Independent Metering of Pneumatic Actuator for Passive Human Power Amplification

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Abstract—In this paper, a method for passive human force amplification with independent metering of valves in a pneumatic actuator is reported. Unlike single valve metering, two independent valves are used to regulate the operating pressure of the actuator. The advantage of independent valves is that they can be controlled such that the actuator operates at minimum possible pressure, while providing a desired force profile. This results in reduction of the energy loss associated with discharge of high pressure air to the atmosphere thereby improving efficiency of operation. A pressure minimization scheme on the discharging side of the actuator is proposed to achieve minimum operating pressure. Passive human power amplification is achieved by commanding suitable inputs to the valve on the high pressure side. An experimental comparison with single valve metering shows that independent metering improves the operating time by nearly 100%, thus improving the efficiency of operation.

I. INTRODUCTION

Human amplifiers are tools that can amplify or attenuate the human force used to manipulate these tools. Due to direct interaction between the human and the device, it is essential to guarantee stability of operation irrespective of the task being performed. From systems theory it is well known that feedback interaction between two passive systems is always stable [4]. It has been shown in [5] that the human muscle dynamics behaves like a passive system. Therefore, by making the amplifier behave as a passive system, safe human power amplification can be achieved. In [6] a novel method for passive human power amplification using hydraulic actuators was presented. In [7], a general framework for passive power amplification with both hydraulic and pneumatic actuators was reported. In [2] these ideas are extended to passive bilateral tele-operation and human power amplification using pneumatic actuators. The experimental studies for evaluating the controller were performed on an anchored system, with a proportional servo valve metering the air flow to the actuator. A schematic of such an actuator moving an inertia is shown in Fig(1). The high power density of fluid powered actuators enables the design of compact human power amplifiers amenable for mobile operation. In such applications with limited supply of pressurized air, the time of operation of such devices can be increased by improving the efficiency of operation. A source of energy loss with single valve metering is the discharge of high pressure air to the atmosphere. One of the ways to mitigate this loss is to reuse the discharging high pressure air for mechanical work. In [8], high pressure air on the discharge side is rerouted to the charging side through an accumulator. In [9], a two way valve is used to achieve cross flow from high pressured discharge side to the charging side. Another approach to efficiency enhancement is to operate the system at lower pressure, thus minimizing the loss associated with energy dissipation on the discharging side. In the current paper, the use of this method for improving efficiency while achieving human power amplification is reported.

In a proportional servo-valve, the flow areas metering the flow in and out of the actuator are mechanically coupled. As a result, it is impossible to control the pressure in both chambers of the actuator independently. Independant metering is an approach [1],[10] that can be used to overcome the coupling between the two flow areas. A schematic of the pneumatic actuator with independent metering is shown in Fig(2). By having two independent valves, the desired tracking objective can be achieved while controlling the operating pressure of the system. In [1], an optimization method is proposed to minimize the operating pressure of a pneumatic actuator for tracking a sinusoidal position trajectory. In [10], the operating pressure is lowered by regulating the pressure on the discharging side to a lower value. This is similar to our proposed method for energy saving, as explained later in the section. None of the above papers are concerned with human power amplification.

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Fig. 1. Operation of pneumatic actuator using a single 5-port, 3-way proportional valve

Human power amplification and minimum operating pressure are the two objectives that determine the command inputs to the two valves. To obtain minimum operating pressure, a function in terms of the error between the current and a desired pressure is defined on the discharge side. By regulating this error function, command inputs to the discharging valve (low pressure side) are obtained. When
the direction of desired force changes, the charging and discharging valves are swapped. It is obtained from analysis that to get a continuously exponentially decreasing error, even in the event of swapping of valves, the desired pressure on the rod side needs to be higher than the desired pressure on the piston side. Note that no formal analysis towards selection of the desired pressure is presented in [10]. The inputs to the charging valve are determined by the force output desired from the actuator.

