Abstract— Pneumatic systems are highly non-linear by their nature. Despite their many advantages like reliability, compliance for interaction tasks requiring backdrivability, high power to weight ratio, the use of such systems is restricted primarily because of its high initial cost. This paper presents the development of a robust observer based controller to eliminate costly pressure sensors and therefore obtain a low cost pneumatic servo system. Experimental results are presented that demonstrate the effectiveness of this pressure observer based controller.

I. INTRODUCTION

The use of pneumatic systems for task such as clamping, grasping, broaching and drilling operations has been utilized in industrial assembly-line settings for decades. Initially, such systems were confined to these limited uses because they were either controlled in an open-loop fashion, or the closed-loop controllers developed were good only for a limited bandwidth. This problem was circumvented by the development of high-bandwidth non-linear controllers for accurate pneumatic motion and force control. This development helped in introducing pneumatic systems to more sophisticated tasks such as assembly (such as IC component insertion), facing and other industrial robotic uses. Pandian et al. [1] presented a sliding mode controller for position control that showed good results at lower frequencies. Richer and Hurmuzlu [2] in their work presented the design of a sliding-mode force controller and showed a good response up to 20 Hz sinusoidal frequency. All these developed controllers concentrated on the position and/or force tracking accuracy and ignored the energetic efficiency and/or initial costs associated with the control system. Many authors focused on the development of more efficient controllers to reduce the operating cost of the system. Sanville [3] suggested the use of a secondary reservoir in an open-loop system to collect exhaust air. This air was in turn utilized as an auxiliary low-pressure supply. Al-Dakkan [4] et al. presented a control methodology that provides significant energy savings. They used two three-way spool valves instead of a conventional four-way proportional spool valve and introduced a dynamic constraint equation that minimizes cylinder pressures resulting in low energy consumption. In other efforts to reduce initial costs, Ye et al. [5], Kunt and Singh [6], Lai et al. [7], Royston and Singh [8], Paul et al. [9], and Shih and Hwang [10] demonstrated the viability of servo-control of pneumatic actuators via solenoid on/off valves in place of proportional valves.

All of these prior works are based on the measurement of all the states (viz. pressures in each chamber, position and velocity) of the system. The measurement of these states mandates the use of two pressure sensors per axis, and a linear potentiometer (or similar) to measure the position. The velocity signal is typically obtained by differentiating the position sensor signal. The problem with pressure measurement is that high-bandwidth, high-pressure sensors required for the control of pneumatic servo systems are expensive and large (relative to the actuator), with a typical cost between $250 and $500. Since pneumatic systems use two pressure sensors per axis, these sensors add $500 to $1000 per axis of a pneumatic servo system. Coupled with valve and cylinder costs, pneumatic systems are not cost-competitive with most electromagnetic types of actuation. One method of avoiding pressure measurements is the so-called PVA, or position-velocity-acceleration controller structure [15]. Although such controllers have met with a certain amount of success, the difficulty of obtaining a clean acceleration signal can be problematic. Additionally, PVA controllers are non-model based and cannot address the often significant nonlinearities associated with pneumatic systems.

Since actuator displacement output is easily measurable, an alternate means of circumnavigating the high cost issue associated with pressure sensors would be to construct a pressure observer to estimate the pressure from the displacement sensor. Two such observer designs are presented in an earlier paper by the authors [11]. The preferred design of the two, and the one utilized for control

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Fig. 1. Schematic of Pneumatic Actuator Servo System
in this paper, requires only position and velocity information. This has the advantage of avoiding noise problems in obtaining acceleration measurements without sacrificing the option of formulating a model-based controller that utilizes pressure information. However, the results of the observer showed some amount of uncertainty in the observed values of the pressures. Therefore, a robust controller based on sliding mode control theory is developed in this paper to take into account observer uncertainties along with the uncertainties present in the system model, like friction, to obtain a low cost pneumatic servo system.

In this paper, the valve dynamics and the delay induced by the connecting tubes are neglected. The valve dynamics are normally much faster than the system dynamics and can be accounted for using a robust controller. The connecting tubes used in the experiment are short and hence do not contribute significant time delays. Pressures in both the chambers are estimated by using observers developed by the authors in [11]. The observers were presented and their stability, robustness, and convergence were discussed in the previous work. The development of both observers is summarized below for convenience.

