TECHNICAL UNIVERSITY OF CLUJ-NAPOCA

ACTA TECHNICA NAPOCENSIS

Series: Applied Mathematics, Mechanics, and Engineering
Vol. 63, Issue IV, November, 2020

STATIC AND MODAL ANALYSIS OF HIGH-SPEED CNC MILLING SPINDLE

Claudiu-Ioan RUSAN, Cornel CIUPAN

Abstract: *The current paper analysis minimum static deflection of high-speed milling spindle at different span length bearing. The spindle design running at 12000 rpm with power rating of 25 Kw and the bearing angular contact value are 15º, the main spindle shaft is design for tool holders HSK-A 63. Static and dynamic analysis have an important role in estimating stresses, strains and displacements of the high speed milling spindle, where the static analysis can determinate the system rigidity and dynamic analysis show the natural frequencies and different mode of shape at different of speed of the milling spindle and obtain frequency response function. Further we evaluate the deflection nose, static and modal analysis by theoretical calculations and comparison with SolidWorks Simulation results.*

Key words: *static deflection, high speed milling, bearing stiffness, SolidWorks Simulation, design spindle, bearing arrangements.*

1. INTRODUCTION

The machine tools industry is an important sector in mechanical and industrial engineering. Machine tools have a strategic and well-established position in the global industry, and with their help we can produce almost all equipment, parts and machinery needed in production. They are present in all processes including metal and among objects containing parts made using machine tools we can mention the following: different machines, turbines, planes, trains, clocks, earthmoving equipment, etc.

Because of these important aspects of machine tools, the global manufacturers and companies which deal with the development and construction of their, are focus in the background on the protection of knowledge or better says the protection of the "know-how". Currently, machine tool building industry worldwide is in constant development and is supported by growing demand that apparently is hardly satisfied. The big players in the world who development these machine tools are in a competition, which leads to a continuous development which increasing the productivity, quality, and economic satisfaction.

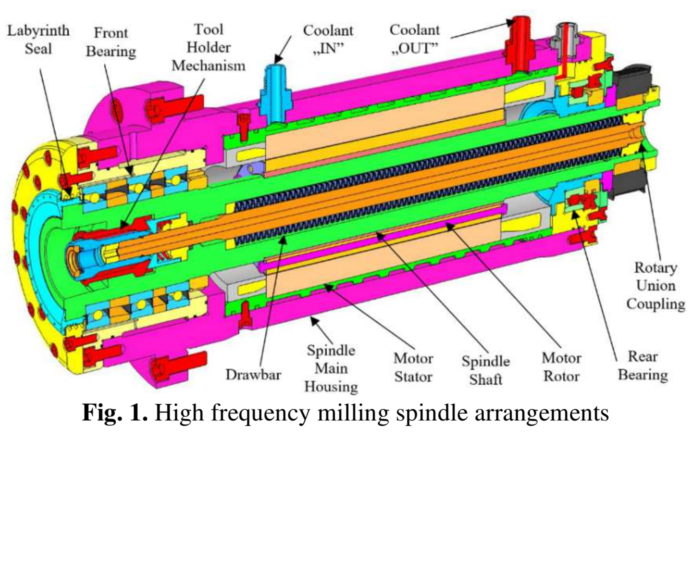

Fig. 1. High frequency milling spindle arrangements

Based on these aspects results it is necessary to expand the knowledge in this field of design and development high speed spindle for CNC machine tools, see figure 1.

Spindles are most important components in the machine tools or a CNC machining center, because the resulting quality is directly influenced by their performance. The productivity and precision of a CNC machining center is dictated by the following parameters: speed, power, torque, rigidity, temperature.

2. HIGH-SPEED CNC MILLING SPINDLE

2.1 Technical specification of spindle shaft

Destination of the spindle is for milling, having power located around 25 Kw, spindle speed of 12000 rpm, shaft type to HSK-A 63 tool holders, bearing contact angle 15^{0}, high rigidity and integrated drive with built-in-motor. The material chosen for the manufacture of the spindle shaft is 16MnCr5, thus designed for gripping HSK tool holders, according to the standard DIN 69063-1 and according to machining standard ISO 2768-mk [4], see figure 2. The contact area of the spindle shaft with the tool holders must have cementation at a depth h = 0,8–1 mm and the heat treatment quenching applied must respect the value of 60 ± 2 HRC.

