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An Optimization Analysis of Cooling Channel Design in a CNC Lathe Spindle System

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Keywords: machine tools, thermal error, spindle, multiphysics analysis, Taguchi method.

ABSTRACT

In this study the heat transfer phenomena, thermal deformation and the design of cooling channels for a standard Computer Numerical Control (CNC) lathe spindle system were conducted numerically. The heat sources taken into consideration include the rotary hydraulic cylinder, the bearings and the spindle pulley. Discussions and conclusions are based on numerical simulations of temperature variations and structural thermal displacement. To reduce the degree of thermal deformation a system of cooling channels was designed for the spindle and tested numerically. Design parameters investigated of the cooling channels were the diameter of the channels (6, 8 and 10mm), the number of channels (4, 6 and 8) and the coolant mass flowrate (0.085, 0.113 and 0.142 kg/s). The Taguchi method was used with simulations of nine sets of combinations of these parameters. The numerical results showed that the best cooling channel design among the aforementioned design parameters has 8 channels, 8mm in diameter and a mass flowrate of 0.085 kg/s. It was also noticed that, when optimizing the cooling system with design parameters comprising the geometrical dimensions of the channel and the coolant flow rate, extra attentions should be taken to assure the independency between these design factors, especially on the flowrate of the coolant to the other design parameters. The method and the results of this study provide a useful reference for the design and optimization of cooling channels for spindle systems in machine tools.

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INTRODUCTION

The recent and very rapid progress of industry towards “Industry 4.0” has made it necessary for the machine tool industry to embrace smart, high precision, and very efficient hybrid technology. Final machining precision requirements are determined by the precision of key components and whole machine assembly of machine tools. It has therefore become extremely important that the machining precision and stability of machine tools should be effectively improved. Machine tools are made up of many components and each one of them contributes thermal errors to some extent. Among them the heat created by the running spindle can create thermal errors that seriously affect the quality of finished work (Mayr, 2012). Therefore, the spindle system is the most crucial component involved in the machining process. In recent years, much work has been done on the design of more efficient cooling systems for spindle assemblies. In most of these, circulating coolant is used to control spindle temperature changes, suppress thermal deformation and maintain cutting stability and precision. Keeping the working temperature down also reduces the wear of vital components such as the bearings, collets and mandrels. In the past, thermal deformation and errors have been analyzed and controlled with thermal compensation techniques based on the results of simplified models and measurement and made real machines (Li, 2015). However, manufacturers and researchers are now focusing more on how to suppress thermal deformation effectively and appropriately, from the design perspective, rather than by compensating for them. The traditional repetitive trials and corrections are being replaced by scientific and data oriented design analysis. Multiphysics numerical analysis of machine tools and spindles has thus been widely applied by designers to minimize thermal errors in recent years (Ma, 2015).

When the spindle is running, most of the heat that contributes to spindle head deformation comes from bearing friction (Harris, 2006). The main factors related to bearing heat include the preload, bearing specifications and speed of rotation. The parameters affecting temperature distribution of the spindle head come from factors such as heat source location, geometric shape, the thermal properties of the materials, heat transfer from the environment and so on. Even with cooling systems of the machine tool, part of the heat is still transferred to other structures near the spindle, raising their temperature and causing deformation or distortion of components. This situation is more severe when the spindle is rotating at high speed since more heat is generated from the bearings. The thermal deformation of the bearings themselves also increases wear and failure. In addition, energy consumption of the spindle system including its cooling system is significant for high speed spindle (Ho, 2017). Thus, efficient conduction of heat away from the bearings can reduce the stress and strain created by thermal deformation, and saving energy. This will not only increase the stability and precision of the rotating spindle but also reduce the chance of deterioration and damage caused by thermal deformation.

