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SELECTION OF IMPROVED DESIGN FOR MACHINE CENTRE TOOL SPINDLE SUPPORT

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Abstract This paper presents a comprehensive design for the main motion drive (MMD) of a multi-operation drilling-milling-boring machine with a modified spindle support. A two-stage drive design with an increased speed range is proposed. A procedure for parametric modeling of the transverse drive layout in a two-criteria formulation is considered. Preliminary design research of a form-generating two-support spindle unit with a reinforced front support mounted according to the "Tandem-X" scheme is carried out. A verification calculation of ball angular-contact bearing (ACB) is implemented in a specialized APM Bear module, based on the general theory of non-ideal surface contact. An analysis of the ACB design parameters is conducted for a typical technological operation: climb milling with an end-mill cutter. A 3D design of a two-stage MMD with a two-support spindle unit is presented in the integrated Creo Parametric CAD environment. Creo tolerance design toolings with stackup-procedure are used to prepare detailed drawing documentation. For photorealistic representation, assembly design rendering options for housing-shaft-bearing and element-by-element exploded views of drive components are proposed. An alternative front spindle support design was developed based on an innovative tapered angular-contact roller bearing. The novelty of the proposed design solution lies in the modified contact geometry between the rolling element and the raceway, aimed at increasing the load-carrying capacity and service life of the spindle-support assembly. Design calculations in the APM Bear module demonstrated a significant increase in durability and load-carrying capacity. The research resulted an alternative competitive design for a two-stage machine tool drive. is proposed.

Keywords: Multioperational machine tool, Gear drive, Spindle unit, 3D modelling, Tapered Bearing, Durability, Calculation module.

1. Introduction

In the main drives of machining centers (MC), a well-known variant with a DC motor and a thyristor strain converter is predominantly used [1, 2]. The required power of the main drive of the machine changes as a function of the spindle speed. In this case, the rated power is not used at high and low speeds. The useful power of the drive, obtained by research, up to approximately 1/3 or even 1/2 of the control range, increases proportionally to the speed and regulation in this part of the range must be carried out with a constant torque. Further, the drive power reaches its maximum and then slightly decreases at the highest speed. In this section, the drive can be regulated with a constant maximum permissible power. Thus, the main drive of the machine requires two-zone regulation, which is provided in different ways [3].

Regulation with a constant maximum permissible power is carried out by changing the excitation current with a constant strain on the armature. In this case, the speed varies upwards from the rated one in a small range, which is determined by the switching capabilities of DC motors. For DC motors of the 2P series, this range $D_r = {}_{\text{const}} = 1:4$. In the case where the technological requirements dictate the range D_r, it is necessary to introduce an additional gearbox. To MMD regulate with a constant maximum permissible torque, the adjustment range must be. However, this range can be increased. This is due to the need for precise positioning of the machine spindle when changing tools [4]. With such a wide range of power characteristic adjustment,

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along with vibration resistance issues, there arise problems of reducing constant and peak loads on the main components of the main drive and the spindle unit of the machine, including its shafts and bearings.

The MC drive efficiency design directly depends on the design solutions for the methods and means of fastening its main components. For the MMD spindle assembly, a reliable approach to the design and modeling of shaft and spindle supports ensures increased durability. In this regard, various aspects of the selection of bearing type and series, their mounted patterns, and the presence and magnitude of preload are important [5]. The key concepts in creating durable machine designs related around developing a preliminary and 3D drive design in modern CAD systems [6, 7, 8]. When selecting a CAD system, it's important to consider the ability to use specialized modules for calculating and modeling bearing assemblies, shafts, and spindle assemblies. Furthermore, multi-criteria search for new design solutions in the area of rolling bearing modernization are relevant. This primarily concerns the forming MC spindle assemblies. This work aims to improve the design of the MC main drive in a wide range of control using innovative solutions in the process of modeling bearing supports.

2. Literature Review

In the field of metal-cutting equipment research, there are many works in the direction of designing and modeling drive devices from the standpoint of reliability criteria and reducing loads on forming units. [9–11]. For the entire variety of machines in a given group (type), it's impossible to use just one or two standard MD designs. Most often, it's necessary to either develop a completely new design using structural optimization methods or create an alternative version of an existing prototype design using parametric optimization. In general-purpose MC for complex machining of workpieces (taking into account preliminary shaping methods), various methods of implementing the main drive are used. The traditional method for complex machining is implemented by a MD design that utilizes a machine tool gearbox with sliding gears and their blocks [12]. The widespread use of adjustable drives with continuously variable spindle speed and gearbox speeds ensures a wide control range. This is achieved through a reduction feed rate to achieve lower speed ranges and accelerator gears to achieve higher speed ranges. At the same time, it is necessary to consider that one of the ways to MD improve is to simplify gearboxes as much as possible. Two-stage gearboxes are the most promising in this regard.

The work [13] is devoted to a theoretical study of single-stage and multi-stage gearboxes from the perspective of various parameters. At the preliminary design stage, gear ratios are determined, based on which various parameters for gear and shaft diameters can be established. In the next step, the type and size of bearings are selected to ensure proper operation of the gearbox. At the final stage, the number of teeth for gears and pinion-shafts, as well as the dimensions of shafts and keys, are calculated. At the stage of creating a 3D model of the gearbox, the authors [13] used SolidWorks CAD for correct visualization and orientation. Furthermore, at this stage, virtual assembly is performed, and tolerances for the selected bearings and keys are determined. Calculations for optimizing the housing shape and reducing weight without compromising the design safety factor based on the 3D model are performed in the ANSYS software environment.