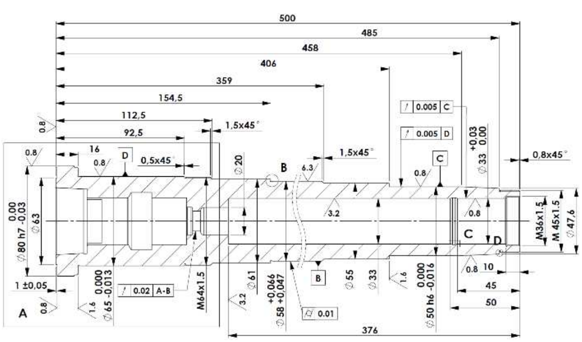

Fig. 2. Spindle shaft drawing – type of connection with the tool holders HSK-A 63

Table 1

Mechanical properties of alloy special steel 16MnCr5

Tensile strength [MPa]	min. 800
Yield stress [MPa]	min. 590
Youngs module [GPa]	210
Poisson´s ratio	0.3
Density [kg/m^3]	7800
Thermal conductivity ambient temp. [W/m°K]	40-45

2.2 Face milling and end milling

The following figure illustrates face milling and end milling operations, where milling cutting tools are known as multi-point cutting tools [1]. For determination of the cutting force components, two milling tools were chosen, first for face milling with carbide inserts, part no. AHX440SR08006CA [5] and the second for end milling in mono-block construction, made of HSS, part no. F2004NNH2000SDA [6].

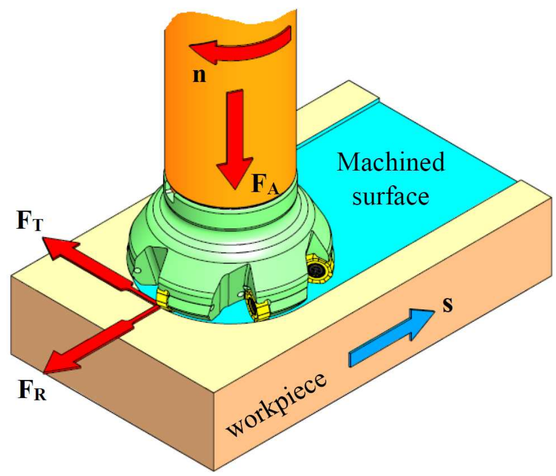

Fig. 3. Cutting force components for face milling

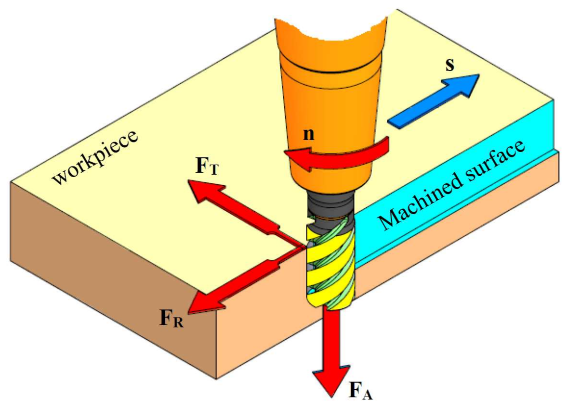

Fig. 4. Cutting force components for end milling

Determination cutting force components for face milling tool, who has 6 cutting teeth, cutting diameter D = 80 mm, the width of the milling area b is 80 mm, cutting parameters recommended by manufacturers: cutting speed v_a = 220 m/min, feed s_d = 0,2 mm/tooth, cutting depth t = 2 mm [5]. The processed material is carbon steel with a hardness of between 180-280 HB, where power constant K_p = 2,59, feed factor for power constant C = 1,08, and tool wear factor at light or medium milling W = 1,25 [7].

$$Q = \frac{s \cdot b \cdot t}{60000}, P_a = K_p \cdot C \cdot Q \cdot W \tag{1}$$

$$F_T = \frac{6120 \cdot P_a}{v_a}, F_R = 0{,}15 \cdot F_T, F_A = 0{,}55 \cdot F_T \tag{2}$$

