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METHODOLOGY FOR DESIGN AND FOR OBTAIN STIFFNESS OF MACHINE TOOL MAIN SPINDLE UNITS WITH ANGULAR BEARING SYSTEMS

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Abstract: *This paper presents a methodology for designing the main spindles of machine tools as well as obtaining their radial stiffness, especially those with angular contact ball bearing type rolling. Detailed information is given on the steps involved in determining the stiffness of main spindles, and a complete design flow for such systems is presented and described. The studies carried out are based on the design of a hand-clamped milling spindle for a chosen purpose, to which different types of arrangements will be applied to the bearing system. The aim is to determine the resulting deflections at the cartridge flange and their stiffness in order to determine the best performing spindle for the light milling application. Two methods of obtaining the deflections, one analytical and one numerical FEA, are used to verify the results obtained, in order to compare the results and the errors.*

Key words: *spindle unit machine, spindle arrangement, milling spindle, bearing arrangement, spindle design, motorized spindle, bearing stiffness, spindle rigidity.*

1. INTRODUCTION

Main spindle units for machine tools are part of the main kinematic chains that provide the rotary motion of the workpiece or tool under the precision conditions required by the type of machine and application. The accuracy of such a system is directly related to the radial run-out of the bearings and the stiffness of the system, which is mainly made up of the cartridge and the bearing system.

In order to meet these requirements, it is necessary to identify the appropriate bearing systems according to the purpose, load and speed, and then to achieve an optimal spindle dimensioning.

According to paper [1], main spindle units are considered to have a direct influence on machined surfaces and productivity.

The requirements of the industrial and mechanical field related to their performance are continuously increasing and higher machining speeds are required, but this leads to an increase in the complexity of the systems, the mechanisms integrated in them, and it is necessary to use integrated motors, Figure 1.

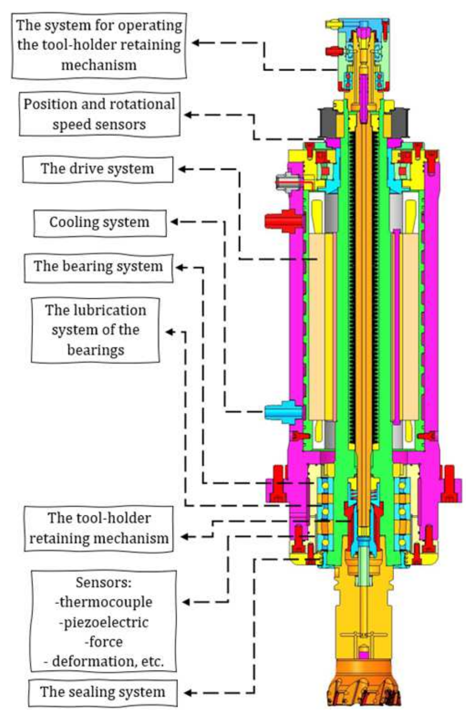

Fig. 1. Systems and mechanisms in a main spindle unit

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2. RESEARCH ON THE STIFFNESS OF THE MACHINE'S MAIN SPINDLES

When theoretically determining the total stiffness of a machine's main spindle, it is necessary to know the cutting forces acting on the system, as well as the axial and radial stiffness of the bearings or sets of bearings used in the design stages.

The stiffness of a main spindle can be determined according to the amount of elastic deformations of the system under the action of a critical machining condition, and these deformations are produced by torsional and bending moments. Manufacturers of such machine tools or spindles provide information on its axial and radial stiffness at the flange.

Radial stiffness (δ) is the most important in such a system because it has the greatest influence on machining accuracy, and this can be determined by calculating the elastic deformations of the cartridge (δ_1) summed with the elastic deformations of the bearing system (δ_2), Figure 2.

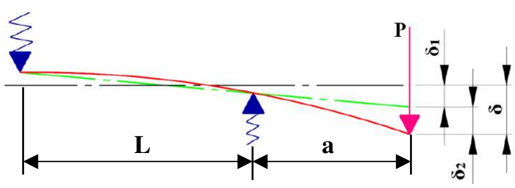


Fig. 2. Elastic deformation of the cartridge and the bearings system, adapted after [2]

Calculating the total elastic deformations of a main spindle system can be done with equation (1) [3], and in order to achieve this we need to set some input data and go through some of the following steps:

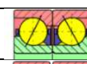
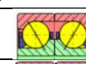
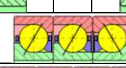
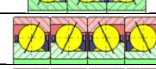
- determination of the force *P [N]* by determining a critical case of machining;
- determining the type of bearing arrangement for the front and rear;
- the distance between the bearings, denoted by *L [mm]* and the distance from the flange where the force *P* is applied to the center of the front bearing set, denoted by *a [mm]*, Figure 2;
- determination of the modulus of elasticity *E* *[N/mm²]* for the cartridge depending on the material chosen;
- calculation of the moment of inertia of the cartridge in the bearing set area for the front side, I_a *[mm⁴];*
- calculation of the moment of inertia of the cartridge between the front and rear bearings system, I_L *[mm⁴];*
- choice of axial stiffness, denoted C_a *[N/µm]*, from the bearing manufacturer's catalogue according to preload class;
- calculation of the total axial stiffness of the bearing sets, denoted by $C_{a\text{-}set}$ *[N/µm]*, according to Table 1;
- calculating the total radial stiffness of the bearing sets, denoted by $C_{r\text{-}set}$ *[N/µm]*, as a function of bearing contact angle according to equations (2)÷(6) and the data from Table 2.

