ACTA TECHNICA CORVINIENSIS
– Bulletin of Engineering
Tome X [2017]
Fascicule 2 [April – June]
ISSN: 2067 – 3809

[1.]Aleksandar ŽIVKOVIĆ, [2.]Milan ZELJKOVIĆ, [3.]Miloš KNEŽEV

THERMAL MODEL OF HIGH SPEED MAIN SPINDLE

[1-3.]University of Novi Sad, Faculty of Technical Sciences, Novi Sad, SERBIA

Abstract: In the last period a high-speed main spindles, by modern machine tools, are used with basic goal - to increase efficiency. The prediction of thermal behavior of the spindle is essential for machining precision. The main source of heat on the spindle occurs due to friction torque on bearings with angular contact. The heat generated on the bearings transfers to the spindle and surrounding air, thus causing the thermal expansion of spindle elements. This paper presents a 3D FEM thermal model of the main spindle with hybrid angular contact ball bearings.
Keywords: angular contact ball bearings, FEM, main spindle, thermal behavior

INTRODUCTION

The development of technology in producing high speed machine tools can be observed from the area of machine tool application [1]. The behavior of machine tools while in exploitation is conditioned by the behavior of certain vital assemblies. Especially significant are the spindle units, as the principle units of machine tools, since they determine the accuracy and productivity [2]. Behavior of this assembly during exploitation is conditioned by numerous parameters which can be grouped in three groups [3]:

a) conceptual,
b) geometric, and
c) other significant parameters.

Within constructive and geometric parameters it is possible to consider an influence of spindle, drive element and bearing support. Other significant parameters are: stiffness and bearing dumping, temperature and limited speed. At high-speed spindle and bearings rotations, a high heat amount is generated, and rotate elements (spindle, inner bearing rings and other revolving assembly elements) present rotational masses in system, which demands a precise cooling adjustment, lubrication and balancing. In this paper, the finite element method is used to simulate the temperature distribution of main spindle system. This model includes the major heat sources, the heat transfer between spindle elements for a transient analysis of the thermal behavior.

THERMAL FEM MODEL OF THE HIGH SPEED MAIN SPINDLE

The solid model of main spindle is built by using Inventor and the structure such as thread hole, keyhole, chamfer, fillet and so on are simplified. This structure has no effects on the analysis result. FEM model of the spindle has been importing to ANSYS APDL. In defining the thermal model of the spindle: the spindle shaft, bearings (rings and balls), and the housing, SOLID 87 element is used. Model consists of 31295 finite elements total. The finite elements CONTA 174 and TARGET170 were used to simulate contact joints. To simulate contact joints, contact pairs have been created at the joints, and the real constant TTC has been defined, i.e. the thermal contact conductivity for each ball, as well as for each contact between the outer ring/housing and inner ring/spindle shaft. The finite element thermal model is shown as Figure 1. Some assumptions are made for the thermal analysis:

1. The spindle model is axisymmetric and assumes a uniform clearance between the outer ring and the housing around the entire perimeter.
2. The thermal resistance in the axial direction, as well as axial conduction between contact elements is not considered.
3. Friction loss of the air is smaller than other heat sources, so it is neglected.
4. The heat radiation is neglected.

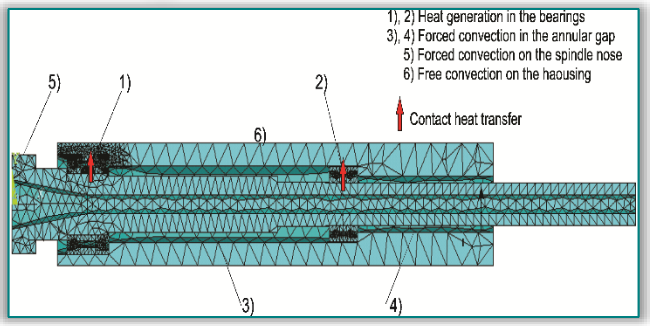

Figure 1. FEM model of the main spindle

Heat generation in the bearings

In analyzed model is assumed that the main heat sources in the assembly are bearings, because the power transmission is enabling by wheeled belt, which is setup on the spindle end, and its influence is discarded. The total frictional torque consists of torque due to applied load, torque due to viscosity of lubricant and spinning friction moment. The friction torque due to applied load and due to viscosity of lubricant at each inner and outer contact with the raceway is formulated similarly as in [4]:

