A servo parameter tuning method for high-speed NC machine tools based on contouring error measurement

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Abstract

In the latest high-speed and high-acceleration NC machine tools, the structural vibration is one of the most critical factors to deteriorate the machine's contouring performance. Particularly on such a machine, the parameters in a CNC servo control system must be carefully tuned, since too high response of the latest CNC units often causes severe structural vibration. This paper presents a practical servo tuning method for high-speed machine tools to optimize its contouring accuracies. In order to reduce the structural vibration with the minimum sacrifice of control bandwidth, the tuning is based on iterative measurement and simulation of the machine's contouring performance. A case study shows that a proper tuning of servo parameters significantly reduces the structural vibration and improves the machine's overall contouring accuracy.

1 Introduction

Recent technological development has commercialized high-speed, high-acceleration machining centres of the feedrate up to 60 m/min, and the acceleration rate up to 1G. In particular, the commercialization of high response CNC servo control systems in the 1990s significantly improved the overall bandwidth of the closed-loop position control system. A majority of high-speed feed drives

in today's commercial NC machine tools are driven by a servo motor and a ball screw. In such a feed drive system, the torque is transmitted to a table via a coupling and a ball screw. Therefore, the dynamics of feed drive system includes internal vibration modes due to the linear and torsional vibration of the ball screw. A special attention must be paid to the control of such vibration modes particularly on high-speed machines. High-speed feed drives typically adopt a roller guideway, which generally exhibits lower damping compared to slide guideways. A ball screw of higher lead also causes lower damping.

A structural vibration is also a critical issue on typical high-speed machines. A high-speed, high-acceleration feed drive naturally imposes a severer impact force on the mechanical structure, which causes the structural vibration of lower frequency, and larger amplitude. This issue becomes more critical on a linear motor driven feed drive. Since it is a direct drive with no transmission mechanism, its driving force is directly transmitted to the mechanical structure. In today's market, the majority of servo motor driven feed drives in machining centres still adopts the "semi-closed loop" control (i.e. the angular position of a servo motor is feedbacked for the position control). On the other hand, a linear motor driven feed drive system must directly feedback the linear position of a table. Therefore, the dynamics of structural vibration directly affects the dynamics of the position closed-loop system. When servo parameters are not properly tuned, it may even cause the instability of the closed-loop system [1]. The structural vibration becomes particularly a critical issue on a large-size machine tool, where the mass of the driven part is generally heavier and/or the travel range is longer.

Conventionally, the gains of position and velocity controllers are set as high as possible, under the condition that the stability (and robustness) of the closed-loop system is secured with some stability margins [2]. On the latest high-speed machines, however, it is often the case that feedback gains must be lowered to reduce the motion error caused by the structural vibration. On some large-sized machines, the maximum acceleration is set lower than the potential capacity of a servo motor, in order to secure required motion accuracies.

This paper presents a servo parameter tuning methodology based on the measurement of the machine's two-dimensional contouring performance by using the cross grid encoder method, or the KGM (Kreuz Gitter Meßsystem in German) method, developed by Heidenhain GmbH [3]. Since the KGM method is non-contact optical measurement, it is more suitable for high-speed and high-accuracy measurement. More importantly, unlike the DBB (Double Ball Bar) method [4] that is restricted to a circular test, it can measure the machine's two-dimensional contouring error on an arbitrary geometry. The background of the research and the basic concept of the tuning methodology were discussed in our previous report [5]. The objective of our research is to propose a practical tool for servo engineers to tune the parameters in a servo control system that is already implemented in a commercial CNC unit.

The remainder of the paper is organized as follows. First, Section 2 presents an overview of the proposed tuning method, with a quick review on the KGM method. Section 3 presents a simulation model of a feed drive system. Section 4 presents the tuning procedure of servo parameters that have particularly a critical
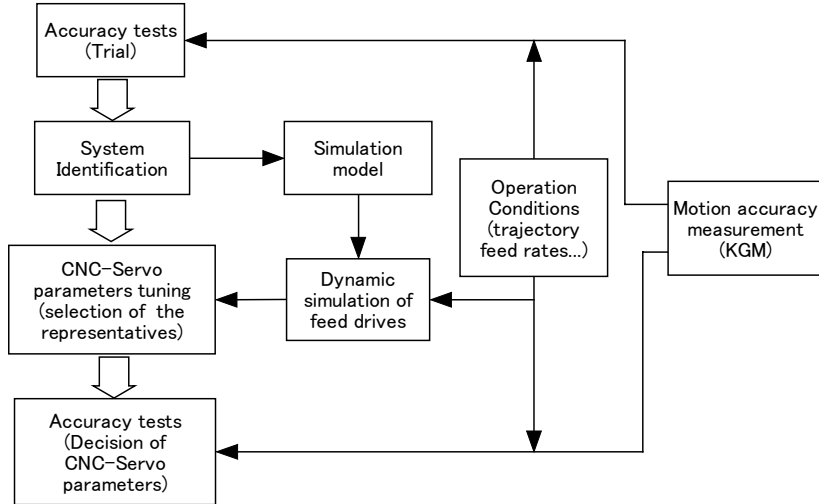
Figure 1: Tuning procedure