The forces and moments generated during the transmission of motion from the electric motor through the drive to the machine spindle, along with the kinematic characteristics, affect its overall dimensions. In [14], these force parameters serve as technical constraints in the form of engineering solutions for optimizing the overall dimensions of the gearbox. A similar influence of cutting forces on deformation and accuracy of shaft-type components was reported in [15], where a steady rest design was proposed to compensate elastic deflections during crankshaft machining. In addition, the role of elastic деformations and dynamic effects becomes especially significant for low-stiffness elements, such as thin-walled parts, where vibration behaviour and resonance conditions determine machining accuracy, as demonstrated in [16]. When calculating the loads acting on the shafts and bearings, the choice of process parameters and coefficients in the analytical dependencies complies with the DIN 3990 standard. To solve the optimization problem (the objective function is the internal volume of the gearbox), the authors [14] developed the corresponding Cambrian software. The result of the optimization procedure is a reduction in the gearbox volume by 18%.

As the demand for complex turning and milling machining centers has grown. New requirements have been put forward for the adjustable speed range of vertical lathe gearboxes: from two gears to three gears, i.e., high-speed, low-speed, and neutral [17]. The main drive system of a vertical turning-milling machining center is driven by a DC motor through a multi-stage gearbox to achieve the required table rotation speed. In addition, the authors note intra-node effects, when the loading mode of the drives along the axes of a five-axis turning-milling machine affects the elements of the carrier system [17]. To compensate for the vibration effects from the drives, the machine structure includes additional slides in the form of roller unloading devices, special clamping devices, and other components that complicate the machine structure.

The MDM final element of the spindle-bearing system of the above-mentioned machines is the spindle assembly, which is supported in most cases by rolling bearings. The need for high manufacturing precision, the challenges of creating preload in duplex and triplex bearings, and preload changes make research into various types of spindle bearings relevant [18]. At the same time, issues related to the handling and positioning of shaft-type components are also addressed in robotics, where the design and kinematics of grippers significantly affect manipulation accuracy and reliability [19]. In [20], an analytical study of various scientific papers on the design and modeling of the spindle-bearing mechanical system for turning and milling machines is conducted. In most cases, spindles are mounted on angular contact ball bearings due to their favourable rigidity characteristics, low friction losses, long service life, and high price-performance ratio.

The work [21] is devoted to the development of a model of tapered roller bearings (TRB) taking into account a contact mode for researching operational defects. This model is used to estimate contact loads and the corresponding distribution of contact pressure between the rollers and raceways. Characteristic features of the TRB design are a large peak contact pressure at the edge of the raceways and an uneven distribution of contact pressure between the roller and raceways. This effect results in a reduction in the TRB friction torque, taking into account the external load and rotational speed.

The TRB developed models at high rotation speeds are characterized by a more accurate determination of influencing parameters, such as cage and ball rotation speeds, gear ratios, and raceway and ball diameters. These same parameters influence the magnitude of internal loads, friction torque, and service life. This approach is presented in [22], where the model allows for more accurate determination of preload characteristics and assessment of bearing rigidity and service life.

A design improvement to a single-row tapered roller bearing by modifying the geometry of the rolling bodies and raceways, resulting in reduced sensitivity to ring misalignment, is presented in [23]. An analytical basis for designing a modified single-row tapered roller bearing while maintaining the primary advantage of the standard roller bearing – high load capacity has been developed.

An analysis of the conducted research has shown that a significant number of works are devoted to improving the quality of bearing supports in the direction of modeling various versions of spindle bearings, improving designs and using effective automated design and engineering analysis systems.

This work sets the task of developing an improved two-stage MDM based on the search for a new design of spindle support and its complex modeling in the environment of a specialized module.

3. Research Methodology

3.1. Sketch Design of a Machining Center Drive with a Horizontal Spindle Head

This paper examines a multi-operation drilling-milling-boring machine, model MC 4V, which can be used both for stand-alone operation and for integration into a flexible manufacturing system capable of retooling for a specified range of parts [24]. The machine is designed to cut complex housing parts up to 600 mm in size, made of special steels, light alloys, and other difficult-to-machine materials. The main machine components are mounted on a rigid base, enabling foundation-free operation (Fig. 1a). The rotary working table and the pallet table changer construction are also mounted on it. A worm drive is used to implement the rotary motion [25].

The ability to automatically change the workpiece and swap the cutting tool, using optimal machining modes, determines economic efficiency throughout the life cycle of the manufactured product. A detailed analysis of this phenomenon is presented in [26, 27], which examines the processштп on CNC machines from a systems perspective. These studies demonstrate the impact of reducing machining cycle time (by approximately 18%) as a result of using optimal cutting conditions and reducing auxiliary time (including automatic tool change).

The frontal cantilever-less spindle head is mountedon a portal-type stand and has cross-displacement along the X- and Y-axes. (Fig. 1, a).

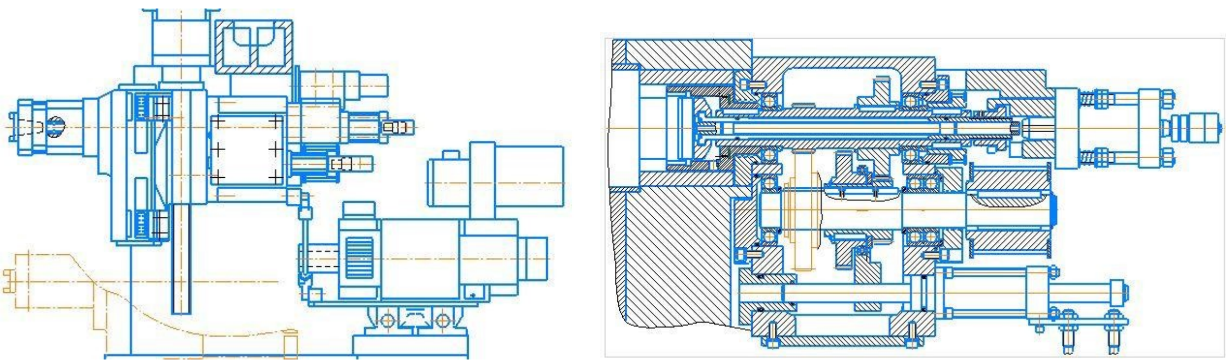


Figure 1: Layout of the machine MC 4V drive: a) general view; b) longitudinal drive layout

