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COMPARISON OF METHODS FOR ADJUSTING AND CONTROLLING THE PRELOAD OF ANGULAR-CONTACT BEARINGS

The effect of preload on the operating conditions of angular-contact bearings is described. The interdependences occurring during mutual displacements of bearing rings at different bearing contact angles are discussed. The criteria of the classification into low-speed and high-speed angular-contact bearings are presented. Selected methods of preloading angular-contact bearings are compared and discussed. Their advantages and disadvantages with regard to the active control and adjustment of the bearing system during its operation are presented. The structure of a developed test stand with a spindle assembly is described. Preliminary measurements of the behaviour of the spindle assembly during operation are reported. Conclusions are drawn and the direction for further research is indicated.

1. INTRODUCTION

Numerous designs of angular-contact ball bearing preload control can be found in the literature on the subject. In most cases, these are prototype systems working exclusively in specific conditions (a given narrow range of spindle rotational speeds, a very small preload force adjustment, etc.), patent pending design solutions or solutions where bearing preload can be adjusted during spindle operation, but it cannot be controlled or adjusted online. In order to develop a new solution which would enable the active adjustment of bearing preload a spindle system was designed and built.

Measurements were performed on a test stand to determine how the bearing race position and consequently the bearing preload change during operation. Proper commercial piezo actuators were selected on this basis. The results of the investigations of the interdependence between outer bearing race displacement and rotational spindle speed will be used to develop an algorithm for controlling the actuators.

Accompanying the work of rolling bearings tight initially from the theoretical described in, [1],[2],[5]. The article presents the results of the selected design solutions. Additionally, the authors present their own research station and test results.

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2. METHODS USED IN INDUSTRY

As the rotational speed of the spindle increases, so does the centrifugal force acting on the balls of the rolling bearings. Consequently, very light or light bearing preload must be applied in high-speed systems [3]. However, if the electrospindle is to be highly stiff in a low range of rotational speeds, medium or high preload should be applied. This means that variable preload needs to be used for operation in a wide range of rotational speeds. In this way one type of spindle can be utilized to a larger extent for different cutting speeds [4],[6], [7]. The preload of the bearings can be effected by means of, e.g., a pressure regulator and a hydraulic system, but this approach has several serious drawbacks [7]:

- no explicit interdependence between rotational speed and preload has been established due to flow disturbances in hydraulic systems,
- it is necessary to build an additional hydraulic system,
- voltage adjustment and control are usually effected stepwise,
- high costs.

In order to prevent uncontrolled changes in bearing preload in high-speed spindles one can use elastic elements to keep bearing preload constant in the whole range of spindle rotational speed. There are also designs in which the elastic elements do not act directly on the bearing and the preload force is transmitted via a special sleeve [4]. The latter moves in the housing, whereby the position of the bearings can be adjusted to, e.g., the changes in spindle temperature. The bearings fixed inside the sleeve remain constantly preloaded with the same force. However, the authors of such a design, described in detail in [4], found that due to defective workmanship, the action of dynamic forces and thermal expansion the sleeve can jam inside the housing.

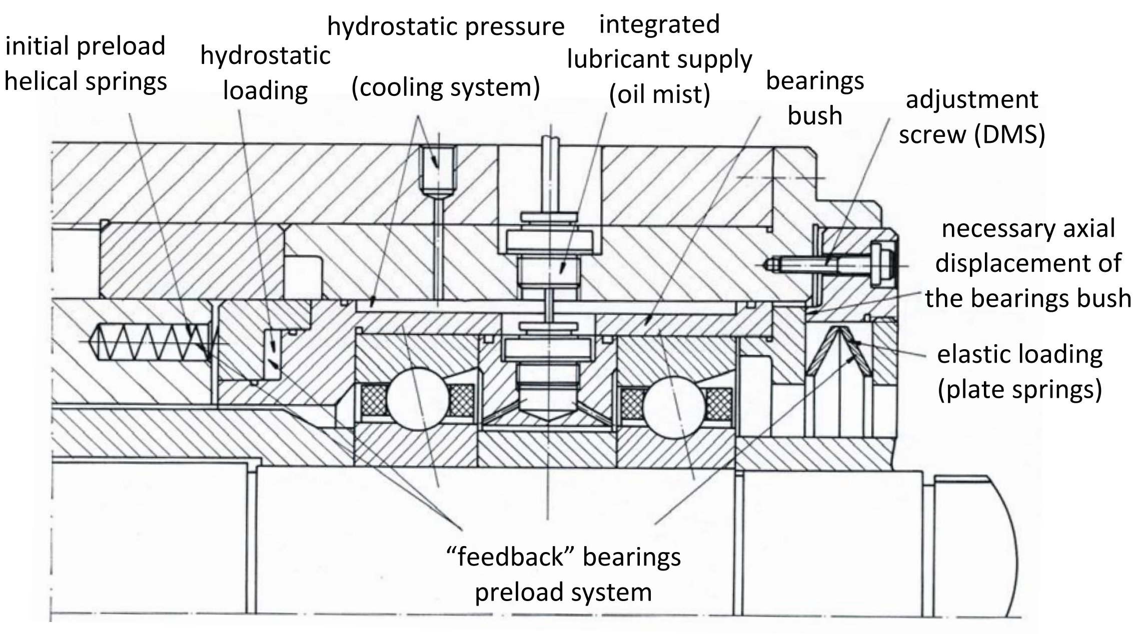


Fig. 1. Diagram of bearing system incorporating sleeve with special stick-slip reducing geometry [4]

Also the friction between the two surfaces inside the spindle can pose further difficulties in designing, analysis and numerical computations, whereby the originally assumed sleeved preload will be different from the actual one. In their manuals the leading manufacturers of bearings inform users that it is necessary to take thermal expansion into account. For example, SKF suggests that only one bearing race should be close fitted while the other one should be slack fitted. In [4] the authors propose a solution incorporating a sleeve with a variable geometry (Fig. 1), working in tandem with an additional hydraulic system the function of which is to eliminate stick-slip (sudden jumps in the velocity of slip) between the sleeve and the spindle housing. An additional purpose of the hydraulic system is to reduce preload in the course of spindle operation. Such a design is to ensure proper spindle operating conditions, but the authors admit that the proposed concept considerably increases system costs and the probability of system failure.

