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CFD Analysis of cooling channels in built-in motorized high speed spindle

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***Abstract*—the aim of this work is to explore, the application of CFD in analyzing the helical cooling channel of a built-in motorized high-speed spindle. A numerical model is based on a high speed spindle, which is used in a printed circuit board industry, of 1.5 kW and maximum speed of 160,000 rpm. The predicted heat transfer coefficient values were compared with the experimental results of published literature. Then the validated model is used for a machine tool spindle of 4.7 kW and maximum speed of 40,000 rpm. The heat transfer coefficients and pressure drop were found for spindle cooling channel. These predicted results were also compared with results of other cooling channel designs, which are adopted by spindle manufacturing industries considering manufacturability and performance.**

It is concluded from the numerical analysis that for a complex geometry heat transfer coefficient can be closely predicted by using the CFD tool. The comparative study of cooling channel designs showed that double start helical cooling channel can be used with some compromise between pressure drop and heat transfer coefficient.

Keywords-component; Turbulence models, Conjugate heat transfer, Kinetic and Dissipation rate

I. INTRODUCTION

High speed machining is the major area of interest in most of the manufacturing firms, as it increases the productivity and reduces the manufacturing cost. In high speed machine tools the spindles are equipped with a built in motor, so that power transmission devices and gears are eliminated. In high speed machining the excessive heat generation in spindle induces uneven thermal expansion in mechanical components which in turn causes friction, wear and large machining tolerances. The general configuration of the spindle in the machine is shown in Fig. 1.

The studies carried out regarding the thermal analysis and cooling of high speed spindle are summarized below:

Bossmanns and Tu [1] presented a power flow model for a more complete thermo mechanical model of high speed spindle. This power flow model considers all the major heat sources within spindle. The major heat sources are (1) heat generated by angular contact bearings, (2) heat generated by electric motor, and (3) heat generated due to viscosity shear of air by the rotating components of spindle.

Later Shuhong xiao and Bolin Zang [2] have analyzed the heat generated in the built-in motor and bearing. They have analyzed the oil-air and oil-water lubrication system. The high speed spindle was modelled and its temperature field were analyzed using ANSYS. Main measures to find uniform temperature field of the high speed motorized spindle were brought forward.

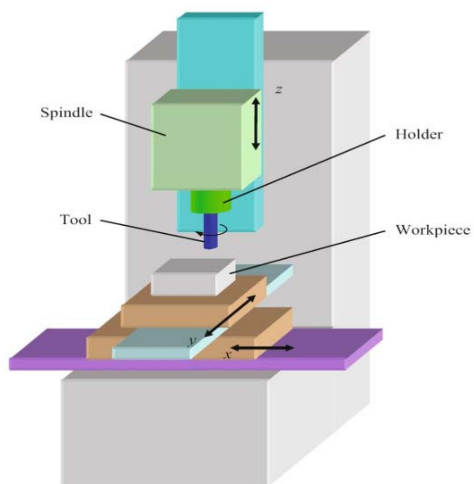

Fig 1: Vertical machining centre showing spindle assembly

Bossman and tu [3] proposed a heat transfer model for the bearing and lubricant oil within high speed spindles. They calculated the heat transfer coefficient of the lubricating oil assuming that heat transfer coefficient is linear with velocity.

Jin Kyung Choi and Dai Gil Lee [4] investigated the thermal characteristic of spindle bearing using finite element analysis. Based on the numerical result, a prototype was built.

C.H. Chien and J.Y. Jang [5] have numerically analyzed the three-dimensional fluid motion and temperature distribution in a motorized high speed spindle with a helical water cooling channel. They used the K-epsilon turbulence model to develop the fluid model. The effects of different heat sources (q = 60 W, 120 W and 240 W, which correspond to heat flux of 1.43, 2.8652 and 5.7304 W/cm2, respectively) and cooling water flow rate were examined in detail. It was shown that the average maximum temperature along the spindle axis is decreased from 24.5°C to 22.2°C, when the water flow rate was increase from 0.4 to 1.2 L/min. It was also shown that 'h' varies with V 0.184.

