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Machine Tool Feed Drives and Their Control—A Survey of the State of the Art

Introduction

Machine tool feed drives control the positions and velocities of machine tool slides or axes in accordance with commands generated by CNC interpolators, and represent the lowest level of motion control hierarchy in machine tools. The requirements on feed drive performance include:

(i) Control over a wide range of speeds. Rapid traverse speeds in machining centers may be tens of m/min, whereas machining speeds in some precision machining applications may be a few mm/min.
(ii) Precise control of position, the desired accuracy of position control varying with the application. Precision of position control during machining affects accuracy of part dimensional control. The desired accuracy may be of the order of a few microns in normal machining operations or in the submicron range in precision machining operations.
(iii) Ability to withstand machining loads while maintaining accuracy of position control.
(iv) Rapid response of drive system to command inputs from the machine tool CNC system.
(v) Precise coordination of the control of multiple axes of the machine tool in contouring operations.

These requirements are best met by closed loop control of feed drives based upon real-time sensing of actuator position and velocity.

Demands on feed drive performance have become more stringent as machining technology has evolved to meet the requirements of a broader range of applications. The use of high spindle speeds for high speed machining has, for instance, necessitated high feed rates to maintain tool loading unchanged and realize productivity improvements. Precision machining applications such as diamond turning have similarly tightened positioning accuracy requirements on feed drives. Feed drive performance improvements have occurred as a result of progress in drive actuation and sensing, and drive control.

The state of the art of machine feed drive control is summarized here, with special emphasis on developments over the last decade and a half. Characteristic features of feed drive hardware are described first, along with the resulting dynamic models to be used in controller design. Feed drive control problems and their classification based on the requirements are discussed next. Recent control approaches and the feed drive control system performance achieved experimentally using these approaches are described next, together with discussion of relevant feed drive mechanical design features where appropriate. Finally, research trends and future areas of research are addressed briefly.

Feed Drive Hardware and Dynamic Models

Feed drive performance assessment and dynamic modeling must begin with consideration of the hardware including drive actuators and associated electronics, power transmission devices and mechanical support structures, and sensors providing information for drive control. Electrical actuation is predominantly used currently in machine tool feed drives. Rotary brushed or brushless DC motors, and rotary synchronous or asynchronous AC motors are used in conjunction with rotary-to-linear power transmission elements. Linear DC or AC motors eliminate the need for the linear power transmission elements (Slocum, 1992). While brushed DC motors retain a cost advantage over the other motors because of simpler associated electronics, they are subject to brush wear and require higher maintenance. Furthermore, brush friction in such motors is nonlinear and can limit control loop performance.

Power amplifiers for DC motors range from high-bandwidth continuous transistorized amplifiers and PWM amplifiers to lower-bandwidth thyristor amplifiers, depending on power level. AC synchronous and asynchronous motors are powered by AC line power, but rely upon sophisticated electronic control of inverters to achieve versatility of control of the currents in the individual stator phases, including control of frequency and amplitude. Smoothness of generated torque is an important performance measure for all the motor types and requires specific design features and electrical power control to minimize torque ripple (Bollinger et al., 1982; Slocum, 1992).

Linear power transmission elements are used in feed drives both to convert rotary actuator motion and to achieve a speed reduction. Ballscrew-nut transmission elements are most commonly used, and are capable of submicron-level motion repeatability. Backlash in such a transmission is reduced by preloading the nut, at the expense of increasing friction at the nut and lowering drive efficiency. The total axial stiffness in a ballscrew-nut transmission depends on the individual stiffnesses of the nut, leadscrew, bearings, and bearing supports, a stiffness range of 100–1,000 Newtons/micron and a feed drive resonant frequency range of 20–90 Hz being typical for medium duty machining centers (Gross, 1983; Weck, 1984). Traction drives are sometimes used in feed drives for high precision machines as they avoid backlash and friction problems. However, they have lower stiffness and lower mechanical advantage, which in turn limits drive force capability. Linear motors eliminate the need for linear power transmission elements and are not limited therefore by transmission inertia, resonance, backlash, and friction.

