Analysis and Investigation on Thermal Behaviours of Ball Bearing in High Speed Spindle

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Abstract---High cutting speeds and feeds are essential requirements of a machine tool structure to accomplish its basic function which is to produce a work piece of the required geometric form with an acceptable surface finish at as high a rate of production as is economically possible. Since bearings in high speed spindle units are the main source of heat generation. Friction in bearings causes an increase of the temperature inside the bearing. If the heat produced cannot be adequately removed from the bearing, the temperature might exceed a certain limit, and as a result the bearing would fail. To analyse the heat flow in a bearing system, a typical ball bearing and its environment has been modelled and analysed using the finite element method. The maximum temperature in the bearing has been calculated as a function of heat generation with the rotational speed as a parameter. The goal of this analysis was to see how fast the temperature changes in the bearing system with respect to rotational speed. and if a given maximum temperature (e.g. maximum temperature of the lubricant or bearing metal) is reached. Steady state thermal-stress simulation is performed exclusively on bearing to investigate the temperature distribution, deformation and thermal stress occurred at various stages, further this work is a more detailed for conducting transient analysis.

Keywords: Heat Generation, Modelling, Thermal Analysis, Static and Steady States.

I. INTRODUCTION

1.1 Background And Motivation

High speed machining is a promising technology to drastically increase productivity and reduce production costs. The technology of high speed machining is still relatively new. Although theories of high-speed metal cutting were reported in the 1930s, machine tools capable of achieving these cutting speeds did not exist until the 1980s.
Only recently, industry has started experimenting with the use of high speed machining in production. The aircraft industry was first, with the automotive industry and mould and die makers now following. Because of little experience in this new field, there are still many problems to be solved in the application of high speed machining. Current problems include issues of tooling, balancing, thermal and dynamic behaviours, and reliability of machine tools [2].

1.2 High Speed Machining

The demand for high speed machine tools and three coordinate measuring machines are rapidly increasing in response to the development of production technology such precision machining which requires high-precision parts and high productivity.

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Research on high speed machine tooling can be approached on the main spindle and feed system. A high speed/precision feed system reduces non-cutting/operating time and tool replacement time, making production more economical.

1.3 Bearing Reliability

The high speed precision ball bearing is a main part in high speed/precision feed system, and there are many joints existing in the ball bearing, such as the interfaces between the bearing and the shaft, the bearing and the bearing support and so on. When two surfaces are in contact, the presence of surface roughness produces imperfect friction at the joint, no matter how much the pressure between the surfaces is. The friction in ball bearings entails a sudden and violent heating of balls that can have very detrimental effects.

The increase of temperature generated by these phenomena can involve mechanical micro-deformations and an overheating of cooling fluid (especially when dealing with cryogenic fluids). Such temperature heating of ball bearing plays a significant role in thermal characteristics of the feed system, causing serious thermal deformation that subsequently degrades the accuracy of machine tool and other mechatronics instrument where the precision ball bearing is used.

II. LITERATURE REVIEW

Jin Kyung Choi et al.**[1]** developed a spindle bearing system in which the thermal characteristics bearing system with a tilting axis were investigated using finite element method to improve the performance. The prototype had been designed for the measurement of the temperature increase which was also analysed by the finite element method and compared to the experimental results. From the comparison of the numerical results with the experimental results, he found that the finite element method predicted well for the thermal characteristics of the spindle bearing system.

Bernd Bossmanns et al.**[2]** developed thermal model to characterize the power distribution of a high speed motorized spindle, in particular the characterization of heat transfer within the spindle and heat sinks under the influence of speed, preload, and lubrication. Without loss of generality, model is based upon a custom-built high performance motorized milling spindle of 32 KW and maximum speed of 25 000 rpm.

X. Hernot et al.**[3]** derived stiffness matrix of angular contact ball bearings by using the analytical approach. This formulation can facilitate the connection of the bearing behaviour model with those of the other components in the assembly The matrix connected to the conventional model in two degrees of

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freedom is first presented. A practical application of this formulation is illustrated through the common problem of sizing a two bearings-shaft arrangement. Variations of displacements, axial forces, and bearing fatigue life related to preload are shown to be easily obtained..

Harris [4] worked on bearing preload and applications, after Jones and palmgrem, harries further summarised and developed Jones theory. he systemized the calculated of bearing load, ball motion, load distribution and high speed bearing load distribution, bearing deflection, fatigue life and friction.

