DISTURBANCE COMPENSATION TECHNIQUES FOR CONTROLLING PNEUMATIC ACTUATOR SYSTEMS

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ABSTRACT
In this paper, three disturbance compensation algorithms are experimentally evaluated considering the control problem of a rodless pneumatic cylinder. Despite its highly nonlinear dynamics, the pneumatic cylinder is modeled as a linear system including an unknown lumped disturbance. Three disturbance estimation algorithms are then employed to compensate for this lumped disturbance: the traditional linear disturbance observer (TDOB), the integral sliding-mode disturbance observer (I-SDOB) and the IMP-based SDOB (IMP-SDOB). Experimental results are presented to compare these three algorithms in terms of positioning accuracy.

Keywords: disturbance compensation; disturbance observer; sliding mode; pneumatic cylinder.

TECHNIQUES DE COMPENSATION DE TURBULENCE DES SYSTÈMES DE CONTRÔLE PNEUMATIQUE D’ACTIONNEURS

RÉSUMÉ
Dans cet article, trois algorithmes de compensation de turbulence sont évalués en considération du problème de contrôle du cylindre pneumatique sans tige. En dépit de sa dynamique non linéaire élevée, le cylindre pneumatique est modélisé comme un système linéaire incluant un groupe de turbulence inconnu. Trois algorithmes d’estimation de turbulence sont alors utilisés pour compenser cette turbulence groupée : la traditionnelle observation de turbulence linéaire (TDOB), l’observation intégrale en mode glissant (I-SDOB) et le principe du modèle interne (IMP-SDOB). Les résultats expérimentaux sont présentés pour comparer ces algorithmes en fonction de l’exactitude.

Mots-clés : compensation de turbulence; observation de turbulence; mode glissant; cylindre pneumatique.
1. INTRODUCTION

Servo control systems usually suffer from significant unknown external disturbances that deteriorate the dynamic performance of servomechanisms. To alleviate the adverse effects of these disturbances, various kinds of disturbance estimation algorithms have been proposed with the attractive feature of “plug-and-play”, which allows the introduction of disturbance observers to the existing control systems without the need to modify the original controllers. Among the various structures of disturbance observers, the traditional linear disturbance observer (TDOB) proposed in [1–6] has received considerable research attention in both its theoretical and practical aspects. It has been applied to many kinds of physical systems, demonstrating its effectiveness and ease of implementation. In [7], a new design was proposed to obtain asymptotic disturbance estimation. Moreover, the TDOB was employed in [8] in order to improve the efficiency of an adaptive feedforward controller. However, a TDOB that utilizes the linear approach would be sensitive to modeling uncertainties of physical plants.

Being a nonlinear approach, sliding-mode control (SMC) [9, 10] is well-known for its robustness against parametric uncertainties of plant models. However, due to the use of discontinuous switching control and the existence of parasitic dynamics or digital implementation, this nonlinear robust control brings about the undesirable phenomenon called chattering, which is a major impediment to the realization of SMC in a broad range of applications. One way to alleviate chattering is to adopt the sliding-mode disturbance observer (SDOB) [11–16]. Similar to the sliding-mode observer for state estimation [17, 18], discontinuous switching control efforts appear in the SDOB for disturbance estimation, rather than being directly applied to the plant, thereby reducing chattering. Integration of the sliding-mode technique with the structure of disturbance observers, on one hand, can enhance the robustness of disturbance observers to modeling uncertainties of physical plants, and on the other hand can alleviate the chattering that commonly appears with the implementation of SMCs. However, in previous SDOBs [11–16], the switching gain needs to be greater than the upper bounds on the unknown disturbance, which limits their effects on the chattering alleviation. To release this limitation, several kinds of integral sliding-mode disturbance observer (I-SDOB) have been proposed [19, 20]. In the I-SDOBs, the switching gain only needs to be greater than the magnitude of the disturbance estimation error. In other words, the switching gain is required only to be greater than the absolute value of the difference between the actual disturbance and its estimate, thus reducing the required magnitude of switching gains. Compared with the existing SDOBs [11–16], the I-SDOBs further alleviate chattering. Nevertheless, stability of the overall controller/observer systems containing the I-SDOBs cannot be guaranteed. In [21], a stability-guaranteed I-SDOB was proposed, in which the overall controller/observer system is proved to be stable based on the Lyapunov method and the assumption that the change rate of actual disturbances with respect to time is upper bounded by some constants.

