Meccanica (2009) 44: 457–464
DOI 10.1007/s11012-008-9187-6

Reference model control for improving motion accuracy of a micro lathe

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Received: 9 October 2007 / Accepted: 1 December 2008 / Published online: 19 December 2008

Abstract A two-axis micro-lathe was developed in 1996 at the former Mechanical Engineering Laboratory in Tsukuba, Japan, for machining of small pieces as a part of the Micro-factory concept. Each axis is driven by a set of built-in PZT actuators. The work presented here concerns the design and implementation of a reference model control algorithm to improve the motion accuracy of the micro-lathe. The motion control was originally assured by a PI action considering dead zone. The performances of the reference model control were experimentally tested and compared with those obtained with PI control for four common trajectories used in manufacturing machines.

Keywords Micro-lathe · PZT actuator · Accuracy · Reference model control · Micro-machines

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1 Introduction

Manufacture industries are facing a new kind of challenges related with environmental issues, agile and flexible manufacturing, as well as fast integration of technological evolutions. Although machined parts or products are getting smaller and smaller, machine tools used to manufacture still in conventional size. Downsizing of manufacturing systems may lead to smart solutions, improving space use, reducing energy consumption, including environmental conditioning such as temperature, humidity and cleanliness, as well as facility investment [1].

A micro-machine research group in the former Mechanical Engineering Laboratory (MEL) in Tsukuba, Japan, has proposed a concept of the micro-factory in 1990. That group estimated that in the case of a $1/10$ size-reduction of production machines, the total energy consumption in the factory decreases to approximately $1/100$ of that of a conventional factory. In such case, some small-size products will be fabricated by on-site manufacturing system at storefront, office or home. To implement the micro-factory, MEL later designed and established a machining micro-factory, which consisted of a micro-lathe and various other small size machining tools, conveyors and assembly devices [2]. As a part of the micro-factory concept [3], a two-axis micro-lathe was developed in 1996 at MEL for the machining of small pieces. Each axis is driven by a set of built-in PZT actuators. In its more recent version, the micro-lathe is equipped with a closed-loop numerical controller using linear micro-encoders as feedback sensors. A very detailed mechanical description of the micro-lathe can be found in [3–5].

The first approach to achieve micro displacements consisted of micro-linear stages based on micro-servo actuators with feed screw mechanisms and linear guides [6]; this approach shows an accuracy of 10–30 µm. The work presented in [7] shows that FTS (Fine Tool Servo) is able to compensate the micro waviness error that is caused by the x-axis translational slide of a miniature ultra-precision lathe. In that case, the lack of accuracy is mainly due to backlash of mechanical parts.

The second approach proposes the use of PZT actuators to achieve micro displacements. However, PZT actuators have two disadvantages when used in positioning devices: (1) Travel is reduced to few microns despite a high resolution. A micro positioning work piece table using three PZT translating actuators [8] has a maximum displacement of 11.8 µm and a resolution of 6 nm. Commonly, that kind of mechanical design uses capacitive sensors for feedback [9]; (2) Nonlinear behaviour of PZT actuators due to hysteresis, which needs a very precise mathematical model [10]. To overcome the reduced travel, some positioning devices use the 'inchworm' principle [11] based on three active PZT elements within the inchworm: two 'clamps' and one 'pusher'. Repetitively advancing and clamping, the pushing element achieve large displacements. Although each pusher step is small, approximately 10 µm, if the step rate is high enough, substantial speeds may be obtained. A micro electro discharge machine with an inchworm type of micro feed mechanism has been presented in [12]; its accuracy is restricted within 2 µm.

In the revised micro-lathe presented here, unlike the conventional inchworm mechanism, the micro-slide uses only one PZT actuator for clamping and one for pushing. In the stationary state, during the step motion cycles, the main stage is held between countering guide-ways by friction. In the feed phase, the feed actuator breaks the friction, and pushes the main slide forward referred to the clamping stage. The motion control of the micro-lathe was originally assured by a PI (Proportional-Integral) action considering a dead-zone. As it is well known, sometimes a single PI action is not the best solution to get a very accurate motion control for both, positioning and tracking tasks. The PI is a linear control, suitable for systems modelled by a transfer function with constant coefficients; proportional action (P) reduces the response time and integral action (I) decreases the steady state error. However, the PZT actuators obviously have a non-linear hysteresis property, which reduces the stability of the control system. This dynamic property of such actuator is similar with that of a "spring-mass-damper" system, which can be regarded as a second-order segment. Since PZT has a high response velocity, under-damped oscillation will take place. In this case, control PI can increase oscillations in the steady state response and the gains should be adjusted for different system regimes. To solve this problem, a more suitable controller is required to minimize the effects of changes in the transfer function coefficients and maintain sufficient stability.