In addition to cooling the spindle and avoiding overheating, the handling of temperature distribution of the spindle is also very important. In general, more uniform temperature distribution of the spindle leads to less thermal bendings of it and consequently less complexity in thermal error compensation (Ko, 2003). Numerical and experimental studies have been conducted to the cooling system designs for CNC machine tools. For example, Chien et al. used a spindle sleeve with helical cooling channels in their study and demonstrated a significant drop of sleeve temperature (Chien, 2008). They further analyzed the heat transfer properties of cooling channels with different geometries and compared the cooling effects of spindle sleeves with single- and double-helical cooling channels. The comparison of heat transfer performance and pressure drops between the two designs of cooling channels showed that a double-helical cooling channel had greater cooling capability and yields better temperature uniformity. Xia et al. compared the spiral cooling channel designs with branch-like spindle sleeves (Xia, 2015). The results revealed that this branch-like design created a lower pressure drop and more even temperature distribution on the spindle. Experimental results also showed that heat dissipation in the branch-like configuration was better and the temperature was more effectively lowered than in the helical design. Huang el al. (Huang, 2016) also concluded that, based on experimental and numerical analysis, the reciprocal cooling channel design in the sleeve has better heat transfer performance than single- or double-helical cooling channels. Other published works also showed that the geometric design of the spindle cooling channels and the mass flow rate of cooling fluid are both important factors related to temperature changes and spindle temperature distribution. On the other hand, optimization methodologies such as the Taguchi method in conjunction with computer aided engineering (CAE) have also been applied on improving thermal deformation of machine tools. Mori et al. (Mori, 2009) proposed the use of the Finite Element Method (FEM) and the Taguchi method in designing an optimal spindle head, the objective being to lower thermal displacement at the center of the spindle and increase the structural precision of the lathe headstock. The Taguchi method has also been used in optimizing the design of spindle motors (Hwang, 2017). In this regard, the Taguchi method is, in general, applied to construct Design of Experiments (DOE) for simulated experiments with an orthogonal array of design parameters. The optimal design parameters could then be attained by statistically minimizing the thermal deformation at the cutting point of the machine tool.

In this study a set of optimal method based on the Taguchi method in conjunction with multiphysics analysis for machine tool spindle cooling channel design in the sleeve of a typical CNC lathe, as shown in Figure 1, was conducted numerically. The heat generated by the spindle is removed by the cooling fluid flowing in the channels as shown in Figure 2. Finite element analysis software ANSYS® was used for fluid-thermal-solid coupling analysis and to build a numerical simulation and analysis model for the spindle thermal displacement of the spindle headstock system. The Taguchi method was used for planning the number of simulation analysis groups, i.e. the design of simulated experiments, based on different numbers of channels, channel diameter and mass flow rate of the coolant, three levels at each, as shown in Table 1. The distribution of the temperature field and the change of thermal deformation of a running spindle under these different conditions were studied as well as adjustment of the design parameters to achieve the best control of temperature rise and thermal displacement.

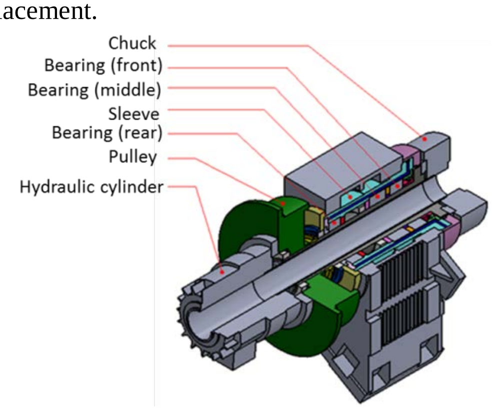


Fig. 1. Schematic diagram of the lathe spindle system in current study.

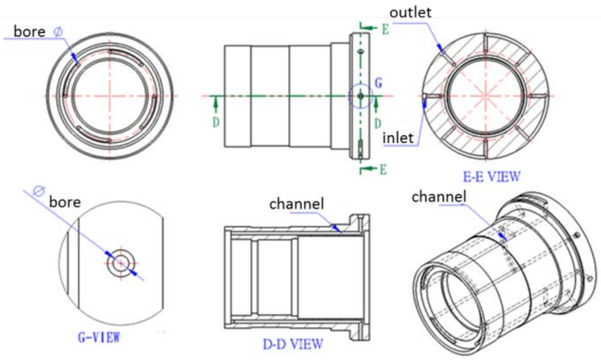


Fig. 2. Schematic diagram of the spindle sleeve cooling channel design

Table 1. $L_9\,(3^3)$ orthogonal array and corresponding values of control factors for the cooling system

case numbers	level			number of channels	bore diameter (mm)	mass flowrate (kg/s)
	1	2	3			
1	1	1	1	4	6	0.085
2	1	2	2	4	8	0.113
3	1	3	3	4	10	0.142
4	2	1	2	6	6	0.113
5	2	2	3	6	8	0.142
6	2	3	1	6	10	0.085
7	3	1	3	8	6	0.142
8	3	2	1	8	8	0.085
9	3	3	2	8	10	0.113