Determination cutting force components for end milling tool, who has 4 cutting teeth, cutting diameter D = 20 mm, depth of the milling b = 25 mm, cutting parameters recommended by manufacturers: cutting speed v_a = 290 m/min, feed s_d = 0,07 mm/tooth, cutting depth t = 3 mm [6]. The processed material is carbon steel with a hardness of between 180-280 HB, where power constant K_p = 2,35, feed factor for power constants C = 1,08 and tool wear factor at light or medium milling W = 1,25 [7].

$$F_R = 0{,}55 \cdot F_T,\;\; F_A = 0{,}25 \cdot F_T \tag{3}$$

Where, Q is metal removal rate, F_T is tangential cutting force, F_R is radial cutting force, F_A is axial cutting force.

Table 2

Value result of cutting force components

Cutting force components	**Face milling**	**End milling**
Tangential cutting force: F_T [N]	2671,527	1406,655
Radial cutting force: F_R [N]	935,034	773,665
Axial cutting force: F_A [N]	1469,339	351,666

2.3 Bearing rigidity and arrangements

For providing rotational movements of the main shaft on a machining center, is takes the bearing systems that must meet certain requirements as follows: to withstand high speeds, to withstand heavy loads and shocks, high accuracy, low noise, reliability and high efficiency, low heating, low cost. The bearing arrangements following figure 5, with triplet bearing set at the front, in which one pair of angular contact ball bearing are arranged in tandem with respect to each other and back to back to a single angular contact ball bearing and a single radial cylindrical roller bearing at the rear mounted [8].

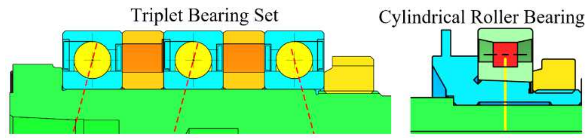

Fig. 5. Bearing arrangements at the front and the rear of main spindle

For the front bearing S7013 CB/HCP4A, axial stiffness for medium preloading, class C, has value C_a=158 N/µm and radial stiffness can be calculated:

$$C_r = 6 \cdot C_a = 948\ \frac{N}{\mu m} \tag{4}$$

Table 3

Technical specification of front bearing and rear bearing

Type and Specification	**S7013 CB/HCP4A**	**N 1013 KTNHA/HC5SP**
Size Details [mm]	d= 65, D= 100, B= 18	d= 65, D= 100, B= 18
Load Rating Static C_0 [kN]	41,6	56
Load Rating Dynamic C [kN]	37,5	44
Attainable Speed [rpm]	16000	17000
Type of Lubrication	grease	grease
Mass [kg]	0.36	0.39

For bearings that have angular contact value α = 15°, the axial stiffness and radial stiffness of the bearing set is calculated by:

$$C_{a-set} = 1{,}64 \cdot C_a = 259{,}12\ \frac{N}{\mu m} \tag{5}$$

$$C_{r-set} = 6 \cdot C_{a-set} = 1554{,}72\ \frac{N}{\mu m} \tag{6}$$

Where, Ca is axial stiffness of bearing, Cr is radial stiffness of bearing, Ca-set is axial stiffness of bearing set, Cr-set is radial stiffness of bearing set.

Table 4

Stiffness of front bearing and rear bearing

Type and Specification	**S7013 CB/HCP4A**	**N1013 KTNHA/HC5SP**
Axial Stiffness	158 N/µm	-
Radial Stiffness	948 N/µm	930 N/µm
Axial Stiffness of Bearing set	259,12 N/µm	-
Radial Stiffness of Bearing set	1554,72 N/µm	-

3. THEORETICAL CALCULATIONS OF DEFLECTIONS

In order to calculate the total deformation δ of the main spindle system, we must take into consideration two type of deformation: an elastic deformation of the spindle shaft under the action of cutting forces, δ_1 and one due to the elasticity of the bearing, δ_2 [2][3].

