

Article

Research on Active Control Method for Thermal Error of Moving-Column Horizontal Machining Centers

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Abstract

This paper investigates the thermal characteristics of machine tools under the influence of multiple heat sources and proposes an active temperature control method. The study was conducted under steady-state conditions with a constant feed rate of 10 m/min and no cutting load. First, finite element simulations are performed to analyze the thermal characteristics of the machine tool subjected to internal heat sources, yielding the corresponding temperature and deformation fields. Second, within the tested temperature range, the thermal characteristics are examined with respect to the layout of temperature control devices, and the best-performing layout schemes for the bed and column are determined. Subsequently, the thermal error resulting from the coupling effect of multiple temperature control devices is studied, and an active control strategy is proposed to obtain the best layout configuration within the tested range. Finally, experiments are conducted using a multi-loop differential active temperature control system, and the validity of the proposed method is verified through comparison between simulation and experimental results. The findings demonstrate that the proposed active control method effectively reduces thermal deformation in machine tool structures.

Keywords: precision machine tools; thermal error; active control method; temperature control devices

1. Introduction

The accuracy of machined parts is greatly influenced by the precision of machine tools [1]. However, the machine tool structure is thermally deformed due to external and internal heat sources [2]. Many studies indicate that forty to seventy percent of machining errors originate from thermal errors [3–5].

Research methods on machine tool structure thermal error fall into two categories: (1) the thermal error model, which is established using finite element analysis, and (2) the relationship between structural components and thermal errors, which is established using thermal distortion theory and experimental results. These methods can be broadly classified into two types: one establishes the relationship between structural components and thermal errors through mathematical derivation based on thermal deformation theory and experimental data; the other models thermal errors by means of finite element analysis.



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Finite element analysis is a key method for studying thermal characteristics. Sun et al. [6] proposed a method to reduce the thermal displacement of machine tools during operation. Renowned machine tool manufacturers such as Siemens and Fanuc have also conducted research on thermal displacement compensation technology [7,8]. Mian et al. [9] proposed a method for characterizing and modeling the large gantry thermal characteristics of machine tools. Zhang et al. [10] proposed a simulation approach for thermal performance utilizing finite element analysis. Yun et al. [11] proposed a thermal error study of the feeding drive system of a CNC lathe and validated their proposed model through experiments. Kim et al. [12] employed the finite element method to analyze the thermal deformation. Weng et al. [13] proposed an analytical modeling method for transient thermal characterization of machine tool structural components and spindles. Li et al. [14] proposed a thermal error analytical model for fully closed-loop feed systems based on equivalent temperature fields (ETFs). Weng et al. [15] proposed a dual-layer analytical modeling approach.

Some scholars have proposed theoretical modeling methods based on numerical theory. Jiang et al. [16] proposed a method for measuring and modeling thermally induced positioning errors using temperature rise process repeatability. Wang et al. [17] proposed a computational model for analyzing the thermal deformation of bed and column structures. Yang et al. [18] proposed an integrated recurrent neural network model that can better track nonlinear and nonstationary thermal errors. Liu et al. [19] proposed a novel method for selecting KTMPs based on the effect of thermal deformations of components (ETDC).

Thermal error can be controlled through three effective methods: (1) Thermal error compensation is used: Errors caused by temperature changes are monitored and compensated for. Nguyen et al. [20] proposed a neural network-based model for predicting thermal deformation. (2) A thermal balance optimization design is employed: Machine tool structure, heat dissipation system, and other related components are optimized. Mori et al. [21] proposed a highly efficient design and optimization scheme for spindle box structures, identifying key design features and optimal structures that are used to reduce thermal displacement. Xia et al. [22] proposed a novel fractal dendritic channel heat dissipation network design for cooling high-speed electric spindles. (3) Active control technology is used for machine tool temperature: The temperature is regulated in an active way by specific technical means. Mitsuishi et al. [23] proposed a novel heat exchanger device, which is installed on the column to control its temperature. Wang et al. [24] proposed an online monitoring and diagnosis method based on the latent variable modeling (LVM) approach, which effectively carries out variable selection and thermal state monitoring with limited sensing resources. Most thermal error compensation methods rely on numerical control systems to achieve position compensation, whereas thermal balance optimization methods are intended only to prevent thermal deformation by optimizing the machine tool structure at the design stage. By contrast, active temperature control methods can achieve real-time temperature regulation.

A horizontal machining center with a moving column is focused on in this article, and the finite element simulation method is employed to analyze the thermal characteristics of the machine tool. Additionally, an active control method for thermal errors is proposed with multi-channel differentiated cooling.

2. Thermal Deformation Simulation Under Inner Heat Sources in Machine Tool

2.1. Distribution of Machine Tool Internal Heat Sources

In Figure 1, the major internal heat sources are located at the motor, bearing seat, guide rail, and motor seat.

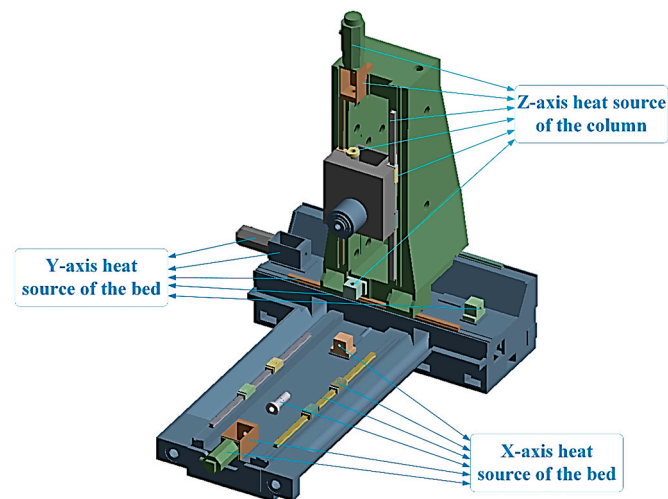


Figure 1. Internal heat sources of a precision machine tool.

2.2. Finite Element Modeling

The material properties of the individual components are presented as follows: structural components are cast iron with an elastic modulus of 1.30×10^{11} Pa, Poisson's ratio of 0.3, and a density of 7030 kg/m^3 ; guide rails and sliders are grade 45 steel, exhibiting an elastic modulus of 2.00×10^{11} Pa, Poisson's ratio of 0.3, and a density of 7850 kg/m^3 ; and the remaining functional units are also grade 45 steel, with an elastic modulus of 2.06×10^{11} Pa, Poisson's ratio of 0.3, and a density of 7850 kg/m^3 . The motion accuracy of the guide rail slider system is focused on in this study. The simulation results are negligibly affected by certain detailed features in the machine tool model, such as holes, slots, and chamfers. Therefore, to ensure simulation accuracy while considering computational efficiency, these subtle structures are removed. This simplification facilitates the establishment of a more effective heat dissipation model, reduces simulation time, and accelerates the research process without compromising quality. Since little effect is exerted on the simulation results by some subtle structures, the machine tool structure should be simplified. The axis structure and screw between the workbench and the base are eliminated, and the simplified structure is shown in Figure 2a.

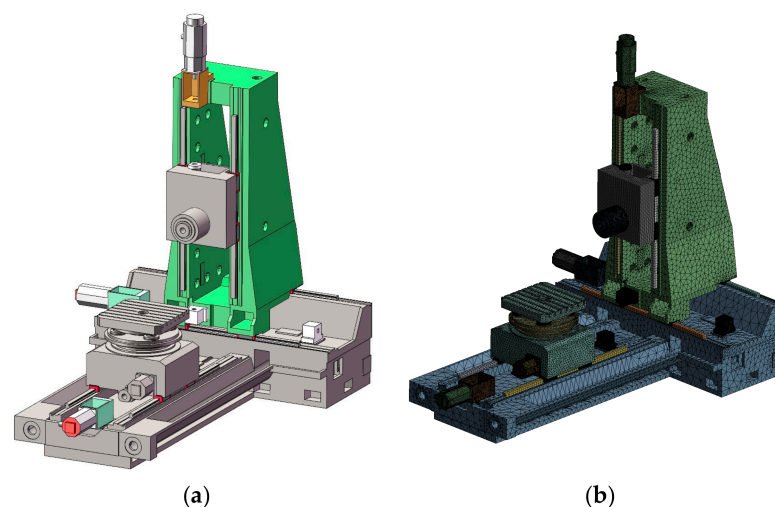


Figure 2. (a) Simplified model of a precision machine tool; (b) mesh model.

