Proceedings of the American Control Conference
Arlington, VA June 25-27, 2001

Passive Control of an Electrohydraulic Actuator Using an Electrohydraulic Passive Valve [1]

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Abstract

A passive control scheme for bilateral teleoperated one degree of freedom hydraulic actuator is proposed. The overall system enables a human operating a motorized joystick to feel as if he is manipulating a rigid mechanical tool with which the work environment is also in contact. By ensuring that the closed loop system behaves like a passive two port device, safety and stability when coupled to other systems are improved. The control scheme is developed by first passifying a four way directional control valve via active feedback, and then by the design of a passive teleoperation control based on the low frequency dynamics of the passified valve. The coordination error between the joystick and the hydraulic actuator converges to 0 for sufficiently low manipulation bandwidth. Experimental results verifies the characterisics of the control scheme.

1 Introduction

Many hydraulic systems are required to touch and contact its physical environments, such as in earth-digging, the transport of materials etc. Of these, many are also controlled by human operators on-site via control levels or joysticks. A typical example is a construction worker operating a hydraulic boom-and-bucket to perform an earth digging task. These systems are two-port devices that simultaneously interact and form closed loop systems with both the human operator and the physical environment. It is critical that these systems remain stable and can safely interact with a broad range of environments and human operators. In addition, these systems must be natural and easy for the human operator to control.

Both the safety and the human friendliness aspects of these applications can be enhanced if the system can be shown to be passive. Roughly speaking, a passive system behaves as if it does not generate energy, but only stores, dissipates and releases it. A passive system is inherently safer than a non-passive system because the amount of energy that it can impart on the environment is limited. The well known passivity theorem [11] ensures that a passive system can interact stably with *any* strictly passive system. The latter include a wide variety of physical objects and environments. The inherent safety that passive systems afford was exploited in machines that interact with humans such as smart exercise machines [7], bilateral teleoperated manipulators [4, 9, 2] and Cobots [1]. It is also exploited in the passive velocity field control (PVFC) methodology [8], which by ensuring that the closed loop system is passive, enables *mechanical* systems to interact safely with the often ill characterized environment while achieving coordination goals, such as for contour following with and without force control [6, 3].

In this paper, we develop a passive control methodology for the bilateral teleoperated control of a hydraulic actuator via a force feedback joystick. The objective is to control the telemanipulation system so that it appears to the work environment and the human operator, as if they are both interacting with a common virtual passive rigid mechanical tool after appropriate power and kinematic scalings. This enables the human to be kinesthetically and energetically connected to its work environment. For example, it allows a human operating an excavator to feel as if he is manipulating a spade. The control objectives in [4, 9, 2] are similar except that they are concerned with electromechanical, rather than hydraulics, machines. An impediment to the development of passive control schemes for electrohydraulic actuators is that, unlike mechanical and electromechanical systems, electrohydraulic valves are not inherently passive. This difficulty was overcome recently in [5] in which passification methods are proposed, via either structural modification or active feedback control, to render a single-stage four way directional control valve a passive two port device. In this paper, we develop a control system based on a valve that has been passified using the active feedback passification method in [5]. The teleoperation controller is designed to be passive with respect to an appropriate supply rate so that the

[1] Research supported by the National Science Foundation.
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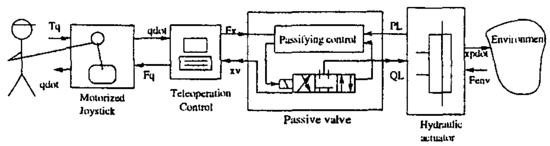

Figure 1: Configuration of the teleoperated system.

overall system is passive, and which, at sufficiently slow manipulation achieves asymptotic coordination of the joystick and the hydraulic actuator.

The rest of this paper is organized as follows. The models for the various subsystems are described in Section 2. Section 3 describes the technique for passifying the four way directional control valve, which is the first step in the control design process. Section 4 describes the passive teleoperation control law. Implementation results are given in Section 5. Section 6 contains concluding remarks.

2 System Modeling and Control Objective

The hydraulic teleoperation system consists of three systems, a direct acting four way directional flow control valve (together with a pump that supplies a constant supply pressure), an ideal one degree of freedom double ended hydraulic actuator, and a motorized joystick. The configuration of the control system is shown in Fig. 1.

