Thermal Characteristics of a Vertical Hydrostatic Guideway System for Precision Milling Machine Applications

Hua-Chih Huang 1, * and Wen-Hao Yang 2

1 Department of Mechanical Engineering, National Kaohsiung University of Science and Technology, Sanmin District, Kaohsiung City 80778, Taiwan
2 Hiwin Technologies Corp., Taichung 40852, Taiwan
* Correspondence: hc.huang@nkust.edu.tw; Tel.: +886-7-381-4526

Abstract: This paper investigates the thermal characteristics of a vertical guideway system for precision milling machine applications by considering three heat sources, namely motor heat, viscous shearing heat of hydrostatic bearings, and friction heat from a ballscrew nut. A finite element (FE) model using ANSYS/Fluent was used to simulate the thermal characteristics of the system by considering the oil film friction of the hydrostatic bearings in the operational feed speed and heat generation in the ballscrew nut. Eight K-type thermocouples were installed in the vertical hydrostatic guideway system to measure the temperature rise in the key components. Nine thermal experiments of the vertical hydrostatic guideway system under three operational feed rates, namely 1.25, 2.5 and 5 m/min were conducted to measure the temperature of seven thermocouples in practical running conditions. The experimental temperature data then was used to adjust the FE model setting to guarantee the accurate prediction of the thermal deformation in real operational conditions. The FE model of the vertical hydrostatic guideway system built in this study can be used to predict the thermal deformation of worktable at center point at any running conditions. At a sliding feed rate of 1 m/s, the thermal positioning error of worktable at center point was 0.1539 μm in the X direction, 0.0009 μm in the Y direction and 2.0246 μm in Z direction.

Keywords: thermal deformation; hydrostatic bearings; vertical hydrostatic guideway; finite element analysis; ballscrew

1. Introduction

There are many factors that may affect the machining accuracy of machine tools, including geometric errors and thermal errors, cutting tool wear, workpiece thermal deformation and the error caused by the external environment [1]. According to the study from Bryan [2], 40–70% of machining errors are caused by thermal errors; hence, the thermal deformation of a machine tool is an important indicator of its machining accuracy. Chen et al. [3] proposed a thermal-error model for a hydrostatic spindle by using a finite-element software at various rotational speeds; it indicated that a spindle that rotates faster experiences greater thermal deformation. Yang et al. [4] used the ANSYS finite-element software to construct a finite-element model of the hydrostatic guideways of a three-axis milling machine. The study analyzed the thermal deformation of the guideway under different oil film thicknesses. Bo et al. [5] studied the thermal characteristics of a hydrostatic spindle used in grinding machines. The results indicated that all of the thermal deformation of the spindle was seemingly caused by temperature changes in the hydrostatic bearings. Heng et al. [6] used the computational fluid dynamics (CFD)-based numerical analysis software FLUENT to simulate the temperature distribution of an oil film and discussed the influence of rotational speed and lubricant viscosity on oil film heating. The results indicated that both of the aforementioned factors have significant effects on the temperature increase in the oil film. Moreover, the high-temperature zones were mainly concentrated...
at the outer edge of the oil sill; further, increases in rotational speed and viscosity caused the high-temperature zones to move closer to the outer edge of the oil sill. Ballscrews are important feed-drive components in precision machine tools. Since the ball rolls within a screw track, the contact between the ball and the track creates frictional heat, which, in turn, increases the temperature of the nut and screw, affecting the manufacturing accuracy of precision machine tools. Horejs [7] examined the thermal stability of a ballscrew system by using a thermocouple to measure temperature. The temperature data was then imported into a finite-element software for use as thermal boundary conditions to obtain the distribution of the heat generated by the nut. Yu [8] found that most researchers measure the surface temperature of nuts or screws using a thermocouple; therefore, gaps exist in the actual results when known temperature zones serve as the boundary conditions in finite-element thermal structures. This demonstrated that frictional heat accumulates in the screw track (contact zone of the screw) located between the nut and the ball. Based on the principles of the ball kinematics, Lin [9] utilized different parameters to calculate the frictional heat flux produced by a ball moving along a screw track under different screw rotational speeds and preloads. The results were then imported into ANSYS as boundary conditions to simulate thermal and structural deformations. Experiments were also performed to compare the experimental and simulation results, where the heat flux derived from the actual sliding speed was imported as the heat source boundary condition. The results indicated that the error between the experimental and simulation results could be maintained to within 5%. This study adapted Lin’s method to estimate the heat flux of ballscrew nut at certain feed rate in the FEM model. Zheng and Dan [10] studied a large machine tool equipped with a hydrostatic guideway and analyzed its dynamic characteristics and bearing stiffness by numerical solution and deduced that the bearing load capacity and stiffness were affected by the size of the rectangular oil recess design. They concluded that an appropriate increase in the lubricant viscosity will help to improve the dynamic performance of the hydrostatic guideway. The study of the dynamic characteristic analysis of a hydrostatic guideway conducted by Liu et al. [11] included modal analysis by ANSYS software, analysis of natural frequencies of each mode and experimental verification of simulation parameters, and a further discussion on the deformation of oil recess on the clamp plate, which is fixed by bolts and screws, when the structure is subjected to recess pressure field. It pointed out that the deformation of the table has a certain influence on the straightness of the hydrostatic guideway, and the deformation of the clamp plate decreases bearing load capacity by 19%. Wang et al. [12] proposed the interaction of static characteristics of the hydrostatic guideway between the fluid pressure field and the structure, in which the flow pressure field acted on the solid structure, and the deformation of the solid affected the characteristics of the flow pressure field. They mentioned that the final oil gap of the hydrostatic bearing was the stable oil film gap after the structural deformation. Yang et al. [13] used the Fluent software to analyze the oil film flow field and obtained the static pressure distribution diagram and dynamic pressure distribution diagram of the oil film of the hydrostatic bearings at the moving feed rate of the hydrostatic guideway. In the contact between the oil recess and oil film, under the action of viscous friction between the oil film and the guideway fixed rail, a high temperature will be generated locally in the forward direction, and temperature rise will reduce the viscosity of the oil and affect the stiffness of the bearings. The dynamic characteristics analysis of the hydrostatic guideway proposed by Dong et al. [14] included the analysis and simulation of the static characteristics of a single recess pad using the FLUENT software, optimization analysis of the dimensions of the orifice diameter, the recess depth and oil film thickness and a modal analysis using the ANSYS software. The simulation parameters were verified by experiments. Chen et al. [15] studied an ultra-precision hydrostatic guideway and they compared its performance to a traditional rolling-contact guideway. They explored the three major performance indexes of a hydrostatic guideway, namely the load capacity, accuracy and stiffness and their relationship to three design parameters such as oil recess pressure, oil film thickness and the effective area of the bearing pad. They pointed out that the proper design of the oil
recess was the primary consideration for the design of a hydrostatic guideway system. Gong et al. [16] investigated the multi-objective optimization and flow field simulation analysis of a hydrostatic guideway. They had conducted a comprehensive comparison between closed-type and open-type hydrostatic guideways under a constant pressure oil supply system. From a fluid–solid coupled analysis they found that the oil recess pressure, which influenced the elastic deformation of the clamp plate on the closed-type hydrostatic guideway, might affect the straightness of the guideway. Jiang et al. [17] proposed that the thermal deformation of a hydrostatic guideway might affect the machining accuracy of machine tools. They constructed a finite-element model of the hydrostatic guideway, and they simulated the thermal deformation analysis of the hydrostatic guideway under different working conditions. The results showed that under higher feed speed, the thermal deformation of the hydrostatic guideway was bigger, and this will have a great influence on the machining accuracy. Yang et al. [4] built a finite-element model of a hydrostatic guideway with an average grid quality of 0.69 using ANSYS software and it pointed out that the grid quality criterion should be higher than 0.5 as the basic condition of a finite-element model for hydrostatic bearings. Other recent literature studied the thermal effect of lubricants for hydrostatic bearings under various service conditions and recess shapes for spindles and lathes [18–22]. In the study, the thermal deformation and structural deformation of a hydrostatic guideway caused by different oil film thicknesses and ballscrew nuts were analyzed.