Therefore the idea arose to reduce the frictional resistance between the sleeve and the spindle housing by replacing the sliding coupling with a rolling coupling. In [9] the authors present a GMN spindle in which the bearing sleeve moves on additional rolling elements in the form of balls.

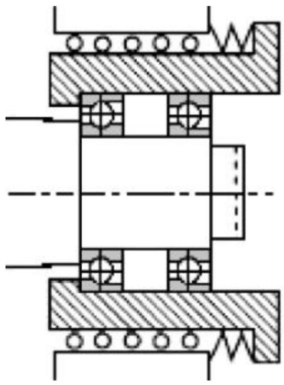


Fig. 2. Schematic of bearing sleeve moving on additional rolling elements [9]

Since this solution substantially reduces the effect of friction, it is increasingly often adopted by the leading electrospindle producers. At the same time it increases the complexity of the whole system. High precision is required in order to mount the sleeve on the rolling elements and an additional lubrication system must be built to lubricate the contact points. As the rotational speed increases, the surface pressures between the sleeve and the rolling elements change significantly, which may affect the precision of the axial movements of the sleeve. By replacing sliding friction with rolling friction in the points of contact between the sleeve and the housing one can eliminate stick-slip, but preload is applied via elastic elements which do not permit to actively adjust it during the operation of the electrospindle. Studies [8] of such spindle systems showed that as a result of abrupt (lasting for a few seconds) changes in rotational speed the spindle tip would shift markedly (a few tens of micrometres) in the axial direction. At changes in speed from 30 thousand to 50 thousand rpm the shift amounts to 50-60 μm. It is very difficult to eliminate shift at the design stage. The main causes of shift are: the low spindle stiffness necessitated by the required weak or very weak preloading of the bearings, and the increase in the centrifugal forces with increasing rotational speed. As the bearing balls press increasingly harder against the outer bearing race, the contact angle of the bearing changes and its stiffness decreases in the axial direction. As a result, the precision of the machine tool decreases considerably.

3. PROTOTYPE SOLUTIONS

Methods of actively or semi-actively introducing and adjusting preload are increasingly often developed. Attempts are made to replace the elastic elements with a more effective solution, which would make it possible not only to keep bearing preload constant, but also to control and adjust it. This can be achieved by using, e.g., various mechanical or mechatronic devices, whereby the bearing can be kept within the nominal contact angle range, which extends its life and limits its heating up and failures. Furthermore, active control and adjustment would make it possible to temporarily increase or decrease spindle stiffness in order to smoothly attain (or operate close to) the system's resonant frequency. In their catalogues the manufacturers of bearings specify the maximum speed at which the given bearing can operate under a given preload. On this basis one can increase the bearing preload, especially at low rotational speeds, and so obtain considerably greater system stiffness while maintaining correct bearing operation. On the other hand, if the preload is decreased, the bearing can work at higher rotational speeds and feed rates, whereby workpiece machining time can be reduced. Considering the advantages stemming from the use of active preload adjustment during spindle operation, successive researchers develop new designs of such a system. Each of the designs has its advantages and disadvantages, but many of the solutions are applicable to exclusively one particular spindle.

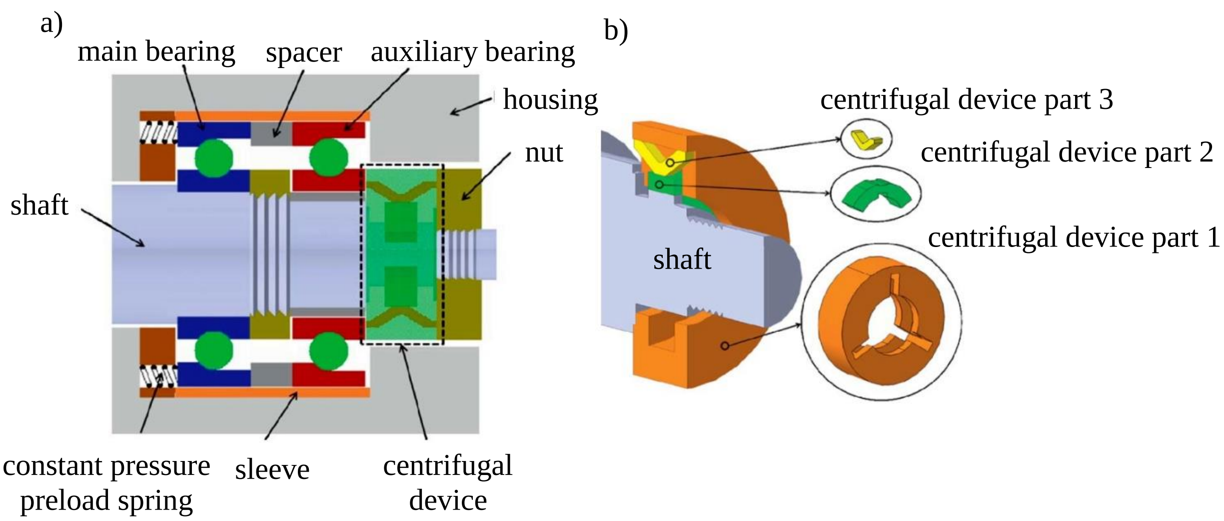


Fig. 3. Concept of automatic preload change by means of centrifugal force: a – schematic of complete system with bearings, b – structure of centrifugal ring [6]

One of the prototype structures based on the variable bearing preload idea, but not implemented through the commonly used elastic elements is the one described [6] and schematically shown in Fig. 3a. The design exploits the principle of centrifugal force action on a compliant element. As the geometry of this element changes with increasing rotational spindle speed, the preload of the bearings decreases.