Feed drives involving high force levels, high bandwidths, and/or high accuracy have frequently employed other forms of

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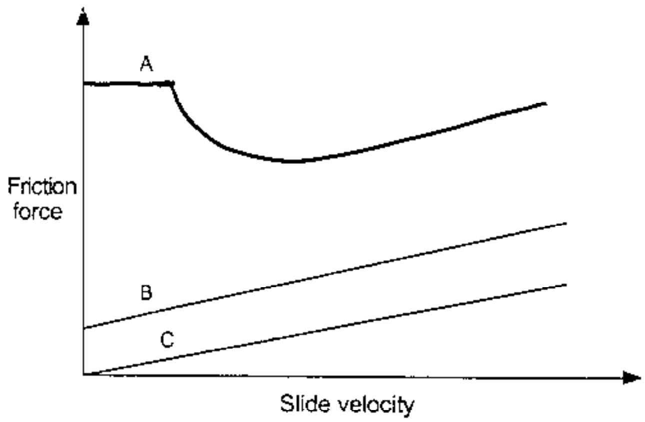

Fig. 1 Types of friction characteristics

actuation. Servovalve-controlled linear hydraulic actuators have been used to achieve stroke lengths in the mm range, high force levels and positioning resolution, along with stiffness due to fluid compressibility of upto 50 N/nanometer (Kanai et al., 1990). Piezoelectric actuation is feasible for stroke lengths of a few microns or tens of microns while retaining high actuator stiffness. Piezoelectric actuation has been used in diamond turning machines and conventional machine tools for correction of error motion due to spindle eccentricity (Patterson and Magrab, 1985) and other systematic machining errors (Kouno, 1984), in both cases the objective being fast and accurate positioning.

Feed drive performance is also affected by the friction characteristics of the bearings used to support elements of the drive (Slocum, 1992). Friction in feed drives limits achievable performance, though the specific mechanisms which are significant, and the resulting performance limitations, may vary from one machining application to another. A general model of sliding friction between lubricated surfaces is described by Armstrong-Helouvry et al. (1994), the time-independent aspects of the general model being shown in Fig. 1 as curve A and indicating the nonlinear nature of friction in sliding contact bearings. Friction in rolling contact bearings is modeled by curve B, the level of the static friction depending on the level of the preload. Hydrostatic bearings have little static friction, and are characterized by the friction characteristic, curve C.

Because of differences in the frictional and damping characteristics of actuators, power transmission components, and bearings, overall frictional characteristics for different feed drives could differ significantly. Moreover, the regions of the friction curves of importance vary with the application, the low velocity region being of importance for diamond turning operations, for instance, and the low and high velocity regions being of importance for high-speed contouring operations involving axis motion reversals.

The sensing options chosen for feed drive control affect achievable performance. Feedback control of position requires on-line sensing of feed drive position, either at the actuator or at the slide. The use of velocity feedback in addition to position feedback enables a greater variety of control objectives to be met, enhancement of damping being one example. Feedback of actuator angular position or velocity introduces a resonance/antiresonance pair inside the control loop in the presence of transmission resonance. Transmission related sources of position error would then be outside the control loop. Feedback of slide position brings these sources of error within the control loop.

Linear models of feed drive dynamics are useful for controller design, the appropriate level of model complexity depending on the application. The simplest model, and the one most often used, results from assuming electrical actuation, a fast current loop, and an infinitely stiff transmission. In this case, the DC motor feed drive model in Fig. 2 reduces to

$$\frac{X_p}{u}(s) = \frac{K_v}{s(\tau_v s + 1)} \tag{1}$$

where the gain K_v and the time constant τ_v are defined appropriately. Such a model may also be used to represent the dynamic behavior of a feed drive with a closed velocity loop, τ_v in this case representing the time constant of the closed velocity loop. More complex dynamic models for feed drives may be more appropriate in many instances, inclusion of transmission resonance and nonlinear friction, and power amplifier nonlinearities being examples of such refinements.

Discrete-time models of feed drive dynamics may be obtained in a number of ways. They may be determined analytically from the discrete-time equivalents of continuous-time models. For example,

$$\frac{X_p}{u}(z^{-1}) = Z\left\{\frac{1 - e^{-sT}}{s}\,\frac{X_p}{u}(s)\right\} \tag{2}$$

relates samples of the slide position x_p and the control input u, T being the sampling time, and it is assumed that a zero-order hold is used to generate values of u between sampling instants. Alternatively, discrete-time models may be determined empirically. In such a case, u represents an appropriate input variable. For example, Kulkarni and Srinivasan (1984) identified third-order discrete-time models for the velocity loop of a feed drive

