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Precise Position Control of Machine Tool Feed Drives in the Presence of Load Perturbation

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ABSTRACT

On machine tools, precise control of the position of the tool tip relative to the workpiece is required to follow the desired path where quality of producing parts is significantly dependent on dynamic performance. The design of such a controller that can perform with all plant uncertainties is a major challenge. It is therefore the aim of this paper to assess the performance of a new design methodology that may enhance the performance of machine tool feed drives. The controller in general has a form of P/PI/PID (Proportional/ Integral/Derivative) compensators that is easy to implement in practice and does not require additional hardware. In addition to a simple structure, the design applies the principles of a phase margin approach to allow the self-tuning controller to automatically update the controller settings. This ability is required to optimise control when the position and the mass of the fixture and/or workpiece change. Effectiveness of the proposed methodology is demonstrated in Matlab simulation, where its performance found good enough for industrial application.

Keywords: P/PI/PID controllers, Phase Margin, Adaptive control, Tracking control, Machine tool feed drives

INTRODUCTION

In machine tools feed drive design, the prevalent principle is to control the position of the tool tip relative to the workpiece as precisely as possible compared to the commanded nominal position, thus ensuring high quality in the parts produced on the machine. Process perturbations such as inertial changes and nonlinear effects, can appear during general machining, which inevitably leads to a reduction in the quality of parts produced. However, conventional feed drive control can be captured with an ideal second order dynamic system in order to implement so-called standard PI/PID controllers. In addition, most of the control techniques applied to conventional feed drives are with fixed gain, where the controller gains are tuned to capture nominal plant conditions. Hence, developing such a control methodology that aims to update the controller setting during machining can have a significant influence on performance requirements and mitigate these adverse effects. To address these challenges, adaptive control is applied to reduce the effects of nonlinearities, model uncertainties, and time variant parameters such as friction and cutting force [1-3].

Some of the recently proposed solutions, which achieve the problem of machine feed drive tracking control in literature include: a zero phase error- tracking controller (ZPETC) proposed by Tomizuka [4] and achieved by means of cancelling the effects of steady state dynamics of the feed drive in a feedforward scheme. The procedure was found to be valid for certain plant dynamics and not appropriate for a time varying parameter (TVP) model such as inertia and friction, which makes it rather difficult to maintain the accuracy. Consequently, Tomizuka [5], and Ye [6] have recommended some improvements by pre-filtering (low pass filter) the position commands to avoid the adverse effects of high frequency components within the control signal. Van Brussel et al[7], have implemented Kalman filter for disturbance estimation, Loop Transfer Recovery (LTR) feedback gain toenhance robustness of the system, and a feedforward term to improve tracking accuracy. Erkorkmaz and Altintas [8],have developed a high performance control scheme by means of pole placement feedback controller, feedforward axis dynamic compensation (ZPETC),feedforward friction compensation and Kalman filter for disturbance compensation. Renton and Elbestawi [9] have developed a control law that takes into account the axis performance capabilities such as position, velocity, and acceleration where the target path in the presence of disturbances does not exceed envelop limits and modifying the velocity accordingly. An additional approach has been presented by Altintas and Erkorkmaz [10] to adaptively enhance the tracking performance of ball screw feed drives in the presence of disturbance with an adaptive Sliding mode controller. All the classical techniques that have been mentioned have a drawback of sensitivity to model errors and their robustness may be insufficient to guarantee high performance of precision motion under broadly varying operating conditions. On the other hand, the adaptive sliding mode controller has to be implemented with accurate modelling and identified parameters of the feed drive mechanism.

In this paper, a self-tuning PI controller using a phase margin approach is introduced which is capable of achieving the desired tracking performance with model parameters changes (i.e. drive inertia) and external disturbances. The control design methodology is applied to the rigid body dynamics of a ball screw feed drive, but its extension to linear drives is applicable where rigid body dynamics of feed drives is captured with the same model. The layout of the paper is organised as follows. the rigid body dynamics model of a ball screw is presented in section 2, the control statement design methodology and simulation results are prescribed in section 3 and 4 respectively and the conclusions are presented in Section 5.