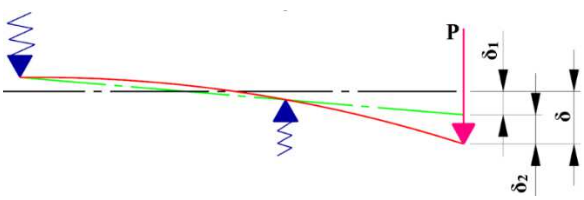

Fig. 6. Total spindle shaft and bearing deformation

$$\delta = \delta_1 + \delta_2 \tag{7}$$

$$\delta = P\left[\frac{1}{S_A}\left(\frac{a+L}{L}\right)^2 + \frac{1}{S_B}\left(\frac{a}{L}\right)^2 + \frac{a^2}{3\cdot E}\left(\frac{L}{I_L}+\frac{a}{I_a}\right)\right] \tag{8}$$

Where, P are cutting force in N, SA is the stiffness of the front bearing set in N/mm, SB is the stiffness of the rear bearing in N/mm, IL are the second moment of area of the spindle shaft at the span in mm^4, I_a are the second moment of area of the spindle shaft at the overhang in mm^4, a is the length of overhang in mm, L is bearing span in mm, E is Young's modulus of spindle shaft in N/mm2.

Due to the design constraints related to the size of the motor integrated in the high-speed spindle with the following characteristics: P = 25 Kw, n = 12000 rpm, the minimum distance possible between the bearing of the front and rear is L0 = 367.5 mm, and analytical calculation will concentrate on three variants or models.

Table 5

Spindle shaft length and bearing span used for analytical calculation and FEM simulation

Variants	Model 1	Model 2	Model 3
Spindle shaft length L-shaft [mm]	500	650	800
Bearing span [mm]	L0: 367,5	L1: 517,5	L2: 667,5

Considering as input data:

- S_A = 1554720 N/mm; S_B = 930000 N/mm;
- E = 210000 N/mm^2; I_a = 762306,564 mm^4;
- I_L = 248582,396 mm^4.

For the calculation of the static deflection on the spindle nose will be considered only the radial force cutting, P for face milling and end milling, subsequently being involved all input data and lengths required for each model of the span bearing being replaced in the equation (8). The results for the calculation of the static deflection are displayed in table 6.

If increase the distance of the bearing span will negatively and directly proportionally influence, by increasing the deformation at the spindle nose and decreasing its rigidity of the spindle shaft.

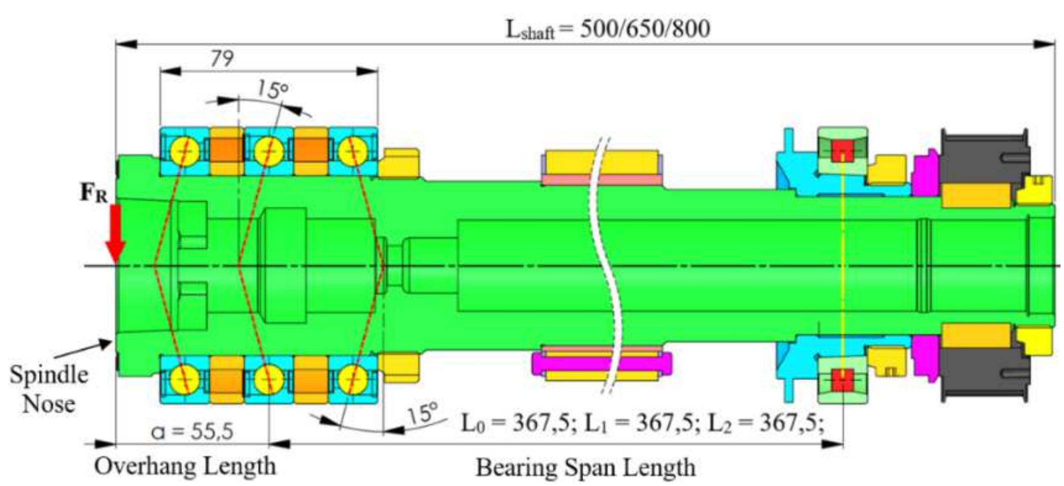

Fig. 7. Highlighting the main spindle shaft with different lengths of bearing span

Table 6

Theoretical results for static deflection on the spindle nose for each models of span bearing