2 An Overview of Tuning Method

Figure 1 illustrates the tuning procedure presented in this paper. The tuning is based on actual measurement of the machine's two-dimensional contouring performance. It should be emphasized that we do not intend to fully automate the tuning procedure. A model-based simulation of the contouring performance, which will be discussed in Section 3 in details, facilitates manual try-and-error tuning procedure by a servo engineer, although it does not completely eliminates his/her decision making.

By using the default set of servo parameters, the machine's contouring accuracy is measured. A test path is given such that each possible source of motion errors can be easily diagnosed. Based on the result of this first trial, the mechanical parameters included in the simulation model are identified. By using the numerical simulation based on this simulation model, candidates for the best combination of CNC servo parameters are selected. The machine's contouring accuracy is again measured with these candidates, and the best combination of servo parameters are chosen.

The cross grid encoder method [4], or the KGM method, is employed for a two-dimensional motion accuracy test in arbitrary shape contouring. On the scale plate, the grating of the interferometer's glass encoders is crossed perpendicularly so that the two-dimensional position of the scanning head can be measured by
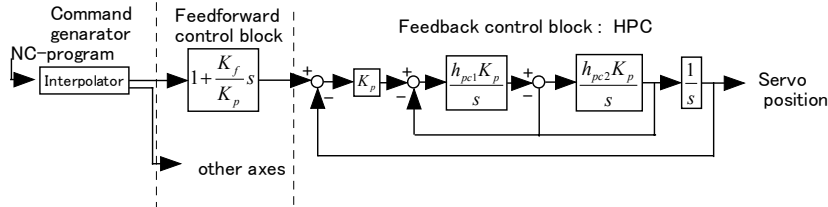
Figure 2: A simplified CNC model of a feed drive system

using the principle of diffracted light. More details about the KGM method can be found in [6]. An advantage of the KGM device is its ability to measure any move within a range of the scale, just as a two-dimensional coordinate measuring machine.

3 A Dynamic Model of a Feed Drive System in an NC Machine Tool

3.1 A CNC model

The tuning method proposed in this paper is based on numerical simulation of the machine's two dimensional contouring performance. To this purpose, we need a simulation model of the dynamics of the entire feed drive system, including both servo control systems and mechanical dynamics.

In this paper, we consider a CNC unit of the dynamics shown in Figure 2. This model assumes the following:

(1) The command generator is exactly the same as the one used in the actual CNC unit. Most commercial CNC units in today's market support an S-curve velocity profile. It can be represented by the combination of two filters, and has two parameters to be tuned.
Typical commercial CNC units employ two ways to distribute a velocity command profile to each axis; the pre-interpolation and post-interpolation acceleration controls. In the post-interpolation acceleration control, a velocity command is distributed to each axis and then is filtered independently. On the contrary, in the pre-interpolation acceleration control, it is first filtered and then distributed to each axis. The pre-interpolation acceleration control must be used for high-accuracy contouring. Our simulation model assumes the pre-interpolation acceleration control.

(2) A feedforward (FF) controller of the first order is assumed. Most commercial CNC units employs a velocity FF controller to obtain faster response while securing the stability of the feedback loops. In Figure 2, K_f represents the feedforward gain [%].

(3) The transfer function of the velocity and current control loop is regarded as ideal (=1) in the simulation model, since the bandwidth of these loops is in general sufficiently large compared to that of the position loop. In other words, we only consider the position loop dynamics in the simulation model.

(4) Many commercial CNC units in today's market implement a higher-order position control loop block to improve the response of the closed-loop system. For example, Takesita et al. [7] presented that the radius reduction in circular interpolation can be improved by using a high order position control loop. In this paper, we consider a second order filter implemented in the position loop, called the high precision control (HPC) filter. In Figure 2, h_{pc1} and h_{pc2} represents HPC gain coefficients. K_p represents the position loop gain [rad/sec].

