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DISCRETE-TIME SLIDING MODE CONTROL OF A DC MOTOR AND BALL-SCREW DRIVEN POSITIONING TABLE

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Abstract: A discrete-time sliding mode control design for a ball-screw driven positioning table is presented. The basic feature of this design is that high speed and high positioning accuracy can be met despite of the fact that the controlled process suffers from friction and mechanical flexibility. Disturbance rejection, in the form of friction compensation and vibration suppression, is the main focus of the paper. Apart from this, the robustness of the proposed controller with respect to model uncertainties is also considered. Experiments on the Y-axis of a Mydata surface mount robot show consistent and robust performance under different load conditions.

Keyword: discrete-time, sliding mode, two-mass system, friction compensation, vibration suppression

1. INTRODUCTION

Positioning systems are used e.g in various phases of microelectronics manufacturing. One application that is targeted for the research reported here is surface mount robots (SMR), in this case represented by Mydata assembly machines for mounting electronic components on circuit boards. The Y-axis of the machine uses a typical ball-screw transmission driven by a DC servo motor to position the circuit boards. For electronic systems manufacturing, high positioning accuracy is required in consequence of the reduced size of modern electronic components, and the high operational speed is desired for achieving high productivity. In addition, robustness must be considered not only for stability but also for performance, since differences in parameters among individual machines represent uncertainties, e.g. uncertainty in friction parameters. Further, different circuit boards represent different loads, and the mass of a particular board changes as components are mounted. In other words, the same controller settings should meet the control specification for all machines and for varying loads, i.e., without individual tuning.

In this application, the major obstacles for high performance are caused by the uncertainties due to friction and mechanical flexibility. Friction, which represents a complicated nonlinear function at low velocities, is the main source of position inaccuracy. Friction effects are usually reduced by introducing a compensator based on an identified friction model. However, an accurate friction model is usually difficult to obtain because of its characteristics of time-variation during changing environmental parameters. Further, in sample-data systems, the accuracy of the friction model and updating speed are highly questionable because of the limitation on sampling rates. The second problem in the system is the uncertain dynamics owing to the inherent flexibility of the machine structure, i.e., the finite stiffness of the long ball-screws as well as of other compliant links. This flexibility is a destabilising factor in the feedback control loop, resulting in substantial limitation of the control bandwidth.

Sliding mode control (SMC) has gained significant interest in recent years due to its superb characteristics in terms of insensitivity to large parameter variations and its capability in disturbance rejection. The concept of discrete-time sliding mode (DSM) (Utkin, 1994) was introduced for the purpose of implementing SMC in sampled data systems. The discrete-time sliding mode control (DSMC) with one step delayed disturbance compensation (Su, *et al.*, 1993) provides an excellent method for disturbances rejection and chattering attenuation. Li & Wikander (2000) have shown that the DSMC is able to compensate unknown friction in positioning systems, despite of the complicated characteristics of friction.

However, the DSMC can not be directly applied to the Y-axis ball-screw system, since the controller excessively excites the mechanical resonance of the process. Many previous researchers have studied the application of the SMC in flexible systems. For example, frequency shaped sliding mode (FSSM) was introduced by Young & Özgüner (1993); H-infinity and μ synthesis based sliding mode control was studied by Nonami, *et al.* (1996); These methods actually introduce a low pass filter with an appropriate cut-off frequency into sliding surfaces, thus high frequency vibrations due to parasitic dynamics as well as their interactions with sliding mode dynamics can be suppressed in the desired frequency band. However, the weakness of such a method is that the transient response may be slowed down to an unsatisfactory degree. To reduce this problem, the combination of a FSSM and a terminal sliding mode control was proposed by Xu & Cao (2000). Other methods can also be found in the literature, such as sliding mode with perturbation estimation and shaped sliding surface (Moura, *et al.*, 1997), sliding mode with shaped command input (Jalili & Olgac, 1998; Singh, 1994) and sliding mode with time-varying or nonlinear sliding surface (Hara & Yoshida, 1996; Li, *et al.*, 1999).