Milling Operation	Radial cutting force F_R [N]	Model 1 L_0=367,5 [mm]		Model 2 L_1=517,5 [mm]		Model 3 L_2=667,5 [mm]	
		Theoretical Deflection [µm]	Spindle Stiffness [N/ µm]	Theoretical Deflection [µm]	Spindle Stiffness [N/ µm]	Theoretical Deflection [µm]	Spindle stiffness [N/ µm]
Face Milling	935,034	7,91	118,209	10,59	88,294	13,32	70,197
End Milling	773,665	6,54	118,297	8,76	88,317	11,02	70,205

4. STATIC ANALYSIS OF SPINDLE ASSEMBLY BY FEM

The finite element analysis method is a numerical analysis solution that can be used to solve of large engineering problems, which may involve analysis of stress, heat transfer or magnetic phenomena. This method is very effective in solving problems such as calculation of specific displacements or deformations, reaction forces, efforts, and safety factors.

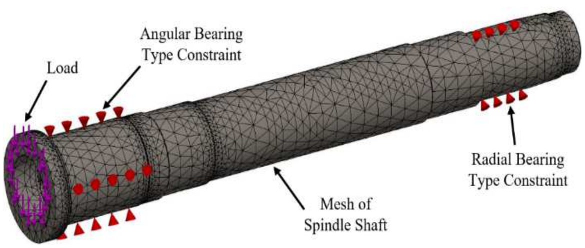

Fig. 8. Meshing and bearing constraints applied to the spindle shaft

Table 7

Material and meshing properties

Models	1	2	3
Nodes	84382	88562	91345
Elements	51557	53602	55732
Material properties	Youngs module [MPa]	Density [kg/m^3]	Poisson´s ratio
	210 x 10^3	7800	0.3

The analytical calculations performed, we must consider that are theoretical and approximate and for their verification and further comparison we will perform the same calculations transposed in the finite element analysis. To perform a FEM study of linear static analysis it is necessary to perform the following steps: creating the mesh of a cad model, setting the material properties, apply the constrains and loads, see figure 8. Setting the constraints for our system involves applying radial force on the spindle nose and setting the bearing type constraints in their seating area [9].

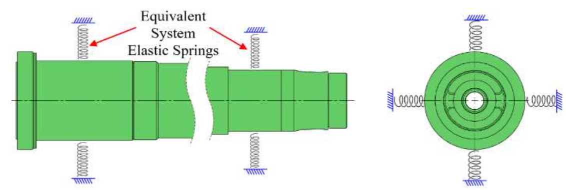

Fig. 9. Equivalent system of the main spindle attached by elastic springs

The application of the bearing type constraints on the 3D model is in fact the realization of an equivalent system attached by elastic springs, see figure 9, that has the radial rigidity and the axial rigidity taken from the bearing, mentioned in table 4. In SolidWorks Simulation give the degrees of freedom as UX,UY,UZ=0 and ROTX, ROTY, ROTZ=1.

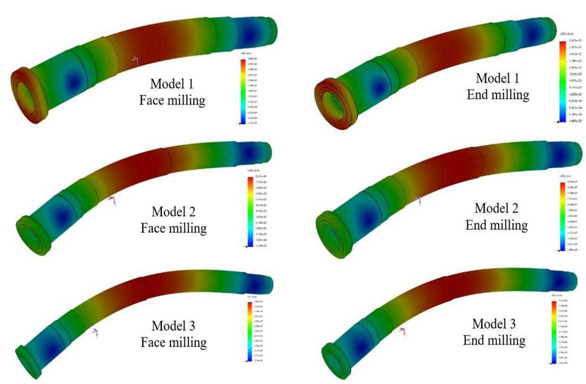

Fig. 10. FEM results for nose deflection of spindle shaft

Finite element analysis was also performed for the three models studied analytically previously, to determine the von Mises stresses and the displacements, see table 8, this led to the strengthening of results and conclusions.

Regarding the evolution of von Mises stresses, the highest value recorded after simulations of the spindles was 1,152e+07 N/m^2 , in conclusion, this value doesn't exceed the material yield stress of 5.905e+08 N/m^2.