In the MC main drives, adjustable speed boxes with a DC motor and thyristor voltage conversion are preferably used. The need to regulate the rotational speed at a constant torque in one part of the variable speed range, and with a constant maximum permissible power in the other, leads to a two-zone regulation [28].

Since the rotational speed varies upwards from the nominal in a small range determined by the switching capabilities of DC motors $(D_r = \text{const} = 1:4)$, it becomes necessary (according to technological requirements) to increase the D_r range. Increasing the range leads to the feasibility of using a two-stage drive device [29]. It is also necessary to take into account that for a wide range of machine tools, structurally proven main units (assemblies) built on the modular principle are offered. The presence of a fixed nomenclature for assembly units in multi-operation machine tools makes the aggregate-modular principle promising; the methodological aspects of which are presented in the work [30].

Motion in the main drive (Fig. 1, b) is transmitted from the electric motor via a poly V-belt to the gearbox input shaft, then via a spur gear to the output shaft. When the movable block gear engages, the other gear must completely disengage from its mating gear. This gear, in turn, transmits motion to the machine spindle via a toothed clutch (specific characteristics of the half-clutch). The speed and position of the spindle are controlled using a feedback sensor - the ROD 700 encoder from HEIDENHAIN, which is based on an optoelectronic reading method.

The efficiency of the main drive depends on the adopted transverse arrangement, including the position of the output shaft. When determining the spatial position of gears transmitting torque to the spindle, two mutually exclusive criteria must be considered: maximum rigidity and minimum reduced load on the spindle's front support.

The spatial layout construction procedure can be improved by using the parameterization toolkit with the APM Graph module [31]. Fig. 2 shows a diagram for determining the spatial position of the MMD shafts in the MC 4V machine tool (Fig. 2, a) and the optimal transverse layout option based on the rigidity criterion (Fig. 2, b). To evaluate the spatial position of the MMD output shaft, we will consider the typical technological operation "Climb milling" using end mill (type 1 with a normal tooth) with a diameter of $d_{a0} = 20$ mm and the number of teeth $z_0 = 6$ with cutting part material T15K6.

It has been experimentally established that the magnitude of the loading for the above technological operation from the cutting forces (projections of the tangential and radial components of the resulting cutting force) is:

$$P_{sx} = 1515 \text{ N } (P_{sx} \approx 0.8 \cdot P);\ P_{sy} = 1316 \text{ N } \left(P_{sy} \approx 0.7 \cdot P\right).$$

With such a balance of forces P_{sx} and P_{sy} the angle of inclination of the resulting force P_s will be about $\alpha \approx 40^0$. This value will be taken into account when finding the position of the output shaft in the MC4V gearbox (Fig. 2, b).

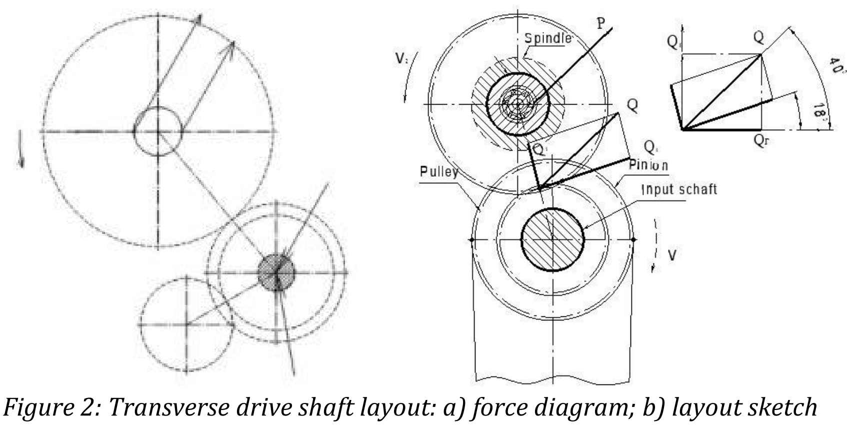


Figure 2: Transverse drive shaft layout: a) force diagram; b) layout sketch

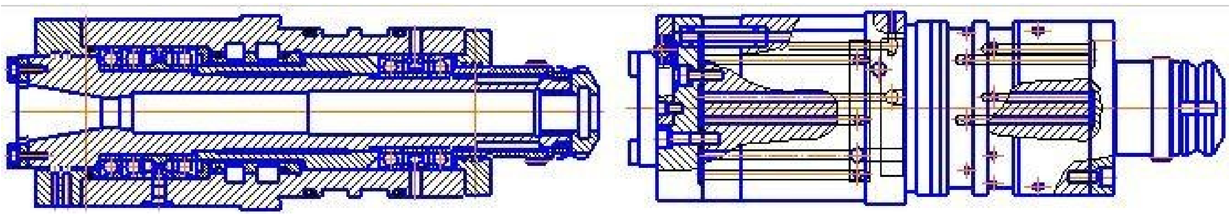


Figure 3: MC spindle unit

Torque transmission from the spindle sleeve (Fig. 1, b) via a gear coupling relieves the spindle from radial forces and vibrations. The spindle assembly's housing facilitates disassembly and assembly, adjusting its supports and carrier system components [32], and improves heat transfer, thereby reducing thermal distortion.