The purpose of this paper is to design a robust servo positioning controller based on DSM for the Y-axis of the Mydata SMR. Regardless of model uncertainties and nonlinear friction, the motor positioning error should be within the range of the measurement resolution while minimizing the excitation of the table vibrations. The remaining sections are organized as follows. The dynamic model of the compliant system and the control problems are presented in section 2. The design of a DSMC and disturbance compensation are given in section 3. In section 4, the method for suppressing vibrations due to unmodeled dynamics is proposed, and the experimental results are presented in section 5. Finally, a conclusion follows in section 6.

2. SYSTEM AND CONTROL PROBLEM DESCRIPTION

2.1 The plant description

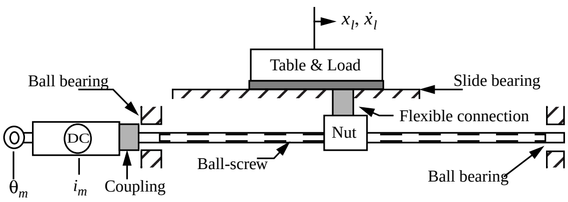

Fig.1. The Y-axis of the surface mount Robot (SMR).

A sketch of a Y-axis of an SMR is shown in Fig. 1, the rotation of a current controlled DC motor is converted into a translational motion by a high precision ball-screw. A slide table attached to the ball-nut carries the load at high velocities. The total moving range along the Y-axis is 1.2 meters. Since the stiffness of the ball-screw and of the connection between the nut and table are finite, the process is simplified as a two-mass system:

$$\bar{J}_m\ddot{\theta}_m + b_m\dot{\theta}_m = K_m i_m - T_l - T_f \qquad (1)$$
$$T_l = k_t(\theta_m - x_l/p)$$
$$\bar{m}_l\ddot{x}_l + b_l\dot{x}_l = T_l/p$$

where the variables θ_m, x_l, are the motor angle and table displacement respectively; i_m is the motor current and T_l the load torque due to the torsion of the ball-screw; T_f is the friction torque acting at motor side. Disturbances acting at the load side are assumed zero in this application. The remaining parameters and their values are given in TABLE 1.

TABLE 1. Parameters of the SMR

	Parameters	Value
J_m	Motor inertia	1.40x10^{-4} (kgm^2)
J_a	Ball-screw inertia	1.70x10^{-4} (kgm^2)
m_n	Nut mass	0.633 (kg)
m_t	Table mass	4.750 (kg)
m_l	Load mass	0~10 (*kg*)
b_m	Damping coefficient	0.003 (*Nm/rad/s*)
k_t	Stiffness	15~20 (*Nm/rad*)
K_m	Torque constant	0.356 (*Nm/A*)
p	Screw pitch	p=0.0064 (*m/rad)*
$\bar{J}_m$	The equivalent motor inertia	$J_m + J_a$ (kgm^2)
$\bar{m}_l$	The equivalent load mass	$m_n + m_t + m_l$ (*kg)*

Clearly, a two-mass model for the distributed masses is a quite coarse simplification of the physical process, and thus some of the model parameters, e.g. the equivalent stiffness are not easily determined from physical parameters. With a 10kg load on the table, the process spectrum analysis shows that the mechanical resonance of the process is in the range of 200-600 rad/s, implying the equivalent stiffness in the range of 15~20 (*Nm/rad*). Moreover, the frequencies of the resonance peaks change with table position, indicating the dynamics uncertainty of the process. In addition, measurements also show that the friction is not only velocity dependent, but also position dependent on a macro scale. Therefore, the main topic of this paper is to design a DSM controller which not only achieves high positioning accurate, but which also is robust against these uncertainties and friction characteristics.