The elastic elements used in this system press against the outer bearing race. There is a specially shaped ring (Fig. 3b) on the other side of the set of two bearings. As this ring acts on the inner ring, its geometry changes under the action of the centrifugal force, whereby the preload decreases. The preload is proportional to the spring rate and the ring's axial dimension. In this way preload can be decreased as rotational spindle speed increases. The drawback of this design is that preload cannot be adjusted in steady operating conditions. Therefore this method is only semi-active. The range of ring size changes is limited to a small elastic area, which significantly limits the range of adjustment. The universality of the concept is questionable because of the narrow ring pressure variation range.

Also the electromagnetic force can be used to control bearing preload. Such a solution was presented in [7]. The proposed device consists of an electromagnet, a movable part, a fixed part, a sleeve, a spring for applying constant preload and a spacer for holding the stretched spring in position. A schematic of the device is shown in Fig. 4. Preload can be adjusted by setting different values of the current flowing through the coil of the electromagnet controlling the spring compression degree. At a low rotational speed the compressed spring maintains a high preload. As the spindle accelerates, the electromagnetic force is decreased, the spring slackens and the bearing preload decreases. The value of the electromagnetic force can be calculated from the formula 1:

$$F = \frac{\mu_0 N^2 A_p i_0}{x_0^2}, \tag{1}$$

where:

F – the electromagnetic force,
μ_0 – vacuum magnetic permeability,
N – the number of windings in the coil,
A_p – the cross-sectional area of the coil's pole shoe,
i_0 – the current in the coil,
x_0 – the size of the air-gap.

According to the above relation, electromagnetic force F is inversely proportional to the square of air-gap x_0 and at the same time directly proportional to current i_0 flowing in the coil. Consequently, even slight changes in air-gap size or in the value of the current result in considerable changes in bearing stiffness. Therefore a very precise control system able to follow the mechanical system is required. It should incorporate a feedback system to correct the set values of the bearing assembly parameters in changeable operating conditions.

Unlike the solution described earlier, this system can be legitimately called a fully active adjustment system. The experiment carried out in [7] confirmed the high repeatability and speed of response of this device to the values set by the control system. However, it was found that during operation the device would cause the magnetization of not only the movable part, but also the elements interacting with it. This is an undesirable phenomenon, particularly in the case of cutting machine tools during the operation of which many chips are produced, since under the action of the electromagnetic field the chips can be attracted to the spindle.

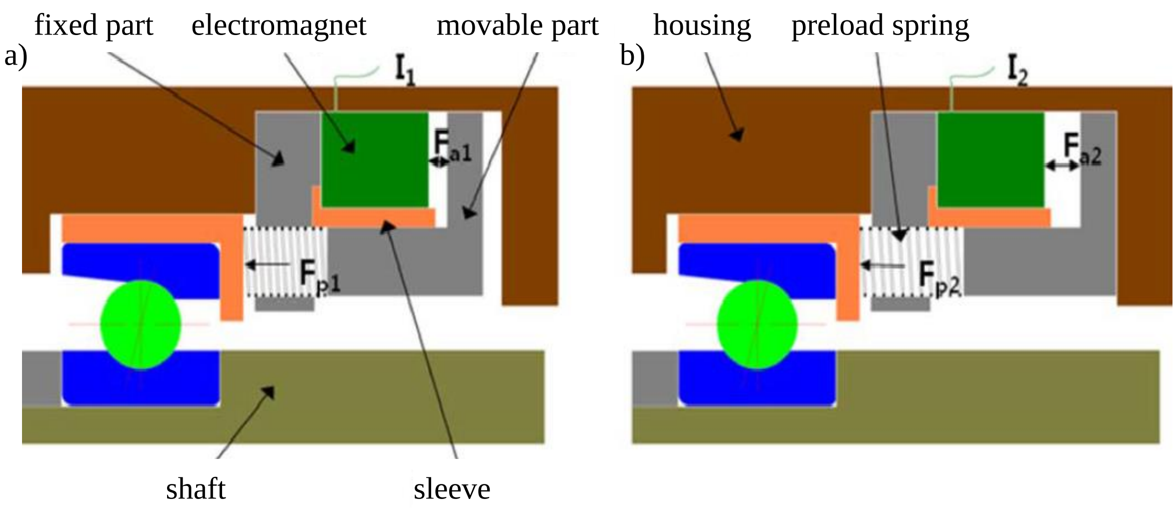


Fig. 4. Principle of preload adjustment by means of controlled coil current: (a) high preload at low speed, (b) low preload at high rotational speed [7]

