System Modeling and Identification of a Precision Fast Tool Servo System

Guang Lu[1], James Cuttino[2], Dale Schinstock[3], and Tim Haskew[4]
[1]University of California at Berkeley, [2]University of North Carolina at Charlotte
[3]University of Tulsa, [4]The University of Alabama

I. INTRODUCTION

Conventional Diamond Turning Machines (DTMs) are limited to rotationally symmetric surfaces due to the low bandwidth of their slide systems. By adding Fast Tool Servo (FTS) systems, DTMs can produce rotationally non-symmetric surfaces without lowering the turning speed. This improves their machining capabilities, and gives engineers many more choices of design. Most of previous fast tool servos used piezoelectric (PZT) actuators, which have high stiffness and relatively low moving mass. The drawbacks to using piezoelectric actuators include the limited range of motion, heat problems, and hysteresis.

At The University of Alabama, a long-range fast tool servo using a voice coil actuator has been developed. Voice coil actuators have a low armature mass and can produce high accelerations while operating at high frequencies. They generate a long range of motion with high resolution and bandwidth. The problems associated with voice coil actuators include considerable heat loss and low structural stiffness.

The system components of this fast tool servo were designed concurrently. Principal system parameters were defined at the beginning of the research. In order to verify the system parameters for the real system, system identification and modeling techniques were conducted to ensure the system would perform as it was predicted. These techniques were significant for the successful development of the FTS system.

II. SYSTEM CONFIGURATION

For precision PID control considerations, the bandwidth of the control loop was designed to be over 3 kHz. In order to separate the dynamics of the current loop from the control loop, the bandwidth of the current loop was designed to be over 30 kHz. Similarly, the bandwidth of each of the components in the current loop must be over 300 kHz in order not to have significant effects on the current loop. The mechanical system was expected to be a second-order system with its mode at 50 Hz. These defined the principal system parameters of the fast tool servo. Successful implement of these parameters was significant to the development of the system.

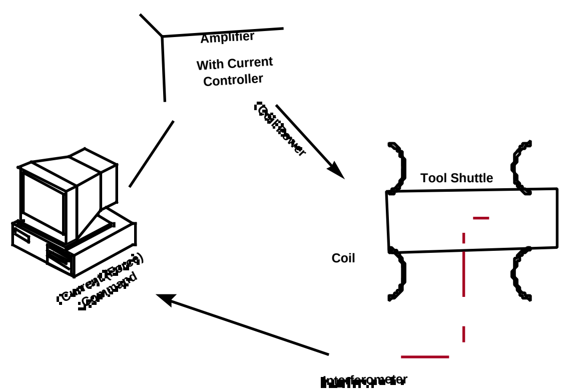

Fig.1 The FTS built at University of Alabama

The FTS system is shown in Fig.1. A flexure system was built as the principal part of the mechanical system. The controller built in the PC sends out current commands to an amplifier, and the amplifier drives the voice coil actuator to move the diamond tool on the shuttle that is suspended by the flexure bearings. A laser interferometer sensor detects the actual position of the shuttle and feeds it back to the controller.

III. SYSTEM COMPONENTS

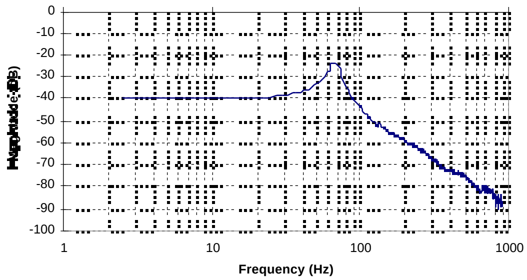

Fig.2 The magnitude plot of the frequency response of the flexure system

The dynamic performance of each of system components has been individually verified in system identification and analyzed in system modeling to ensure that each of them satisfies the system requirements.

A. Flexure System

The flexure system in the FTS, is a monolithic structure fabricated from one piece of 316 stainless steel. Four curved flexures connect the moving part in the middle, the shuttle, to the fixed part that is attached to the base. The shuttle is used to hold the diamond tool and is fixed to the coil in the actuator. The monolithic design helps to ensure alignment and material uniformity, and it also makes the stiffness in all flexures the same to reduce the error motions caused by unbalanced stiffness. Compared to conventional flexures that are usually flat, curved flexures provide higher stiffness, lower stress, and occupy more compact space.

The mechanical system was designed and modeled to be a second-order system. The transfer function of this system is $\frac{z(s)}{F(s)} = \frac{1}{ms^2 + bs + k}$. This model was verified using finite element analysis method.