$$\delta = P\left[\frac{1}{S_A} * \left(\frac{a+L}{L}\right)^2 + \frac{1}{S_B} * \left(\frac{a}{L}\right)^2 + \frac{a^2}{3E}\left(\frac{L}{I_L} + \frac{a}{I_a}\right)\right] \quad (1)$$

According to equation (1), S_A and S_B represent by convention the radial stiffnesses of the $C_{r\text{-}set}$ bearing sets for the front and rear but having the units of measurement converted from [N/µm] to [N/mm].

Table 1

Total axial stiffness of the angular bearing set, adapted [4] [5]

Types of bearing arrangements				$C_{a\text{-}set}$ [N/µm]
Coding	Figure			
DB (DF)		or		C_a
DT		or		$2 \cdot C_a$
TBT				$1{,}64 \cdot C_a$
QBC				$2 \cdot C_a$
QBT				$2{,}24 \cdot C_a$
QTTTQ				$2{,}64 \cdot C_a$

For α = 15°: $C_{r-set} = q_1 \cdot 6 \cdot C_{a-set}$ (2)

For α = 18°: $C_{r-set} = q_1 \cdot 4.5 \cdot C_{a-set}$ (3)

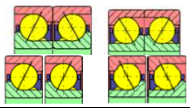
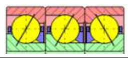
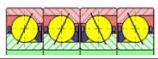
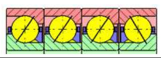
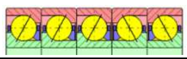
For α = 20°: $C_{r-set} = q_1 \cdot 3.5 \cdot C_{a-set}$ (4)

For α = 25°: $C_{r-set} = q_1 \cdot 2 \cdot C_{a-set}$ (5)

For α = 30°: $C_{r-set} = q_1 \cdot 1.4 \cdot C_{a-set}$ (6)

Table 2

Radial stiffness factors of the bearing set for angular contact ball bearings, adapted [4] [5]

Bearing Arrangement Model		Radial stiffness factor q_1
Coding	Figure	
DB, DF, DT		1
TBT		1,36
QBC		2
QBT		1,6
QTTTQ		2,72

The radial stiffness of the main spindle system, denoted by K [N/μm] can be determined, as the ratio between the cutting force P *[N]* applied to the cartridge flange and the total deformation resulting in equation (1).

3. MAIN SPINDLE DESIGN FLOW

Modern machine tools require the use of high cutting speeds and high accuracy, which means that the spindles used by them will have to meet ever higher technical requirements, [6]. The detailed flow of spindle design, figure 3, presented in this paper addresses two harmoniously merged design concepts, which will ultimately lead to achieving the axial and radial stiffness of the designed spindle.

The design concepts used are based on theoretical-analytical calculations and FEA analysis applied to a CAD prototype, and the results obtained can be compared.

The advantage of CAD design based on the virtual model is that it shortens the design and development time of the spindles, at the same time facilitating FEA studies of static, dynamic, modal and thermal studies, as well as the necessary optimizations and verifications.

A brief description of the design flow and the necessary steps will be made as follows:

- The first step in the CNC main spindle design flow is to determine its purpose and the machining operations it needs to perform. At this stage, the degree of universality of the main spindle must also be taken into account as it can perform several cutting operations, such as turning, milling, boring, drilling and threading in a machining center.
- Once these aspects have been established, it is necessary to outline a critical machining case, resulting in the choice of the material to be machined, characterized by hardness and high mechanical strength, followed by the choice of the tool used. By following the indications provided by the manufacturers, the cutting parameters, cutting forces, power and time required for the operation will be determined according to the tool used.
- For the preliminary drawing of the cartridge contained in the main spindle system it is necessary to choose the type of bearing, the lubrication system of the bearings, the type of drive, the type of clamping (manual or automatic) and the interface of the holder with the cartridge. All these will contribute to the dimensional and geometric parameters of the cartridge.
- Calculating the radial and axial stiffness of the bearings and determining the optimum bearing spacing is an important and necessary step in the analytical and static FEA calculation of the total spindle flange deflection. In this phase, the CAD model of the cartridge together with the bearing system will be completed in more detail. By determining from the two methods the deformations at the spindle flange and knowing the cutting forces acting on it, the maximum stiffnesses of the system can be deduced.
- The results obtained will be compared and validated in accordance with the designer's expectations for the chosen application, and if the accuracy obtained does not meet the spindle requirements, optimization and recalculation will be carried out by re-calculating the bearing system, recalculating the axial and radial stiffnesses of the bearing system, the optimum distance between bearings, resizing the cartridge and updating the other factors involved.
- The final stage will consist of the detailed design of the spindle system including lubrication, sealing, cooling and the cartridge

holder-release system. The complete design of the spindle will be completed by adding the assembly and execution drawings together with the technical documentation, etc.