Valve model We consider a symmetric, matched, critically centered four way directional valve. Let x_v be the spool displacement x_v. Ignoring flow saturation, the output flow equation in terms of x_v is usually given as [10]:

$$Q_L(x_v, P_L) := \begin{cases} \frac{C_d}{\sqrt{\rho_h}} w x_v \sqrt{P_s - \frac{x_v}{|x_v|} P_L}; & \operatorname{sgn}(x_v) P_L < P_s \\ -\frac{C_d}{\sqrt{\rho_h}} w x_v \sqrt{\frac{x_v}{|x_v|} P_L - P_s}; & \operatorname{sgn}(x_v) P_L \geq P_s \end{cases} \tag{1}$$

where $C_d > 0$ is the orifice coefficient, $\rho_h > 0$ is the fluid density, $w > 0$ is the gradient of the orifice area with respect to the spool position, P_s is the constant supply pressure. In normal operation, $|P_L| < P_s$ so only the first case statement in (1) is commonly used. In the abnormal case $\operatorname{sgn}(x_v)P_L \geq P_s$, Eq.(1) is valid as long as the valve does not cavitate. Following [5], we can rewrite (1) into:

$$Q_L(x_v, P_L) = K_q x_v - K_t(x_v, P_L) P_L \tag{2}$$

where

$$K_t(x_v, P_L) = \int_0^1 \frac{\partial Q_L}{\partial P_L}(x_v, l \cdot P_L) dl. \tag{3}$$

A convenient formula for $K_t(x_v, P_L)$ for the usual situation when $P_s \geq \operatorname{sgn}(x_v)P_L$ is

$$K_t(x_v, P_L) = \frac{K_q |x_v|}{\sqrt{P_s}\left(\sqrt{P_s} + \sqrt{P_s - \operatorname{sgn}(x_v) P_L}\right)}.$$

It can be shown that $K_t(x_v, P_L)$ is non-negative for all x_v and P_L. Thus, the valve can be interpretated to comprise an ideal flow source $K_q x_v$, and a shunt conductance $K_t(x_v, P_L)$.

For the purpose of the present paper, we can identify from Eq.(1), a load adjusted flow gain $\bar{K}_q(\operatorname{sgn}(x_v), P_L)$ such that

$$Q_L = \bar{K}_q(\operatorname{sgn}(x_v), P_L) x_v. \tag{4}$$

Notice that $\bar{K}_q(\operatorname{sgn}(x_v), P_L)$ depends only $\operatorname{sgn}(x_v)$ but not on x_v itself and $\bar{K}_q(\operatorname{sgn}(x_v), P_L)$ decreases as $\operatorname{sgn}(x_v)P_L$ increases. Moreover, whenever $|P_L| < P_s$, it is possible to choose an appropriate x_v to achieve any desired flow Q_L.

The dynamics of the spool are given by:

$$\epsilon \ddot{x}_v = u \tag{5}$$

where ϵ is the inertia of the spool, u is the control force. In an actual direct acting proportional control valve, in addition to the control input, u may include also centering spring forces, and flow forces etc.

It has been shown [5] that given any constant $x_v > 0$, the directional control valve is *not passive* with respect to the supply rate $s(P_L, Q_L) := -P_L Q_L$.

Hydraulic Actuator We assume that the hydraulic actuator is an ideal double ended cylinder with area A_p. Its flow-velocity and pressure-force relationships are given by:

$$A_p \dot{x}_p = Q_L; \qquad F_{env} = A_P P_L \tag{6}$$

where x_p is the actuator displacement, F_{env} is the environment force acting on the piston, Q_L and P_L are the flow into the actuator and the load pressure. If Q_L is provided by the valve as in Eq.(2), then:

$$\dot{x}_p = Q_L / A_p = \frac{\bar{K}_q(\operatorname{sgn}(x_v), P_L)}{A_p} x_v. \tag{7}$$

The double ended hydraulic actuator is passive with respect to the supply rate:

$$s_{piston}((F_{env}, \dot{x}_p), (Q_L, P_L)) := -F_{env}\dot{x}_p + P_L Q_L.$$

Motorized Joystick A motorized joystick is an inertia with dynamics given by:

$$M\ddot{q} = F_q + T_q \tag{8}$$

where F_q is the control force, and T_q is the force supplied by the operator. Using its kinetic energy $\frac{1}{2}M\dot{q}^2$ as its storage function, it is easy to see that the motorized joystick is passive with respect to the supply rate:

$$s_{joystick}((F_q, \dot{q}), (T_q, \dot{q})) := F_q \dot{q} + T_q \dot{q}.$$

2.1 Control objectives

Given a kinematic scaling $\alpha \in \Re$ and power scaling $\rho > 0$, our goal is to control the hydraulic actuator and the motorized joystick so that

1) the joystick and the hydraulic actuator is coordinated: $\alpha\dot{q} - \dot{x}_p \to 0$;

2) the closed loop control system is passive with respect to the supply rate:

$$s_{total}((F_{env}, \dot{x}_p), T_q\dot{q}) := -F_{env}\dot{x}_p + \rho T_q \dot{q},$$

where $T_q\dot{q}$ is the power input by the human operator and $-F_{env}\dot{x}_p$ is the power input by the work environment.