\[ P_1 \text{V}_1 - P_2 \text{V}_2 \]

where \( \text{m}_i \equiv \frac{P_i \text{V}_i}{RT} \), \( P_i = P_{\text{in}} - P_{\text{atm}} \) corresponds to the gauge pressure, \( A_1, A_2 \) are the areas on either side, \( F \) is the actuator force, and \( F_{\text{e}} \) is the external environment force. The human force, \( F_{\text{h}} \), and the inertia \( M_p \ddot{x}_p \) are obtained moving the actuator up and down in the vertical plane until the reservoir is discharged to a low enough pressure. The dynamics of the inertia being moved by the actuator is given,

\[ M_p \ddot{x}_p = F_{\text{h}} + F_{\text{e}} + F_a \]

where, \( F_{\text{h}} \) is the human force, \( F_{\text{e}} \) is the external environment force, and \( F_a \) is the actuator force. For the purpose of comparing single valve metering with independent metering, the velocity of motion was kept very similar in both the experimental runs. It is to be noted that the proposed method is not restricted to systems with single d.o.f. and can be easily extended to multi d.o.f. systems.

The dynamics of pneumatic actuators has been well studied and reported in literature [3]. In this paper the thermodynamic process in the actuator is assumed to be isothermal. However, the controller design can be suitably extended to adiabatic models, as shown in [2] for human power amplification. The pressure dynamics for an isothermal process can be obtained from ideal gas law as,

\[ \dot{P}_1 = \frac{m_1RT}{V_1} - \frac{P_1}{V_1} \dot{V}_1 \]

\[ \dot{P}_2 = \frac{m_2RT}{V_2} - \frac{P_2}{V_2} \dot{V}_2 \]

where \( P_i, V_i \) correspond to the pressure and volume in the actuator chambers shown in Fig(2), \( T \) is the constant temperature of operation, \( m_i \) corresponds to the mass flow rate to individual chambers and \( R \) is the universal gas constant. The mass flow rate controls the pressure dynamics and is given by [3],

\[ \dot{m}_i = \Psi_i \dot{u}_k \]

\[ \Psi_i = \begin{cases} \frac{C_1 P_i}{\sqrt{T}} & \text{if } \frac{P_k}{P_u} \leq P_{cr} \\ \frac{C_2 P_i}{\sqrt{T}}(\frac{P_k}{P_u})^\frac{1}{2} - \frac{C_2 P_i}{\sqrt{T}}(\frac{P_k}{P_u})^\frac{1}{2} & \text{if } \frac{P_k}{P_u} > P_{cr} \end{cases} \]

where \( u_k = C_f A_o \) is a measure of the flow area open in the valve and is designated as the control input to the actuator, \( k \in (H, L) \) where \( H \) refers to valve on high pressure side and \( L \) refers to the valve on low pressure side, \( C_1 = \sqrt{\frac{2}{R(\gamma - 1)}} \), \( C_2 = \frac{2^{\gamma - 1}}{R(\gamma - 1)} \), and \( P_{cr} = \left( \frac{2}{\gamma + 1} \right)^{\gamma \over \gamma - 1} \). The terms \( F_u, P_d \) correspond to pressures upstream and downstream of the valve, the condition \( \frac{P_u}{P_d} \leq P_{cr} \) corresponds to choked flow and \( \frac{P_u}{P_d} > P_{cr} \) corresponds to unchoked flow. In designing the control input, the valve dynamics are ignored as they are much faster than the system dynamics. Note that these pressures vary depending on whether the chamber is charging or discharging. The force exerted by the actuator is given by,

\[ F_a = P_1 A_1 - P_2 A_2 - P_{\text{atm}}(A_1 - A_2) \]