From the aforementioned literature, it is clear that, for a closed-type hydrostatic guideway system, the sliding heat generated between a small oil-film of hydrostatic bearings and the frictional heat flux in the ballscrew nut are the main heat sources of a vertical hydrostatic guideway system. This paper proposed a novel FE modeling method of a vertical hydrostatic guideway system for milling machine applications. It built a finite element model of a vertical hydrostatic guideway system by considering the heat generated from the hydrostatic bearings and ballscrew nut incorporated with thermal experiments to investigate the thermal characteristics of the system. In order to verify the setting of the temperature boundary conditions for the finite element model, thermal experiments at different feed speeds were conducted to validate the model as close as possible to the physical system. In the end, this FEM model of a vertical guideway system can be used to predict the positioning error of a worktable at the center point at any operational feed rate.

2. Modelling Methodology and Theoretical Background

2.1. FEM Model of Vertical Hydrostatic Guideway

A vertical hydrostatic guideway system consists of a drive motor, hydrostatic bearings, a ballscrew and a worktable [23]. Figure 1 shows the photograph of the test setup and the simplified FEM model of the main structural parts and oil-film area of the vertical hydrostatic guideway system of this study. In this setup, a close-form design of a hydrostatic guideway applied in a single rail is shown in Figure 2, and the red-colored parts are the bearing pads of hydrostatic bearings. The rectangular recess with a depth of 1 mm was chosen for the oil pocket design of the hydrostatic bearings. For each bearing pad, the load-carrying capacity is [24] \( W = P_r \cdot A_e \), where \( P_r \) is the recess pressure, \( A_e \) is effective area of the pad and \( A_e = (L - l_1) \cdot (B - b_1) \). The layout design of the hydrostatic bearings (normal/side pad) in the worktable is shown in Figure 3, and the dimensions of each pad are shown in Figure 4 and Table 1.
Figure 1. Vertical hydrostatic guideway system.

Figure 2. Closed-form design of a hydrostatic worktable.

Figure 3. Bearing pad design in worktable.

Figure 4. Schematic bearing pad of a rectangular hydrostatic bearing.
### Table 1. Dimensions of bearing pads.

| Dimension     | Normal Recess Pad | Side Recess Pad |
|---------------|-------------------|-----------------|
| L (mm)        | 170               | 170             |
| B (mm)        | 20                | 22              |
| \( l_1 \) (mm)| 5                 | 3.5             |
| \( b_1 \) (mm)| 5                 | 3.5             |
| \( A_e \) (mm²)| 2475             | 3080.25         |
| Oil film gap (µm) | 10                | 10              |
| Recess depth (mm) | 1                | 1               |