In the SMR, only a single position feedback device, an optical encoder on the motor side, is provided. Therefore, the nominal model used in the controller design utilise merely the first equation of (1) and the the load torque due to torsion is treated as a disturbance. Converting the motor angle position to linear position and rewriting the equation in the state space form gives:

$$\dot{x}_c = Ax_c + B(u_c - d) \qquad (2)$$

$$A = \begin{bmatrix} 0 & 1 \\ 0 & -a \end{bmatrix},\ B = \begin{bmatrix} 0 \\ b \end{bmatrix} \qquad (3)$$

where $x_c = [p\theta_m \ \ p\dot{\theta}_m]^T$, $u_c = i_m$, $a = b_m/\bar{J}_m$ $b = pK_m/\bar{J}_m$ and $d = (T_l + T_{f1})/(pK_m)$ is the lumped disturbance of the load torque, friction and other external disturbances. The parameters of the second mass may not be known exactly due to the load uncertainty, however, knowing the resonance frequency of the process is necessary since it provides useful information for the later control design.

2.2 Control problem formulation

The primary goal of the considered control application is high speed point-to-point positioning. However, due to the characteristics of the process, care must be taken not to induce structural vibration through bang-bang type of control signals. In state space control design, the states of the trajectory can be generated by a reference model which was named an x_d-generator in Misawa (1997). The x_d-generator has the form of

$$\dot{x}_d = Ax_d + Bu_d \tag{4}$$

The dynamics of the x_d-generator is obtained from the nominal process through feedback, i.e., the hypothetical u_d is designed by

$$u_d = -L_f x_d + \rho y_d \tag{5}$$

where L_f is the feedback gain which can be determined by any appropriate design method such as pole placement or LQR control. y_d is the command distance and ρ is a scalar in SISO system. The generated output of interest should satisfy the desired control specification. The significance of using *the* x_d-generator is that the vector x_d is consistent with the nominal system described by the pair (A, B). Thus the control problems are converted to regulation problems in the error state space with the same matrices A and B. Subtracting (4) from (2) gives the error dynamics:

$$\dot{x} = Ax + Bu - Bd \tag{6}$$

where $x = [x_1, x_2]^T = x_c - x_d$, x_1, x_2 are position and velocity error, respectively; $u = u_c - u_d$.

3. DSM CONTROL DESIGN

Denoting $x_k = x(kT), u_k = u(kT)$ and $d_k = d(kT)$ where T is the sampling interval, the ZOH discrete-time model of the error dynamics is:

$$x_{k+1} = \Phi x_k + \Gamma(u_k - d_k) \tag{7}$$

$$\Phi = e^{AT}, \Gamma = \int_0^T e^{A\tau} d\tau B, \tag{8}$$

$$d_k = \int_0^T e^{A\tau} d((k+1)T - \tau) d\tau \tag{9}$$

The magnitude of Γ and d_k are both $O(T)$ if $d(t)$ is matched and satisfy the boundedness and smoothness conditions (Su, *et al.*, 2000).

Define the discrete-time sliding surface:

$$S = \{s_k | s_k = \Lambda x_k = 0, \quad k = 0, 1, \ldots\} \tag{10}$$

where, $\Lambda = [\lambda \;\; 1]$, λ is a positive real number and $\Lambda\Gamma$ is invertible. Using the definition that the equivalent control u_k^{eq} is the solution of $s_{k+1} = 0$ (Utkin, 1994) gives that

$$u_k^{eq} = -(\Lambda\Gamma)^{-1}\Lambda\Phi x_k + d_k \tag{11}$$

3.1 The DSM with one-step delayed disturbance compensation

The above u_k^{eq} can not be directly realized because the disturbance d_k is unknown. An effective disturbance compensation method was used in Su, *et al.* (1993, 1996, 2000) and Young, *et al*. (1999) to solve this problem, i.e., using the one step delayed disturbance d_{k-1} to approximate the current disturbance d_k. The approximate disturbance d_{k-1} can be easily calculated from equation (7) as follows,

$$d_{k-1} = -(\Lambda\Gamma)^{-1}\Lambda x_k + (\Lambda\Gamma)^{-1}\Lambda\Phi x_{k-1} + u_{k-1} \tag{12}$$

