GLOBAL ASYMPTOTIC VELOCITY OBSERVATION OF NONLINEAR SYSTEMS
Application to a Frictional Industrial Emulator

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Abstract: Development of velocity observers for mechanical systems with friction deserves a special emphasis, because as evidenced in numerical and experimental tests when a state-of-the-art observer is armed, friction can induce high-frequency oscillations in the estimated signal. In this short paper, two new velocity-observation algorithms are designed, based on this previously reported observer, that eliminate the high-frequency oscillations noted. Numerical and experimental performance comparisons are carried through making use of LuGre model and a mechanical PID control system that incorporates the estimated velocity into the feedback loop.

1 INTRODUCTION

Velocity-dependent control laws such as PD, PID, and most robust control laws, among many others, theoretically require direct access to velocity. In reality, there are many applications in which this is not available either due to considerable manufacturing savings in cost, weight, and volume, or because the velocity measurements are highly contaminated with noise. In the latter case, for instance when measuring robot joint velocities, it may not even be desirable to do so (Arteaga and Kelly, 2004). Consequently, if the full-state information is missing, it is necessary to estimate the unmeasurable velocity through the use of an observer and feed it back into the controller. Such is the case with the frictional industrial emulator ECP model 220 used in our experiments. It has been documented when studying mechanical closed-loop control systems, that friction causes tracking errors, limit-cycles, and stick-slip motions, among other difficulties and usually unwanted phenomena (Armstrong-Hétlouvry et al., 1994). As evidenced in (Canudas de Wit and Fixot, 1991), (Canudas de Wit and Fixot, 1992), (Berghuis and Nijmeijer, 1993), and (Arteaga and Kelly, 2004) velocity observer design is a topic that has been and continues to be extensively studied. State observation of nonlinear inexact plants has been treated by utilizing discontinuous observers (Choi et al., 1999), (Xiong and Saif, 2001), and (Xian et al., 2004). However, little research focused on velocity observation of mechanical systems with friction at low velocities which, when incorporating existing observers, exhibit high-frequency oscillations in the velocity-estimation signal. A state-of-the-art, globally asymptotic, discontinuous velocity-estimation scheme for second-order mechanical systems has been presented in (Xian et al., 2004). Though very reliable, this observer is not specialized for mechanical systems in the presence of friction. Thus, a high-frequency component of the velocity observation is detected when numerical simulations and experimental testing are performed. The two newly proposed observers claimed here are an attempt to alleviate this unwanted oscillatory effect and try offer an increase in the observation reliability.

The remainder of this document proceeds as follows: Section 2 postulates two modified observers based on the velocity estimator stated in (Xian et al., 2004). The next section presents numerical simulations evidencing the performance of the observers. Section 4 begins with a description of the frictional experimental testbed, the ECP industrial emulator

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(ECP, 1995), and illustrates the results obtained when implementing of the original observer as well as the two modifications proposed in the PID mechatronic system. Finally, conclusions are drawn in Section 5.

2 VELOCITY OBSERVATION

Consider a class of mechanical systems expressed by (Xian et al., 2004)

$$\ddot{x} = h(x,\dot{x}) + G(x,\dot{x})u, \tag{1}$$

where $x \in \mathbb{R}$ is the system output, $u(t) \in \mathbb{R}$ is the control input, and $h(x,\dot{x}) \in \mathbb{R}$ as well as $G(x,\dot{x}) \in \mathbb{R}$ are nonlinear functions[1]. The system (1) satisfies the following assumptions (Xian et al., 2004).

Assumption A1. Both $h(x,\dot{x})$ and $G(x,\dot{x})$ are C^1 functions.

Assumption A2. The control input is a C^1 function and $u(t), \dot{u}(t) \in L_\infty$.

Assumption A3. The system state is bounded for all time; *i.e.*, $x(t), \dot{x}(t) \in L_\infty$.

The goal of the velocity observer is to estimate the unmeasurable velocity signal $\dot{x}(t)$ using only-position measurement and assuming that $h(x,\dot{x})$, $G(x,\dot{x})$ and $u(t)$ are unknown (Xian et al., 2004). Let $\hat{\dot{x}}(t) \in \mathbb{R}$ be the estimated velocity and $\tilde{\dot{x}} = \dot{x} - \hat{\dot{x}}$ the velocity estimation error. Then, the objective of the velocity observer is to ensure that $\tilde{\dot{x}}(t)$ converges to zero as the time tends to infinity.

