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A novel cascade control principle for feed drives of machine tools

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ABSTRACT

Most feed drives in machine tools are designed using ball screw drives. The bandwidth of such drives with conventional PPI control principle is limited by the first natural frequency of mechanical transmission elements. A novel feedback control principle is presented to improve feed drive dynamics. It is cascade structured and consists of a weakly set motor speed controller, a disturbance observer with the feedback velocity of machine table and a PD position controller. This principle increases the bandwidth of the position loop over 200% without any extra sensor or actuator. Experimental results validate the effectiveness and robustness of this principle.

1. Introduction

Ball screws with cascaded PPI control principle, which consists of an outside proportional position controller and a proportional integral motor speed controller inside, is most commonly used in machine tool industries due to its cost effectiveness, easy tuning and robustness [1]. However, with such a control principle the bandwidth of the position loop i.e. *Kv* factor (or *Kpp* in some countries) is limited by the first natural frequency of the mechanical transmission elements [2]. In order to dampen the mechanical resonance and increase the bandwidth, various solutions have been proposed. They can be divided into two classes: alternatives and supplements to the PPI principle.

Most alternative control principles are based on sophisticated control theories with state space representation. Altintas et al. developed a sliding mode controller via rigid-body model [3] and it was then improved by utilizing two-mass model [4]. Erkorkmaz proposed pole placement controller for the control of feed drives [5]. H_∞ controllers with good robustness can also be seen for ball screw drives [6,7]. The supplement solutions are based on the cascade control structure. Pritschow et al. introduced an additional control loop with the feedback of table acceleration to raise the phase of the table tracking response [8]. Besides it, many constructive solutions based on the PPI principle are also regarded as the supplements of PPI principle. Semi-active damper is proposed to dampen the vibration in the feed direction [9]. Pritschow and Croon modified the drive mechanism with a soft axial bearing and a strong damper, so that the *Kv* factor is no longer limited by the first mechanical resonance [10].

All solutions mentioned above increase the bandwidth of feed drives. However, because of the complexity and insufficient robustness of control engineering solutions and extra hardware costs of constructive solutions, none of them can substitute for the PPI principle. The open question is, how to increase the bandwidth of feed drives with the similar features of the PPI controller today: no additional cost, sufficient robustness and easy tuning. In Ref. [11], the idea of vibration damping through the weakly set motor speed controller was proposed. Based on this damping concept, a novel control principle is presented in this paper. The remainder of this paper is as follow. The control principle will be introduced and analysed in Section 2. A test bench will be introduced in Section 3. In Section 4, parameters of the presented controller will be tuned. The effectiveness and robustness of this control principle will be experimentally verified in Section 5. Section 6 concludes with comments.

2. Principle analysis of the new control structure

The analysis of control principles is based on a mechanical model with two masses, which is shown in Fig. 1. The equivalent mass of the motor and spindle inertia is summarized as m_1. Its position and velocity are captured by the rotary encoder, which is built in the motor. The mass of the table with its load is summarized as m_2. Its position and velocity are gathered by the linear encoder, which is fixed on the table. c is the overall axial stiffness and d represents the damping factor between two masses. The friction of the guides and bearings is neglected in this model.

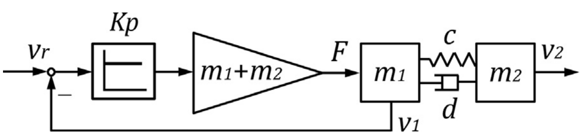

Fig. 1. Two-mass model of feed drives with a proportional motor speed controller.

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In the standard control structure, the motor speed is fed back to the speed controller, so that mechanical vibrations can be excluded from the speed control loop. The motor speed here is controlled by a proportional controller for the simplification of the derivation procedure. The tracking performance of the table speed v_2/v_r is investigated firstly, since it is the control plant in the position loop and its behaviour dominates the position controller directly. The transfer function from the reference speed v_r to the actual table speed v_2 can be formulated as:

$$\frac{v_2}{v_r} = \frac{ds + c}{\frac{m_1 m_2}{(m_1+m_2)K_p}s^3 + \left(\frac{d}{K_p} + m_2\right)s^2 + \left(\frac{c}{K_p} + d\right)s + c}. \quad (1)$$

The control gain Kp (or Kvp in some countries) is normally set to a high value for sufficient feed stiffness. However, a too stiffly tuned motor excites natural vibrations of drive mechanism significantly. On the contrary to the standard principle, the motor speed controller here is set to a low value, so that the resonance vibrations can be directly damped by the motor. With the weakly set Kp, the transfer function Eq. (1) can be approximated as:

$$\frac{v_2}{v_r} \approx \frac{K_p}{s + K_p} \cdot \frac{ds + c}{\frac{m_1 m_2}{m_1+m_2}s^2 + (d + K_p m_2)s + c} \quad (2)$$

$$\text{with} \quad K_p < \sqrt{c/m_2}. \quad (3)$$

This approximation is based on the fact that the cut-off frequency of the first multiplier of Eq. (2) is constant lower than those of the second multiplier, which is guaranteed through Eq. (3). For the practical application, the value of Kp is suggested about five times smaller than the mechanical resonance $(c/m_2)^{0.5}$.