The main structural components, such as the bed, column, and guide rails, are made of structural steel with a density of 7850 kg/m^3 , Young's modulus of 2×10^{11} Pa, Poisson's

ratio of 0.3, a coefficient of thermal expansion of $1.2 \times 10^{-5} \text{ }^{\circ}\text{C}^{-1}$, a thermal conductivity of $60.5 \text{ W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$, and specific heat at a constant pressure of $434 \text{ J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$. For the mesh, free meshing was used with no fixed global element size; the minimum edge length was set to $4.5393 \times 10^{-4} \text{ m}$; a medium mesh smoothing level was applied; no automatic inflation was employed, with the inflation algorithm set to pre and the element type set to wedge; mesh quality check was enabled with a mechanical strength error limit; the total number of nodes was 5,844,950 and the total number of elements was 1,711,962; and Solid70 8-node hexahedral thermal solid elements were used for thermal analysis, while Solid186 20-node hexahedral and Solid187 10-node tetrahedral solid elements were used for structural analysis. A bonded contact was assigned to the bed–column bolted joint surface with a thermal contact conductance of $800 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$, representing a bolt-clamped cast iron–cast iron contact surface; bonded contact was assigned to the guide rail–slider assembly surface with a thermal contact conductance of $1200 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$, representing metal-to-metal contact with a lubricating oil film; and bonded contact was assigned to the motor seat–machine base contact surface with a thermal contact conductance of $1500 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$, representing the motor mounting interface. A uniform initial temperature was applied as the initial thermal temperature; the thermal-structural reference temperature was specified as the reference temperature for thermal deformation calculation; a fixed support was applied to the bottom mounting surface of the bed, with all translational and rotational degrees of freedom constrained and no additional displacement constraints imposed, so that all deformation was generated solely by thermal strain; the steady-state thermal convergence criterion was based on the energy residual with a maximum number of iterations of 50; and the static structural convergence criterion was based on the force residual and the displacement increment convergence tolerance. For convective boundary conditions, all outer surfaces of the worktable were assigned a convective heat transfer coefficient of $16 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ with an ambient reference temperature of $20 \text{ }^{\circ}\text{C}$; the exposed surfaces of the spindle box housing were assigned a convective heat transfer coefficient of $22.3 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ with an ambient reference temperature of $20 \text{ }^{\circ}\text{C}$; and all exposed surfaces of the bed, column, and motor housings were assigned a convective heat transfer coefficient of $9.7 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ with an ambient reference temperature of $20 \text{ }^{\circ}\text{C}$. The mesh model is shown in Figure 2b.

2.3. Boundary Conditions

The ambient temperature is prescribed as 20 degrees Celsius and the feed rate of the Z-, Y-, and X -axes is prescribed as 10 m/min. Figure 3 presents the calculation process. Firstly, the heat generation model (including the slider and guide rail, motor, and bearing pair) and the heat dissipation model are calculated according to thermodynamic and static empirical formulas. The calculated results cannot be directly applied as loads to the boundary conditions of the simulation software. The heat generation must be converted into heat flux density loads and heat generation rate loads can be used in the finite element simulation. The boundary condition calculation results are presented in Tables 1–3.

Table 1. Loading of the convective heat transfer coefficient.

	Remaining Positions	Worktable	Spindle Box
Convective heat transfer coefficient [$\text{W}/(\text{m}^2 \cdot \text{K})$]	9.7	16	22.3

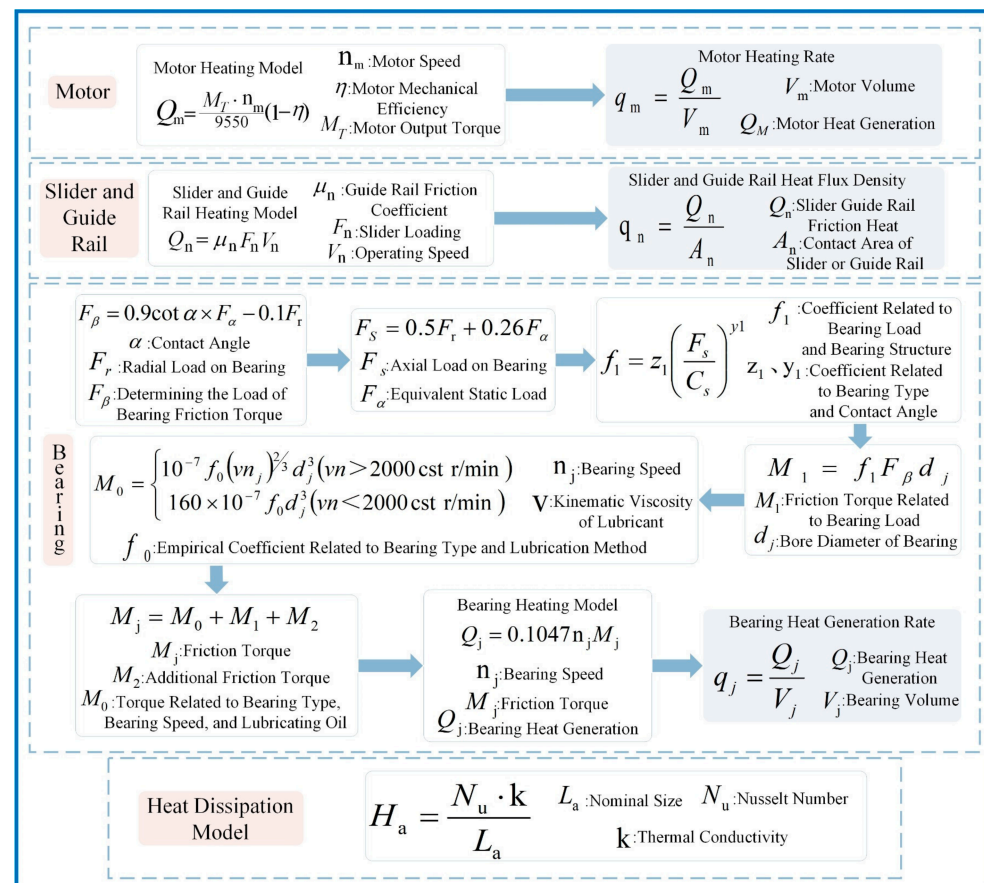


Figure 3. Boundary condition calculation process and results.

Table 2. The application of thermal loads (heat generation rate).

Coordinate Axis	X-Axis		Y-Axis		Z-Axis	
Part	Motor	Bearing	Motor	Bearing	Motor	Bearing
Heat generation rate [W/m ³]	2600	141,300	2530	162,600	141,300	2310

Table 3. The application of thermal loads (heat flux density).

Coordinate Axis	X-Axis		Y-Axis		Z-Axis	
Part	Guide rail	Slider	Guide rail	Slider	Guide rail	Slider
Heat Flux Density [W/m ²]	350.8	331	355	283	271.6	308

Sources of the thermal load parameters are as follows:

The motor heat generation rate is calculated from the rated power and mechanical efficiency of the motor. The bearing heat generation is calculated based on the Palmgren bearing friction torque formula, in combination with the bearing type, actual load, and rotational speed. The heat flux density of the guide rail–slider interface is calculated from the frictional heat formula, with the friction coefficient of the rolling guide pair taken as 0.005 by referring to classical empirical values. The convective heat transfer coefficient is calculated using empirical formulas for natural convection over vertical and horizontal flat plates, with reference to classical heat transfer theory and the methods reported in the literature.

2.4. Finite Element Analysis Results

In Figure 4, the deformation and temperature fields at steady state are attained following the boundary conditions from Section 2.3.

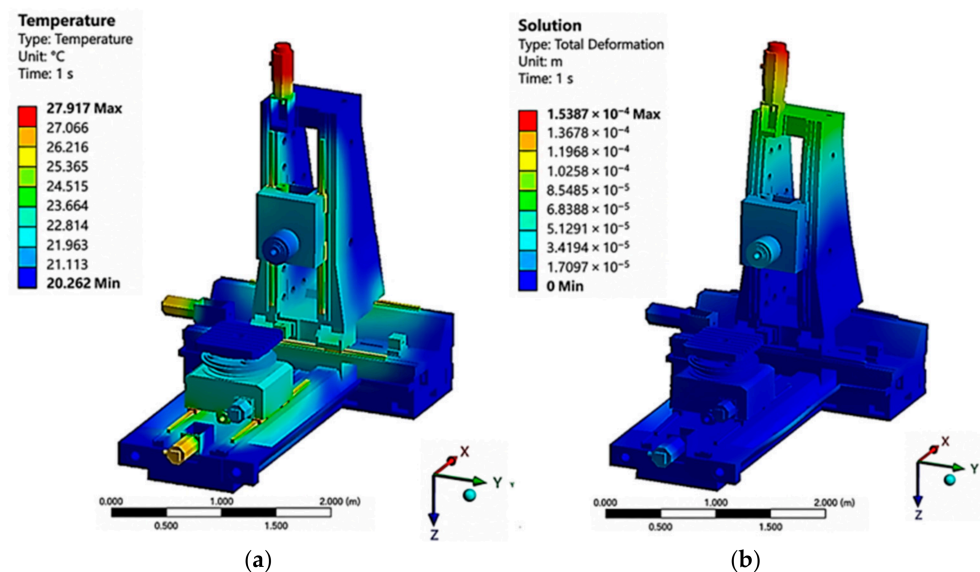


Figure 4. Deformation field with internal heat sources and their temperature field: (a) temperature field; (b) deformation field.

The temperature field varies from 20.2 °C to 27.9 °C in Figure 4 and the highest temperature is observed at the Z-axis motor. From the deformation field, it is observed that the column demonstrates a “backward” trend and the center of the bed exhibits a “convex” trend.

3. Analysis of the Layout of Temperature Control Devices for Machine Tools

3.1. Layout Analysis of Temperature Control Devices on Bed

The temperature control unit is primarily composed of a base plate, a cover plate, and a coolant. The base plate is fabricated from aluminum, the cover plate is fabricated from copper, and No. 5 spindle oil is employed as the coolant. The relevant material properties are presented as follows: copper—density 8978 kg/m³, specific heat capacity 381 J/(kg·K), thermal conductivity 387.6 W/(m·K); aluminum—density 2719 kg/m³, specific heat capacity 871 J/(kg·K), thermal conductivity 202.4 W/(m·K); No. 5 spindle oil—density 878 kg/m³, specific heat capacity 2090 J/(kg·K), thermal conductivity 0.14 W/(m·K), and dynamic viscosity 4.302×10^{-3} Pa·s. The temperature control device is integrated with a differentiated multi-loop active temperature control system. As shown in Figure 5, three layout schemes for the temperature control device are proposed based on the simulation results obtained from the previous section.

