Analysis of High Speed Spindle with a Double Helical Cooling Channel

R.Sathiya Moorthy, V. Prabhu Raja, R.Lakshmipathi

***Abstract* -** The purpose of this paper is to numerically analyze the fluid motion and temperature distribution in built-in motorized high-speed spindle housing with double helical water cooling channel. A three dimensional finite element model of high speed spindle housing is developed and simulated using computational fluid dynamics software to determine the temperature distribution. The model is based upon a machine tool spindle of 5KW and maximum speed of 40,000 rpm. The effect of heat flux (q"= 310867 W/m^2) with different cooling water velocity are examined in detail. The results are compared with a single helical cooling channel and it indicates that the designed cooling loop is more effective and increase in temperature can be reduced significantly.

***Keywords* –** High speed spindle, spindle housing, double helical cooling channel and Turbulence analysis.

INTRODUCTION

In order to achieve the goal of high productivity, the concepts of high speed machining are established in manufacturing industries. In high speed machine, spindle is usually equipped with a built-in motor; it leads to the reduction in vibration by means of eliminating the power transmission devices, such as belts and gears. It soon became noticeable that the very high heat dissipation of built-in motors pushes the spindle components to their limits. In high-speed machining, the excessive heat generation in the spindle induces uneven thermal expansion. Therefore, better modelling of the thermo-mechanical behaviour of the motorized spindle system is required to control the temperature of the machine tool spindle

Bernd Bossmanns, Jay F. Tu [2] developed thermal model based upon a custom-built high performance motorized milling spindle of 32 KW and maximum speed of 25 000 rpm to characterize the power distribution of a high speed motorized spindle, heat sources within the spindle systems are (a) heat generation by angular contact ball bearings under the influence of speed, preload, and lubrication. (b) Heat generation by the electric motor in rotor and stator as a function of torque and speed. (c) Heat generation due to viscosity shear of air by the rotating components of the spindle.

K. J. H. Al-Shareef and j. A. Brandon [1] presented an analysis of the dynamic characteristics of machine tool spindle-bearing systems subjected to harmonic excitation at the spindle nose including the influence coefficient and it is validated using Maxwell's reciprocity theorem. The calculation compares natural frequencies and mode shapes of the continuous system and its replaced by a discrete lumped mass system. Jenq-Shyong Chen, Wei-Yao Hsu [6] is reported the characterizing and modelling of the thermal growth of a motorized high speed spindle. The centrifugal force and thermal expansion occurring on the bearings and rotor change the thermal characteristics of the built-in motor, bearings and assembly joints. An auto-regression dynamic thermal error model, that considers the temperature history and spindle-speed information, has been proposed and proved to improve the model accuracy. The relationship between temperature measurements and thermal displacements is highly nonlinear, time-varying and non-stationary. A new thermal model which correlates the spindle thermal growth to thermal displacements measured at some locations of the rotating spindle shaft has been proposed.

Choi and Lee [4] adopted Kreith's [5] heat transfer coefficients for a rotational spindle to obtain the temperature distribution in a spindle bearing system by using finite element analysis to calculate the heat transfer coefficient at different flow rates. Because the heat transfer coefficient was assumed to be constant throughout the working fluid, the calculated temperature distribution in the spindle bearing system was different from the actual situation. The thermal analysis of a spindle is not easy because the heat transfer coefficient of the liquid cannot be precisely determined due to the geometric complexity of a spindle. The thermal analysis of a spindle is not easy because of the geometric complexity of a spindle.

The intention of this paper is to numerically analyze the three-dimensional fluid motion and temperature distributions in a built-in motorized high-speed machine tool spindle with a helical water cooling channel. The effect of heat flux (q″= 310867 W/m^2) with three different cooling water velocity(0.28m/s,0.48m/s,0.72m/s, 0.8m/s and 0.9m/s) are examined in detail.

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MODELLING DESCRIPTION

Fig. 1 shows the physical model of the built-in motorized spindle housing with a double helical water-cooling channel. The heat generated by the motor is distributed in the inner wall of the housing with a length of 70mm.

The high speed machine is operated in an air conditioned room with the ambient temperature is 20^0C. Therefore, the heat is removed both by the forced convection of cooling water and the natural convection of ambient air. The assumptions are made to simplify the analysis is thermal conductivity of the spindle housing is isotropic and temperature independent.