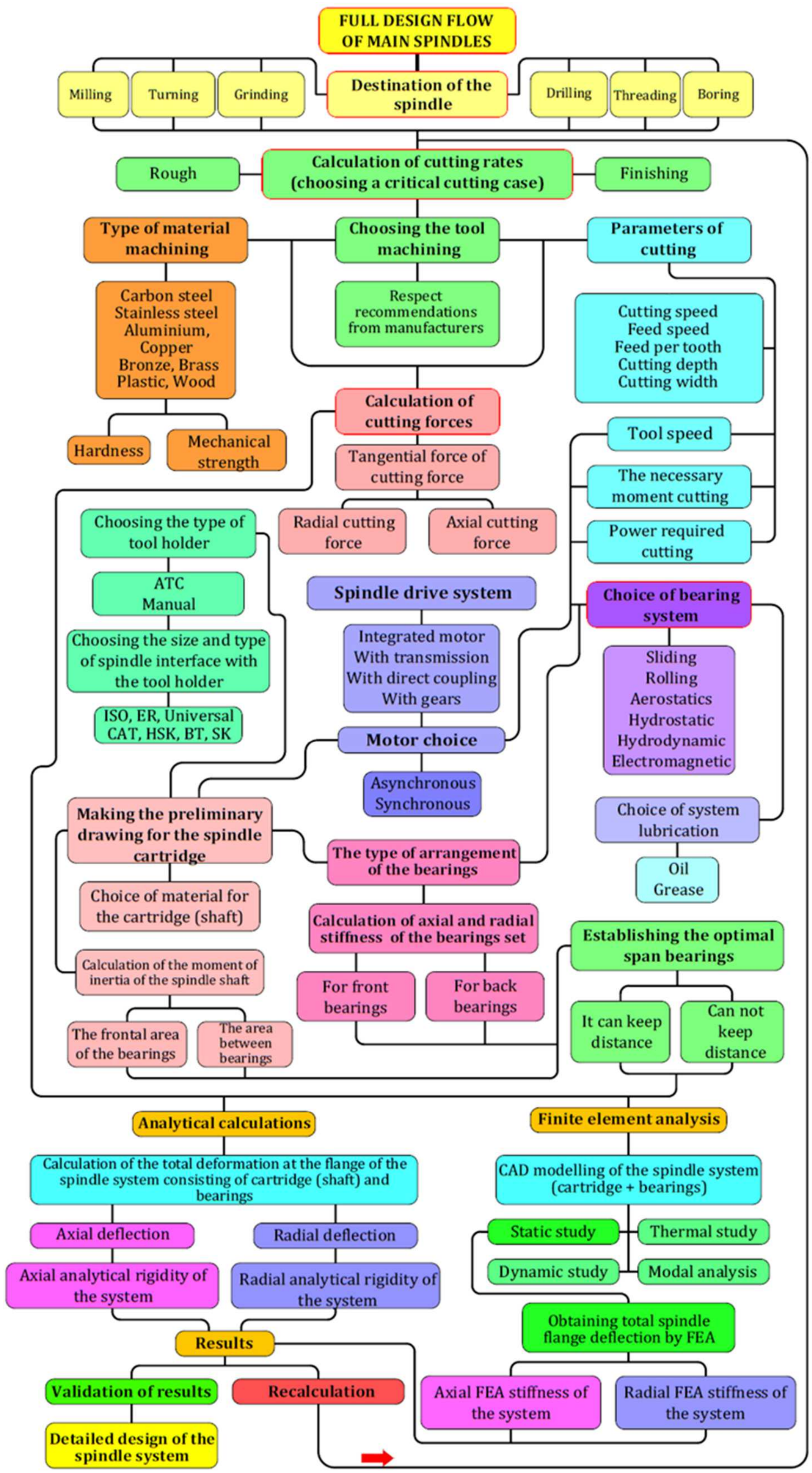


Fig. 3. The complete design flow of main spindles for CNC machine tools

4. DESIGN OF A MAIN MILLING SPINDLE FOR LIGHT CUTTING

In order to carry out the radial stiffness studies in the following work we consider the design of a main spindle used for light milling operations on aluminium, plastic or wood materials on a CNC router.

The construction of the main spindle should be as simple and compact as possible using low-cost technical solutions. The lubrication system will be achieved by using grease lubricated angular contact bearings, and in order to meet the requirements and the degree of precision required for its purpose, a series of combinations and arrangements of bearings at the front and rear of the spindle will be made to determine the most appropriate solution in terms of performance.

