Comparison of Disturbance Suppression for Servo Drives

S. Doenitz
Darmstadt University of Technology
Department of Power Electronics and Control of Drives
Landgraf-Georg-Str. 4
64283 Darmstadt
Germany
tel.: +49 / (0)6151 / 162366
fax.: +49 / (0)6151 / 162613
email: sdoenitz@srt.tu-darmstadt.de

Abstract:

This paper compares two methods for estimation and compensation of acting disturbance force of a linear position controlled servo drive. Both methods have been combined with common control structures like cascade and state space control. The final paper will give experimental results of a test stand with a disturbance force actuator.

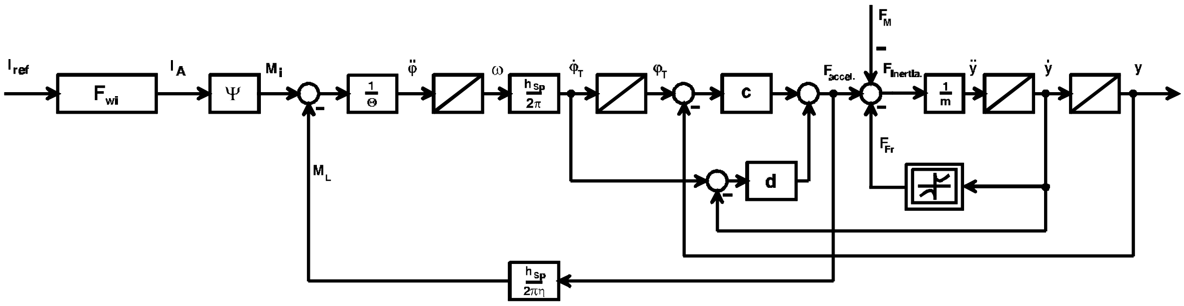
Summary:

1 Introduction

The main item in servo control applications is position control, which is decisive for the accuracy to be obtained. The classical cascade control of position, velocity and current is state of the art, but nowadays state space controller have an increasing significance. Fig. 1 shows the typical model of a plant of a servo system consisting of a mechanical two mass system and a closed current control loop. As measuring signals are at disposal the position signal y, the rotary angle φ and the armature current I_A. Shaft velocity and slide velocity can be calculated by using the position signals. With these values it is simple to realize either the classical cascade control or a state space control.
The accuracy of a position controlled system depends among other things, like controller gains and control structure, on currently acting disturbance forces. Such forces are friction forces, which act very closely at the load, and machining forces, which include reactions from the machining process to the drive. According Fig. 1 the disturbance forces F_{Fr} and F_M act on the last sum point, that means a changing of these forces will be detected by the position and velocity feedback. The current controller only influences the torque of the motors. To maintain stiff position control in order to suppress load disturbances implies the need for large controller gains which trend to reduce system stability. Therefore methods for disturbance compensation are very important [3],[4].

2 Compensation with explicit estimation of disturbance forces

Fig. 2 shows the principle of force estimation



F_{accel} - accelerating force	I_A - drive current	Θ - inertial moment of rotor and screw spindle
F_{Fr} - friction force	I_{Ref} - drive current reference value	m - load mass
$F_{inertia}$ - inertial force	y - load position	Ψ - flux
F_M - machining force	φ^T - transposed rotor angle	c - elastic constant
F_{wi} - closed current loop	φ - rotor angle	**d** - damping constant
M_i - motor torque	ω - mechanical angular frequency	η - transmission efficiency of the screw spindle

Fig. 1: Block diagram of the process

and compensation, which was introduced in [1]. The measured values of the positioning system are fed into the "block estimation of parameters", which calculates guessed model parameters c*, d*, m* by using a least-square-method, and the blocks "derivation", which calculate the two velocities and the slide acceleration. The equation for calculating disturbance forces can be read directly from the force sum point in Fig. 1.

$$F_{dist} = F_{Fr} + F_M$$
$$F_{dist} = c*(j_T - y) + d*(\dot{j}_T - \dot{y}) - m*\ddot{y} \qquad (1)$$

The task of the digital filter is the damping of noise caused by the derivation of the position signals and natural frequencies of the mechanical part of the process overlaying the signal of identified forces. The block "feedforward compensation" decides about the value of the compensation signal and whether the signal will be switched on.