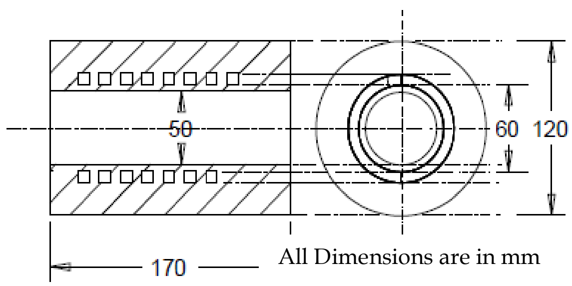
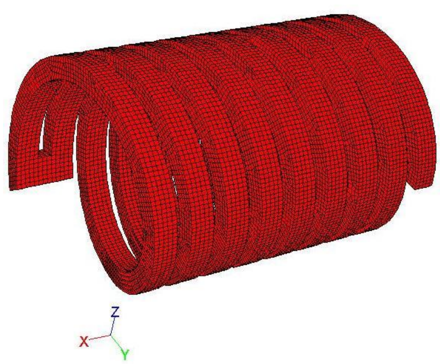
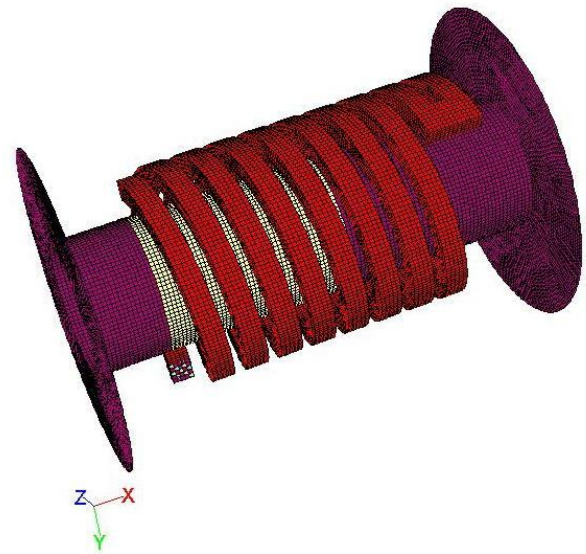


Fig. 1. The physical model of the built-in motorized spindle with a helical channel.



a. Cooling channel grid



b. Total Grid

Fig.2 Meshed model

All the analysis is carried out here is based on effect of cooling water having density of 998.2kg/m^3, through a double helical channel on the spindle housing, whose density is 7800Kg/m^3. The thermal conductivity of cooling water, spindle housing are 0.6 W/m-K and 60 W/m-K respectively.

NUMERICAL METHOD

In this study, the governing equations were solved numerically using a finite volume. A brief description of the methodology is as follows.

A. Modeling and meshing.

Initially, the model created using Pro/ENGINEER Wildfire 4.0 with respected to the dimensions. Then it was imported into the meshing software (GAMBIT 2.4) in the required format. The model which represents a single volume is being segregated into two volumes, fluid and solid. To ensure accuracy and validity of the result the grid independence study is carried out and it is found that a structured mesh shown in Fig.2 having the Element Size 0.02.is most suitable. In this work the model is discritized to 1092083elements.

B. Boundary conditions

The selections of boundary areas are the next and most important steps in computational fluid dynamic methods. In the case of high speed machine tool spindle the majority of heat is generated by the electric motor in rotor and stator. The hot spots are almost all concentrated near the center of the spindle axis. The following elaborates the physical boundary condition of the this work that is the fluid at the inlet zone is provided with different velocity (0.28m/s, 0.48m/s, 0.72m/s, 0.8m/s, 0.9m/s) and uniform temperature of 17^0C. The heat generated by the motor in the stator surface (q″=310867 W/m²) is distributed in the inner wall of the housing with a length of 70mm. In reaming inner walls the adiabatic boundary condition is assumed. At the outlet of the water channel of the computational domain, Dirichlet boundary condition is applied.

C. Solving process

The governing equations are non-linear and coupled, so several iterations of the solution loop must be performed before a converged solution is obtained and each of the iteration is carried out as shown in Fig.3

First Fluid properties are updated in relation to the current solution, if the calculation is at the first iteration, the fluid properties are updated consistent with the initialized solution. Then three momentum equations are solved consecutively using the current value for pressure so as to update the velocity field. Since the velocities obtained in the previous step may not satisfy the continuity equation, one more equation for the pressure correction is derived from the continuity equation and the linearized momentum equations finally, the convergence of the equations set is checked and all the procedure is repeated until convergence criteria are met when the residuals of all the variables are less than e^{-6}.