3 Passification of the control valve

As mentioned in Section 1, the first step in the control design is to passify the valve. The following theorem from [5] provides a feedback control law that renders the valve a passive two port system, with respect to a supply rate that consists of the hydraulic power and a supply function related to the command.

Theorem 1 ([5]) *Let F_x be the auxillary, exogenous command input to the valve, and let*

$$z := \dot{x}_v - \frac{1}{B}(F_x - AP_L).$$

Consider the control law of the spool given by

$$u = -B\dot{x}_v - AP_L + F_x + F_{act} \tag{9}$$

where B is a damping coefficient, A is the pressure feedback gain, F_{act} is the passification term given by:

$$F_{act} = -\gamma B x_v - \left[\widehat{\dot{F}_x} - \widehat{A\dot{P}_L}\right] - g_2(t)sgn(z) \tag{10}$$

where γ is a positive constant and $\hat{\cdot}$ denotes the best estimate of the argument.

If $g_2(t)$ in (10) is defined so that

$$sgn(z(t))g_2(t) > (\dot{F}_x(t) - A\dot{P}_L(t)) - \left[\widehat{\dot{F}_x(t)} - \widehat{A\dot{P}_L(t)}\right],$$

then the critically lapped four way two-port valve is passive with respect to the supply rate

$$s((P_L, F_x), x_v) := [-P_L Q] + \frac{K_q}{A}F_x x_v. \tag{11}$$

Remarks:

- The passified valve in theorem 1 can be thought of as a two-port device with the command port variables F_x, x_v, and the output port variables P_L and Q_L. In this case, the 2-port device has associated with it a power gain of K_q/A.
- The passification control requires estimates of the time derivatives of the load pressure P_L and of the auxillary valve command F_x. These can be computed from the pressure chamber dynamics and from the teleoperation control (to be designed).
- When the estimate of the term $\dot{F}_x(t) - A\dot{P}_L(t)$ is accurate, the valve behavior is given by:

$$\frac{x_v(s)}{F_x(s) - AP_L(s)} = \frac{s + \frac{B}{\epsilon}}{Bs(s + \frac{B}{\epsilon}) + \gamma B^2} \tag{12}$$

  Thus, at low frequency operation, the passified valve dynamics are given by:

$$F_x - AP_L = K_{sp}x_v, \qquad K_{sp} = \gamma B\epsilon, \tag{13}$$

  where K_{sp} is the equivalent spring rate. If the dynamics of the passified valves are designed to be critically damped, the bandwidth within which the low frequency approximation is appropriate will be $B/(2\epsilon)$. The subsequent teloperator control design will be based on the low frequency approximation in Eq.(13) of the passive vale dynamics. Thus, for good performance, the bandwidth of operation should be less than $B/(2\epsilon) rad/s$.
- As will be apparent in Section 4 after we have presented the properties of the teleoperated hydraulic system, the equivalent spring rate K_{sp} contributes to dissipation in the passified valve.

The approach to designing a passive hydraulic teleoperation system is via the interconnection of passive two-port systems with compatible supply rates according to the following key lemma.

Lemma 1 *Consider the two two-port systems A and B with respective port variables (u_A^1, y_A^1), (u_A^2, y_A^2), and (u_B^1, y_B^1), (u_B^2, y_B^2). Suppose that system A is passive with respect to the supply rate*

$$s_A((u_A^1, y_A^1), (u_A^2, y_A^2)) = u_A^1 \cdot y_A^1 + \gamma_1 u_A^2 \cdot y_A^2$$

and system B is passive with respect to the supply rate

$$s_B((u_B^1, y_B^1), (u_B^2, y_B^2)) = -u_B^1 \cdot y_B^1 + \gamma_2 u_B^2 \cdot y_B^2$$

where $\gamma_1 > 0$, $\gamma_2 > 0$. Then, the interconnection given by: $u_A^2 := y_B^1$, $u_B^1 := y_A^2$ is passive with respect to the supply rate:

$$s_{AB}(u_A^1, y_A^1), (u_B^2, y_B^2)) := u_A^1 \cdot y_A^1 + \gamma_1\gamma_2 u_B^2 \cdot y_B^2.$$