In linear control theory, there is an approach to ensuring precise tracking based on the internal model principle (IMP) [22], which states that a model of the non-decaying exogenous signal in the loop transfer function ensures perfect asymptotic tracking and disturbance rejection. In [23], an SDOB design is devised to combine the best features of the IMP and the sliding mode. More specifically, an internal model principle-based SDOB, referred to as the IMP-SDOB, was proposed in [23]. With the internal model of the disturbance, the IMP-SDOB asymptotically identifies an unknown disturbance that can be strongly time-varying. In [24], the IMP-SDOB is improved to guarantee stability of the overall controller/observer system.

In this paper, various disturbance estimation algorithms are reviewed, including the TDOB, the I-SDOB and the IMP-SDOB. To evaluate the effectiveness of these disturbance estimation algorithms, they are experimentally applied to a pneumatic servo system that has highly nonlinear dynamics with unknown disturbances.
2. EXPERIMENTAL PNEUMATIC CYLINDER

2.1. Plant Description and Nominal System Model

Figure 1 shows the schematic layout of the pneumatic servo system, in which a proportional pneumatic directional control valve (Festo MPYE-5-1/8) is employed to adjust the flow rate to a pneumatic rodless cylinder (SMC MY1C-25-100) that moves an external payload. The maximum supply air pressure is 0.7 MPa, and the piston diameter and the stroke of the pneumatic rodless cylinder are 25 mm and 100 mm, respectively. The motion of the payload is sensed by a linear scale (Carmar LE0150-1A3) and an accelerometer (Silicon Designs 2210-005) with a full-scale measurement range of \(\pm 5g\). As shown in Fig. 1, the signals from the linear encoder are fed through differential line receivers to a field-programmable gate array (FPGA) (Xilinx XCV50PQ240-C6) that processes these signals to yield a resolution of 1 \(\mu m\). In addition to position counting, the FPGA detects velocity by a digital tachometer that measures the time interval of the encoder pulses to achieve more accurate estimation than direct differentiation of the position signal. Furthermore, the FPGA provides an interface to the A/D converter (ADC) and the D/A converter (DAC), in which the ADC acquires the analog acceleration signal while the DAC sends the control signal to the proportional valve. After A/D conversion, the FPGA generates an interrupt request to the digital signal processor (DSP) (TI TMS320C6711) at the rate of 12.2 kHz. In an ISR that implements control laws, the DSP obtains the position, velocity and acceleration information from the FPGA, calculates the control algorithms, and sends the control efforts to the proportional valve through the DAC and some analog signal processing circuits.

In the experimental system, a personal computer was used to develop the control program written in C language, to compile it, to download the resulting code into DSP for execution and to acquire experimental data. Figure 2 shows a photograph of the experimental pneumatic system.

The pneumatic system has highly nonlinear dynamics and can be approximated as a third-order system when the response of the valve is fast enough to be neglected compared to the response of the whole system [25]. In this work, a Dynamic Signal Analyzer (DSA) (Spectral Dynamics SigLab-20-42) automatically
steps a sine wave over a specified frequency range and measures the frequency response of the pneumatic system around the midpoint of the full stroke. Then, a Matlab function `invfreqs(,·)` performs the least-squares fit to the frequency response data and yields the following nominal third-order transfer function of the plant

\[
P(s) = \frac{x(s)}{u(s)} = \frac{4012}{s(s^2 + 93.62s + 5272)} \text{ (m/V)},
\]