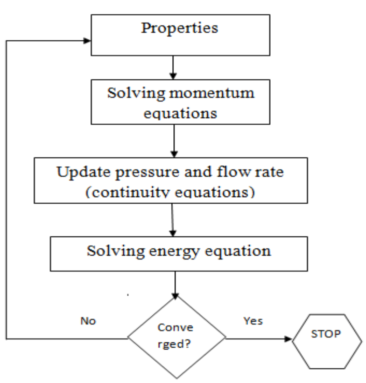


Fig.3. Solving process

Here the pressure based solver is used to solve the energy equation bases on RNG k-epsilon turbulence model. Prior to computation, a thorough verification of the grid-independence of the numerical solution was performed in order to ensure the accuracy and validity of the numerical results. The solution reaches convergence after approximately 200 iterations.

RESULTS AND DISCUSSIONS

Fig. 4 represents temperature distribution of cooling water, with inlet velocity of 0.48 m/s and inner wall heat flux heat flux q″= 310867 W/m². Cooling water take the heat from heat source results in increasing the temperature the cooling water from 17^0C to 43^0C.The water velocity distribution inside the channel as shown in the fig.6 was well mixed and quite uniform due to the swirl effect generated by the helical channel with rectangular Cross-section. The hot spots for the cooling water are almost all concentrated near the center of the spindle axis. This is due to the fact that the heat generated by the motor is distributed in the middle of the inner housing with a length of 70mm.

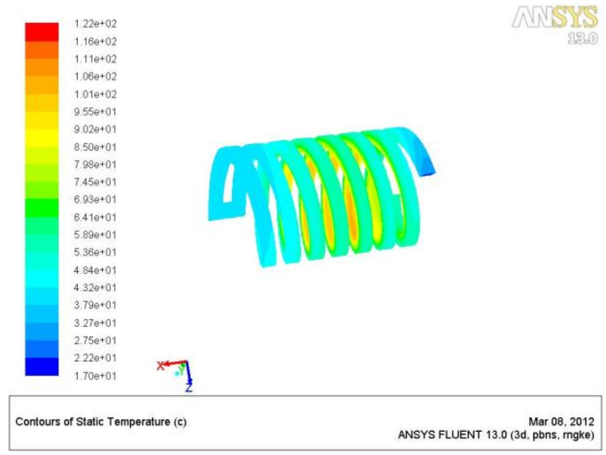


Fig.4.The temperature distribution of cooling water

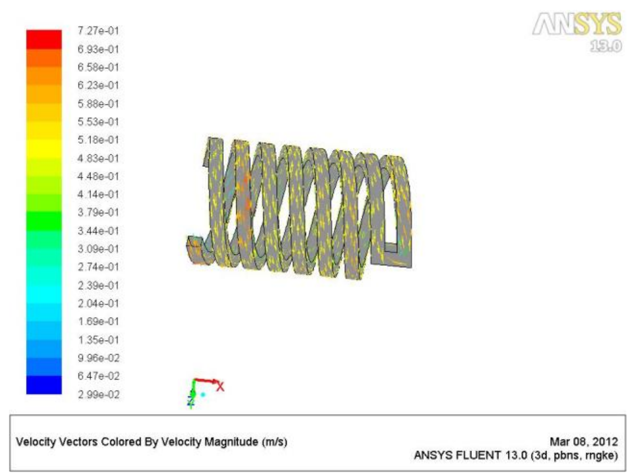


Fig.5. The velocity distribution of cooling water

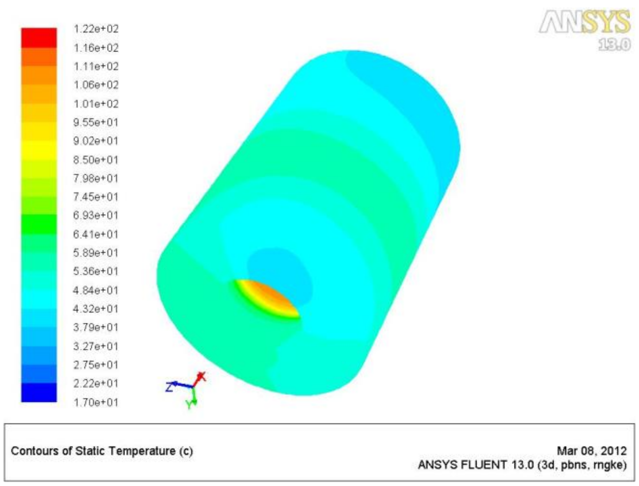


Fig.6. Temperature distribution with heat flux 0.31 W/mm^2

Fig 6 shows the maximum temperature on the outer wall of the spindle housing was 64^{0}C. Most of the heat is conducted along the radial direction rather than the axial direction. Due to the effect of double helical cooling channel small amount of heat source at inner wall (q″=310867 W/m^2) conducted through the wall.

