Journal of Machine Engineering, 2019, Vol. 19, No. 4, 58–69
ISSN 1895-7595 (Print)   ISSN 2391-8071 (Online)

Received: *05 November 2019* / Accepted: *04 December 2019* / Published online: *20 December 2019*

preload, angular-contact bearings, machine tool

Pawel TUREK[1*]
Waclaw SKOCZYNSKI[1]
Marek STEMBALSKI[1]

DEVELOPMENT OF FEM MODEL OF AN ANGULAR CONTACT BALL BEARING WITH ITS EXPERIMENTAL VERIFICATION

The article presents FEM model of an angular contact ball bearing used in spindle systems with active preload control. A two-dimensional replacement model for a single rolling element was developed. Its elastic characteristics were determined and the stress distribution was presented for the FEM 2D model. Based on the elastic characteristics for a single rolling element, a complete 3D bearing was modelled. The substitute model of a bearing developed in this way was used to model the spindle system. The elasticity curve of this spindle was determined. The last stage of the work involved the experimental verification of the FEM model using a custom-built test bench, in which piezoelectric elements were used to preload the bearings.

1. INTRODUCTION

In machine tool spindles, their proper stiffness is an important parameter. For the correct operation of the machine tool, it is determined, for example, in an analytical manner as shown in [1]. These calculations are based on the principle of operation of angular contact ball bearings explained in detail in publication [2]. Based on the formulas contained in it, it is possible to determine the values of individual components of forces acting in a given bearing.

Modern machining centres often use electric spindles as their main drives, whose bearing system is usually based on angular contact ball bearings. These types of bearings can transmit both radial and axial forces, and require proper preload value for proper operation. This value strictly determines the working parameters of the spindle, and mainly its stiffness. If we assume that bearings work with a light [3] preload, used as described by the author of the publication in high-speed spindles, we get a small spindle stiffness, which results in the possibility of cutting smaller material allowances, but at a high speed. Using a heavy preload, the opposite effect will be obtained. The low preload also reduces the power loss of the

[1] Wroclaw University of Science and Technology, Department of Machine Tools and Mechanical Technologies, Wroclaw, Poland
[*] E-mail: pawel.turek@pwr.edu.pl
https://doi.org/ 10.5604/01.3001.0013.6230

bearings so that the spindle heats up less. Research in this direction has been included in [4]. The authors determined the relationship between preload and power losses based on the numerical model and experimental research.

Modern machine tools are increasingly equipped with various types of mechatronic systems. They are used, for example, to damp the vibration of a cutting knife on lathes using a magnetic actuator [5], to change the value of the preload of angular bearings using piezoelectric actuators and displacement sensors [6]. In [7], the authors describe the concept of changing the value of bearing preload using electromagnetic force. However, as they themselves note the magnetic actuator in the machine spindle can cause strong sticking of chips to its housing. One of the latest works presenting the mechatronic solution used in the machine tool spindle is the structure proposed in work [8], in which the authors using the piezoactuator and beveled spacer ring change the preload of the bearing spindle system in a controlled manner. The use of an active system to change the preload allows changing the spindle stiffness during its operation and to better adapt the preload to the current needs of the machining process. A large number of such solutions make you look critically at their various limitations. In [9], selected active and passive methods for changing or correcting bearing preload during spindle operation were compared.

The purpose of the paper is to develop a FEM model of an angular contact ball bearing that would be useful in numerical calculations of the static and dynamic properties of spindle units subjected to active preload of bearings, and an experimental verification using a real system.

2. REVIEW OF MODELS DESCRIPTIONS IN THE LITERATURE

Development of machine tool spindle models is a very common approach that helps determine many different properties. By using the theoretical knowledge on the subject of bearing systems in machine tools, presented in detail in [10, 11], and the increasingly common computational systems of various types, it is possible to significantly accelerate the performance of advanced engineering analyzes. These analyzes assist designers in developing prototypes and improving existing structures. They also allow you to describe the behavior of e.g. rotating parts inside a machine tool spindle. Descriptions of this type were described in detail by the author in [12, 13] where the behavior of various angular contact bearing models was explained based on the model.

In this and in similar cases, models come to the rescue. Of these, mainly analytical and numerical models can be distinguished. They are used to describe static and dynamic properties as well as thermal behavior of a given spindle. An example of an analytical model is included in the publication [14]. The authors assumed that the whole model will consist of a spindle shaft, angular contact ball bearings and a housing. The proposed model takes into account the gyroscopic effects, allows the application of cutting forces to the spindle, changing the preload tension and determining the frequency response of the system.