Using the ideal gas law, the net force exerted by the actuator in Eq(5) can be rewritten as,

\[ F_a(m, x_p) = \frac{RT}{L_{1o} + x_p} \left( \frac{m_{1g}}{L_{1o} + x_p} - \frac{m_{2g}}{L_{2o} - x_p} \right) \]

where \( m_{1g} \equiv \frac{P_{ig} V_i}{RT} \), \( P_{ig} = P_i - P_{\text{atm}} \) corresponds to the gauge pressure, \( A_1, A_2 \) are the areas on either
side of the piston, \(x_p\) is position of the piston, \(L_{1o}, L_{2o}\) correspond to the dead volume (including hose volume) of respective chambers and are obtained as \(L_{1o} = V_{1o}/A_1\), \(L_{2o} = L_s + V_{2o}/A_2\) and \(L_s\) is the stroke length. The position of actuator corresponding to zero force is obtained as,

\[
x_p = \frac{m_1}{m_1 + m_2}L_o - L_{1o}
\]

where \(L_o = L_{1o} + L_{2o}\). Using the above equation, the actuator force in Eq(6) can be written as a spring force,

\[
F_a = -K(m, x)(\bar{x}(m)p) - \bar{x}(m)p
\]

where \(K(m, x) = \frac{RT(m_1 + m_2)}{(L_{1o} + x_p)(L_{2o} - x_p)}\) is the nonlinear spring stiffness. As the relative degree from the actuator force to the designated command input \(u_k\) is one, it is differentiated Eq(5) for control design. On differentiating we get,

\[
\dot{F}_a = -K(m, x_p)\dot{x}_p + \frac{K(m, x_p)}{m_1 + m_2}((L_{2o} - x_p)\dot{m}_1
\]

\[
- (L_{1o} + x_p)\dot{m}_2) + \delta(m, x_p)\dot{x}_p
\]

where \(\delta(m, x_p) = \frac{P_{atm}(\frac{A_1}{L_{1o} + x_p} - \frac{A_2}{L_{2o} - x_p} + K(m, x)\Delta x(\frac{1}{L_{1o} + x_p} - \frac{1}{L_{2o} - x_p}))}{\dot{x}_p}\). Depending on which side is charging, either \(\dot{m}_1\) or \(\dot{m}_2\) will control the force amplification.

**III. CONTROL OBJECTIVE**

The objectives of the controller are two-fold, 1) to obtain human power amplification from the actuator in a passive manner and 2) to operate at minimum possible system pressure. As there is direct interaction between the human and the device, human power amplification is achieved by amplifying the input human force. Therefore, the control objective for high pressure side valve is,

\[
F_a = \rho F_h
\]

where \(\rho\) is the amplification factor. Passive operation is achieved by monitoring the energy flow from the system. The power supplied to the system by external forces is known as the supply rate, and is given by,

\[
s_p(F_h, F_c, \dot{x}_p) := ((\rho + 1)F_h + F_c)\dot{x}_p
\]

where the desired external force \(F_{des}\) is given by amplified human force \(\rho F_h\). To achieve passive operation, the supply rate must satisfy the following condition \(\forall t\),

\[
\int_0^t s_p(F_h, F_c, \dot{x}_p)d\tau \geq -C_o^2
\]

where \(C_o^2\) represents the upper bound on the energy that can be extracted from the system by external forces. From Eq(5) it can be seen that a desired positive force can be obtained for a smaller value of \(P_1\) by keeping \(P_2\) as low as possible. As the direction of force changes, the current discharging side (i.e chamber 1) will now be at a much lower pressure than normal mode of operation, thus requiring lower pressure on the charging side to attain the desired force. Therefore, minimum operating pressure can be achieved by maintaining low pressure on the discharge side.