Chi-Wei Lin et al.**[5]** who developed an integrated model with experimental validation and sensitivity analysis for studying various thermo-mechanical-dynamic spindle behaviours at high speeds. They investigated the effect of bearing stiffness with respect to the preload and showed that the bearing stiffness increases with increase of preload and the effect of rotational speed makes the bearing stiffness decrease with increase of speed. *Mohammed A. Alfares* et al.**[6]** Presented a mathematical model based on a five degrees of freedom dynamic system to study the effects of axial preloading of angular contact ball bearings on the vibration behaviour of a grinding machine spindle. They ended by saying the larger the initial preload applied, the less vibration amplitudes are generated and as the initial preload increases, the stiffness of the bearing increases that makes dominant frequencies of the system shift to higher values. As the preload increases up to a certain value, the peak to- peak amplitude decreases. Beyond this value the reduction in vibration amplitude is insignificant which indicates that larger values of preloading will not further reduce the vibration levels of the machine spindle system.
S.P. Harsha et al.**[7]** presented a model for investigating structural vibrations in rolling element bearings. He derived the mathematical formulation of contact stiffness between the inner race and ball, ball and outer race based on Hertzian elastic contact deformation theory. He also formulated the contribution of inner ring, ball and the outer ring in deformation of bearing separately.

Yuan Kang et al.**[8]** determined stiffness of angular-contact ball bearings by using neural network method. He developed the BPNN algorithm which the stiffness can be determined for all of the same type of bearings, with the same inputs of varied rotating speed, varied thrust force, and varied radial force. He validated the tool by conducting case studies for different series of bearing sets taken from SKF catalogue and compared with JHM. The better results are drawn that the computer time for the stiffness determination by using the JHM is several hundred times greater than by using BPNN.

Xu Min et al.**[9]** developed thermal model for machine tool spindle based on heat source models and the heat transfer models from Bossmanns**[2]**. He calculated the amount of heat generated by the bearing including the thermal contact resistance at the solid joints based on a fractal model and the change of the heat generation power, viz. the amount of the heat generation per second. Here the complete thermal model is used to simulate the temperature distribution in grinding machine housing with a conventional spindle bearing.

III. PROBLEM DEFINITION

The thermo-mechanical model for the spindle needs to include all interacting effects inside the spindle relevant to the objective. The model should account for all heat sources, heat transfer, heat sinks and relative thermal expansion within the system. In this case bearings in the model are one of the problems. Friction in bearings causes an increase of the temperature inside the bearing. If the heat produced cannot be adequately removed from the bearing, the temperature might exceed a certain limit, and as a result the bearing would fail.

Proposed possible way to solve the problem

1. Literature study of different approaches to model roller bearings
2. Set-up of a 3-D reference FE-model
3. Identification of the main sources for the nonlinearity of roller bearings
4. Define of a simplified bearing model
5. Validation and recommendations for the use of simplified bearing models

IV. METHODOLOGY

To analyse the heat flow in a bearing system, a typical ball bearing and its environment has been modelled and analysed using the finite element method. The maximum temperature in the bearing has been calculated as a function of heat generation and with the rotational speed as a parameter.

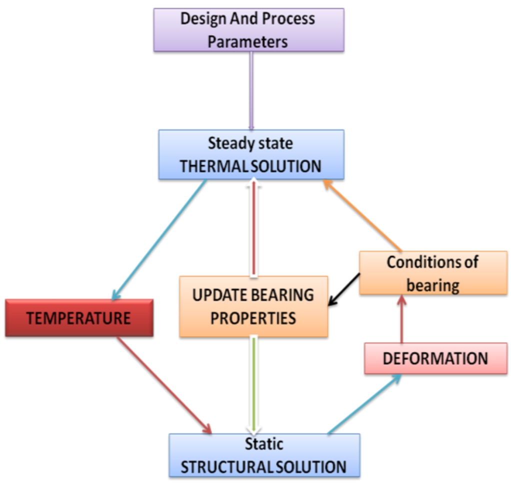

Figure 4.1: Flow Chart of Methodology

The design model which is created is imported to Ansys by giving all process parameters which the model undergoes steady state thermal analysis by giving all the boundary condition. Here the temperature distribution in the model that occurred in the bearing comes out as output. Now the structural boundary conditions are applied to the bearing by updating the properties to undergo steady state structural analysis which finally gives the deformation of the bearing at various points.