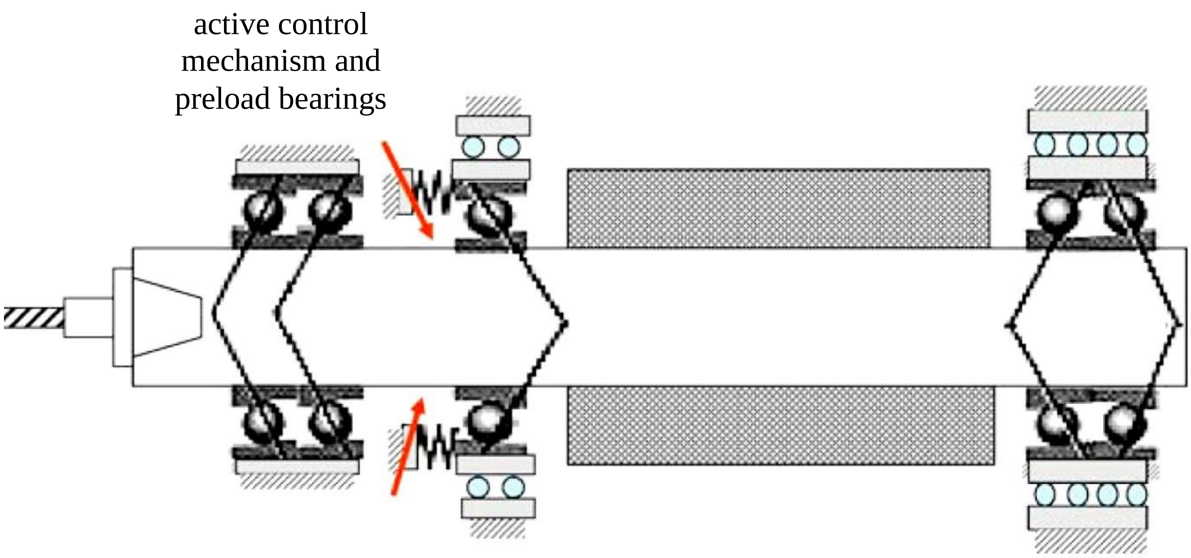


Fig. 5. System of electrospindle bearings with active preload adjustment mechanism [3]

Still another concept is used to change spindle system stiffness by actively tensioning an additional angular-contact bearing mounted at one third of the shaft length. The solution presented in [3] uses tensometric systems and piezoelectric actuators. The spindle assembly consists of two separate bearing sets. One of them operates in closed system "O" and is mounted in a special sleeve which makes it possible for it to move in the axial direction.

On the opposite side of the spindle the other bearing set is fixed directly in the housing and constitutes a half of system “O”. Its complement is the mentioned above additional single bearing which also can move in the axial direction owing to the fact that it is mounted in a sleeve. The preloading of the additional bearing is effected by a spring and is aided by an active adjustment system (Fig. 5).

Since piezoelectric actuators are characterized by a supply voltage hysteresis and drift, strain gauges operating in the full bridge system were used to actively adjust the bearing preload force as the piezo actuator length changed. Calibration was carried out on a separate test stand where the active preload system and the investigated bearing were mounted. The bearing is preloaded by three piezoelectric actuators set up at every 120° on the bearing perimeter (Fig. 6). Commercial force gauges made by Kyowa and a strain gauge were used to determine the characteristics of the piezo actuators. In this way the interdependence between bearing preload and piezo actuator length change was determined. On this basis an algorithm for the control of the active system was developed. The elastic elements in this structure are also used to protect the bearing system from active control system failure or power loss in the actuators.

An analysis of this solution shows that the individual bearings in this spindle will heat up with different intensity, but this should not directly affect spindle operation accuracy. Also the forces of the reaction between the bearing outer race and the spindle housing will differ because of the different mounting of the bearings, which may lead to different failure-free operation times. The two bearings situated closest to the tool jigging place are set directly in a socket in the housing and there is linear contact between the bearing ring and the housing. The other bearings are set in sleeves and additional rolling elements were incorporated between the sleeve and the housing so that the bearings can move freely in the axial direction. Consequently, the balls in abutment with the housing and the sleeve contact point, which may result in a different load distribution and a different elastic characteristic in the radial direction of the bearings.

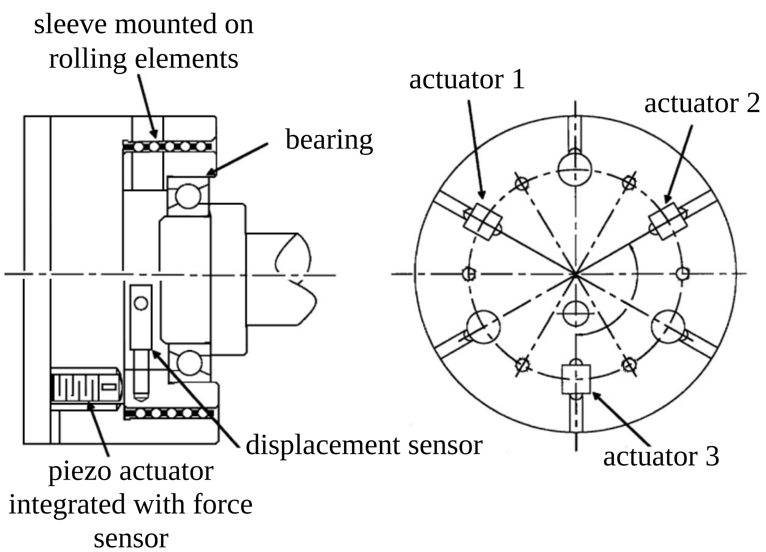


Fig. 6. Arrangement of actuators and displacement sensor on test stand [3]

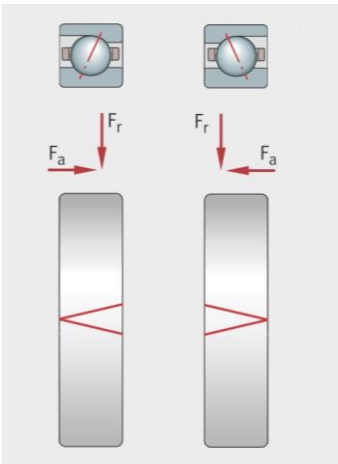
Thanks to the use of such a system of bearings as presented in [3], the effect of temperature can be largely eliminated since the elongation of the shaft will not cause a change in the preload of the bearings, but only the back spindle bearing support will shift slightly.