Nomenclature

A, B^+, B^- = polynomials in z^{-1}
B_m = motor damping coefficient
B_s = slide damping coefficient
E_c = contour error orthogonal to commanded trajectory
E_n = tracking error component nearly orthogonal to contour error
E_x, E_y = x and y components of position error
e_a = motor armature voltage
e_b = motor back emf
e_r = radial error
F_c = cutting force opposed to direction of slide motion
F_{fs} = slide friction force
G_c = closed-loop feed drive transfer function
G_{ff} = feedforward transfer function
i_a = motor armature current
J = motor moment of inertia
K_E = motor back emf constant
K_t = current feedback gain
K_T = motor torque constant
K_x = axis loop gain
K_u = gain parameter
L_a = motor armature inductance
M = slide mass
N = transmission ratio; 2π/pitch for ballscrew, unity for direct drive
R = commanded trajectory radius
R_a = motor armature resistance
s = laplace variable
T = sampling interval
T_{fm} = motor friction torque
T_L = load torque on motor shaft
T_m = motor torque
u = control input
V = contouring speed
x_p = slide position
z = z transform variable
θ_m, $\dot{\theta}_m$ = motor angular position and velocity
ω = angular frequency
ω_n = undamped natural frequency
τ_x = axis time constant

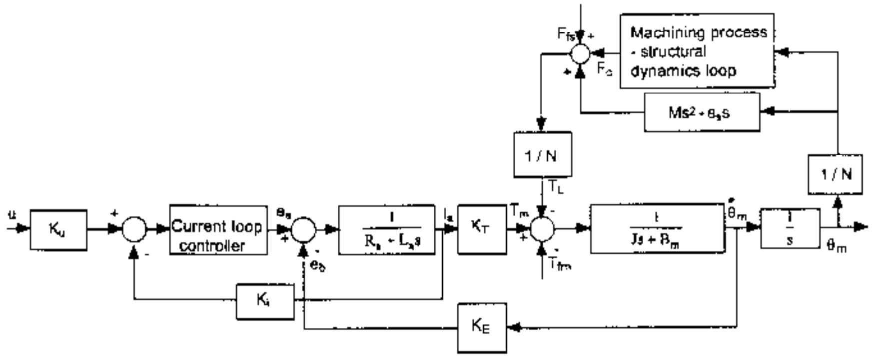

Fig. 2 Feed drive dynamic model and process interaction-brushed DC motor

with a brushed DC motor powered by a thyristor amplifier. Tung and Tomizuka (1993) identified discrete-time models of order varying from five to seven for the closed position loop of a feed drive with an AC motor, and achieved a good frequency response match up to 100 Hz. Inclusion of a closed loop in the identified model is also one way of including nonlinear effects such as amplifier nonlinearities and nonlinear friction components, as the closed loop linearizes to some extent nonlinear relationships within the loop.

Drive Control Problems and Classification

Feed drive control problems may be classified in a number of ways, depending on their requirements. Point-to-point control is relevant for rapid traverse operations between cuts and requires that the drive move rapidly to its final location with a short settling time, the actual trajectory being of secondary importance. Continuous-path control is relevant during machining, the actual drive trajectory and its closeness to the desired trajectory being of primary importance. The resulting control problem is either one of tracking the reference input or of regulating the drive position in the presence of disturbances such as cutting forces. Continuous-path control problems are the more challenging drive control problems, point-to-point control problems being addressed satisfactorily by simple control approaches.

Continuous-path control may be further classified as single axis control and contouring (or multiaxial) control. Single-axis tracking control problems in normal machining have been motivated by applications such as high speed machining of aluminum alloys and high-speed finishing of dies and molds, involving feed rates of over 10 m/min, and non-circular turning of nonaxisymmetric workpieces. Effective feed drive control in the former class of applications requires higher bandwidth of the closed position loop for accurate tracking of a wider variety of reference inputs. Non-circular turning applications involve rapid actuation in the cutting direction, the desired motion being repetitive with a period equal to the period of rotation of the spindle. In both of these applications, feed drive positioning accuracy must be maintained in the presence of normal cutting loads.