The control law is then in the form of

$$u_k = -(\Lambda\Gamma)^{-1}\Lambda\Phi x_k + d_{k-1} \tag{13}$$

By applying (13) to (7), the closed-loop dynamics are described by

$$x_{k+1} = (I - \Gamma(\Lambda\Gamma)^{-1}\Lambda)\Phi x_k - \Gamma(d_k - d_{k-1}) \tag{14}$$

Obviously, if $d_k - d_{k-1} = 0$, the error vector will asymptotically converge to zero as long as the matrix $(I - \Gamma(\Lambda\Gamma)^{-1}\Lambda)$ is contractive. Furthermore, the eigenvalues can be assigned arbitrarily by selecting a suitable Λ. In the case of $d_k - d_{k-1} \neq 0$, since both Γ and d_k are of the order $O(T)$, the steady state error of (14) is $O(T^2)$ and it can be characterized by

$$s_{k+1} = -\Lambda\Gamma(d_k - d_{k-1}) \tag{15}$$

i.e., $s_{k+1} = O(T^2)$. Taking also into account the actuator saturation value u_{max}, Utkin (1994) has proved that the control

$$u_k = \begin{cases} u_k & \text{if } \|u_k\| \le u_{max} \\ u_{max}\dfrac{u_k}{\|u_k\|} & \text{if } \|u_k\| > u_{max} \end{cases} \tag{16}$$

is able to force the system into the $O(T^2)$ boundary layer of the sliding surface.

Note that when the term $d(t)$ contains nonlinear friction, the smoothness of $d(t)$ will not be satisfied in the vicinity of zero velocity, but away from zero velocity, $d(t)$ can simply be considered as a continuous and/or a slow time-varying disturbance. Fortunately, in this application, zero velocity happens only at motion start and final position reaching phases. According to the control specifications, friction must be handled in the reaching phase such that the DSMC brings the system (7) to the sensor resolution vicinity of the origin and keeps it there. Young (1998) revealed that in continuous-time SMC, the discontinuous control may work cooperatively with Coulumb friction and drive the system in accordance to hierarchy of sliding mode $x = 0 \rightarrow \dot{x} = 0$. In Li & Wikander (2000), it has also been experimentally shown that the discrete-time sliding mode is realizable with a small enough sampling time. In that paper, a DSMC with the control law (16) was used to perform a point-to-point positioning control and to cope with large friction. When the sampling time was decreased to $T \le 0.012s$, the unknown friction was well compensate and precision positioning was achieved.

4. VIBRATION SUPPRESSION

Note that the DSMC designed in the previous section is based on the nominal model (2) which assumes the process being rigid, and d_{k-1} in the control law (13) represents the estimated disturbances. Care should be taken when the controlled process contains unmodeled dynamics. Due to flexibility, the term d_{k-1} will be not only friction and low frequency disturbances due to parameter uncertainty, but also potentially high frequency disturbances due to the unmodeled dynamics. Directly feeding d_{k-1} to the input may magnify those high frequency components thus resulting in high frequency oscillations in the closed-loop. In order to suppress the vibrations, it is here proposed that a disturbance compensation filter is added to the control (13). I.e., a discrete-time low pass filter $Q(q)$ is introduced to filter the signal d_{k-1} and the control law becomes

$$u_k = -(\Lambda\Gamma)^{-1}\Lambda\Phi x_k + d_{fk} \tag{17}$$

where $d_{fk} = Q(q)d_{k-1}$. The state-space realization of the transfer function $Q(q)$ can be written

$$\begin{aligned} \eta_{k+1} &= A_z\eta_k + B_z d_{k-1} \\ d_{fk} &= C_z\eta_k + D_z d_{k-1} \end{aligned} \tag{18}$$