METHODOLOGY

An L_9 (3×3) Taguchi orthogonal array as shown in Table 1 was used in this study to perform analysis of the cooling channel designs with different numbers of channels (parameter A: 4, 6 and 8 channels), channel diameters (parameter B: 6, 8 and 10mm) and mass flowrates of the coolant (parameter C: 0.085, 0.134 and 0.142 kg/s). The Taguchi method considerably reduces the number of cases and thus the time taken for analysis. For instance, in the current study, only nine groups of simulation analyses (see Table 1) need be performed and not 27 as originally planned for three factors with three levels.

The structure of the headstock and the interior space of the spindle system under simulation are shown in Figure 1. VG 32 cooling oil was assumed to be used as the coolant. It is pointed out that the structure and other specification of the simulated lathe headstock were based on a real CNC lathe from a local manufacturer.

In the numerical simulation, steady state continuity equation, momentum equation and heat transfer equation were solved simultaneously using ANSYS Fluent® in the fluid region:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}\delta_{ij}\frac{\partial u_k}{\partial x_k}\right] + \rho\vec{g} + \frac{\partial}{\partial x_j}\left(-\rho\overline{u'_i u'_j}\right) \tag{2}$$

$$\frac{\partial}{\partial x_i}[u_i(\rho E + p)] = \frac{\partial}{\partial x_j}\left[k_{eff}\frac{\partial T}{\partial x_j} + u_i(\tau_{ij,eff})\right] \tag{3}$$

Where the last term in equation (2) denotes the Reynolds stress (i.e. the turbulence effects) that was computed by using Standard k-ε model. The effective thermal conductivity of the fluid, k_{eff} and the deviatoric stress, $\tau_{ij,eff}$, in the energy equation also take the turbulence effects into account.

For solid region, the standard heat conduction equation considering volumetric heat sources in the solids as will be discussed later was solved in conjunction with the fluids on the surroundings for temperature distributions:

$$\nabla \cdot (k\nabla T) + S_h = 0 \tag{4}$$

With the knowledge of the temperature distributions in the structures, the thermal deformation of the structure was calculated following the thermoelasticity model in ANSYS®.

As shown in Figure 3, the bearings, the belt pulley and the rotary hydraulic cylinder are the main heat sources in the spindle system. When the spindle is running, heat is generated by friction in the ball bearings in the front, middle and rear of the spindle, and transferred to the surrounding structures. The bearings were defined as volumetric heat sources in our simulation. Heat is also generated by friction between the surfaces of the belt and the pulley and surface heat flux is determined by the contact surfaces between them. The amount of heat being generated from the bearings and pulley was estimated by using the empirical theories in the reference (Harris, 2006). The jaws of the collet are tightened or loosened by the draw tube which is controlled by the hydraulic pressure generated by the rotating hydraulic cylinder. The internal components of the hydraulic cylinder were ignored in this study for simplicity, and the whole cylinder is defined as a volumetric heat source because its oil temperature is raised significantly by the heat generated by the bearings during operation. The heat sources of the front, middle and rear bearings, the rotary hydraulic cylinder and the heat flux from the belt pulley are calculated using empirical formulae and are the setting values used for the main heat-generating components in the whole spindle system, as shown in Table 2. It should be pointed out that the amount of heat being generated in each component listed in Table 2, with considering the volume, is compatible to typical spindle systems (Ma, 2015). It is emphasized that the current study focused on the effect of the cooling channel designs in the sleeve. Therefore, the flow field and heat transfer of the moving belt was ignored in this study.