**IV. CONTROLLER DESIGN**

The pneumatic actuator can be modeled as a combination of ideal velocity source and a nonlinear spring [6]. By controlling the ideal velocity as the velocity of a virtual inertia an energetically passive structure is obtained. This is as shown in Fig(4), wherein the net external forces acting on the co-ordinated system is given by

\[
\begin{align*}
\rho \bar{F}_h - F_a &= \rho \bar{F}_h - F_a \\

\end{align*}
\]

Therefore, the velocity of the virtual mass (ideal velocity) can be inferred as the causal effect of the error between the desired and the actual actuator force. Note that the virtual mass is fictional and its dynamics are only evaluated as part of the controller design. As shown in [2],[6] the desired human power amplification can be achieved through velocity co-ordination between the virtual mass and inertia. This can be verified from Fig(4), wherein the net external forces acting on the co-ordinated system is given by

\[
V_{\dot{E}} := \dot{x}_p - \dot{x}_v \rightarrow 0
\]

On the discharging side of the actuator, the following weighted error in pressure is defined,

\[
J_p(P_1, P_2, F_a) = \begin{cases} (P_2 - (1 + \epsilon_2)P_{atm})A_2 & \text{if } F_a \geq 0 \\
(P_1 - (1 + \epsilon_1)P_{atm})A_1 & \text{if } F_a < 0
\end{cases}
\]

where \(\epsilon_1 > 0\) and \((1 + \epsilon_1)P_{atm}\) is the target operating pressure. The valve command inputs are determined by regulating the above error function.
A. Low pressure side controller

**Theorem 1.** For a smooth actuator force $F_a \in C^n$, if the command input to the valve on low pressure (discharging) side is given by,

$$ u_L = \begin{cases} 
  \frac{V_L}{\Psi^2_{RT}} \left( \frac{P_v}{V_1} + \eta((1 + \epsilon_2)P_{atm} - P_2) \right) & \text{if } F_a \geq 0 \\
  \frac{V_L}{\Psi^2_{RT}} \left( \frac{P_v}{V_1} + \eta((1 + \epsilon_1)P_{atm} - P_1) \right) & \text{if } F_a < 0
\end{cases} \tag{16}
$$

where $\eta$ is a positive constant, and $\epsilon_1$, $\epsilon_2$ are small positive constants satisfying the following condition,

$$ \frac{\epsilon_2}{\epsilon_1} = \frac{A_1}{A_2} \tag{17} $$

and $A_1, A_2$ are the areas on either side of the piston, then the error function in Eq(15) exponentially converges to zero.

**Proof:**

When the actuator force is positive, chamber 2 in Fig(2) is the discharging side. The pressure dynamics on this side of the actuator are given in Eq(3). By substituting Eq(16) in Eq(3) we get,

$$ \dot{P}_2 = -\eta(P_2 - (1 + \epsilon_2)P_{atm}) \tag{18} $$

Using Eq(15), the above equation can be written in terms of the error function as,

$$ \dot{J}_p = -\eta J_p \tag{19} $$

When the actuator force is negative, chamber 1 is the discharging side. The pressure dynamics on this side of the actuator are given in Eq(2). On substituting Eq(16) in Eq(2), the following dynamics for pressure and error function are obtained,

$$ \dot{P}_1 = -\eta(P_1 - (1 + \epsilon_1)P_{atm}) \tag{20} $$

$$ \dot{J}_p = -\eta J_p \tag{21} $$

From Eq(19) and Eq(21) we notice that the error function has same dynamics for both positive and negative force directions. When the direction of force changes, i.e $F_a = 0$, the following condition is obtained on pressure values,

$$ P_1 A_1 - P_{atm} A_1 = P_2 A_2 - P_{atm} A_2 \tag{22} $$

Adding $\epsilon_1 P_{atm} A_1$ to both sides of the above equation and using the condition in Eq(17) we get,

$$ (P_1 - (1 + \epsilon_1)) A_1 = (P_2 - (1 + \epsilon_2)) A_2 \tag{23} $$

which satisfies the following condition,

$$ J_p(F_a = 0^+) = J_p(F_a = 0^-) \tag{24} $$

Thus, the error function is continuous with dynamics given by Eq(19,21). Therefore control input in Eq(16) coupled with the condition in Eq(17) results in an exponentially convergence of the error function.

