological capabilities of MMT drilling, milling and boring type, additional technological fixtures is introduced into their composition [20, 21]. Such quick-change modular units as horizontal (HSH), vertical (VSH) and angular (ASH) spindle heads provide high-performance machining of any complexity parts and are effective in various types of production from single to mass production.

Consider a universal multioperational machine model SF68VF4, the kinematics of which main motion drive is shown in Fig. 1.



a b

Fig. 1. Kinematics of the main movement drive: a – gearboxes with vertical spindle head; b – kinematics of the angular spindle head

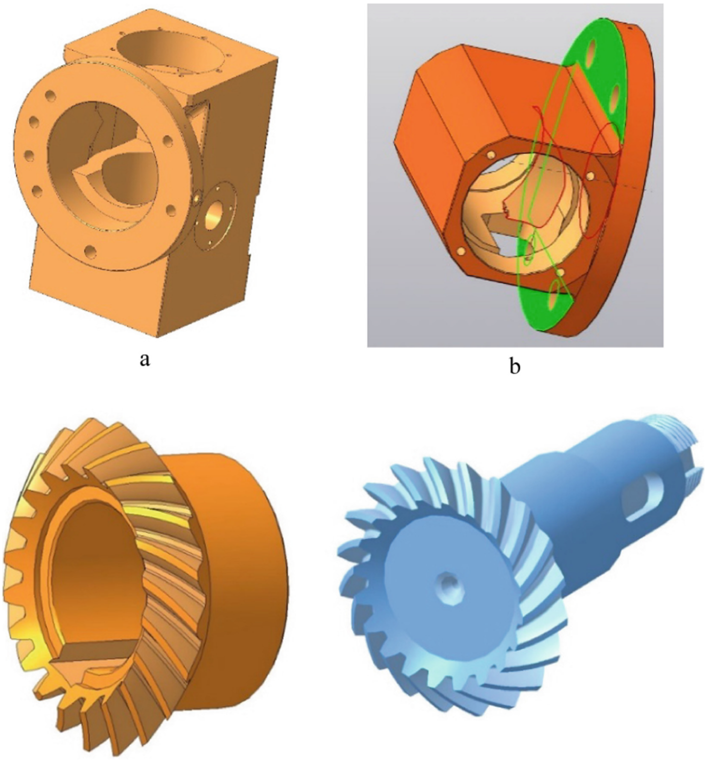
According to the kinematic diagram, the rotation from the electric motor through the poly V-belt is transmitted to the input shaft and then through a two-stage gearbox to the output shaft from which rotation is transmitted to the vertical head- and angular head clutch, or to the horizontal spindle pinion.

In order to carry out a comprehensive procedure for researching the modular tooling of a machine tool and obtain numerical estimates of the stress-strain state, it is necessary at the initial stages to build computational solid models using CAD KOMPAC-3D [18, 22].

Let's use the "bottom-up" principle, i.e. first, we will build 3D-models (Fig. 2) of the constituent parts (housing, shafts, gears) with their subsequent integration into the assembly structure of the angular and vertical spindle heads (Fig. 2). Three-dimensional modeling of the considered parts was carried out in the environment of the integrated CAD using an application of the system "Shafts and mechanical transmissions 3D".

The system software includes: geometric and parametric cores of C3D Modeler and C3D Solver, respectively. The geometric core of C3D Modeler allows you to effectively implement the construction of models of such complex housing parts (Fig. 2, a; b) and calculate the geometric characteristics of the objects being modeled. New functionality introduced in C3D Modeler designed to remove holes and fillets from the model, which are typical for cast iron housing workpieces (Fig. 2, a; b). This leads to a simplification of the 3D model, which is prepared for further calculation in the CAE system [23–25].

It is important to be able to operate not with geometric primitives of a plane or elements of space, but with combined objects and parts in general in the application environment "Shafts and Mechanical Transmissions 3D". The procedure for express-building of high-precision models of bevel gear- and pinions rims with geometrically correct tooth surfaces has been improved [26–28]. This is achieved by using the bevel gear hobbing simulation method and a new type of "Conical surface section". At the same time, in the sense of kinematics, this is realized by moving the curve of the conical



a b c d

Fig. 2. Solid models of machine drive parts: a – VSH housing; b – ASH housing; c – bevel gear; d – bevel pinion shaft

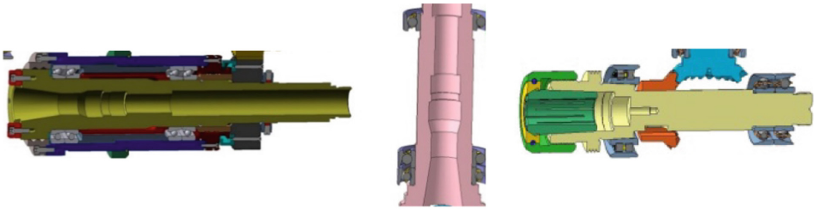
section along two guides with the possibility of changing the parameters of this section. As a result, a very smooth surface is formed along its entire length. In Fig. 2, c; d shows 3D models of a bevel gear and pinion shaft.