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V. HEAT GENERATION IN BEARING – ANALYTICAL SOLUTION

5.1 *Heat Generation*

The major heat generation of the system is caused by the cutting process and the friction between the balls and races of the bearings. Assumed that the majority of cutting heat is taken away by coolant and chips, the heat generated by bearings is the dominant cause of temperature change. In angular contact ball bearings heat is generated mainly by three sources. The heat generated by a bearing can be computed as

$$H_f = 1.047X10^{-4}\, n\, M \qquad (5.1)$$

Where H_f is the heat generated power (W), n is the rotating speed of the bearing (rpm), M is the total frictional torque of the bearing (N mm). The total frictional torque M consists of two parts, one is the torque M_1 due to applied load and the other one is the torque M_2 due to viscosity of lubricant. That is

$$M = M_1 + M_2 \qquad (5.2)$$

5.1.1 *Frictional torque due to applied load*

The torque due to applied load can be empirically approximated by the following:

$$M_1 = f_1\, F_\beta\, d_m \qquad (5.3)$$

In which f_1 is the factor depending upon bearing design and relative load. For ball bearing

$$f_1 = z\, (F_s/C_s)^{Y} \qquad (5.4)$$

Where F_s is the static equivalent load and is given by

$$F_s = X_o\, F_r + Y_o\, F_a \qquad (5.5)$$

X_o, Y_o are the values taken from the table A_4, and F_r is the radial force acting on the bearing which is consider has zero, F_a is the axial force on bearing and the values are given has From table at 15^0 angle $X_o = 0.5$, $Y_o = 0.46$ and the axial force F_a = 945 N. By solving the equation (45) yields

F_s = 434.7 N

C_S is basic static load rating gives has, C_S = 4850 N
Table A_1 gives appropriate values of Z and Y. For angular contact ball bearing having contact angle 150 the values are given has

Z = 0.001 Y = 0.33

So by solving equation (44) f_1 is obtained has

f_1 = 0.00045

F_β is dynamical equivalent load, for angular contact ball bearings generally depends on the magnitude and direction of the applied load, F_β is given has

$$F_\beta = F_a - 0.1F_r \qquad (5.6)$$

Since F_r, the force in radial direction is zero then the equation (46) gives

F_β = 945 N

Finally by solving the equation (43) by taking pitch diameter of bearing d_m = 36 mm, the torque developed due to applied load is given has

M_1 = 15.309 N mm

5.1.2 *Viscous Friction Torque*

For bearings that operate at moderate speeds and under non-excessive load, the viscous friction torque can be empirically expressed as follows:

$$M_v = 10^{-7} f_o\, (v_o n)^{2/3} d_m^3,\ v_o n \geq 2000 \qquad (5.7)$$

$$M_v = 160x10^{-7} f_o\, d_m^3,\ v_o n \leq 2000 \qquad (5.8)$$

In which ν_o is the kinematic viscosity of lubricated oil given in centistokes, the value of v_o is taken from table A_3, depending up on the temperature. Here we assumed the uniform temp has 40 and the value taken is,
v_o = 20

And n is the revolution speed of the bearing in *rpm*

n = 5000 *rpm*

f_o factor depending upon the type of bearing and method of lubrication. The value is taken from harries [4]. It is taken has

f_o = 2
Since we are taking the speed greater than 2000 the equation (6.7) gives the final viscous friction torque has

M_v = 23.32 N mm

The overall torque is given has
M = 38.367 N mm
Finally the heat generated by a bearing is given has

$$H_f = 1.047X10^{-4}\, n\, M$$

Therefore at 5000 rpm the heat generated in angular contact ball bearing is

$$H_f = 20.085 \text{ W}$$

Similarly the heat generated at various speeds are calculated and listed below

Table 5.1: heat generated at various speeds

S.no	Rotational speed (rpm)	Heat generated (W)
1.	5000	20.08
2.	10000	54.47

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3.	15000	99.60
4.	20000	154.07
5.	25000	217.03
6.	30000	287.85
7.	35000	366.07
8.	40000	451.32
9.	45000	543.28
10.	50000	641.70
11.	55000	746.35

VI. MODELING AND ANALYSIS

6.1 Model Description

The physical model of angular contact ball bearings used in the high speed feed system that is to be studied is shown in the fig.6.1. This conventional bearing is composed of a retainer and ceramic balls having the mass 0.075kg. Its dimensions are D=47 mm, d=25 mm, $r_{1,2}$= 2 mm, $r_{3,4}$=4mm, B=12mm, a_o=15^0 and Z=14 balls. The bearing is supposed to operate under maximum basic dynamic load rating of C=9.56kN and static load rating of C_0=5.6kN and at rotate speed of 5000-55000rpm.