Single-axis tracking control in precision machining applications requires higher positioning accuracies and consequently consideration of effects not critical in normal machining applications. The rejection of disturbances such as motor torque ripple, tachogenerator ripple, and disturbances from bearings, by the control loop becomes significant in precision machining applications (Donaldson and Maddux, 1984). The solution requires, in addition to appropriate feed drive mechanical design, a high position control loop bandwidth for effective disturbance rejection. Consideration of friction also becomes more critical in feed drives for diamond turning machines, because of the small displacements and velocities involved. The underlying friction mechanisms in ballscrew drives are different, for example, depending on whether the slide displacement amplitudes are a few tens of nanometers or a few hundreds of nanometers, and call for different control actions (Ro and Hubbell, 1993). Some precision machining applications also require positioning accuracies of a few nanometers and control loop bandwidths in the tens of Hz. The stiffness requirement of the control loop to machining forces varies with the machining application, a static stiffness range of 0.1–1 N/nm being acceptable in diamond turning involving low forces (Kouno, 1984), and a static stiffness of more than 10 N/nm being desired in precision grinding of brittle materials (Kanai et al., 1990).

In contouring control applications, the objective is to coordinate the motion of multiple feed drives so that the contour error, defined in Fig. 3 for biaxial motion, is minimized. The contour error is the error component E_c which leads to part dimensional error, the tracking error E_n being nearly orthogonal to it and contributing little to part dimensional error. The contour error for a specified trajectory increases more than linearly as a function of contouring speed if the control algorithm were to remain unchanged, and becomes significant in high speed machining applications involving contouring feedrates of a few m/min and higher.

The transmission between the primary actuator and the slide also affects the nature of the feed drive control problem significantly. Feed drives including gears or ballscrews for speed reduction have high values of the transmission ratio N (Fig. 2). Consequently, the feed drive control loop is effectively decoupled from the machining process/structural dynamics loop or chatter loop, because of the low values of the coupling terms $1/N$. In effect, the stiffness of the servocontrolled drive as seen

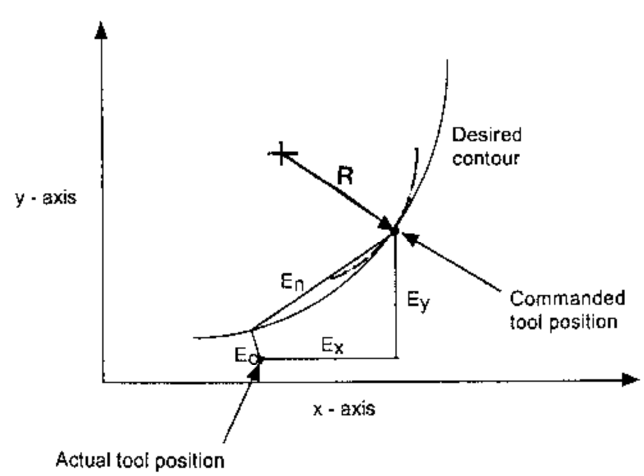

Fig. 3 Contouring motion and error definitions (circular contour is special case, $E_c = e_r$)

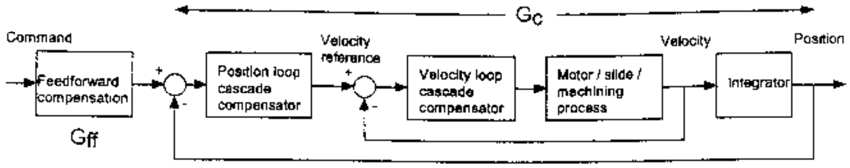

Fig. 4 General feed drive controller structure

by the chatter loop is high enough for its effect on the chatter loop dynamics to be neglected. Also, the feed drive controller design need not consider interaction with the chatter loop. In fact, slide dynamic effects would also be of secondary importance for feed drive control in such cases, unless transmission resonance is significant.

The transmission ratio N would be lower for feed drives using ballscrews with larger pitch. In the limiting case of feed drives using direct linear actuation, the actuator motion itself would be linear and N would equal unity. The advantages of using direct drives stem from the absence of power transmission elements and include lower drive inertia, higher acceleration capability, elimination of transmission resonance, and transmission-related backlash and frictional effects (Pritschow and Philipp, 1990). However, the chatter loop and the feed drive control loop interact significantly, as is evident from the higher value of the coupling terms $1/N$ for lower values of N. Stabilization of the chatter loop then becomes an important control objective for direct feed drives in addition to positioning requirements (Alter and Tsao, 1994b).

Developments and Results in Feed Drive Control

Figure 4 represents a relatively general form of the feed drive controller structure for single-axis applications involving tracking control. Analog as well as digital implementations of the controller are common, as are hybrid implementations involving analog inner control loops and digital outer control loops. Other variations of the structure are also used, for example, controllers employing state estimation and feedback.