The three-dimensional models of the spindle assemblies for the HSH, VSH and ASH developed in the CAD KOMPAS environment are shown in Fig. 3.

Based on the constructed 3D models of the spindle heads (Fig. 3), rendering was performed in the Artisan Rendering module [29, 30], which is integrated into the KOMPAS-3D system.

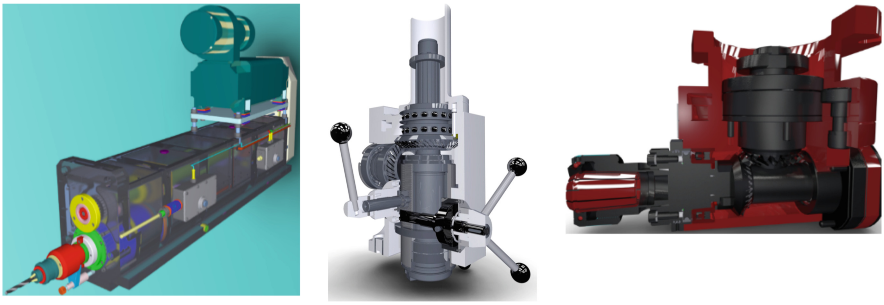
This creation of a photorealistic image and analysis of the appearance of the spindle heads design form an integral idea of the design, while it becomes possible to select materials taking into account the colors and textures. It is important to create subsequent feedback in the course of adjusting the geometry of the product for its improvement. In Fig. 4 shows a rendering of the HSH, VSH and ASH assemblies.

In the considered designs of the spindle heads for a multioperational machine, which is part of the horizontal machining centers group, an important role is played by a gear drive, considered as a gearbox with two bevel gears BG2. At the first and second stages of converting motion from horizontal HSH to vertical – VSH and from vertical



a b c

Fig. 3. Three-dimensional models of spindle assemblies: a – horizontal; b – vertical; c – angular



a b c

Fig. 4. Rendering of spindle heads: a – horizontal; b – vertical; c – angular

to angular ASH) that transform the movement of the machine forming units in the machine coordinate system. The improvement of their designs by technological and design methods is carried out. Equally important are the issues of effective lubrication of bevel gears.

4 Results. Increase in the Resource of the Gear Drive

As you know, the main criterion for evaluating the performance of bevel gears with straight and circular teeth is the characteristic of contact endurance, which is used to calculate the basic parameters – the outer pitch diameters in the first $d_{e2(I)}$ and second $d_{e2(II)}$ stages. Calculations show that the diameters of the driven wheels of BG2 gearbox with both stages can differ significantly in this case. This results in the smaller of these wheels not being directly immersed in the oil reservoir. Accordingly, in the engagement of this stage, the contact conditions of the teeth deteriorate, which ultimately reduces both its resource and the resource of the gearbox as a whole. This problem can be solved by such partitioning of the BG2 gearbox gear ratio U according to its stages I and II, in which the outer pitch diameters of the wheels of both stages – $d_{e2(I)}$ and $d_{e2(II)}$, will be the same, that is, to implement the condition $d_{e2(I)} = d_{e2(II)}$. Consider the problem of partitioning the gear ratio in two versions – wheels with circular and straight teeth.

4.1 Partitioning the Gear Ratio of the BG2 Gearbox with Bevel Gears with Circular Teeth

Parameters $d_{e2(I)}$ and $d_{e2(II)}$ are determined from the main criterion for the performance of bevel gears – contact endurance of the teeth, [31]:

The relationship of the torques $T_3 \approx T_2 \cdot U_{II}$ in the formula for $d_{e2(II)}$ is given without taking into account the efficiency of the bevel gear, and the load factors K_H in the Ist and IInd stages of the BG2 gearbox are taken to be the same.

Coefficients of the influence of the longitudinal shape of the teeth on the contact stresses of the bevel gears circular teeth, [15, 31]:

$$d_{e2(I)} = 1650 \cdot \sqrt[3]{\frac{K_H \cdot T_2 \cdot U_I}{\theta_{H(I)} \cdot [\sigma_H]_I^2}};\ d_{e2(II)} = 1650 \cdot \sqrt[3]{\frac{K_H \cdot T_3 \cdot U_{II}}{\theta_{H(II)} \cdot [\sigma_H]_{II}^2}} = 1650 \cdot \sqrt[3]{\frac{K_H \cdot T_2 \cdot U_{II}^2}{\theta_{H(II)} \cdot [\sigma_H]_{II}^2}}. \tag{1}$$

where the constants $c_{1(i)}$ and $c_{2(i)}$ are taken depending on the hardness of the pinion- and wheel teeth of i-stage.