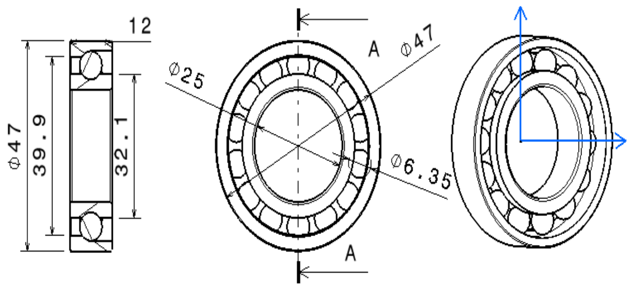

Figure.6.1: The Physical model of single row angular contact ball bearing

6.2 Tool Used

Table 6.1 Tools used

Design tool	Analyses tool
PRO-E	ANSYS WORK BENCH 12.0

6.3 Properties Of Materials Used

Table 6.2 properties of materials

Properties At Ambient Temp	UNITS	CERAMIC (Si_3n_4)	STEEL $100Cr_6$
Density	g/cm^3	3.2	7.8
Coefficient Of Expansion	10^{-6}/k	3.2	11.5
Young's Modulus	GPa	315	216
Poisson's Ratio	-	0.26	0.3
Thermal Conductivity	W/mk	30-35	40-45
Tensile Yield Strength	MPa	700	2500

VII.RESULTS AND DISCUSSIONS

7.1 Thermal-Stress Analysis

The CAD model which is used for analysis undergoes a steady-state thermal analysis. In this steady-state numerical analysis temperature distribution in the bearing is measured with respect to different rotational speeds. Here the heat generation value plays the major role in the thermal analysis which is calculated, discussed previously.

Table 7.1: Analysis Results

Rotational Speed(rpm)	Heat Generation(W)
5000	20.08
10000	54.47
15000	99.60
20000	154.07
25000	217.03
30000	287.85
35000	366.67
40000	451.32
45000	543.28
50000	641.70
55000	746.35

8.1 Heat generation with effect of rotational velocity

The change in heat generation value with respect to the rotational velocity is plotted in fig.8.1. At high speed, thermal effects on the dynamic response significant and must be considered. When bearing speed increases, there is an increase of heat generation at the bearing contact locations which generates thermal load to the bearing..

Dynamic heat generation in bearing

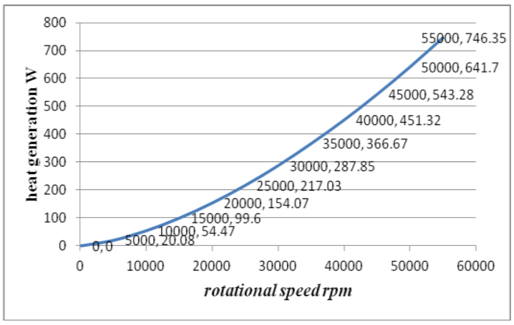

Figure 7.1: Bearing heat generation rate as function of rotational speed

VIII. CONCLUSION

A thermal model is developed to study the heat generation rate, temperature distribution, deformation and thermal stress occurred in the bearing system at various stages with rotational

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speed as parameter and preload load applied to a feed system . Based on the characteristics of dynamic behaviour of the bearing system, the thermal stress simulation is conducted, and it is observed from the simulation that the temperature in the bearing increases with increase in heat generation developed by bearing and also it is found that bearing inner ring temperature is higher than the outer ring temperature due to centrifugal forces that make inner ring contact forces and their corresponding heat generation rate higher than those of the outer ring. Further the increase of rotation speed the inner ring centrifugal displacement value increases greatly, this increase in inner ring centrifugal displacement causes larger contact deformation and stress. Furthermore, the effect of the inner ring centrifugal displacement is larger on inner raceway than on outer raceway.

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