DYNAMICS OF A HIGH PERFORMANCE WORKPIECE TABLE FOR ACTIVE CONTROL DURING PRECISION GRINDING

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Introduction

In the surface grinding process, a high speed work table motion is involved. In order to achieve active control of the process, the commands will include substantial amount of high frequency signals, which could not be sufficiently implemented by the wheel infeed system that typically has a small bandwidth. In order to solve this problem, a separate actuating unit will be required to support the workpiece in order to implement small but fast control actions. The workpiece actuating unit and the wheel infeed unit will work together under a composite control scheme [1], in which the advantages of the wheel infeed unit and the workpiece actuating unit are combined, such that both the large driving force capability and the fast active control can be achieved.

To realize the workpiece actuating unit to implement positioning functions that are fast but at the level of micrometers, a piezoelectric translator is used. Compared with the traditional feed mechanisms, workpiece micro-positioning tables using PZT are advantageous in resolution, static and dynamic stiffness, and most importantly, dynamic response. In precision positioning, this type of positioning tables has been used to generate compensation actions [2-9].

However, the tables have some inherent shortcomings such as hysteresis and parasitic motion that exists in multi-dimensional positioning. An additional one is overshoot. Using closed loop control and correctly tuned controllers, the hysteresis and parasitic motion could be solved. However, the overshoot problem during high speed positioning remained critical. It can be seen that high speed dynamic micro-positioning has not been achieved [2-9]. Typically, the settling time was 150ms and the resolution was 500nm [9]. Further improvement in high speed positioning accuracy is necessary.

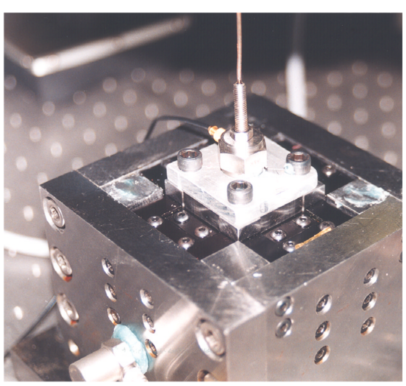

Fig. 1 Testing of the micro-positioning table

Design of the Micro-positioning Table

To achieve the above objectives, a new workpiece micro-positioning table has been developed (Fig. 1). It has a piezoelectric translator at its center to generate a displacement up to 47.7μm and a pushing force up to 1000N. This would give the workpiece table a working range of approximately 45μm after the removal of the effect due to the spring stiffness on the expansion range of the piezoelectric translator. The rigidity of the translator is approximately 17N/μm. The PZT is used to push the moving part in the workpiece table. Four precision ball bearing guides are used to provide support for the moving part. This is to ensure that a good rigidity in the horizontal plane is maintained to reduce the possible deformation generated by the tangential grinding force and to reduce the parasitic errors. The moving part is used to support the top piece that in turn is used to mount a workpiece. The area of the top piece is 100x100mm.