4. DESCRIPTION OF TEST STAND WITH SPINDLE ASSEMBLY

The prototype preload control systems described above have many advantages, but also some drawbacks. Practically, only one of them can be called fully active. The other systems respond to changing spindle operating conditions merely in a passive way and do not permit active bearing preload adjustment. For this reason the present authors decided to create a design which would ensure active preload adjustment. A test stand was designed and erected to study the effect of changes in the preload of angular-contact bearings on the operation of the spindle assembly. A specially constructed spindle incorporating angular-contact bearings made by FAG, with a contact angle of 15°, in the "O" arrangement [11] was used in the tests.

The bearings can work individually and also in tandem. For this purpose the housing and the spindle were so designed as to enable testing in the two bearing configurations. Initially single bearings were to be tested and so spacer sleeves corresponding to the dimensions (width and inside diameter) of the additional bearings were used in the places intended for the mounting of additional bearings on the shaft. The spacing of the bearings, i.e. the distance between the centres of gravity of the rolling elements in the spindle, was set at 0.154 m. The directions of the forces which a single bearing can transmit are shown in Fig. 7a. The basic specifications of a single bearing are presented in Table 1.

a)



LP PP

LP – left bearing support
PP – right bearing support

b)

Fig. 7. Bearings: a) bearings operation system [8], b) actual bearing

Table 1. Specifications of bearing incorporated into spindle assembly [8]

Type of bearing		B7206C
Contact angle [^{O}]		15
Weight [kg]		0.19
Specific dynamic capacity C_r [kN]		23.2
Specific static capacity C_{0r} [kN]		14.6
Maximum speed for grease/oil [rpm]		24000/38000
Preload F_v [N]:	light preload	122
	medium preload	412
	heavy preload	856
Axial stiffness C_a [N/μm]	light preload	42.1
	medium preload	75.5
	heavy preload	112.3

Grease FAG Arcanol MILTITOP (base oil viscosity: 85 mm^2/s and operating range: -40 to +150°C) was used in order to simplify the model spindle design. Thanks to this, additional cooling and lubricating systems could be eliminated. On the other hand, grease significantly affects the maximum rotational speed of bearings, which also depends on the type of preload applied to the system of bearings, as shown in Table 2.

Table. 2. Maximum rotational speed with correction factors for grease taken into account [8]

Bearing preload	Max. speed [rpm]:
Light	19200
Medium	16800
Heavy	12000

Figure 8 shows the investigated spindle with a drive system. The spindle housing was made in the form of two separate cases, i.e. front bearing housing (4) and back bearing housing (3), mounted on a base plate. Thanks to this solution spindles of different length can be tested. The mounting holes in the base plate enable changes in the distance between the bearing housing, being a multiple of 0.05 m. Each of the bearing housing can be positioned by means of locating pins with an accuracy of below 0.01 mm in the axial and radial directions. In addition, each of the bearing housing is equipped with a thermocouple recording the temperature of the bearing outer race during operation. A high-speed three-phase induction motor made by TEKNOMOTR, with a Siemens frequency converter drives the spindle to ensure smooth rotational speed adjustment. The motor has a dynamically balanced rotor and its bearing assembly (ceramic bearings) enables the transmission of axial and radial forces. Power from the motor is transmitted to the spindle via a Rotex clearance-free claw clutch (2) designed for work at elevated rotational speeds owing to the use of a clamping sleeve (no key-type joints). This ensures flexible coupling between the drive unit and the tested spindle.

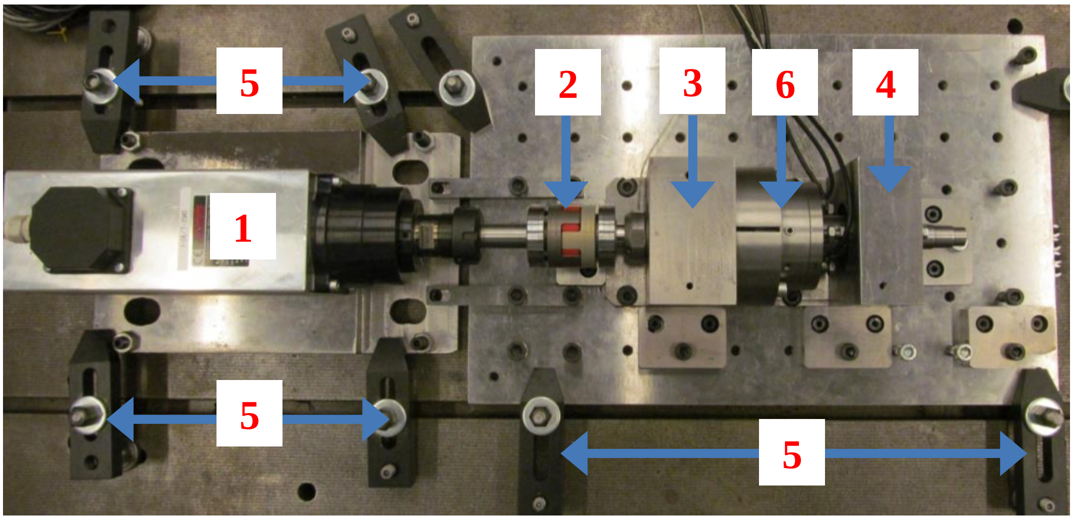


Fig. 8. Tested spindle system, 1 – motor, 2 – clutch, 3 – back bearing housing, 4 – front bearing housing, 5 – mounting clamps, 6 – preload system

Because of its weight the spindle system was installed on a metal table with T-slots in which clamps (5) fixing the base plate of the spindle were mounted. The same solution was adopted for fixing the spindle driving motor to the table.