Over the years, there has been continuous development in each of the modeling areas. One of the latest works on analytical models used to describe the properties of angular contact bearings, which is worth mentioning is work [15] in which the author presents a detailed description of the distribution of forces prevailing inside the angular contact ball bearing. The continuous development of computers and increasing their computing power means that for simulation and determination behavior of machine tool components, models using FEM (finite element method) technology are quite often developed. It is based on discretization on finite elements, for which the solution is approximated by specific functions, and calculations are carried out only at nodal points. Using this technique, the bearing models presented were used for calculations in machine tool spindles. An example here can be the work [16], in which the developed model provides the ability to predict the instantaneous rigidity of bearings, and the static and dynamic behavior of the spindle depending on the current operating conditions. The model also enables simulation of cutting forces during spindle operation.

The use of the FEM method does not only apply to bearings or other individual machine components. Using this technique, you can successfully develop a complete machine model. Nowadays, there is a need to look at the machine tool as a whole in modeling. For this reason, there are attempts to develop a comprehensive model for the entire machine. This type of example can be found in [17], in which the authors emphasize the need for holistic modeling of machine tool thermal properties. It is to provide a solid foundation to support the design process of new machines.

In the development of numerical models using the FEM technique, there is still a problem in the dynamic description of the modeled bearing properties. This problem arises at the junction of a treadmill rolling element. Due to the lack of unambiguous information on the ball behavior in the angular contact bearing during its work, various researchers use various types of simplifications to describe this contact. They were presented extensively in [18], in which the author describes various replacement models and proposes his own concept for modeling of large bearings. The FEM model presented in the article uses simplification to eliminate bearing rolling elements and replace them with beam elements. The authors decided on this type of solution because it is simple to implement in Abaqus, and at the same time allows calculations to be carried out in an acceptable time.

3. DETERMINATION OF INITIAL ASSUMPTIONS FOR THE MODEL

As part of research on a spindle system with an active change in the preload of angular contact ball bearings, a model of a single bearing was developed. This was mainly associated with the fact that it is very difficult to determine the actual pressure value between the race and the rolling element in bearings. There is practically no physical possibility to measure this force and, control it during an active preload change. For this reason, the value of this force is determined analytically by developing more or less accurate models. A commonly used theory in this area is the Hertz theory, which assumes that:

- Bodies in contact are homogeneous, isotropic and linear-elastic,
- External surfaces of bodies around the point of contact are smooth with regular curvature,
- Body deformations are small,
- The contact surface in relation to the surface of the bodies is small,
- There are no tangential stresses on the contact surface, only normal stresses.

Knowledge of the phenomena prevailing in rolling bearings is necessary to determine the behaviour of a bearing at the contact place between the rolling element and the race. The total bearing capacity is stated by the manufacturer. Various types of simplifications are used to determine this value in an analytical manner, e.g. using FEM. Replacement elements are introduced, which are given specific properties, the symmetry of the structure is used, or the modelled element is reduced to a two-dimensional form. In the case of bearings, a practical approach is to use symmetry and to analyse a section of a bearing race with one rolling element.

Manufacturers of this type of bearing usually indicate in their catalogues the value of the permissible maximum bearing load in the respective preload ranges (light, medium, heavy preload). After determining the properties of such a bearing on the basis of product specifications, at points lying on the boundaries of the change in the preload type, we obtain step transitions that are physically never present in a real system. Therefore, the development of a model of a bearing is intended to better reflect its behaviour in real operating conditions. A correct model also allows obtaining valuable data related to the work of such a bearing in a machine tool spindle.

In order to build the numerical model of the tested spindle system, the following work was assumed:

- Determination of the stress distribution curve of a single rolling element,
- Construction of a substitute model of angular contact ball bearing,
- Development of a model of a spindle using a substitute model of bearings.

The use of bearing FAG B7206C is assumed in the paper. To properly model a single rolling element along with a section of the bearing race, it is necessary to determine the bearing dimensions. For the bearing adopted for use, the manufacturer does not provide such values in the specifications. For this reason, one of the bearings was divided into components (races, cage, rolling elements). Using the CMM measuring machine made by Zeiss, the dimensions of individual elements were measured (Table 1). Placement rings (internal and external) and five randomly selected balls were placed on the table. They were attached with clamps so that they would not move. Then, by approaching the probe to the curvature of the bearing race around 30 measuring points for each ring were collected. Based on these points, using the software of the CMM, the torus was interpolated, whose radius determined the radius of the bearing raceway. Each of the measurements was performed with five repetitions and in Table 1, mean value rounded to 0.01 mm.

