The test rig for the research of the spindle bearings behavior.

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Abstrakt

This paper presents a test rig, which was developed for research thermo-mechanical behavior of spindle bearings lubricated grease and system oil-air. Angular contact bearings are most commonly used in high speed spindles due to their low friction and cost-effective maintenance. Besides a motor the bearings are the main source of heat in spindle, leading to thermal expansion spindle parts. Dilatation of the shaft and bearings changed of preload, which caused change friction and dynamic stiffness of the bearings. The ability to correctly predict the thermal-mechanical states of the spindle is one of the main precondition for optimization of design spindles and increased condition states. The results of the experiments for lubricated grease and system oil-air are compared with thermo-mechanical model of bearings.

Keywords: Spindle; Spindle Bearing; Lubrication; Test Rig

1. Introduction

The spindles are key component in the machine tool. The reliability of the spindle is determined mainly by the reliability of the spindle bearings. In the field of spindle technology occurred to a relatively great of progress during last 30 years (**Fig. 1**), from the belt driven spindle, to the motor spindle with integrated torque motors [1].

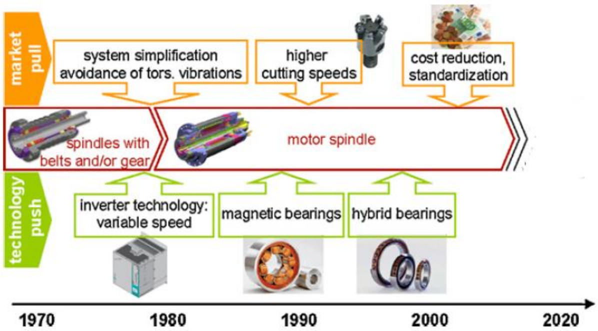

Fig. 1. *Historical review of spindles.***[1]**

This progress enabled to reduce the installation dimensions for spindles and increase the maximum speed of milling spindles. Increase the maximum speed was caused by the development of hybrid bearings with ceramic balls (Si_3N_4), which are lubricated direct oil-air lubrication (DLS- Direct Lubrication System). Oil-air lubrication system used for high-speed machining, where achieved the speed factor A > 2 000 000 mm/min.

$$A = n \cdot d_m = n \cdot \frac{(d+D)}{2} \quad (1)$$

The current trend in the field of spindle technology is aimed at increasing reliability, service life and prediction maintenance while monitoring operating conditions [1].

That is one reason virtual prototyping, to simulate and predict subsequently the spindle behavior [2]. Another reason for modeling and analysis of spindle is to optimize the dimensions, maximizing dynamic stiffness and usage power in cut or modeling of the cutting process [3,4]. Detailed models of spindles can be effectively used also for the analysis balancing of the dynamic properties of spindle and structure of the machine [5].

The most important prerequisite for the creation of advanced simulation models for the thermo-mechanical behavior of spindle is bearing model. In addition to the electric motor, friction in the bearings is the main heat source in spindles (**Fig. 2**), leading to thermal expansion of spindle parts. Heating of the bearing influences the final reliability of running of spindle. In the cased of thermo-mechanical instabilities due to the exponential increase of temperature on the bearings, can lead to damage bearings [6].

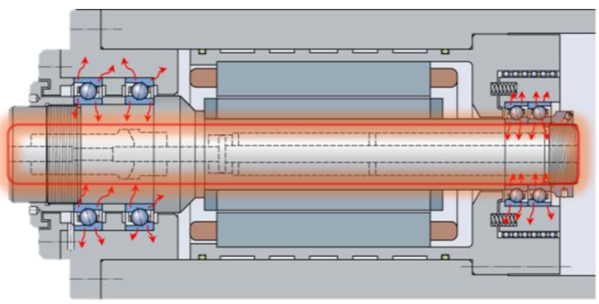

Fig. 2. *The spindle bearings behavior .*

Thermo-mechanical model of bearing (spindle) describe the overall behavior of the system, in which occurs due to friction loss in the bearings to thermal growth, caused to thermal expansion, a change of the contact force and contact angle of the bearing, leaded to change in friction loss and thermal growth. This created a closed loop of thermal effect in spindle bearings (**Fig. 3**)

