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THERMAL BEHAVIOUR OF ULTRA-PRECISION SPINDLE

The thermal behavior of ultra-precision spindles is crucial for improving machining accuracy and productivity. One of the primary heat sources in machine tool spindles is the bearings. Aerostatic and hydrostatic spindle bearings are widely used for ultra-precision machine tools [1]. Aerostatic bearings have the main advantage of high temperature resistance at high speeds. However, they have poor damping and rigidity characteristics compared to oil-lubricated bearings.

On the other hand, oil-lubricated hydrostatic bearings are advantageous for ultra-precision spindles in achieving high machining accuracy due to their higher bearing rigidity. However, using oil to lubricate hydrostatic bearings results in power loss due to heat generation from the oil flow at high speeds [2]. A spindle with water-lubricated hydrostatic bearings was proposed to cope with the disadvantage of both the aerostatic and oil hydrostatic bearings [3].

This study investigated the thermal behavior of a spindle with water-lubricated hydrostatic bearings via simulation with experimental data. The simulation shows the influence of thermal deformations on the thermal expansion of the bearing surfaces, the gap change in bearings, and load capacity and stiffness.

The precision water-lubricated spindle [4] under study has two four-recessed journal bearings and four recessed double-sided thrust bearings with restrictors for each recess. The temperature variations of the water and spindle flow were estimated using simulation and experimental data. The temperature distribution inside the spindle-bearing gap and the associated flow channels was obtained by numerical simulation using the experimentally obtained data as boundary conditions.

The simulation was divided into two stages. In the first stage, the temperature field in the spindle was determined using the CFD software. At this stage, the rotor is assumed to be stationary. To investigate the effect of lubricant temperature change caused by rotor rotation, the temperature boundary conditions gathered experimentally at a specific spindle speed were used. In the second stage, a thermal analysis of the spindle was performed using FEM software. This analysis aimed to define the spindle displacement caused by thermal deformations. This study did not consider the influence of the rotor deformation caused by an external load and rotor displacement due to the water pressure distribution.

It was established that the spindle rotor ends elongated by around 2.7 μm due to the influence of thermal deformation. This affects the accuracy of the machined workpiece's linear dimensions. Furthermore, thermally induced displacements of bearing surfaces lead to changes to the initial design gaps for the journal and thrust bearings. The gap size of the hydrostatic bearings is a critical parameter that determines the bearing load factor and stiffness. It was shown that the thermal deformation decreased the thrust bearing gap up to 14% on average and caused a journal bearing gap variation up to 15%. The primary influence on the bearing gap size for thrust and journal bearings causes the deformation of the bearing casing.

It should be noted that thermal deformation causes the bearing surfaces' imperfection in both the axial and radial directions. A surface imperfection is characterized by a longitudinal profile deviation in the axial direction: a barrel-shaped deviation for the casing bore and a taper-shaped deviation for the rotor journal.

It was defined that, in the radial direction, thermal deformation caused the rotor journal to expand uniformly by 0.63 μm on average. In addition, the spindle casing deformed as well.

Because the spindle casing is fixed to a base plate, the bottom surface of the casing can be regarded as a reference position. In contrast, the upper part of the casing can freely deform upward. As a result, the profile of the casing bore shifted in the vertical direction causing a motion error in the vertical position of the cutting tool or workpiece attached to the rotor end.

As it is well known, a non-uniform bearing gap causes deterioration of spindle accuracy. Therefore, the influence of bearing gap variation on the spindle output characteristics was investigated. For the thrust bearing, it was established that the average value of the gap standard deviation in the circumferential direction is 0.046 μm, which is less than 0.5% of the design bearing gap size. Thus, we could estimate the bearing characteristics based on the average gap value with satisfactory accuracy assuming the linearity of the gap change. Unlike thrust bearings, the thermal effect on journal bearings leads to a significant non-uniform radial gap along the bearing's circumferential direction. A Fourier series was proposed to define the bearing surface imperfection. Considering the form of computed profiles for the casing bore and bearing journal surfaces, the number of harmonics of the series can be limited to 4 for the casing and 1 for the rotor. Then, the radial gap function $h = f(\varphi)$, which accounts for the bearing surface imperfection, was represented as

$$h(\varphi) = h_0 + A_0 + \sum_{k=1}^{4} A_k \sin\left(k\varphi + \psi_k\right) - \left[B_0 + B_1 \sin\left(\varphi + \chi_1\right)\right],$$

where h_0 is the design gap value; A_0, A_k, and ψ_k are the zero harmonic, subsequent harmonics, and the initial phase characterizing the contour position of the casing bore profile, respectively; and B_0, B_1, and χ_1 are the zero harmonic, the first harmonics, and the initial phase of the first harmonic characterizing the contour position of the rotor journal profile, respectively.