The simpler tracking control problems in normal machining applications involve cascade compensator design in the position and velocity loops, the compensators being versions of PID compensators or lead/lag compensators (Gross, 1983). Design criteria include achieving satisfactory closed position loop bandwidth, steady state accuracy, and/or rejection of disturbances such as cutting forces. Digital implementation of the compensators requires consideration of the effects of sampling as well.

Relatively simple controller designs have also proven to be effective in precision machining applications, provided the feed drive mechanical design is appropriate. For example, a high order lead compensator within the velocity loop was used by Donaldson and Maddux (1984) to achieve a velocity loop bandwidth of 161 Hz for a feed drive on a precision machining research lathe, a good dynamic model of the feed drive including transmission resonance being needed for effective compensator design. The choice of a traction drive with an appropriate roller radius lowered the rotational drive speeds significantly, reducing the frequency of the tachogenerator ripple in the actuator velocity measurements to a value where the control loop could reject the disturbance effectively. A hydrostatic bearing was chosen for the same application to reduce static friction and increase damping in the feed drive. Piezoelectric actuation for the correction of systematic errors in a diamond turning machine was able to achieve an accuracy of 10 nm, a range of a few microns, a closed loop bandwidth of 50 Hz, and a static stiffness of 300 N/micron, using simple control algorithms (Kouno, 1984). Closed loop control of position was effective in linearizing actuator hysteresis. Kanai et al. (1990) have described the design of a special hydraulic actuator for the feed drive on a precision grinding machine for brittle materials, with a resolution of one nanometer, stroke of 20 mm, closed loop bandwidth of 80 Hz, and a static stiffness of over 50 N/nm. A hybrid hydrostatic/sliding contact bearing was also used to obtain adequate rejection of the high frequency components of the machining force.

Controller design refinements are capable of improving the performance of feed drives in precision machining operations. Ro and Hubbell (1993) used model reference adaptive control to improve the performance of a DC motor-ballscrew slide for displacement amplitudes ranging from a few nanometers to a few microns, the machining application of interest being diamond turning. Because of the differences in the friction mechanisms in the ballscrew near the two displacement limits, two forms of the controller were necessary to obtain consistent performance over the full range of motion.

Feedforward compensation has been effective in improving the closed position loop bandwidth of feed drives, when implemented digitally (Tomizuka, 1989). Feedforward compensator design in this case involves approximate inversion of the asymptotically stable closed loop transfer function G_c by G_{ff}. G_c is typically given by

$$G_c(z^{-1}) = \frac{z^{-d}B^{-}(z^{-1})\,B^{+}(z^{-1})}{A(z^{-1})} \tag{3}$$

where d is the number of sample delays, the zeros of the polynomial B^- are either outside the unit circle in the complex z-plane or within the unit circle but on or close to the negative real axis, and the zeros of the polynomial B^+ are all within the unit circle. Cancellation of the zeros of B^- would give rise to unstable poles of G_{ff} or poles which would result in a very oscillatory response of the feedforward compensator, both of which are undesirable.

Tomizuka (1989) and others have proposed a number of alternative forms of G_{ff} using frequency domain criteria. The resulting G_{ff} is noncausal and requires preview of the reference input, which is not a problem since future values of the desired trajectory are known from the interpolation algorithm. The feedforward compensator increases the overall bandwidth of $G_{ff}G_c$ by employing high compensator gains at high frequencies and, in most cases, also achieves an overall zero phase characteristic. The high compensator gains at high frequencies may, however, result in saturation problems for some trajectory commands. Furthermore, unmodeled effects such as cutting forces degrade the effectiveness of feedforward compensation (Koren and Lo, 1992). Using such a feedforward compensation scheme and an experimentally identified model of G_c, Tung and Tomizuka (1993) improved the 3-db bandwidth of the feed drive on a medium-duty machining center with an AC motor-ballscrew transmission, from just over 10 Hz to about 40 Hz. The position loop in this case was closed around the motorshaft position, leaving the transmission resonance outside the loop. Feedforward compensation can also be designed so as to optimize a measure of time domain tracking performance subject to constraints on the reference trajectory and preview length (Alter and Tsao, 1995). The latter approach can avoid the saturation problems referred to earlier.