$\eta \in R^h$, h is the order of the filter and A_z, B_z, C_z and D_z are matrices with proper dimensions. The transfer function of the filter is then

$$Q(q) = C_z(qI - A_z)^{-1}B_z + D_z \tag{19}$$

Incorporating the past disturbance d_{k-1} and the auxiliary state η_k into the control is equivalent to induce additional $m+h$ order dynamics to the closed- loop system. The augmented system is

$$\begin{bmatrix} x_{k+1} \\ \eta_{k+1} \\ u_{k+1} \end{bmatrix} = \bar{\Phi}\begin{bmatrix} x_k \\ \eta_k \\ u_k \end{bmatrix} + \Gamma_1 d_k + \Gamma_2 d_{k-1} \tag{20}$$

where,

$$\bar{\Phi} = \begin{bmatrix} \Phi & 0 & \Gamma \\ 0 & A_z & 0 \\ -(\Lambda\Gamma)^{-1}\Lambda\Phi^2 & C_z A_z & -(\Lambda\Gamma)^{-1}\Lambda\Phi\Gamma \end{bmatrix}, \tag{21}$$

$$\Gamma_1 = \begin{bmatrix} -\Gamma \\ 0 \\ (\Lambda\Gamma)^{-1}\Lambda\Phi\Gamma + D_z \end{bmatrix}, \Gamma_2 = \begin{bmatrix} 0 \\ B_z \\ C_z B_z \end{bmatrix} \tag{22}$$

As long as A_z is stable, the stability of the augmented system can be proved with the following lemma.

Lemma 1: The eigenvalues of $\bar{\Phi}$ are eig$\{A_z\}$, eig$\{\Phi - \Gamma(\Lambda\Gamma)^{-1}\Lambda\Phi\}$ and 0.

This lemma can be proved in a similar way as in Su, *et al.* (2000).

It is clear that the filter introduces dominant poles into the closed-loop system. Therefore, with a proper cut-off frequency, the unmodeled dynamics can be effectively damped out by the filter. The error (15) now becomes

$$s_{k+1} = -\Lambda\Gamma(d_k - d_{f,k}) \tag{23}$$

Obviously, if $d_k = O(T)$, it is also true that $d_{f,k} = O(T)$, hence s_{k+1} is still within $O(T^2)$.

In the FSSM proposed by Young & Özgüner (1993), frequency dependent weighting matrices are introduced in the design of the sliding surface, i.e., the surface is given a low-pass character. In other words, the sliding surface dynamics is slowed down in order to avoid excitation of elastic vibrations; In the suppression method proposed here, the sliding surface is still treated as the intersection of the hyperplanes defined in the state space of the nominal process, and the unmodeled dynamics are suppressed by a low pass filter within the closed loop. The advantages of the proposed method are: a) the design is simpler and straightforward, b) it is easy to test the different roll-off frequency of the filter to satisfy the control specifications. Moreover, the transient response does not slow down as much as that in FSSM, since the filter damps out high frequency components of the process, which thus behaves more or less as a rigid body, enabling the sliding surface to keep its fast dynamics.

5. EXPERIMENTS

The proposed controller is verified in position control of the Y-axis of the SMR with the nominal parameters listed in TABLE 1. The resolution of the position encoder is 20,000 pulses per revolution, equivalent to 2µm per pulse in the translatory motion. The velocity signal is obtained from the difference of two consecutive position measurements. $T = 2ms$ is used as the sampling period for the model discretization and the controller design. The scheme of the implementation is shown in Fig. 2.

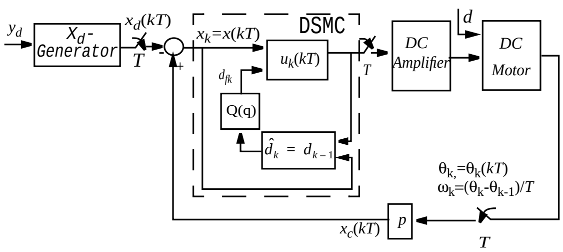

Fig.2. Block diagram of the proposed controller structure.

