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Accurate control of ball screw drives using pole-placement vibration damping and a novel trajectory prefilter

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A B S T R A C T

This paper presents a pole-placement technique to achieve active vibration damping, as well as high bandwidth disturbance rejection and positioning, in ball screw drives. The pole-placement approach is simple and effective, with an intuitive physical interpretation, which makes the tuning process straightforward in comparison to existing controllers which actively compensate for structural vibrations. The tracking performance of the drive is improved through feedforward control using inverted plant dynamics and a novel trajectory prefilter. The prefilter is designed to remove tracking error artifacts correlated to the velocity, acceleration, jerk and snap (fourth derivative) of the commanded trajectory. By applying the least-squares method to the data from a single tracking experiment, the prefilter can be tuned quickly and reliably. The proposed controller has been compared to the P-PI position–velocity cascade controller commonly used in industry. The controller design, stability analysis and experimental results are discussed in the paper.

1. Introduction

In order to meet industry demands for improved productivity and part quality, machine tools must be equipped with faster and more accurate feed drives. Over the past two decades, research has focused on the development of new control strategies and smooth trajectory generation techniques. These developments, along with advances in actuator and sensor technology, have greatly improved the motion delivery accuracy in high speed machine tools.

In ball screw drives, the high accelerations required for high speed motion often excite the structure's axial and torsional vibration modes. These modes also place a limit on the achievable closed-loop bandwidth, thus limiting the positioning and trajectory tracking accuracy. Several strategies for dealing with these vibrations have been proposed in literature, including prefiltering (i.e., input shaping) of the motion commands [1,2] and the control signal [3], as well as inserting notch filters inside the loop [4]; in order to prevent the excitation of structural modes. Notch filtering can provide a modest increase in the bandwidth, by recovering loop stability through attenuation of the sharp gain increase due to resonance. However, this method also comes with the cost of extra phase delay prior to the notch frequency, which poses another stability limitation on the achievable closed-loop bandwidth. Another issue common to input shaping and notch filtering is the requirement to have approximate knowledge about the natural frequency, and sometimes the damping ratio.

In search of more robust and effective designs, recent research has focused on actively damping the vibrations inside the control loop. Various techniques have been investigated, such as H_∞ control [5], adaptive sliding mode control [6,7], observer-based state feedback [8,9], and predictive control [10]. These techniques generally allow the closed-loop bandwidth to be increased further, compared to using notch filters inside the loop. However, the design becomes more challenging since the vibratory dynamics needs to be considered as part of the plant model. In addition, most of the advanced techniques like H_∞ control and adaptive sliding mode control require a fair bit of familiarity with advanced level theory, in order to be able to design and implement such controllers successfully.

There have also been practical solutions proposed, such as vibration cancellation inside the position or velocity feedback loops [11,12], or placing zeros nearby oscillatory poles, in order to shift the root locus away from problematic regions [13]. The latter is applicable when the closed-loop bandwidth is significantly lower than the frequency of the vibratory dynamics; which opposes the goal of maximizing the responsive frequency range of feed drives.

In addition to a robust feedback loop capable of attenuating disturbances and structural vibrations, it is also necessary to design an appropriate feedforward controller and/or a trajectory prefilter which improves the command following properties of the drive

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system. The role of this prefilter is different than the vibration avoidance function of input shaping prefilters mentioned earlier on [1,2]. Rather, the objective is to widen the bandwidth of the tracking transfer function, and bring its phase closer to zero, for as wide a frequency range as possible. Several feedforward controllers and prefilters have been proposed for this purpose in literature, which rely on stable model inversion and/or elaborate analytical solution or optimization techniques [14–18].

In industrial application, the tracking transfer function is usually improved through the use of velocity and acceleration feedforward terms, which help to mitigate the tracking error due to inertial and viscous friction forces. This scheme can still leave an error profile that is correlated with higher order derivatives of acceleration. Observing this, Boerlage et al. [19] have proposed using acceleration and the derivative of jerk (i.e., snap) in their model inversion scheme.

In this paper, we propose a control scheme which utilizes pole-placement as the main feedback design. Trajectory tracking is improved by: (1) Compatible state command generation between the translational motion of the table and rotational motion of the screw; (2) Inversion of the drive dynamics; and (3) A novel trajectory prefilter which removes correlations of the velocity, acceleration, jerk, and snap commands from the tracking error. To improve the motion accuracy at velocity reversals, feedforward friction compensation is also added.