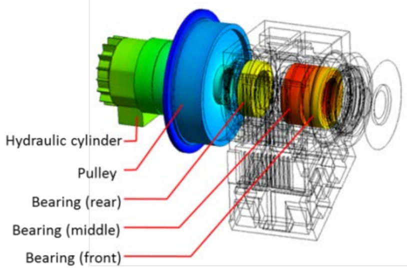


Fig. 3. Schematic diagram of heat source distribution in the spindle system.

Table 2. Heat generated by major heat sources considered in the lathe spindle system

	Rotary hydraulic cylinder	Pulley	Rear bearing	Middle bearing	Front baring
Heat source (kW/m^3)	11.5	0.185	34.8	100.4	60.4

As shown in Figure 2, a re-circulating design for the channels in the spindle sleeve was used in this study. The temperature and mass flowrate of the coolant according to Table 1 are set at the inlet of the channel. The boundary condition for the fluid at the outlet of the channels is 1 atm (i.e., gauge pressure = 0).

RESULTS AND DISCUSSIONS

1. Lathe spindle system without cooling channel

In this study, analysis of the lathe spindle temperature distribution and fluid flow fields was done with the spindle running at a fixed speed at 3000rpm. To start with, simulation of a spindle with no cooling channels was conducted as the baseline case and Figure 4 shows the temperature distribution in the interior and exterior of the spindle. The high temperature field is located near the bearings, the highest being on the front and middle bearings near the front end of the mandrel. The temperature of the structures near the rotary hydraulic cylinder and belt pulley, which are main heat-generating components, is also significantly higher than room temperature.

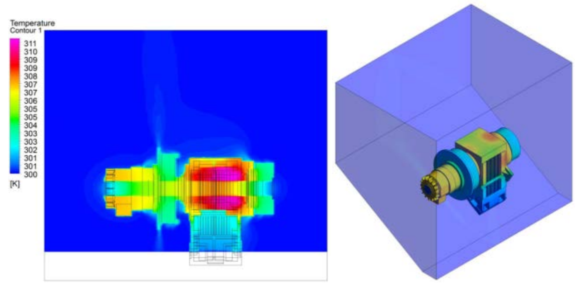


Fig. 4. Temperature field distribution inside the lathe spindle system (base). (a): interior, cross-section along the center of the spindle (b) exterior of the spindle system.

The velocity distribution of air in the spindle system is shown in Figure 5. The simulation results reveal that the heat generated by the internal bearings will heat the air inside spindle head structure and induce natural convection, but this effect is not significant. The tight inner space inside the structure makes hot air discharge and cold air intake very difficult and reduced heat dissipation causes the temperature of the structure to rise appreciably. External air flow fields, like those driven by the belt pulley, are more obvious.

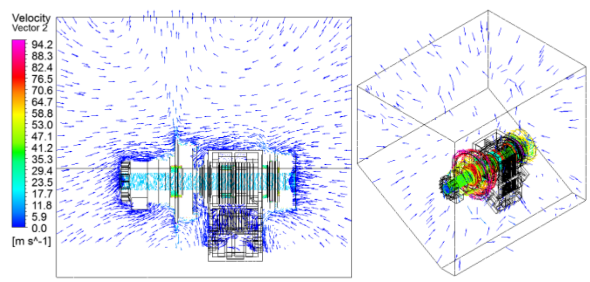
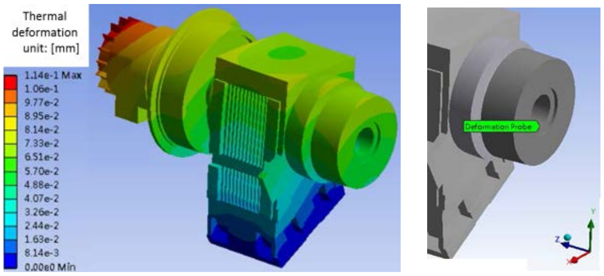


Fig. 5. Distribution of flow fields of the lathe spindle system (base). (a): interior, cross-section along the center of the spindle (b) exterior of the spindle system.