When studying the deformations and stiffnesses conferred by the different bearing arrangements for the spindle system, it is necessary to determine the cutting forces by choosing a critical use case on the machine.

Consider a situation of rough milling of a material such as aluminium EN AW 6012-T6, with a mechanical strength of 310 MPa, with a 20 mm diameter end mill with four cutting edges, using the following cutting parameters: cutting depth a_p= 1 mm, cutting width a_e= 20 mm, feed per tooth f_z= 0.2 mm/tooth, cutting speed v_a= 500 m/min. Using the Kennametal simulator the following characteristics and cutting forces will result: cutting power P_a= 2.48 Kw, motor power required for cutting P_m= 2.76 Kw, cutting moment M_a= 1. 56 Nm, material removal rate Q=1528 cm^3/min, tangential cutting force F_T= 156.24 N, radial cutting force F_R= 85.93 N and axial cutting force F_A= 39.06 N.

In the case studies carried out the maximum cutting force denoted as P= 156.24 N will be considered and will be applied to the cartridge flange in all the case studies addressed.

According to what was previously obtained, the designer can choose the characteristics required for the main spindle in terms of the performance of the integrated asynchronous motor, and these are: power of 3 kW, torque of 1.6 Nm, maximum speed of 18,000 rpm, manual clamping of the tool type ER32, and single-phase 220 V supply.

The basic specifications of a main spindle such as power, rpm and torque are available when buying such equipment, but most of the time other information about its construction and what is inside is relevant, and most manufacturers will not give such information because it is part of their "know-how", even mentioning the loss of warranty if they do not service the unit with them. A very important feature when choosing or designing a main spindle is related to the bearing system, the type of bearings and the arrangement of the bearings, which will ultimately lead to the determination of the stiffness as well as the machining accuracy and lifetime.

Figure 4 shows the position of the components of the main milling spindle designed for this application and used for the studies.

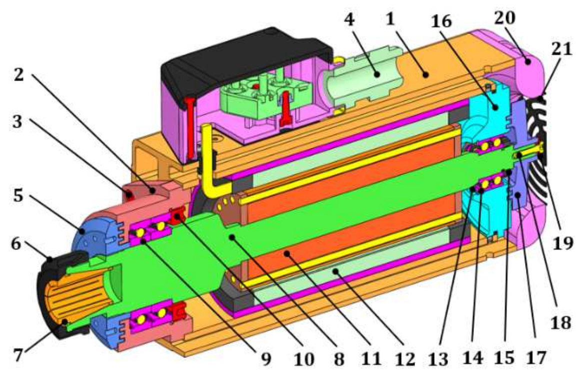


Fig. 4. Spindle configuration and its components
(1- main housing, 2- front bearing set housing, 3- fastener M5x16, 4- electrical connections, 5- front cover for pre-tensioning the bearings and sealing, 6- tightening nut ER32, 7- elastic collet, 8- spindle cartridge, 9- front bearing set, 10- bearing nut, 11- integrated rotor motor, 12- integrated stator motor, 13- elastic retaining ring, 14- back bearing set, 15- bearing nut, 16- back bearing set housing, 17- sealing cover, 18- fastener M4x16, 19- plain washer, 20- back cover for housing, 21- ventilation mesh)

The main spindle cartridge, Figure 5, has a relatively simple geometry that is easy to make, facilitating both its machining and the assembly of components and the bearing system.

In its front part, an internal feature is machined to match the ER32 tool retention system. The contact surfaces of the cartridge with the tool clamping spring bushing and the bearing seats will require a surface hardening treatment of the cementing type.

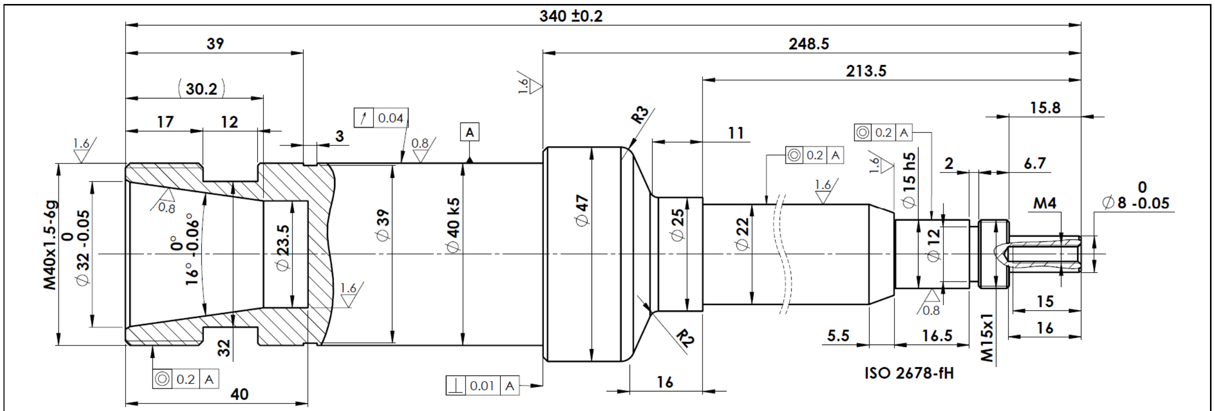


Fig. 5. Design drawing of the main spindle cartridge with ER32 clamping

The material from which the cartridge is made is 16MnCr5 (numerical coding: 1.7131), a case-hardened steel with soft core and abrasion-resistant surfaces which is widely used in the machine-building industry, i.e. in the machine tool industry due to its high performance in terms of mechanical properties: mechanical strength 800 MPa, yield strength 590 MPa, [7].