The model shown in Figure 2 represents a typical CNC unit available in today's market. However, a different simulation model must be used for a different CNC unit.

3.2 A dynamic model of structural vibration

As overviewed in Section 1, a high-speed and high-acceleration feed drive is often subject to a coupled vibration of a foundation, a bed, and a column (such a vibration is referred to as the base vibration in this paper). While the linear and torsional vibrations of a ball screw typically have a frequency above 100 Hz, the base vibration typically appears at a lower frequency, 10~50Hz. Therefore, it has a more direct effect on the machine's contouring accuracy.

Figure 3 shows a dynamic model of a feed drive system including its base vibration dynamics [8], where: x_t: position of table [m], M_t: mass of the table [Kg], K_t: stiffness of the table [N/m], C_t: damping coefficient of guideway [Ns/m], J_m: inertia of the motor [Kgms^2], R: transformation coefficient ($R = p/(2\pi)$, where p is the ballscrew pitch), θ_m: angular position of the motor [rad], T: torque of the motor [Nm], x_b: position of the bed [m], M_b: mass of the bed [Kg], K_b: stiffness of the bed [N/m], C_b: damping coefficient of the bed [Ns/m].

. This model considers a feed drive driven by a rotary servo motor and a ball screw. The dynamics of the ball screw system can be modelled as a two-mass dynamic system with the natural frequency of 100~200Hz (only the longitudinal vibration mode is considered) [2]. The bed is assumed to be a solid body. The bed is usually supported by levelling blocks or stud bolts. Their dynamics is simply modelled by a spring and a damper. The equations of motion can be written as follows:

$$M_t \ddot{x}_t + C_t \dot{x}_t + K_t(x_t - x_b) = K_t x_r$$
$$M_b \ddot{x}_b + C_b \dot{x}_b + K_b x_b + K_t(x_b - x_t) = -K_t x_r$$

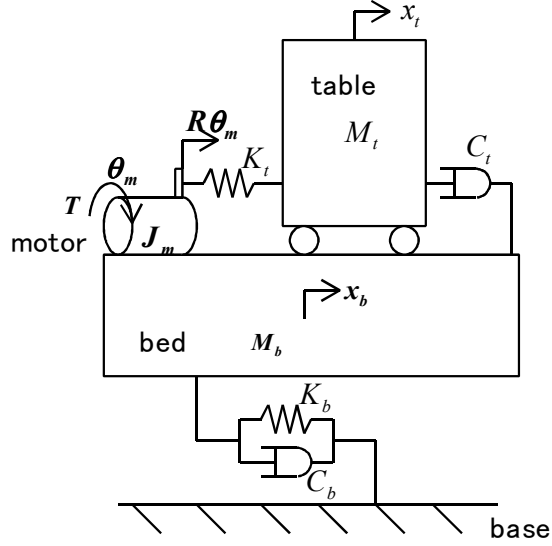where $x_r \equiv R\theta_r$. It can be rewritten as:

Figure 3: Dynamic model of feed drive system

$$[M][\ddot{X}] + [C][\dot{X}] + [K][X] = [F] \qquad (1)$$

where

$$[X] = [x_t \quad x_b]^T,$$

$$[M] = \begin{bmatrix} M_t & 0 \\ 0 & M_b \end{bmatrix}, \qquad [C] = \begin{bmatrix} C_t & 0 \\ 0 & C_b \end{bmatrix},$$

$$[K] = \begin{bmatrix} K_t & -K_t \\ -K_t & K_b + K_t \end{bmatrix}, \qquad [F] = \begin{bmatrix} K_t x_r \\ -K_t x_r \end{bmatrix}$$

Figure 4 shows the overall block diagram of the simulation model, with the servo controller part and the mechanical part combined.

4 A Tuning Procedure of Servo Parameters

4.1 Servo parameters to be tuned

As servo parameters that have particularly a critical effect on the machine's contouring accuracy, this paper presents a tuning procedure for the following five parameters: 1) an acceleration time (the first-order time constant for linear acceleration and deceleration), 2) a position loop gain, K_p, 3) a time constant of a
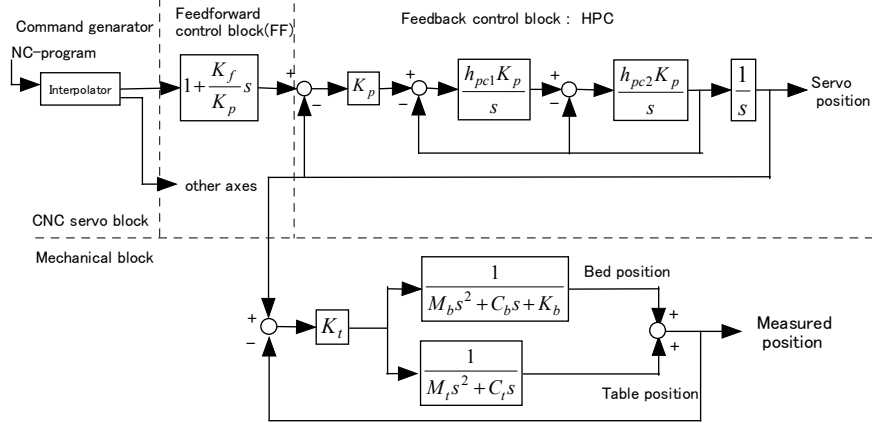
Figure 4: Simulation model