II. PRESSURE OBSERVERS

A. Energy Based Lyapunov Observer Design

This observer is based on the state equation of the pressure in a variable volume chamber with the assumption of isothermal process during charging (mass flow in) and discharging (mass flow out). The rate of change of pressures in one side of a pneumatic piston or the other (a or b) is estimated by following state equation (refer to [12-14] for a derivation of the state equation) of the pneumatic system:

\[
\dot{P}_{(a,b)} = \frac{RT}{V_{(a,b)}} \dot{m}_{(a,b)} - \frac{V_{(a,b)}}{V_{(a,b)}} \dot{P}_{(a,b)}
\]

where, \(P\) (Pa) is the observed absolute pressure in the chamber; \(R\) (J/kg-K) is the gas constant; \(V\) (m\(^2\)) is the volume of the chamber; subscripts “a” and “b” represent properties of chambers “a” and “b” respectively (see Figure 1). \(\dot{m}\) is the observed mass flow rate and \(\dot{m}\) is the actual mass flow rate flowing either in or out of the chamber and is given by the expression [12-14]:

\[
\dot{m} = \frac{C_e A_p p_u}{\sqrt{T}} \left( \frac{2\gamma}{\gamma-1} \right) \frac{\frac{p_d}{p_u} \left( \frac{p_d}{p_u} \right)^{\frac{\gamma+1}{\gamma}}}{\frac{p_d}{p_u} > \frac{2}{\gamma+1}}
\]

\[
\dot{m} = \frac{C_e A_p p_u}{\sqrt{T}} \left( \frac{2\gamma}{\gamma+1} \right) \frac{2}{\gamma+1} \left( \frac{p_d}{p_u} \right)^{\frac{\gamma}{\gamma+1}} \text{ if } \frac{p_d}{p_u} \leq \frac{2}{\gamma+1}
\]

where, \(C_e\) is the discharge coefficient of the valve; \(A_p\) (m\(^2\)) is the exhaust area of the valve; \(p_u\) (Pa) and \(p_d\) (Pa) are the upstream and downstream pressure respectively; \(T\) (K) is the exhaust temperature of gaseous products; \(\gamma\) is the ratio of the specific heat at constant pressure \((C_p)\) to the specific heat at constant volume \((C_v)\).

It was shown in the experimental results of [11] that a maximum multiplicative error of + 0.4 and – 0.9 atmospheric pressure exists when the velocity of the piston is zero. Therefore, the controller should be robust enough to handle this uncertainty.

B. Force-Error Based Lyapunov Observer Design

This observer is derived based on a state equation with the corrective term which is based on the error between the actual and estimated resultant force across the piston of the cylinder. The value of rate of change of pressures is estimated by following equations:

\[
\dot{\hat{P}}_{(a,b)} = \frac{\gamma RT}{V_{(a,b)}} \dot{\hat{m}}_{(a,b)} - \frac{\gamma V_{(a,b)}}{V_{(a,b)}} \hat{P}_{(a,b)} + k_{(a,b)} \Delta F
\]

where,

\[
\Delta F = (P_a A_a - P_b A_b) - (\hat{P}_a A_a - \hat{P}_b A_b)
\]

In the above equation, the term \(P_a A_a - P_b A_b\) is calculated using force balance equation of the pneumatic system (Figure 1):

\[
M \ddot{x} + B \dot{x} + F_c = P_a A_a - P_b A_b - P_{atm} A_r
\]

where, \(x\) (m) is the position of the piston; \(M\) (kg) is the mass of the load; \(B\) is the viscous friction coefficient; \(F_c\) (N) is the Coulomb friction; \(P_{atm}\) (Pa) is the absolute environmental pressure; \(A_r\) (m\(^2\)) is the cross-sectional area of the rod, and \(A_a\) and \(A_b\) (m\(^2\)) are the effective piston areas in chambers “a” and “b”, respectively.

\(k_{(a,b)}\) in the equation (3) is the variable gain and its value is dependent on the position of the load.

\[
k_a = \frac{-\gamma V_{a}}{V_{a} A_a} \text{ for } V_a > V_b \left( \frac{A_b}{A_a} \right)
\]

\[
k_a = \frac{-\gamma V_{a} A_a}{V_{b} A_b} \text{ for } V_a \leq V_b \left( \frac{A_b}{A_a} \right)
\]

In this observer, a multiplicative error of ± 0.6 atmospheric pressure is noticed in the experimental results. Finally, it should be noted that the observer of equation (3) requires acceleration information, as reflected through equations (4) and (5), and therefore has the disadvantage of requiring
either an accelerometer or obtaining a potentially noisy and/or phase shifted signal through differentiation. Given that the difference in performance between the energy based observer of equation (1) and the force-error observer of equation (3) was negligible [11], and given the advantage of the energy based observer not requiring acceleration information, the observer of equation (3) was selected to be incorporated into the controller/observer design here.