Table 1. Bearing dimensions for the construction of a geometric model

Inner race radius [mm]	Outer race radius [mm]	Ball diameter [mm]
4.95 +/–0.01	5.04 +/–0.01	9.53 +/–0.01

4. 2D MODEL

The geometrical model of the bearing was used to build the FEM model in Abaqus software. The following assumptions were made:

- Identical material for the ball and bearing races,
- The bearing operates only within the range of elastic deformations,
- Ball rotation during bearing operation is not taken into account,
- The presence of a lubricant is not taken into account,
- The inner race remains fixed and does not move while the bearing is under load,
- Load is applied to the bearing by a displacement of the outer race.

The model of a bearing was developed as two-dimensional. The assembly consists of three elements (bearing rings and rolling element). They were set in such a way that there was no play between them. Then, based on the catalog data for the tested bearing, it was determined that in order to induce preload of the bearing system, corresponding to the average value, it was necessary to displace the bearing raceways by 6.82 µm. For this purpose, in the model the outer ring of the bearing was set displacement in the direction of the X axis, which allowed to obtain the correct preload value for the 2D model. Simplification has also been introduced, involving the use of symmetry in the design of the bearing. One rolling element was modelled in combination with the races. This allowed to significantly accelerate the calculations, while maintaining the accuracy of the results at a satisfactory level. CPS4R coating elements reflecting a flat stress state were used to build the model. Material properties are presented in Table 2.

Table 2. Bearing dimensions for the construction of a geometric model

Material property	Value
Density [kg/m^3]	7860
Young's modulus [GPa]	211
Poisson's ratio	0.3

All degrees of freedom were removed from the inner race, while the outer race could move in the direction of the X axis. Tangential behaviour with a coefficient of 0.002 (friction coefficient for bearings lubricated with grease) was applied between individual parts (ring – rolling element – ring) at the surface contact. Uneven density of the mesh built of tetrahedron elements was adopted (Fig. 1). At the contact points of the race with the ball, a single finite element had a dimension of 0.0005 mm.

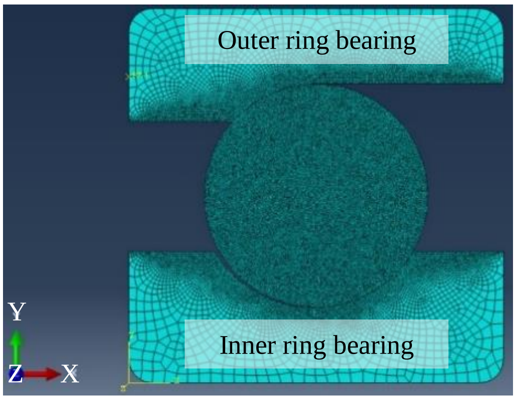

Fig. 1. Substitute model for a single rolling element – mesh view

The simulation was carried out for a static state in which the influence of centrifugal force on the bearing ball is neglected. The total analysis time was 1 second and was divided into 116 steps. Their number was selected for analysis automatically, assuming that a single step will last for a maximum of 0.01 s. The bearing considered in tests, according to its characteristics in the catalog [19], can work in three different ranges of initial stress (light, medium, heavy). It was assumed that the medium preload range will be used during its operation. On this basis, the range for which the maximum preload force is 412 N was focused on in the numerical model developed. According to the specifications, the bearing stiffness for the medium preload value is 75.5 N/μm. Calculations performed in Abaqus software were used to determine the value of the preload force and to determine the elasticity curve of the segment of the bearing race and one rolling element of the bearing. The elasticity curve for the entire bearing was obtained on the basis of the model, taking into account the number of rolling elements of the bearing being tested (in this case twelve). In this way, the model elasticity curve of the entire bearing in the range of 0–403 N was determined (Fig. 2).

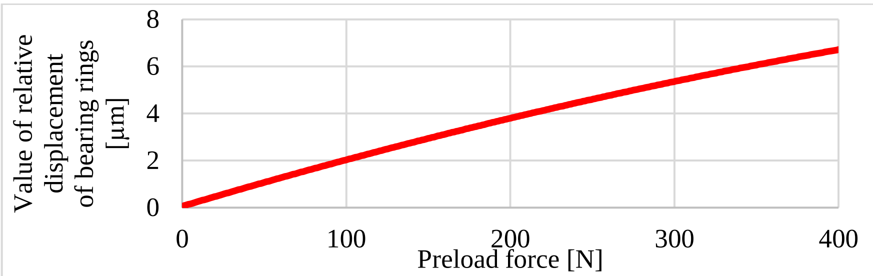

Fig. 2. Elasticity curve of the bearing determined on the basis of a numerical model

Based on the analyses, it was shown that the use of one value of bearing stiffness for the preload force range of 0–403 N does not cause significant errors (i.e. +/–0.06 μm).