In the room temperature synchronization strategy, the temperature control devices are maintained at 20 °C, identical to the constant-temperature chamber. In the differential temperature control strategy, the plates are set at 17 °C. The ambient temperature is set to 20 °C, and two strategies are proposed: the room temperature synchronization strategy and the differential temperature control strategy. In the room temperature synchronization strategy, the temperature control devices are set to 20 °C in the simulation, while in the differential temperature control strategy, the temperature control devices are set to 17 °C. As described in Section 2.3, the remaining internal heat source loads of the bed are applied to obtain the simulation results. In Figure 6a,b, the thermal deformation along

the Z-axis of the left guide rail on the X-axis is extracted from the simulation results for comparative analysis.

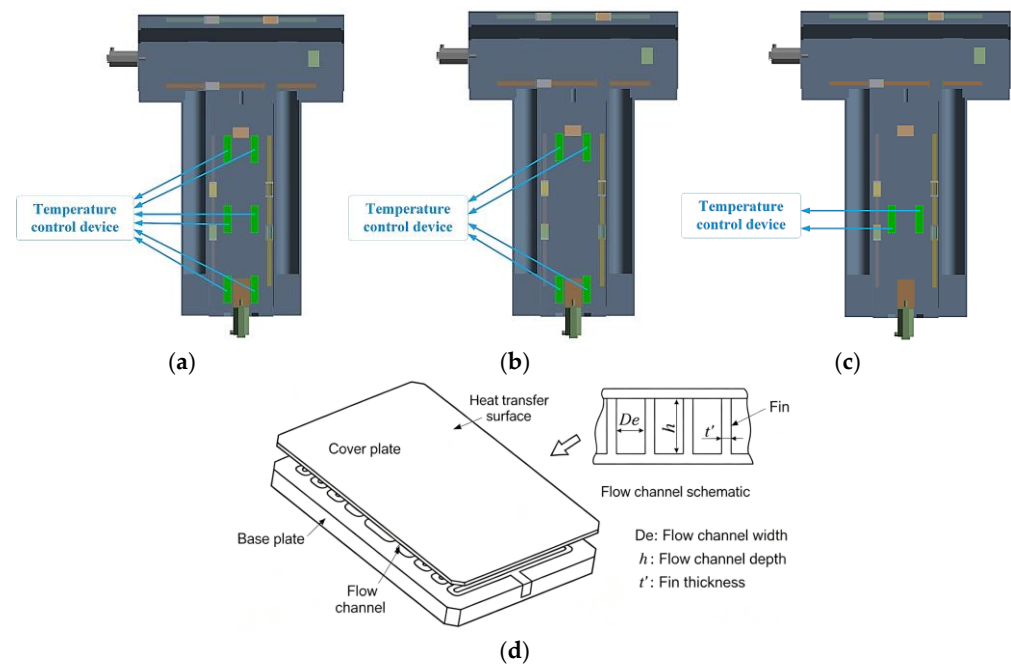


Figure 5. Layout schemes of temperature control devices for the bed: (a) layout scheme 1 for the temperature control device; (b) layout scheme 2 for the temperature control device; (c) layout scheme 3 for the temperature control device; and (d) geometry of the temperature control device.

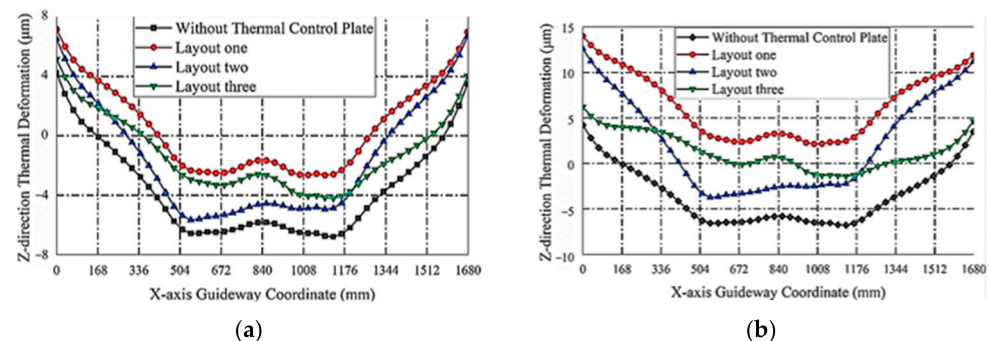
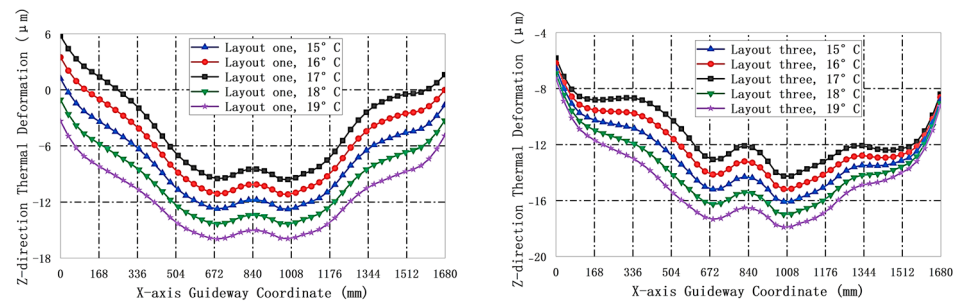


Figure 6. Two temperature control strategies: (a) room temperature synchronization strategy; (b) differentiated temperature control strategy.

From Figure 6a,b, layout scheme 3 is observed to exhibit the best X-axis Z-direction straightness variation between the two thermal control strategies. In contrast, layout scheme 1 shows the largest thermal deformation variation under both thermal control strategies when comparing the results with and without the thermal control plate. Since the machine tool structure and its heat sources are symmetrically arranged, resulting in consistent deformation, analyzing a single guide rail is sufficient for this study.

Further research is conducted to investigate the differentiated thermal control strategies in layout scheme 1 and layout scheme 3, where the temperature control device is sequentially set to 15, 16, 17, 18, and 19 degrees Celsius. In Figure 7a,b, for comparative analysis, the Z-direction thermal deformation of the left guide rail of the X-axis is extracted from the simulation results.



(a) Different temperatures in layout scheme 1 (b) Different temperatures in layout scheme 3

Figure 7. Two layout schemes of differentiated temperature control strategy.

According to Figure 7a,b, under layout scheme 1 at diverse temperatures, the results show no significant variation in the Z-direction straightness at each temperature. The fluctuation of thermal deformation remains around 15 μm. For layout scheme 3, it is observed that the straightness reaches its minimum value when the temperature control device is set to 17 °C, with a deformation range of about 10 μm.

Therefore, layout scheme 3 is adopted to arrange the temperature control device on the bed with the application of differentiated temperature control strategies; within the tested temperature range, 17 °C was identified as the best-performing temperature for the temperature control device.

3.2. Analysis of the Layout of Temperature Control Devices on Column

Similarly, the layout scheme of the temperature control device on the column is studied using the FEA method. Three layout schemes are proposed in Figure 8.

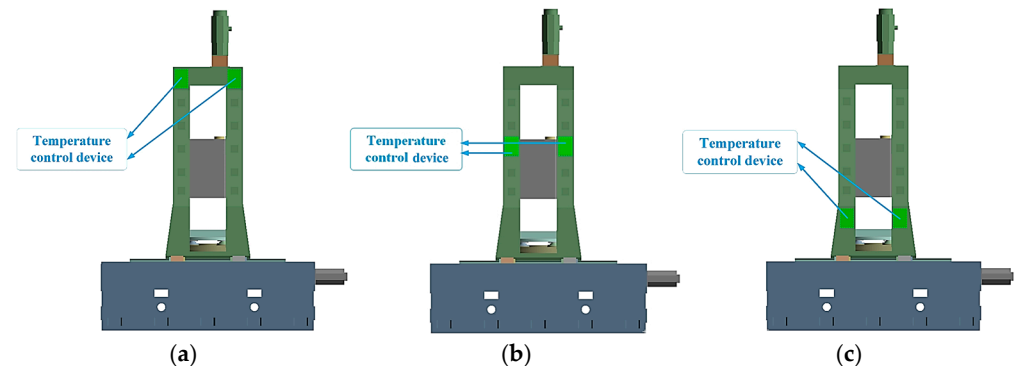


Figure 8. Temperature control device layout scheme for the column: (a) layout scheme 1 for the temperature control device; (b) layout scheme 2 for the temperature control device; (c) layout scheme 3 for the temperature control device.

The thermal deformation is analyzed using two strategies: the room temperature synchronization strategy and the temperature difference control strategy. The temperature control devices in the simulation are set at 20 °C and 23 °C, respectively, with the ambient temperature set at 20 °C. The motor located on the upper part of the column acts as the main heat source, causing the column to tilt backward. Accordingly, the control temperature was set to counteract this deformation. The heat source load in Tables 1–3 is applied to the column as a boundary condition, as shown in Figure 9a,b, and the X-direction thermal deformation of the Z-axis guide rail is obtained.