Model-based compensation for nonlinear friction is effective in improving tracking performance at low feed drive velocities and during reversals in high-speed motion. Tung et al. (1993) have used a nonparametric model of drive friction versus motor velocity to estimate drive friction as a function of commanded drive velocity. Stiction errors in the medium duty machining center referred to above, during direction reversals, were reduced from about 10 microns to 3 microns.

Since noncircular machining involves a periodic reference input, the period corresponding to that of spindle rotation and hence known, a special form of the cascade compensator is used in the control of such operations, the resulting control being referred to as repetitive control. The compensator in this case has very high gains localized at the discrete frequencies in the periodic reference input, and consequently results in very low values of the error at these frequency components. The localization of the high gains at the discrete frequencies of interest allows stable loop operation at higher gain magnitudes and lower errors than would have been the case if the high gains were distributed more evenly over the frequency spectrum. Tsao and Tomizuka (1994) have implemented repetitive control on a hydraulic actuator for noncircular machining, and have achieved positioning error in the steady state of less than five microns for a sinusoidal commanded motion of amplitude 1.25 mm and frequency 20 Hz.

Contouring accuracy is an important performance measure of multi-axial feed drives, and tends to degrade markedly as contouring speeds increase. Poo et al. (1972) studied contouring accuracies for biaxial feed drives assuming the simplified model forms of Eq. (1) for each drive, though the parameters themselves could be different for the two drives. Their conclusions were that the feed drive controller parameters for each axis should be tuned to match the closed loop frequency responses for the two axes as closely as possible. Assuming dominant second order dynamic response of the closed loop systems, a damping ratio of 0.707 minimizes the steady state radial error e_r for circular contours. In practice, the radial error for such contours varies approximately as the square of the ratio ω/ω_n, ω_n being the closed loop undamped natural frequency. ω is the angular contouring frequency of the commanded motion and is equal to V/R, where V is the contouring speed and R is the trajectory radius. The radial error achieved by just such an adjustment of the axis feedback controllers on a medium-duty machining center, for a circle of radius 10 mm traversed at 7.5 m/min (angular contouring frequency of 2 Hz), was about 110 microns in the absence of machining (Tung and Tomizuka, 1993). The corresponding axis closed loop 3-dB bandwidth was a little over 10 Hz. The closed loop bandwidth was subsequently increased to about 40 Hz, and the radial error reduced to less than 10 microns, with the use of feedforward compensation based on an experimentally identified model of the closed loop feed drive dynamics.

An alternative approach to contouring accuracy improvement (Koren, 1980; Kulkarni and Srinivasan, 1989; Chiu and Tomizuka, 1995) directs more control effort toward reduction of the contour error E_c rather than the orthogonal component E_n (Fig. 3), since only the former results in part geometry error. Since the contour error component E_c depends on both the axis errors E_x and E_y, the control action couples the responses of the axes, and hence these controllers are referred to as cross-coupled controllers. Figure 5 shows a general representation of a cross-coupled controller for a biaxial system. In its simplest form, the additional control action introduced by the cross-coupled controller depends only on the contour error E_c (Koren, 1980). Inclusion of the contour error in a quadratic index of performance to be optimized by controller design results in the additional control action depending on the individual axis velocities as well as the contour error (Kulkarni and Srinivasan, 1989). Because of the time-varying relationship between the contour error and the individual axis errors for curved contours, all cross-coupled controllers are time-varying controllers when implemented for curved contours. Chiu and Tomizuka (1995) have reported on the performance of a cross-coupled controller designed using a task-based coordinate frame approach and applied to a medium duty machining center. The radial error for a circular contour of radius 10 mm traversed at a contouring speed of 6 m/min, (resulting in an angular contouring frequency of about 1.5 Hz) was kept below about 7 microns in the absence of machining forces. The error component in the tangential direction was allowed to be higher, about 45 microns, as it was less critical. The performance of these controllers has also been shown experimentally to be less sensitive to feed drive model accuracy and knowledge of cutting loads and frictional forces than are controllers which depend only on feedforward control action (Koren and Lo, 1992).