In the milling operation discussed above, radial forces significantly exceed axial ones. This necessitates the use of angular-contact bearings, particularly on the spindle's front support.

During milling with a multiedge cutter, dynamic impact loads occur, especially during interrupted cutting. This places a premium on spindle bearings for increased stability. Duplexed angular-contact ball bearings with a tandem (DT-pattern) mounting arrangement best meet this requirement. Analysis of the system of forces acting on the workpiece and tool revealed bidirectional axial loading. However, the DT mounting arrangement supports axial forces in one direction. For such loading situations, an X-arrangement mounting is used. Therefore, a Tandem-X (TFT) mounting arrangement is recommended for the front support.

The basic design implementation of the machine tool spindle node is a two-support structure mounted on two rolling bearings (Fig. 3).

Also, to eliminate internal clearance and increase the rigidity of duplexed-contact bearings, spacer bushings of the required thickness are used between the rings of two ball bearings, followed by tightening the outer (or inner) rings until the gap between them disappears.

3.2. Calculation of Angular-Contact Bearings in the APM Bear Module

For comprehensive refined calculations of spindle bearing supports, the specialized APM Bear module is used. This module is a system for calculating the parameters of non-ideal rolling bearings. The term "non-ideal" refers to a bearing whose geometric dimensions deviate from the nominal values. This refers to the need to account for the shape errors of rings, raceways, and rolling elements, the contact surfaces of which always exhibit shape errors. Moreover, the amplitude of these errors is comparable to the magnitude of the contact displacements.

Such refined calculations are based on the general theory of contact of non-ideal surfaces [33]. A characteristic feature of such calculations is the fact that the actual contact between the rolling elements and raceways is statistical in nature.

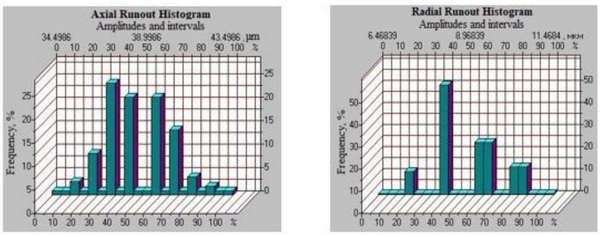
Fundamental calculation parameters such as stiffness and displacement can only be determined statistically – as a sample realization, average value, variance, etc. A similar approach is used in modeling and calculating bearings in other industries [34], which increases the reliability of the obtained results.

In APM Bear a sample realization of the bearing's contact displacements, consisting of 100 elements, is calculated. Using this model, it is possible to determine the average values of displacements and stiffness, their variances, maximum, minimum, and most frequent values, the shape of the scatter range, etc.

In other words, the parameters calculated using contact displacements (frictional torque, power losses, loads acting on the rolling elements) are also represented as arrays of 100 elements (with corresponding statistical characteristics). These results can be displayed in various ways – as a table, histogram, graph, diagram, and even animation. This gives you a realistic picture of the bearing's behavior in all its complexity and completeness.

Fig. 4 shows the results of a complex calculation in the APM Bear module for the angular-contact ball bearing 2-46113 (SKF 7013 AC, contact angle $\alpha = 26^0$) mounted on the front support of the MC spindle unit.

Average durability, hours, L_{10}, hr	8538.254
Max. cont. stress, σ_H, MPa	2108.737
Axial runout, μm	38.689
Radial runout, μm	8.868
Lateral beats μm	-0.040
Friction moment, N x m	0.209



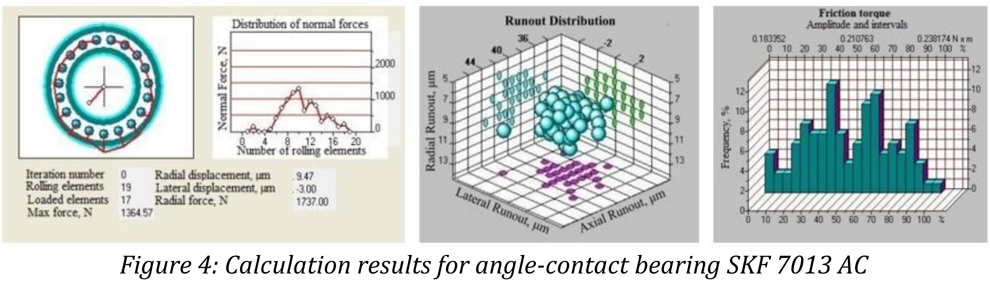


Figure 4: Calculation results for angle-contact bearing SKF 7013 AC

Let us analyze the results obtained under the given loading that occurs during the typical technological operation discussed above: climb milling with an end mill cutter.

Maximum contact stresses $\sigma_H = 2108.737\ \mathrm{N/mm^2}$ is within the acceptable limit $[\sigma_H] = 4200\ \mathrm{MPa}$ for angular-contact ball bearings according to ISO standards.

An analysis of the required precision parameters of angular-contact rolling bearings according to ISO and SKF standards shows compliance with the radial runout indicators (for outer rings) $\Delta_r = 8.868 < [\Delta_r] = 11.0\ \mu\mathrm{m}$. According to the axial runout parameter value, the bearing in question does not comply with the standards: $\Delta_a = 10\ \mu\mathrm{m}$.

Durability of one radial thrust bearing $L_{10} = 8538.254$ hr significantly lower than the permissible values, around 30000–40000 hours. Moreover, the estimated service life of SKF bearings should be 100000 hours or more.