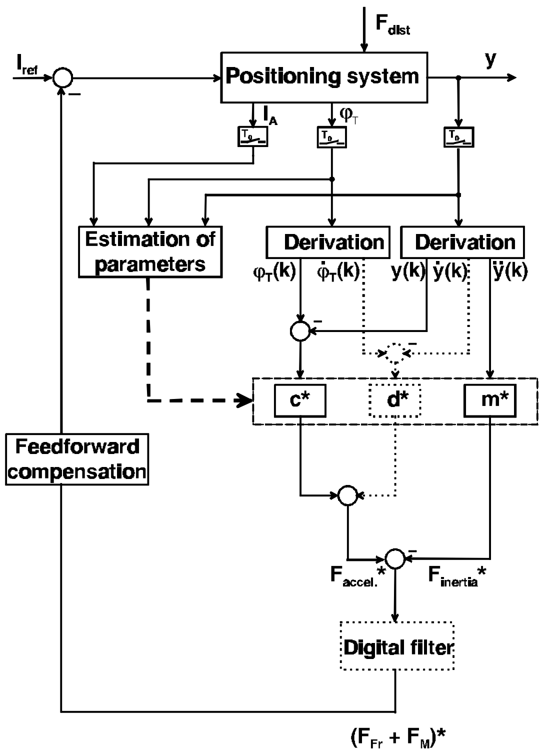


Fig. 2: Explicit disturbance force compensation

3 Compensation with disturbance observer

Another solution to estimate disturbances is the realization of a disturbance force observer which can be seen in Fig. 3. Equation (3) describes the structure of the observer according [2] by using a simple disturbance differential equation (2) and the description given in equation (1).

$$\dot{F}_{dist} = 0 \qquad (2)$$

The problem is to find the right choice of the observer parameters l_0, l_1, l_S.

$$\begin{bmatrix}\dot{x}\\ \dot{y}\\ F_{dist}\end{bmatrix} = \begin{bmatrix}0 & -\frac{c}{m} & -\frac{1}{m}\\ 1 & -\frac{d}{m} & 0\\ 0 & 0 & 0\end{bmatrix} \cdot \begin{bmatrix}x\\ y\\ F_{dist}\end{bmatrix} + \begin{bmatrix}\frac{c}{m}\\ \frac{d}{m}\\ 0\end{bmatrix} \cdot j_T + \begin{bmatrix}l_0\\ l_1\\ l_s\end{bmatrix} \cdot (y - \hat{y}) \qquad (3)$$

Good results delivers a linear quadratic regulator design with solution of algebraic Riccati equation, in which weight factors for l_0, l_1 are chosen equally and the weight factor for l_S has to be some powers higher.

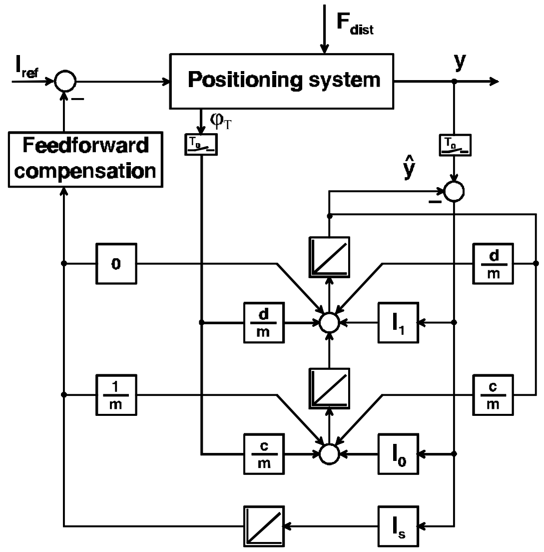


Fig. 3: Compensation with disturbance force observer

The block "feedforward compensation" also decides about the compensation value in dependence on the current state of slide.
A filter to reduce noise is not necessary due to the integrator which delivers the disturbance signal. That also means that this signal is not useful for diagnostic purposes.

4 Results

Fig. 4 and Fig. 5 show simulation results which were achieved with a classical cascade control structure by using a P position controller and a PI speed controller and a state space control structure. All simulations have been made with a stick slip friction characteristic and constant machining force between 0.3s and 0.7s. The position set value has been chosen so that the system operates on the negative slope of the friction characteristic.
As reference curve has been selected the position error for cascade control in Fig. 4.b and in Fig. 5.b for state space control. The controller gains have been determined in a way that both structures have the same