The Reference Model (RM) control [13] has the advantage to fit the overall performance of the system to a pure second-order system, allowing a direct way to predetermine the transient behaviour [14], the steady state performance [15], and the oscillations level, even for systems with complex dynamics such as robots [16]. The RM control, combined with a real time identification process, has the advantage of enduring changes in the parameters of the system. Such capacity has turned the RM control on a tool used frequently in adaptive control algorithms, called MRAC (Model Reference Adaptive Control) [17–20], where the control parameters can be updated to change the response of the system. The work presented in this paper concerns the design and implementation of a RM control algorithm to improve the motion accuracy of the micro-sliders.

2 Revised micro-lathe with NC

2.1 Mechanical configuration

The lathe is 32.0 mm long, 25.0 mm wide and 30.5 mm height, and weighs only 100 g (Fig. 1). The spindle is directly driven by a 1.2 W d.c. coreless motor up to 15,000 RPM; it is mounted on the orthogonal stacked micro-sliders. The displacement of each slide is detected by newly developed embedded linear micro-encoder with 62.5 nm resolution. Its mechanical configuration provides with guiding-and-hold rigidity since there is no backlash between sliders

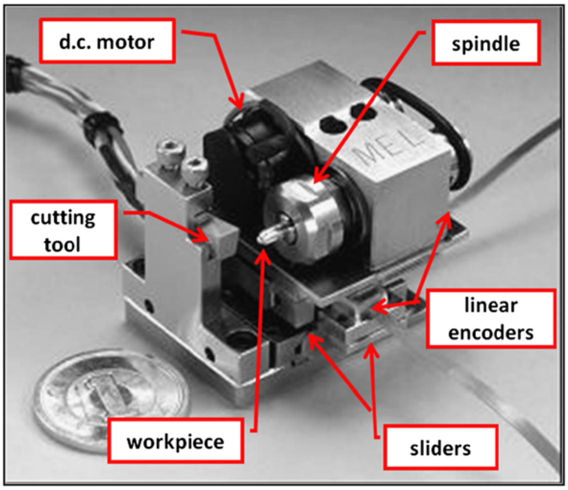

Fig. 1 Revised micro-lathe with NC

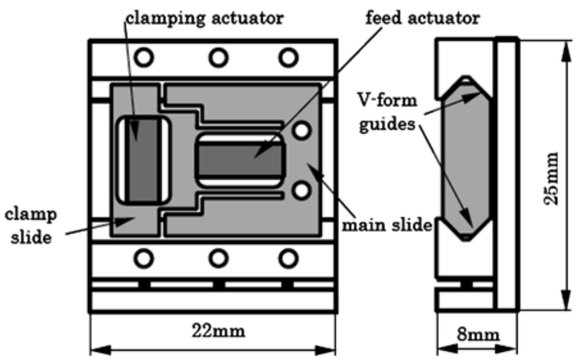

Fig. 2 The step-feed configuration

and guides. The thrust force was adjusted to about 3 N using screws to fix the guides. Total occupying desktop space including a laptop computer is about 550×450 mm.

2.2 Slide motion

The micro-lathe uses a pair of micro-sliders based on a unique step-feed configuration driven by two PZT actuators (Fig. 2). The stage moves like an inchworm so that it realizes both fine positioning and long travel. The sequence of the stepwise motion is optimized to get feed motion as smooth as possible, and continuously repeated at 200 Hz, when the feed rate reaches 0.4 mm/s.

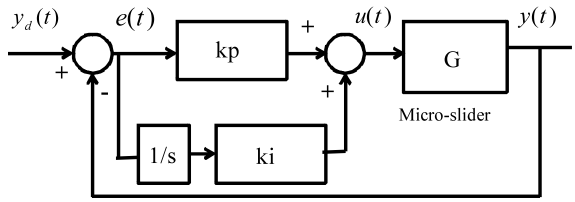

Fig. 3 Block diagram of PI control

2.3 Motion controller

The displacement of each slide is feedback to the servo controller based on a single board laptop computer. A numerical controller processes part programs and feeds the servo controller with a command pulse train. The servo cycle is 5 ms and the system resolution is 0.1 µm. The control PI (Proportional-Integral), implemented using the NC, is described in the block diagram in Fig. 3.