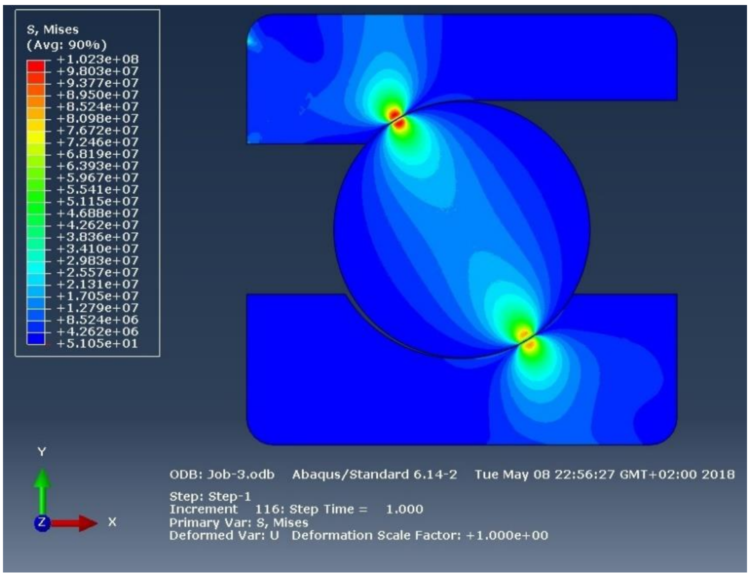

Fig. 3. Stress distribution when determining the curve for a single rolling element

Fig. 3 shows the stress distribution for a single rolling element. Here, we can see the characteristic load line in accordance with the bearing angle. The results obtained in this way are approximate. This is due to the fact that an elliptical surface is created at the contact place between the race and the ball (according to Hertz's theory), whose reproduction in a two-dimensional model is problematic. Based on the model, we can also observe slightly higher stress values at the contact place between the ball and the outer race of the bearing. The conducted numerical analysis showed that for medium preload value, the working angle of the analysed bearing is 14.6 degrees, and its results are presented in the SI system (Fig. 3).

5. 3D MODEL

With such a model prepared for a single rolling element, it was possible to make an attempt to develop a model of a full bearing. In order for this analysis to be carried out within the calculation duration acceptable to constructors, various simplifications and so-called substitute elements are used. In this case, the theory used in modelling of large bearings was used. The theory has been widely described in [18]. In it, the author proposes introducing an original super element: race – rolling element – race (BEB), which allows us to take into account all the significant phenomena occurring at the connection of these elements with each other. A schematic diagram of this type of element is shown in Fig. 4. Its nominal length is l_{BEB}, which is equal to:

$$l_{BEB} = r_{b1} + r_{b2} - D \quad (1)$$

where: r_{bi} – race radii, D – ball diameter.

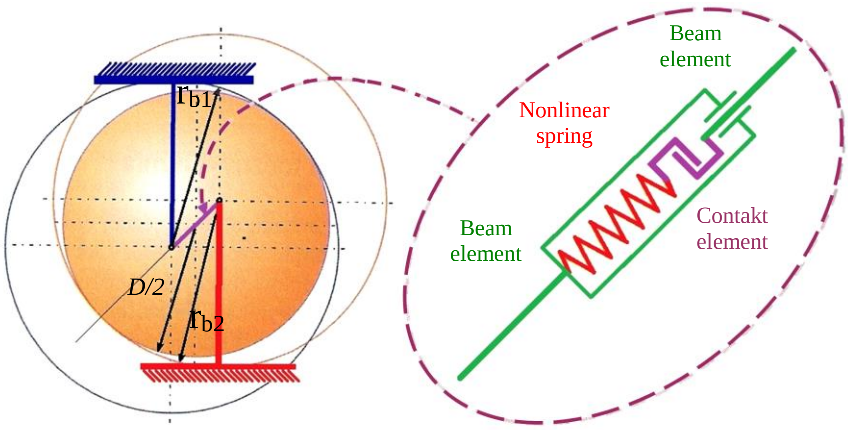

Fig. 4. Diagram of the race – rolling element – race BEB super element [17]

B7206C ball bearing was modelled based on the theory using the BEB substitute element. Based on [18], the substitute element was built using finite beam elements with articulations, an elastic element with nonlinear properties and a gap element. The calculations were carried out using the Newton-Raphson method [18]. All elements were adapted for the non-linear analysis. The calculation procedure consisted in increasing the displacement of the outer bearing race, which resulted in an increase in force at each iterative step. The non-linear elastic element (Fig. 5a) received a characteristic developed on the basis of the 2D model. The local coordinate system adopted for each beam element allowed determining the direction of this deformation and distributing the loads of this element. The connection of beam elements with bearing races was made based on the coupling procedure. An important stage of modelling was determining the radius of the circle (and not an ellipse as in Hertz's theory) using which the beam element will be "coupling" to the bearing races (Fig. 5b). This radius was selected on the basis of a number of analyses, and its value was the same for both races.