The second multiplier of Eq. (2) describes a second order (PT2) system, whose natural frequency can be formulated as:

$$\omega_r = \sqrt{\frac{c}{m_2}\left(1 + \frac{m_2}{m_1}\right)}. \quad (4)$$

Compared to the mechanical resonance, ω_r is increased by the factor of the mass ratio m_2/m_1. However, the bandwidth of Eq. (2) is still limited by its first multiplier, whose cut-off frequency is much lower than the mechanical resonance due to the weakly set Kp. In order to counter this first order (PT1) behaviour, a PD controller is introduced in the position loop. The functional principle of the PD position controller is shown in Fig. 2.

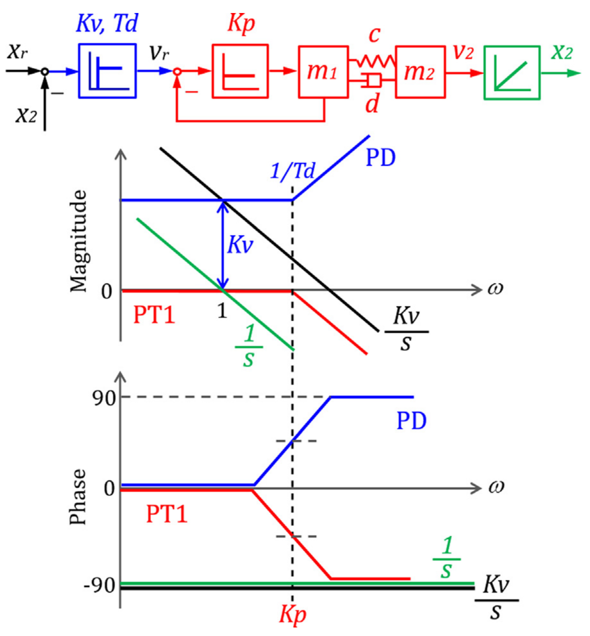

Fig. 2. Open position loop with its schematic frequency response showing the functional principle of the PD controller.

Blue, red and green curves in Fig. 2 describe the Bode plot of the PD controller, the PT1 system and the integrator, respectively. The second multiplier of Eq. (2) is here neglected. It can be seen that the PD controller cancels the PT1 behaviour by setting the reciprocal value of the derivative time T_d with the same value of the motor speed control gain Kp:

$$G_{PD}G_{PT1}G_{int} = K_v(1 + T_d s) \cdot \frac{K_p}{s + K_p} \cdot \frac{1}{s} = \frac{K_v}{s} \quad (5)$$

where $Kp = 1/T_d$. The open loop bandwidth of the summarized system (black lines) is therefore dominated by the proportional part of the position controller Kv.

Compared with the table speed control loop introduced in Ref. [11], this new principle can further increase the Kv factor, since the stability reserve of the absent table speed loop needs not to be considered. However, the control structure shown in Fig. 2 has a drawback. It cannot compensate the static tracking error, which is caused by the disturbance forces, since neither the position controller nor the speed controller has an integral part. In order to solve this problem, a disturbance observer (DOB) is introduced in this control principle.

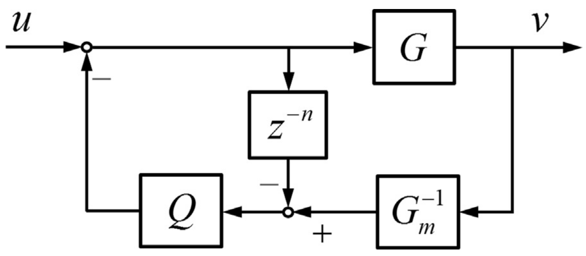

Fig. 3. Structure of a disturbance observer.

The structure of a DOB is shown in Fig. 3, where G is the real plant, G_m is the nominal plant model of G, z^{-n} is the time delay and Q is a low pass filter. The close loop transfer function of this observer is formulated as:

$$\frac{v}{u} = \frac{G}{1 + Q\left(\frac{G}{G_m} - z^{-n}\right)}. \quad (6)$$

If the delay and the filter could be ignored, the system performance would be perfectly dominated by the defined plant model G_m.

$$\frac{v}{u} = G_m \quad \text{if} \quad Q = 1 \quad \text{and} \quad n = 0 \quad (7)$$

However, the time delay in the reality cannot be eliminated and the modelled nominal plant G_m can never perfectly describe the real plant G, so that the filter with a too large bandwidth leads to the system instability. According to the robust control theory, the observer is robust stable, only if the following condition is satisfied [12].

$$\| Q(j\omega) \cdot \Delta(j\omega) \|_\infty < 1 \quad \text{with} \quad \Delta = \frac{G - G_m}{G_m} \quad (8)$$

A DOB combined with a PD controller is also used in Ref. [12] to improve the disturbance rejection of linear motors. There, the DOB is implemented on the inner control loop and the nominal plant is approximated by a PT2 system. In contrast, this paper introduces a DOB implementation between the position and the motor speed loop, so that the nominal plant model can be simplified to a PT1 system. Fig. 4 shows the complete control structure. The proposed control principle consists of an outside PD position controller, a DOB loop in the middle and an inside proportional motor speed controller and will be abbreviated as PDOP in the following text.

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