RIGID DYNAMICS OF BALL SCREW FEED DRIVES

Rigid body models of feed drives are obtained to capture the low frequency range of the system (i.e. excitation of structural resonance is excluded). The approach is valid to be implemented for ball screw and linear feed drives. The model parameters consist of inertia, viscous damping and friction effects. Erkorkmaz and Altintas [11] have shown how to identify the parameters of a simple linear model of feed drives by using unbiased least square and Kalman filter for friction model . In this approach, a series of control signal $u(s)$ is applied to the motor to deliver as much excitation as possible to the axis dynamics. Subsequently, the inertia and damping can be identified using measured velocity and motor torque with consideration of the following models:
Axis position in Laplace domain can be written as [12]:

$$X(s) = \frac{r_g}{s} . \frac{1}{J_s + B} [K_t K_a u(s) - T_d(s)] \quad (1)$$

$$T_m = K_t K_a u(s) \quad (2)$$

From the above equations, the drive velocity can be obtained as:

$$\omega(s) = \frac{1}{J_s + B} [T_m - T_d(s)] \quad (3)$$

The moment of inertia of the table and workpiece reflected on the ball screw shaft is:

$$J_{tw} = (m_t + m_w)(\frac{h_p}{2\pi}) \quad (4)$$

The moment of inertia of the ball screw with a pitch diameter of d is:

$$J_l = \frac{1}{2} m_l (\frac{d}{2})^2 \quad (5)$$

The total inertia reflected on the motor's shaft is calculated as:

$$j_e = \frac{J_{tw} + J_l}{r_g^2} + J_m \quad (6)$$

The parameters of the above mechanical system are equivalent inertia $J(\mathrm{Kg\,m^2})$, viscous damping $B(\mathrm{Kg\,m^2/s})$, motor torque $T_m(\mathrm{Nm})$, motor torque constant $K_t(\mathrm{Nm/A})$, amplifier gain$K_a(\mathrm{A/V})$, and disturbance torque on the motor shaft $T_m(\mathrm{Nm})$.The disturbance torque is a combination of cutting force, friction effects and axis inertia variation (i.e. component and fixture mass change). The case of jogging the axes back and forth with no load, the overall disturbance will be considered as drive inertia and friction effects. A similar technique has been applied in [13-16] for identifying the rigid body dynamics of the ball screw. The main disadvantage of the aforementioned model is the lack of ability to capture the adverse effects of structural vibrations on control design performance.

Rigid body dynamics applied on a small vertical milling machine velocity loop:

The parameters of the machine feed drives are calculated and identified in [17], using experimental work and machine data sheets. Table 1 indicates the required parameters for system modelling and control statement design.

Table 1: Feed drive control system parameters

Feed drive parameters	X Axis	Y Axis
Feed Drive		
Amplifier Gain K_a[A/V]	15.7311	15.7311
Motor Torque Constant K_t [N.m/A]	1.39	1.39
Damping B $[(Kg.m^2/sec)]$	14×10^{-4}	14×10^{-4}
Inertia J [Kg.m^2]	2.8828×10^{-3}	3.8028×10^{-3}
Ball screw pitch[mm]	12	12
Velocity Control Loop parameters		
Sampling time t_s [ms]	0.250	0.250
Proportional gain K_p	1.5735	1.5735
Integral gain K_i	5.4976	5.4976

By applying the parameters values from the table on (3), the velocity control loop can be obtained, which thereafter will be used for controller design.

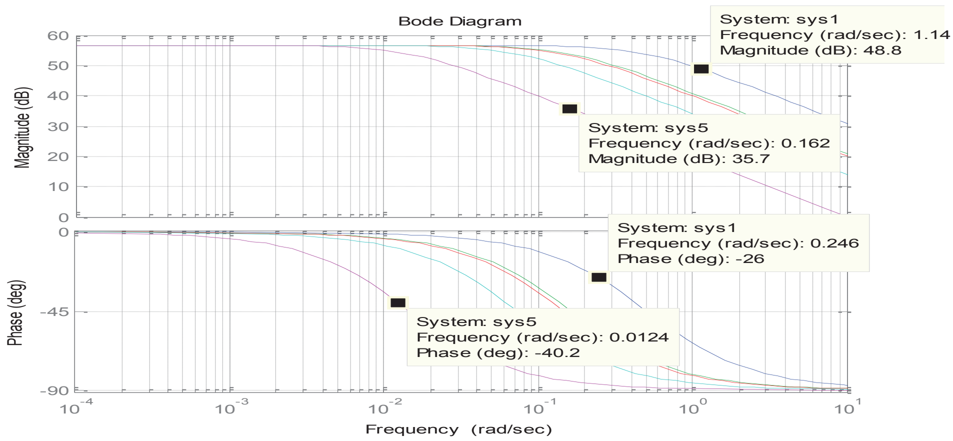

Figure 1: Feed drive phase and magnitude as function of inertial change

CONTROLLER STATEMENT DESIGN

By considering the open loop dynamics of the ball screw feed drive, the open loop transfer function can be written as follows:

$$L(S) = G(S)K(S) \tag{7}$$

Where G(s) defined as a finite set of plants $G_1(S), \dots \dots G_n(S)$,and K(S) is a PI compensator with proportional ab (K_p)and integral a (K_I) actions

$$K(S) = \frac{a(1+bS)}{S} \tag{8}$$

The objectives are to design a controller that satisfies the following criterion of closed-loop sensitivity function with the gain margin (GM) and phase margin (PM) conditions[18]:

$$\left|\frac{1}{1+kL(S)}\right| \leq M \; for \; S = j\omega, \forall\omega \geq 0, k \in [1, k], \tag{9}$$