In order to ensure work safety the inverter was placed in a control panel attached to the side of an aluminium table made of aluminium profiles. The drive's master power switch, a safety cut-out switch and a simple (ON/OFF) inverter control panel were installed in the cabinet. The table is equipped with castors allowing it to be moved freely. A computer with monitors, enabling spindle control and measurement recording, were placed on the table.

During the planned tests of the behaviour of the bearings three actuators arranged symmetrically relative to the spindle axis will be used. Each of the actuators consists of a housing, a piston and an elastic element enabling preload change through spring compression. The spring is compressed by means of a screw with a known thread pitch. Various springs are to be used in the tests in order to study the behaviour of the system of bearings under a wider range of preloads. The force exerted on the tested bearing by the actuators introduces preload by displacing its outer race and it depends on the change in the actuator length. Three wire touch inductive sensors arranged (similarly as the actuators) symmetrically at every 120 degrees relative to the spindle axis are used to measure the displacement of the outer race. The sensors' measuring range is 4 mm and their measuring accuracy is 0.4 μm.

The interdependence between the force exerted by the actuator and a change in the length of the spring was determined using an HBM U2A force gauge with a measuring range of up to 1 ton. The actuator characteristics for different springs are shown in Fig. 9.

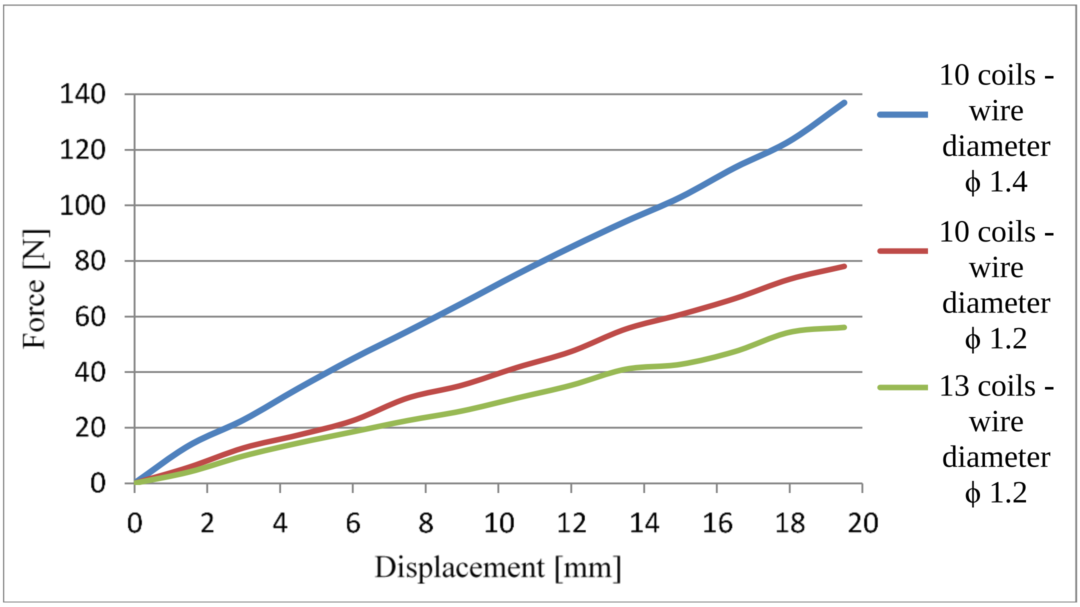


Fig. 9. Elastic characteristics for different springs

Table. 3. Range of forces produced in actuator workspace for different springs

Displacement		Force exerted by springs [N]		
[mm]		10 coils – wire ϕ 1.2	13 coils – wire ϕ 1.2	10 coils – wire ϕ 1.4
	0	0	0	0
	1.5	5.78	4.04	13.56
	3	12.72	9.82	22.83
	4.5	17.34	14.45	34.11
	6	22.54	18.5	44.8
	7.5	30.64	22.54	54.64
Actuator operational range	9	35.27	26.01	64.75
	10.5	41.62	30.64	75.15
	12	47.41	35.26	84.99
	13.5	55.5	41.05	94.24
	15	60.71	42.78	102.92
	16.5	66.48	47.41	113.61
	18	73.42	54.34	123.15
	19.5	78.05	56.08	136.95

Table 3 specifies force values for a given change in spring length. Bearing system can`t work with zero preload. Because of it from all the characteristics only a certain range - a shift from 9 do16.5 mm. was excluded. This range will be used to the study. This area was marked as the spring actuator operational range.

5. MEASUREMENTS

Ultimately the model spindle is to be preloaded by piezo actuators. In the current version of the test stand passive elastic elements described in sect. 4 were used to determine the spindle system characteristic, mainly the relative displacement of the bearing races during operation. A spring with 10 coils and a wire diameter of 1.2 mm was used to preload the bearing system. The preload of approximately 105 N and 200 N was applied. Six speeds (3000, 6000, 9000, 12000, 15000 and 18000) were distinguished in the spindle rotational speed range of 0-18000 rpm. The displacement of the outer bearing race was measured for each of the speeds. A National Instruments card and a control panel implemented in LabVIEW were used to control the motor driving the spindle. Displacement was measured by three inductive sensors in a setup with a universal amplifier Quantum X made by HBM. HBM's Catman 4.1 software dedicated to the amplifier was used to record the measurements. The sampling frequency was set at 9600 Hz. In addition, thermocouples were installed in the places where the bearings were mounted in the housing in order to control the operating temperature of the bearings. The main specifications of the measuring system are presented in Table 4.