After substitution $\theta_{H(i)}$ in the given dependencies for $d_{e2(I)}$ and $d_{e2(II)}$ corresponding transformations, the condition $d_{e2(I)} = d_{e2(II)}$ is transformed into a function $U_I = U_I(U)$ of the following form:

$$U_I^3 + a \cdot U_I^2 + b \cdot U_I + c = 0 \tag{2}$$

where $a = \frac{c_{2(II)} \cdot U}{c_{1(II)}}$; $b = -\frac{U^2 \cdot c_{2(I)} \cdot [\sigma_H]_I^2}{c_{1(II)} \cdot [\sigma_H]_{II}^2}$; $c = -\frac{U^2 \cdot c_{1(I)} \cdot [\sigma_H]_I^2}{c_{1(II)} \cdot [\sigma_H]_{II}^2}$.

It should be noted that during operation, the forces and contact stresses in the engagement are changed, which leads to a certain level of error in the center distance. This phenomenon is associated with changes in the distance between gearbox shafts. An interesting approach was proposed in [32] to estimate such an error. Based on the analysis of the change in the distance between the shafts of two gears, a procedure for determining the influence of various geometric errors of the gears. This task is accomplished by evaluating the sinusoidal component of the variation curve, obtained by means of the Fourier transform was proposed.

The solution of the cubic Eq. (2) is performed for 3 variants of the combination of the teeth hardness of the Ist and IInd stages for the BG2 gearbox:

a) both stages are "soft", $\{\overline{H}_I < 350HB,\ \ \overline{H}_{II} < 350HB\}$: $c_{1(I)} = c_{1(II)} = 1.22$; $c_{2(I)} = c_{2(II)} = 0.21$; $[\sigma_H]_I = [\sigma_H]_{II} = 507\,MPa$;
b) -Ist stage "soft", $\{\overline{H}_I < 350HB\}$; IInd stage "hard", $\{\overline{H}_{II} > 350HB\}$: $c_{1(I)} = 1.22$; $c_{2(I)} = 0.21$; $c_{1(II)} = 0.81$; $c_{2(II)} = 0.15$; $[\sigma_H]_I = 507\,MPa$; $[\sigma_H]_{II} = 814\,MPa$;
c) both stages are "solid", $\{\overline{H}_I > 350HB,\ \ \overline{H}_{II} > 350HB\}$; $c_{1(I)} = c_{1(II)} = 0.81$; $c_{2(I)} = c_{2(II)} = 0.15$; $[\sigma_H]_I = [\sigma_H]_{II} = 814\,MPa$,

here $\overline{H}_I$ and $\overline{H}_{II}$ are the average hardness of the teeth at the Ist and IInd stages.

For the convenience of using dependence (1) in computational practice, we approximate its power function $U_I = k \cdot U^x$: option a): $U_I \approx 0.87 \cdot U^{0.78}$; option b): $U_I \approx 0.7 \cdot U^{0.76}$; option c): $U_I \approx 0.86 \cdot U^{0.79}$ (approximation error less than 0.5%).

For the found value U_I the amount U_{II} is found by the usual method: $U_{II} = U/U_I$.

For clarity of the analysis of functions $U_I = U_I(U)$, the calculations according to (1) and (2) were performed with some excess of the maximum recommended value of the gear ratio U_I in standard bevel gears [5] (in Fig. 5 the limiting value U_I is marked by the size $U_{i(\max)} = 6.3$).

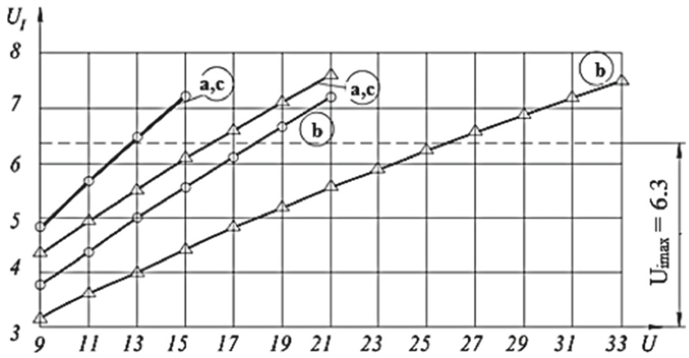


Fig. 5. Graphs of the function $U_I = U_I(U)$: lines with symbols ○ – for circular teeth, formula (2); with symbols Δ – for straight teeth, formula (3)

4.2 Partitioning the Gear Ratio of the BG2 Gearbox with Bevel Gears with Straight Teeth

In spur bevel gears, the coefficients of influence $\theta_{H(I)} = \theta_{H(II)} = 0.85$, [15, 31]. As a result, the function $U_I = U_I(U)$ for partitioning U into stages of the BG2 gearbox from the condition $d_{e2(I)} = d_{e2(II)}$ takes on a simpler form:

$$U_I = k \cdot \sqrt[3]{U^2} \tag{3}$$

where $k = \sqrt[3]{[\sigma_H]_I^2/[\sigma_H]_{II}^2}$.