While the combination of feedback control, plant inversion and trajectory prefiltering has been suggested before in literature, the main differences and advantages of the proposed design can be summarized as follows:

1. *The pole-placement controller (PPC) not only achieves position control, but also actively dampens out vibrations of the first axial mode*. As demonstrated in Sections 3 and 4, doing so allows higher control bandwidth to be achieved compared to conventional designs which only consider rigid body motion, like P-PI position–velocity cascade control.
2. *Compared to [5–10], the proposed PPC is very simple to tune and implement*. Of the modeled system states, two of them are the rotational and translational position of the screw and table, which are readily measurable on most ball screw drives. The other states are constructed by either differentiating or integrating these measurements with respect to time. Hence, there is no need for a state observer. Also, a simple and effective tuning strategy is proposed for selecting the desired pole locations. The tuning is done concurrently with graphical frequency domain analyses, which help to ensure that unmodeled vibration modes and other dynamics, like delays, are implicitly taken into account using the drive's measured frequency response.
3. *In contrast to [6,14,15,17,18], design of the proposed trajectory prefilter is very simple and can be realized automatically, using data from a single tracking experiment*. The coefficients for the velocity, acceleration, jerk, and snap commands are computed by treating these profiles as regressors in a least-squares solution that attempts to reconstruct the tracking error. By using direct experimental data, deficiencies in the inversion due to modeling errors are automatically remedied. While the proposed prefilter has some similarity to [19], the key differences are that our prefilter inverts the closed-loop dynamics rather than the open-loop plant, and that velocity and jerk correlations can be considered, in addition to acceleration and snap.
4. *The proposed scheme is significantly more effective in disturbance rejection and command tracking, compared to the current industrial solution of P-PI position–velocity cascade control with velocity and acceleration feedforward terms*. This is demonstrated in simulation and experimental results in Sections 3 and 4.

Henceforth, the paper is organized as follows. The ball screw modeling, controller design and the trajectory prefilter are presented in Section 2. Simulation results and frequency domain analyses, comparing the proposed scheme to the current industrial solution of P-PI cascade control, are discussed in Section 3. Section 4 presents experimental results that compare the two schemes in rope snap experiments as well as high speed tracking and machining tests. Finally, the conclusions are presented in Section 5.

2. Controller design

The proposed overall control scheme is shown in Fig. 1. The feedback loop is closed using pole-placement control, which provides active vibration damping for the axial mode, as well as position control. Feedforward terms, including inverted system dynamics and Stribeck-type friction compensation are included in order to improve command tracking. A new trajectory prefilter has been developed, which removes the artifacts of the commanded velocity, acceleration, jerk and snap from the tracking error. Finally, a filter pack comprising of notch and low-pass filters is added to ensure stability. In the following subsections, the design of the individual components is explained. Their individual contributions are validated through time domain simulations and frequency domain analyses in Section 3.

2.1. Drive model

The ball screw drive test bed is shown in Fig. 2. The table is supported by an air guideway system and driven by an NSK W2010FA-3P-C5Z20 precision ball screw with 20 mm pitch and 20 mm diameter. A 3 kW AC servomotor provides actuation through a diaphragm type coupling. The setup was built to facilitate research on control laws which achieve active vibration damping on feed drives. Hence, no external passive dampers are attached to the table. Feedback is provided by a high-resolution rotary encoder which is mounted on the ball screw, and a linear encoder mounted on the table. The rotary encoder produces 5000 sinusoidal signals per revolution which can be interpolated by 400 times, giving a position measurement resolution equivalent to 10 nm of table motion. The catalogue rated accuracy of this encoder is equivalent to 200 nm of table motion. The linear encoder, which has a rated accuracy of 40 nm, has 4 μm signal period and is also interpolated 400 times, providing a measurement resolution of 10 nm. The control laws are implemented at 20 kHz sampling frequency on a dSpace system. The maximum encoder line count frequency for the dSpace encoder board is 750 kHz, allowing travel speeds up to 3 m/s (= 4 μm × 750,000 1/s) to be measured. The consistency of the sampling period was validated to be around 0.15% of its nominal value (in terms of standard deviation), by monitoring the DAC output signals with an oscilloscope while running the control law.

The controller is designed considering the simplified dynamic model shown in Fig. 3, which is captured by Equation (1):

$$\begin{aligned} m_1\ddot{x}_1 &= -b_1\dot{x}_1 + k(x_2 - x_1) + c(\dot{x}_2 - \dot{x}_1) + u + d_1 \\ m_2\ddot{x}_2 &= -b_2\dot{x}_2 + k(x_1 - x_2) + c(\dot{x}_1 - \dot{x}_2) + d_2 \end{aligned} \quad (1)$$

As shown in earlier work [6], this model captures the dynamics of the *first* axial mode with sufficient closeness. If necessary, more accurate models can also be used with the proposed PPC scheme, such as those presented in [18,20] which have a full mass matrix and take the form:

$$\begin{bmatrix} m_{11} & m_{12} \\ m_{12} & m_{22} \end{bmatrix}\ddot{x} + \begin{bmatrix} c_{11} & c_{12} \\ c_{12} & c_{22} \end{bmatrix}\dot{x} + \begin{bmatrix} k & -k \\ -k & k \end{bmatrix}x = F \quad (2)$$