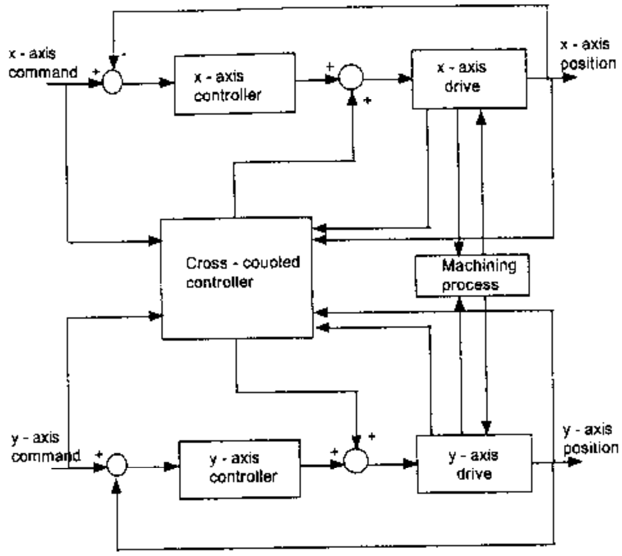

Fig. 5 Cross-coupled controller structure

The higher positioning performance achievable using direct drive motors is indicated by the results reported by Pritschow and Philipp (1990) for a linear asynchronous motor with a 2000 N force capacity and slide mass of 50 kg. A position loop bandwidth of over 130 Hz was reported along with a minimum dynamic stiffness of about 200 N/micron, using discrete-time implementations of proportional and proportional-integral controllers in the position and velocity loops, respectively. The higher positioning performance would translate to improved machining accuracy if the servo dynamic stiffness is high enough to ensure stability of the specific machining process being performed. Alter and Tsao (1994b) have evaluated analytically the stability of machining processes which employ actively controlled linear motor feed drives, and have also experimentally investigated the effect of static and dynamic servo stiffness on machining accuracy and stability. They have also demonstrated that servo controller refinements are capable of improving achievable dynamic stiffness and hence stability of the machining process (1994c).

Future Areas of Research and Research Trends

An important research issue of current interest in the control of direct drives for machining is the simultaneous enhancement of the dynamic servo stiffness of such drives for improved chatter stability, along with improvement of positioning accuracy. Robust controller design techniques are good candidates because of the need to accommodate changes in the process behavior with feed and depth of cut, tool geometry, and workpiece materials, as well as change in slide inertia due to workpiece changes (Alter and Tsao, 1994c). Controller refinements may also be expected to result in further feed drive performance improvements in those precision machining applications which have relied primarily on feed drive mechanical design enhancements in the past.

Two-stage actuation involves the cascading of a fast, high-resolution microactuator with a lower bandwidth macroactuator, and has been evaluated for a variety of applications ranging from noncircular turning of engine piston profiles, diamond turning of nonrotationally symmetric surfaces requiring high range-to-resolution ratios of up to 10^8:1, and high-speed multiaxial contouring. The development of advanced control algorithms which partition the control effort between the macro and microactuators optimally for the application of interest is another research issue of current interest.

Development of novel machine tool architectures is also likely to result in new demands on feed drive performance. Recently, six degree-of-freedom machine tools based on parallel-link kinematic structures have become commercially available and seek to achieve high maneuverability, high feedrates, and high positioning accuracy. Feed drive control for such machines poses important challenges because of the inherent dynamic coupling between the axes and nonlinear dynamic effects, these aspects becoming more significant at higher speeds of operation.

References

Alter, D. M., and Tsao, Tsu-Chin, 1994a, "Exact 2-D Model Matching with Application to Repetitive Control," ASME *Journal of Dynamic Systems, Measurement, and Control*, Vol. 116, Mar., pp. 2–9.

Alter, D. M., and Tsao, Tsu-Chin, 1994b, "Stability of Turning Processes with Actively Controlled Linear Motor Feed Drives," ASME JOURNAL OF ENGINEERING FOR INDUSTRY, Vol. 116, Aug., pp. 298–307.

Alter, D. M., and Tsao, Tsu-Chin, 1994c, "Dynamic Stiffness Enhancement of Direct Linear Motor Feed Drives for Machining," *Proceedings of the American Control Conference*, Baltimore, Maryland, June, pp. 3303–3307.

Alter, D. M., and Tsao, Tsu-Chin, 1995, "Optimal Feedforward Tracking Control of Linear Motors for Machine Tool Drives," *Proceedings of the American Control Conference*, Seattle, Washington, June, pp. 210–214.

Armstrong-Helouvry, B., Dupont, P., and Canudas de Wit, C., 1994, "A Survey of Models, Analysis Tools, and Compensation Methods for the Control of Machines with Friction," *Automatica*, Vol. 30, No. 7, pp. 1083–1138.