M.A. Donmez, M.H. Hahn and J.A. Soons [6] have studied the feasibility of using compressed air to reduce the thermal drift of machine tools. This method uses inexpensive, specially shaped tubing with small silts. Compressed air forced through such tubing increases the heat transfer rate. Experimental setup has also been developed. These methods of cooling have been proved to be effective.

From this, it is evident that, studies have been carried out on cooling channels and to develop thermal models for high speed spindle. But a comparative study of various cooling channels design is required so that a design engineer can make a better selection for a specific spindle application.

In this work a numerical model is developed for a high speed spindle of 1.5 kW and maximum speed of 160,000 rpm [5].The predicted heat transfer coefficient values were compared with the experimental results of published literature [5].Then the validated model was used for the spindle of 4.7kW and maximum speed of 40,000 rpm. The heat transfer coefficient and pressure drop of the spindle were predicted. These predicted results were also compared with results of other cooling channel designs, which are adopted by spindle manufacturing industries considering manufacturability and better performance.

II. ANALYSIS OF SPINDLE FOR PCB APPLICATION

Numerical analysis is performed on a spindle used for PCB manufacturing, of 1.5 kW and a maximum speed of 160,000 rpm. It uses a double start cooling channel. Water is the working fluid, which is actually pre-cooled before inlet. The Inlet temperature of water is 17ºC.The heat generated from the motor is drawn by water flowing through the cooling channel. This is a problem of conjugate heat transfer where the mode of heat transfer is solid conduction, conduction in no slip region and convection.

A. *Mathematical analysis*

The physical model of the spindle is shown in Fig. 2. The spindle is operated in an air conditioned temperature T Env = 290 K. The heat generated by the motor is distributed throughout the length of the housing. The total heat is taken away by the housing material, the coolant by forced convection and by natural convection too.

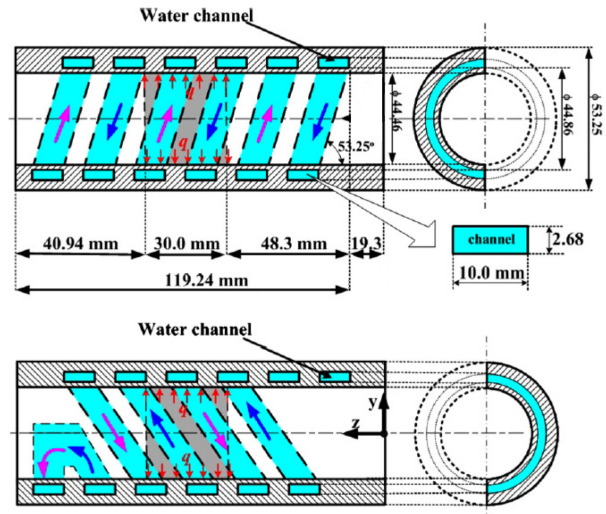

Fig 2: The physical model of the built in motorized spindle with a helical channel [5]

The following assumptions are made to simplify the analysis:

1. The heat generated by the motor is assumed to be 60,120 and 240 kW.
2. Thermal conductivity of the spindle housing material is 16.3W/m0C (AISI 302), and isotropic.
3. The fluid is considered to be incompressible with constant physical properties.
4. The flow is assumed to be steady and turbulent with no viscous dissipation.
5. The natural convection and radiation effects are ignored.