### 2.2. Flow Field Model of Hydrostatic Bearing

The hydrostatic bearings considered in this study were recess-type hydrostatic bearings with a capillary-type restrictor as shown in Figure 5. While using ANSYS Fluent to model the flow field of the hydrostatic thrust bearing, one important parameter in the analysis was the length of the capillary restrictor in the designed film gap. In this study, we constructed the hydrostatic bearing model in Fluent, including the capillary restrictor. The length of capillary restrictor was calculated by using theoretical formula [24] (shown in Equation (1)) and was then simulated by using Fluent. Equation (1) is the formula used to calculate the length of the capillary restrictor. \( P_s \) is the supply pressure of the oil and \( P_r \) is the recess pressure. \( l_c \) is the length of capillary in mm, \( d_c \) is the inner diameter of the capillary (2 mm) and \( \mu_t \) is the viscosity of the oil. \( Q_{h0} \) is the outlet flow rate of the bearing recess at the designed film gap.

\[
l_c = \frac{\pi d_c^4 \times (P_s - P_r)}{128 \mu_t Q_{h0}} \tag{1}
\]

Equation (2) calculates the load-carrying capacity \((W)\) of the hydrostatic bearing. \( A_e \) is the effective area of the recess. \( R_{c0} \) is the flow resistance of the capillary restrictor at the designed film gap. \( R_c \) is the flow resistance of capillary restrictor in the final film gap. \( R_{h0} \) is the flow resistance of the bearing sill at the designed film gap. \( \varepsilon \) is the film gap displacement rate (or eccentricity rate) \( \varepsilon = e/h_0 \). \( e \) is the reduced film gap during the loading conditions. \( h_0 \) is the designed film gap. Since the capillary restrictor was a fixed restrictor, the flow resistance of the capillary restrictor was constant regardless of the film gap. Therefore \( R_c / R_{c0} \) is one in any case. Figure 5 is the schematic diagram of the capillary restrictor.

\[
W = A_e \times P_r = \frac{P_s A_e}{1 + \frac{R_{c0}}{R_{h0}} \left(1 - \varepsilon\right)^3 \times \frac{R_c}{R_{c0}}} \tag{2}
\]

### Figure 5. Schematic diagram of capillary restrictor.

### 2.3. Continuity Equation of a Hydrostatic Bearing

Figure 6 depicts the outlet flow rate at four sills of the rectangular hydrostatic bearing pad. The equation used to describe the oil flow of the hydrostatic bearing pad was the continuity equation. That is to say, the inlet flow rate of the capillary must be equal to the sum of the outlet flow rates through the four bearing pad sills.
For the continuity equation of flow rate [12]: \( Q_{\text{inlet}} = Q_{\text{outlet}} \)

\[
Q = Q_{\text{inlet}} = Q_{\text{outlet}} = \frac{\pi d^4}{128 \mu l_c} (P_s - P_r)
\]  \( (3) \)

where \( Q_{\text{inlet}} \) is the inlet flowrate of the capillary restrictor and \( Q_{\text{outlet}} \) is the outlet flow rate of the four sills of the bearing pad. To calculate the flow rate \( q_1 \) to \( q_4 \) for each sill [12]:

\[
q_1 = \frac{h_0^3 \times p_{r0} \times (L - (\frac{t}{2}) - (\frac{b}{2}))}{12 \times \mu_l \times b_1}
\]  \( (4) \)

\[
q_2 = \frac{h_0^3 \times p_{r0} \times (B - (\frac{t}{2}) - (\frac{b}{2}))}{12 \times \mu_l \times l_1}
\]  \( (5) \)

\[
q_3 = \frac{h_0^3 \times p_{r0} \times (L - (\frac{t}{2}) - (\frac{l}{2}))}{12 \times \mu_l \times b_1}
\]  \( (6) \)

\[
q_4 = \frac{h_0^3 \times p_{r0} \times (B - (\frac{t}{2}) - (\frac{l}{2}))}{12 \times \mu_l \times l_1}
\]  \( (7) \)

Total outlet flow rate \( Q_{\text{ho}} \):

\[
Q_{\text{ho}} = q_1 + q_2 + q_3 + q_4
\]  \( (8) \)

From the flow continuity equation, the recess pressure and the load-carrying capacity of the bearing pad can be calculated. The flow rate of all the hydrostatic bearings in this research FEM model were checked for their correctness by using the continuity equation of oil flow. The boundary conditions of the oil film at a hydrostatic bearing modeled by ANSYS Fluent software is shown in Figure 7. For the FEM model, the mesh size of the worktable structure was 5 mm, that of the vertical guideway structure was 20 mm, that of the recess oil film was 0.4 mm, that of the oil film at the orifice hole was 0.1 mm, and that of the oil film at the sills was 0.0025 mm.

**Figure 6.** Outlet flow rate at four sills of the rectangular hydrostatic bearing pad.

**Figure 7.** Boundary conditions of oil films at a hydrostatic bearing.

![Boundary conditions of oil films at a hydrostatic bearing](image-url)
2.4. Thermal Temperature Rise of Oil

The temperate rise (ΔT) of the oil lubricant of a hydrostatic bearing is due to two contributions. Firstly, the friction-induced temperature rise is the temperature rise caused by the friction power (P_f) of the oil film in a small gap, which relates to the feed speed, viscosity grade of the oil, oil film clearance and effective area of the bearing pad. Secondly, the pumping power-induced temperature rise is the temperature rise caused by pumping oil lubricant out of the oil supply system [24]. Note that the theoretical temperature rise ΔT here is the extreme (maximum) value under the assumption of all the pumping power (P_p) and friction power being 100% contributed into heating the oil without any energy loss.

\[ P_f = \mu V \frac{V^2}{h} \]  
\[ P_p = P_s Q_p \]  
\[ \Delta T = \frac{P_f + P_p}{c \rho Q_p} \]

where \( \mu \) is the viscosity of the oil (Pa·s), V is the sliding feed rate (m/s) of the bearing pad, A is the total area of the bearing pads (m²), h is the oil film gap (m), \( P_s \) is the supply pressure of the oil (N/m²), \( Q_p \) is the volume flow rate, c is the specific heat capacity of the oil and \( \rho \) is the density of the oil.

