Available online at www.sciencedirect.com

SciVerse ScienceDirect

Procedia CIRP 1 (2012) 605 – 608

www.elsevier.com/locate/procedia

5th CIRP Conference on High Performance Cutting 2012

Improving motion accuracy of tool center point using model-reference feedforward controller

Kotaro Nagaoka[a] and Atsushi Matsubara[b]*

[a]Advanced Technology R&D Center, Mitsubishi Electric Corporation, 8-1-1 Tsukaguchi-honmachi, Amagasaki, Hyogo 661-8661, Japan
[b]Department of Micro Engineering, Kyoto University, Yoshida-honmachi, Sakyo-ku, Kyoto 606-8501, Japan
* Corresponding author. Tel.: +81-6-6497-7621; fax: +81-6-6497-7285.E-mail address: Nagaoka.Kotaro@da.MitsubishiElectric.co.jp.

Abstract

To achieve high-precision machining with NC machine tools, their feed drive systems must minimize position errors at the tool center point. When acceleration and deceleration occur, such mechanical errors as elastic deformation of the machine and mechanical vibration primarily cause these position errors. In this study, we discuss a suitable compensator design for mechanical errors through case studies. The numerical simulation results indicate that an appropriate design of a dynamic model improves the motion accuracy of the tool center point.

Keywords: Computer numerical control (CNC); Servosystem; Modelling; Motion; Vibration; Measurement; Control

1. Introduction

Improvement of the motion accuracy by applying precise motion control of the feed drive systems is strongly needed in order to achieve high-precision machining with NC machine tools. Although the NC controller carries out servo control so as to make the detected position measured by a position detector such as a linear scale track the position command precisely, the tool center position has a deviation from the position command because the detected position differs from the position of the center of the real tool due to mechanical errors. The errors consist of static errors and dynamic errors. The dynamic errors such as elastic deformation of the machine and mechanical vibration become major components when the velocity of which the machine changes transiently.

To increase response of the servo control and obtain highly accurate motion accuracy of the NC machine tools, a model-reference feedforward controller [1] has been introduced [2]. The controller consists of a reference model and compensators for mechanical dynamics. The reference model has a function that provides a nominal response to the position command. The compensators for mechanical dynamics create feedforward control signal so as to make the tool center point track to the nominal response. Using this controller, the tool center point tracks to the position command precisely. However, a mechanical model is necessary to design the controller.

In this study, measurement and modeling of the dynamic characteristics of the mechanical system are conducted to obtain a relationship among the driving force, the feedback position, and the tool center position. Then, the compensators for mechanical dynamics are designed based on the obtained mechanical model. The relationship between the design of the compensator and the accuracy of the motion trajectory is investigated in particular detail through some case studies. Numerical simulations are carried out to show that the proposed controller design makes a contribution to improvement of the motion accuracy of the tool center point.

2. Measurement of mechanical characteristics

2.1. Measurement system

Figure 1 illustrates a measurement system that analyzes the mechanical errors by using synchronized measurement of the driving force of the motor (f_m), the

 http://dx.doi.org/10.1016/j.procir.2012.04.107

tool center position (x_{tcp}) and the feedback position (x_{fb}) [3]. In this system, the feedback position is detected by a linear scale, and the tool center position is measured by a grid encoder. In addition, the driving force is calculated by the motor current. A signal for identification, for example a swept sinusoidal wave, is given to the servo system before the identification is carried out. Then, a frequency response from the driving force to the feedback position and that from the feedback position to the tool center position are calculated.

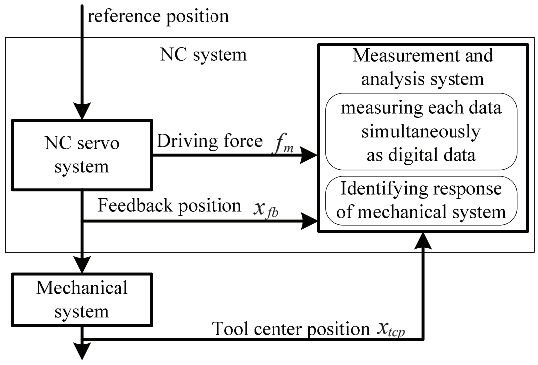

Fig. 1. Measurement system

2.2. Machine tool used in measurement experiment

Table 1 shows major specifications of the small vertical machining center used in the case study. The tool center position is measured by a grid encoder KGM182.