(1)
in which \( P(s) \) denotes the transfer function of the plant, \( x(s) \) is the Laplace transform of the load position, and \( u(s) \) is the Laplace transform of the control signal to the proportional valve. Rewriting (1) in companion form gives

\[
x^{(3)} = f(x) + b(x)(u + d),
\]

(2)
in which \( x = [x \; \dot{x} \; \ddot{x}]^T \) is the state vector, \( f(x) = 93.62\ddot{x} - 5272\dot{x} \) and \( b(x) = 4012 \). Here, \( d \) includes unknown disturbances and modeling uncertainties partially due to linearizing the nonlinear dynamics. The state variables for the pneumatic system are the position, velocity and acceleration of the load position, which results in a system model with system perturbations satisfying the so-called matching condition [16] for disturbance rejection and insensitivity to parameter uncertainties. Other physical quantities, such as the chamber pressures of the pneumatic cylinder as shown in [26], can also be chosen as state variables. However, this leads to a system model, in which system perturbations cannot satisfy the matching condition and system robustness to these perturbations is reduced. Another benefit of choosing the acceleration as a state variable is the ubiquitous availability of acceleration sensors. Moreover, the acceleration signal can be employed to facilitate the estimation of velocity, as illustrated in [27].

### 2.2. Design of a Nominal Controller

Let the tracking error \( e = x - x_d \), in which \( x_d \) denotes the reference assumed to be three-times differentiable with respect to time. Moreover, define the filtered tracking error \( s \) to be

\[
s = (p^2 + c_1 p + c_0)e,
\]

(3)
in which the differential operator \( p = d/dt \), and the \( c_i' \)'s are constant parameters chosen so that the dynamics associated with \( s = 0 \) is asymptotically stable. Let the control be in an additive form

\[ u = u_{pa} + u_{do}, \tag{4} \]

in which \( u_{do} \) denotes the disturbance compensation by a disturbance observer, and the nominal control \( u_{pa} \) is described by

\[ u_{pa} = b^{-1}[-f - \lambda s + x^{(3)}_d - (c_1 p^2 + c_0 p)e], \tag{5} \]

in which \( \lambda \) is a constant parameter. Substituting (3–5) into (2) gives

\[ \dot{s} = -\lambda s + b(u_{do} + d). \tag{6} \]

In the ideal situation when \( u_{do} = -d \), we have \( \dot{s} + \lambda s = 0 \). Thus, the \( c_i' \)'s and the parameter \( \lambda \) are to specify the desired closed-loop dynamics. Subsequently, rather than attenuating the disturbance effect through high feedback gains, disturbance observers are to be designed to generate \( u_{do} \) in order to compensate for the disturbance \( d \).

3. DISTURBANCE COMPENSATION TECHNIQUES

In this section, three disturbance estimation algorithms are revisited and tailored for our experimental system. They are: the TDOB [1-3], the I-SDOB [21] and the IMP-SDOB [24].

3.1. Traditional Linear Disturbance Observer (TDOB)

Figure 3 shows the block diagram of the TDOB-based control system, in which \( Q(s) \) is a lowpass filter. It can be seen that realization of the TDOB requires the implementation of \( P^{-1}(s) \), which involves triple differentiation of the position signal of the payload, \( x \), with respect to time. The lowpass filter \( Q(s) \) is used to attenuate high-frequency gain for robustness to high-frequency unmodeled dynamics. Since the acceleration of the payload, \( \ddot{x} \), can be measured by an accelerometer, the lowpass filter is chosen to be of first-order and described by

\[ Q(s) = \frac{1}{(\tau s + 1)}. \tag{7} \]

with \( \tau \) being a small positive constant.
3.2. Integral Sliding-Mode Disturbance Observer (I-SDOB)