Consider the following velocity observer (Xian et al., 2004)

$$\begin{aligned} \hat{\dot{x}} &= p + k_0\tilde{x}, \\ \dot{p} &= k_1 sgn(\tilde{x}) + k_2\tilde{x}, \end{aligned} \tag{2}$$

where k_0, k_1, and k_2 are positive constants, and $sgn(\cdot)$ is the signum function.

Remark 1. *System (2) can be expressed as*

$$\begin{aligned} \dot{y} &= f(y,x), \quad y \in R^2, \quad x \in R \\ z &= g(y,x), \quad z \in R \end{aligned}$$

where $y = [\hat{x} \quad p]^T$, $f(y,x) = [p + k_0\tilde{x} \quad k_1 sgn(\tilde{x}) + k_2\tilde{x}]^T$, $g(y,x) = p + k_0\tilde{x}$, *and* $z = \hat{\dot{x}}$ *is the output velocity estimation.*

To state the main result in (Xian et al., 2004), let

$$N_0(x,\dot{x},t) = h(x,\dot{x}) + G(x,\dot{x})u(t).$$

[1]Without loss of generality, we have assumed a one-degree-of-freedom (DOF) mechanical system.

Theorem 1 (Xian et al., 2004). *The velocity observer (2) ensures global asymptotic regulation of* $\tilde{\dot{x}}(t)$ *(i.e.,* $\tilde{\dot{x}}(t) \to 0$ *as* $t \to \infty$*) provided that* k_1 *satisfies*

$$k_1 > ||N_0(x,\dot{x},t)||_\infty + ||\dot{N}_0(x,\dot{x},t)||_\infty.$$

For detailed proof, see Theorem 2 in (Xian et al., 2004). Let us now put forward the following observer

$$\begin{aligned} \hat{\dot{x}} &= p + k_0\tilde{x}, \\ \dot{p} &= -k_1 sgn(\tilde{x}) + k_2\tilde{x}. \end{aligned} \tag{3}$$

Theorem 2. *The velocity observer (3) ensures global asymptotic regulation of* $\tilde{\dot{x}}(t)$ *(that is,* $\tilde{\dot{x}}(t) \to 0$ *as* $t \to \infty$*) provided that* k_1 *satisfies exactly the same conditions as in Theorem 1.*

Proof. Identical to that of Theorem 1.

Another alternative proposal to the original observer is the following

$$\begin{aligned} \hat{\dot{x}} &= p + k_0\tilde{x}, \\ \dot{p} &= k_1 sgn(\hat{x}) + k_2\tilde{x}. \end{aligned} \tag{4}$$

Theorem 3. *The velocity observer (4) also ensures global asymptotic regulation of* $\tilde{\dot{x}}(t)$ *(i.e.,* $\tilde{\dot{x}}(t) \to 0$ *as* $t \to \infty$*) on the condition that* k_1 *satisfies the same restriction as in Theorem 1.*

Proof. The same case as for the proof of Theorem 2 above.

These innovative observers are very similar to the one in Theorem 1, but they present significant differences. The observer in Theorem 2 introduces an inversion of the sign in the estimation dynamic that produces a filtering effect. The observer in Theorem 3 brings forward an estimation error $\hat{x}$ inside the signum function, with the intent of reducing the high-frequency content the error signal $\tilde{x}$ present in (2).

3 NUMERICAL EXPERIMENTS

Consider a linear motion of unit mass

$$\ddot{x} = u - f, \tag{5}$$

where f is the friction force. Assuming for a moment that $f = 0$ and $k_1 = 10$, the requirement imposed by all theorems is satisfied. We complete the velocity observers by setting $k_0 = k_2 = 10$. We also construct the following PID controller, which makes the closed-loop system globally asymptotically stable (if $f = 0$)

$$u = -k_p(x - x_d) - k_i \int (x - x_d)dt - k_d\dot{x}, \tag{6}$$

with $k_v = 6$, $k_p = 3$, $k_i = 4$, and the position reference set at $x_d = 1$ [m], see details in (Canudas de Wit et al., 1995). Because friction force can produce limit cycles within the system when the control law has an integrating action (Canudas de Wit et al., 1995), we incorporate a friction force in (5) by invoking the LuGre model and its standard parameters given in (Canudas de Wit et al., 1995); *i.e.*, f is obtained as a nonlinear dynamic. Simulation results are depicted in Figure 1 where the position and velocity of the system are pictured. These plots are a recreation of the experiment presented in (Canudas de Wit et al., 1995). Obviously, the PID controller (6) incorporates velocity measurement, which for this simulation was assumed to be available.