Table 1. Specifications of machine tool

Axis	X	Y
Travel	230 mm	220 mm
Drive type	Linear motor	
Rated force	400 N	
Guide way	Linear (ball)	
Control resolution	1 nm	
Feedback resolution	0.1 nm	

2.3. Frequency response of the machine

The frequency response from the driving force to the feedback velocity (i.e. derivative of the feedback position) and that from the feedback position to the tool center position appear in Figures 2 and 3, respectively. The driving force is obtained from the motor current by multiplying a force constant.

In these figures, solid lines represent frequency responses calculated by the periodgram method [4] from measurement results. Dashed lines represent frequency responses of the estimated linear model calculated so as to fit the measured responses. The estimated models are written as transfer function which has 5 pairs of conjugate poles and 5 pairs of conjugate roots, respectively.

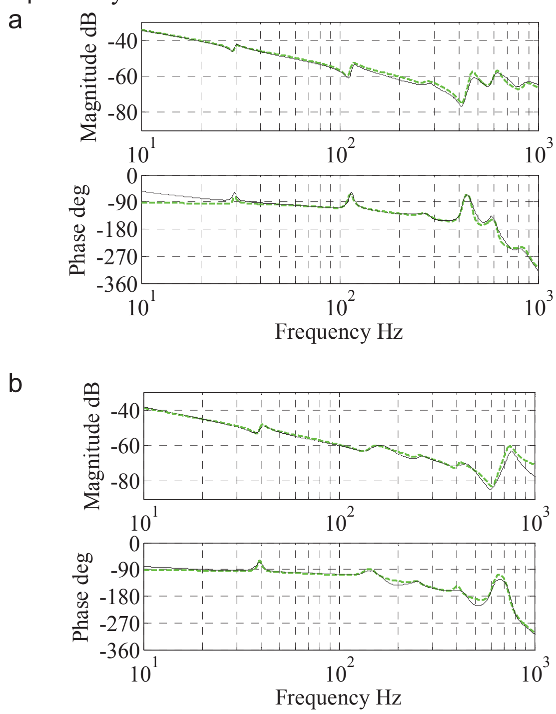

Fig. 2. Frequency response from force to feedback velocity (solid: measured, dashed:estimated) : (a) X axis; (b) Y axis

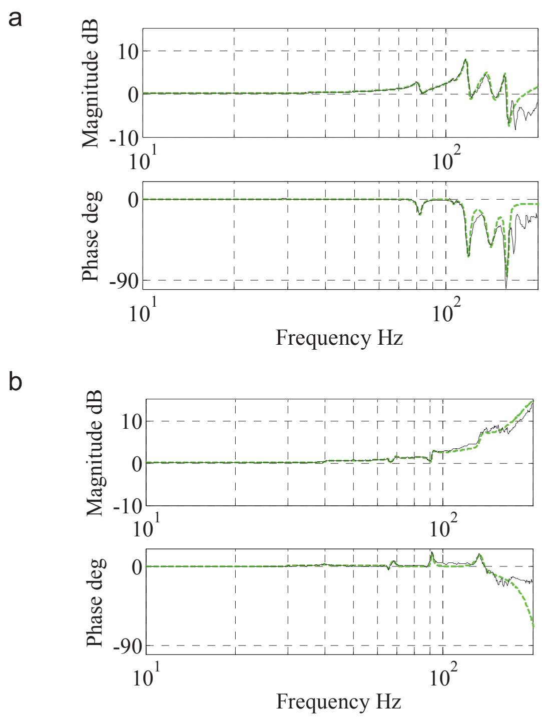

Fig. 3. Frequency response from feedback position to tool center position(solid: measured, dashed:estimated) : (a) X axis; (b) Y axis

2.4. Verification of the estimated model

To verify the validity of the modelling of the mechanical system, an experiment of measurement of the trajectory is carried out using the machine introduced in Table 1. A square-shaped path shown in Fig. 4 is chosen as the tested pattern of the trajectory since dynamic motion errors such as vibration are likely to occur at the corner of the trajectory. In the experiment, the feedrate is set to 5000mm/min. At each corner, the federate is temporarily lowered in order to maintain path accuracy at the corner edges.

A measured trajectory using the machine and simulated trajectory using the estimated mechanical model are shown in Fig. 5 and Fig. 6, respectively. In each figure, dashed line denotes the reference position, thick solid line denotes the feedback position, and thin solid line denotes the position of the tool center point. The graphs are magnified around the right-top corner P_2 of Fig. 4. It is observed that vibration amplitude (peak-to-peak value) of the trajectory of the tool center point is up to 1.0 μm. The error between feedback position and tool center point is up to 0.2μm. The tendency of the measured trajectory nearly corresponds to that of the simulated trajectory. However, there exists difference in high frequency component of these trajectories. It requires further investigation in the future.