2.5. Modelling and Thermal Analysis of a Ballscrew Nut

In a ballscrew drive system, thermal deformation occurs owing to the temperature rise caused by the large amount of friction between the nut and the ball. Since the heat is generated by the friction between the ball moving in the groove of the nut as well as between the nut and the ball, the frictional heat flux of a ball moving at a high speed can be regarded as being evenly distributed in the inner surface of the nut groove, and can be taken as the heat-source boundary condition in the FEM. Figure 8a shows the frictional heat source of a nut groove. From Lin’s study [9], the heat generated by the nut varies according to the changes in the rotational speed, external load and friction of the ballscrew. The influence of the rotational speed is greater than that of the external load. Since the nut groove serves as the heat zone caused by friction, the heat source is substituted into its contact zone. On the other hand, since the nut is exposed to air, some of the heat is lost due to convection; therefore, the temperature of the nut would reach equilibrium. Another boundary condition for the heat analysis of the nut shown in Figure 8b is the convection boundary of a nut surface exposed to air. In this paper, the heat resistance between the nut groove and nut surface was neglected. That is to say that the temperature of the nut groove is the same as the temperature of nut surface. The boundary condition of heat transfer from the worktable surface to air is natural convection, and the heat transfer between the ballscrew nut and worktable is structural conduction.

![Figure 8](image-url)

**Figure 8.** Boundary conditions for the thermal analysis of the ballscrew nut. (a) Frictional heat source of groove. (b) Convection boundary of nut surface.

Since the one-dimensional heat transfer pathway between the nut groove and the surface of the nut is very short, this study regarded it as an expression of low thermal resistance. Therefore, under natural convection conditions, the temperature gradient of
the nut groove and the surface of the nut would not be too high. When simulating the temperature rise of the nut, this study neglected the temperature gradient between the nut groove and the surface of the nut. One thermal couple was attached on the surface of the nut to measure the temperature. Thermal experiments of three different rotational speeds were performed to measure the surface temperature of the nut, which served as the boundary conditions of the nut groove in a steady state. These three rotational speeds (125, 250 and 500 rpm) specified for the ballscrew nut can be derived into three corresponding worktable feed rates (1.25, 2.5 and 5 m/min) by converting the pitch (10 mm) of the ballscrew. The purpose of the thermal experiments was to tune the boundary conditions of the finite element model of the nut towards real running conditions. Table 2 lists the measurement results of nut temperature at the three feed rates in the three experiments. The mean temperature of the nut was then set as the boundary condition of thermal analysis in the finite element model of the system.

Table 2. Measured temperature of the ballscrew nut in three experiments.

| Feed Rate          | 1st Exp. (°C) | 2nd Exp. (°C) | 3rd Exp. (°C) | Mean Temp (°C) |
|--------------------|---------------|---------------|---------------|----------------|
| 5 m/min (500 rpm)  | 43.95         | 44.13         | 43.97         | 44.01          |
| 2.5 m/min (250 rpm)| 37.61         | 37.73         | 37.80         | 37.71          |
| 1.25 m/min (125 rpm)| 31.80       | 32.61         | 32.49         | 32.30          |

The thermal effect of the vertical guideway system comes from three main contributions: one is from the motor of the vertical drive system, another is from the oil film sliding effect of the hydrostatic bearings, and the other is from the friction heat of ballscrew nut. The heat generated by the motor can be considered as a fixed heat source with rated power in the finite element model. Figure 9 shows the thermal deformation simulation of a vertical guideway system resulting from a ballscrew nut at a feed rate of 5 m/min at the location of (a) nut, (b) nut set (or holder) and (c) worktable. The mechanical contacts of the structure connection with the top and the bottom of the ballscrew were fixed connections.

![Figure 9. Thermal deformation of ballscrew nut and worktable at feed rate of 5 m/min.](image)

2.6. Thermal Experiments of a Vertical Hydrostatic Guideway System

The thermal experiments of a vertical hydrostatic guideway system were carried out in order to measure the temperature rise on the thermocouples at three different feed rates, namely 5 m/min, 2.5 m/min and 1.25 m/min. Temperature data were then imported into the FE model to simulate the thermal deformation of the system. Figure 10 depicts the test setup of a vertical hydrostatic guideway system. Figure 11 illustrates the schematic diagram of the temperature measurement system. There were eight thermocouples installed in this system to measure the temperature at six locations of the worktable, one thermocouple at the nut set, and one thermocouple at the wall for environmental room temperature (T₇).
The attached locations of seven thermocouples (T₁, T₂, T₃, T₄, T₅, T₆ and T₈) are indicated in Figure 12.

**Figure 10.** Experimental test setup.

**Figure 11.** Schematic diagram of the temperature measurement system.
2.6.1. Experimental Conditions and Test Results

For the thermal experiments, the operating rotational speeds of the ballscrew were specified as 125, 250 and 500 rpm; the corresponding linear feed rates of the worktable were 1.25, 2.5 and 5 m/min, respectively. The oil supply pressure was 20 bar, the initial temperature of the inlet oil was 27 °C and the ambient room temperature ranged from 25 °C to 28 °C. Thirty minutes after startup [8,9], the data-acquisition system started to record the temperature data of thermocouples, and the data were continuously recorded for six hours in order to construct the curves of the temperature changes in the thermocouples over time. Thermal experiments based on three different feed rates were conducted in this study. Each experiment was replicated three times under the same conditions to examine the repeatability of the results. The temperature rise curves over time were obtained and processed by using data-acquisition software; the values were then tabulated, and the initial and steady-state temperatures were recorded.