Bollinger, J. G., Stute, G., and Van Brussel, H., 1982, "Digital Controls and Feed Drives: State of the Art and New Developments," *Annals of CIRP*, Vol. 29/2, pp. 497–506.

Chiu, G. T. C., and Tomizuka, M., 1995, "Contouring Control of Machine Tool Feed Drive Systems: A Task Coordinate Frame Approach," *Proceedings of the ASME Dynamic Systems and Control Division, DSC* Vol. 57-1, International Mechanical Engineering Congress and Exposition, San Francisco, California, Nov., pp. 503–510.

Donaldson, R. R., and Maddux, A. S., 1984, "Design of a High-Performance Slide and Drive System for a Small Precision Machining Research Lathe," *Annals of CIRP*, Vol. 31/2.

Gross, H., 1983, *Electrical Feed Drives for Machine Tools*, John Wiley & Sons Ltd., New York.

Kanai, A., Sano, H., Yoshioka, J., and Miyashita, M., 1990, "Positioning of a 200 kg Carriage on Plain Bearing Guideways to Nanometer Accuracy with a Force-Operated Linear Actuator," Nanotechnology, pp. 43–51.

Koren, Y., 1980, "Cross-Coupled Biaxial Computer Control for Manufacturing Systems," ASME *Journal of Dynamic Systems, Measurement, and Control*, Vol. 102, Dec., pp. 265–272.

Koren, Y., and Lo, C. C., 1992, "Advanced Controllers for Feed Drives," *Annals of the CIRP*, Vol. 41/2, Aug., pp. 1–10.

Kouno, E., 1984, "A Fast Piezoelectric Actuator for Servo Correction of Systematic Errors in Precision Machining," *Annals of the CIRP*, Vol. 33, Aug., pp. 369–372.

Kulkarni, P. K., and Srinivasan, K., 1984, "Identification of Discret Time Dynamic Models for Machine Tool Feed Drives," ASME *Symposium on Sensors and Controls for Automated Manufacturing and Robotics*, ASME Winter Annual Meeting, New Orleans, pp. 1–20.

Kulkarni, P., and Srinivasan, K., 1989, "Optimal Contouring Control of Multi-Axial Feed Drive Servomechanisms," ASME JOURNAL OF ENGINEERING FOR INDUSTRY, Vol. 111, May, pp. 140–148.

Patterson, S., and Magrab, E., 1985, "Design and Testing of a Fast Tool Servo for Diamond Turning," *Precision Engineering*, Vol. 7, No. 3, pp. 123–128.

Poo, A. N., Bollinger, J., and Younkin, G. W., 1972, "Dynamic Error in Type 1 Contouring Systems," IEEE *Transactions on Industrial Applications*, Vol. IA-8, No. 4, pp. 477–484.

Pritschow, G., and Philipp, W., 1990, "Direct Drives for High-Dynamic Machine Tool Axes," *Annals of the CIRP*, Vol. 39, pp. 413–416.

Ro, P. I., and Hubbel, P. I., 1993, "Model Reference Adaptive Control of Dual-Mode Micro/Macro Dynamics of Ball Screws for Nanometer Motion," ASME *Transactions Journal of Dynamic Systems, Measurement, and Control*, Vol. 115, Mar., pp. 103–108.

Slocum, A., 1992, *Precision Machine Design*, Prentice-Hall, Englewood Cliffs, NJ.

Tomizuka, M., 1989, "Design of Digital Tracking Controllers for Manufacturing Applications," ASME *Manufacturing Review*, Vol. 2, No. 2, June, pp. 134–141.

Tsao, T.-C., and Tomizuka, M., 1994, "Robust Adaptive and Repetitive Digital Tracking Control and Application to a Hydraulic Servo for Noncircular Machining," ASME *Journal of Dynamic Systems, Measurement, and Control*, Vol. 116, Mar., pp. 24–32.

Tung, E. D., and Tomizuka, M., 1993, "Feedforward Tracking Controller Design Based on the Identification of Low Frequency Dynamics" ASME *Journal of Dynamic Systems, Measurement, and Control*, Vol. 115, Sept., pp. 348–356.

Tung, E. D., Tomizuka, M., and Urushisaki, Y., 1993, "Experiments in High-Speed Machining Using a Digital Feedforward Controller," *Transactions of the NAMRI/SME*, Vol. 21, Stillwater, Oklahoma, pp. 279–285.

Weck, M., 1984, *Handbook of Machine Tools, Volume 3, Automation and Controls*, Wiley, New York.