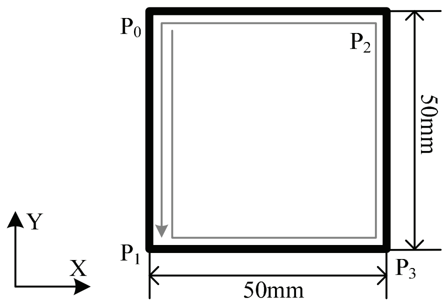

Fig. 4. Tested trajectory

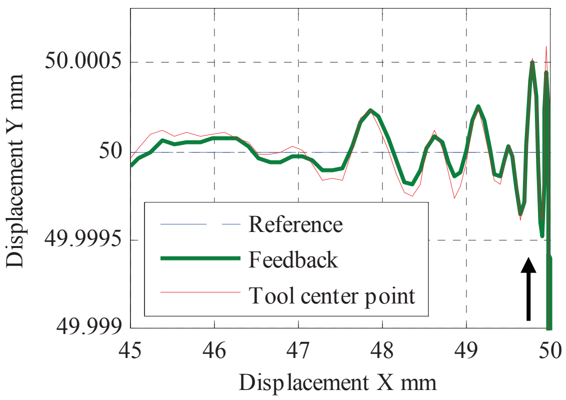

Fig. 5. Measured trajectory

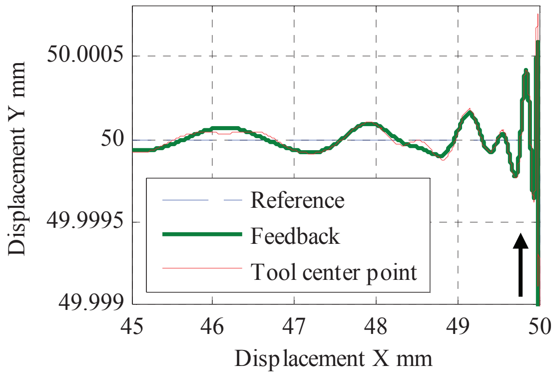

Fig. 6. Simulated trajectory

3. Controller design and trajectory error analysis

3.1. Design of the proposed controller

A model reference feedforward (MR-FF) controller is a controller based on the model of the mechanical system. An appropriate design of nominal models of the mechanical system used in the controller is necessary to achieve precise control of the tool center point.

Figure 7 shows a block diagram of the MR-FF controller. In this figure, x_r is a reference position, $G_r(s)$ is a reference model, $C_p(s)$ is a position loop controller, $C_v(s)$ is a velocity loop controller, $G_{fd}(s)$ is a transfer function from the driving force to the feedback position, and $G_{dt}(s)$ is a transfer function from the feedback position to the tool center point. $G_{c1}(s)$, $G_{c2}(s)$, and $G_{c3}(s)$ are feedforward compensators calculated by the mechanical model. According to a previous report [5], the tool center position becomes equivalent to the output of the reference model if the feedforward compensators are designed as follows:

$$G_{c1}(s) = G_{dtm}^{-1}(s) \tag{1}$$

$$G_{c2}(s) = sG_{dtm}^{-1}(s) \tag{2}$$

$$G_{c3}(s) = G_{fdm}^{-1}(s)G_{dtm}^{-1}(s) \tag{3}$$

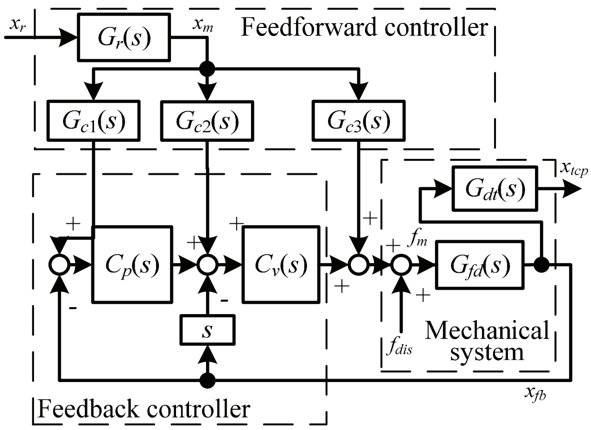

Fig. 7. Structure of model reference controller

where $G_{fdm}(s)$ and $G_{dtm}(s)$ are the nominal models of $G_{fd}(s)$ and $G_{dt}(s)$, respectively. The reference model is designed as 5th-order low-pass filter in order to achieve high tracking response.