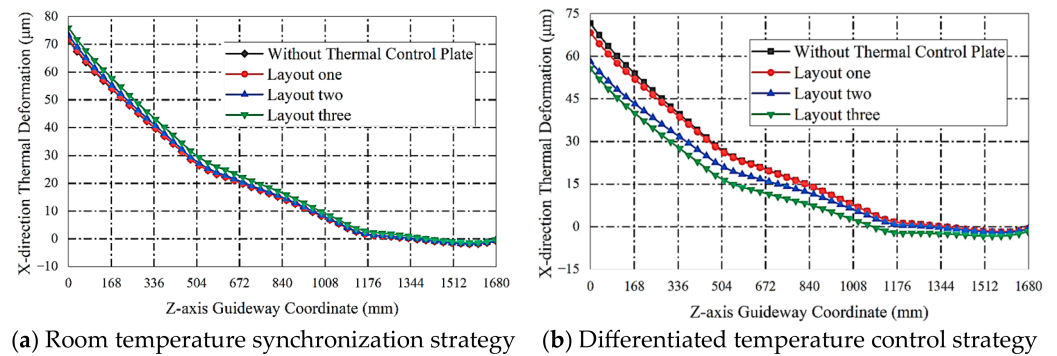


Figure 9. Two temperature control strategies.

It is evident from Figure 9a,b that the most effective straightness correction is achieved for the Z-axis X-direction thermal deformation curve in layout scheme 3. The inclination of the column is mitigated and reduced by the temperature control devices heated at the upper end of the column. Conversely, no significant improvement is observed in the thermal deformation curve straightness in layout scheme 1, in which the deviation remains approximately 70 μm. Further research is conducted on the differentiated thermal control strategy of layout scheme 3, where the temperature control device is sequentially set to 22, 23, 24, 25, and 26 degrees Celsius. Because the precise optimal temperature was initially uncertain, a range of temperatures was tested experimentally to determine the optimum. As shown in Figure 10, thermal deformation along the X-direction of the Z-axis is extracted from the right guide rail to be compared with simulation results. The temperature control device is arranged on the column according to layout scheme 3 and set to 25 °C. A minimum value of about 40 μm is reached for the straightness of thermal deformation. The thermal deformation fluctuations at other temperatures are greater than those at 25 °C.

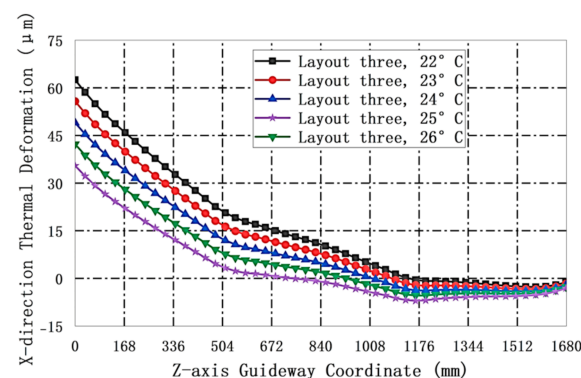


Figure 10. Thermal deformation at different temperatures in layout scheme 3.

Therefore, the optimal placement for the temperature control device is determined to be near the base of the column as shown in layout scheme 3, and a differentiated temperature control strategy should be adopted with the temperature set at 25 °C.

4. Active Temperature Control Method for Machine Tools to Suppress Thermal Errors

4.1. Thermal Error Analysis of Machine Tool Under Coupling of Multiple Temperature Control Devices

Active temperature control refers to the strategic placement of heat dissipation modules near the moving components and heat-generating structural parts of a machine tool, enabling rapid heat removal or heat input to actively regulate thermal deformation. As described in Section 3, the temperature control device is installed in the middle of the bed

(numbered 1 and 2) and at the lower end of the back of the column (numbered 3 and 4), as shown in Figure 11.

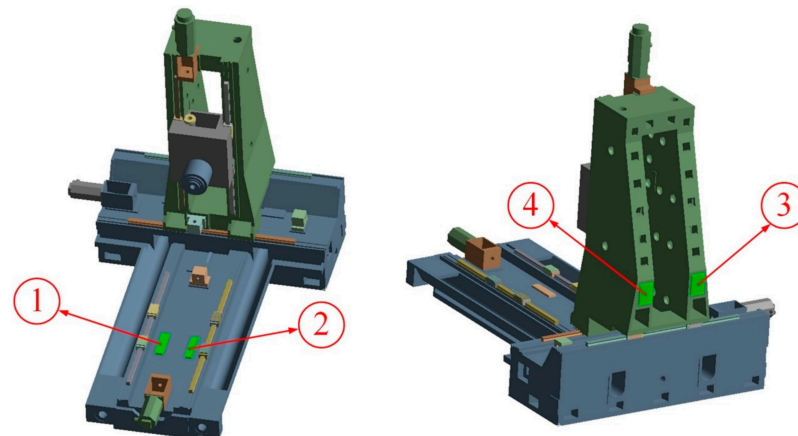


Figure 11. Layout of temperature control device.

The thermal deformation under various active temperature control conditions in Table 4 is analyzed based on the layout of temperature control devices in Figure 11. The temperatures of devices numbered 1 and 2 are set to 17 °C, and the temperatures of devices numbered 3 and 4 are set to 25 °C. According to Tables 1–3, the boundary conditions for other heat sources are set. The thermal deformation of the guide rails is mainly affected by the temperature control devices on the bed and column, respectively. Through combined control, thermal errors in both directions can be suppressed simultaneously, and collaborative optimization of multi-directional thermal errors can be achieved. For the roll error of the guide rail system, the reduction achieved by combined control is slightly greater than the linear superposition of the reductions obtained by individual structural control, which is mainly attributed to the more uniform temperature distribution of the whole machine structure under the combined action.

Table 4. The different loading conditions of the temperature control devices.

Condition Number.	Number of Temperature Control Devices Regarded as Heat Sources in FEA
Condition one	1
Condition two	2
Condition three	1, 2
Condition four	3
Condition five	4
Condition six	3, 4
Condition seven	1, 2, 3, 4

4.2. FEA Results

The machine tool was simulated according to the seven conditions in Table 4, and the temperature field and deformation field were obtained, as shown in Figures 12 and 13.

In Figure 12, a range of 20 to 27.9 °C is exhibited by the temperature field at steady state. Higher temperatures are exhibited by three motors on the Y-, Z-, and X-axes than by other structural components. The highest temperature of 27.9 °C is observed for the Z-axis motor. A significant change is observed in the temperature field near the external temperature control devices, while the temperature of the parts farther away from the temperature control devices is unchanged.

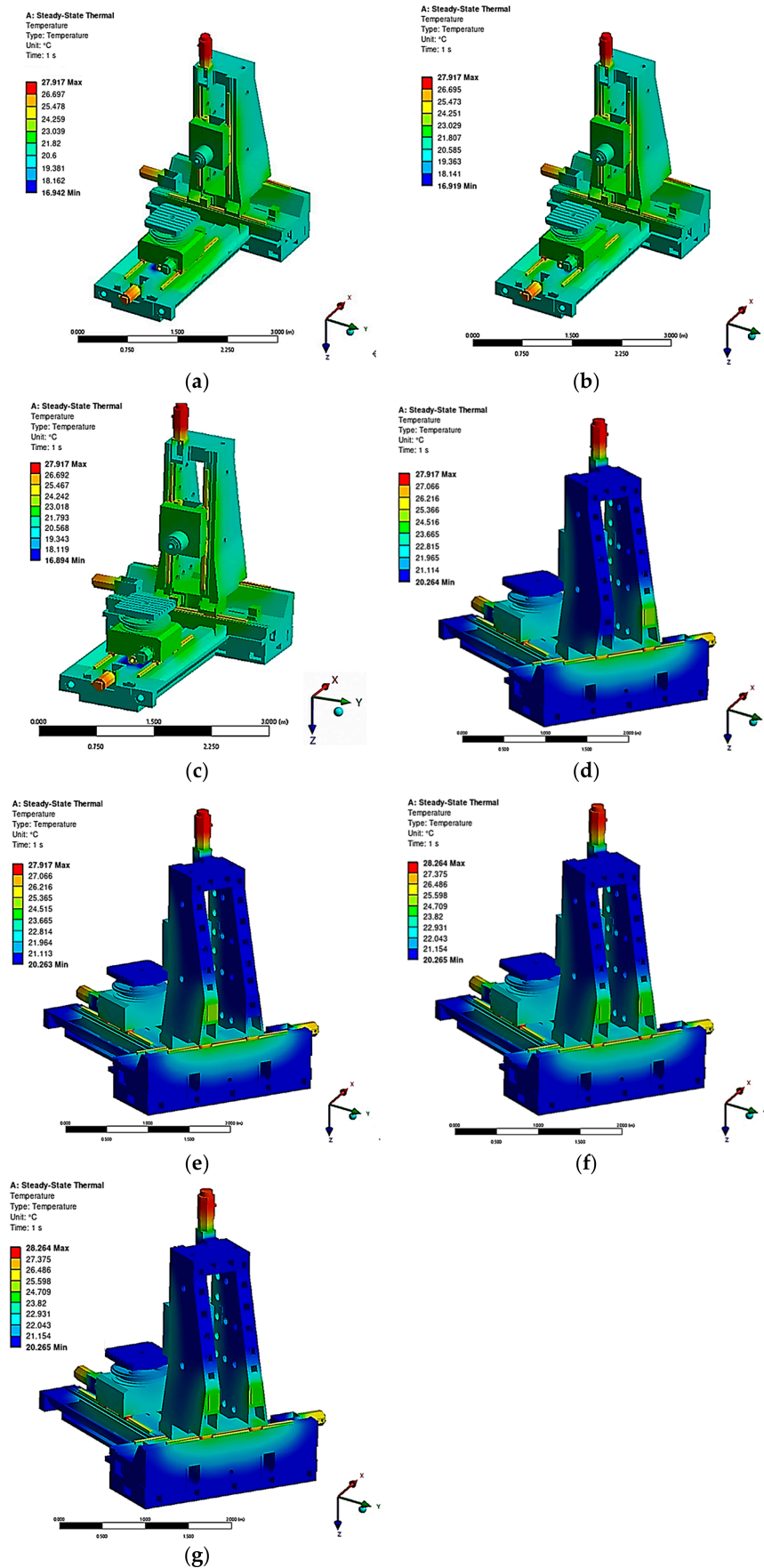


Figure 12. Temperature field of the machine tool under seven conditions: (a) Condition one; (b) Condition two; (c) Condition three; (d) Condition four; (e) Condition five; (f) Condition six; (g) Condition seven.