Note that the controller design for the low pressure side is independant of the control objective on the high pressure side. The minimum attainable pressure for a pneumatic system is atmospheric pressure. However, from Eq(5) it can be noticed that this value for operating pressure will lead to a singularity ($\Psi_i = 0$, if $P_a = P_d = P_{atm}$). In our experiments, a value of 0.5 was arbitrarily assigned to $\epsilon_1$ and so $\epsilon_2$ had a value of 0.45.

B. High pressure side controller

The valve connected at the high pressure side is used to achieve velocity co-ordination control. As described in [2] a passive transformation is used to transform the system states $[\dot{x}_p, \dot{x}_v]$ to obtain the dynamics of the relative system (a.k.a Shape system), and the dynamics of the center of mass (a.k.a Locked system). These dynamics are given by,

$$ M_L\dot{V}_L = F_e + F_h + \rho F_h \tag{25} $$

$$ M_E\dot{V}_E = \phi F_L + F_a - \rho F_h \tag{26} $$

where $M_L = M_p + M_o$ and $M_E = M_p M_o / (M_p + M_o)$. Once co-ordination is achieved i.e $V_E \rightarrow 0$ and $\dot{x}_p = \dot{x}_v = V_L$, the locked system dynamics show that the co-ordinated system would interact with the environment with an amplified human force. The control input is also taken through a series of transformation as given in the following. The details of these transformations are provided in [2] and [7]. Define the input command $u_H$ as,

$$ u_H = \frac{1}{\gamma_3} u_{H_1} + \frac{1}{\gamma_1} \dot{x}_v - \frac{\zeta(\bar{m}_i, x_p, T)}{\gamma_1 F_a} \tag{27} $$

where, if chamber 1 is charging,

$$ \zeta(\bar{m}_i, x_p, T) = \bar{m}_i \Psi_2 RT \ln \left( \frac{L_{2o}}{L_{2o} - \bar{x}_p} \right) \tag{28} $$

$$ \gamma_1 = \frac{1}{F_a} \Psi_1 RT \ln \left( \frac{L_{1o} + \bar{x}_p}{L_{1o} + x_p} \right) \tag{29} $$

$$ \gamma_3 = (L_{2o} - x_p) \Psi_1 / (m_1 + m_2) \tag{30} $$

$$ \beta = (L_{1o} + x_p) \bar{m}_i \tag{31} $$

and if chamber 2 is charging,

$$ \zeta(\bar{m}_i, x_p, T) = \bar{m}_i \Psi_2 RT \ln \left( \frac{L_{1o} + \bar{x}_p}{L_{1o} + x_p} \right) \tag{32} $$

$$ \gamma_1 = \frac{1}{F_a} \Psi_2 RT \ln \left( \frac{L_{2o} - \bar{x}_p}{L_{2o} - x_p} \right) \tag{33} $$

$$ \gamma_3 = (L_{2o} + x_p) \Psi_2 / (m_1 + m_2) \tag{34} $$

$$ \beta = (L_{2o} - x_p) \bar{m}_i \tag{35} $$

and,

$$ u_{H_1} = \bar{u}_{H_1} + \frac{\gamma_3}{\gamma_1} - 1 \bar{x}_v - \beta(x_p, \bar{m}_i) + K(m, x)^{-1} \delta(m, x) \tag{36} $$

The control input to the high pressure side depends on the input to the low pressure through the $\zeta(\bar{m}_i, x_p, T)$ term. The equations of interest for velocity co-ordination are given by,

$$ \dot{V}_E = V_E \tag{28} $$

$$ M_E\dot{V}_E = \phi F_L + F_a - \rho F_h \tag{29} $$

$$ \dot{F}_a = -K_m V_E + K_m \bar{u}_{H_1} \tag{30} $$

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Theorem 2. Given a continuous desired force \( F_{\text{des}} = \rho F_h \), the following choice for input \( \tilde{u}_{H1} \) will achieve asymptotic convergence the co-ordination errors \( q_E \) and \( V_E \):