Sensitivity boundary condition of the plant defined by $M > 1$, and uncertain gain with interval $[1, k]$

By assuming that the model can be presented as First Order plus Dead Time (FOPDT) or as Second Order plus Dead Time (SOPDT) systems, then with consideration of ZOH the discrete model of the feed drive system

$$P(s) = [\frac{K_p}{\tau_s+1}]e^{-\theta s} \tag{10}$$

Where K_p is gain of stable system, $\tau$is time constant and θ represents dead-time
The real feed drive model can be expressed by a discrete transfer function model as:

$P(q^{-1}) = \frac{B}{A} q^{-d-1} = \frac{b}{1-aq^{-1}} q^{-d-1}$ Where a=$e^{-\frac{1}{\tau}T_S}$ b= $k_p(1-e^{-\frac{1}{\tau}T_S})$ (11)

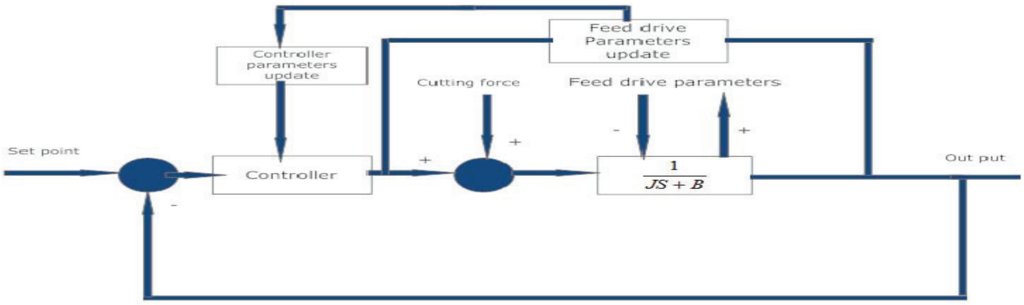

Figure 2: The block diagram of adaptive PI controller [19, 20]

In machine tool feed drives, periodic disturbances are very common such as viscous friction, cutting forces and load variation. In particular, the disturbance can be seen on the level of feed drive as a time variant with periodic values, which are likely to be captured with different models as shown in [19].The parameters of the model can be estimated by Recursive least Square (RLS) with exponential forgetting factor combined with model approaching to zero frequency for unwanted parameters. The adaptive controller is designed to satisfy the following general model [20]

$P(q^{-1}) = \frac{B}{A} q^{-d-1} = \frac{b_1 q^{-1} + b_2 q^{-2} + \cdots + b_m q^{-m}}{1-aq^{-1}}$ (12)

Yield, the estimated parameters in a vector form as:

$\theta = [a, b^1, b^2, \dots \dots \dots, b^m]^T$

Therefore, the system output can be written as:

$$y(k) = \theta^T h(k) \ \ where \ \ h(k) = [-y(k-1), u(k-1), u(k-2), \dots \dots, u(k-m)]^T$$

To implement the recursive algorithm, and in order to update model parameters in each sample time, it is necessary that the error must be defined as follows:

$\varepsilon(k) = y(k) - \dot{\theta}^T(K-1)h(k)$

The error relation defines the difference between the actual system and the estimated outputs, where parameter estimation updates are carried out by:

$\dot{\theta}(k) = \dot{\theta}(k-1) - G(K)\varepsilon(k)$

And the estimator gain matrix, with forgetting factor ρ (chosen from 0 to 1) is defined as:

$G(k) = \frac{\rho(k-1)h(k)}{\rho + h^T P(K-1)h(k)}$ (13)

The developed estimation methodology takes into consideration the whole range of acceptable parameters, which let RLS estimation approaches to zero for unwanted parameters.

PI controller design:

In order to implement phase margin design, we must ensure that the gain of frequency response of the open loop is unity(=1) when its phase shift is $< G_{OL}(j\omega) = -\pi + \emptyset_m$

The next step of the design is to find a frequency, which satisfies the following condition:

$< G_{OL}(j\omega) = -\pi + \emptyset_m$ when $\omega(rad) = \omega_{gm}$

For the sake of this design, the nominal plant and controller parameters of the feed drive are available. Thus, the cross over frequency (ω_gm) can be numerically obtained with Matlab. The adaptive control low is designed to satisfy $|G_{OL}(j\omega)| = 1$,hence controller parameters are updated accordingly.