The worktable of a vertical hydrostatic guideway system achieves thermal steady-state status (thermal equilibrium) when its thermocouple temperature no longer changes with time. The time required to achieve thermal steady-state is defined as the time from the start of the experiment to the time when the steady-state temperature is achieved. The experimental results showed that the system took approximately six hours to reach the steady-state temperature. Figure 13 demonstrates the three test results of the thermal experiments at a feed rate of 5 m/min over six hours. Table 3 shows the temperature measurements of the seven thermocouples and their mean temperatures over the three experiments at a feed rate of 5 m/min. Figures 14 and 15 show the three test results of the thermal experiments at feed rates of 2.5 m/min and 1.25 m/min, respectively, over six hours. Tables 4 and 5 show the temperature measurements of the seven thermocouples and their mean temperatures over the three experiments at feed rates of 2.5 m/min and 1.25 m/min, respectively.
Table 3. Measured temperature of thermocouples at feed rate of 5 m/min.

| Thermocouple No. | 1st Exp. (°C) | 2nd Exp. (°C) | 3rd Exp. (°C) | Mean Temp (°C) |
|------------------|---------------|---------------|---------------|----------------|
| T1               | 31.48         | 31.56         | 31.52         | 31.52          |
| T2               | 31.20         | 31.22         | 31.30         | 31.24          |
| T3               | 31.25         | 31.35         | 31.33         | 31.31          |
| T4               | 33.46         | 33.76         | 33.77         | 33.72          |
| T5               | 33.32         | 33.65         | 33.65         | 33.54          |
| T6               | 43.95         | 44.13         | 43.97         | 44.01          |

Table 4. Measured temperature of thermocouples at feed rate of 2.5 m/min.

| Thermocouple No. | 1st Exp. (°C) | 2nd Exp. (°C) | 3rd Exp. (°C) | Mean Temp (°C) |
|------------------|---------------|---------------|---------------|----------------|
| T1               | 30.01         | 30.12         | 30.17         | 30.1           |
| T2               | 30.03         | 30.19         | 30.21         | 30.14          |
| T3               | 29.73         | 29.89         | 29.92         | 29.84          |
| T4               | 29.82         | 29.99         | 30.03         | 29.94          |
| T5               | 31.28         | 31.39         | 31.45         | 31.37          |
| T6               | 31.13         | 31.27         | 31.30         | 31.23          |
| T8               | 37.61         | 37.73         | 37.80         | 37.71          |

Figure 13. Thermal experimental results of three tests at feed rate of 5 m/min.

Figure 14. Thermal experimental results of three tests at feed rate of 2.5 m/min.

Figure 15. Thermal experimental results of three tests at feed rate of 1.25 m/min.
2.6.2. Comparison between FE-Simulated and Experimental Temperatures

This study utilized the “Path Line” post-processing function in ANSYS to obtain accurate temperature measurements at the locations on the worktable. This method is shown in Figure 16, where three paths were established between thermocouple T1 and T6. The temperatures acquired at both ends of the line served as the basis for comparing the FE-simulated temperatures and experimental temperatures. Tables 6–8 present a comparison between the experimental and FE-simulated temperatures of thermocouples T1 to T6 and their mean absolute errors (MAE) acquired according to the aforementioned “Path Line” function. The simulated temperature results were slightly higher than the experimental results, with the temperature difference being less than 1.29 °C; however, the errors were all below 3.97% and the simulated temperature distribution was similar to the experimental temperature distribution in terms of tendency. The high-temperature zones of the worktable were within the proximity of the nut holder (T5, T6), and the heat was transferred to the surrounding low-temperature zones (T1 to T4). The lower the feed rate, the smaller the distribution of the temperature difference of the worktable. The FE simulation results were concordant with the experimental results. Therefore, the thermal analysis FE model for the hydrostatic guideway system established in this study can be utilized as a reference model to predict thermal displacement of the worktable center point.

![Figure 16. Establishment of three paths on the worktable.](image)

Table 5. Measured temperature of thermocouples at feed rate of 1.25 m/min.

| Thermocouple No. | 1st Exp. (°C) | 2nd Exp. (°C) | 3rd Exp. (°C) | Mean Temp (°C) |
|------------------|--------------|--------------|--------------|---------------|
| T1               | 27.28        | 27.79        | 28.37        | 27.81         |
| T2               | 27.33        | 27.84        | 28.39        | 27.85         |
| T3               | 27.03        | 27.54        | 28.12        | 27.56         |
| T4               | 27.21        | 27.68        | 28.27        | 27.72         |
| T5               | 28.00        | 28.52        | 28.97        | 28.49         |
| T6               | 27.88        | 28.40        | 28.86        | 28.38         |
| T8               | 31.80        | 32.61        | 32.49        | 32.30         |

Table 6. Comparison of experimental and FE simulation results at feed rate of 5 m/min.

| Thermocouple No. | Av. Exper. Temp (°C) | Simul. Temp. (°C) | MAE (°C) | Error (%) |
|------------------|----------------------|-------------------|-----------|-----------|
| T1               | 31.48                | 32.57             | 1.09      | 3.35      |
| T2               | 31.52                | 32.63             | 1.11      | 3.40      |
| T3               | 31.24                | 32.53             | 1.29      | 3.97      |
| T4               | 31.31                | 32.58             | 1.27      | 3.90      |
| T5               | 33.72                | 34.05             | 0.33      | 0.97      |
| T6               | 33.54                | 34.06             | 0.52      | 1.53      |
Table 7. Comparison of experimental and FE simulation results at feed rate of 2.5 m/min.