The basic description of the behavior of angular contact bearings published Harris [7], the theory of interaction between the balls and the ring is based on the Hertz

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theory. The basic relationship describes the normal load of deformation balls railways:

$$Q = k \cdot \delta^{3/2} \tag{2}$$

Several studies have been published which focus on developing a thermo-mechanical spindle model in order to predict spindle behavior under various operating conditions. Bossmanns and Tu presented [8] a model of heat transfer among spindle parts, but did not consider the feedback effect on the thermal properties of the bearing. Shin's group presented a Finite-Element (FE) model which considers the interaction between heat, thermal growth and resulting steady state stiffness changes in the bearings using two-dimensional axisymmetric elements [9,10], similarly Jedrzejewski et al. [11]. However, the transient changes, which lead to thermal instability (**Fig. 4**)and bearing seizure, were not considered.

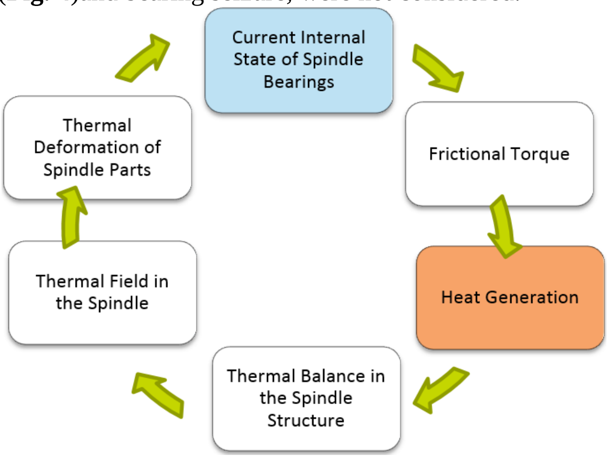

Fig. 3. *Closed loop of thermal effect in spindle bearings.*

The existing model Holkup [12,14,15] allows the prediction of temperature distribution and thermal growth, together with transient changes in bearing stiffness and contact loads under specified operating conditions. The proposed predictive thermo-mechanical model allows a transient simulation which considers the bearings, the spindle parts and their thermo-mechanical interaction. It is based on the mechanical model of rolling bearings uses Jones' [13]

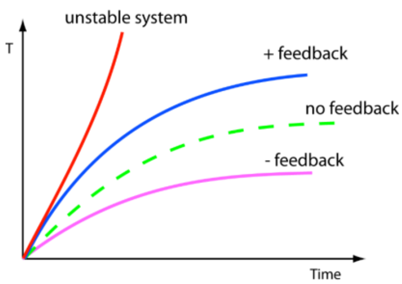

Fig. 4. *Graph stable and unstable of spindle behavior.*

The practical implementation of the mentioned state-of-the art models still remains a challenge. The main issue for the mechanical model of the bearing is its specific internal geometry. The main issue for the thermal-mechanical model of the bearing are unknown losses caused internal heat generation. These losses are dependent on the rotational speed, preload, external temperature, type and quantity of lubricant. The main issue for the thermal-mechanical models of the spindle are complex influence of all spindle parts and working conditions.

Motivation and the aim is the proposal of the test rig for enabling identification thermo-mechanical behavior of spindle bearings lubricated grease and system oil-air.

2. Heat sources in the bearing model

The issue of thermal preload arises especially in hard preloaded assemblies of angular contact bearings and may cause a potential threat to thermal stability of the spindle. Thermal expansion of bearings and spindle components produce a change in preload and thus in the quantity of heat produced. This in turn affects the temperature field in the spindle. Subsequent thermal deformations get into a closed loop of cause and effect. As a result of this, the bearing assembly can get unstable and overloaded with excessive wear or bearing seizure can occur. This phenomenon is known as thermally induced preload.