Dynamic testing using impact method was conducted to verify this model and identify the dynamic parameters of the flexure system, which were important to the controller design. The resonance frequency was found to be 66.8 Hz. The frequency response of the flexure system is shown in Fig.2 and Fig.3. However, after the coil in the actuator was attached to the flexure system, the resonance of the system was later verified to be at 48.5 Hz.

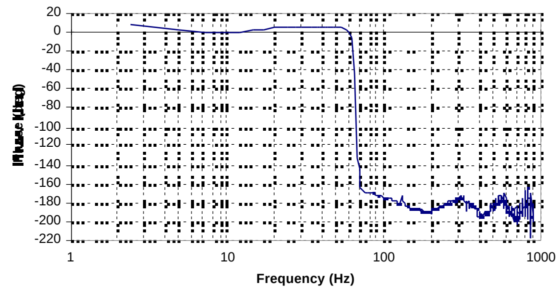

Fig.3 The phase plot of the frequency response of the flexure system

B. Voice Coil Actuator

The voice coil actuator is composed of a coil assembly and a permanent magnet. The coil assembly is suspended in the magnetic field by the flexure system. The displacement of the coil in the fixed magnetic field is proportional to the current through the coil in a limited range of motion. Voice coil actuators have high bandwidth and resolution. The linear actuator used in FTS is a BEI linear voice coil actuator, model number LA25-42-000A.

In system design, the electrical circuit in the voice coil actuator was modeled as a first-order dynamic system. Its transfer function is $\frac{1}{Ls+R}$.

C. Amplifier

Until recently, linear operational amplifiers were the only considerations for high bandwidth amplifiers. However, they were limited in current and power at high frequencies. Compared to the linear amplifiers, PWM (Pulse-Width Modulation) types are much more efficient, typically more than 95% efficiency. But one of the major disadvantages of PWM amplifiers is that their bandwidths are restricted by the switching frequencies. However, recently developed PWM amplifiers switch at much higher frequencies, which makes PWM amplifiers with broad bandwidths available.

A closed loop PWM amplifier was built with an APEX SA02 PWM amplifier chip. The SA02 is an 80 volt, 250 kHz switching PWM amplifier. The full bridge output circuit can provide 10 Amps of continuous drive current. The pole in the voice coil, $Z = R/L$, is detrimental to the system performance since it results in a phase lag. Therefore, a PI controller is used to cancel it by placing a zero at R/L. It increased the system bandwidth and improved dynamics of the current loop. Two power resistors were used to measure the currents in two opposite directions in the H bridge in the PWM as the feedback. The frequency response of the system is shown in Fig.4 and Fig.5. These figures show that the bandwidth of the amplifier is over 30 kHz.

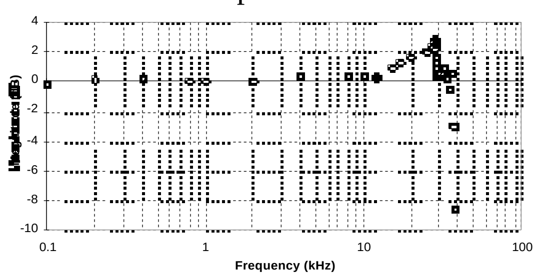

Fig.4 Magnitude plot of the frequency response of the PWM amplifier

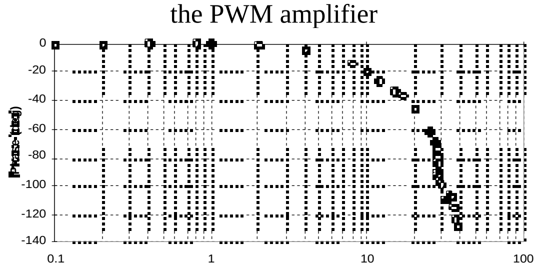

Fig.5 Phase plot of the frequency response of the PWM amplifier

D. Controller

The mechanical system of the FTS, including the tool shuttle and the coil assembly, was assumed to be a spring-mass-damper model. Its transfer function is $\frac{z}{F} = \frac{1}{ms^2 + bs + k}$. Note that the mass includes the mass of the shuttle in the flexure system and the mass of the coil assembly. Using the identified parameters m, b, and k, the theoretical frequency response was characterized and computed. It shows that the system (including mass of the shuttle and coil) has a resonance at 48.5 Hz, while the resonance in the flexure system alone was verified during dynamic testing to be at 66.8 Hz. The additional mass from the coil assembly lowered the resonance frequency in the mechanical system.