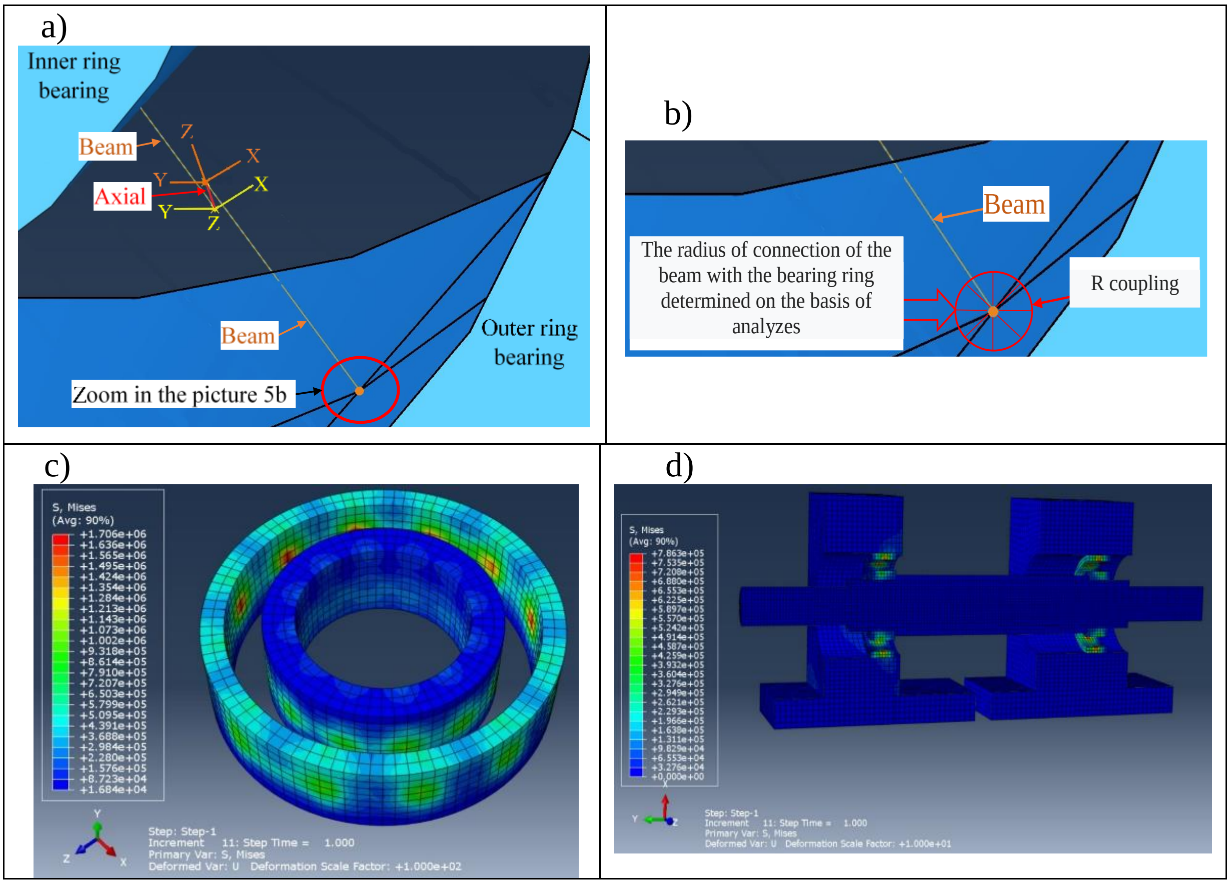

Fig. 5. a) View of the substitute elements used in the model, b) Explanation of the method of attaching the beam element to the bearing surface c) Stress distribution for the B7206C bearing tested, d) Stress distribution for the spindle system

C3D20R finite elements were used to build the numerical model. Material properties were adopted identically as for the 2D model. The bearing preload was exerted by displacing the outer race. Functions available in Abaqus software allow the distribution of displacements of a point, a node or their groups. Such point displacement can cause local deformation of the lateral surface of the bearing ring. To avoid this, the model adopts a substitute point to which the entire lateral surface of the bearing ring was "glued" using the coupling function. Then the displacement value according to the bearing specifications was applied to this point. Thanks to this, the same effect was obtained as if the bearing ring was displaced using a constant pressure value for the entire lateral surface. In this way, the stress value in the bearing was calculated as a result of inputting the preload value. Stress distribution is presented in Fig. 6.