Where,

$y_d(t)$: Input, desired position,
$u(t)$: Command to the micro-slider,
$y(t)$: Output, current position,
$e(t)$: Following error,
G: Transfer function describing the micro-slider.

The PI transfer function is given by,

$$\frac{U(s)}{E(s)} = kp\left(1 + \frac{ki}{s}\right). \tag{1}$$

The kp coefficient represents the amount of corrective action that is applied for a given error. Power is applied in direct proportion to the current measured error, in the correct sense so as to tend to reduce the error. At low kp values, only a small corrective action is applied when errors are detected; errors will remain uncorrected for relatively long periods of time. If kp is increased, the system becomes more responsive and the error decreases quickly, but if kp increases too much an oscillating behaviour appears. The ki coefficient magnifies the effect of long-term steady-state errors, applying ever-increasing power until they reduce to zero. As the kp coefficient, if ki increases too much, oscillations are produced on the system response. A more detailed description of the feed mechanism, feedback device and control parameters is presented in [4].

2.4 Machined samples

The micro-lathe can cut brass with an accuracy of 1.5 µm roughness in the feed direction and 2.5 µm

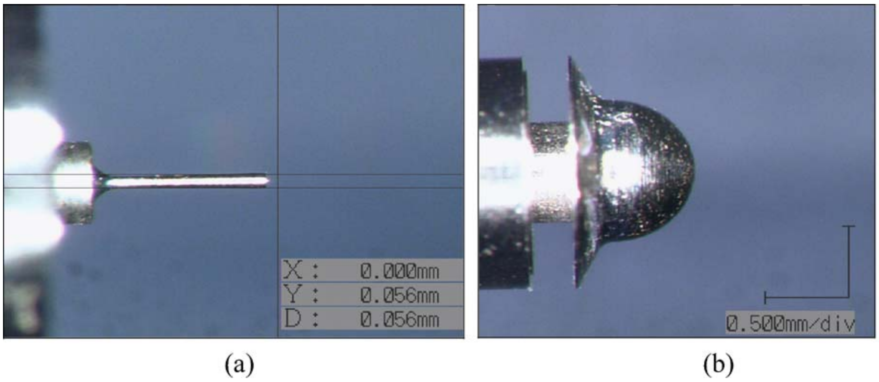

Fig. 4 Machined samples, (**a**) Machined needle, (**b**) Machined "micro-hat"

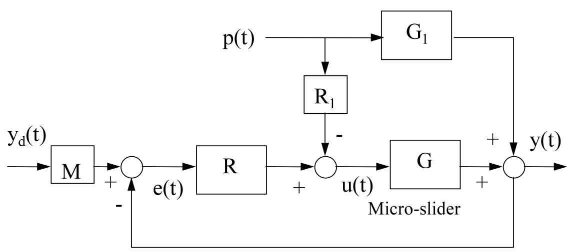

Fig. 5 General diagram of a single-input RM control

roundness. Work pieces made of brass of 2 mm diameter were machined using a diamond tool, Fig. 4.

3 Reference model control

The motion control was originally assured by a PI action considering non-linear hysteresis of PZT actuators. As is well known, sometimes a single PI action is not the best solution to get a very accurate motion control for both positioning and tracking tasks. De Larminant and Thomas proposed the RM control some years ago [13]. At that time, it was assumed that a real plant should be well enough identified to get good experimental results in terms of stability. However, the experimental result presented in this paper demonstrates that only an estimation of plant parameters is sufficient to get very acceptable performances. The general diagram of the single-input RM control for the micro-slider is shown in Fig. 5.

Where,

$p(t)$: Deterministic perturbation,

G, G_1: Transfer functions describing the micro-slider,

R, R_1, M: Transfer functions of the controllers.