To increase the durability rating, we will consider the option of implementing duplexed angular-contact ball bearings on the front spindle support. In accordance with standard 832-2022, the dynamic load rating for this option increases by 1.62 times. At the same time, the durability at a given load rating accordance with the dependence $\left(L_{10} = \frac{C}{P}\right)^3 \cdot 10^6$ increases to 36300 hours. However, this support option is also insufficient in terms of durability. Perhaps, as an alternative, a different bearing type should be considered – for example, a roller bearing.

3.3. Three-Dimensional Modeling of the Machining Center Drive

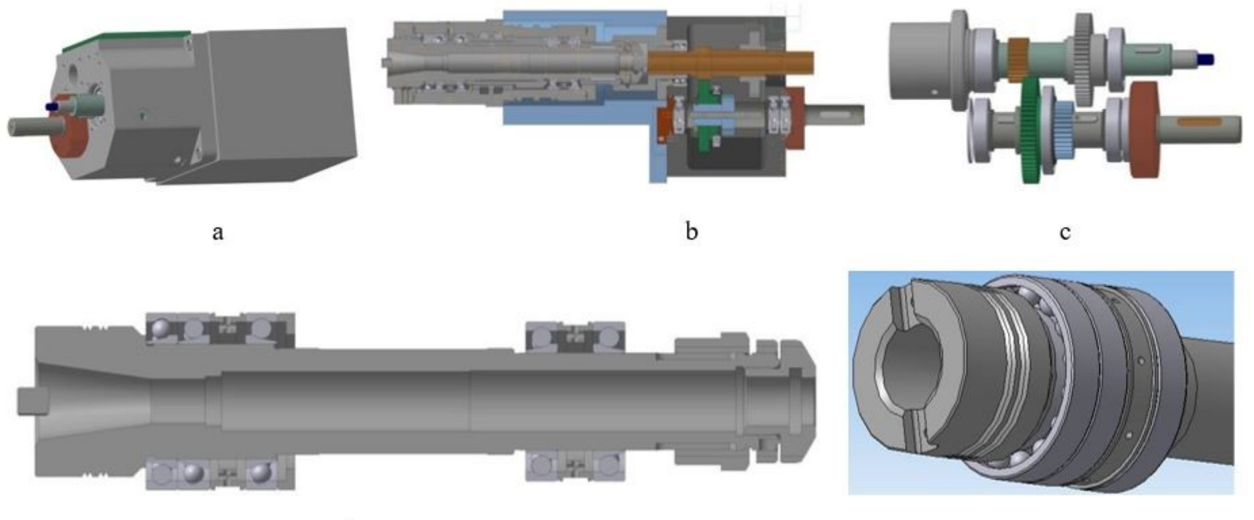
Research of various technical objects to assess the stress-strain state, vibration and precision characteristics is carried out using 3D models developed in modern CAD systems [35, 36].

To study the MMD design features, refine its kinematic and layout parameters, and evaluate its technical and economic performance, we will perform a 3D modeling procedure. Solid models of the drive's main components will be constructed using the integrated CAD Creo Parametric environment [37]. Figure 5 shows a 3D design of a machining center drive with a two-stage gearbox and a two-support spindle with a horizontal axis.

The Creo system emphasizes that a 3D CAD model is an ideal theoretical representation of a product. To ensure that the design meets actual technical requirements during assembly, Creo's tolerance design tools are effectively used. A typical assembly for the Drive product is the "shaft (spindle) – bearings" assembly unit shown in Fig. 5, d. The assembly operation is based on the Creo superposition procedure. The preceding step is to define the tolerance on the mating surfaces. To determine the maximum allowable tolerance value, it is necessary to take into account the accumulated variation of individual dimensions. The combination of dimensional variations results in a critical distance deviation in the final product, typically between two parts of an assembly.

To implement the tolerance setting operation, the EZ (e7) *Tolerance* module is used. The shaft tolerance value is specified in the *Options* dialog box. The bearing fit dimensions on the shaft are calculated using tolerances according to ISO 286 Standard. A typical shaft tolerance value is $10\ \mu\mathrm{m}$. In practice, this assignment is associated with the desire to optimize drive performance. The primary alternative for this assignment is a worst-case analysis (description of the worst-case scenario).

Setting a tolerance allows you to automatically generate a bearing stackup on a shaft.



a b c d e

Figure 5: Main drive of the machine: a) general view; b) section of the drive with spindle; c) kinematics; d) spindle; e) front support

To do this, the EZ Tolerance section will analyze the clearance between the Shaft part and the BEARING part. In the *New Stackup* dialog box for this stackup, select the BEARING flat surface closest to the shaft. Similarly, select the shaft flat surface closest to the bearing and select the *Asm_Right* reference plane. The stackup size is entered in the same graphics window. In the presented algorithm, the assembly constraint for this stackup is selected in the *Assembly* area of the above-mentioned dialogue box. To further design the drive, the Creo procedure *Adding Features* is used. For example, when designing the front spindle support, spacer rings are selected as a new feature to create a preload in the front support. To do this, in the EZ Tolerance section, use the *Add Feature* command and select a transition surface (on the inner ring of the *Bearing 1* part). The transition surface on the opposite side of the *Bearing* 2 part is selected similarly. As a result of the above actions, a structurally defined 3D unit of the horizontal spindle front support is formed in the Creo Parametric environment (Fig. 5, e).

3.4. Rendering of the Machining Center Drive

In Creo [37], a photorealistic representation of a machining center drive model (Fig. 5) is implemented in the specialized *Rendering Studio* section. This clearer representation of the design allows customers to focus on the drive's design features and enhances the product's competitiveness. In addition to rendering, Creo equipped an Explosion feature, which evaluates the product structure, analyzes the drive's assembly manufacturability, and then develops a corresponding technological process. Fig. 6 shows a rendering and exploded view of a MC drive unit. The 3D modeling of the machining center drive formed the basis for the technical design, development of working drawings and specifications, and refinement of the design and geometric parameters of the gearbox main components and dual-support spindle.

