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Enhancement of feed drive dynamics using additional table speed feedback

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ABSTRACT

The tracking performance and disturbance response of feed drive systems with P–PI cascade control principle are limited by the first natural frequency of mechanical transmission elements. In order to overcome this limitation, a new control principle for feed drives with large inertia ratio is presented. It synthesizes a weakly set motor speed controller with an additional table speed control loop. The bandwidth is increased without any extra sensor or actuator. The large inertia ratio can be achieved by using a high dynamic motor with small inertia and a ball screw with high pitch.

1. Introduction

Both the machining time and the quality of the machined workpiece are dominated by the dynamic performance of feed drives of machine tools. Most feed drives for industrial applications are designed using ball screw drives for cost reasons and controlled by a cascaded proportional integral (PI) controller in the motor speed loop and a proportional (P) controller in the position loop. As pointed out in [1], the first natural frequency of the ball screw at a low frequency range confines the bandwidth as well as the control gain (K_v factor) of the position loop.

In order to increase the bandwidth, various solutions have been presented. Many of them are summarized in a CIRP key note paper [2]. The existing methods can mainly be divided into two classes. One class is based on the *conventional cascade control structure*. Pritschow et al. extended this structure with an additional feedback of acceleration, which was directly measured by a Ferraris sensor [3]. Verl and Frey designed a semi-active damper between the table and the machine bed [4]. Pritschow and Croon reconstructed the feed drive mechanism with a soft axial bearing and a strong parallel damper [5]. All these concepts attenuate resonance vibrations and increase the bandwidth but require an extra sensor or actuator.

The other class belongs to the *sophisticated control theories* with the state space representation. Altintas et al. presented an adaptive sliding mode controller to increase the dynamic stiffness [6]. Erkorkmaz et al. improved the feed drive performance via the combination of a pole placement controller and the loop shaping strategy [7]. In recent years, the H_∞ control theory has also been introduced into feed drive systems with ball screws [8,9]. These solutions require no further hardware; however, they are not widely applied in industries because of their complexity or insufficient robustness.

In this paper, a new control method for feed drives with large inertia ratio is presented. It is based on the cascade structure with an additional table speed control loop to increase the achievable bandwidth of the position control loop. The functional principle will be explained and the effectiveness will be verified by experiments.

2. Principle of the new control method

As shown in Fig. 1, mechanical drives can be abstracted in a model with two masses, since they are typically instrumented with two encoders. The rotary encoder is built in the motor (v_1, x_1) and the linear encoder (v_2, x_2) is fixed on the table. For a translational view on the drive system, the inertia of motor J_m, coupling J_c and threaded spindle J_s can be summarized as equivalent mass

$$m_1 = (J_m + J_c + J_s) \cdot \left(\frac{2\pi}{h} \cdot i \right)^2, \qquad (1)$$

with the screw pitch h and the gear ratio i. The mass of the table with its load is summarized as m_2. The resulting inertia ratio in this paper is defined as m_2/m_1.

Both masses are coupled by a stiffness c and a damping factor d. The friction of bearings and guides is neglected in this model.

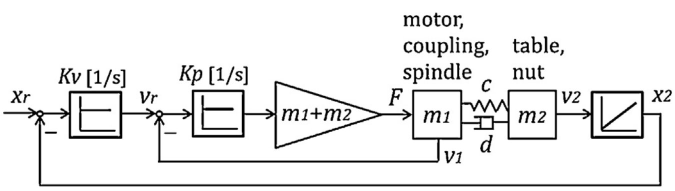

Fig. 1. Simple model of ball screw with conventional motor speed and position controllers.

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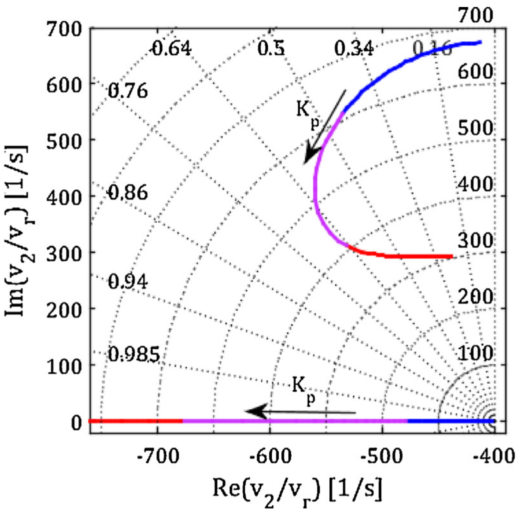

Fig. 2. Root locus of closed motor speed loop with mechanics.

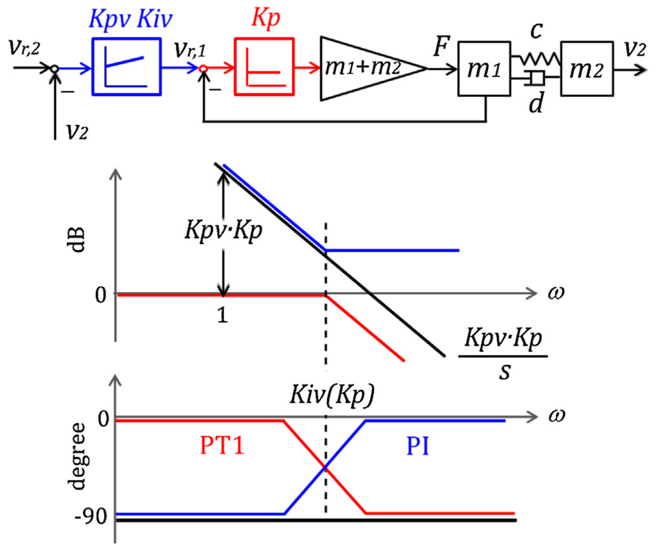

Fig. 3. Open table speed loop shows the functional principle of a table speed controller.