Due to the problem of closure (contains Reynolds stresses) standard k-epsilon model is used. Time averaged Navier-stokes equation for continuity, momentum and energy can be expressed as

$$\frac{\partial \overline{u_i}}{\partial x_i} = 0 \quad\text{....................................(1)}$$

$$\frac{\partial \rho(\overline{u_i u_j})}{\partial x_j} = -\frac{\partial \overline{p}}{\partial x_i} + \frac{\partial}{\partial x_i}\left[\mu\left(\frac{\partial \overline{u_i}}{\partial x_j} + \frac{\partial \overline{u_j}}{\partial x_i}\right) - \rho\overline{u_i' u_j'}\right] \quad\text{....................(2)}$$

$$\frac{\partial}{\partial x_i}\left(\rho \overline{u_i} k\right) = -\frac{\partial}{\partial x_i}\left(\frac{\mu_{eff}}{\sigma_k}\frac{\partial k}{\partial x_i}\right) + \rho(G - \varepsilon) \quad\text{............(3)}$$

$$\frac{\partial}{\partial x_j}\rho c_p(\overline{u_j T}) = \overline{u_j}\frac{\partial \overline{p}}{\partial x_j} + \overline{u_j'}\frac{\partial \overline{p'}}{\partial x_j} + \frac{\partial}{\partial x_j}\left(k\frac{\partial \overline{T}}{\partial x_j} - \rho c_p \overline{T' u_j'}\right) \text{.....(4)}$$

$$\frac{\partial}{\partial x_i}\left(\rho \overline{u_i}\varepsilon\right) = -\frac{\partial}{\partial x_i}\left(\frac{\mu_{eff}}{\sigma_k}\frac{\partial \varepsilon}{\partial x_i}\right) + \rho\frac{\varepsilon}{k}\left[\left(c_1 + c_3\frac{G}{\varepsilon}\right)G - C_2\varepsilon\right] \text{.....(5)}$$

where

$$G = \frac{\mu_t}{\rho}\left[2\left(\frac{\partial u_i}{\partial x_i}\right)^2 - \frac{2}{3}(\nabla u_i)^2\right]$$

$$\mu_{eff} = \mu + \mu_t$$

$$\mu_t = \rho c_\mu \frac{k^2}{\varepsilon}$$

$$c_\mu = 0.09, c_1 = 1.15, c_2 = 1.90, c_3 = 0.25, \sigma_k = 0.75,$$

The heat conduction of the solid housing is expressed as

$$\frac{\partial^2 T_S}{\partial x_j} = q'' \text{.......... ....(6)}$$

Because of the interface created between the solid and fluid region, continuity of energy and heat flux can be stated as

$$-k_{solid}\frac{\partial T_{Solid}}{\partial n} = -k_{fluid}\frac{\partial T_{fluid}}{\partial n}; T_{Solid} = T_{fluid} \text{..........(7)}$$

The local heat transfer coefficient and average heat transfer are given as

$$h = \frac{q^n}{T_w - T_b} \text{...(8)}$$

$$\bar{h} = \frac{\int h.dA}{\int dA} \text{.......... ....(9)}$$

$$T_b = \frac{\int v.T_f dA}{\int v dA} \text{...(10)}$$

Where, dA is an infinitesimal area of the wall surface and ‘v’ is the local fluid velocity. The equations from (1) - (10) will be used by the numerical tool to solve problem.

B. Numerical Model

In this work a numerical model is developed, using standard k-epsilon turbulence model. Turbulence intensity was taken as 15%. Polyhedral finite volume element was used to discretize the solid and fluid domain with a base size of 5mm. A conformal interface was made between the solid and fluid boundary to enable energy transfer. A suitable grid independence test was performed to ensure the accuracy and validity of the results.