smoothing filter on the reference trajectory, 4) a feedforward controller gain, K_f, and 5) a corner velocity. Their tuning is performed in the same order.

4.2 Tuning procedure

(1) An acceleration time: a shorter acceleration time is favourable as long as it does not induce the structural vibration. By using numerical simulation based on the model identified in the trial measurement, the acceleration time must be determined so that 1) an overshoot error is smaller than the prescribed tolerance in one axis operation, and 2) a relative error between two axes is smaller than the prescribed tolerance in two dimensional contouring.

(2) A position loop gain: a higher position loop gain is favourable for disturbance rejection, although too high a gain may induce the vibration and even make the closed-loop system unstable. The position loop gain, K_p, on each axis is tuned such that the gain of the open-loop transfer function of the position feedback loop becomes –6dB at the mechanical resonance frequency.

(3) A time constant of a smoothing filter on the reference trajectory: the smoothing filter must be designed such that it cancels the mechanical resonance. Its time constant must be set at the inverse of the resonance frequency.

(4) The feedforward (FF) controller gain: it is well known that the phase delay of the feedback loop causes the radius reduction in a circular operation. By setting the feedforward gain properly, the radius reduction can be prevented. When a third-order position loop controller is used, it is theoretically shown by Takeshita et al. [8] that there will be ideally no radius reduction by setting the FF gain at $K_f=0.5$. In practice, however, due to the unmodelled dynamics such as the time delay and round-off errors in servo control system, this ideal value may not minimize the radius reduction.

Set the FF gain at K_{f1} and K_{f2}, and measure the radius reduction in a circular interpolation with the reference radius of r_0, and the angular velocity of ω. Suppose that the measured radii are r_1 and r_2, respectively. Suppose that the transfer function of the position closed-loop system (the trans-

fer function from the output of the FF block to the table position) is given by $G(s)$. Then, the following equations hold:

$$\left(1+\omega^2 K_{f1}\right)\left|G(j\omega)\right|^2 = r_1^2 / r_0^2$$

$$\left(1+\omega^2 K_{f2}\right)\left|G(j\omega)\right|^2 = r_2^2 / r_0^2 \qquad (2)$$

$$\left(1+\omega^2 K_f\right)\left|G(j\omega)\right|^2 = 1$$

where $K_f{}^*$ is the optimal FF gain, which makes the radius $r = r_0$. By cancelling ω and $G(j\omega)$ from the equations above, the optimal FF gain, $K_f{}^*$, can be derived as follows:

$$K_f{}^* = \sqrt{\frac{K_{f1}^2\left(r_0^2 - r_2^2\right) - K_{f2}^2\left(r_0^2 - r_1^2\right)}{r_1^2 - r_2^2}} \qquad (3)$$

(5) A corner velocity: a higher corner velocity is favourable from the viewpoint of shortening cycle time, but if it is too high, it easily causes the structural vibration at the corners. By increasing it from a small value on the simulation, one can find the maximum value that does not induce the structural vibration above the tolerable level.

5 A Case Study

5.1 Machine specifications

As a case study, the proposed tuning method is applied to a small-sized vertical type machining centre, Machine A. Its major specifications are shown in Table 1.

5.2 Tuning Results

A test trajectory to measure the machine's motion accuracy is designed as shown in Figure 5 [12]. It has straight lines, a sharp corner and a right angle corner where the mechanical vibration is likely induced, a dull corner and arcs to check the smoothness. Figure 5 (a) shows a contouring error profile measured by the KGM method when the default set of servo parameters, shown in Table 2 ("be-

Table 1: Major specifications of machine A.