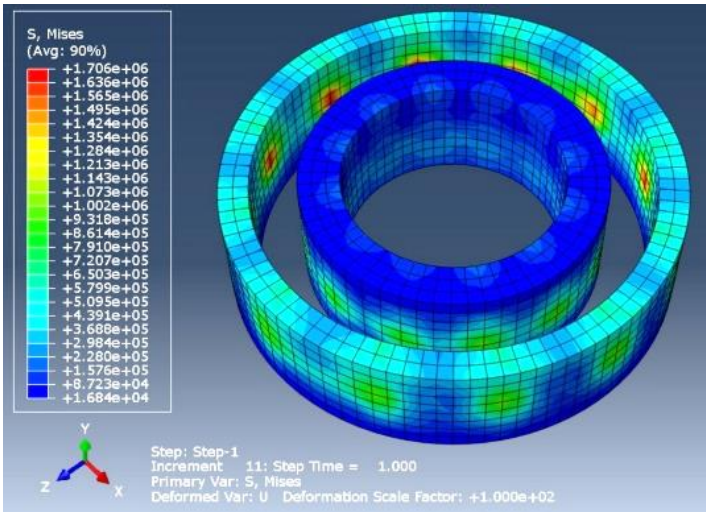

Fig. 6. Stress distribution for the B7206C bearing tested

After developing the model for a single bearing, the spindle system was modelled with two identical angular contact ball bearings with the O arrangement. In the case of the spindle model, the inner bearing rings were glued to the shaft using the tie procedure. The body was reinforced from below, and the preliminary analyses carried out showed that the simplifications applied to the real object did not affect the correctness of the model. For this model, lower maximum stresses were observed in the left (front) bearing support (Fig. 7). This is due to the elongation of the shaft under the influence of the preload force and the friction occurring between the outer bearing race and the spindle housing.

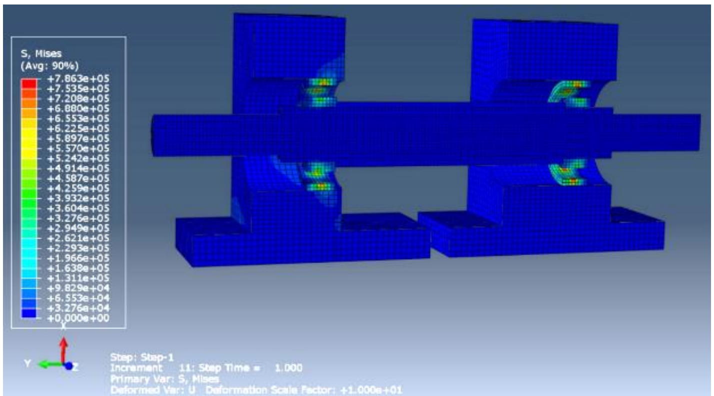

Fig. 7. Stress distribution for the spindle system

6. EXPERIMENTAL VERIFICATION OF THE MODEL

Correctness of the model presented was verified on the test bench, the schematic diagram of which is presented below (Fig. 8).

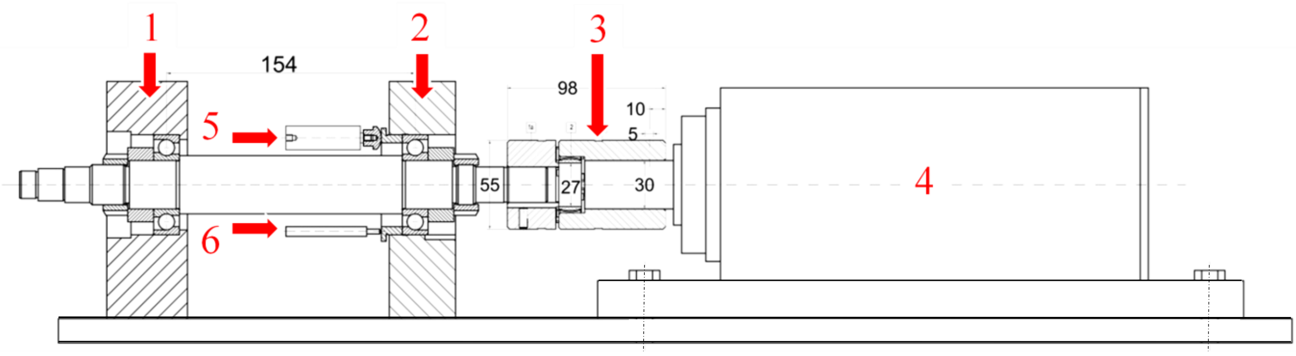

Fig. 8 . Schematic diagram of the tested spindle with a drive motor, 1 – Front support, 2 – Rear support, 3 – ROTEX coupling from KTR, 4 – C5160D-DB-PER32 high speed drive motor from TEKNOMOTOR, 5 – PSt 150/10/40 VS15 piezo actuator from Piezomechanik GmbH, 6 – MDKa - F1 displacement sensor from VIS