Fig. 3 shows the meshed model of the fluid and solid domain. Along with residual plot other parameters such as outlet velocity, mass flow rate and outlet temperatures were monitored to ensure the convergence of the solution. Convergence criterion was satisfied when the residual of the entire plot reached1.0x10^-4.Boundry conditions used for the model are the following:

For Fluid domain

- Inlet : velocity inlet boundary normal
- Outlet : pressure outlet (gauge pressure)

For solid domain

- Heat input : heat flux

The input values for turbulence model used are given in Table 1.Computation time consumed were in the range of 20-30 hours each case, with Pentium Dual- core (RAM-1GB)

Flow rate (L/min)	Kinetic energy (J/kg)	Dissipation rate (J/kg-sec)
0.4	0.004	0.08
0.8	0.014	0.63
1.2	0.031	2.13

Table 1: Input parameters for the turbulence model

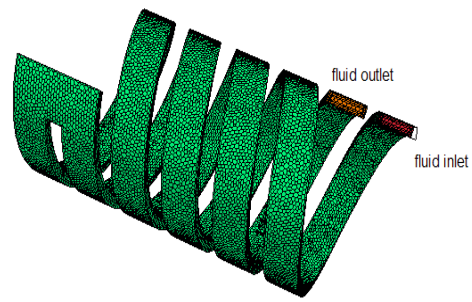

Fig 3: Fluid channel grid

Fig. 4 shows distribution of local heat transfer coefficient in the cooling passage. The local heat transfer coefficient values have increased to 6000W/m2K at the junction of helical coils. This is because of the drastic change in the fluid passage geometry, which could increase the velocity and cause the pressure to drop.

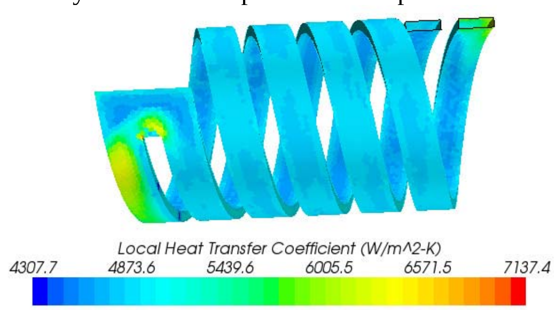

Fig 4: Local heat transfer coefficient distribution of cooling channel for the flow rate Fr=0.8 L/min

Fig. 5 is evident to fact that there is an increase in velocity from 0.48 to 0.78 m/sec. This is because of change in fluid passage, which increases turbulence and hence leading to increase in local velocity.

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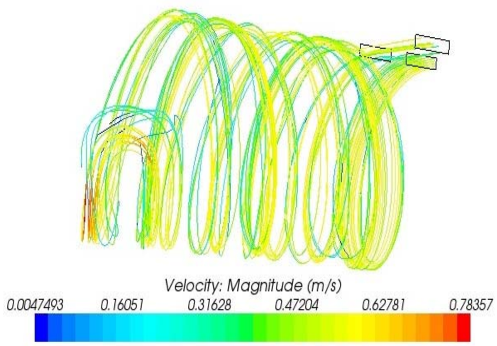

Fig 5: Velocity vector of the fluid flow for the flow rate Fr=0.8 L/min

C. Comparison of numerical and experimental results

From the numerical analysis the average heat transfer coefficients were predicted for the flow rates of 0.4L/min,0.8L/min and1.2 L/min. Fig. 6 shows the comparative plot for average heat transfer coefficients.

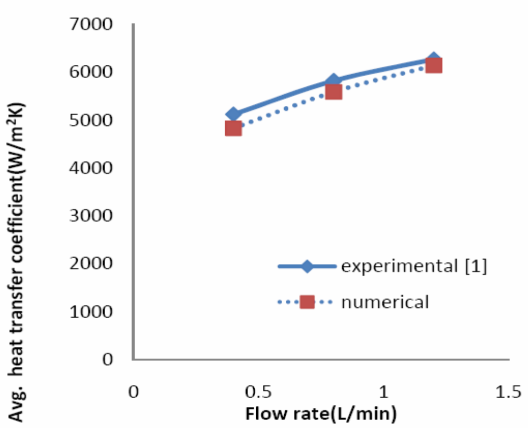

Fig 6: Numerical and experimental average heat transfer coefficient for the flow rates of 0.4L/min,0.8L/min and1.2 L/min

It is evident from the plot, of the fact that, heat transfer coefficient increases with increase in flow rate. The error percentages between the numerical and experimental results are tabulated in the Table 2.