The modeling process identified issues related to optimal layout and kinematics, as well as the specifics of selecting and calculating shaft and spindle supports. This particularly applies to the front spindle support, the initial design of which is based on a triplex mounting arrangement. The main challenges of this type of support include the complexity of the multi-row design and preload adjustment. 1…4 μm reduced damping. Furthermore, the technological capabilities of bearing assemblies, especially under heavy loads, are limited by insufficient speed under tension, as required for roughing operations. In other cases, they are limited by insufficient load capacity under the tension required for high-speed machining.

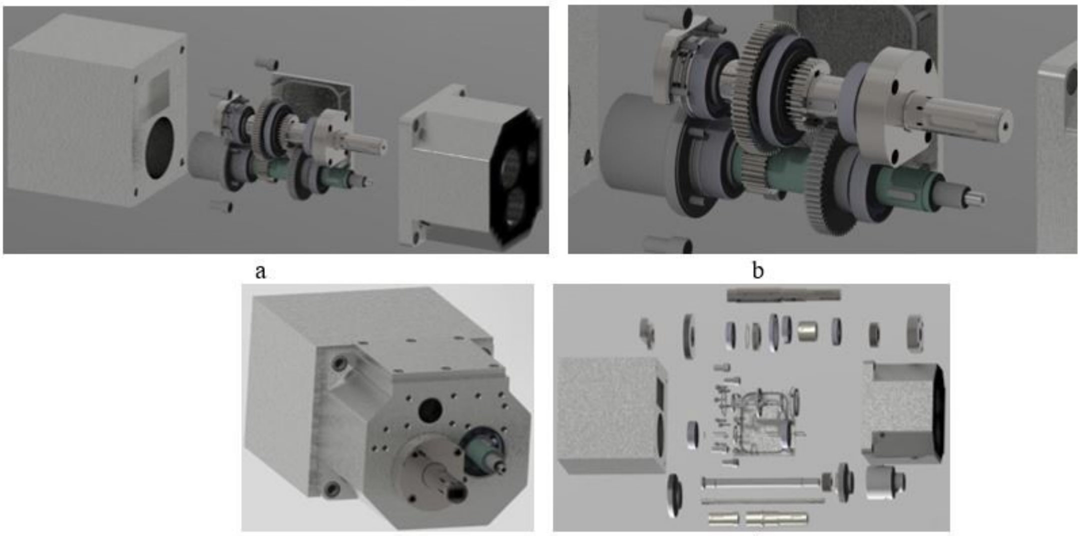
The design alternative of spindle assemblies on tapered roller bearings is increasingly being considered as a promising option for supporting the design.

4. A Modified Version of the Front Spindle Support Based on the Innovative Design of a Single Row Tapered Roller Bearing

4.1. Innovative Component

The shaft and spindle supports use tapered bearings containing rolling elements by means of cut cones with rectilinear generatrices in accordance with SKF 32013 Bearing.

A disadvantage of the known tapered bearing is its increased sensitivity to ring misalignment caused by elastic deformation of the shafts, which reduces the bearing's load-carrying capacity and service life [5]. This paper aims to improve the tapered bearing in terms of load-carrying capacity and service life.



a b c d

Figure 6: Photorealistic capabilities of Creo: a); b); c) – rendering; d – explode

The basic idea (supported by a patent [38]) consists of giving the rollers and raceways a curved shape on the inner and outer rings. This reduces the sensitivity to ring misalignment, resulting in

The stated objective is achieved by the fact that in a single row tapered roller bearing (according to the proposed concept), the generatrices of the tapered rollers are concave, while the raceways of the inner and outer rings are convex (Fig. 7). Moreover, the concave generatrices of the tapered rollers and the convex raceways are outlined by equal arcs of circles of radius R_K. This will ensure, firstly, linear contact of the rollers with the rolling raceways, and secondly, it will reduce the sensitivity of bearing to the misalignment of the inner ring relative to the outer ring. In this case the full length of contact between the rollers and the rolling raceways of the rings along the common lines is maintained.

Based on the proposed idea, a design of a single row tapered roller bearing Fig. 7) has been developed. In the proposed design, tapered rollers 1, outlined by a concave generatrix, contact with the convex rolling raceways of the inner ring 2 and outer ring 3. increased load-carrying capacity and durability. Furthermore, the single-row tapered bearing can be mounted on shafts with low bending rigidity.

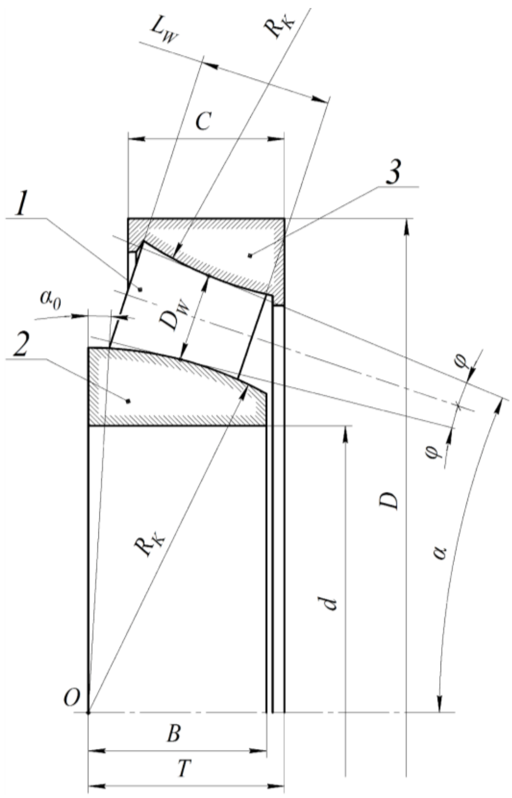


Figure 7: Innovative version of tapered roller bearing

Angle of inclination of the tangent α to the midpoint of the circular arc of the concave rolling raceway of the outer ring 3, is selected within the limits provided for the prototype: $\alpha = 11^0 \ldots 16^0$. Radius of curvature R_K is determined from the equation (1):