The goal of the RM control is to find the controllers R, R_1, M that impose a desired behaviour to the whole system. The desired behaviour is mathematically described by two conditions (expressed in the Laplace domain, using letter s as complex variable),

$$\frac{Y(s)}{P_1(s)} = 0, \quad \text{and} \quad \frac{Y(s)}{Y_d(s)} = H_d(s). \tag{2}$$

The first condition ensures the perturbation rejection feature, the term $Y/P_1 = 0$ means that the form of the controller transfer function should cancel (zero value) the effect of the disturbance P_1 on the output Y. The second condition forces the closed-loop response to follow a behaviour described by transfer function $H_d(s)$. From the block diagram in Fig. 5,

$$Y = G_1 P_1 + G\big[R(MY_d - Y) - R_1 P_1\big]. \tag{3}$$

Then,

$$Y = \frac{G_1 - GR_1}{1 + GR} P_1 + \frac{GR}{1 + GR} MY_d. \tag{4}$$

Considering only motion control, without machining, the perturbation rejection term can be neglected. In such case, only the second condition was considered. Assuming $M = 1$,

$$R = \frac{H_d}{G(1 - H_d)}. \tag{5}$$

If $G(s)$ is known in advance, a convenient choice of $H_d(s)$ can be found using Graham and Lathrop polynomials [13]. They found the polynomials coefficients by simulating the transfer functions, with zero steady-state error, which minimize the criterion,

$$J = \int_0^\infty t|e(t)|dt. \tag{6}$$

The following transfer functions are issued from first, second and third order polynomials,

$$\text{First order:} \quad H_{d1}(s) = \frac{\omega_n}{s+\omega_n}, \tag{7}$$

$$\text{Second order:} \quad H_{d2}(s) = \frac{\omega_n^2}{s^2+1.4\omega_n s+\omega_n^2}, \tag{8}$$

Third order:

$$H_{d3}(s) = \frac{\omega_n^3}{s^3+1.75\omega_n s^2+2.15\omega_n^2 s+\omega_n^3}. \tag{9}$$

The numerical value of ω_n is selected depending on the desired response time, at least theoretically speaking, since a mechanical system has its particular response time. Actually, for practical purposes, ω_n is the only gain to be experimentally tuned. Since the micro-slider has been considered as a servomechanism, we are assuming that the stage model $G(s)$ is represented by a second order linear transfer function,

$$G(s) = \frac{Y(s)}{U(s)} = \frac{1}{\alpha_1 s^2+\alpha_s s}. \tag{10}$$

This model simplifies the parameter identification process. Then, a logical choice for $H_d(s)$ will be $H_{d2}(s)$, that leads to,

$$R(s) = \frac{\omega_n^2(\alpha_1 s+\alpha_2)}{s+1.4\omega_n}. \tag{11}$$

Using a discrete differential form, (11) can be written as follows,

$$y(k) = -d_1 y(k-1) + d_2 u(k) + d_3 u(k-1). \tag{12}$$

With, $d_1 = 1.4\omega_n\Delta - 1, d_2 = \alpha_1\omega_n^2, d_3 = \alpha_2\omega_n^2\Delta - \alpha_1\omega_n^2$ where Δ is the sampling time, equal to 5 ms. Using the Least Squares Identification Method, the estimated coefficients of the micro-slider transfer function were $\alpha_1 = 6.67e-6, \alpha_2 = 0.0033$, considering μm as units of the output signal.

4 Results

Four different desired trajectories, commonly used for machining applications, were considered during the experimental implementation of the RM control to the real system: (a) Linear trajectory at constant velocity, (b) Sinusoidal trajectory, (c) Step input, (d) Circular trajectory in the X-Y plane. The best performances were obtained by setting $\omega_n = 320$. For each type of trajectory, the performances obtained with the RM control were compared with those obtained with the PI control.

The first was a linear trajectory at constant velocity equal to 100 μm/s with a change of direction at 1.25 seconds. The micro-slider output position and following error obtained with the PI control are shown in Fig. 6a, the response obtained with the RM control is shown in Fig. 6b. The RM control strongly reduces the