Table 4. Measuring system specifications

Sensor resolution	0.4 μm
Measuring range:	0 - 4 mm
Sampling frequency	0.1 – 19200 Hz
A/D converter	24 bits
Supply voltage	10 – 30 V
Nominal operating temperature	-20 - +60°C
Max. length of data transmission cable	100 m

The measurements which were carried out show that in the investigated bearing system a change in preload from the nominal value (about 105 N) to the maximum value (about 200 N) results in the shift of the outer bearing race by about 22 μm in the positive direction. For illustration purposes Fig. 10 shows a schematic of the bearing system with the axes marked.

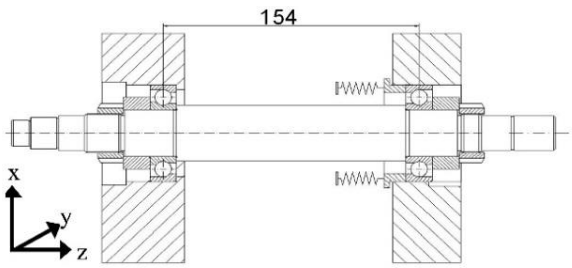


Fig. 10. Schematic of the tested spindle bearing system

By carrying out measurements on the test stand it was possible to determine how much the outer bearing race displaces as a result of the increasing (with rotational speed) centrifugal force acting on the rolling elements. The displacement values for the particular speed ranges are shown in the diagrams below (Figs 11 and 12, the preload amounting to about 105 N and 200 N, respectively). Figures 11 and 12 show the measurement variable cycle after the initial warm-up and stabilization of the heat of the spindle at a speed of 3000 rpm. Before the cycle of warming displacement sensors were calibrated on the same instrument.

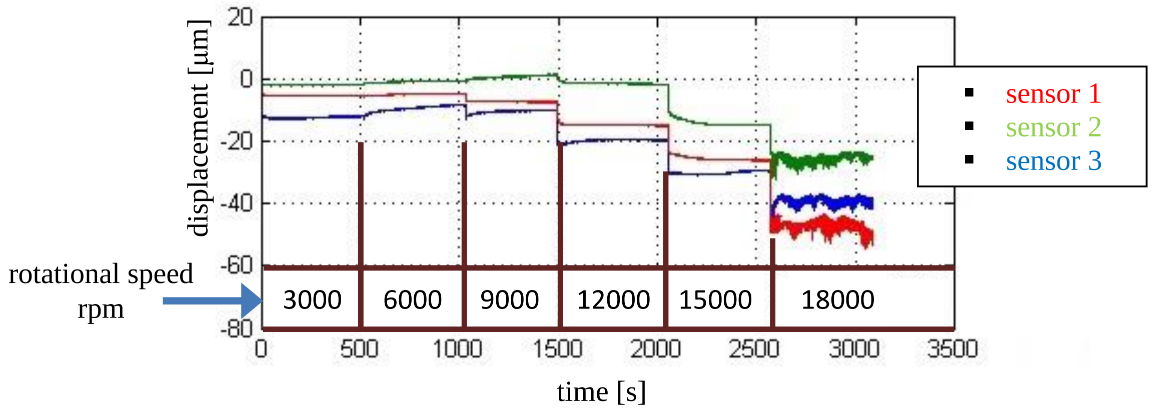


Fig. 11. Relative displacements for bearing ring preload of 105 N in three points (three sensors) at spindle rotational speed increasing from 3000 to 18000 rpm

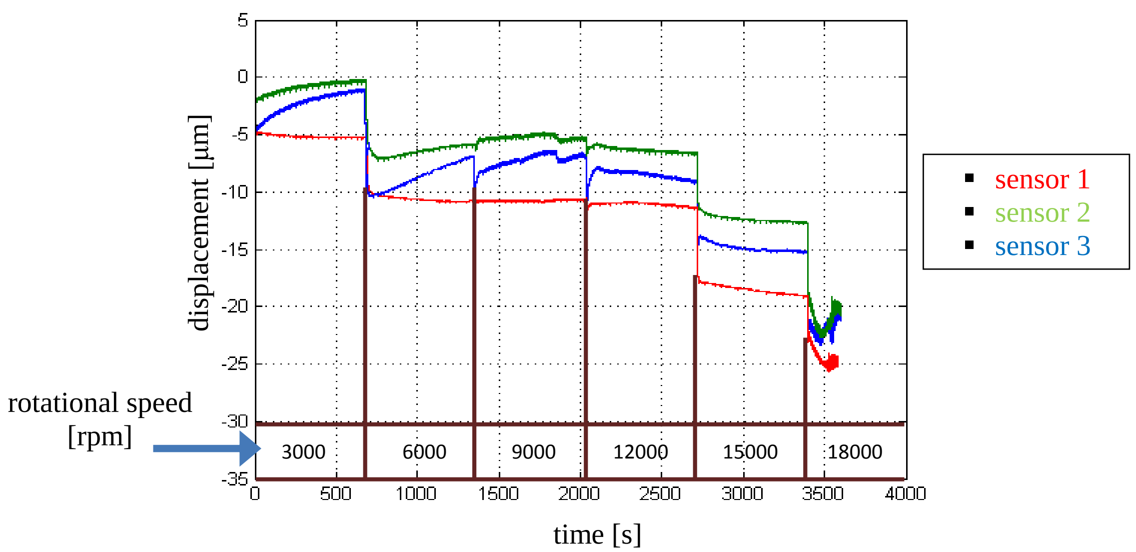


Fig. 12. Relative displacements for bearing ring preload of 200 N in three points (three sensors) at spindle rotational speed increasing from 3000 to 18000 rpm

The diagrams show that each time the speed is increased the bearing outer race shifts towards negative Z-axis values. As the rotational speed increases, so does the bearing temperature. The interdependences are presented in Figs 13 (preload 105 N) and 14 (preload 200 N).