III. SLIDING-MODE CONTROLLER

The proposed position controller in this paper is based on sliding mode control theory. Sliding mode controllers are generally well suited for pneumatic servo actuators due to the highly non-linear behavior and uncertainties present in the model. The equivalent control input is calculated such that the rate of change of the positive-definite \( \frac{1}{2} s^2 \) Lyapunov scalar function is zero \( \dot{V} = 0 \). The corrective term is then added to the equivalent control input to make \( \dot{V} \) negative in the face of uncertainty, which implies the robustness of the controller, and provides uniform asymptotic stability. With this condition satisfied, all trajectories will move towards the surface \( s(t) \) and once they reach the surface, remain on it for all the future time.

For the system shown in Figure 2, the desired output is the position of the end-effector. The control input to the system is the area of the valve. In order to derive the control law, define a time-varying sliding surface as:

\[
s = \left( \frac{d}{dt} + \lambda \right)^{n-1} e
\]

where, \( \lambda \) is a strictly positive number, \( n \) is the number of times the output must be differentiated to get the input, and \( e \) is the error between the actual and desired position.

The above equation can be rewritten as:

\[
s = (\ddot{x} - \dot{x} \dot{d}) + 2 \dot{\lambda} \dot{e} + \lambda^2 e
\]

Substituting the expression of \( \ddot{x} \) from equation (5) in equation (8):

\[
s = \frac{1}{M} (P_a A_d - P_b A_b - P_d A_d - B \ddot{x} - F_d) - \dot{x} \dot{d} + 2 \dot{\lambda} \dot{e} + \lambda^2 e
\]

Differentiating equation (9) results in:

\[
\dot{s} = \frac{1}{M} (P_a A_d - P_b A_b - B \ddot{x}) - x_d^{(3)} + 2 \dot{\lambda} \dot{e} + \lambda^2 \dot{e}
\]

In the control of the pneumatic system shown in Figure 2, two four-way proportional spool valves were used. However, they were constrained to act as a one four-way proportional spool valve. Accordingly, the following constraint equation was imposed on the control input, which is the effective or signed valve area in this case:

\[
A_v = A_{eq} = -A_{eq}
\]

A positive valve area corresponds to the charging of chamber “a” and discharging of the chamber “b” while a negative area corresponds to the charging of chamber “b” and discharging of the chamber “a”.

Using constraint equation (11), substituting the value of \( \dot{p}_a \) and \( \dot{p}_b \) in equation (10), and forcing \( s \) to zero yields the equivalent control law:

\[
A_{eq} = \frac{\alpha \left( \frac{P_a A_a^2}{V_a} + \frac{P_b A_b^2}{V_b} \right) + B \ddot{x} + M (x_d^{(3)} - 2 \dot{\lambda} \dot{e} - \lambda^2 e)}{RT \alpha \sqrt{T} \left( \frac{P_a \Psi(P_a, P_d A_d A_d) A_a + P_b \Psi(P_b, P_d A_b A_b)}{V_a} + \frac{2 \dot{\lambda} \dot{e} + \lambda^2 e}{V_b} \right)}
\]

where,

\[
\Psi = \left\{ \begin{array}{ll}
\rho_u & \text{for choked flow} \\
\rho_d \left( \frac{\gamma}{\gamma+1} \right)^{(\gamma+1)/(\gamma-1)} & \text{for unchoked flow}
\end{array} \right.
\]

The function \( \Psi \) captures the shift in dynamic behavior that occurs in the transition between choked and unchoked flow through the valve. The mass flow rate through the valve initially depends both on the upstream and downstream pressures and increases with the pressure difference. Once the velocity of air at the venturi reaches the speed of sound, i.e. sonic, the mass flow rate is only a linear function of the upstream pressure. The switching condition in equation (12) ensures that the controller uses the right equivalent control law.

The equivalent control input from equation (12) provides marginal stability in the sense of Lyapunov and uses model and error information. As noted earlier, a robustness term is added to this control input to provide uniform asymptotic stability. Thus the final control input is given by:

\[
A_v = A_{eq} - k \cdot \text{sat} \left( \frac{s}{\phi} \right)
\]

where \( k \) is a strictly positive gain and captures uncertainties of the model and the pressure observer; \( \Phi \) is the boundary layer thickness and selected such as to avoid excessive chattering across the sliding surface while maintaining the desired performance of the system.