Proof: Using the fact that $s_{AB}(u_A^1, y_A^1), (u_B^2, y_B^2))$ $= s_A((u_A^1, y_A^1), (u_A^2, y_A^2)) + \gamma_1 s_B((u_B^1, y_B^1), (u_B^2, y_B^2))$, and the passivity properties of systems A and B, the result is obtained by the direct computation of $\int_0^t s_{AB}(u_A^1, y_A^1), (u_B^2, y_B^2))d\tau$. ∎

As an example, this lemma shows that the interconnection of the passified valve and the hydraulic actuator is passive with respect to the supply rate:

$$s_{valve/piston}((F_{env}, \dot{x}_p), (F_x, x_v)) = -F_{env}\dot{x}_p + \frac{K_q}{A}F_x x_v.$$

The passivity property of a teleoperation controller (Fig. 1) for the passified valve input F_x and for the joystick input F_q that would generate the desired passivity property of the overall teleoperator system is given by the following corollary.

Corollary 1 *Suppose that the critically lapped proportional control valve has been passified as in Theorem 1. Consider a two port system which serves as a controller with port variables* $(\dot{q}, T_q)$, (x_v, F_x). *Then the interconnection of the hydraulic actuator* (6), *the passified valve, the controller, and the motorized joystick is passive respect to the supply rate:*

$$s_{total}((F_{env}, \dot{x}_p), T_q\dot{q}) := -F_{env}\dot{x}_p + \rho T_q \dot{q} \tag{14}$$

where $\rho > 0$ *is a positive power scaling factor, if the controller is passive with respect to the supply rate:*

$$s_{controller}((x_v, F_x), (\dot{q}, F_q)) = -\frac{K_q}{A} F_x x_v - \rho F_q \dot{q} \tag{15}$$

4 Passive teleoperation controller

In addition to ensuring that closed loop control system is passive with respect to the supply rate in Eq.(14), the joystick and the hydraulic actuator should also be coordinated up to a kinematic scaling α. Let the coordination error be $E := \alpha q - x_p$. To preserve passivity, our controller will be designed so that it is passive with respect to the supply rate in Eq.(15) as specified by Corollary 1. We shall design the control law to have good coordination performance for relatively slow manipulations, specifically when the passified valve dynamics can be well approximated by the static relationship Eq.(13):

$$K_{sp} x_v = F_x - AP_L$$

where $K_{sp} = \gamma B \epsilon$, is determined by the control used in the passification algorithm in Theorem 1. Roughly speaking, Eq.(13) will be valid if the frequency of operation is lower than $B/2\epsilon$. If the frequency of operation is higher, coordination performance will degrade but the passivity property of the overall system will still be valid. *Hence, safety will not be compromised.*

If the static valve dynamics (13) are valid, then we can manipulate F_x to control x_v and hence from (4), also $\dot{x}_p$, so that $x_p(t) \to \alpha q(t)$. One such control is:

$$F_x = AP_L + \frac{K_{sp} A_p}{\bar{K}_q(t)} [\lambda(t) E(t) + \alpha \dot{q}(t)], \tag{16}$$

where $\lambda(t) > 0$ will be determined later. Because $Q_L = A_p \dot{x}_p = \bar{K}_q(t) x_v$ (Eq.(7)), this control law will generate $\dot{x}_p = \lambda(t) E + \alpha \dot{q}$, so that

$$\dot{E} = \alpha \dot{q} - \dot{x}_p = -\lambda(t) E \tag{17}$$

and $E(t) \to 0$ exponentially if $\lambda(t) \geq \underline{\lambda} > 0$.