At this point, we repeated the previous simulation using the true velocity in the control law and implementing the three observers solely to estimate the velocity, obtaining the results shown in Figure 2. Since in the experimental application the velocity would not be available for use, the simulation was again repeated, this time invoking the observers from Theorems 1, 2, and 3 in the control law (*i.e.*, replacing $\dot{x}$ by its corresponding $\hat{\dot{x}}$ in (6)). The results are portrayed in Figures 3, 4, and 5, respectively. Note that chattering appears with the observer from Theorem 1 and not with the others. It is important to stress that chattering is undesirable in a physical system because the high-frequency switching can damage the system, as well as activate unmodelled dynamics (G. Bartolini, 1998) and (Hung, 1993).

Let us return to the previously considered case where the PID controller (6) is employed using the exact velocity, which is only observed. If we slightly modify the observer parameter values to $k_1 = 5$ and $k_0 = k_2 = 1$, the velocity observation obtained with the observer in (2) becomes highly oscillatory whereas the two observers proposed still provide good results, as shown in Figure 6. If furthermore we now feed the corresponding velocity estimation into the PID controller (6), the first observer (stated in Theorem 1) is much more sensible to variations in its parameters. This causes instability of the closed-control-loop (simulation pictures were then omitted for the first case); nevertheless, the other two new observers yield acceptable good simulation results, see Figures 7 and 8.

4 APPLICATION TO AN INDUSTRIAL EMULATOR

To have a more realistic comparison among the observers stated in Section 2, we proceed to implement

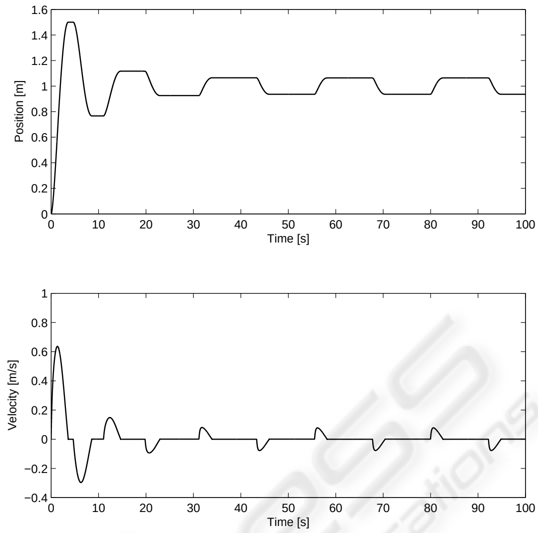

Figure 1: PID position control of a second-order system that incorporates Lugre friction model - see (Canudas de Wit et al., 1995).

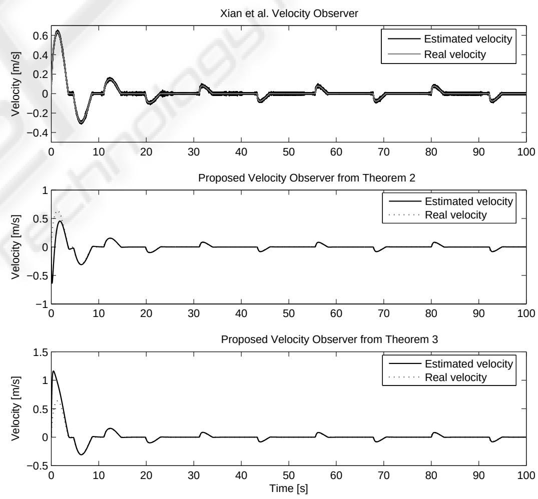

Figure 2: Comparison of the three estimators (for $k_{0..} = k_1 = k_2 = 10$): 1) Top with Theorem 1, 2) Middle with Theorem 2, and 3) Bottom with Theorem 3.

them on an experimentation testbed.

4.1 Experimental Setup

The experiments were performed on an ECP Model 220 industrial emulator which includes a PC-based control platform and a DC brushless servo system (ECP, 1995). The mechatronic system includes two motors, one as servo actuator and the other as disturbance input (not used here), a power amplifier, and two encoders which provide accurate position