The parameters of the sliding surface are selected as $\Lambda = [50 \;\; 1]$, which implies that the closed-loop bandwidth is allocated at around 50 rad/s. Since the lowest resonant frequency of the process is around 200 rad/s, a first order low pass filter with cut-off frequency of 100 rad/s is used for the vibration suppression, the discrete-time transfer function of the filter is

$$Q(q) = \frac{0.0912(q+1)}{q - 0.8176} \tag{24}$$

A step command is first input to the x_d-generator that produces a smooth trajectory for the motion. Since the x_d-generator is acting as a prefilter in a 2 DOF control design, it is worth to consider the dynamics of the trajectory together with the dynamics of controlled system. For example, in order not to excite resonance oscillations, one useful method is to take away those frequency components from the input command signal (Singh, 1994). So the dynamics of the x_d-generator is expected to be equal or even slower than the dynamics of the DSMC controlled loop. Recall that the dynamics of the x_d-generator is obtained from the nominal model through feedback. The feedback gain can be obtained by placing the two poles at $-50 \pm 5i$, which gives $L_f = (0.2507\ 0.0068)$, and $\rho = 0.2507$ is the reciprocal of the steady state gain of the feedback controlled nominal model. The generated trajectory arrives at the desired position in about 0.12 s.

In order to test the robustness of the proposed controller, the experiments are first performed without load, i.e., $m_l = 0$ and then with load, $m_l = 10kg$. Moreover, the initial positions of the table are placed arbitrarily along the Y-axis. Fig. 4 and Fig. 5 show the experimental results of 10 *mm* positioning with the two load cases. In Fig. 4 and Fig. 5, the upper plots (a) show the positioning result without using the filter $Q(q)$. High frequency vibrations due to the unmodeled dynamics are observed in both cases. It is also seen in Fig. 5 that when the *10 kg* load is added, the resonant mode due to the torsion of the ball-screw is more obvious. The lower plots (b) show the same experiments with the filter $Q(q)$, and it can be seen that the vibration is well damped out and the position error falls into $\pm 2\mu m$ within 0.22s. The controller is also robust for small distance positioning. Fig. 3 displays the performance of a 1 *mm* positioning with 10 kg load. The slide table motion is observed with another linear encoder installed on the table. Fig. 6 shows the motion of the table with and without the filter, 10 kg load is put on the table. The effectiveness of the filter is clearly shown.

6. CONCLUSIONS

This paper has proposed a simple and effective control method for the positioning control of a ball-screw driven system. The algorithm is based on DSMC with disturbance compensation and vibration suppression. The design only requires knowledge of the first mass and the frequency range of the first resonant mode. Using the proposed control, the unknown friction in the system can be well compensated and all the high order vibrations due to the inherent flexibility of the system can be effectively damped out. The experiments demonstrate consistent performance under different load conditions and for different positioning disturbances, indicating that the proposed controller has strong robustness regarding stability and performance.

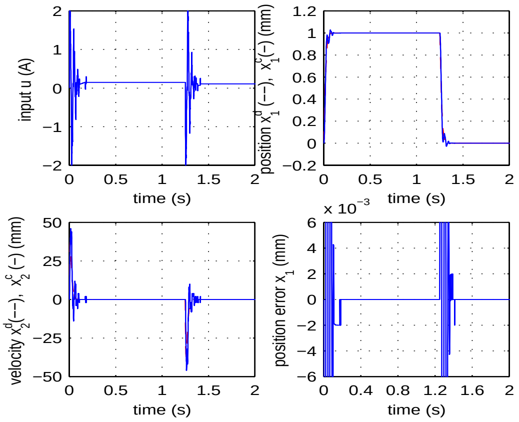

Fig.3. 1 *mm* positioning with the 10gk load and the filter.