SIMULATION RESULTS

Two simulation tests have been carried out; the first test is shown in figure3 to clarify how the proposed controller is adaptive and capable of tracking the set point. However, the phase margin of the first test is deliberately chosen to be incorrect (the phase margin of the nominal plant and machine controller can be obtained with Matlab or can be calculated from [20]) to indicate its effect on the performance of the controller. Consequently, applying incorrect phase margin leads to large settling time and overshoot. On the other hand, the second simulation test as indicated in figure 5 compares the output of the controller with respect to the existing machine controller, where the performance of the adaptive PI controller is enhanced with the desired phase margin. From the simulation results, it is evident that the proposed design methodology is competent in achieving the industry demands of velocity changes and high acceleration and deceleration due to accurately tracking the set point.

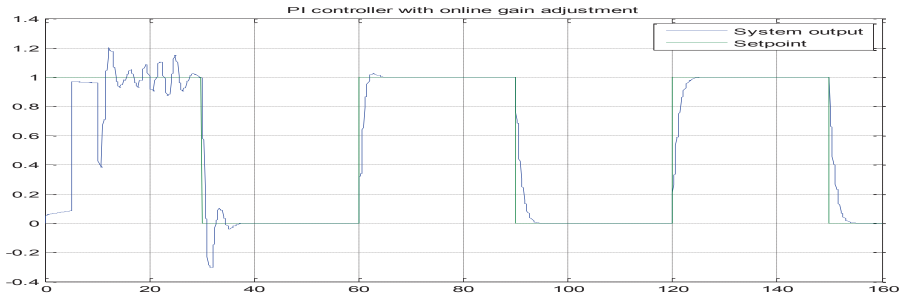

Figure 3: PI controller with online gain adjustment

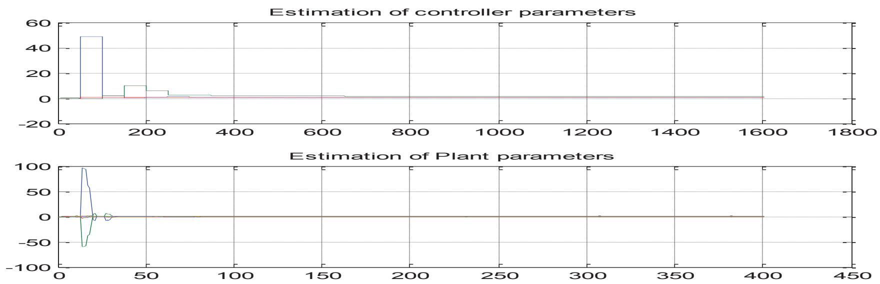

Figure 4: Parameters estimation

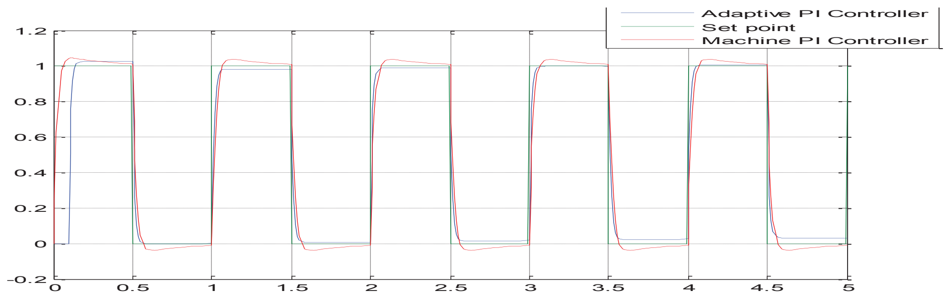

Figure 5:Tracking performance. Adaptive PI controller vs Machine PI controller

CONCLUSIONS

The practical design of a new adaptive PI controller for machine tool feed drives has been presented in this paper. Moreover, it is shown that the adaptive PI controller is able to cope with changes in feed drive parameters and machining perturbation. The idea behind improving the velocity control loop of the feed drive is due to its significant influence on the performance of position, acceleration and deceleration characteristics of the feed drive in high-speed machining. The performance of the proposed methodology can be extended for implemention in industrial applications for the following reasons:

- In terms of robustness, the design methodology is robust, where the gain and phase margin are measure of robust in control engineering.
- Methodology is competent to be accepted by industry for the following points:
  - Does not require additional hardware to be implemented.
  - Its structure has the form of PID filters.
  - Dose not include advanced control theory

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