Consider an artificially introduced auxiliary process described by

\[ \dot{z} = -\lambda s + b \phi \text{sgn}(\sigma), \]  

(8)

in which \( z \) is the state variable of the auxiliary process, \( \phi \) is a switching gain, \( \text{sgn}(\cdot) \) denotes the signum function, and the switching function \( \sigma \) is defined by

\[ \sigma = s - z. \]  

(9)

Taking the derivative of (9) with respect to time and substituting (6) and (8) into the resulting equation gives

\[ \dot{\sigma} = b(u_{do} + d - \phi \text{sgn}(\sigma)). \]  

(10)

Provided that the switching gain \( \phi > |u_{do} + d| \), then the sliding condition is satisfied, i.e. \( \sigma \dot{\sigma} < 0 \) if \( \sigma \neq 0 \). Assigning the initial condition of the auxiliary process such that \( z(0) = s(0) \), we have \( \sigma(0) = 0 \). This together with the satisfaction of the sliding condition yields \( \sigma(t) = 0 \) for \( t \geq 0 \). Since \( \sigma(t) = 0 \), we have \( \dot{\sigma} = 0 \), which gives according to (10)

\[ \phi \text{sgn}(\sigma) = u_{do} + d, \]  

(11)

in the sense of equivalent values [10]. Since the switching signal, \( \phi \text{sgn}(\sigma) \), reveals the discrepancy between \( u_{do} \) and \( d \) it represents the compensation error by \( u_{do} \) and can be used to adjust the disturbance compensation, \( u_{do} \).

Consider the following integral law for disturbance compensation

\[ \dot{u}_{do} = -k_{do} \phi \text{sgn}(\sigma) - k_{sv} bs, \]  

(12)

in which \( k_{do} \) and \( k_{sv} \) are arbitrary positive constant parameters. Figure 4 shows the structure of the I-SDOB-based controller.
3.3. IMP-Based Sliding-Mode Disturbance Observer (IMP-SDOB)

Assume that the model of the disturbance is given by [24]

\[ \dot{\xi} = A \xi, \quad d = C \xi, \]  

where the state vector \( \xi \in \mathbb{R}^m \), and the pair \((A, C)\) is assumed to be observable. Consider the same artificial auxiliary process (8) and the same switching function (9). Then the result (11) is also valid for the IMP-SDOB in the sense of equivalent values. Consider the following law for disturbance compensation

\[ \dot{\hat{\xi}} = A \hat{\xi} + g \varphi \text{sgn}(\sigma) + P^{-1} C^T b s, u_{do} = -C \hat{\xi}, \]  

in which \( \hat{\xi} \in \mathbb{R}^m \) is the state vector of the disturbance estimator, \( g \in \mathbb{R}^m \) is a gain vector, and \( P \in \mathbb{R}^{m \times m} \) is a gain matrix required to be symmetric positive-definite. The design of the IMP-SDOB requires the determination of \( g \) and \( P \): Given an observable pair \((A, C)\), the pole placement technique is employed to find \( g \) such that \((A - gC)\) has desired eigenvalues with negative real parts. Subsequently, choose a symmetric positive-definite matrix \( Q \), and solve the Lyapunov equation \((A - gC)^T P + P(A - gC) = -Q\) for a symmetric positive-definite matrix \( P \) with the previously determined \( g \) [24]. Figure 5 shows the block diagram of the IMP-SDOB, in which the nominal controller that generates \( u_{pa} \) is not shown.