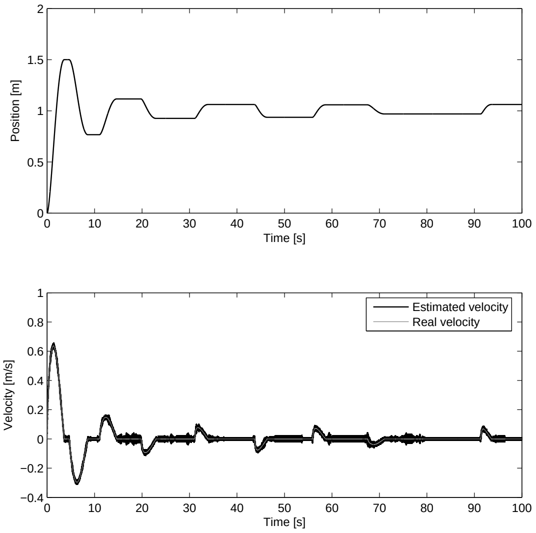

Figure 3: PID control incorporating velocity estimation from Theorem 1.

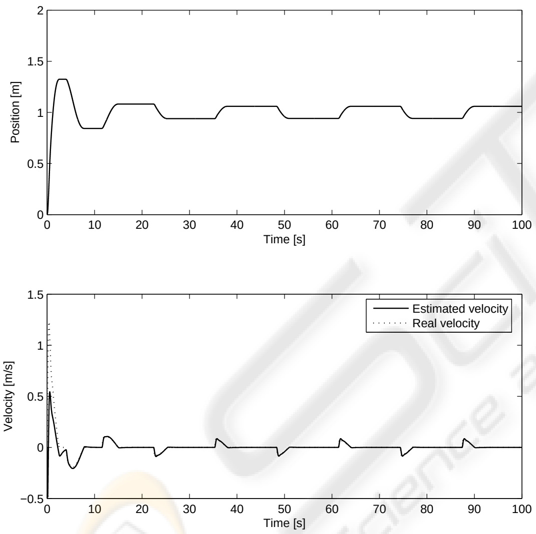

Figure 4: PID control employing velocity estimation given in Theorem 2.

measurements; i.e., 4000 lines per revolution with $4\times$ hardware interpolation giving 16000 counts per revolution to each encoder; 1 count (equivalent to 0.000392 radians or 0.0225 degrees) is the lowest angular position measurable (ECP, 1995). The system was set up to incorporate inertia and friction brake. The drive and load disks were connected via a $4:1$ speed reduction (see Figure 9). In order to demonstrate that the system is subject to the notorious effects of friction, we calculated according to the procedure described by (R. Kelly and Campa,

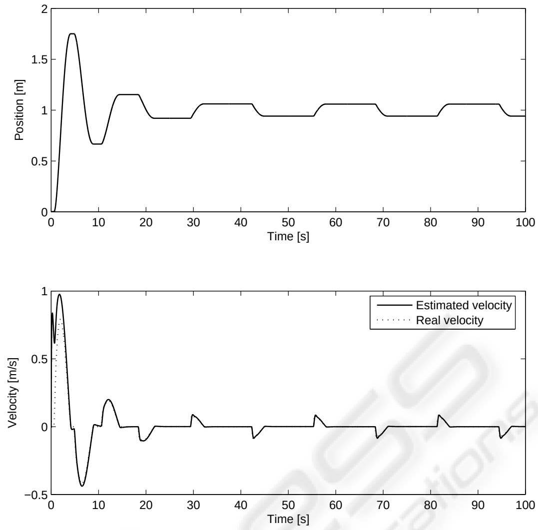

Figure 5: PID control utilizing the observer stated in Theorem 3.

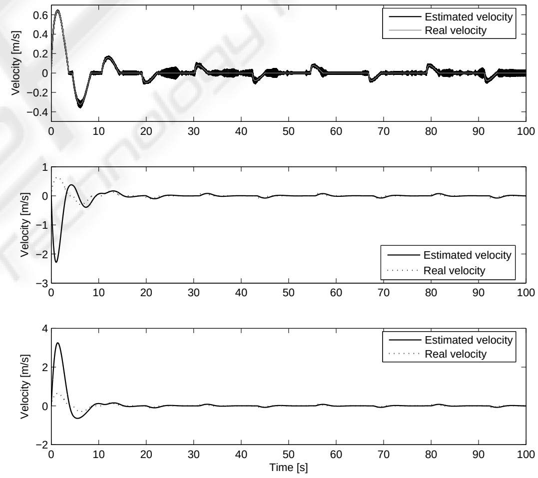

Figure 6: Comparison of the three estimators (for $k_{0..} = k_2 = 1$ and $k_1 = 5$): 1) Top with Theorem 1, 2) Middle with Theorem 2, and 3) Bottom with Theorem 3.