The distance between the front bearing of the cartridge and the rear secondary bearing will depend on the length of the integrated motor and the type of bearing arrangements used in the construction of the bearings system.

The use of the ER32 hand tool clamping system will further dictate the geometry of the cartridge and the mounting diameters of the bearing system, thus for the front side of the cartridge the minimum bearing mounting diameter will be ϕ40 mm and for the rear side will result in a maximum diameter of ϕ15 mm.

5. THE PROPOSED ARRANGEMENTS AND THE RESULTING STIFFNESSES FOR THE BEARING SETS

The bearing system is composed of angular bearings and the arrangements are based on the literature, according to [8][9][4] and are correlated with the application served in the following cases of three spindle unit models, according to the arrangements in Table 3.

In all three cases, Figures 7-9, three preload classes light (L*), medium (M*), high (H*) will be considered for the front side bearing sets and for the rear spindle bearing system we will use a light (L*) preload class. The bearings used have precision class P4S (ABEC 9 /ABEC 7 Blend) and in terms of radial runout they show better values, 2.5 µm, compared to the DIN620 standard, see figure 6.

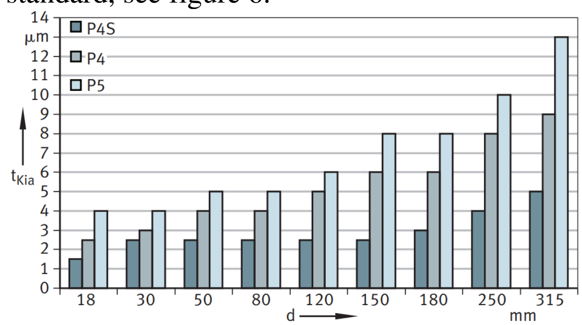
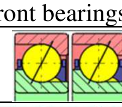
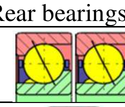
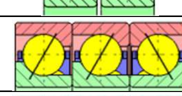
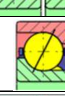
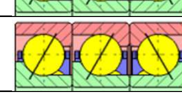
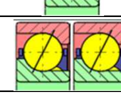


Fig. 6. Radial runout tolerances the inner ring (t_{Kia}) for P4S bearing precision compared to DIN620 [4]

Bearings can be ordered with side seals for grease lubrication systems and the axial stiffnesses, preload forces and limit forces at rolling element separation will be found in the catalogues of the manufacturers according to precision and coding.

Table 3

Case study bearing arrangements of the designed spindles

Spindle Coding	Type of arrangement Front bearings set	Type of arrangement Rear bearings set
DT-DT		
TBT-R		
TBT-DT		
Specs of the bearings	HS7008-C-T-P4S d= ϕ40 mm D= ϕ68 mm B= 15 mm, α= 15° n= 30000 rpm (grease) Precision class: P4S	HS7002-C-T-P4S d= ϕ15 mm D= ϕ32 mm B= 9 mm. α= 15° n= 70000 rpm (grease) Precision class: P4S

General input data are the cutting force P=156 N, the cartridge material 16MnCr5, the modulus of elasticity E= 210.000 N/mm^2, the moment of inertia of the cartridge area for the front side I_a= 125663,7 mm^4 and for the back side I_L = 26087,1 mm^4.

For case 1, a main milling spindle is considered, Figure 7, which consists of a DT arrangement (with two bearings in tandem) for the main bearing at the front of the spindle and the rear of the spindle where the secondary bearing is located.

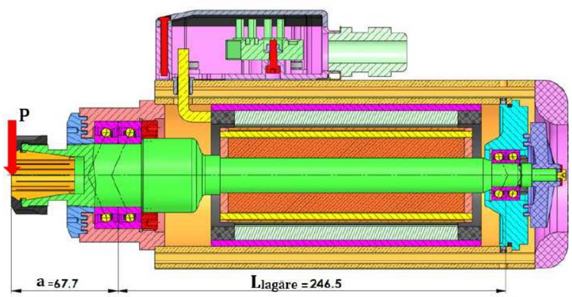


Fig. 7. Highlighting the milling spindle bearing span with DT-DT arrangement

For the main spindle designed in the established DT-DT arrangement, the distance between the bearings is equal to L_{lagar} = 246.5 mm and the minimum distance between the center of the driving bearing and the point of application of the cutting force is equal to a = 67.7 mm.