$$R_K = 0.25 \cdot (d + D) / \cos(\alpha - \varphi) - 0.5 \cdot D_w, \quad (1)$$

where d and D– inner and outer diameters of a tapered roller bearing (coincides with the values d and D known single row tapered roller bearing); the angle of inclination of the generatrices of the tapered rollers φ selected from the standard range: [5]. Parameters of conical rollers 1 with concave generatrices – diameter D_w and length L_w are determined from the relations (2):

$$D_w \approx 0.25 \cdot (D - d) \cdot \cos(\alpha - \varphi);$$

$$L_w \approx 2 \cdot R_K \cdot \sin(\alpha - \alpha_0 - 2 \cdot \varphi) \cdot \cos\varphi. \quad (2)$$

Corner $\alpha_0 = 3^0$ provides an increased length of the rolling raceway on the inner ring 2 compared to the length of the generating tapered rollers, due to which the skewness of the inner ring 2 relative to the outer ring 3 is within the value of the angle that does not affect the performance of the tapered roller bearing. The location of the center of the radius of curvature R_K on the wide end of the inner ring 2 allows for the assembly of a tapered bearing, as well as for adjusting the clearances in it due to the axial movement of the inner ring 2.

Rollers 1, inserted into bearing cage (not shown in Fig. 7), together with outer ring 3, which has shoulders on its ends, form a one-piece structure. Inner ring 2 can be freely removed from the roller bearing in the axial direction.

A single-row tapered roller bearing operates as follows (Fig. 7). Inner ring 2, pressed onto the shaft, rotates with the shaft as a single unit, while outer ring 3, secured in the housing, remains stationary. Between rings 2 and 3 are tapered rollers 1 with concave generatrices. When inner ring 2 is skewed relative to outer ring 3, the contact between rotating rollers 1 and rings 2 and 3 remains linear along the entire length of the generatrix of the rollers 1, since the contact line between them is a common arc of a circle radius R_K.

Misalignment of rings 2 and 3 will cause rollers 1 to move along arced contact lines relative to outer ring 3 without causing edge contact. Furthermore, the longitudinally convex-concave contact between rollers 1 and rings 2 and 3, all other things being equal, will yield lower contact stresses than in a conventional single-row tapered roller bearing, thereby increasing load-carrying capacity and extending service life.

4.2. Calculation of a Single row Tapered Roller Bearing in the APM Bear Module

Using APM Bear [31] it is possible to calculate the most commonly used types of roller bearings: a) radial for operation under radial load conditions; b) spherical with two rows of rolling elements for operation under radial load conditions; c) angle-contact, operating under combined load conditions – axial and radial force; d) thrust bearing, operating under conditions of exclusively axial load.

The module uses databases for standard bearings, the parameters of which can be entered automatically. A manual input dialog allows you to enter bearing parameters that are not included in the database. Geometric parameters are entered for all bearing types: ring and rotating elements diameters, contact angle, and, for roller bearings – roller length. A special feature is the ability to check for acceptable input parameter ranges. Thus, roller length can vary widely from 0 to 700 mm. Accuracy for all types of bearings in the APM Bear module is assessed by the runout of the outer and inner rings, with a maximum permissible value of up to 10 μm. An analysis of the obtained results showed that the stresses in the contact zone of the raceways and rolling elements are reduced by almost half (Fig. 8, a). This is one of the factors that significantly increases the service life of the proposed innovative option. The runouts in the alternative option are comparable and are within the standard values, except for the axial runout parameters, which significantly exceed the permissible ones. This means the need to use duplexed supports. The choice of bearing fit on a shaft depends primarily on the magnitude, direction, and nature of the loads acting on the bearing. The resulting 100 normal force distribution variants allow us to determine the number of simultaneously applied forces (Fig. 8, c) for the first iteration, the maximum load, and the amount of displacement (runout) within acceptable limits. Duplexing is achieved by mounting two bearings on the front support: 1) an innovative single row tapered roller bearing developed by the authors; 2) a double-row SKF 22344(00) series angle-contact ball bearing [17]. The SKF double-row bearing design has a one-piece housing and two rows of ball cage assemblies with a contact angle of the rolling elements and raceways equal to 60^0. These two rows are separated by spacer sleeves, which create a preliminary load in the axial direction equal to $F_{pl} = 490$ N. (Bore diameter $d = 65$ mm).

The proposed front support design reduces the negative impact of multi-row design problem by switching from triplex (angle-contact ball bearings) to a Duplex mounting arrangement based on a tapered roller and double angle-contact ball bearing, which supports bidirectional axial loads. Design calculations show that the service life of the upgraded front spindle support increases by an order of magnitude $L_{10} = 272563$ hr. In this module angle-contact bearings are considered to be mounted in pairs with mandatory axial tightening, the maximum value of which, regardless of the mounted type: "O" or "X" – ranges from 0 to 10^5 N. When choosing the mounted type, it should be taken into account that if the shaft temperature is higher than the housing temperature, then the preload is reduced in the "O" arrangement, while it is increased in the "X" arrangement. After selecting the initial variable load on the bearing assembly, the *Set Variable Load* button becomes active. The designer can select the loading mode (constant, heavy, average probability, average normal, light, or very light) from the drop-down menu. This automatically launches the variable load schedule editor, allowing for consideration of the specifics of the machining process.