2000) the following friction coefficients for the system: $F_v = 0.05772[Nmsec/rad]$ (viscous friction coefficient) and $F_c = 0.43032[Nm]$ (Coulomb friction).

A Pentium 4, 2.80 GHz CPU, 512 MB RAM, computer running under Windows XP is programmed to implement the controller together with the interface medium ECP USR Executive 5.1, a C-like programming language (ECP, 1995). The system contains a data-acquisition board for digital to analog conversion and a counter board to read the position encoder outputs into the servo system. The minimum servo-loop closure sampling time T_s is 0.884 ms.

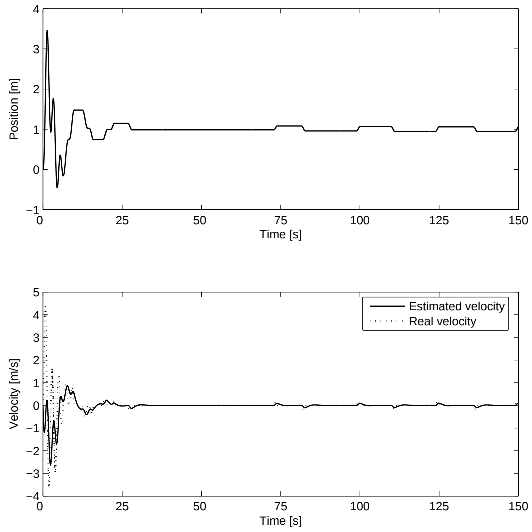

Figure 7: PID control containing the velocity as given by Theorem 2.

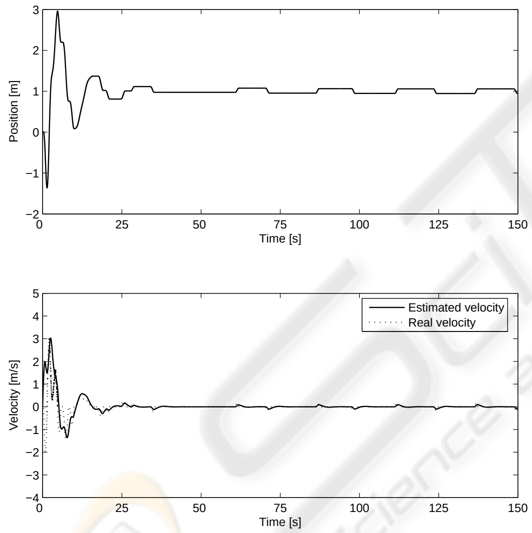

Figure 8: PID control making use of the observer in Theorem 3.

The output voltage signal generated by the system is in the range of $\pm 5V$ and is delivered to the motor drive via the DAC, the measurement feedback is a position signal (in counts or radians), measured at the shaft of each of the two disks by the optical rotary incremental position encoders, then it is read by the microcomputer by means of the counter board and delivered into the PC. A software interface has been built to easily transfer the raw data collected from the plant (by means of the ECP USR Executive program) to the Matlab workspace environment, in order to

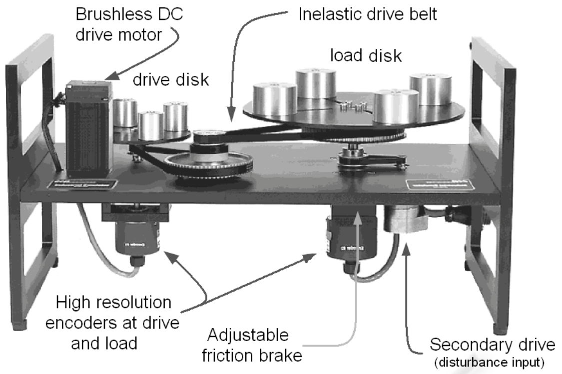

Figure 9: Mechanical system with friction.

display the results. The load disk is weighted with 4 masses of 0.50 kg each (at a radius of r = 10.0 cm) while the drive disk remains unweighted (see Figure 9). It is worth mentioning that the mechanical system has encoders which give accurate position measurements, nevertheless no direct velocity sensing is available (ECP, 1995). In this scenario, we implemented the aforementioned velocity observers, obtaining the results pictured in Figures 10 – 12.

4.2 Experimental Results

The implemented control law is as follows

$$u = -k_p(x - x_d) - k_i \int (x - x_d)dt - k_d\dot{\hat{x}},$$

with $k_d = 0.0011$, $k_p = 0.135$, and $k_i = 0.4$. The desired reference position was set to $x_d = 100\ [counts] = 0.0392\ [rad] = 2.25\ [deg]$.