A PID controller using a pole-zero cancellation method was developed to provide an initial benchmark with which other controllers could be compared. The system bandwidth was chosen to be 4 kHz, and the zeroes were used to cancel the poles in the mechanical system. The theoretical error of this controller design is 1.25% at 50 Hz. The PID controller was implemented using an MC8-DSP-ISA board made by Precision MicroDynamics Inc. This board contains an Analog Devices 2106X family DSP processor. The standard configuration is a 40MHz ADSP-21061. The ADSP-21060 SHARC (Super Harvard Architecture Computer) is a single-chip, 32-bit computer optimized for signal computing applications. A DSP program has been built to run the control loop with the sampling frequency of 40 kHz.

IV. SIMULATION

Simulation was used to facilitate the concurrent design process of the system. It indicated the predicted adequacy of the controller, the amplifier, and the other system components and pointed out potential problems and sources of errors. Corresponding to the system identification and modeling process of the system components, the simulation models and parameters were made close to the real system, and further simulation was conducted to predict the system performance based on the identified system components.

The simulation of the FTS system was conducted in SIMULINK. The inputs provided to the system were sinusoidal signals to represent the position commands. The commanded position was a 50 Hz sinusoid with the amplitude of 0.5 mm. Fig.6 shows the error between the simulated position command and the actual position. The peak-valley amplitude of the position error is 11.6 micrometers, representing a 1.16 % error. This error is close to what was predicted in the controller design.

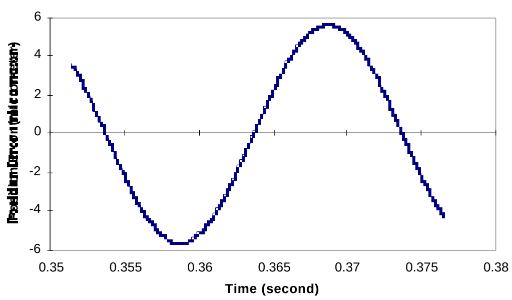

Fig.6 Simulated position error

V. TESTING

The DSP program generates sinusoidal commands at a sampling frequency of 40 kHz. In this GUI program, users can easily change the frequencies and magnitudes of the position commands. Users can also change the PID gains (K_p, K_i, K_d). The DSP program generates current commands and outputs them to the amplifier through a DAC port of the board. When the amplifier drives the voice coil actuator to move the flexure, the DSP program reads the measured position from the laser interferometer through a digital input port. The measured current in the voice coil is input into the DSP board through an ADC port. Data was stored into the external memory of the DSP.

The position commands were sinusoidal signals with the magnitude of 0.5 mm at different frequencies to test the system's ability to operate in the desired frequency region. The position error is shown in Fig.7. In this figure, a 680 Hz component is superimposed on top of a 50 Hz component. The 50 Hz component was caused by the natural resonance of the mechanical system and its appearance was normal. However, the 680 Hz component was undesirable. It could be caused by the compliance in the fixture used to hold the flexure system or compliance in the flexures whose natural frequency was close to or higher than the bandwidth of the impact signal and thus was not identified in the system identification tests of the flexure system.

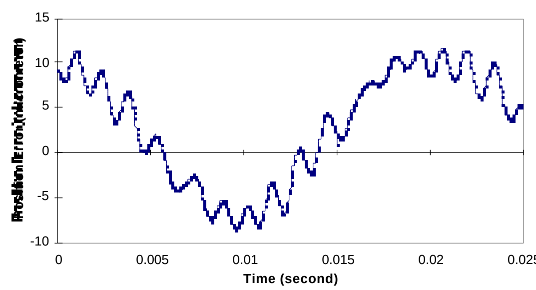

Fig.7 Position error for sinusoidal signal testing with magnitude of 0.5 mm at 50 Hz

VI. CONCLUSIONS

The developed fast tool servo can be controlled to move across $\pm$ 0.5 mm motion range with 10 μm position error at 50 Hz. The amplifier provides over 30 kHz bandwidth required by the system. The monolithic flexure system was designed and verified to be a second-order system with the resonance at 50 Hz. An unusual laser interferometer setup has been applied to this fast tool servo. The resolution of this interferometer is $\pm$ 5 nm. Only 5-10 nm noise is generated from the interferometer. These parameters satisfy the system requirements, and the real system performs close to the theoretical model. Research is still being conducted on the amplifier to reduce the noise and to solve the other problems that limit the system performance. A new flexure system is going to be designed to be able to move across a longer range of motion safely. Other control algorithms are going to be implemented to the FTS system to improve the system performance.