The internal state changes significantly in high speed angular contact bearings compared to static state. The internal contact angles and contact forces of rolling elements change due to centrifugal forces. The external contact force usually grows with a decreasing external contact angle, and the internal contact angle grows simultaneously. This leads to a significant change in bearing stiffness, which determines the dynamic behavior of the spindle. If the radial loading of the bearing is missing, the symmetrical state is considered. In any other case the state of rolling elements is different and each ball has different loading which causes a change in the position of ball excursion [14,15].

The bearing model solves the internal state of the bearing based on its external boundary conditions. This model is based on a system of the following equations:

- Equations of normal contact deformation
- Equations of force equilibrium for rolling elements
- Equations which describe internal bearing geometry, i.e. relationship between the relative position of the centers of curvature of bearing rings and rolling elements and contact deformations

Assumptions of the thermal model:

- The model of the bearing is axisymmetric.
- The heat source is separated between the ball and its raceways – separation is done for inner and outer contact.
- Heat power in one contact is divided in halves.
- The temperature of balls is homogeneous.
- The heat resistance of contacts of rings with the shaft or housing are dependent on the current state of the bearing.

Models heat sources can be verified only indirectly through measurable variables (temperature, torque, preload). The aim is to measured the greatest number of parameters, operating conditions for the possibility of more extensive compared with the model.

3. Test rig

Research thermo-mechanical behavior of spindle bearings lubricated grease and system oil-air and for verification the model of head produced in bearings proposed the test rig (**Fig.4**).

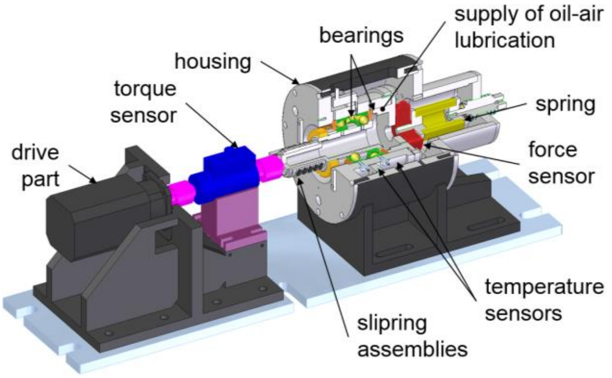

Fig. 4. *Test rig main components.*

This proposed test rig allows to measured the passive friction of angular contact ball bearings and measured the key parameters (torque moment, temperature, preload) for prediction the thermal behavior of the bearings due to the supplied quantity of lubricant.

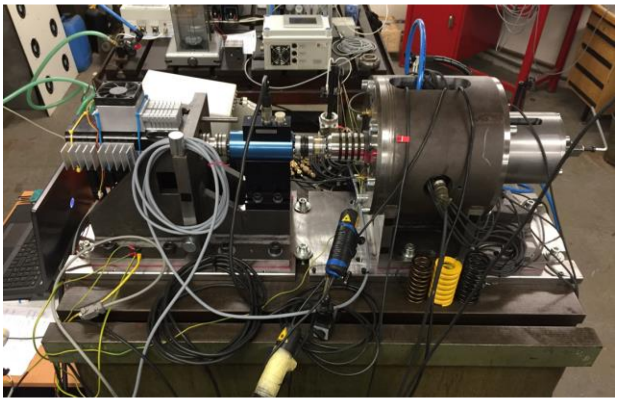

Fig. 5. *Measurement of test rig*

The test rig (**Fig. 5**) was designed for testing specific pairs of spindle bearings with an inside diameter of 70mm, from FAG manufacturer B7214C-T-P4S with contact angle α = 15°; B7214E-T-P4S with contact angle α = 25 (**Tab.1**).

Tab. 1. *Bearings parameters.***[16]**

	α [°]	$n_{G\ grease}$ [min^{-1}]	$n_{G\ grease}$ [min^{-1}]
B7214C-T-P4S	15	11 000	18 000
B7214E-E-P4S	25	10 000	17 000

The design of the test rig is proposed as coaxial arrangement. The test rig is aligned on a panel connecting the drive part and the test part of the rig. The drive part consists of the VUES 2AM406B-S motor with parameters listed in (**Tab.2**) and torque sensor KISTLER 4503A with increased sensitivity at low torques (with two nominal ranges) is placed between the couplings on the drive shaft.