Next, we design the control law for the motorized joystick. The goal here is to ensure that the controller is passive with respect to the supply rate $s_{controller}(\cdot)$ in (15) as suggested in Corollary 1. If the coordination error $E(t) = \dot{E}(t) = 0$, we have $\bar{K}_q(t) x_v = A_p \dot{x}_p = A_p \alpha \dot{q}$ and $s_{controller}((x_v, F_x), (\dot{q}, F_q)) = 0$ if

$$\begin{aligned} 0 &= AP_L x_v + \frac{K_{sp} A_p}{\bar{K}_q(t)} \alpha \dot{q}(t) x_v + \frac{\rho A}{\bar{K}_q(t)} F_q \dot{q} \\ &= \frac{AP_L A_p \alpha}{\bar{K}_q(t)} \dot{q} + \frac{K_{sp} A_p}{\bar{K}_q(t)} \alpha \dot{q}(t) x_v + \frac{\rho A}{K_q} F_q \dot{q} \end{aligned}$$

Therefore, after $E(t) \to 0$, the joystick control should be:

$$F_q = \frac{K_q}{\rho A} \left[-\frac{K_{sp} A_p \alpha}{\bar{K}_q(t)} x_v - \frac{A_p \alpha}{\bar{K}_q(t)} AP_L \right]. \tag{18}$$

We now propose a dynamic control law that guarantees that the desired passivity property in Corollary 1 is satisfied. In addition, the control law should generate the valve control Eq.(16) at nearly all times, and the joystick control Eq. (18) after $E(t) \to 0$. The controller contains the dynamics of a fictitious flywheel, with inertia M_f and speed $\dot{f}$, which is used to store energy temporarily. The control outputs and the flywheel dynamic update law are given by:

$$\begin{pmatrix} \frac{A_p}{K_q} F_q \\ F_x \\ M_f \ddot{f} \end{pmatrix} = \Omega_2(t) \begin{pmatrix} E \\ \dot{q} \\ x_v \\ \dot{f} \end{pmatrix} \tag{19}$$

where

$$\Omega_2(t) = \begin{pmatrix} -\gamma\alpha & 0 & -\frac{\alpha K_{sp} A_p}{\bar{K}_q(t)} & -\frac{\alpha A_p AP_L}{\bar{v} \bar{K}_q(t)} \\ \frac{\lambda(t) K_{sp} A_p}{\bar{K}_q(t)} & \frac{\alpha K_{sp} A_p}{\bar{K}_q(t)} & 0 & \frac{AP_L}{\bar{v}} \\ 0 & \frac{\alpha A_p AP_L}{\bar{v} \bar{K}_q(t)} & -\frac{AP_L}{\bar{v}} & 0 \end{pmatrix} \tag{20}$$

$$\bar{v} = \begin{cases} \dot{f} & \text{if } |\dot{f}| > f_0 \\ \text{sgn}(\dot{f}) f_0 & \text{otherwise} \end{cases}, \tag{21}$$

$$\lambda(t) = \frac{\gamma \bar{K}_q^2(t)}{K_{sp} A_p^2} \geq 0, \tag{22}$$

and $\gamma > 0$ is a gain. Notice that $\lambda(t)$ will be strictly positive if $|P_L| < P_s$, that F_x is exactly (16) if $\bar{v} = \dot{f}$, and that F_q is exactly Eq.(18) when $E(t) = \dot{E}(t) = 0$.

Proposition 1 *The controller given in Eqs.*(19)-(22) *is a passive two-port system with respect to the supply rate Eq.*(15)*:*

$$s_{controller}((x_v, F_x), (\dot{q}, F_q)) = -\frac{K_q}{A} F_x x_v - \rho F_q \dot{q}.$$

Therefore, by Corolloary 1, the overall teleoperator system is passive with respect to the supply rate:

$$s_{total}((F_{env}, \dot{x}_p), T_q \dot{q}) := -F_{env} \dot{x}_p + \rho T_q \dot{q}$$

where $\rho > 0$ *is a power scaling factor.*

Proof: Consider the storage function

$$W = \frac{1}{2}\gamma E^2 + \frac{1}{2}M_f \dot{f}^2.$$

Then, if we define $\Psi^T(t) = [E, \dot{q}, x_v, \dot{f}]$, we have

$$\dot{W} + \frac{A\rho}{K_q} F_q \dot{q} + F_x x_v = \Psi^T(t)\Omega(t)\Psi(t) = 0$$

where the matrix $\Omega(t) :=$

$$\begin{pmatrix} 0 & \gamma\alpha & -\frac{\gamma \bar{K}_q}{A_p} & 0 \\ -\gamma\alpha & 0 & -\frac{\alpha K_{sp} A_p}{\bar{K}_q(t)} & -\frac{\alpha A_p}{\bar{v}\bar{K}_q(t)} A P_L \\ \lambda(t)\frac{K_{sp}A_p}{\bar{K}_q(t)} & \frac{\alpha K_{sp} A_p}{\bar{K}_q(t)} & 0 & \frac{AP_L}{\bar{v}} \\ 0 & \frac{\alpha A_p}{\bar{v}\bar{K}_q(t)} A P_L & -\frac{AP_L}{\bar{v}} & 0 \end{pmatrix}.$$