As it can be noted in Figure 10, when the observer stated in Theorem 1 is employed, this produces an oscillation of relatively high-frequency and amplitude into the system. The observer (4), after the transient, eliminates both the amplitude and the frequency of this oscillation (Figure 11). The estimator from Theorem 3 further reduces the amplitude and duration of the transient, even more than observer (4), eliminating chattering as seen in Figure 12. Moreover, note that the position limit cycles in Figures 11–12, caused by friction in PID control of servo drives (Canudas de Wit et al., 1995), have the same rectangular-like waveform pattern as in Figure 1, and can be distinctly identified.

5 CONCLUSIONS

Two new velocity-observation designs are presented, and experimentally validated, for use in mechanical

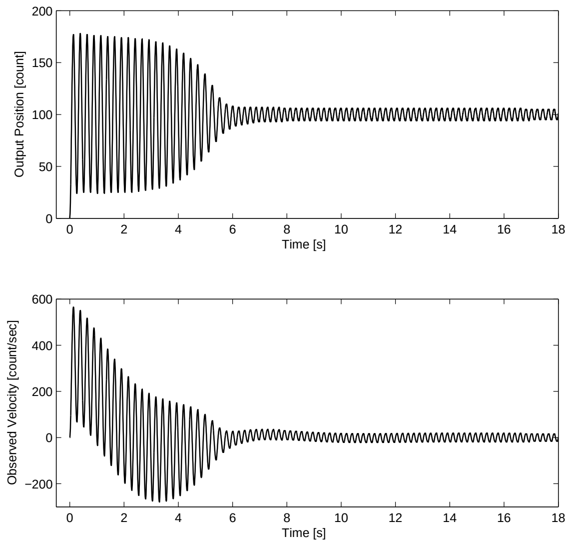

Figure 10: Experimental results using the observer from Theorem 1.

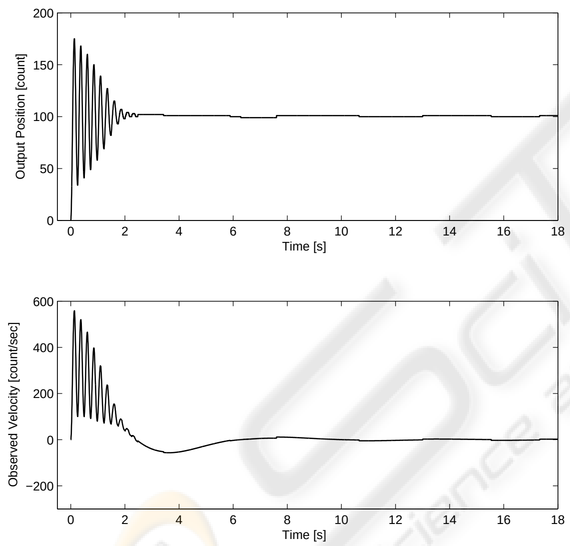

Figure 11: Experimental results employing the observer from Theorem 2.

systems with friction where only-position measurements are available. As it can be appreciated from numerical and experimental results, the proposed observer schemes are more efficient than their precursor in that chattering is eliminated from the velocity-observed signal. It is worth emphasizing that the presented observers (3) and (4) are especially interesting for industrial purposes, for they assure that the velocity-acquisition hardware can, without difficulty, be replaced by an analogous inexpensive software performing the same function.

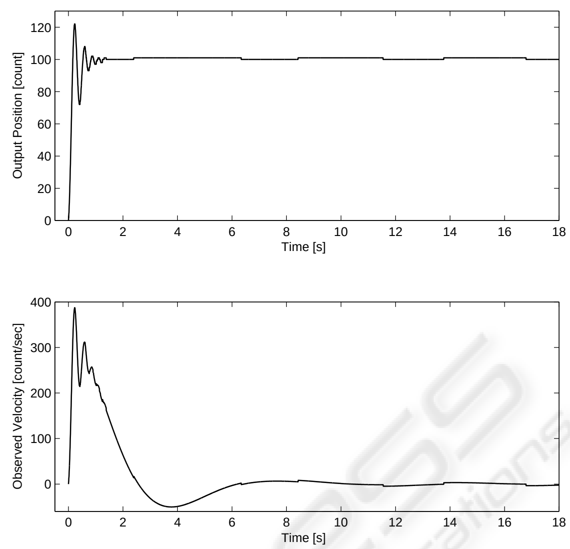

Figure 12: Experimental results utilizing the observer from Theorem 3.

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