Table 8

FEM results for static deflection on the spindle nose for each models of span bearing

Details and results	Milling Operation	
	Face Milling	End Milling
Radial cutting force F_R [N]	935,034	773,665
Model 1: L_0=367,5 [mm]		
FEM Deflection [μm]	3,84	3,17
Spindle Stiffness [N/μm]	243,498	244,058
Model 2: L_1=517,5 [mm]		
FEM Deflection [μm]	5,39	4,45
Spindle Stiffness [N/μm]	173,475	173,857
Model 3: L_2=667,5 [mm]		
FEM Deflection [μm]	7,11	131,509
Spindle stiffness [N/μm]	5,87	131,799

5. MODAL ANALYSIS OF SPINDLE ASSEMBLY

Modal analysis is the process of determining all the modal parameters, including also of natural frequencies. This analysis leads to prevent cracking of parts due to resonance phenomena. Any structure subjected to frequencies tends to vibrate and this analysis leads to prevent cracking of parts due to resonance phenomena. These frequencies that lead to the vibration of the structure are called natural frequencies and occur when no external disturbances act on the structure because they result from their own elastic and inertial forces. For each natural frequency that leads to the vibration of the structure, it is assigned its own mode of vibration. The main purpose of a modal analysis is to find its own vibration frequencies and avoid possible resonance phenomena. Figure 11 shows different mode shapes at different natural frequencies of spindle shaft for bearing span model 1 and we have got six natural frequencies, for the other two models are similarly analysed.

Mode shapes mainly depend on the density of the material, boundary conditions, stiffness of the shaft, the following table shows natural frequencies for all three models at different bearing span.

One of the requirements of the designed shaft is rated speed to be 12000 rpm, respectively the frequency will be 200 Hz.

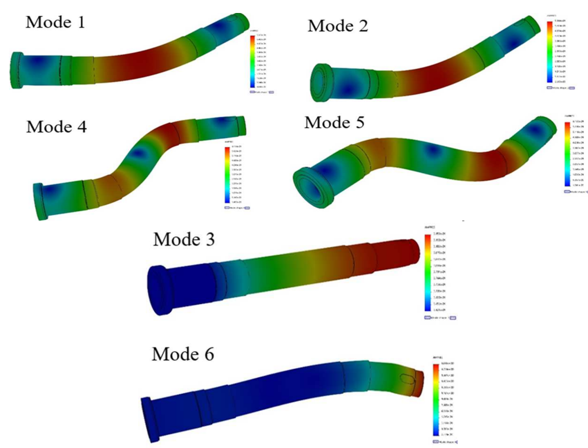

Fig. 11. Natural Frequencies and mode shapes for span bearing model 1

As can be seen in the results of this study, none of the six natural frequencies obtained by finite element analysis, for the three spindle variants, doesn't approach the spindle shaft frequency, during the milling operation. From this point of view, any of the three studied models will not reach the resonance phenomenon, as can be seen in table 9.

Table 9

Modal analysis results for each models of span bearing

Natural Frequency and mode shapes	Number of modes					
	1	2	3	4	5	6
Natural Frequency in Hz for model 1	893,29	893,37	949,26	3033,2	3033,7	3977
Mode shapes for model 1	Bending	Bending	Elongation	Buckling	Buckling	Bending
Natural Frequency in Hz for model 2	468,56	468,82	836,51	1712,4	1712,7	3450,1
Mode shapes for model 2	Bending	Bending	Elongation	Buckling	Buckling	Buckling
Natural Frequency in Hz for model 3	285,26	285,37	751,46	1087	1087	2260
Mode shapes for model 3	Bending	Bending	Elongation	Buckling	Buckling	Buckling

6. CONCLUSION

The analytical study and finite element analysis were performed on three-shaft models for two milling operations and with different bearing lengths, but with unchanged rigidity values for bearings. The value of span length bearing is imposed constructively by the chosen integrated motor and the minimum value is 367,5 mm.

According to the study, the precision of the main spindle is closely related to its stiffness, where the overall stiffness of the shaft is influenced by span length between the bearing and the rigidity of the front and rear bearings.

With increasing the span length bearing, spindle stiffness and deflection on the spindle noise also increase.

Static analysis of spindle assembly and modal analysis was carried out by using SolidWorks Simulation and the results give a good correlation between theoretical calculation and simulation results, see the summary of results in table 10.