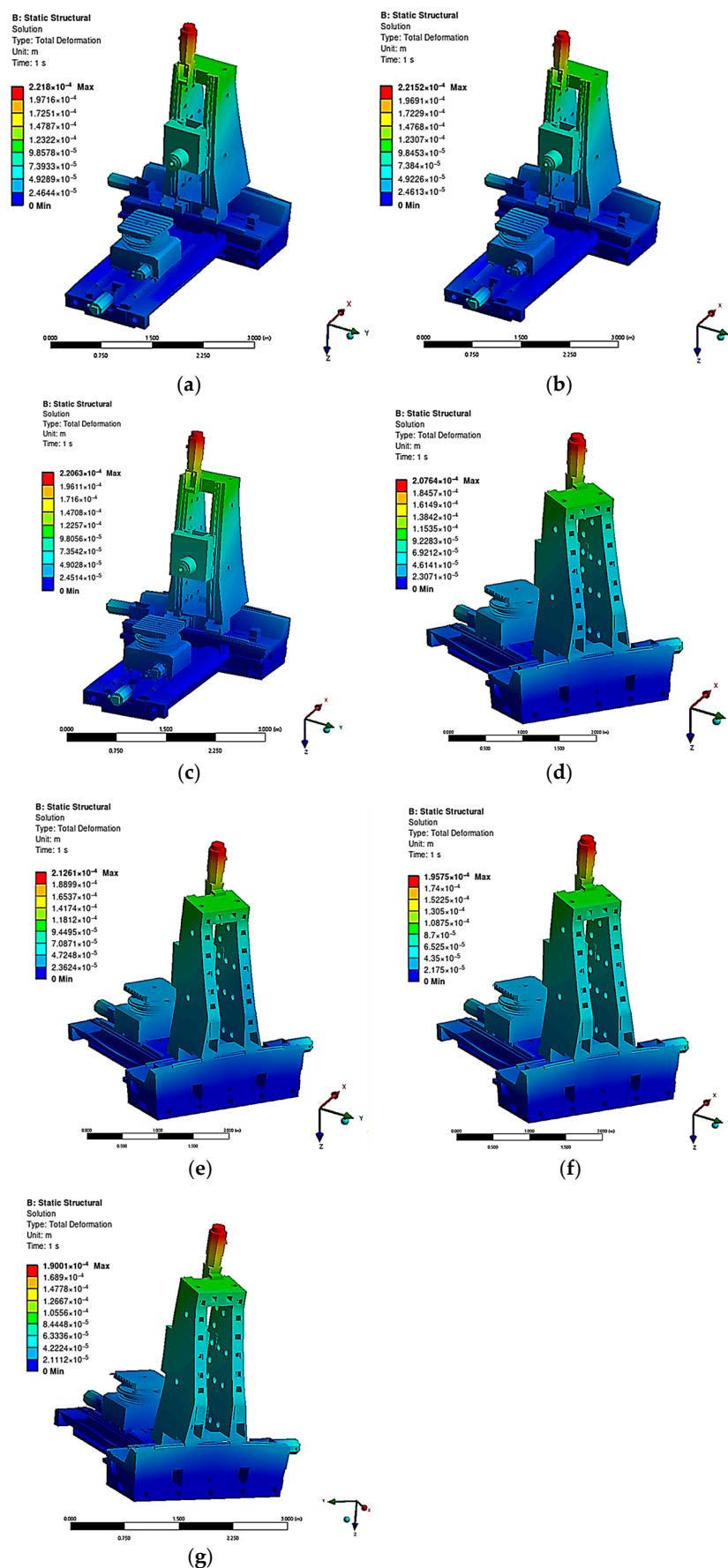


Figure 13. Deformation field of the machine tool under seven conditions: (a) Condition one; (b) Condition two; (c) Condition three; (d) Condition four; (e) Condition five; (f) Condition six; (g) Condition seven.

To visually observe thermal deformation, the thermal deformation of guide rails is regarded as the primary subject of investigation. The extracted coordinate directions and paths are shown in Figure 14.

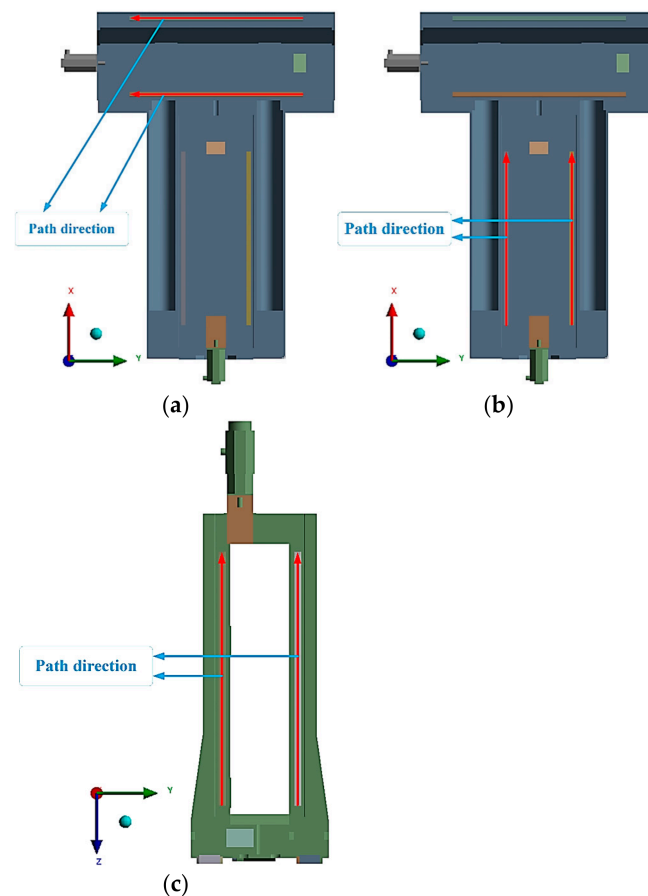


Figure 14. Extraction path of guide rail: (a) Extraction path of Y-axis guide rail; (b) extraction path of X-axis guide rail; (c) extraction path of Z-axis guide rail.

4.2.1. Thermal Deformation Along the Z-Direction of the X-Axis Left Guide Rail

During the study of machine tool thermal deformation, finite element simulations are conducted based on the seven conditions shown in Table 4. The thermal deformation of the guide rail is presented in Figures 15 and 16.

In Figure 15, the maximum value of the thermal deformation under the action of only the internal heat source is determined to be $22\text{ }\mu\text{m}$, and it shows an “upward convex” shape. In Conditions one, two, and three, thermal deformation is reduced to varying degrees by external heat sources. The thermal deformation in Conditions four, five, and six is similar to that without temperature control devices. The thermal deformation is reduced by $4\text{ }\mu\text{m}$ in the simulation results of Condition one, $2\text{ }\mu\text{m}$ in Condition two, and $5\text{ }\mu\text{m}$ in both Conditions three and seven. Therefore, thermal deformation is significantly superimposed in the bed structure due to the influence of a single heat source and multiple temperature control devices.

A comprehensive analysis of the bed deformation data under Condition 7 (Figure 15) indicates that this condition yields the best straightness.

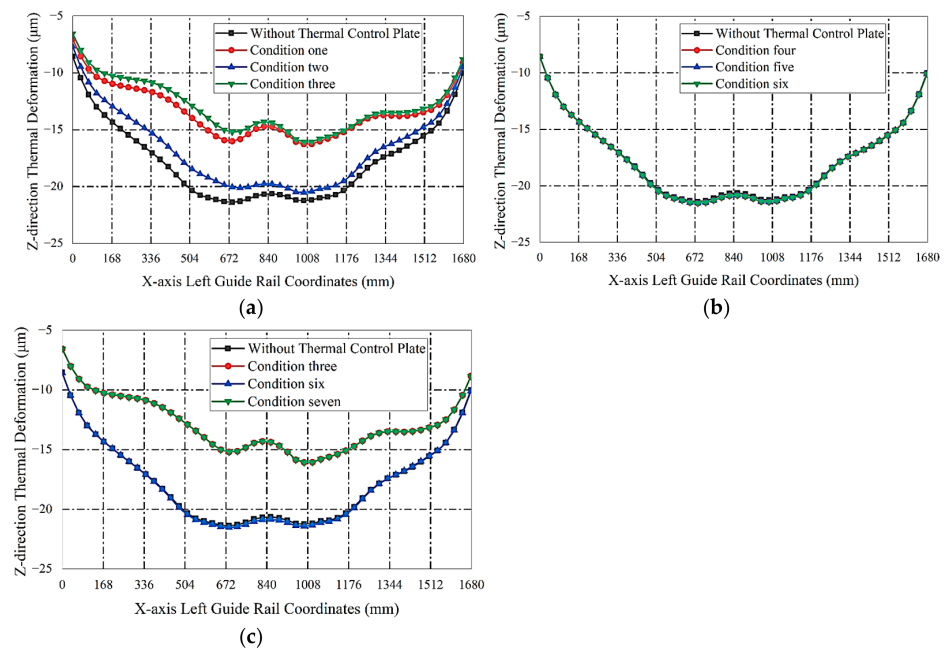


Figure 15. Comparative thermal deformation (X-axis): (a) thermal deformation under active temperature control of bed; (b) thermal deformation under active temperature control of columns; (c) thermal deformation under active temperature control of bed and columns.