This system was used for research on the impact of an active change in bearing preload on the amplitude of spindle tip vibrations [20]. In the spindle system, three piezoelectric actuators PSt 150/10/40 VS15 from Piezomechanik GmbH equipped with a strain gauge were used to change the preload value. Each of them was calibrated on a separate calibration bench. The actuators were mounted symmetrically along the spindle axis. This enabled the bearing preload to be changed actively, as well as the constant monitoring of the preload force. In addition, tactile displacement sensors were installed inside the body, whose task was to continuously measure the displacement value of the bearing's outer race. These sensors were mounted in an additional housing attached to the rear bearing support. During assembly, they were positioned so that they would touch the sleeve through which the preload force was exerted on the bearing. Their measuring range was 4 mm and the measuring resolution was 0.4 μm. They were arranged symmetrically along the spindle axis and moved relative to the piezo actuators by 30 degrees.

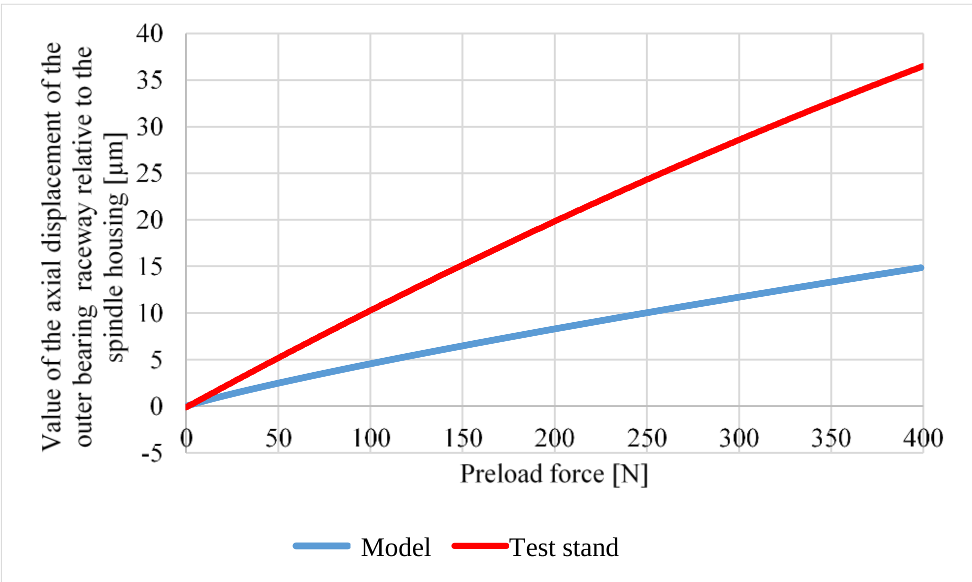

Fig. 9 Elasticity curve of the spindle system determined on the basis of a numerical model and of real spindle system

Each sensor had individual fastening, thanks to which it was certain that each of them remained in contact with the sleeve. This type of mounting sensors can lead to small errors because the sensors are not permanently attached to the sleeve and only touch it. This can lead to loss of contact with the measured element especially during high spindle speeds and sudden changes. This results in erroneous measurement results.

Before starting the measurements, the so-called starting position was set. In this position, the bearing system had a zero preload value. The preload force value is zero but there is no looseness in the system. Increasing the supply voltage in piezo actuators by 1 V resulted in recordable displacement values for the outer race and an increase in the preload force registered through the strain gauge system. Then the preload was increased in a stepwise manner by increasing the power supply in the range of 0–150 V by 30 V, which corresponded to 27 N (+/– 1 N) for each piezo actuator. As a result, the characteristics were obtained, which are given in Fig. 9. Comparisons with data obtained on the basis of the FEM model.

7. CONCLUSION

The presented measurements were carried out under stable temperature conditions, for five iterations, while maintaining each time the loosening of the entire system to eliminate any hysteresis error, which usually occurs in piezo actuators. The percentage discrepancies of the results for the consecutive five measurement steps are presented in Table 3.

Table 3. Percentage discrepancies in the results of the required displacement of the bearing's outer race for the numerical model and the real spindle system

Preload force [N]	0	80	160	240	320	400
Percentage [%]	-	48.3	41	42.6	40.5	40.8

Therefore, it can be seen from Table 3 that with the same preload values, the displacement of the bearing's outer race for the numerical model account for approximately 43% of displacements occurring in the real spindle system. This is due to the fact that the model does not take into account many aspects that occur in a real test bench. These include geometric errors associated with the implementation of bearing seats, errors associated with the correct determination of the contact point between the outer ring of the bearing and the piezoelectric element. In the model, the displacement method is always perfectly parallel to the spindle axis, which cannot be guaranteed in the test bench. With such small displacements, even minimal errors can lead to a slanting of bearings race when the preload force value changes. The model also assumes the tangential behaviour coefficient of 0.2 at the contact point between the outer race and the bearing body, which may be different in the spindle tested. Generally, however, the authors have concluded that the approach to modelling angular contact ball bearings in this way allows determining the correct values for a given individual bearing and can be successfully used for other types of angular contact ball bearings. When analysing bearing arrangements, taking into account a greater number of parameters would significantly improve the accuracy of the results obtained.