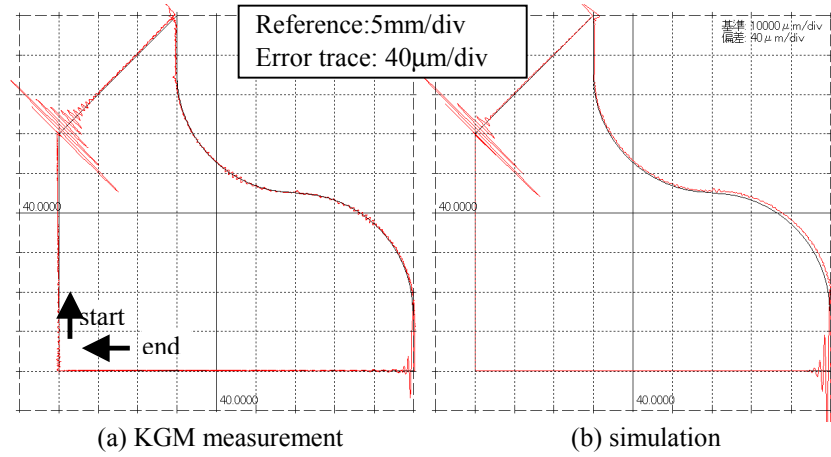|	
Stroke [mm]	X:560, Y:430, Z:350
Maximum feedrate [mm/min]	20,000
Maximum acceleration [G]	0.9 G (X,Y)
Diameter and lead of ball screw [mm]	32, 16
Type of table guideway	Linear ball guideway
Servo motor power [KW]	1.5 (X, Y)
Type of position control	Full-closed control
Spindle speed [min^{-1}]	12000

(a) KGM measurement   (b) simulation

Figure 5: Motion error traces before tuning

Reference:5mm/div
Error trace: 40μm/div

基準 10000μm/div
偏差 40μm/div

40.0000

40.0000

40.0000

40.0000

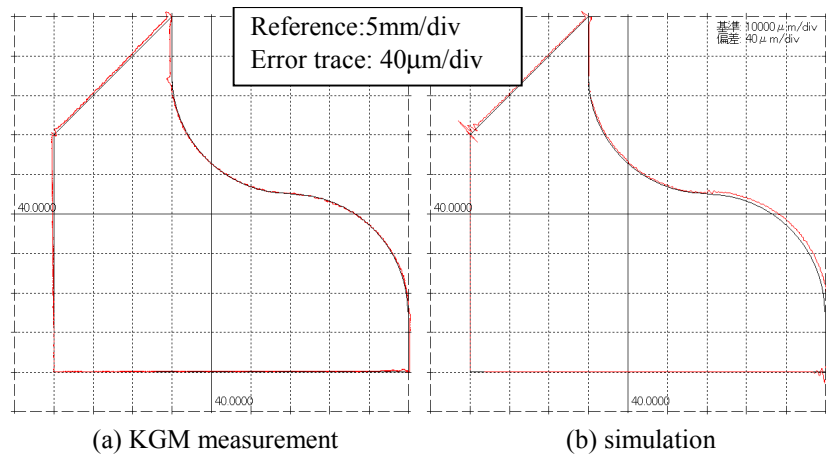(a) KGM measurement   (b) simulation

Figure 6: Motion error traces after tuning

fore tuning"), is used. This combination of parameters is too "aggressive," which resulted in the induction of the mechanical vibration of the frequency about 40Hz in Y axis direction at a corner.

The simulation parameters are identified such that the simulated trajectory coincides well with the measured trajectory. The simulation result, shown in Figure 5(b), shows a good agreement with Figure 5 (a). Based on the identified simulation model, the servo parameters are tuned by the procedure outlined in Section 4. A simulated error trajectory after the tuning is shown in Figure 6 (b). Then, the KGM measurement is again performed to validate the effectiveness of the tuning (Figure 6(a)). Table 2 shows the servo parameters before and after the tuning.

6 Conclusion

Table 2: The servo parameters before and after the tuning

Parameter	Before tuning	After tuning
Acceleration time [ms]	10	50
Position loop gain [rad/sec]	100	48
Time constant of the smoothing filter [ms]	3.5	28.4
FF gain K_f [%]	51	51
Corner velocity [mm/min]	600	240
Feedrate [mm/min]	3,000	

In this paper, a practical tool is proposed for servo engineers to tune the parameters in a servo control system for an NC machine tool. A simple dynamic model of a feed drive system, which contains the base dynamics to simulate structural vibration, is identified by using a trial test of contouring error measurement. The tuning is based on iterative simulation and measurement. A case study is conducted to investigate the practical applicability and effectiveness of the proposed tuning method on a small-sized vertical type machining centre.

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