Transient Displacement Models of the Workpiece Table

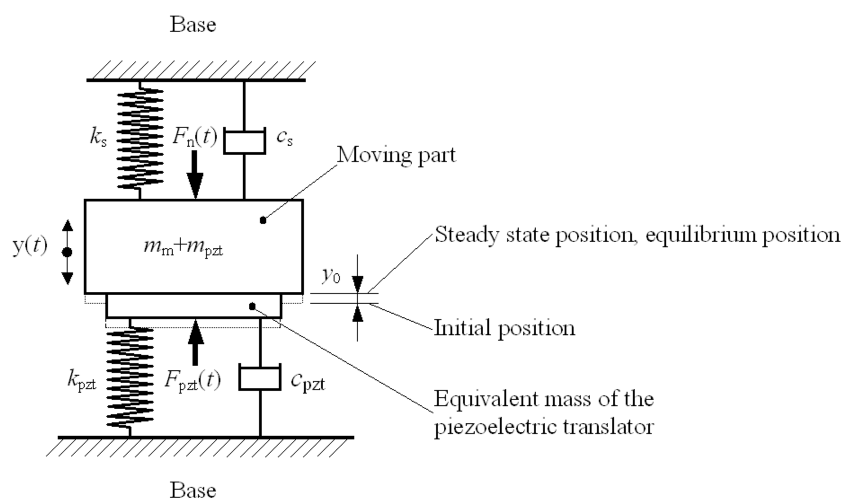

Fig. 2 Mechanics model of the workpiece micro-positioning table

In order to investigate the dynamics of the table, a mechanics model is used (Fig. 2). The moving part and the four springs are included. The PZT is modeled as a mass and spring system with a damping coefficient. $F_{pzt}(t)$ is the force generated by the PZT, $F_n(t)$ is the normal grinding force, $y(t)$ is the table displacement, and y_0 is the steady state displacement.

The mass-spring system will respond dynamically if it is subject to the piezoelectric driving force, $F_{pzt}(t)$, as the input to the system. The differential equation of the displacement $y(t)$ can be obtained as follows:

$$(m_{pzt} + m_m)\ddot{y}(t) + (c_{pzt} + c_s)\dot{y}(t) + (k_{pzt} + k_s)y(t) = F_{pzt}(t) - F_n(t) \tag{1}$$

where m_m is the mass of the moving part, m_{pzt} is the mass of the piezoelectric translator, c_{pzt} is the equivalent damping coefficient of the piezoelectric translator, and c_s is the equivalent damping coefficient of the springs used for preloading. To solve Eq. (1) for the above objectives, the normal grinding force $F_n(t)$ is assumed to be zero. Use the PZT force model during the expansion stage, noted as $F_{pzte}(t)$, the displacement $y(t)$ can be obtained. The acceleration of the table can then be obtained as:

$$\begin{aligned}\ddot{y}(t) = {} & \frac{A}{\omega_n}\frac{e^{-\zeta\omega_n t}}{\sqrt{1-\zeta^2}}(2\zeta\omega_d\cos(\omega_d t+\phi)+(1-2\zeta^2)\omega_n\sin(\omega_d t+\phi)) \\ & +NA\frac{e^{-\zeta\omega_n t}}{\sqrt{1-\zeta^2}}((2\zeta^2-1)\omega_n^2\sin(\omega_d t-\phi)-2\zeta\omega_n\omega_d\cos(\omega_d t-\phi)) \\ & -\frac{LA}{\omega_n}\frac{e^{-\zeta\omega_n t}}{\sqrt{1-\zeta^2}}((2\zeta^2-1)\omega_n^2\sin(\omega_d t)-2\zeta\omega_n\omega_d\cos(\omega_d t))-\frac{GA}{R^2C^2}e^{-\frac{t}{RC}}\end{aligned} \tag{2}$$

where ζ is the damping factor, ω_n is the natural frequency, and

$$2\zeta\omega_n = \frac{c_{pzt} + c_s}{m_{pzt} + m_m},\ \omega_n^2 = \frac{k_{pzt} + k_s}{m_{pzt} + m_m},\ A = \frac{1}{m_{pzt} + m_m} k_{pzt} d_{33} n k_{amp} v_0,\ \phi = \mathrm{arctg}((1-\zeta^2)^{0.5}/\zeta)$$

$$G = -\frac{RC}{2\zeta\omega_n - \omega_n^2 RC - 1/RC},\ N = -G,\ L = \frac{2RC\zeta\omega_n - 1}{2\zeta\omega_n - \omega_n^2 RC - 1/RC}$$

where k_{amp} is the amplification factor of the piezoelectric drive amplifier, RC is the time constant of the piezoelectric translator circuit, d_{33} is the piezoelectric constant, n is the number of layers in the translator, k_{pzt} is the stiffness of the piezoelectric translator, and k_s is the stiffness of the springs for preloading. From Eq. (2), it can be seen that the acceleration of the table is proportional to the coefficient A that is proportional to the input voltage v_0 of the step command signal to the drive amplifier.