Table 4

Input data for case study 1 (DT-DT)

Bearing: HS7008-C-T-P4S	The arrangement of the front bearing set	DT
Bearing: HS7002-C-T-P4S	The arrangement of the rear bearing set	DT
L*	Axial stiffness of the bearing set for the front of spindle, $C_{a\text{-}set}$ [N/mm]	$60{,}2 \cdot 10^3$
M*		$94{,}8 \cdot 10^3$
H*		$129.8 \cdot 10^3$
L*	Radial stiffness of the bearing set for the front of spindle, S_A [N/mm]	$361{,}2 \cdot 10^3$
M*		$568{,}8 \cdot 10^3$
H*		$778{,}8 \cdot 10^3$
L*	Axial stiffness of the bearing set for the rear of spindle, $C_{a\text{-}set}$ [N/mm]	$27{,}4 \cdot 10^3$
L*	Radial stiffness of the bearing set for the rear of spindle, S_B [N/mm]	$164.4 \cdot 10^3$

Table 4 summarizes all the input data for the realization and determination of spindle deflections and stiffness in DT-DT arrangement at the cartridge flange, necessary for theoretical-analytical and numerical FEA calculations at different bearing preload classes.

The main milling spindle in figure 8 will represent case 2, which is a TBT arrangement (with three bearings, two in tandem and one opposed) for the driving bearing and a single radial axial angular contact bearing (R) for the rear.

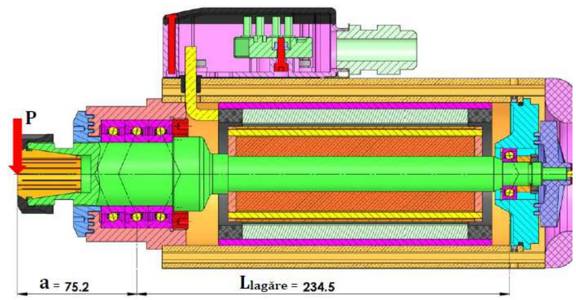


Fig. 8. Highlighting the milling spindle bearing spacing with TBT-R arrangement

For the bearing arrangements established for the main spindle in case 2 of type TBT-R, the distance between the bearings as equal to L_{lagar} = 234.5 mm and the distance between the center of the driving bearing and the point of application of the cutting force is equal to a = 75.2 mm.

The input data summarized in Table 5 are needed in the analytical calculations that follow as well as in the FEA simulations for determining the cartridge flange deformations and stiffnesses, for the main spindle in TBT-R arrangement.

Table 5

Input data for case study 2 (TBT-R)

Bearing: HS7008-C-T-P4S	The arrangement of the front bearing set	TBT
Bearing: HS7002-C-T-P4S	The arrangement of the rear bearing set	1B
L*	Axial stiffness of the bearing set for the front of spindle, $C_{a\text{-}set}$ [N/mm]	$49{,}4 \cdot 10^3$
M*		$77{,}7 \cdot 10^3$
H*		$106{,}4 \cdot 10^3$
L*	Radial stiffness of the bearing set for the front of spindle, S_A [N/mm]	$403{,}1 \cdot 10^3$
M*		$634{,}0 \cdot 10^3$
H*		$868{,}2 \cdot 10^3$
L*	Axial stiffness of the bearing set for the rear of spindle, $C_{a\text{-}set}$ [N/mm]	$13{,}7 \cdot 10^3$
L*	Radial stiffness of the bearing set for the rear of spindle, S_B [N/mm]	$3{,}45 \cdot 10^3$

Consider the milling spindle in Figure 9, for case 3, which consists of a TBT arrangement (with three bearings, two in tandem and one opposed) for the main bearing at the front of the spindle, and a DT arrangement for the rear of the secondary bearing.

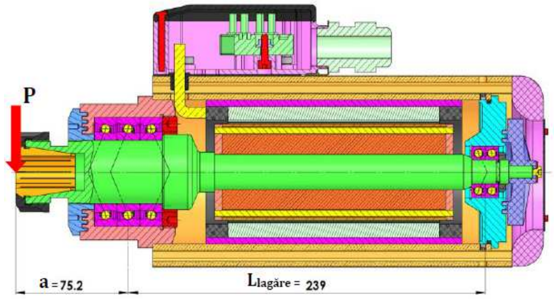


Fig. 9. Highlighting the distance between milling spindle bearings with TBT-DT arrangement

Thus, for the established TBT-DT bearing arrangement, the distance between the bearings span is equal to L = 239 mm and the minimum distance between the center of the bearing and the point of application of the cutting force is equal to a = 75.2 mm.