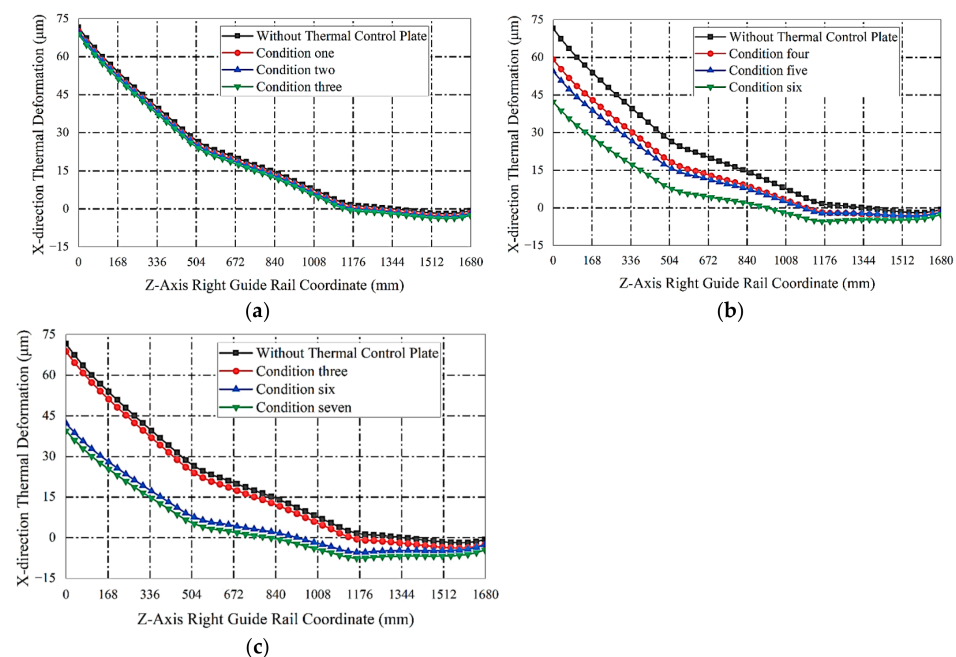


Figure 16. Comparative thermal deformation (Z-axis): (a) thermal deformation under active temperature control of bed; (b) thermal deformation under active temperature control of columns; (c) thermal deformation under active temperature control of bed and columns.

4.2.2. Thermal Deformation Along the X Direction of the Z-Axis Left Guide Rail

Similarly, in Figure 16, the maximum value of the thermal deformation under the action of only the internal heat source is determined to be 70 μm, and it shows a “backward” shape. The thermal deformation in Conditions one, two, and three is similar whether with or without a temperature control device. Thermal deformation is reduced to varying degrees in Conditions four, five, and six. The thermal deformation is reduced by 12 μm in the simulation results of Condition four, 17 μm in Condition five, 30 μm in Condition six,

and 30 μm in Condition seven. Similar to the bed, the thermal deformation of the column structure is also significantly superimposed as a result of the impact of a single heat source and multiple temperature control devices.

A comprehensive analysis of the column deformation data under Condition 7 (Figure 15) similarly shows that Condition 7 yields the best straightness within the tested temperature range.

4.3. Analysis of Six Thermal Errors in the X-Axis

Six motion errors are present in the linear axis of the machine tool: one positioning, two straightness, and three angle errors. The bed is used as an example for calculation.

(1) Positioning Error

The positioning error is calculated according to Equation (1):

$$\delta_x(x) = \frac{d_{x_left}(x) + d_{x_right}(x)}{2} \quad (1)$$

where $\delta_x(x)$ is the X-direction positioning error of the X-axis guide rail at the coordinate position x . $d_{x_left}(x)$ and $d_{x_right}(x)$ are the thermal deformations in the X-direction at coordinate position x for the left and right guide rails of the X-axis. x denotes the guide coordinate position.

(2) Straightness Errors

The straightness error is calculated according to Equations (2) and (3):

$$d_y(x) = \frac{d_{y_left}(x) + d_{y_right}(x)}{2} \quad (2)$$

$$d_z(x) = \frac{d_{z_left}(x) + d_{z_right}(x)}{2} \quad (3)$$

where $d_z(x)$ and $d_y(x)$ are the thermal deformations in the Z- and Y-directions belonging to the X-axis guide rail at coordinate position x . $d_{y_left}(x)$, $d_{y_right}(x)$, $d_{z_left}(x)$, and $d_{z_right}(x)$ represent the thermal deformations of the left and right guides in the Y- and Z-directions at coordinate position x of the X-axis.

As an example, the straightness in the Y-direction of the X-axis is considered, using the method of connecting the start and end points. The X-direction guideline for the Z-axis is defined according to the following equation:

$$\delta_{y_path}(x) = K_{xy}x + d_y(x_0) \quad (4)$$

where $\delta_{y_path}(x)$ is the Y-direction thermal deformation at coordinate position x on the straight trajectory of the X-axis guide. $d_y(x_0)$ is the Y-direction thermal deformation of the guide at the X-axis origin. $K_{xy}x$ denotes the slope of the guideline.

Thermal deformation at the midpoint of the two guides in the Y-direction and the trajectory line separation are computed. The error in the Y-direction is determined by incorporating the deviation between the trajectory and guidelines at the origin:

$$\delta_y(x) = \frac{|K_{xy}x + d_y(x_0) - d_y(x)|}{\sqrt{1 + K_{xy}^2}} + d_y(x_0) \quad (5)$$

where $\delta_y(x)$ is the Y-direction straightness at position x of the guide.

(3) Angular Errors

The pitch error is determined using Equation (6):

$$\varepsilon_y(x) = \frac{d_{z_front}(x) - d_{z_back}(x)}{L_1} \quad (6)$$

where $\varepsilon_y(x)$ is the pitch error of the X-axis of the bed. $d_{z_front}(x)$ and $d_{z_back}(x)$ represent the average Z-direction thermal deformations of the front and back sliders of the X-axis at specific coordinate positions, respectively. L_1 is the X-direction spacing of the X-axis guide sliders.

The yaw error is calculated according to Equation (7):

$$\varepsilon_z(x) = \frac{d_{y_front}(x) - d_{y_back}(x)}{L_1} \quad (7)$$

where $\varepsilon_z(x)$ is the yaw error of the column.

The roll error is calculated according to Equation (8):

$$\varepsilon_x(x) = \frac{d_{z_left}(x) - d_{z_right}(x)}{L_2} \quad (8)$$

where $\varepsilon_x(x)$ indicates the roll error, and L_2 is the Y-direction span of the X-axis guide sliders.

The six thermal errors of the X-axis are calculated to obtain Figure 17.

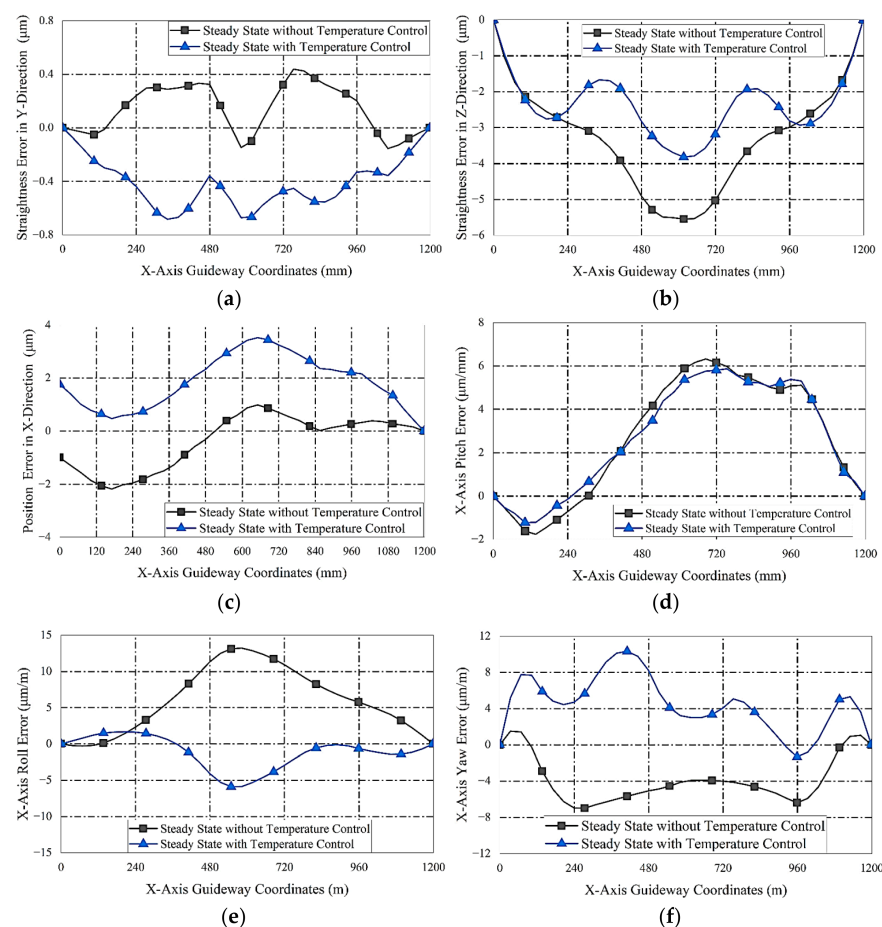


Figure 17. Six thermal errors: (a) Y-axis straightness error; (b) Z-axis straightness error; (c) X-axis positioning error; (d) pitch error; (e) roll error; (f) yaw error.