A simulation of the thermal deformation of the spindle head structure for the base case is shown in Figure 6(a). It can be seen that the greatest thermal deformation occurs in the rotary hydraulic cylinder. Thermal deformation of the cylinder is subject to less constraint because it is far away from the fixed boundary condition of the headstock that assumed to be perfectly anchored on the base of the lathe. However, the thermal displacement of the nose of the spindle where workpiece is mounted, i.e. the cutting point is more important. To compare and analyze the influence of thermal errors on the cutting point (between different cooling system designs at a later time) a fixed probing point was specified on the collet chuck, as shown in Figure 6(b). The thermal displacements of the probe point of the spindle system, without a cooling channel system, are shown in Table 3.



(a) (b)

Fig. 6. Thermal deformation (a) and the probing position of the spindle system.

2. Lathe spindle system with cooling channels

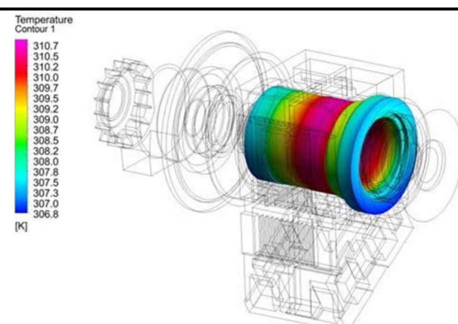
To improve the heat transfer of the running lathe spindle system, a design of single inlet/outlet recirculation cooling channels inside the spindle sleeve has been adopted. The injection of coolant and its temperature are assumed to be managed by an oil cooler unit.

As mentioned earlier, the greatest heat sources in such a spindle system are the three bearings. The reason for having the cooling channels inside the spindle sleeve is because the sleeve is the nearest structure to these heat sources and also presents a feasible housing for cooling channels. The recirculating coolant flows through the channels, absorbs heat in the higher temperature regions, and is cooled gradually by the structure when it reaches the lower temperature regions (Huang, 2016). The spindle sleeve channel design parameters used in this study are the number of longitudinal channels, their diameter and the flowrate of the coolant. An important consideration, when deciding the number of cooling channels and their diameter, was not to change the original exterior geometric dimensions of the spindle sleeve, so the dimension of the spindle head is not affected with adding cooling channels. The choice of the number of channels was 4, 6 or 8 with diameters of 6, 8 or 10mm. The mass flowrates of coolant used in the simulations were 0.085, 0.113 and 0.142kg/s, decided with reference to 30%, 40% and 50% of the highest flow capacity of oil the coolers commonly used with this type of lathe. The coolant temperature at the channel inlet was fixed at 293K.

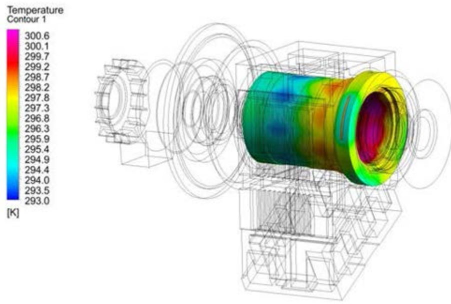
Nine groups of cooling channel design were studied using the Taguchi method, as shown in Table 3. The simulation results reveal that the spindle temperature drops significantly with the use of a cooling sleeve, as also shown in Figure 8. The overall temperature drop in the spindle head becomes more obvious and temperature is distributed more evenly in general with an increase in the number of channels. Thus, the use of cooling channel is accompanied by an obvious improvement in overall thermal displacement.