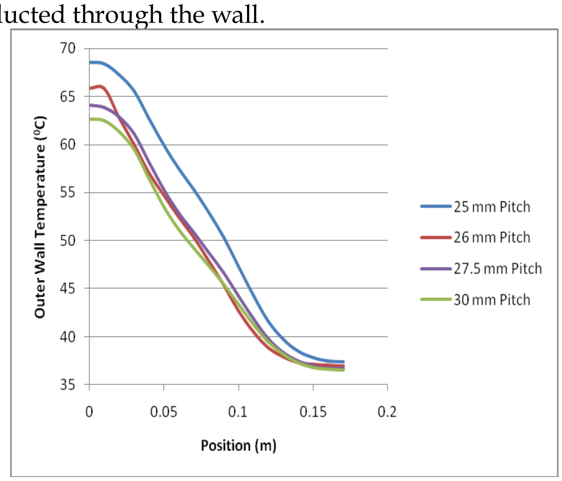


Fig7: The outer wall Temperature Distribution for Different Pitches for a velocity of 0.48 m/s

The pitch of the double helical cooling channel has been varied from 30mm to 25mm. From Fig 7 it is inferred that even if we reduce the pitch from27.5mm to 25mm, the variation in the outer wall temperature is 4^{0}C, that to an end only, by considering the manufacturing difficulties we can conclude that 26mm pitch as an optimized one.

It is observed that the machine tool spindle temperature is reduced by increasing the cooling water velocity also. The temperature at the outer wall of spindle with respected to the velocity of cooling water (0.28m/s, 0.48 m/s, 0.72 m/s, 0.8 m/s, 0.9 m/s) are 82^{0}C, 63^{0}C, 51^{0}C, 49^{0}C and 46^{0}C respectively.

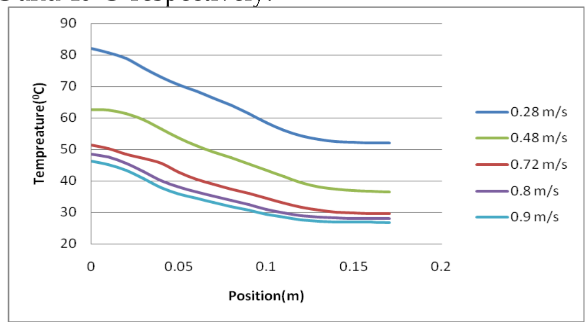


Fig 8: The outer wall Temperature Distribution for Different velocity with a Pitch of 26

The inlet velocity of the double helical cooling system has been varied from 0.28m/s to 0.9m/s. From the Fig 8 it is found that even if we increase the velocity greater than 0.72m/s the variation in the outer wall temperature has been 5^{0}C only. This is due to reason the turbulence has attained in the velocity of 0.72 m/s itself.

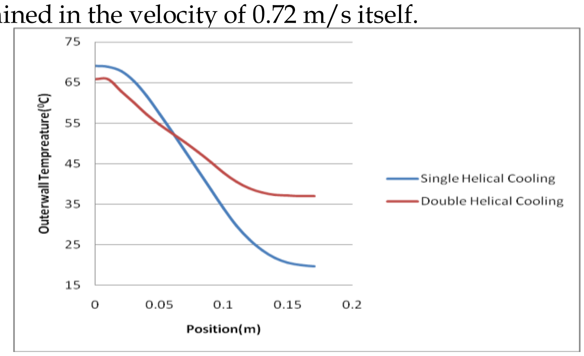


Fig 9:The Outerwall Temperature Distribution for Single Helical and Double Helical Cooling Chanel

Form Fig.9 the double helical cooling reduces the maximum temperature to from 69^{0}C to 66^{0}C and the outer wall temperature distribution is from 65^{0}C to 37^{0}C. In single Helical cooling the outer wall temperature distribution is from 69^{0}C to 20^{0}C. The results indicate that the double helical cooling channel is more effective than single helical cooling channel.

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