The optimal spindle shaft variant of the study is model 1 who have a span length bearing 367,5 mm with a stiffness of 243,498 N/µm and deflection on the spindle nose of 3.84 µm for face milling, and stiffness of 244,058 N/µm and deflection on the spindle nose of 3.17µm, see table 11.

Further studies will be done on the rigidity of the high-speed spindles for CNC machine tools and how it is influenced when they have attached tool holders and tools with significant lengths.

Table 10

Results of theoretical study and finite element analysis

Operation	**Face milling**			**End milling**		
Model	**Model 1: L_0=367,5 [mm]**	**Model 2: L_1=517,5 [mm]**	**Model 3: L_2=667,5 [mm]**	**Model 1: L_0=367,5 [mm]**	**Model 2: L_1=517,5 [mm]**	**Model 3: L_2=667,5 [mm]**
Radial cutting force	935,034 [N]			773,665 [N]		
Theoretical analysis	. **Theoretical/Analytical results for static deflection on the spindle nose**					
Deflection [µm]	7,91	10,59	13,32	6,54	8,76	11,02
Stiffness [N/µm]	118,209	88,294	70,197	118,297	88,317	70,205
FEM analysis	**FEM results for static deflection on the spindle nose**					
Deflection [µm]	3,84	5,39	7,11	3,17	4,45	5,87
Stiffness [N/µm]	243,498	173,475	131,509	244,058	173,857	131,799

Table 11

Deflection and stiffness value for optimized configuration – model 1

Operation	**Theoretical**		**SolidWorks Simulation**	
	Deflection [µm]	Stiffness [N/µm]	Deflection [µm]	Stiffness [N/µm]
Face milling	7,91	118,209	3,84	243,498
End milling	6,54	118,297	3,17	244,058

7. REFERENCES

[1] Basavaraj, Dr. S. Subrahmanya Swamy, *Design, static, and modal analysis of high speed motorized milling spindle*, International Research Journal of Engineering and Technology (IRJET), pp. 787-799, 2018.

[2] Hareesha, *Static and dynamic analysis of a cnc milling spindle*, International Research Journal of Engineering and Technology (IRJET), Volume: 02 Issue: 07, 2015.
[3] Tata McGraw-Hill, *Machine tool design handbook*, pp. 539-540-653, 1982
[4] Material database, *https://www.totalmateria.com*
[5] Milling tools, *http://www.mitsubishicarbide.com/*
[6] Milling tools and tool holders, *https://www.haimer.biz/*
[7] Book of spindle part 2, *https://dynospindles.com/vault/technical/Book-of-Spindles-Part-2.pdf*
[8] Super-precision bearing, catalogue SKF, 2016
[9] SolidWorks Simulation, *http://www.solidworks.com/education*

Analiza statică şi modală pentru un arbore principal de frezat CNC

Rezumat: *Lucrarea curentă analizează deformația statică minimă, a unui arbore principal de frezat pentru diferite poziții și distanțe ale rulmenților. Arborele este proiectat să funcționeze la 12000 rpm având puterea nominală de 25 Kw iar unghiul de contact al rulmenților fiind de 15˚, design-ul cartușului este realizat pentru port-scule de tipul HSK-A 63. Analizele statice și dinamice au un rol important în estimarea tensiunilor, a rezistenței precum și a deformațiilor arborelui principal de frezat, unde analiza statică poate determina rigiditatea sistemului iar analiza dinamică ne arată frecvențele naturale, modurile de vibrație la diferite turații ale arborelui de frezat și funcția de răspuns la frecvență. `Mai departe se vor studia deformațiile flanșei de capăt al arborelui, analiza statică şi modală prin calcule teoretice şi compararea acestora cu rezultatele simulării din SolidWorks.*

Claudiu-Ioan RUSAN, PhD Student, Eng., Technical University of Cluj-Napoca, Design Engineering and Robotics, 103-105 Muncii Blvd., 400641 Cluj-Napoca, email: rusanclaudiuioan@gmail.com, phone: +4 0755 178 271,

Cornel CIUPAN, Professor, Ph.D. Eng., Technical University of Cluj-Napoca, Department of Industrial Design and Robotics, 103-105 Muncii Blvd., 400641 Cluj-Napoca, email: cornel.ciupan@muri.utcluj.ro