Table 3. Thermal displacements in the lathe spindle system

	channels-bore -diameter(mm) -mass flowrate(kg/s)	X-thermal displacement ΔX (mm)	Y- thermal displacement ΔY (mm)	Z- thermal displacement ΔZ (mm)	Overall thermal displacement $\lvert\Delta X\rvert+\lvert\Delta Y\rvert+\lvert\Delta Z\rvert$
base	0-0-0	6.46×10^{-3}	3.76×10^{-3}	-3.65×10^{-3}	1.39×10^{-2}
1	4-6-0.085	3.89×10^{-3}	1.34×10^{-2}	-1.06×10^{-2}	2.79×10^{-2}
2	4-8-0.113	1.03×10^{-3}	-5.21×10^{-4}	-7.78×10^{-3}	9.34×10^{-3}
3	4-10-0.142	2.77×10^{-3}	6.47×10^{-3}	-3.13×10^{-3}	1.24×10^{-2}
4	6-6-0.113	8.69×10^{-4}	2.98×10^{-3}	-1.06×10^{-3}	4.91×10^{-3}
5	6-8-0.142	6.49×10^{-4}	2.16×10^{-3}	-1.21×10^{-3}	4.02×10^{-3}
6	6-10-0.085	9.81×10^{-4}	3.15×10^{-3}	-8.58×10^{-4}	4.99×10^{-3}
7	8-6-0.142	3.00×10^{-4}	-5.36×10^{-4}	2.24×10^{-3}	3.07×10^{-3}
8	8-8-0.085	8.63×10^{-4}	1.18×10^{-3}	6.54×10^{-4}	2.70×10^{-3}
9	8-10-0.113	1.13×10^{-3}	2.27×10^{-4}	2.87×10^{-3}	2.87×10^{-3}
10	8-8-0.142	1.62×10^{-3}	1.85×10^{-2}	-1.69×10^{-2}	3.70×10^{-2}



(a)



(b)

Fig. 8. Temperature of the spindle sleeve (a) without cooling channel, (b) with cooling channel (8 channels, bore diameter: 8mm, coolant flow rate: 0.085kg/s).

OPTIMIZATION AND VERIFICATION

The S/N ratios of the thermal displacement corresponding to the number of channels (A), the channel diameter (B) and the mass flow rate of coolant (C) were calculated, as shown in Figure 9. With the aim of reducing the thermal displacement as much as possible, the S/N ratio in this case was defined as:

$$S/_{N} = -10log\left(\frac{1}{n}\sum \beta_i^2\right) \tag{5}$$

where β is the overall thermal displacement of the representative point on the spindle as listed in Table 3, n is the number of samples for each set of control factor. In the current study, since the goal is to minimize the thermal displacement at the cutting point, the best combination of design parameters A, B and C should correspond to the biggest S/N ratio. Thus, the results shown in Figure 9 suggested that the best combination of design parameters were: 8 channels (A3), 8mm in diameter (B2) with a mass flow rate of 0.1134kg/s (C2). However, the simulation done with this combination of parameters (see group 10 in Table 3) is obviously not an optimal combination with the lowest total thermal displacement.

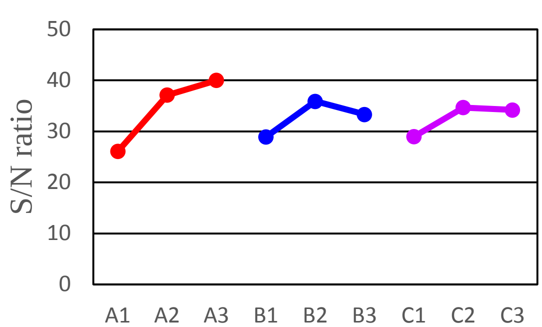


Fig. 9. The S/N ratios of number of channels (A), channel diameter (B) and mass flowrate (C).

A closer look at the results reveals that group 10 is not the optimal combination of parameters because the diameter change influences coolant flow speed at a constant mass flowrate. Therefore, the flowrate is not an independent parameter. Since convective heat transfer rate is actually related to the flow speed according to the dependency of the Nusselt number on the Reynolds number, flowrate is thus not an appropriate control factor in this case. Therefore the coolant flowrate was replaced by flow speed as a control factor. The flow speed in the nine groups of parameters was divided into three sets of levels (D1: high flow speed, D2: medium flow speed, and D3: low flow speed) as shown in Table 4. The S/N ratios were then calculated from the new control factors. And it was found that the total thermal displacement becomes smaller and the S/N ratio rises when flow speed gets smaller, see Figure 10. Therefore, the lowest flow speed (D3) instead of the medium mass flowrate (C2) should be chosen as a parameter for any specific channel diameter. A re-examination and comparison of the design parameters shown in Table 4 reveals that group 8 (A3, B2 and D3) has the lowest total thermal displacement of 2.70×10^{-3}mm. Group 8 was clearly the optimal combination of parameters, which matches the analytical prediction given by the Taguchi method.