Table 6

Input data for case study 3 (TBT-DT)

Bearing: HS7008-C-T-P4S	The arrangement of the front bearing set	TBT
Bearing: HS7002-C-T-P4S	The arrangement of the rear bearing set	DT
L*	Axial stiffness of the bearing set for the front of spindle, $C_{a\text{-}set}$ [N/mm]	$49{,}4 \cdot 10^3$
M*		$77{,}7 \cdot 10^3$
H*		$106{,}4 \cdot 10^3$
L*	Radial stiffness of the bearing set for the front of spindle, S_A [N/mm]	$403{,}1 \cdot 10^3$
M*		$634{,}0 \cdot 10^3$
H*		$868{,}2 \cdot 10^3$
L*	Axial stiffness of the bearing set for the rear of spindle, $C_{a\text{-}set}$ [N/mm]	$27{,}4 \cdot 10^3$
L*	Radial stiffness of the bearing set for the rear of spindle, S_B [N/mm]	$164{,}4 \cdot 10^3$

The input data in Table 6 refer to the main spindle designed in case 3 for TBT-DT arrangement and are needed for analytical calculations and FEA simulations to determine displacements and stiffnesses at the main spindle flange taking into account the bearing preload classes.

6. RESULTS OF THE CASE STUDIES FOR THE DESIGNED SPINDLES

Table 7 and Figure 10 shows the theoretical optimum distances between bearings span, the value of the minimum possible flange deflection and the maximum radial stiffness of the spindle for the different classes of bearing preloads depending on the type of bearing arrangement used. In order to determine the minimum deformations and maximum radial stiffnesses that can be obtained, the cutting force P, initially determined in the input data, will also be taken into account.

These values, which are related to the optimal distance, were obtained by the theoretical-analytical method and subsequently the designer, by knowing them, is helped to visualize and orient the geometry and dimensions of the spindle unit.

Optimum bearing distances can only be maintained in the case of spindles driven by direct coupling to a motor, or those driven by pulleys or gears, with the proviso that the radial forces resulting from these types of drive shall be taken into account in the calculations.

Tabel 7

Optimum bearing span distances, deflections, and radial stiffness of designed spindle variants as a function of preload class

Different bearings arrangement	**DT-DT**	**TBT-R**	**TBT-DT**
Bearing preload class	**Optimum distance between bearings span [mm]**		
U*	87,42	220,78	91,22
M*	77,52	217,59	80,36
R*	65,04	216,11	74,96
Bearing preload class	**Flange deflection under force P [µm]**		
U*	6,35	18,65	7,67
M*	5,67	18,39	6,91
R*	4,85	18,27	6,55
Bearing preload class	**Radial spindle stiffness at flange [N/µm]**		
U*	24,59	8,37	20,35
M*	27,51	8,49	22,59
R*	32,15	8,55	23,84

* Bearing preload classes (U-light, M-moderate, R-heavy);

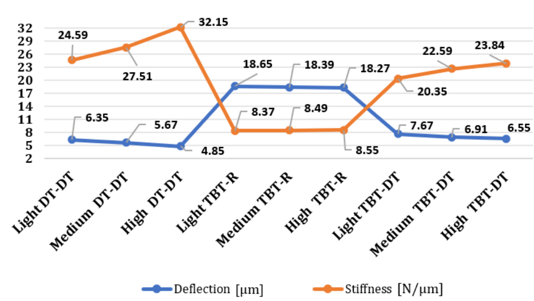


Fig. 10. Variation of deflections and radial stiffness for optimum span bearing distances as a function of arrangement and preload class of the bearings

Theoretical-analytical results according to the used span bearing distance for the designed main spindles are included in Table 8, as well as the resulting spindle flange displacements and radial stiffnesses obtained. According to the results, the best stiffness values are obtained in the main spindle variant in DT-DT arrangement for all front bearing preload classes.

Tabel 8

Bearing span distances, deflections and radial stiffness resulting from the analytical method for designed variants of main spindles

Bearing preload class	Type of bearing set arrangement	Distance between bearings span [mm]	Flange deflection under force P [μm]	Radial spindle stiffness at flange [N/μm]
U*	DT-DT	246,5	12,12	12,88
	TBT-R	234,5	18,78	8,32
	TBT-DT	239	14,45	10,81
M*	DT-DT	246,5	11,87	13,16
	TBT-R	234,5	18,53	8,43
	TBT-DT	239	14,20	10,99
R*	DT-DT	246,5	11,75	13,29
	TBT-R	234,5	18,41	8,48
	TBT-DT	239	14,09	11,08

Figure 11 shows the results of the numerical analyses for the main spindle in the DT-DT arrangement with light preloading of the bearing system, the value of the displacement in the area of the cartridge's flange where the tool holder is clamped is taken into account. For the other types of arrangements and class of preloading the same procedure was followed.

In order to verify and compare the results obtained for the designed main spindles in terms of the deformation at the cartridge flange and stiffness, a numerical study with the finite element method was performed and the values obtained are summarized in Table 9.