$$M_{i,o(j)} = \left\{ 0.675 f_0 \left(\nu \omega_b \right)^{2/3} d_m^{\ 3} + f_1 \left(\frac{Q_{i,o(j)}}{Q_{i,o(\max)}} \right)^{1/3} Q_{i,o(j)} d_b \right\} \quad (1)$$

where are: f_0 - a factor depending upon the type of bearing and method of lubrication; f_1 - factor depending upon bearing design and relative load, ν -kinematic viscosity of lubricate, d_b - the diameter of the roller, Q_i and Q_o - contact loads at inner and outer raceways respectively, ω_b - angular velocity of the ball and Q_{max} - maximum contact load.

The order component of heat generation within the bearing is the spinning friction moment, which is formulated for every contact with the inner and outer raceway, based on [5]:

$$M_{S,i,o(j)} = \frac{3\mu Q_{i,o(j)} a_{i,o(j)} E^{'}}{8} \quad (2)$$

where μ is friction coefficient and a is semi-major axis of the contact area.

The total heat generated on the inner and outer raceway can be obtained as the sum of heat generated for each ball as [6]:

$$H_{i,o(j)} = \sum_{j=1}^{Z} \omega_b M_{i,o(j)} + \sum_{j=1}^{Z} \omega_{S,i,o(j)} M_{S,i,o(j)} \quad (3)$$

Convection Boundary Condition

Heat transfer coefficient at spindle rotation, assumed that the temperature difference is minor, can be calculated from the next equation, under condition that the forced convection is considered:

$$h_v = N_u k_a / d_v \quad (4)$$

where k_a is the heat conductivity of fluids (air), N_u is Nusselt's number and d_v is the diameter of the spindle nose. In case of annular gap of the spindle, d_v changes to the size of gap δ_{gap}. The detailed determinate of the Nusselt's number is available in [6,7].

Conduction between balls and raceways

The thermal contact conductance between balls and raceways is obtained empirically in function of the rotation speed [8]:

$$h_b = \frac{Z\sqrt{14 + 2\ln \nu - 2\ln d_b}\, d_b^2}{2400} \quad (5)$$

$$\nu = \frac{n\left(d_m + d_b\right)}{19099} \quad (5a)$$

The thermal resistance is obtained as [9]:

$$R_b = \frac{d_b}{h_b A} \quad (6)$$

where A is contact area between ball and raceway.

Conduction between rings and spindle/housing

Since the ring thickness is not a section constant, the ring width can be divided into five parts based on the cross-section, as shown in Fig. 2. Heat resistance between the bearing inner ring and the spindle shaft and outer ring and the housing is determined by using the relation for cylinder heat resistance [10]:

$$R_{i,o} = \frac{\ln(R_{m(i,o)} / R_{n(i,o)})}{2\pi L_{l(i,o)} k} \quad (7)$$

where $R_{n(i,o)}$ and $R_{m(i,o)}$ are the inner and outer section radii which are observed in the inner or outer bearing ring according to Fig. 2, and $L_{l(i,o)}$ the length according to the inner and outer section (Figure 2). Heat conductivity (k) for steel is 46.6 W/m-K and it is valid for 20-200 °C [11].

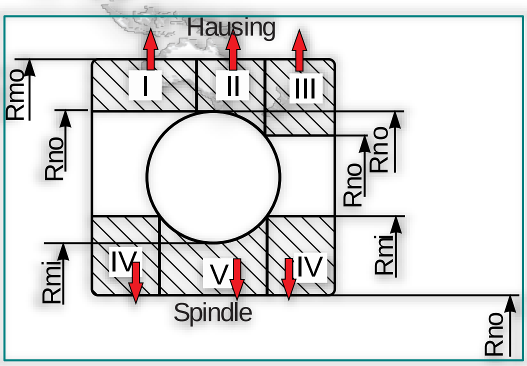

Figure 2. Conduction model of the bearing

RESULTS AND DISCUSSION

The analysis of thermal characteristics of the spindle includes stationary and non-stationary temperature change for different rotation speeds. Figure 3 predicts the temperature distribution of the spindle for rotation speed of 6000 rpm and reference temperature of 22 °C is shown.