| Thermocouple No. | Av. Exper. Temp (°C) | Simul. Temp. (°C) | MAE (°C) | Error (%) |
|------------------|----------------------|-------------------|----------|-----------|
| T1               | 30.1                 | 30.77             | 0.67     | 2.18      |
| T2               | 30.14                | 30.81             | 0.67     | 2.17      |
| T3               | 29.84                | 30.74             | 0.90     | 2.93      |
| T4               | 29.94                | 30.78             | 0.84     | 2.73      |
| T5               | 31.37                | 31.67             | 0.30     | 0.95      |
| T6               | 31.23                | 31.67             | 0.44     | 1.39      |

Table 8. Comparison of experimental and FE simulation results at feed rate of 1.25 m/min.

| Thermocouple No. | Av. Exper. Temp (°C) | Simul. Temp. (°C) | MAE (°C) | Error (%) |
|------------------|----------------------|-------------------|----------|-----------|
| T1               | 27.81                | 27.85             | 0.04     | 0.14      |
| T2               | 27.85                | 27.87             | 0.02     | 0.07      |
| T3               | 27.56                | 27.83             | 0.27     | 0.97      |
| T4               | 27.72                | 27.85             | 0.13     | 0.47      |
| T5               | 28.49                | 28.45             | 0.04     | 0.14      |
| T6               | 28.38                | 28.45             | 0.07     | 0.25      |

Based on the aforementioned findings in both the FE simulations and experiments, the upper surface of the worktable was hotter than the lower surface (T1 > T3, T2 > T4). As a consequence, the deformation of the worktable along the center of the axis was asymmetrical. This can be seen in Figures 17 and 18, where heat transfer occurred between the contact surfaces of the nut flange and the nut holder. As shown in Figure 17a, the high-temperature zones on the nut holder were closer to the flange, and the upper area was hotter than the lower area. Figure 17b shows that the worktable and the nut holder were fastened together by screws. Figure 17c shows the temperature distribution between the contact areas of the nut holder and the worktable, in which the nut holder transfers the asymmetrical heat source to the worktable. Figure 17d shows that when the temperature was asymmetrically distributed along the center of the axis, the thermal strain and thermal deformation was asymmetrical as well.

![Figure 17](image-url)

**Figure 17.** Cont.
3. Other FE Simulation Results and Discussions

Since the FE thermal simulation model of this vertical hydrostatic guideway system was verified in terms of its correctness with respect to the aforementioned experiments, this FE model can be used to simulate different service conditions and to predict the performance of hydrostatic bearings, the structural deformation and the thermal deformation of the system, and even to calculate the thermal displacement (errors) of the worktable at the center point.

3.1. Simulation Results of Hydrostatic Bearings

The temperature rise in the vertical hydrostatic guideway system was analyzed with respect to different oil film clearances (10 µm~30 µm), sliding feed rate/speeds (0.1 m/s~1 m/s) at different supply pressures (10 bar, 15 bar 20 bar) and with oil of different viscosity grades (VG46, VG32 and VG15) [25–27]. Some selected results are shown in Figures 19 and 20. Note that the temperature rise happened mainly on the bearing sill due to the small film clearance. Figure 21 depicts the load-carrying capacity of the normal recess pad and side recess pad at a recess pressure of 10 bar in different oil-film clearances. Figure 22 displays the load-carrying capacity of the normal recess pad at a recess pressure of 10 bar with oil of different viscosity grades (VG46, VG32 and VG15).
Figure 19. Temperature rise w.r.t. different supply pressures (VG32).

Figure 20. Temperature rise w.r.t. different viscosity grades of oil ($P_s = 15$ Bar).

Figure 21. Load-carrying capacity versus oil-film clearance at $P_r = 10$ bar.

Figure 22. Load-carrying capacity versus three viscosity grades at $P_r = 10$ bar.

3.2. Simulation Results of Structural Deformation

The forces acting on the normal recess pad (2443 N) and side recess pad (3049 N) due to the pressurized hydrostatic bearing at $P_r = 10$ bar were obtained from ANSYS Fluent analysis as shown in Figure 23. Figure 24 shows the structural deformation of the clamp plate due to the pressurized hydrostatic bearing, and the structural deformation of clamp plate ranged from 17.8 to 24 $\mu$m at $P_r = 10$ bar [23]. The structural deformation of the clamp plate may have caused the further increasing in the oil-film clearance to approximately 18 $\mu$m. It may have decreased the stiffness of the...
hydrostatic bearing at the clamp plate significantly owing to the dramatic increase in the oil-film gap. Table 9 shows the opening of the oil-film gap due to structural deformation, which resulted in the load-carrying capacity of the hydrostatic bearing decreasing by 6.83%. Note that the clamp plate was fastened to the worktable by multiple bolts and screws [28]. The mechanical contact between the clamp plate and the worktable flange was friction contact, with a coefficient of friction value of 0.1. The structural deformation analysis in this study was considered in the range of elastic deformation of material. The recess pressure acting on the fixed rail of the guideway system is illustrated in Figure 25, and its result indicated a minor effect on the rail deformation (deformation of 0.076162 µm only), therefore its effect was ignored in this study. A mesh quality analysis was performed on the proposed FEM models. According to reference [25], the mesh quality of the FEM model of machine tools should be bigger than 0.5 in analysis. For the FEM model in this paper, the majority (98%) of the mesh quality was bigger than 0.5, and mostly fell in the range between 0.75 and 0.95, which is considered acceptable for analyzing machine tools. The mesh quality analyses of the vertical guideway structure and worktable are shown in Figures 26 and 27.