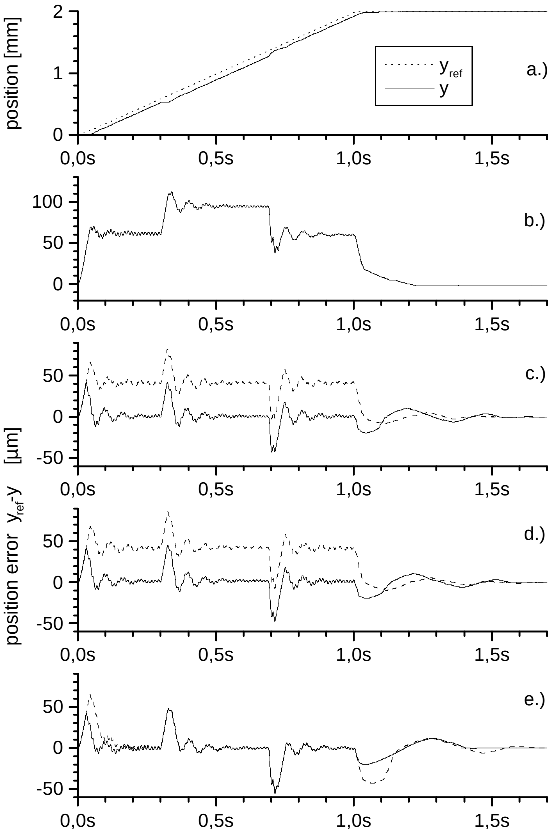


Fig. 4: Cascade control
with speed forward compensation
without speed forward compensation

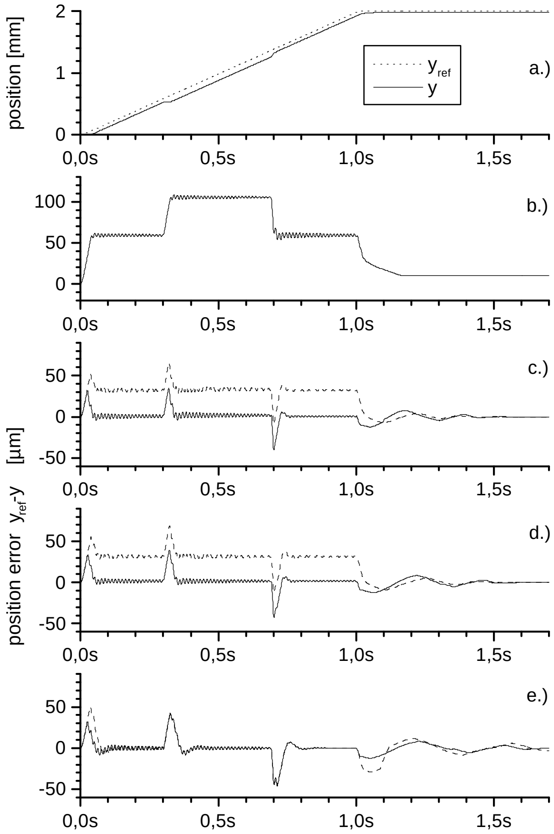


Fig. 5: State space control
with speed forward compensation
without speed forward compensation

following error of about 60µm without machining forces. That leads to oscillations within the cascade control loop on one hand and on the other hand the state space controller gains are not as huge as possible. In figures Fig. 4.a and Fig. 5.a are shown the shapes of set and actual position for that case. Figures Fig. 4.c and Fig. 5.c show the position error by using the compensation method according chapter 2 and figures Fig. 4.d and Fig. 5.d depict the results which were achieved with the disturbance observer according chapter 3. At last in Fig. 4.e and Fig. 5.e is presented the position error by using a PI position controller without any compensation method.

All control structures are able to supress disturbances especially the compensation methods according to chapter 2 and 4 in connection with a state space control show good results. The differences between the explicit disturbance compensation and the disturbance observer are small in simulations and it remains to examine this in praxis. By using a PI position controller only the combination with speed feedforward compensation is useful due to the large amplitude of overshooting around the kink of the reference position. All simulations underline the better response of the state space control structure. Especially if higher weight factors for the position in controller design can be used as in the simulations above additionally a reduction of system oscillations occurs.

The final paper will present experimental results of a test stand with disturbance force actuator and a dc motor driven machining slide. Advantages and disadvantages will be discussed in more detail.

5 References

[1] Singer, B.: A new method of identification and compensation of friction and machining forces in positioning systems, EPE 1993, p. 327-331

[2] Föllinger: Regelungstechnik, Hüthig Verlag, 8.Auflage, 1994

[3] Serri, A.: Speed control improvement by torque observer for electrical drives, EPE 1995, p. 3.475-3.480

[4] Kokes, M.: Vorrichtung und Verfahren zur Regelung von Systemen, deren Störgrößen einen zeit- oder winkelperiodischen Anteil enthalten, patent no. DE 4026091 A1, 1990