The six errors were calculated using the deformation data under Condition 7 (Table 4), and the results are presented in Figure 17. In Figure 17, the roll error, Z-direction straightness error, and X-direction positioning error are reduced by the active temperature control device. In Figure 17b, the maximum Z-direction straightness error without active temperature control is 5.5 μm , while that under the effect of active temperature control is 3.8 μm . In Figure 17c, the maximum X-direction positioning error without active temperature control is 3.8 μm , while that under the effect of active temperature control is 2 μm . In Figure 17e, the maximum roll error without active temperature control is 14 μm , while that under the effect of active temperature control is 6 μm . The above shows that thermal errors can be effectively suppressed by the active temperature control method.

5. Experiment

To verify the correctness of the analysis, experiments with active temperature control were carried out, and the obtained results were compared with those obtained without active temperature control. The experimental flowchart is shown in Figure 18.

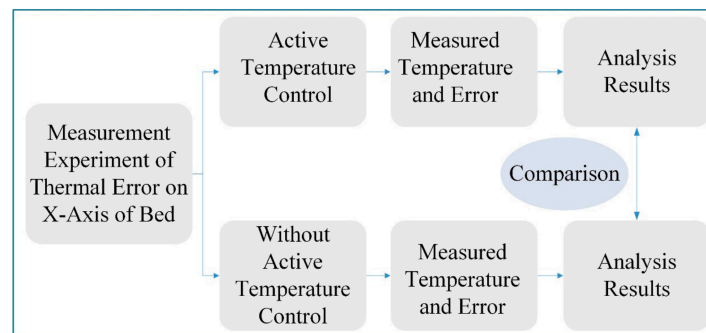


Figure 18. Experimental design for thermal error on the X-axis of the bed of a horizontal machining center.

5.1. Experimental Setup

The machine tool was housed in a constant-temperature laboratory maintained at 20 ± 0.5 °C. A multi-loop differentiated active temperature control system was established, with a temperature control accuracy of ± 0.5 °C. The temperature of the motor, motor seat, bearing seat, and environment was recorded using the PT100 thermistor sensor. An Omega PT100 temperature sensor (Omega Engineering, Norwalk, CT, USA), relatively large in size, was used and held in place by magnetic attachment. The thermal deformation was detected using eddy current sensors. Linear feed system motion errors were monitored by the Renishaw XM60 (Renishaw plc, Wotton-under-Edge, UK). Experimental equipment and measuring instruments are depicted in Figure 19. The differentiated multi-loop bath recirculation system is based on two recirculation coolers with identical supply pressures, several independent coolant blenders, and control units. As illustrated in Figure 19b, each recirculation branch is connected to two trunks via an input/output electric valve group; each group contains two electric valves that direct the recirculation coolant to or from the two trunks, respectively, as detailed in Figure 19c. All control units of the independent coolant blenders and bath recirculation coolers are linked to the host computer through a communication unit (USB to RS485) and execute instructions using PID control [25].

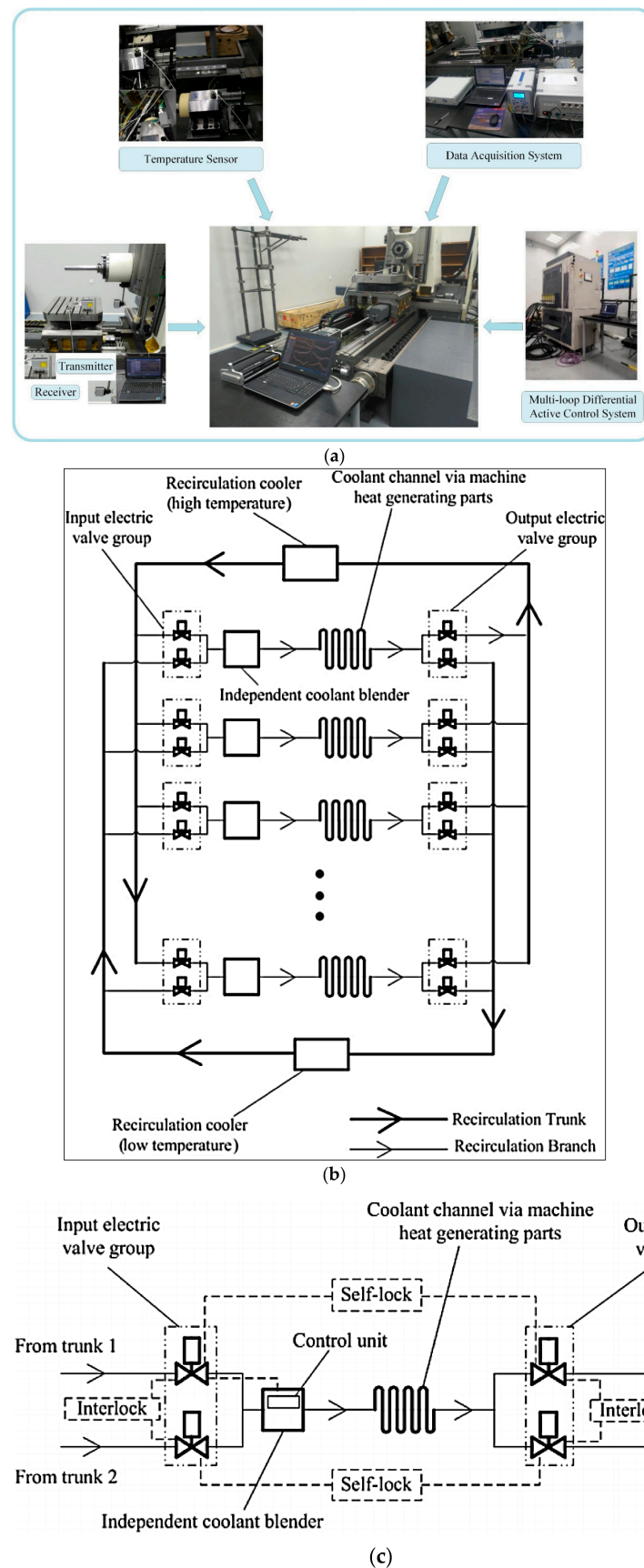


Figure 19. Thermal error measurement experimental platform of machine tool X-axis: (a) experimental equipment and measuring instruments; (b) structure and principle of differentiated multi-loop bath recirculation system; (c) accomplishment of differentiated supply temperature in branch.

5.2. Experimental Process

Machine tool thermal balance refers to the continuous heat exchange between the various internal heat sources (such as friction in guide rail pairs and motor losses) and the external environment during machine operation, eventually reaching a dynamically stable thermodynamic state. A feed rate of 10 m/min is used to conduct the experiment. Firstly, the motion error is detected using an XM60 laser interferometer (Renishaw plc, Wotton-under-Edge, UK) and is used as initial data. Then, the machine tool is kept running continuously without applying temperature control equipment, with the motion error recorded after thermal equilibrium is achieved. Finally, as depicted in Figure 20, the temperature control device is installed according to Condition seven in Table 4. The machine tool is connected to a multi-loop differentiated cooling cycle system. The temperature control device on the bed is filled with 17 °C cooling oil, and the temperature control device on the column is filled with 25 °C cooling oil. The temperature of the machine tool is continuously monitored, and its motion error is reassessed after reaching thermal equilibrium.

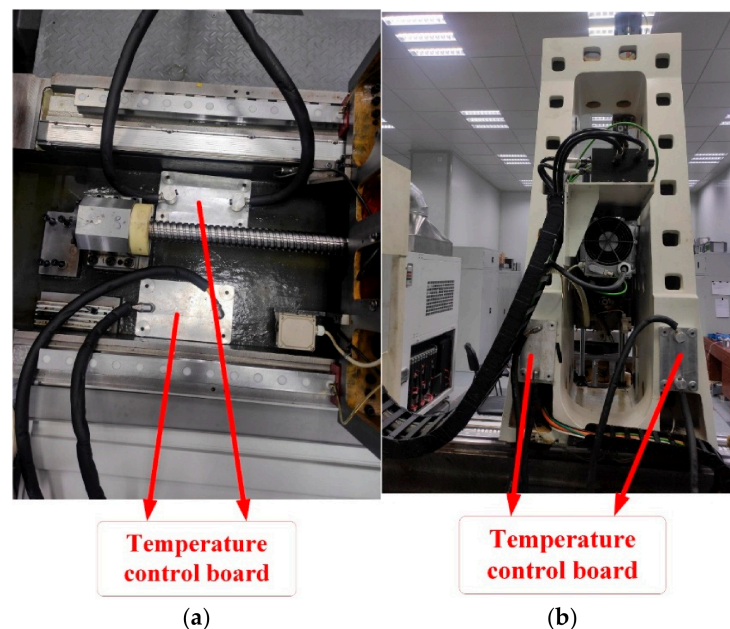


Figure 20. Installation position of temperature control devices: (a) layout of temperature control devices of bed; (b) layout of temperature control devices of column.