Flow rate (L/min)	Avg. Heat transfer coefficient (W/m^2K)		Error (%)
	Numerical	Experimental[1]	
0.4	4825.708	5119.78	5.74
0.8	5584.304	5816.224	3.98
1.2	6136.754	6266.744	2.07

Table 2: Error estimation between numerical and experimental results

The results obtained from the numerical analysis are in good agreement with the experimental data [5].Further this model can be used for analysis of machine tool spindle.

III. ANALYSIS OF MACHINE TOOL SPINDLE

Numerical analysis will be performed for a typical machine tool spindle. Spindle has a capacity of 4.7 kW and maximum speed of 40,000 rpm. The heat generated by the motor is carried away by the coolant through the cooling channel. The coolant used is SERVOSPIN 2 oil, which is pumped by a coolant pump with capacity of 20L/min-10L/min. A chiler unit is incorporated with the spindle having capacity of 2500 Kcal/hr. It can maintain the inlet temperature of the coolant from 18ºC to 24ºC. The 2D model of the spindle is shown in Fig. 7.The cooling channel has a square cross section of 6x6mm, with a pitch of 13.5 mm. The sweep length is limited to 75 mm. Now a numerical model is developed for the specifications of 4.7 kW spindle.

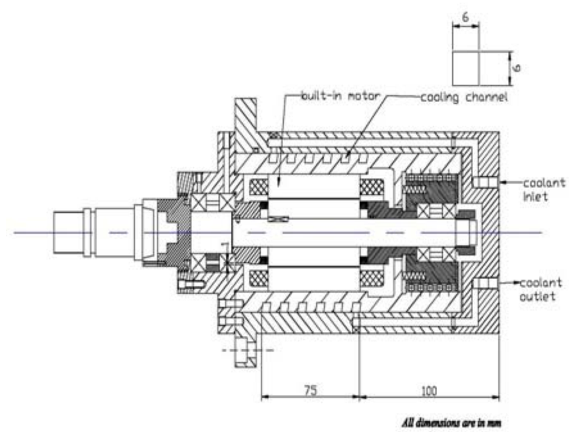

Fig 7: Physical model of 4.7 kW spindle

The following were the assumptions made for the ease of analysis:
1. The fluid is assumed to be incompressible.
2. The flow is fully developed and turbulent.
3. Convection coefficient of 10 W/m2K is applied on the housing for natural convection.
4. Radiation effects are ignored.

Cross sectional view of 4.7 kW machine tool spindle is depicted in the Fig. 7.It shows that the placement of cooling channels is just above the built-in motor, which is the major heat source.

A. Numerical Results for Double Start Helical Channel In 4.7 kW Spindle

The methodology adopted for the validated model is extended to the machine tool spindle, for the numerical analysis. Table 3 shows the predicted average heat transfer coefficient for the flow rates 20L/min, 15L/min and 10L/min. For the flow rate of 20 L/min the average heat transfer coefficient is maximum.

Flow rate (L/min)	Avg. heat transfer coefficient (W/m^2-K)
20	3016.08
15	2391.53
10	1734.78

Table 3: Predicted average heat transfer coefficients

Fig .8 shows the local heat transfer coefficient distribution for the flow rate of 20 L/min. The heat transfer coefficient is maximum at the junction of the channels, due to higher turbulence.