Table 9. Load-carrying capacity affected by structural deformation of clamp plate.

| Recess Pad Location | Before Struct. Deformation | After Struct. Deformation | Load Capacity Drop (%) |
|---------------------|---------------------------|--------------------------|------------------------|
|                     | Film-Gap (µm) | Load Capacity (N) | Film-Gap (µm) | Load Capacity (N) |                     |
| At clamp plate      | 10            | 2443                    | 27.8             | 2275.9             | 6.83%                |
Figure 24. Clamp plate deformed due to recess pressure.

Table 9. Load-carrying capacity affected by structural deformation of clamp plate.

| Recess Pad Location | Before Struct. Deformation | After Struct. Deformation | Load Capacity Drop (%)
|---------------------|----------------------------|---------------------------|------------------------
| Recess Pad Location | Film-Gap (µm) | Load Capacity (N) | Film-Gap (µm) | Load Capacity (N) |
| At clamp plate 1 | 2443 | 27.8 | 2275.9 | 6.83% |

Figure 25. (a) Recess pressure acting on fixed rail. (b) Structural deformation on fixed rail.

Figure 26. Mesh quality analysis of the structure of a vertical guideway.

Figure 27. Mesh quality analysis of the structure of a worktable.

3.3. Simulation Results of Thermal Deformation in Worktable

The thermal deformation at a fixed location of the ballscrew nut under the worktable is indicated in Figure 28. The worktable center appeared to be convex due to the thermal deformation caused by the frictional heat source of ballscrew nut. The thermal displacement of the path line in the center worktable is shown in Figure 29. Figure 30 explains the shifting of the worktable center point resulting from thermal deformation at a feed rate of 1 m/s. The axial thermal displacements of the worktable at the center point shown in Table 10 are the simulation results from the FE analysis at a feed rate of 1 m/s.
**Table 10.** Axial thermal displacement of worktable at center point.

| Displacement of Center Point (µm) | ΔX   | ΔY   | ΔZ   |
|----------------------------------|------|------|------|
|                                  | 0.1539 | 0.0009 | 2.0246 |

**Figure 28.** Thermal deformation of the worktable.

**Figure 29.** Displacement of path line in X, Y and Z direction.

**Figure 30.** Shifting of worktable at center point resulting from thermal deformation.
4. Conclusions

The study of the thermal characteristics of a vertical hydrostatic guideway system for
milling machines obtained some conclusions as follows:

(1) This study proposed an FE model to simulate the thermal characteristics of a vertical
hydrostatic guideway system by considering three heat sources, namely motor heat,
viscous shearing heat generated from the oil-film of hydrostatic bearings and friction
heat generated from the ballscrew nut. Nine thermal experiments were implemented
at three different feed rates to verify the boundary conditions of the FE model. The
temperature difference between FE model simulations and experimental results were
less than 3.97%.

(2) The load-carrying capacity of the hydrostatic bearing at the clamp plate was 6.83%
lower in the design of this study owing to the structural deformation of the clamp
plate caused by recess pressure.

(3) The FE model of the vertical hydrostatic guideway system built in this study can
be used to predict the performance of hydrostatic bearings, structural deformation
of components and the thermal deformation of the worktable at the center point at
different running conditions (e.g., feed rates).

(4) Under the sliding feed rate of 1 m/s, the predicted positioning error of worktable
at center point was 0.1539 µm in the X direction, 0.0009 µm in the Y direction and
2.0246 µm in the Z direction, due to thermal deformation.

Author Contributions: Conceptualization, methodology, supervision, funding acquisition and writing
original draft: H.-C.H. Software, formal analysis, investigation, visualization, and data validation:
W.-H.Y. All authors have read and agreed to the published version of the manuscript.

Funding: The financial support of this research from the Ministry of Science and Technology of
Taiwan (project no. MOST 106-2221-E-151-021) is gratefully acknowledged.

Data Availability Statement: Not applicable.

Acknowledgments: The test setup of the vertical hydrostatic guideway system provided by the
Intelligent Machinery Technical Center, Industrial Technology Research Institute, Taichung, Taiwan is
gratefully acknowledged.

Conflicts of Interest: The authors declare no conflict of interest.