5.3. Experimental Results

The experiment is conducted within a thermostatic laboratory set at 20 ± 0.5 °C. As shown in Figure 21, the bearing seat, motor, and motor seat temperatures in the machine tool are measured with temperature sensors. Without active temperature control, thermal equilibrium is reached by the machine tool after three hours of operation. The highest temperature of 27.9 °C is observed at the Z-axis motor among the three motors. Three hours after the multi-loop differential active temperature control system is connected, the machine tool reaches thermal equilibrium once more. Temperatures at all measuring points are significantly lowered by the temperature control device. Then, the main heating components are continuously tested for one hour. A gradual decrease in temperature is observed at each measuring point. The subsequent slow temperature decrease represents a gradual convergence process after thermal equilibrium and does not affect the thermal error measurement. Moreover, the experiments demonstrate low energy consumption and good thermal control stability.

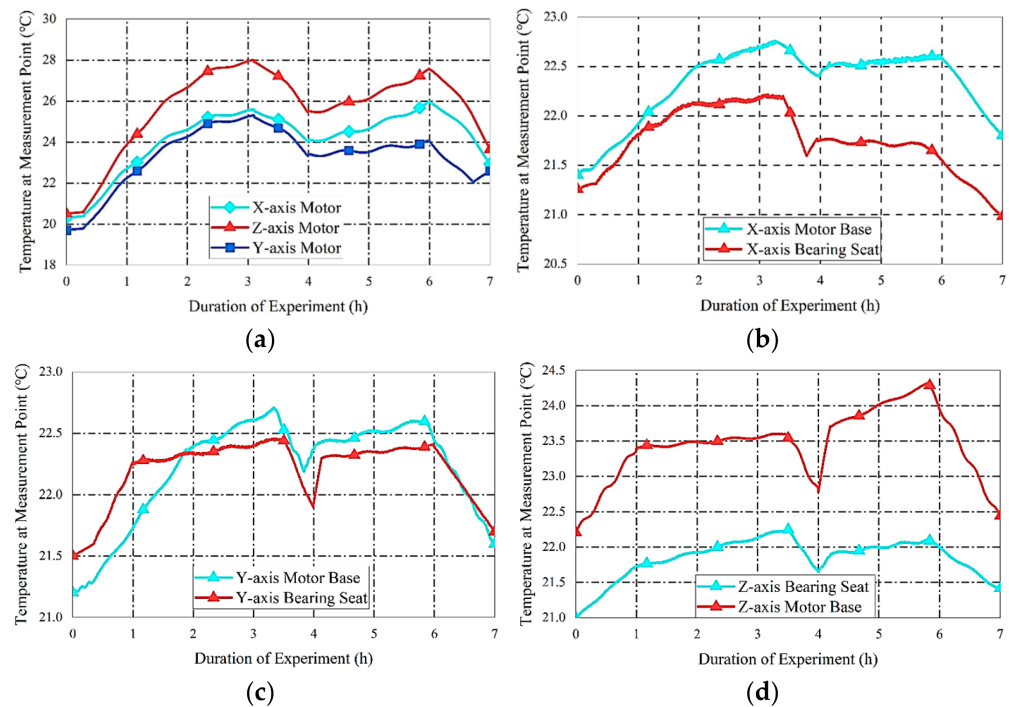


Figure 21. Machine tool temperatures at key points: (a) XYZ three-axis motor; (b) X-axis motor base and bearing base; (c) Y-axis motor base and bearing base; (d) Z-axis motor base and bearing base.

The six motion errors in the initial state, the no-active temperature control state, and the active temperature control state are analyzed.

In Figure 22, the roll error, X-direction positioning error, and Z-direction straightness error are reduced by the active temperature control device. In Figure 22b, the Z-direction straightness error in the initial state is $2.6 \mu\text{m}$ and the error with active temperature control is reduced by $2.4 \mu\text{m}$ compared to that without active temperature control. In Figure 22c, the X-direction positioning error in the initial state is $1.5 \mu\text{m}$ and the error with active temperature control is reduced by $2.5 \mu\text{m}$ compared to that without active temperature control. In Figure 22e, the roll error in the initial state is $2.6 \mu\text{m}$ and the error with active temperature control is reduced by $1.2 \mu\text{m}$ compared to that without active temperature control. The correctness of the active temperature control method is verified through comparison with the simulation results in Figure 17.

The simulations presented in this work were conducted under steady-state operating conditions with all three axes running at a constant speed of 10 m/min with no cutting load. In practice, however, machine tools usually operate under unsteady dynamic conditions involving variable speeds, multiple loads, and multi-axis coordination during the machining process. Future research could explore the influence of factors such as machining paths, feed rate variations, and cutting load fluctuations on the evolution of temperature fields and the development of thermal errors, in order to establish a more realistic dynamic thermal error model applicable to actual machining conditions. This study is also applicable to horizontal machining centers with symmetrically distributed heat sources.

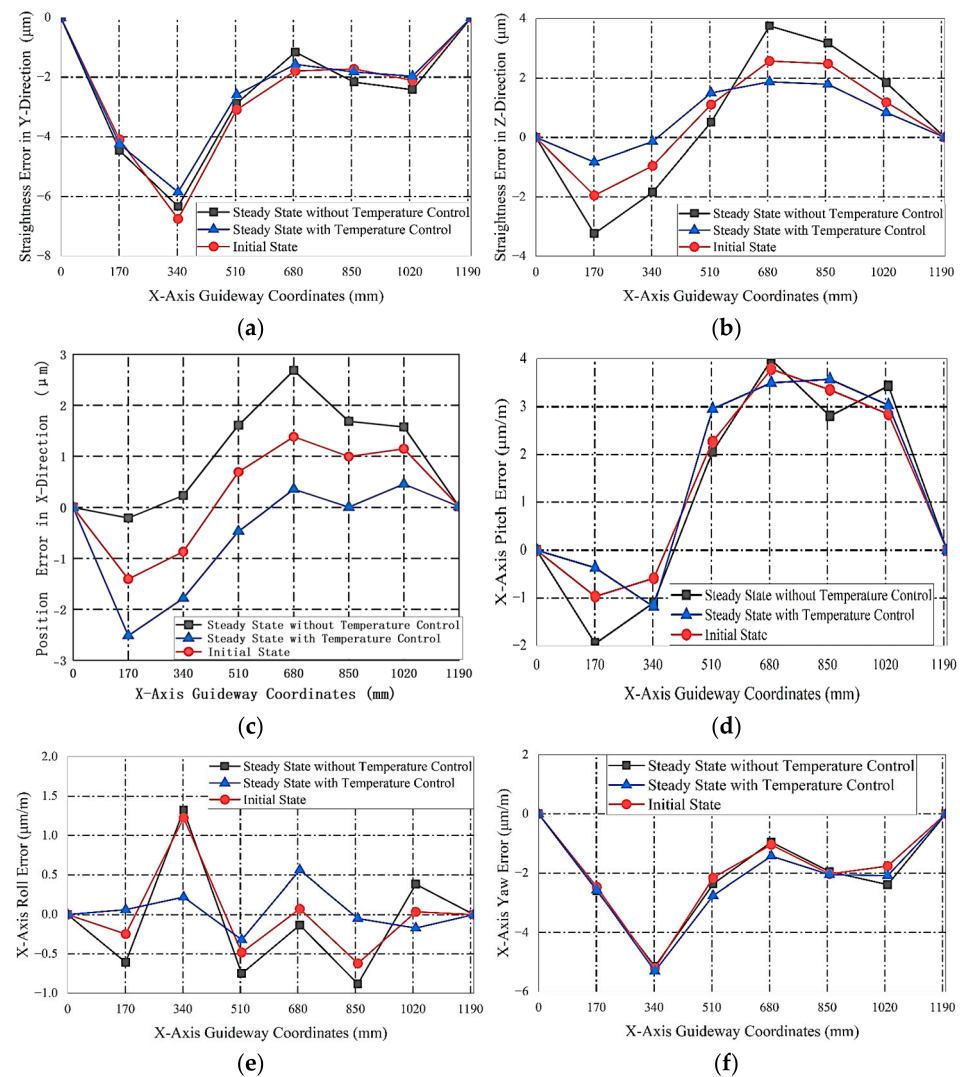


Figure 22. Comparison of six motion errors for the X-axis: (a) Y-axis straightness error; (b) Z-axis straightness error; (c) X-axis positioning error; (d) pitch error; (e) roll error; (f) yaw error.

6. Conclusions

This study proposes an active temperature control method utilizing a multi-loop differentiated system for machine tools. The following conclusions are drawn:

- (1) Providing a new idea for machine tool design, the proposed active temperature control method is an operative approach to reduce thermal deformation.
- (2) The placement of the temperature control device directly affects the effectiveness of suppressing thermal errors. The temperature control device of the bed should be placed in the middle of the bed, and the temperature control device on the column should be arranged at the lower part of the column.
- (3) A coupling effect exists among multiple temperature control devices, resulting in a superposition of thermal deformation. Simulation and experimental results demonstrate that applying different cooling oil temperatures to the temperature control devices on the bed and column can achieve the best performance within the tested temperature range.

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