The temperature of the front bearings is much higher than that of the rear. The maximum temperature occurs on the outer ring of the front bearing. This happens mostly because the higher generated heat and higher load occur on the front bearings. As the front bearings are very near to the top of the spindle (small length of the spindle top) heat, from the bearing in axial and radial direction, is transferred to the top of the spindle.

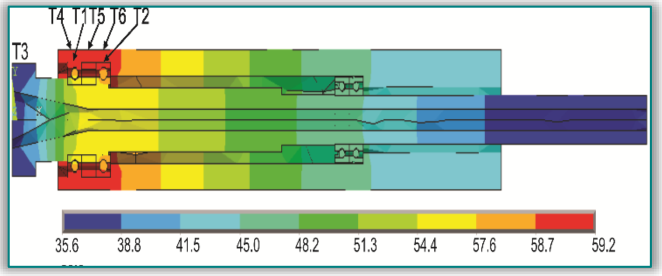

Figure 3. Temperature distribution
of the spindle for 6000 rpm

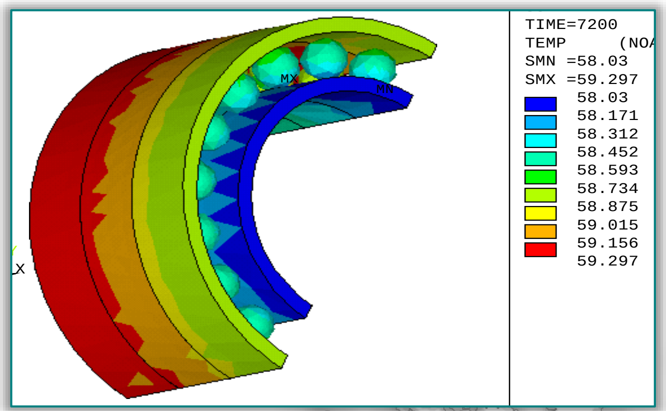

Figure 4. Temperature distribution
of front bearing at 6000 rpm

Figure 5 shows the simulated temperatures on the front bearing, housing, and spindle nose during time, for characteristic points in Figure 3. Unlike the temperature of the front bearing, the temperatures in other observed places increase slowly in the initial phase and rapidly in the later phase, until they reach stationary state. Their growth in time takes longer than on the bearings. Essentially, it is important to mention that the increase of temperature for observed places follows the same trend, but in different times of reaching the stationary state.

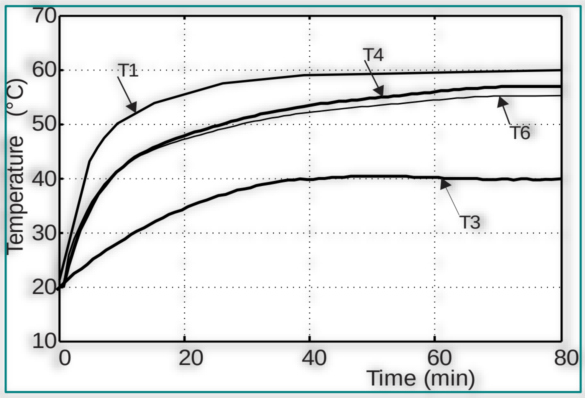

Figure 5. The temperature changes with time
at the selected points for 6000 rpm

In table 1 is shows the comparison between simulation and measured temperatures difference for stationary temperature state for different rotational speed.