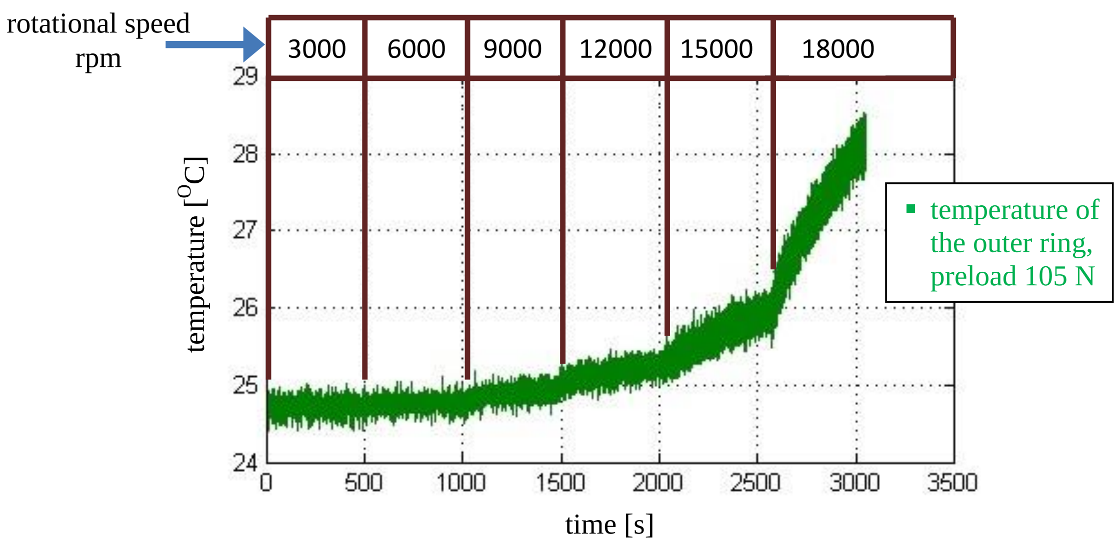


Fig. 13. Dependence between bearing temperature and spindle rotational speed of 3000-18000 rpm

The graphs show that at a lower preload value the temperature of the outer bearing race increases more rapidly. This may suggest that this preload value is insufficient for the particular bearing system. Consequently, the rolling elements undergo slip, whereby they excessively heat up the bearing races. It may happen that the bearings will roll with not all their circumference being on the bearing race, which also considerably increases the working temperature of the bearings.

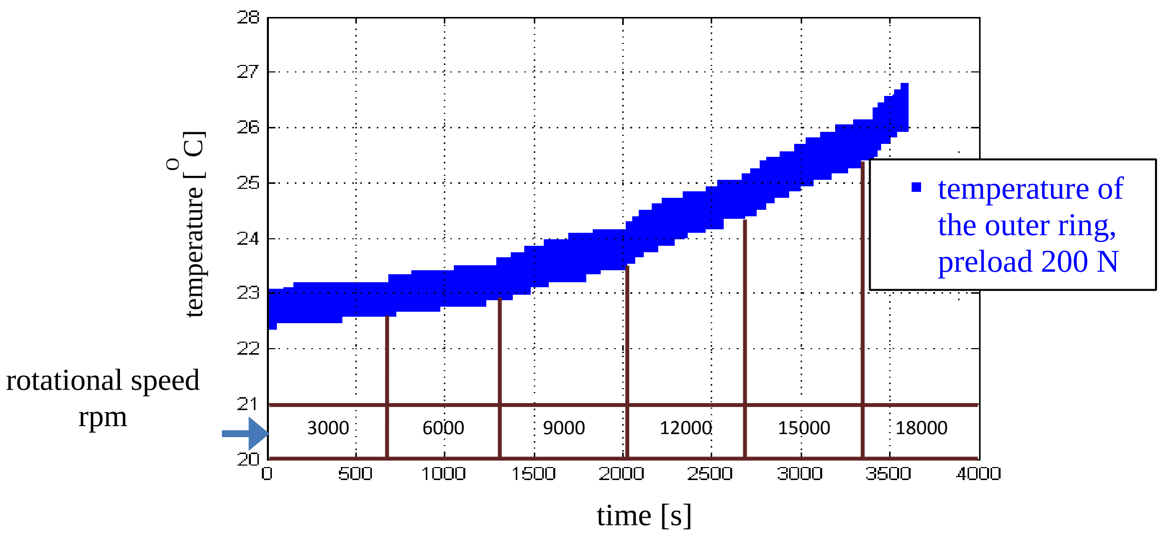


Fig. 14. Dependence between bearing temperature and spindle rotational speed spindle of 3000-18000 rpm

6. CONCLUSIONS

On the basis of the measurements presented in this paper one can determine the size of elongation of which a piezo actuator should be capable in order to be used to adjust the preload of the bearings during the operation of the system. The test stand was found to have several shortcomings which should be remedied prior to the next stage in the tests and active system construction. One should mention here the need for retainers to help fix the drive to the tested spindle. In the test stand as it is now each time the motor and the spindle are disconnected, it is necessary to carry out a long-lasting alignment procedure. The fixing of the displacement sensors is not fully rigid, which causes measurement inaccuracies, particularly at speeds above 1000 rpm. It would also be worthwhile to repeat the measurements, using a more flexible jaw clutch insert. This would reduce the adverse effect of the imprecise alignment of the spindle and motor shafts.

The authors succeeded in confirming the occurrence of angular-contact bearing opening in high-speed systems, widely reported in the literature on the subject. This phenomenon causes a reduction in angular-contact bearing stiffness in the axial direction and a simultaneous increase in radial stiffness. Proper measurements will enable further work on the design and implementation of a system of the active adjustment of bearing assembly preload depending on the speed, and of the possible correction of the preload. As a result, it will be possible to change the stiffness of the spindle to ensure operation at the minimal required preload, which will extend the life of the bearings.

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