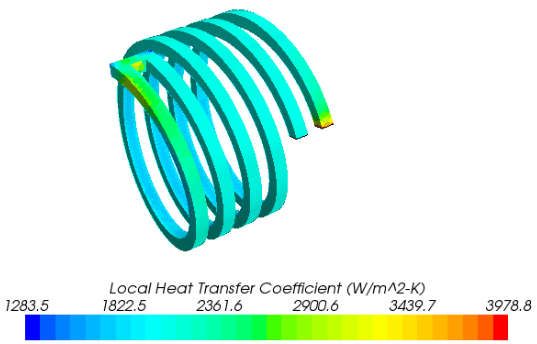

Fig 8: Local heat transfer coefficient distribution of cooling channel for the flow rate Fr=20 L/min

B. *Alternative Designs for Cooling Channel*

Apart from double start helical channel design there are various other designs which are used by the spindle manufacturing units. Most commonly seen designs for cooling channel are shown in the Fig. 9 and 10.

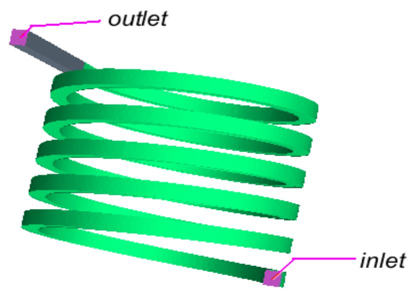

Fig 9: Single start helical coil

These cooling channels designs are adopted in spindles may be because of manufacturing feasibility and better performance. The right justification cannot be made, unless these designs are compared under same working conditions. Thus a comparative study is made numerically for all these designs based on two important factors .The two factors are average heat transfer coefficient and pressure drop. Apart from the heat transfer requirements an important consideration in cooling channel design is pressure drop or pumping cost. The cooling effect can be increased by increasing the inlet velocity of the coolant. But higher velocity will cause a huge pressure drop and correspondingly large pumping cost.

The power requirement to pump fluid in steady state is given by

$$power = \int v dp = \frac{\dot{m}}{\rho}\Delta p \cong \dot{m}^3$$

where m is the mass flow rate of flow

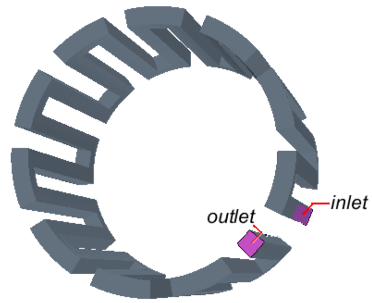

Fig 10: Axial channel

CAD models for single start helical coil and axial channel were made with a pitch of 13.5 mm , the cross section of 6x6mm and sweep length of 75 mm, same as that of double start helical channel used in 4.7 kW spindle.

C. *Numerical results for single start helical and axial channel*

From the numerical analysis average heat transfer coefficient and pressure drop was predicted. Fig. 11 and Fig.12 shows the local heat transfer distribution for single start helical coil and axial channel.

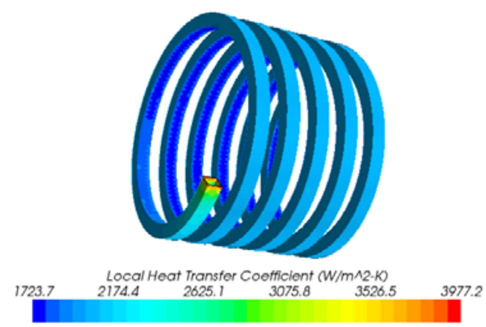

Fig 11: Local heat transfer coefficient distribution of cooling channel for the flow rate Fr=20 L/min

For the single start helical channel there is no significant increase in local heat transfer coefficient in the cooling passage.

But for the axial channels local heat transfer coefficient can be seen around 3500 W/m2K at all the junctions. This is a significant increase in local heat transfer coefficient when compared to single start helical channel.