ACKNOWLEDGEMENTS

The calculations were made using the resources made available by Wroclawskie Centrum Sieciowo-Superkomputerowe (http://wcss.pl), calculation grant No. 109. The paper has been supported by Wroclaw University of Science and Technology from Statutory project No. 049U/0014/19.

REFERENCES

[1] ZHANG J., FANG B., ZHU Y., HONG J., 2017, *A comparative study and stiffness analysis of angular contact ball bearing under different preload mechanisms*, Mechanism and Machine Theory, 115, 1–17.
[2] JĘDRZEJEWSKI J., KWAŚNY W., 2010, *Modelling of angular contact ball bearings and axial displacements for high-speed spindles*, CIRP Annals–Manufacturing Technology, 59, 377–382.
[3] CHEN J. S., CHEN K. W., 2005, *Bearing load analysis and control of a motorized high speed spindle*, International Journal of Machine Tools & Manufacture, 45, 1487–1493.
[4] JIANG S., MAO H., 2010, *Investigation of variable optimum preload for machine tool spindle*, International Journal of Machine Tools and Manufacture, 50, 19–28.
[5] CHEN F., LIU G., 2017, *Active damping of machine tool vibrations and cutting force measurement with a magnetic actuator*, International Journal of Advanced Manufacturing Technology, 89/1–4, 691–700.
[6] HADI HOSSEINABADI A.H., ALTINTAS Y., 2014, *Modeling and active damping of structural vibration in machine tools*, CIRP Journal of Manufacturing Science and Technology, 7, 246–257.
[7] HWANG Y.K., LEE Ch.M., 2010, *Development of a newly structured variable preload control device for a spindle rolling bearing by using an electromagnet*, International Journal of Machine Tools & Manufacture, 50, 253–259.
[8] CIOU Y.S., LEE C.Y., 2019, *Controllable preload spindle with a piezoelectric actuator for machine tools*, International Journal of Machine Tools and Manufacture, 139, 60–63.
[9] TUREK P., SKOCZYŃSKI W., STEMBALSKI M., 2016, *Comparison of methods for adjusting and controlling the preload of angular-contact bearings*, Journal of Machine Engineering, 16/2, 71–85.
[10] HARRIS T.A., 2001, *Rolling bearing analysis (4^{th} Edition)*, John Wiley and Sons, New York.
[11] HARRIS T.A., KOTZALAS M.N., 2007, *Rolling Bearing Analysis, Essential Concepts of Bearing Technology*, Taylor&Francis Group, New York, 371.
[12] KOSMOL J., GATYS R., 2016, *Simulation studies on the effect of speed rotation on forces contact us in rolling bearing*, Inżynieria Maszyn, 21/1, 32–45, (in Polish).
[13] KOSMOL J., STAWIK K., 2016, *Influence of contact stiffness coefficient and coefficient of friction on contact forces in a rolling bearing*, Modelowanie Inżynierskie, 58, 65–74, (in Polish).
[14] CAO Y, ALTINTAS Y., 2004, *A General Method for the Modelling of Spindle-Bearing Systems*, Journal of Mechanical Design Transactions of the ASME, 126, 1089–1104.
[15] KOSMOL J., 2019, *An extended contact model of the angular bearing*, Journal of Theoretical and Applied Mechanics, 57/1, 59–72.
[16] ALTINTAS Y., CAO Y, 2005, *Virtual design and Optimization of machine tool spindles*, Annals of the CIRP, 54/1, 379–382.
[17] JEDRZEJEWSKI J., KWASNY W., KOWAL Z., WINIARSKI Z., 2014, *Development of the Modelling and Numerical Simulation of the Thermal Properties of Machine Tools*, Journal of Machine Engineering, 14/3, 5–20.
[18] SMOLNICKI T., 2002, *Physical aspects of coherence of large roller bearings and deformable support structures*, Oficyna wydawnicza Politechniki Wrocławskiej, Wrocław, (in Polish).
[19] Super precision bearings, FAG, MATNR 036884715-0000/SP 1/GB-D/2016061.5/Printed in Germany.
[20] TUREK P., 2019, *Modeling of dynamic properties of the spindle system with the active regulation of bearing stiffness*, PhD Thesis, Wrocław University of Technology.