Then, the obtained function $h(\varphi)$ is used to compute the characteristics of a journal bearing. A pressure distribution function was the basis for calculating the output characteristics of journal bearings. The calculation of the pressure distribution function is grounded in the Reynolds equation. Bearing recess pressure, which is the boundary condition for solving the Reynolds equation, is determined by solving the equation of the balance of fluid flow rate.

The thermal stability of a spindle under the influence of an external temperature field was evaluated using the bearing load capacity and stiffness as indicators. The present study shows that the load capacity and stiffness decrease when thermal effects are accounted for. However, this trend is quantitatively different for spindle bearings: it is relatively minor for the recessed journal bearings, and it causes a stiffness drop of up to 14% for the opposing-pad thrust bearings. This also contributes to decreased accuracy of the linear dimensions of machined parts.

The effect of rotor rotation on the bearing stiffness was also evaluated. For this purpose, a simulation was conducted considering the rotor rotation at a maximum speed of 3000 min^{-1}. The spindle rotation slightly increases the journal bearings' stiffness without changing the general pattern. Moreover, at small rotor displacements from its equilibrium position, the indicated increase in stiffness does not exceed 1%.

A separate study investigated the effect of changing the viscosity due to an increase in lubricating fluid temperature on bearing performance. The obtained spindle's thermal characteristics are planned to be used to develop a spindle temperature control system.

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MECHANISM WITH VARIOUS FUNCTIONS OF KNIFE BLADE MOVEMENT

There is a wide variety of methods for trimming sheet materials, which significantly influence the quality indicators of the final finishing of printed products. Continuous trimming methods and devices have a number of advantages over stationary cutting methods, namely: increased productivity; reduced time for equipment adjustment; energy savings during the process; improved cut quality; reduced metal consumption and dimensions, etc. For this reason, machines with movable knives have become widespread [1–7].

The author has proposed an improved version of the trimming mechanism for sheet materials (hereinafter – SM) described in works [6–7], which includes a knife (1) mounted in a frame (2) (Fig. 1), kinematically connected to a slider (3), which in turn is installed in a guide (4). A pair of pins (5) are mounted on the frame (2), and a shaft (6) on which a pair of eccentrics (7) are installed with adjustable positions, kinematically connected to the pair of pins (5) by means of a pair of double slotted links (8), installed in a pair of guides (9). Shaft (6) is mounted in a pair of supports (10).

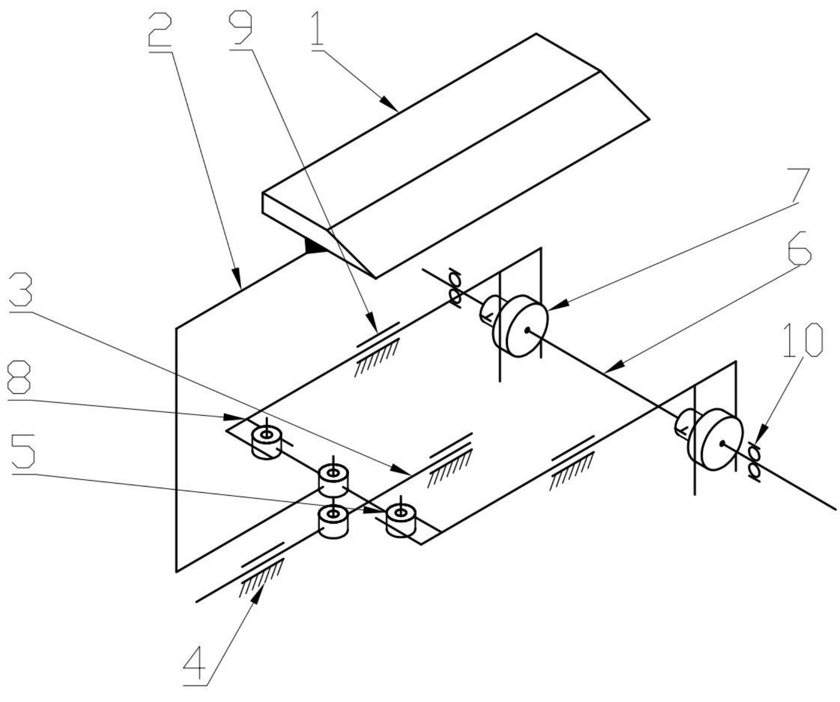

Fig. 1 – Kinematic scheme of the sheet material trimming mechanism

The presence of two driving links (eccentrics 7) in the mechanism is due to the fact that the knife receives two types of motion when the eccentrics 7 are set with a phase angle of 90°: reciprocating motion in a direction perpendicular to the SM movement, and reciprocating-rotary motion in a direction perpendicular to the SM movement. The trajectory of point A located on knife (1) and coinciding with the center of oscillation is a straight line (OO1) perpendicular to the feed plane of the SM (Fig. 2a). The trajectory of any other point on the knife edge has a complex path. From the initial position (1-1), the blade performs a linear motion in a plane perpendicular