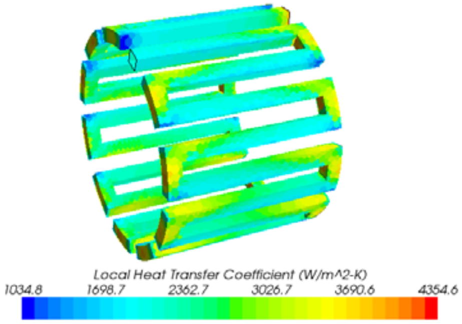

Fig 12: Local heat transfer coefficient distribution of cooling channel for the flow rate Fr=20 L/min

Table 4. shows the predicted average heat transfer coefficients for single start helical channel and axial channel. The average heat transfer coefficients for single start helical channel is comparatively lower than the axial channel.

Flow rate (L/min)	Avg. heat transfer coefficient channel (W/m^2-K)	
	Single start helical channel	Axial channel
20	2523.39	2969.87
15	2020.66	2350.75
10	1485.38	1700.25

Table 4: Predicted average heat transfer coefficients for single start helical channel and axial channel

Table 5. presents the pressure drop of single start helical channel and axial channels. The pressure drop for axial channel is higher due to drastic change in fluid passage geometry. This can result in a huge pumping loss.

Flow rate (L/min)	Pressure drop (bar)	
	Single start helical channel	Axial channel
20	0.29	14.5

Table 5: Predicted pressure drop for single start helical channel and axial channel

D. Comparative Analysis of Studied Designs

Thus from the numerical results of the three cooling channel designs, a comparative study can be made and a better design can be proposed.

Fig. 13 presents the comparative plot for the average heat transfer coefficient for all the three designs of cooling channel, namely double start helical channel, single start helical channel and axial channel.

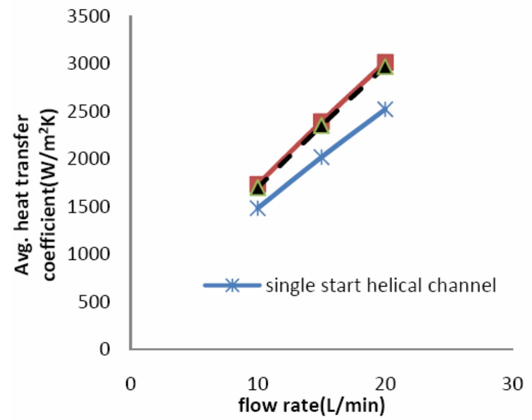

Fig 13: Comparative plot for double start

The plot shows that the average heat transfer coefficient for double start helical channel is higher than other two designs. Table 6. presents the comparative results of pressure drop for all the three designs.

Flow rate (L/min)	Pressure drop (bar)		
	Double start channel	Single start channel	Axial channel
20	2.98	0.29	14.5

Table 6: Pressure drop

Axial cooling channel will cost a huge pumping loss where the pressure drop is 14.5 bar. For the single start helical channel design the pressure drop is minimal, whereas for the double start helical channel design pressure drop is 2.98 bar. For the right selection of the cooling channel design, a compromise between average heat transfer coefficient and pressure drop has to be made.

IV. CONCLUSION

Generally high speed machines are prone to uneven thermal deformations. Thus it is required to develop a thermal model to predict the temperature distribution. For a better thermal modeling of high speed spindles, CFD can be recommended as a better tool to predict the heat transfer coefficient and to access the performance of the cooling channel design. From the comparative study of cooling channel designs it can be concluded that double start helical channel has a better average heat transfer coefficient value and with an acceptable pressure drop of 2.98 bars .Though single start helical channel experiences minimum pressure drop of 0.29 bar, it cannot be used because of its lower average heat transfer coefficient value. Thus by making a compromise between pressure drop and average heat transfer coefficient, double start helical channel is suggested as a better design. During the numerical analysis k-_ turbulence model was used. The numerical results were satisfactory when compared to the experimental results and hence k-_ turbulence model can be a better model to solve conjugate heat transfer problems. This work can be further extended by optimizing the cooling channel geometry for minimum pressure drop and maximum average heat transfer coefficient.

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