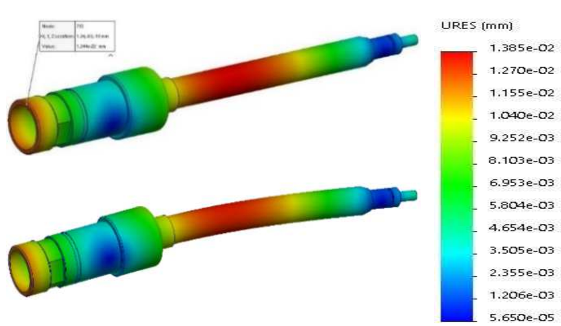


Fig. 11. Deformations recorded at the flange in DT-DT arrangement with light preloading of the front bearing set

Tabel 9

Bearing span distances, deflections and radial stiffness resulting from the FEA method for designed variants of main spindles

Type of preload of the main bearing set	Type of bearing set arrangement	Distance between bearings span [mm]	Flange deflection under force P [μm]	Radial spindle stiffness at flange [N/μm]
U*	DT-DT	246,5	12,44	12,56
	TBT-R	234,5	19,60	7,97
	TBT-DT	239	15,72	9,93
M*	DT-DT	246,5	12,26	12,74
	TBT-R	234,5	19,42	8,04
	TBT-DT	239	15,63	9,99
R*	DT-DT	246,5	12,07	12,94
	TBT-R	234,5	19,31	8,09
	TBT-DT	239	15,35	10,18

7. CONCLUSION

The study was carried out on three milling spindle models with different types of angular bearing arrangements assigned to the driving and secondary bearings.

The design of the spindles was based on keeping the original requirements and the studies considered the main casing, the casing of the main bearing and the support and fixing cover of the secondary bearing as non-deformable, having infinite stiffnesses and the rest of the components are removed from the analysis.

The variation of deflections and radial stiffness for the designed spindle, resulting from the analytical method and the FEA method, also taking into account the preload classes of the driving bearing set are shown in Figure 12. Thus, according to the obtained results the best performing type of spindle arrangement designed for the proposed application in the paper is the DT-DT arrangement variant of the bearings system.

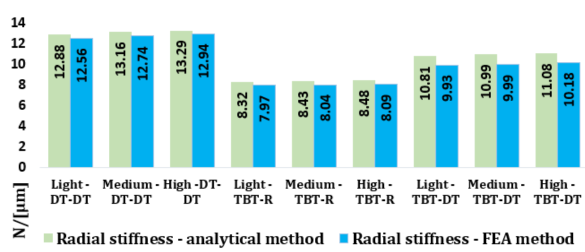


Fig. 12. Variation of radial stiffness as a function of the class of bearing preload and arrangement type

For a better overview of the results obtained by the analytical method and FEA used to determine the radial stiffness for the studied spindle variants, the error margins obtained are shown in Figure 13. The largest range of error is obtained for the TBT-DT variant with a medium preload of the driving bearing, with an error value of 9.09%.

From the authors' point of view, the percentages obtained in the studies can be accepted as they are relatively small values, but also due to the fact that the exact shape and geometry of the cartridge could not be taken into account in the analytical calculations.

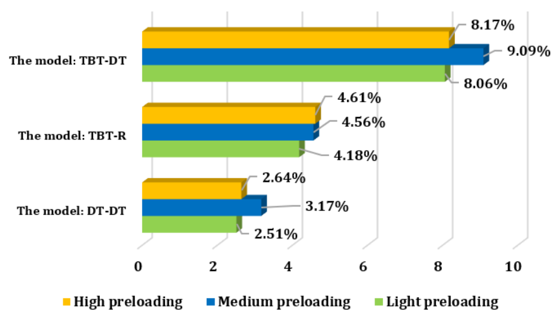


Fig. 13. The values of the range of error reported for the analytical method and the theoretical results

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METODOLOGIA DE PROIECTARE ŞI DE OBŢINERE A RIGIDITĂŢII ARBORILOR PRINCIPALI CU SISTEME DE LĂGĂRUIRE DE TIP RULMENŢI UNGHIULARI

Aceasta lucrare prezinta o metodologie de proiectare a arborilor principali ai maşinilor unelte precum şi obţinerea rigidităţii radiale ale acestora, în special ai celor cu lagăre de rostogolire de tip rulmenţi cu bile cu contact unghiular. Se vor regăsi informaţii detaliate in ceea ce privesc etapele de stabilire a rigidităţii arborilor principali si este prezentat şi descris un flux de proiectare complet pentru astfel de sisteme. Studiile realizate se bazează pe proiectarea unui arbore de frezat cu prindere manuală pentru o anumita destinaţie aleasă la care se va aplica diferite tipuri de aranjamente la sistemul de lăgăruire. Se urmăreşte stabilirea deformaţiilor rezultate la flanşa cartuşului şi rigiditatea acestora pentru a stabili care este cel mai performant arbore pentru aplicaţia de frezari uşoare. Totodată pentru a verifica rezultatele obţinute se utilizează doua metode de obţinere a deformaţiilor una pe cale analitica şi una numerica de tip FEA, cu scopul de a realiza comparaţia rezultatelor şi a erorilor.

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