Table 4. Cooling fluid flow speed conversion table

Number of channels	Bore diameter (mm)	Mass flowrate (kg/s)	Flow velocity (m/s)	Level of flow velocity
4	6	0.085	3.53	D1
4	8	0.113	2.65	D2
4	10	0.142	2.21	D2
6	6	0.113	4.71	D1
6	8	0.142	3.31	D2
6	10	0.085	1.27	D3
8	6	0.142	5.89	D1
8	8	0.085	1.98	D3
8	10	0.113	1.96	D3

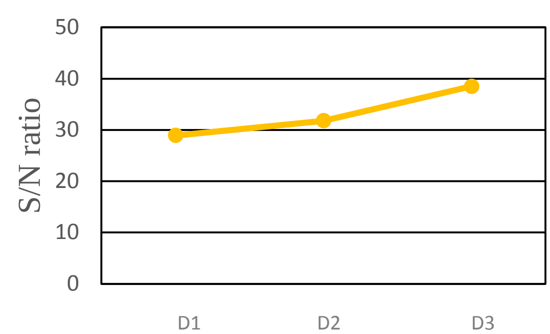


Fig. 10. The effect of coolant flow speed based on the S/N ratio.

CONCLUSIONS

The effects of the design parameters for a lathe spindle sleeve cooling system for control of the spindle head temperature and the reduction of thermal deformation were studied analytically. The design parameters include the number and diameter of channels in the spindle sleeve and mass flowrate of cooling fluid. Nine groups of parameters were used for simulation and the results were analyzed by the Taguchi method. The temperature of the spindle head was decreased significantly by cooling sleeves compared to a spindle headstock without cooling channels. The spindle temperature field and thermal deformation were also appreciably affected by different cooling sleeve design parameters. The optimal design of a cooling sleeve was confirmed by further exploration of the simulation results using the Taguchi method. Furthermore, the results of this study also showed that, when applying the Taguchi method, the flow speed of cooling fluid should be used as a control factor and design parameter rather than the flowrate. The reason is that the flowrate of the cooling fluid can interfere with another important design parameter for cooling channels, the channel diameter, and causes inaccuracy in the prediction from the analysis.

ACKNOWLEDGMENT

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NOMENCLATURE

E	total energy
$\vec{g}$	gravity
k_{eff}	effective thermal conductivity of fluids
p	pressure
S_h	volumetric heat source
T	temperature
u	fluid velocity
x	spatial coordinate
u'	velocity fluctuation due to turbulence
β	overall thermal displacement
δ_{ij}	Kronecker delta
μ	fluid viscosity
ρ	density
τ_{eff}	deviatoric stress

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臥式車床主軸頭熱變形模擬與冷卻流道優化分析

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摘要

本研究探討一典型臥式車床主軸系統熱傳現象以及散熱流道設計。首先根據臥式車床主軸在運轉的情況之下所產生的熱源,運用有限元素分析軟體,進行車床主軸系統的熱流固耦合熱傳模擬，探討這些熱源在車床主軸系統所產生的溫升與熱變形的現象,以及對整體車床主軸系統結構熱變位之影響。原本的主軸套管並無散熱流道的設計，所以本研究接著以模擬模型於主軸套管上設計內部往復式散熱流道，希望利用油冷機將冷卻液注入散熱流道進而適當地控制主軸頭結構溫度的變化與改善熱變形。本研究所探討的散熱流道設計參數為流道孔徑、流道數量與質量流率，依據田口法配對出 9 組不同的流道設計參數，接著使用有限元素分析軟體進行溫升與熱變形分析模擬。經由分析結果得知有流道設計的套管熱變形明顯減少；並且根據模擬結果的統計與分析,得知熱變位最小的設計參數,以及討論在運用田口法進行冷卻流道模擬實驗規劃與優化分析時應選擇的適當參數。本研究成果與研究方法可作為工具機主軸冷卻系統設計建模分析與優化的參考。