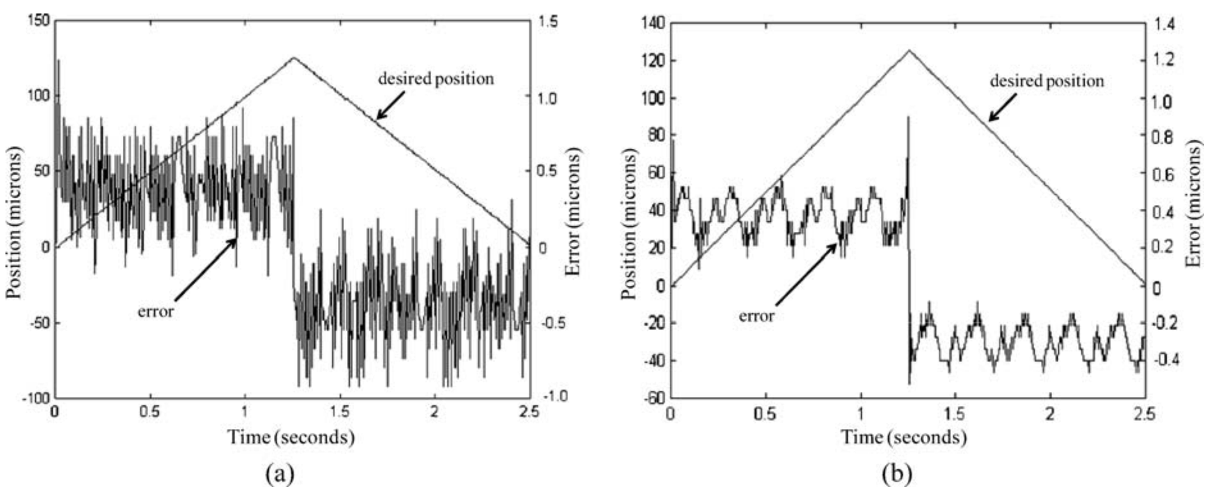

Fig. 6 Response and error to a linear trajectory with: (**a**) PI control, (**b**) RM control

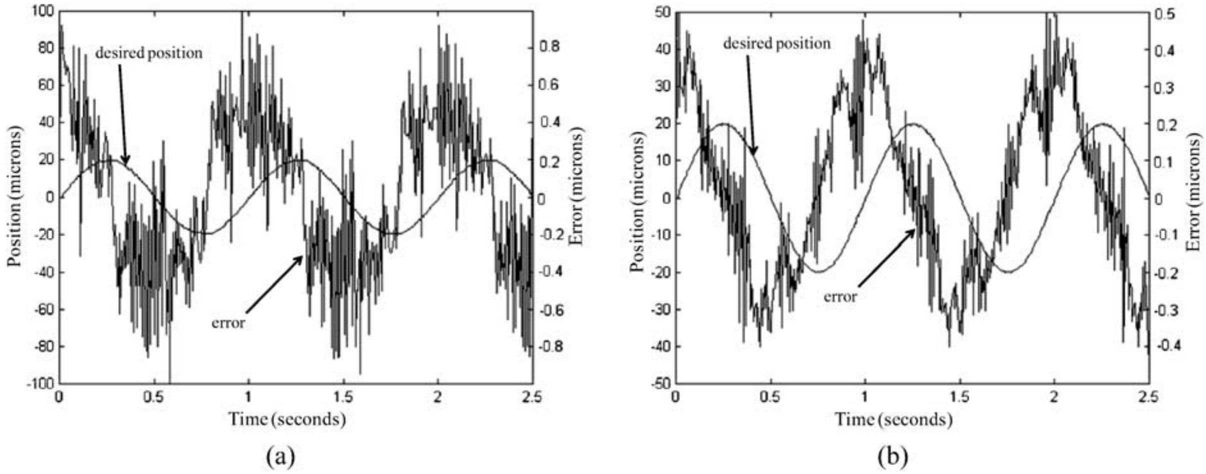

Fig. 7 Response and error to a sinusoidal input with: (**a**) PI control, (**b**) RM control