The saturation function in equation (14) is defined by the following:
\[
\text{sat}(s/\phi) = \begin{cases} 
\text{sgn}(s/\phi) & \text{if } \vert \text{abs}(s/\phi) \vert \geq 1 \\
 s/\phi & \text{otherwise}
\end{cases}
\] (15)

IV. EXPERIMENTAL SETUP AND RESULTS

Experiments were conducted to implement and check the accuracy and effectiveness of the proposed controller with pressure observers. Two sets of experiments were conducted. The first set utilized the measured pressures, and the second set utilized the observed pressures, for the control of the motion trajectory of the end-effector using the sliding mode controller. A schematic of the system setup is illustrated in Figure 1 and the actual setup is shown in Figure 2. For this work, a two-degree of freedom linear robotic manipulator manufactured by Festo Corporation is used. However, for the results, only one degree of freedom is used, which is a double acting pneumatic cylinder (Festo SLT-20-150-A-CC-B). Two four-way proportional spool valves (Festo MPYE-5-M5-010-B) are used for controlling charging and discharging process of both chambers of the cylinder. A linear potentiometer (Modori LP-150F) with a travel length of 150 mm is used to measure the position of the load. The velocity signals are obtained by an analog differentiator with a first order roll-off at 50 Hz. Similarly, acceleration signals (used for the controller but not the observer) are obtained by an analog differentiation of the velocity signals with a first order roll-off at 50 Hz. Two pressure transducers (Festo SDE-16-10V/20mA) are also used in the setup for the measurement of actual pressures. The control and the observer algorithms are implemented using Real Time Workshop (RTW) from Mathworks in a 2.4GHz, 512MB RAM, Pentium IV processor based PC. The communication between the computer and the experimental setup is established through the digital input and analog output channels of an A/D card (National Instruments PCI-6031E).

The maximum pressure supply used for this experiment is 620kPa (90 psig). Some of the parameters (example, area of piston, area of rod, stroke length, pay-load mass) for this experiment are known accurately. The discharge coefficient ($C_d$), which primarily represents frictional losses, is a function of the valve area among other parameters such as the size and shape of the valve opening, surface finish and similar parameters. For this experiment, the average discharge coefficient is calculated based on the volumetric flow chart provided by the valve manufacturer. Other parameters, like the viscous friction coefficient, are difficult to measure. Therefore, these parameters are tuned to get the best control results.

The sliding mode control gains and the boundary layer thickness used in the experiment were selected via simulation using the system model and the uncertainties noted in the observer results. The gains were later tuned to provide good tracking performance.

The results of the controller with a mass of 3.6 kg at the end-effector are demonstrated in Figures 3 and 4. In all the figures shown, the solid line shows the desired trajectory and the dashed line shows the actual trajectory followed by the end-effector. Figure 3a shows the tracking of the end-effector at 0.25 Hz sinusoidal frequency when the controller uses pressure sensors present in the system. Figure 3b shows the result of sinusoidal tracking when the controller uses the energy-based pressure observer. It can be seen that the results obtained using pressure sensors versus pressure observers demonstrate essentially the same tracking performance. A small deviation in the tracking is observed in both cases when the velocity of the end-effector reaches zero. This is presumably because of the neglected Coulomb friction in the controller design. Figure 4 demonstrates the results at 2.5 Hz sinusoidal frequency. At this frequency a phase lag and attenuation in the amplitude is observed in the response. The detailed frequency response is shown in Figure 6. The results of the step response are shown in Figure 5. The results are similar to the sinusoidal tracking where the response of the system is almost identical using pressure sensors (figure 5a) or using pressure observers (figure 5b).

The results presented here are obtained using the energy-based pressure observer. Results of the force-error based pressure observer are very similar and are not presented in this paper. The energy-based pressure observer is preferred here due to its structural simplicity, its independence on the change of parameters (like payload mass), and its lack of dependence on acceleration.
Figure 6 shows the measured closed-loop frequency response of the controlled system. The bandwidth is observed to be about 5 Hz. It should be noted that the 5 Hz bandwidth is not a limitation of the controller. At this frequency, saturation in the valve output was observed which limited the bandwidth. The bandwidth can be increased with the use of valves with higher mass flow rates (larger maximum orifice sizes) or by reducing the mass at the end-effector. An increase in bandwidth can also be obtained by increasing the supply pressure.

Fig. 3a. Desired (solid) and actual (dashed) position at 0.25 Hz sinusoidal frequency tracking using pressure sensors

Fig. 3b. Desired (solid) and actual (dashed) position at 0.25 Hz sinusoidal frequency tracking using pressure observers

Fig. 4a. Desired (solid) and actual (dashed) position at 2.5 Hz sinusoidal frequency tracking using pressure sensors

Fig. 4b. Desired (solid) and actual (dashed) position at 2.5 Hz sinusoidal frequency tracking using pressure observers

Fig. 5a. Desired (solid) and actual (dashed) position at 0.5 Hz square-wave frequency tracking using pressure sensors
V. CONCLUSION

In this paper, the design of a controller for a pneumatic system that uses pressure observers has been presented. The results presented demonstrate that the tracking performance using pressure observers versus using pressure sensors is in essence indistinguishable. This shows that the system can be accurately controlled using pressure observers and hence it results in a low cost system. Additionally, the use of pressure observers along with the controller developed results in lower weight, more compact, and lower maintenance system.

REFERENCES