Graphic functions (1) and (2) are shown in Fig. 5 for options a), b), c) in relation to bevel gears with circular and straight teeth.

Based on the calculations performed, we will analyze the features of the procedure for partitioning the overall gear ratio.

1) For the combination $\{\overline{H}_I < 350HB; \overline{H}_{II} > 350HB\}$ – these are lines b) for circular and straight teeth in Fig. 5. Values U_I and U_{II}, which do not exceed the maximum recommended value $U_{i(\max)} = 6.3$, are obtained for spur gears at gear ratios of the gearbox $U \le (25 \div 26)$, and for circular teeth at $U \le 18$. Hence, the conclusion is that a partitioning U by stages of the BG2 gearbox from the condition $d_{e2(I)} = d_{e2(II)}$ for this combination of hardness of the teeth of stages I and II is possible, but for straight teeth it is feasible over a larger range of values U.
2) The smallest range $U : U \le 13$, at which it is possible to partition it into steps from the condition $d_{e2(I)} = d_{e2(II)}$ with observance of the limitation $U_i \le U_{i(\max)} =$

6.3, is given by gears with circular teeth at $\{\overline{H}_I < 350HB;\ \overline{H}_{II} < 350HB\}$ and $\{\overline{H}_I > 350HB;\ \overline{H}_{II} > 350HB\}$ – the line a) and c) with symbols "○" in Fig. 5. The values U_I and U_{II} for these combinations of hardness practically coincide, therefore they are shown by one line.

3) A conclusion similar to 2) can be made in relation to the same combination of hardness for straight teeth, line a), c) with the symbols "Δ" in Fig. 5. The difference is only in a slightly larger maximum value $U : U \le 15$.

The partitioning of the gear ratio by stages of the BG2 gearbox from the condition $d_{e2(I)} = d_{e2(II)}$ can be recommended primarily for long-term operating modes, when abundant lubrication of the gearing is one of the most important factors that positively affects the transmission resource of power gearboxes. At the same time, the greatest possibilities of using this method of partition into stages of the BG2 gearbox, which follows from Fig. 5, will be for spur gears with a variant $\{\overline{H}_I < 350HB;\ \overline{H}_{II} > 350HB\}$, where the implementation of the partitioning condition $d_{e2(I)} = d_{e2(II)}$ is possible for U reaching the value $25 \div 26$.

5 Conclusions

In this work, the following results are obtained:

1. The complex of 3D models of horizontal, vertical and angular spindle heads for multioperational drilling, milling and boring machine with a two-stage gear drive in the CAD KOMPAS-3D environment has been developed. This 3D project became the winner of the International Competition "Future Aces of Computer 3D Modeling.
2. Created three-dimensional models of housing parts of machine tool spindle heads with complex spatial shape. The strategy of creating solid models based on an extended range of geometric primitives of the KOMPAS system in the form of combined functional elements and parts as a whole has been effectively used. This makes it possible to increase the productivity of the designer at the stage of creating a 3D project for the designed product.
3. In the specialized application "Shafts and mechanical transmissions 3D", three-dimensional models of bevel gears with circular teeth are created. In the process of creation, a unique procedure for constructing gear rims was used by the imitation method of gear milling for bevel gears and a new functionality of the geometric core module "Conical section surface".
4. In the Artisan Rendering module, the machine tool heads are rendering using high-quality hardware OpenGL, the advantages of which are the simplicity and speed of installing a complete scene (Snapshot) and the ability to view and generate several Snapshots for software rendering. This realistic image of the projected spindle heads allows you to get a more complete view of the designs and outline ways to improve them.
5. A search procedure for the design of a two-stage gear drive with an increased parameter of teeth contact endurance due to improved lubrication of the main drive elements has been developed. For this purpose, analytical dependences are proposed for partitioning the gear ratio of a two-stage drive according to its stages I and II, at which

the outer pitch diameters of the wheels of both stages will be the same. A derivative analytical dependence $U_I = k \cdot U^x$ of the first stage gear ratio is introduced for three variants of the gear materials hardness, which makes it possible to effectively solve the basic cubic equation of the first stage gear ratio. The approximation error was less than 0.5%. The results obtained are recommended primarily for long-term operating conditions, when abundant lubrication of the gearing is one of the most important factors that positively affect the gear resource of the drive mechanisms for spindle heads of various designs.

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