References
1. Chang, E.S. Precision Feed System Technology. Mach. Tool Accessory Mag. 2009, 55.
2. Bryan, J. International Status of Thermal Error Research. CIRP Ann. 1990, 3, 645–656. [CrossRef]
3. Chen, D.; Bonis, M.; Zhang, F.; Dong, S. Thermal error of a hydrostatic spindle. Precis. Eng. 2011, 35, 512–520. [CrossRef]
4. Yang, S.; Wu, P.; Liu, S.; Sun, L.; Zhao, P.; Long, X.; Jiang, Z. Deformation and Thermal Analysis of the Guideways of a Large Scale
Aspheric Machine Tool. Procedia CIRP 2015, 27, 181–186. [CrossRef]
5. Bo, S.K.; Gyeong, T.B.; Gwi, N.K.; Hong, M.M.; Jung, P.N.; Sun, C.H. A Study on the Thermal Characteristics of the Grinding
Machine Applied Hydrostatic Bearing. T Can. Soc. Mech. Eng. 2015, 39, 717–728.
6. He, F.; Huang, Z.; Chen, X.; Xu, K.; Chen, S. Simulating and Optimizing Temperature Field of Oil Film for CNC Vertical Lathe
Hydrostatic Rotary Table. Mech. Sci. Tech. Aerosp. Eng. 2015, 34, 1733–1737.
7. Horejs, O. Thermo-Mechanical Model of Ball Screw With Non-Steady Heat Sources. Int. Conf. Therm. Issues Emerg. Technol. Theory
Appl. 2007, 133–137.
8. Yu, S.W. The Thermal Analysis of Nuts in Ball Screw Systems. Master’s Thesis, National Cheng Kung University, Tainan,
Taiwan, 2009.
9. Lin, C.Y. Thermal Analysis and Verification for Double-Nut Ball Screw. Master’s Thesis, National Formosa University, Yun-Lin,
Taiwan, 2014.
10. Dianrong, G.; Dan, Z. The Theoretical Analysis and Numerical Simulation of Dynamic Characteristics of Hydrostatic Guides for
Heavy NC Machine Tool. WASE Int. Conf. Inform. Eng. 2010, 3, 114–117.
11. Liu, Y. Analysis of Dynamic and Static Characteristics of Hydrostatic Guide for Ultra-Precision Machine Tools and Modal
Parameter Identification. Master’s Thesis, Harbin Institute of Technology, Harbin, China, 2010.
12. Wang, Z.; Zhao, W.; Li, B.; Zhang, J.; Lu, B. Fluid-structure interactions (FSI) on static characteristics of hydrostatic guideways. In Proceedings of the IEEE International Symposium on Assembly and Manufacturing (ISAM), Tampere, Finland, 25–27 May 2011; pp. 1–5.
13. Yang, L. Performance Analysis and Research of Closed Hydrostatic Guide. Master’s Thesis, Northeastern University, Shenyang, China, 2012.
14. Dong, P. Research on the Static and Dynamic Characteristics of Ultra-Precision Hydrostatic Guide and Its Control Technology. Master’s Thesis, Harbin Institute of Technology, Harbin, China, 2013.
15. Chen, X.;Hong, H.; Yin, Y. Ontological Knowledge Capture for the Design of ultra-precision Hydrostatic Guideways Based on Imaginal Thinking. *Enterp. Syst. Conf.* 2014, 119–124.
16. Gong, J. Multi-Objective Optimization and Flow Field Simulation Analysis of Hydrostatic Guide. Master’s thesis, Guangdong University of Technology, Guangzhou, China, 2014.
17. Jiang, Y.; Hou, G.; Sun, T. Finite element Analysis of Thermal Characteristics of hydrostatic Guide. *Aeronaut Precis. Manuf. Tech.* 2011, 23–25.
18. Liu, T.; Li, C.; Duan, R.; Qu, H.; Chen, F.; Zhou, Z.; Zhang, J.; Shi, Z. Viscous Oil Film Thermal Modeling of Hydrostatic Bearings with a Rectangular Microgroove Surface. *Front. Energy Res.* 2022, 10, 81380. [CrossRef]
19. Chen, D.; Zhang, X. Flow-Heat-Solid Coupling Thermal Deformation Analysis of Hydrostatic Spindle under Viscosity Temperature Effect. *Res. Square.* 2021. [CrossRef]
20. Liu, F.; Jiang, S. Temperature Rise Characteristics of High-speed Water-lubricated Hydrostatic Thrust Bearing. *J. Phys. Conf. Ser.* 2021, 106, 012050.
21. Zhang, Y.; Kong, P.; Feng, Y.; Guo, L. Hot Oil Carrying Characteristic about Hydrostatic Bearing Oil Film of Heavy Vertical Lathe in High Speed. *Ind. Lubr. Tribol.* 2019, 71, 126–132. [CrossRef]
22. Liu, P.; Chen, W.; Su, H.; Chen, G. Dynamic Design and Thermal Analysis of an Ultra-precision Flycutting Machine Tool. *Proc. Inst. Mech. Eng. Part B J. Eng. Manuf.* 2016, 232, 404–411. [CrossRef]
23. Yang, W.-H. Characteristics Analysis and Verification of Vertical Guideway System. Master’s Thesis, National Kaohsiung University of Science and Technology, Kaohsiung, Taiwan, 2018.
24. Rowe, W.B. *Hydrostatic, Aerostatic and Hybrid Bearing Design*; Butterworth-Heinemann: Oxford, UK, 2012.
25. Yang, S.; Zhao, P.; Xu, Y.; Sun, I.; Wu, P.; Long, X.; Jiang, Z. Hydrostatic Worktable Performance of an Ultra-precision Optical Aspheric Machine Tool. *Procedia CIRP* 2015, 27, 187–191. [CrossRef]
26. Liu, Z.; Zhan, C.; Cheng, Q.; Zhao, Y.; Li, X.; Wang, Y. Thermal and tilt effects on bearing characteristics of hydrostatic oil pad in rotary table. *J. Hydrodyn. Ser. B* 2016, 28, 585–595. [CrossRef]
27. Huang, H.-C.; Yang, S.-H. Thrust-Bearing Layout Design of a Large-Sized Hydrostatic Rotary Table to Withstand Eccentric Loads for Horizontal Boring Machine Applications. *Lubricants* 2022, 10, 49. [CrossRef]
28. Liu, Y.L. Analysis of the Static and Dynamic Characteristics and Identification of Model Parameters of a Hydrostatic Guideway for Ultra-Precision Machine Tool Applications. Master’s Thesis, Harbin Institute of Technology, Harbin, China, 2010.