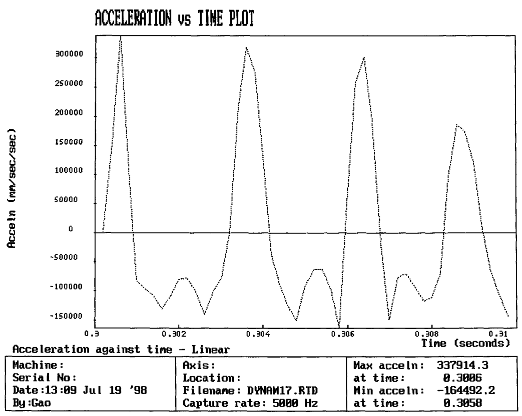

Fig. 3 Experimentally obtained acceleration to a step command signal

Experimental Testing and Discussion

In order to verify the performance of the developed micro-positioning table and the dynamic model, a series of experimental tests have been carried out. A Renishaw ML10/EC10 laser interferometer was used in the tests. To reduce the effect of external disturbance such as vibration, the tests were carried out on a Newport RS-2000 optical table with four I-2000 stabilizer vibration isolators. The environmental noise after the vibration isolation was approximately 10nm. Through a 12-bit D/A board, a computer generated a control command signal of a voltage range 0-10 volts. A PI E505 piezoelectric drive amplifier was used to generate a drive voltage of 0-100V to be applied to the piezoelectric translator. The average output power of the drive amplifier is 30W and the average output current is 300mA. The peak output current (<5ms) is 2000mA.The drive was selected to match the demand for the dynamic operation of the PI P-

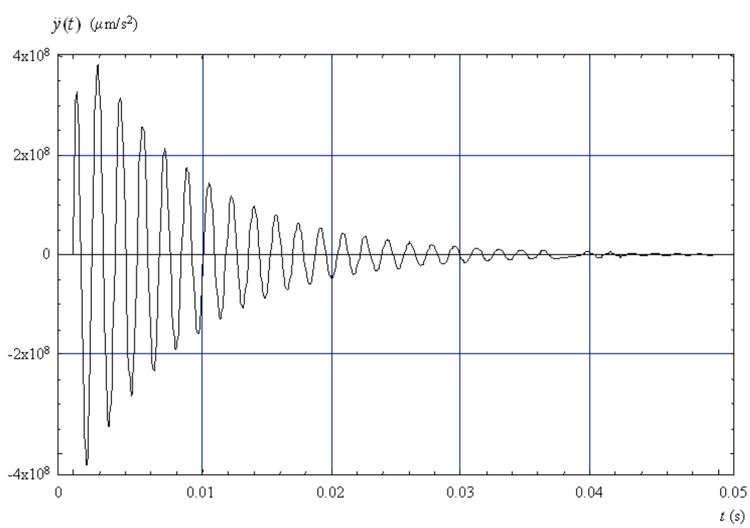

Fig. 4 Theoretical acceleration to the step command signal

840 piezoelectric translator with a current coefficient of 15μA/Hzμm.

From the tests, the table was found to have a resolution of 10nm after the noise effect is removed, a closed loop stiffness of approximately 400N/μm, a natural frequency of approximately 579.2Hz (Fig. 1), and a rising time of 400μs. Due to the high response speed to the step command, the acceleration was found to be up to 337914.3mm/s^2 that is 34.48 times of the gravitational acceleration (Fig. 3). Using Eq. (2), the theoretical acceleration is obtained (Fig. 4), where the theoretical maximum acceleration is found to be approximately 3.5x10^{8}μm/s^2. This result is quite close to the experimental result shown in Fig. 3, except the distortion at the bottom of the cycles due to the effect of detachment that is modeled separately, thus validates the dynamics model in Eq. (2).

Conclusions

A new workpiece micro-positioning table is developed to offer improvements in dynamic responses that are required for many high speed machining and motion applications. The piezoelectric translator is simplified as a spring and mass system driven by a piezoelectric force. A dynamic model is established to describe the acceleration of the table. The acceleration is mainly affected by the time constant of the drive amplifier and piezoelectric translator system, the damping factor of the spring and mass system in the workpiece table, and the input command signal to the drive. The model is validated by the experimental tests in which the maximum acceleration was many times of the gravitational acceleration.

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