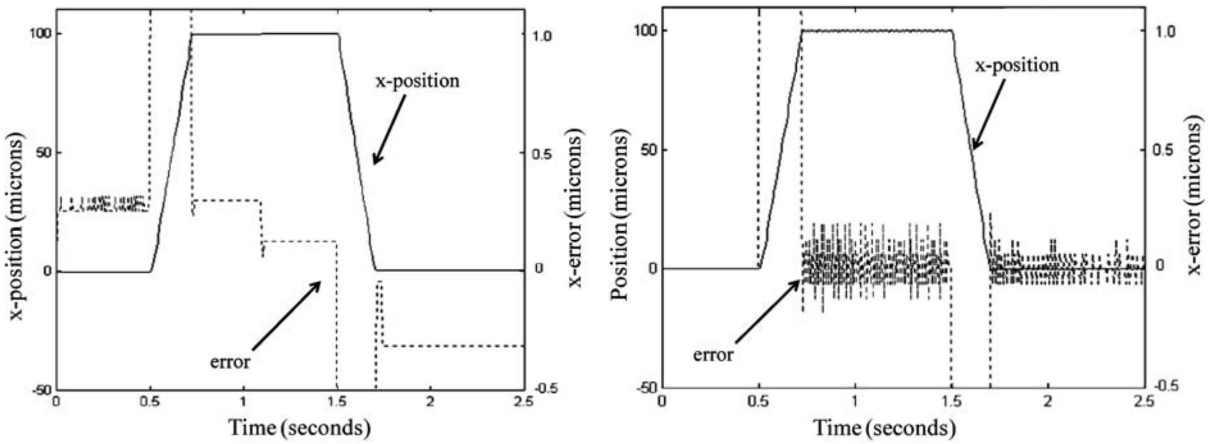

Fig. 8 Response to step input with: (**a**) PI control, (**b**) RM control

oscillations observed with the PI control, due to integral action. The following error decreases from 0.9 µm with PI control to less than 0.5 µm with RM control.

The second trajectory considered to test the control performances was a sinusoidal signal of 20 µm of amplitude and 1 second period. The response obtained with the PI control is shown in Fig. 7a and with the RM control in Fig. 7b.

Again, oscillation decreases with RM control as well as the following error that was less than 0.5 µm.

The third trajectory was a 100 µm step input. The responses are shown in Fig. 8. Unlike the response for tracking trajectories, it can be observed that RM control increases oscillations in the steady state. However, the following error remains around zero (less than 0.1 µm). On the other hand, the PI control keeps a larger steady state error close to 0.3 µm.

As is common for testing accuracy of machine tools, the circular test was also considered. The combined response for both micro-sliders, in the X-Y plane, is shown in Fig. 9 for tracking a circle of 20 µm radius. The RM control presents a better fitting to the circular trajectory.

5 Conclusion

The motion control accuracy of the micro-lathe was improved using the RM control instead of the classic PI control, in spite of an approximate mathematical

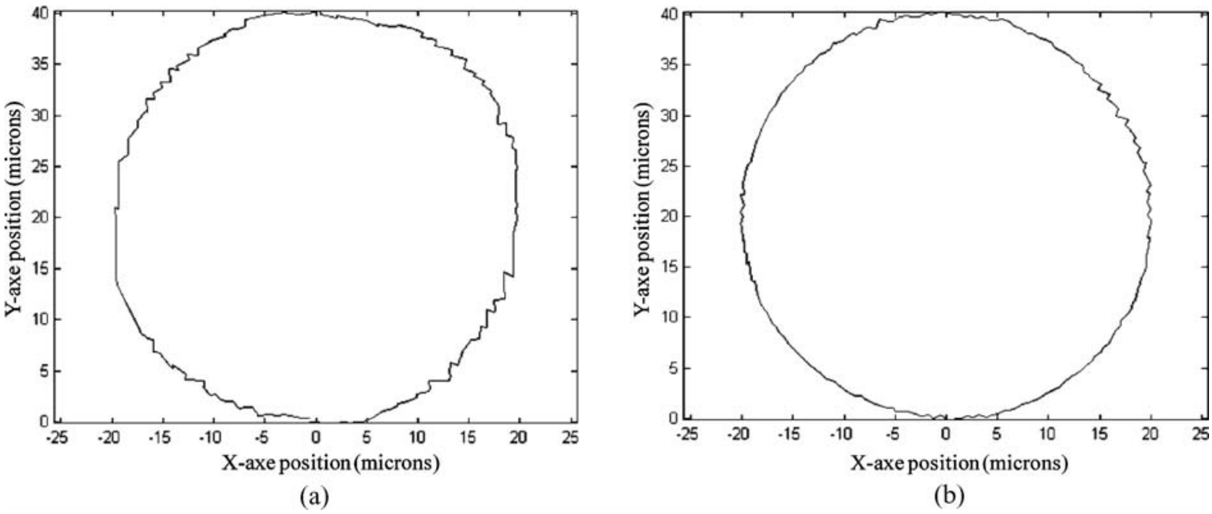

Fig. 9 Circular test with: (**a**) PI control, (**b**) RM control

model of the system is considered. The experimental result shows that the RM control is much more robust to system uncertainties as is assumed theoretically. To complete this study, the RM control will be evaluated under machining conditions performing a real turning task experiment. In such case, a perturbation-rejection term should be included, and then the controller R_1 must be considered.

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