Tab. 2. *Motor technical data*

Engineering data			
Rated torque	M_N	0.7	Nm
Rated speed	n_N	25,000	rpm
Limiting data			
Max. torque	M_{max}	7	Nm
Max. speed	n_{max}	40,000	rpm

The test part of the rig consisted of the drive shaft, which are resiliently preloaded the bearings in the “X”-arrangement. Bearings preload is solved by means coaxial spring causing a preloading force in the bearings.
Test rig is designed for measuring the (**Fig. 6**):

- Temperatures in three regions
- Torque
- Force
- Acceleration.

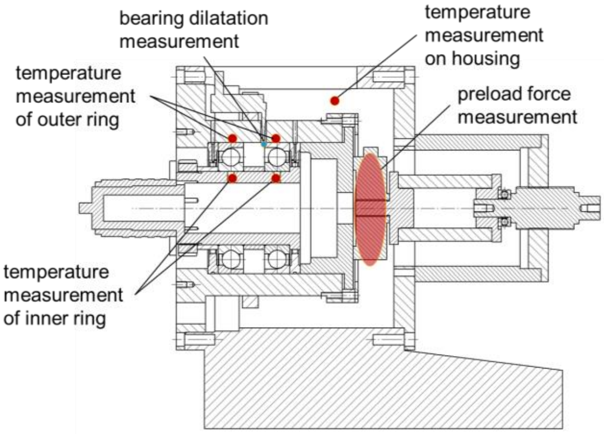

Fig. 6. *Measurement of test rig*

The designed shaft enables temperature measurement of spindle bearing inner rings using slipring assemblies (**Fig 7.**). The temperature of the outer ring is measured in the inner housing, in which temperature sensors are placed. Another temperature sensor is placed on the outer ring of the housing and measures the temperature under the surface. The force sensor LUCAS TENZO S-35 is placed in the housing and measures the magnitude of the preload. The IMI SENSORS IMI 622B01 diagnostic sensor is placed on the outer housing and measures the level of vibrations during testing the bearings.

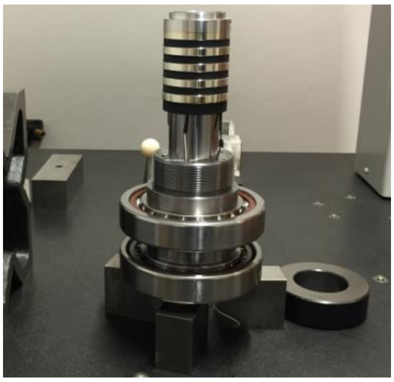

Fig. 7. Slipring assemblies for temperature measurement of spindle bearing inner rings

The lubricant supply (**Fig. 8**) was solved by the oil-air centralized lubrication system, which supplies quantities of the lubricant on the outer side of bearings using distribution rings.

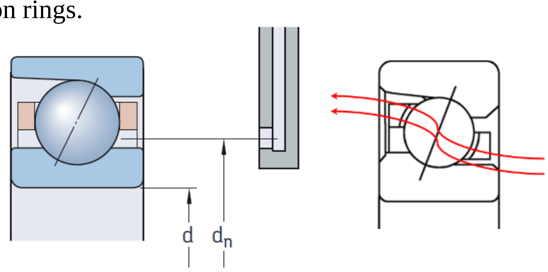

Fig. 8. *Supply oil-air to the bearing* ***[16]***

The core of lubrication unit (**Fig. 9**) is from LubTec manufacturer. Key components of the system include mixing valve, pressure valve and oil reservoir. Other accessories are built for measuring purposes. Control unit is built for measuring purposes too. Control unit allow set the duration of the dose and period of lubrication cycle. This unit is set for 0,05 g of oil VH68 per dose to each bearing.