B APM Bear, the absolute and relative displacements of the bearing center (depending on geometry, accuracy and external load) coincide, so the terms displacement and runout are used interchangeably.

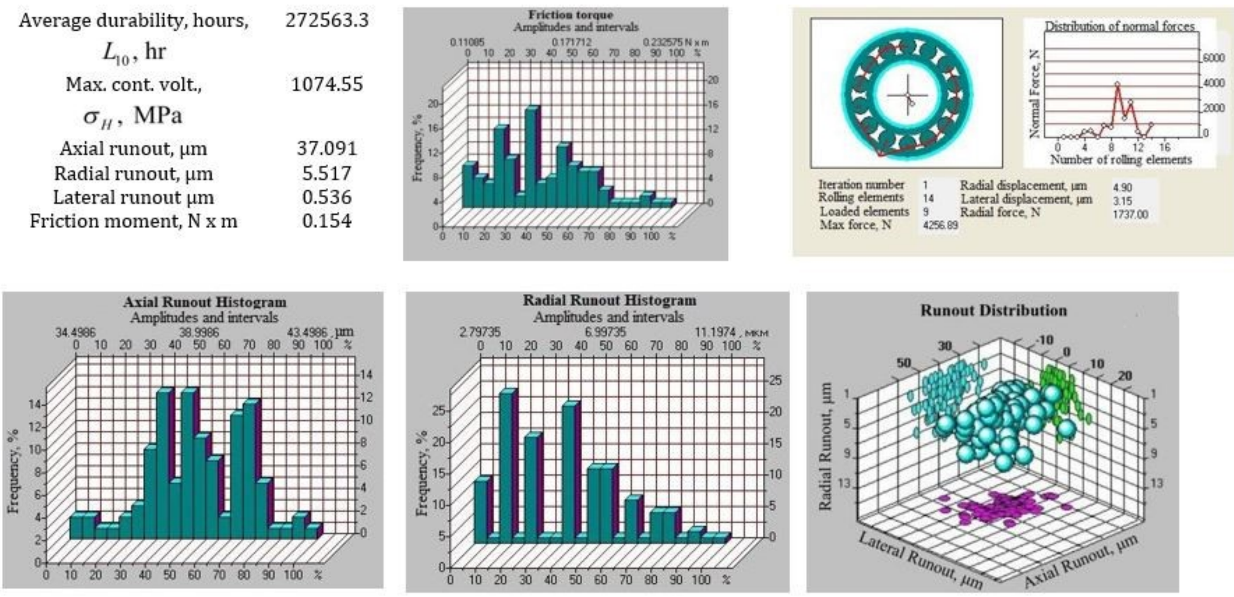


Figure 8: Calculation results for SKF 32013 single row tapered roller bearing

The calculation of displacements is the APM Bear core. The system gives the user a unique opportunity to determine the actual motion of a bearing. To characterize the movements, an array of 100 bearing center positions is calculated (simulated). For each position, up to three components (axial, radial, and lateral) can be calculatedThe results of displacement calculations can be presented in the form of: a table with static characteristics (mean, variance, standard deviation), a histogram, a field of bearing center positions, or an animation. In this paper, a calculation was made of a tapered roller bearing (Fig. 8), mounted on a front support, as an alternative to a triplex, the calculation of which was presented above. The resulting innovative solutions increase load capacity and durability. Furthermore, it becomes possible to implement a compromise design between the insufficient load capacity required for high-speed machining and the insufficient speed required for roughing operations.

5. Conclusions

As a result of the research, the following results were obtained.

A comprehensive design has been developed for a two-stage drive for the main movement of a multi-operation milling machine with a two-support spindle, mounted on a patented innovative front support with a modified single-contact tapered roller bearing.

A preliminary design for a two-stage MC gear drive with dual-zone adjustment and an extended speed range has been developed, enabling a wide range of technological processes from roughing to finishing. The spatial arrangement of the drive shafts was determined to maximize the rigidity of the spindle assembly, resulting in a force inclination angle of approximately 40°, acting on the output shaft.

A spindle assembly design with front support, which is mounted on SKF 7013 AC angle-contact ball bearings, was developed. This support is characterized by Tandem-X arrangement with preload in the form of two bushings of different heights. A comprehensive calculation of the bearings' key parameters was performed using the specialized APM Bear module. This module is based on a system for calculating the parameters of non-ideal bearings whose contact surfaces have shape errors. The resulting calculated strength parameters for bearing SKF 7013 AC are within acceptable limits, and durability is significantly below acceptable values. Duplexing the angle-contact bearings does not solve the problem of the required operating life of the spindle unit.

The 3D design of a two-stage gear drive, as the basis for the engineering study, was created in the integrated Creo Parametric CAD system. The technical requirements for the shaft-bearing assembly are implemented using the tolerance design tool (e7 Tolerance module) with the Creo-stackap procedure. The completed rendering and exploded view of the drive unit provide a clearer understanding of the design and manufacturability of the assembly.

An innovative front support for the spindle assembly was developed based on a modernized single-row tapered rolling bearing design, supported by the author's patent. The APM Bear module performed a comprehensive calculation of force parameters (normal forces and friction moments) and accuracy parameters (radial, axial, and lateral runout). Analysis of the results showed a reduction in contact stress (almost twice) to 1074 MPa and an increase of an order of magnitude ($2.7\cdot10^4$ hr) the durability of the most loaded front support of the spindle unit.

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