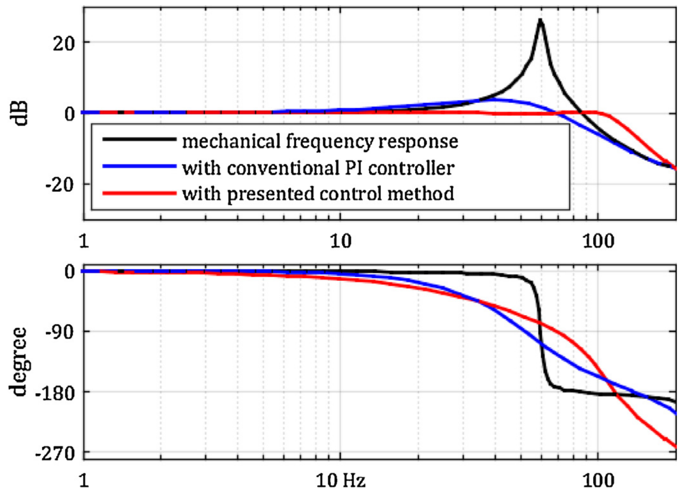

Fig. 4. Simulated effectiveness of the presented control method.

The additional friction in reality, however, has always a positive influence on the system stability due to its damping behaviour.

In order to simplify the derivation procedure, the motor speed here is controlled by a proportional controller. The tracking performance from the reference speed v_r to the table speed v_2 will be analyzed, since it is directly enclosed in the position loop and its behaviour dominates the selection of the K_v factor. The transfer function from v_r to v_2 is written as:

$$\frac{v_2}{v_r} = \frac{ds + c}{(m_1 m_2/(m_1 + m_2)K_p)s^3 + ((d/K_p) + m_2)s^2 + ((c/K_p) + d)s + c}. \tag{2}$$

The root locus of Eq. (2) is plotted in Fig. 2. The K_p factor is typically chosen as a high value (red curves of Fig. 2) for sufficient dynamic stiffness against disturbances. With the new control method, the proportional gain K_p is set lower than the first natural frequency (blue curves of Fig. 2). As a result, the transfer function Eq. (2) can be approximated to Eq. (3).

$$\frac{v_2}{v_r} \approx \frac{K_p}{s + K_p} \cdot \frac{ds + c}{(m_1 m_2/(m_1 + m_2))s^2 + (d + m_2 K_p)s + c}, \quad K_p < \sqrt{\frac{c}{m_2}}. \tag{3}$$

The second multiplier describes a mechanical PT2 system. However, its resonance frequency is no longer $(c/m_2)^{0.5}$ but increased by the factor of the inertia ratio.

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

Besides the increasing of the resonance frequency, the weak control gain leads to a PT1 behaviour (the first multiplier of Eq. (3)), whose cut-off frequency is much lower than ω_r. Consequently, the bandwidth is still limited. To counter this limitation, an additional PI controller is superimposed outside with the feedback of the table speed.

Fig. 3 shows the functional principle of the table speed controller $v_{r,1}/(v_{r,2} - v_2)$, where the PT2 behaviour of Eq. (3) is neglected. Blue and red curves describe the Bode plot of the PI controller and the PT1 system respectively. It can be seen that the integral part of the PI controller raises the amplitude and the phase of the PT1 system by setting the integral gain K_{iv} with the same value like the proportional gain of the motor speed controller K_p:

$$G_{PI} \cdot G_{PT1} = K_{pv}\left(1 + \frac{K_{iv}}{s}\right) \cdot \frac{K_p}{s + K_p} = \frac{K_{pv} \cdot K_p}{s}, \tag{5}$$

when $K_{iv} = K_p$. The bandwidth of the summarized system (black lines) can therefore be increased by the proportional part K_{pv} of the controller. The value of K_{pv} or the bandwidth of the table speed loop is limited by the second part of Eq. (3), with its critical frequency shown in Eq. (4).

The effectiveness of this control principle can be seen by simulation in a Bode plot. Fig. 4 shows the mechanical frequency response (black) and the frequency response from the table reference speed to the actual speed. For the system with the conventional PI controller (state of the art, blue), the table reference speed is equal to the motor reference speed. Compared to the state of the art method, the new method (red) reduces the amplitude peak and increases the critical frequency, where the phase crosses -90°.

3. Experimental validation

The experimental setup is depicted in Fig. 5. A table is driven by a ball screw with 40 mm pitch and 40 mm diameter without any additional gear ($i = 1$). The primary section of a linear motor is integrated into the table for the generation of disturbance force. The equivalent mass of the rotational parts is 162 kg. The table with the load can be varied between 260 kg and 600 kg. All the measurements, except the robustness tests (Section 4), were conducted with a nominal table mass of 430 kg. The test bench was controlled using a dSPACE system, where the