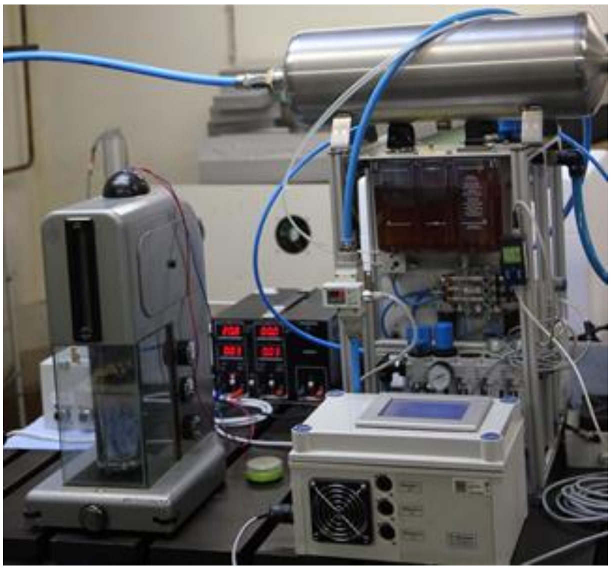

Fig. 9. *Oil – air lubrication unit*

The test rig allows the test bearings subjected to for the widest possible spectrum of load conditions characterized by speed, preload and bearing lubrication.

3. Experimental results

The experiment results of the friction torque spindle bearings B7214C-T-P4S lubricated grease and system oil-air, It is show in the graph (**Fig. 10**). Bearing lubrication system oil-air was dosage in interval 7,5 min. The bearings was preloaded force Fa = 400N. The experiment was always measured at constant speed e.g.: 8 000 rpm.

The experiment results were compared with mathematical simulation identifying friction torque in bearings on the based the thermo-mechanical models of bearings [12,14,15].

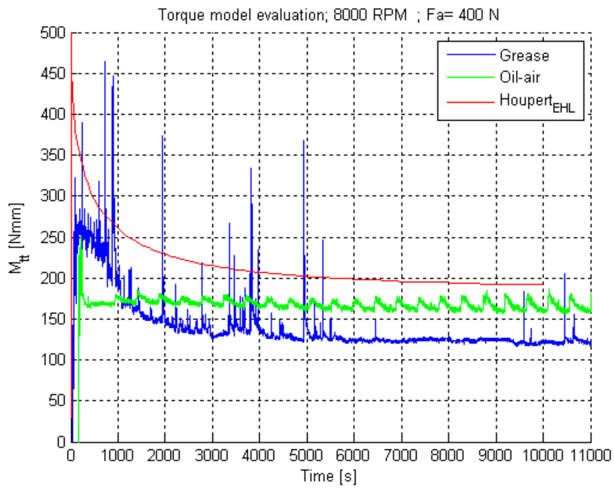

Fig. 10. *Comparison of measured data and simulation model 8000 RPM, 400 N axial preload, oil-air and grease lubrication; FAG B7214-C-T-P4S*

4. Acknowledgement

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5. Conclusion

The proposed test rig enabling identification behavior the pair of spindle bearings B7214C-T-P4S with contact angle α = 15°; B7214E-T-P4S with contact angle α = 25 in the "X"-arrangement. Operating parameters of test rig are speed, bearings preload, lubrication method and the quantity of lubricant supplied to the bearing.

The rig was designed to measure the temperature of spindle bearing inner, outer rings and environment, friction torque moment, preloaded force and vibration identifying the condition and behavior the bearings of the tests.

The test rig during experiments has proved to function and it is appropriate for identification behavior of spindle bearings lubricated grease and system oil-air

Symbols

A speed factor [mm/min]
d inter bearing diameter [mm]
D outer bearing diameter [mm]
d_m bearing mean diameter [mm]
F_a axial preload [N]
k rigidity [N/mm]
M_N rated torque [Nm]
M_{max} max. torque [Nm]
M_{tt} friction moment [Nmm]
n rotation speed [r/min; rpm]
n_N rated speed [rpm]
n_{max} max speed [rpm]
t time [s]
Q load [N]
α contact angle [°]
δ deformation [mm]

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