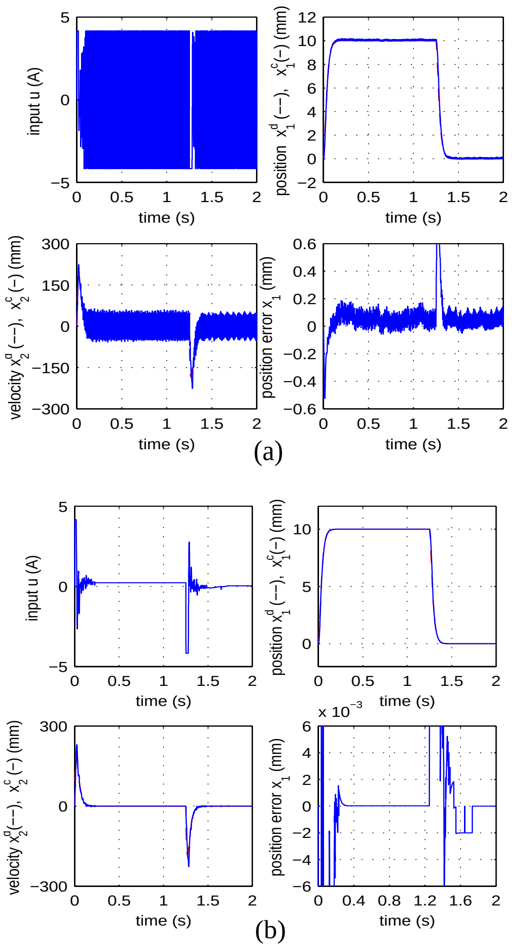

Fig.4. 10 *mm* positioning with the load $m_l = 0$; (a) without the filter, (b) with the filter. The dashed lines in the position and velocity plots are the reference states, the solid lines are the actual measured states.

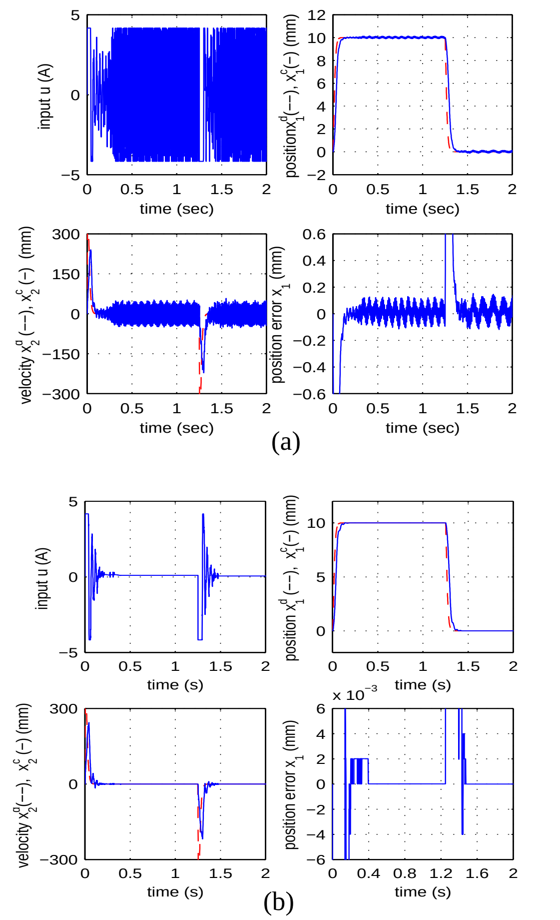

Fig.5. 10 *mm* positioning with the load $m_l = 10kg$; (a) without the filter, (b) with the filter. The dashed lines in the position and velocity plots are the reference states, the solid lines are the actual measured states.

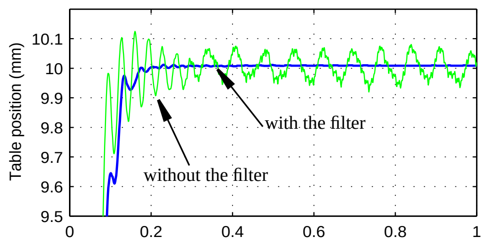

Fig.6. The table position with the 10kg load.

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