3.2. Modeling of mechanical system

The following two ways of modeling are considered for a design of feedforward compensators.

(1) Complete modeling

The estimated mechanical model (multi-mass model) is thoroughly considered in constructing the compensators. In this case study, the nominal mechanical models $G_{fdm}(s)$ and $G_{dtm}(s)$ are set equal to the transfer functions of the estimated model shown in Fig. 2 and Fig. 3, respectively.

(2) Approximate modeling

The mechanical model is considered as two-mass model, the simplest model for a modeling of vibrations, during design of the compensators. In this case study, the transfer function $G_{fdm}(s)$ is set to the transfer function which has one vibration pole and one vibration root. The pole and roots is calculated by the lowest vibration mode of the estimated model. The transfer function $G_{dtm}(s)$ is set to 1.

3.3. Evaluation of trajectory errors

Influence of the design of the feedforward compensator to the trajectory of the tool center point is investigated through numerical simulations. The reference trajectory used in the simulation is configured as the same as in the experiment described in 2.4.

Graphs shown in Fig. 8 are simulation results of the trajectory of the feedback position and the tool center point. In the case where the feedforward compensator is designed using the complete modeling depicted in Fig. 8 (a), the trajectory of the tool center point perfectly tracks the reference. Meanwhile, a fluctuation of the feedback trajectory of 0.1 μm amplitude is observed. On the other hand, in the case where the compensators are designed with the approximate modeling, it remains 0.2μm amplitude of vibration in the trajectory of the tool center position illustrated as Fig. 8 (b).

Though the vibration in the trajectory of the tool center point is effectively suppressed when the complete modeling is applied, thorough implementation of the feedforward controller using the high-order model is difficult to achieve because resources of servo-control processors are limited. On the contrary, the approximated two-mass model is easy to implement and it can adequately suppress the vibration of the tool center point. The effectiveness of these modeling methods is planned to be verified through measurement experiments at the next step.

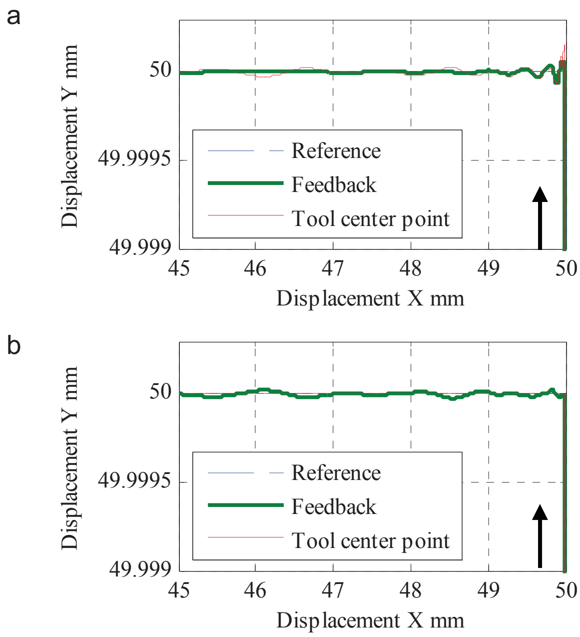

Fig. 8. Simulation result: (a) Complete modeling; (b) Approximate modeling

4. Conclusion

We designed a feedforward compensator to suppress the dynamic motion errors that appear in the trajectory of the tool center point by using the model of the mechanical system of the NC machine tools. The complete multi-mass model and approximated two-mass model are compared as the model for designing the feedforward compensator through numerical simulations. It is revealed that the vibration of the trajectory is completely suppressed by using the feedforward compensator based on the complete modeling and effectively suppressed by using the feedforward compensator based on the approximate two-mass mechanical model.

References

[1] Koyama M, Yano M. Two degrees of freedom speed controller using reference system model for motor drives. Proc 4th European Power Electronics Conf 1991; 60–65.

[2] Nagaoka K. Control Method of Motion Error Compensation for NC Machine Tools. Intl J Automation Technology 2009; 3/3, 292–297.

[3] Nagaoka K, Matsubara A, Fujita T, Sato T. Measurement and Analysis of Motion Accuracy Using NC System with Synchronized Acquisition of External Sensor Data. Intl J of Automation Technology 2008; 3/4, 394–400.

[4] Ljung L. System Identification:Theory for the User. Englewood Cliffs, New Jersey: P T R Prentice Hall; 1987.

[5] Matsubara A, Nagaoka K, Fujita T. Model-reference feedforward controller design for high-accuracy contouring control of machine tool axes. CIRP Annals 2011; 60/1, 415–418.