\[
\tilde{u}_{H1} = K_s^{-1}(\dot{F}_d - \hat{F}_{ex1} - L_z \varepsilon q_E - (L_z - K_p - K_s) V_E - K_d M_E^{-1} \varepsilon q_E)
\]

where \( L_z, K_p, K_d, K_s, \varepsilon \) are positive constants, \( M_E \) is the mass of shape system, \( K_s \) is the spring stiffness, \( \hat{F}_c \) is the error in force and is given by,

\[
F_c = F_a - \rho F_h
\]

\[ Z = F_c - F_{\text{ex}} \] is the difference between the actual and desired force error, where the desired error is given by,

\[
F_{\text{ex}}^d = -K_p q_E - K_d V_E - \hat{F}_{ex1}
\]

and \( \hat{F}_{ex1} \) is the estimate of unknown forces \( F_{ex1} \), and for a positive definite \( \Lambda \) the estimation error \( \hat{F}_{ex1} \) is obtained from the following update law,

\[
\hat{F}_{ex1} = \Lambda^{-1}(-M_E^{-T}K_d^T L_z^{-T} Z - (V_E + \varepsilon q_E))
\]

Proof of the above theorem is similar to the one reported in [2].

V. RESULTS

The controller is implemented on a single d.o.f pneumatic actuator. The experimental setup is as shown in Fig(3). The areas for the pneumatic actuator in the experimental setup are \( A_1 = 0.002 m^2 \) and \( A_2 = 0.0018 m^2 \). Thus the area ratio is obtained as \( \frac{A_1}{A_2} = 0.9 \). To implement independent metering, two FESTO MPYE-5-LF010, 5 port-3 way proportional valves are modified to be 4 port-3 way proportional valves. The amplification factor for the input human force is set to be 7. For the purpose of comparison, both the single valve metering and independent metering control schemes are put through similar velocity profile. The force tracking and velocity co-ordination performance of the controller with single valve metering is shown in Figs(5, 6). The controller performance begins to degrade as the supply pressure decreases. This is more apparent in the velocity co-ordination performance. After about 38s, the supply pressure is too low to achieve the desired force and velocity tracking. From Fig(7), it can be seen that the valve command begins to saturate at this point but cannot provide the desired performance. Note that as the operating pressure decreases, the force magnitude required to achieve the desired velocity decreases. This was experimentally determined to be due to reduction in friction at lower operating pressure.

With independent metering, the force and velocity tracking performance are shown in Figs(8, 9). The velocity co-ordination performance over the same time scale, is much better with independent metering. The force tracking performance is good and is comparable to that obtained with single valve metering. The variation of the supply pressure with for single and independent valves are shown in Fig10. After the same time duration, the pressure drop with independent metering is lower than that with single valve metering. The performance of the actuator degrades considerably when the supply pressure falls below \( 2e5 (N/m^2) \). With single valve metering this happens at around 38s, as observed in Fig(6), while with independent valves it happens at around 76s. Thus in human power amplification with independent metering, the operational time of the actuator is improved by nearly 100%. The error function convergence is shown in Fig(11).

VI. CONCLUSIONS AND FUTURE WORK

In this paper a simple scheme for independent metering with proportional servo valves has been provided. The valve on the discharging side was controlled to lower the operating pressure of the actuator, while the one on charging side was used to obtain the desired force profile. Experimental results show that the proposed method has a doubled the operation time, thus dissipating less energy than a single valve flow metering. A drawback of the proposed method is that the throttling losses at the valves are still present. This loss can be greatly diminished by using on-off valves instead of proportional valves. The cost of two on-off solenoid valves is also much lower than a single proportional servo valve. Independent metering with on-off valves will be investigated in the future.
Fig. 7. Command input to the single valve metering the flow to the actuator

Fig. 8. Human force amplification with independent valve metering

Fig. 9. Velocity co-ordination with independent valve metering

Fig. 10. Comparison of reservoir pressure variation with single and independent valve metering

Fig. 11. Error function

REFERENCES