Table 1. Comparison of the simulated
to the measured temperature

	Simulation			Experimental		
rpm	T1	T2	T3	T1	T2	T3
2000	45.2	42.6	29.4	48.1	43.2	28.2
4000	49.8	47.3	32.3	57.5	56.7	31.2
6000	59.2	59.1	35.6	67.3	63.4	32.4

The maximum difference from 2 to 10 % in stationary state, depending on the number of revolutions, is acceptable, especially since the suggested model does not consider the effects of bearing elements' change in dimensions. These changes affect the accuracy of several parameters, such as contact resistance, forced convection and generated heat in the bearing.

CONCLUSION

This paper discussed the mathematical model for predicting the spindle's temperature distribution in steady-state and non-steady-state. This model can be separated into two sub-models: the heat generated model, and the model of the spindle. In this paper, the thermal model of the spindle was created by using the finite element approach. This approach was chosen because of its major advantage, which is that heat conduction can be easily integrated with complex geometries and physical conditions, especially for any further analyses of the influence of temperature on static and dynamic behavior of the spindle. On the other hand this paper presents an analytic method for calculating the heat generated on the bearing, based on the internal distribution of load. Additionally, the simulation errors of the temperature fields are less than 10%, which in turn confirms the effectiveness of the simulated model proposed in this paper.

Acknowledgment

In this paper some results of the project: Contemporary approaches to the development of special solutions related to bearing supports in mechanical engineering and medical prosthetics – TR 35025, carried out by the Faculty of Technical Sciences, University of Novi Sad, Serbia, are presented. The project is supported by the Ministry of Science and Technological Development of the Republic of Serbia.

Note

This paper is based on the paper presented at The 3rd International Scientific Conference on Mechanical Engineering Technologies and Applications (COMETa 2016), organized by the Faculty of Mechanical Engineering, University of East Sarajevo, in Jahorina, Republic of Srpska, BOSNIA & HERZEGOVINA, December 7–9, 2016.

References

[1] Abele, E., Altintas, Y., Brecher, C. (2010). Machine tool spindle units, CIRP Annals - Manufacturing Technology, 59 (2), p.p. 781-802.

[2] Zverev, I.A., Eun, I.U., Chung, W.J., Lee, C.M. (2003). Simulation of Spindle Units Running on Rolling Bearings, International Journal of Advanced Manufacturing Technology, 21 (10-11), p.p. 889-895

[3] Zeljković, M. (1996). Sistem za automatizovano projektovanje i predikciju ponašanja sklopa glavnog vretena mašina alatki, Doktorska disertacija, Fakultet tehničkih nauka, Novi Sad.
[4] Li. H., Shin, Y.C. (2004). Integrated dynamic thermo-mechanical modeling of high-speed spindles, part 1: Model development, Journal of Manufacturing Science and Engineering, Transactions of the ASME, 126 (1), p.p. 148-158
[5] Harris, T. A., Michael., N. K. (2007). Rolling bearing analysis: Advanced Concepts of Bearing Technology, Taylor & Francis Group.
[6] Zivkovic, A. (2013). Računarska i eksperimentalna analiza ponašanja kugličnih ležaja za specijalne namene, Doktorska disertacija, Fakultet tehničkih nauka, Novi Sad.
[7] Zivkovic, A., Zeljkovic, M. Tabakovic, S., Milojevic, Z. (2015). Mathematical modeling and experimental testing of high-speed spindle behavior, International Journal of Advanced Manufacturing Technology, 77, p.p. 1071-1086
[8] Uhlmann, E., Hu, J. (2012). Thermal Modelling of a High-speed Motor Spindle, Procedia CIRP, 1 (0), p.p. 313-318.
[9] Kim, S.M., Lee, S.K. (2001). Prediction of thermo-elastic behavior in a spindle-bearing system considering bearing surroundings, International Journal of Machine Tools and Manufacture, 41 (6), p.p. 809-831
[10] Incropera, F.P., Lavine. A.S., DeWitt, D.P. (2011). Fundamentals of heat and mass transfer, John Wiley & Sons.
[11] Yovanovich, M.M. (1971). Thermal constriction resistance between contacting metallic paraboloids: Application to instrument bearings. AIAA Progress in Astronautics and Aeronautics: Heat Transfer and Spacecraft Control 24, p.p. 337-358

ISSN:2067-3809