4. EXPERIMENTAL RESULTS

The considered pneumatic system has highly nonlinear dynamics, though it is simply treated as a third-order linear system with constant coefficients in the controller design. Between the plant and the linearized model, there are modeling discrepancies, including unknown disturbances and modeling uncertainties partially due to linearizing the nonlinear dynamics. These modeling discrepancies act as real disturbances and need to be compensated for by a disturbance observer in order to improve system performance. The control law (4) in an additive form is \( u = u_{pa} + u_{do} \). The nominal controller (5) is specified by \( \lambda = \omega_n, c_1 = 2\omega_n, \) and \( c_0 = \omega_n^2 \), in which the constant \( \omega_n \) determines the nominal closed-loop poles. Here, the nominal controller with \( \omega_n = 50 \) is utilized in the following experiments. Consider the following reference in the unit of mm

\[ x_d(t) = 20[H(t) - H(t - 0.547)] + 50[\dot{H}(t - 0.547) - H(t - 1.094)] + 80[H(t - 1.094) - H(t - 1.634)], \]
in which \( H(\cdot) \) denotes a unit-step function. Figures 6 and 7 show the dynamic responses by the TDOB with \( 1/\tau = 60 \) and 70, respectively. It can be seen that the TDOB with \( 1/\tau = 70 \) leads to severe oscillations in the control signal. Therefore, the TDOB with \( 1/\tau = 60 \) is utilized for performance comparison with other disturbance estimation algorithms. For the I-SDOB, let \( \varphi = 70, k_{do} = 50 \) and \( k_{sv} = 1.5 \times 10^{-4} \). Figure 8 shows the dynamic responses by the TDOB and the I-SDOB, demonstrating that the I-SDOB yields performance superior to the TDOB. For the IMP-SDOB, consider a third-order disturbance satisfying \( \ddot{d} + (2\pi)^2 \dot{d} = 0 \), which includes a step and a sinusoid of 1 Hz and which yields
The gain vector $g$ is determined such that $(A - gC)$ has eigenvalues of $-20$ with multiplicity of three. Moreover, the matrix $Q$ is chosen to be $1000I$, and the matrix $P$ is obtained by solving the Lyapunov equation $(A - gC)^T P + P(A - gC) = -Q$. Figure 9 shows the dynamic responses by the I-SDOB and the IMP-SDOB, demonstrating that the IMP-SDOB yields somewhat smaller positioning error than the I-SDOB.
Let the following reference be a step function with a step size of 20 mm. Moreover, an input disturbance $d(t) = 0.2 \sin(2\pi t) + 0.1 H(t - 2.5)$ [V] is artificially introduced to the system. Figures 10 and 11 show the dynamic responses by the I-SDOB and IMP-SDOB, respectively. By comparing Figs. 10 and 11, it can be seen that the IMP-SDOB can cope with the disturbance better than the I-SDOB. Subsequently, consider the tracking control problem with a 1-Hz reference of 3-4-5 curves. Figure 12 shows the tracking performance of the TDOB and the I-SDOB, and it can be seen that the I-SDOB performs better than the TDOB. As seen from Figs. 8 and 12, although the I-SDOB yields high-frequency switching in the control signals, this high-frequency control serves as high-frequency dither signals to alleviate the adverse effect of dry
friction on tracking precision. Figure 13 shows the tracking performance of the I-SDOB and the IMP-SDOB, demonstrating that the IMP-SDOB further improves the tracking precision of the pneumatic system. Figure 14 shows the tracking errors in ten consecutive experiments, further confirming the effectiveness of the IMP-SDOB.

5. CONCLUSIONS

In this paper, a nonlinear pneumatic system is modeled as a linear system with a lumped disturbance that includes the external disturbance and the model mismatch. Three disturbance estimation algorithms are
Fig. 14. Ten consecutive experimental results by the I-SDOB and the IMP-SDOB.

individually integrated into the controller to compensate for this lumped disturbance. Experimental results show that the nonlinear SDOBs outperform the linear TDOB in terms of positioning accuracy. Moreover, the SDOBs by nature can generate high-frequency dither signals in the control input, thus alleviating the adverse effect of friction. By further incorporating the model of exogenous signals into the SDOB, the IMP-SDOB yields the minimum deviation from the desired position and achieves the most precise positioning without causing much chatter in the control signal.

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REFERENCES