Notice that the last 3 rows of $\Omega(t)$ are $\Omega_2(t)$ in (19), and that $\gamma \dot{E} = \Omega_{1*}(t)\Psi$. With the choice of $\lambda(t)$ in Eq.(22), $\Omega(t)$ will be skew symmetric. Hence,

$$W(t) - W(0) = \int_0^t s_{controller}((x_v, F_x), (\dot{q}, F_q))d\tau$$

$$\Rightarrow \int_0^t s_{controller}((x_v, F_x), (\dot{q}, F_q))d\tau \geq -W(0).$$

■

Flywheel speed If the flywheel speed is always larger than the design threshold, $|\dot{f}(t)| \geq f_0, \forall t$, then the F_x entry in the controller in Eq.(19) is the same as Eq.(16), which in turn ensures that that the coordination error dynamics are given by Eq.(17) which is convergent. This is indeed the case since we can choose an initial value $\dot{f}(0)$ such that $|\dot{f}(t)| \geq f_0$ for all t. To see that this is true, suppose that at some time t, $\dot{f}(t) > f_0$,

$$\begin{aligned} \frac{d}{dt}\frac{M_f}{2}\dot{f}^2 &= \dot{f}\left(\frac{\alpha A_p}{\bar{\dot{f}}\bar{K}_q(t)} A P_L \dot{q} - \frac{AP_L}{\bar{\dot{f}}} x_v\right) \\ &= \frac{AP_L}{\bar{K}_q(t)}(\alpha A_p \dot{q} - \bar{K}_q(t) x_v). \end{aligned}$$

Now since $\bar{K}_q(t)x_v = A_p \dot{x}_p = A_p(\alpha\dot{q} - \dot{E})$,

$$\frac{d}{dt}\frac{M_f \dot{f}^2}{2} = \frac{AP_L}{\bar{K}_q(t)}\dot{E}.$$

This means that as long as $\bar{K}_q(t) \geq a > 0$, for some a, (which is guaranteed if $|P_L(t)|$ is strictly smaller than the supply pressure P_s), one can choose initial flywheel speed $\dot{f}(0)$ such that:

$$\frac{M_f}{2}\dot{f}(0)^2 - f_0^2 > \frac{AP_L}{a}E(0). \tag{23}$$

Then, the energy in the flywheel will not be lower than f_0. Hence, $\bar{v}(t) = \dot{f}(t)$ at all times.

Haptics Property We know investigate what the human operator feels when operating the joystick. Considering the F_q row in Eq.(19) and the joystick dynamics Eq.(8), it is easy to show that after convergence of $E \to 0$,

$$M\ddot{q} = -\left(\frac{\alpha K_q K_{sp} A_p}{A\rho \bar{K}_q(t)}\right)\dot{q} - \left(\frac{\alpha K_q}{\rho \bar{K}_q(t)}\right) F_{env} + T_q. \tag{24}$$

Eq.(24) can be interpreted to be a scaled version of the the environment force F_{env} and the operator force T_q acting commonly on the joystick with damping. This is consistent with the design philosophy for teleoperation in [4, 9, 2] that the teleoperator should appear to be a rigid mechanical tool to the human operator.

Remarks:

1. The force scaling factor is $\frac{\alpha K_q}{\rho \bar{K}_q(t)}$, which is smaller than the expected value α/ρ for a lossless system with power scaling ρ and kinematic scaling α. Moreover, the damping coefficient $\left(\frac{\alpha K_q K_{sp} A_p}{A\rho \bar{K}_q(t)}\right)$ also increases as $|P_L|$ increases (hence $\bar{K}_q(t)$ decreases). These nonlinear effects are due to the fact that as the load pressure increases, the apparent power loss in the passified valve increases and the effectiveness of the valve to deliver flow decreases.

2. The passified valve's equivalent spring rate K_{sp} in Eq.(13) contributes to the sluggishness of the joystick. Thus, as remarked in Section 3, K_{sp} corresponds to the power loss in the passified valve. Ideally, it should be small to decrease the joystick damping. Unfortunately, this would compromise the bandwidth in which the low frequency approximation is valid. It can be shown from (13) that the optimal tradeoff is limited by the spool mass ϵ (smaller the better).

We summarize the property of the passive hydraulic teleoperation controller.

Theorem 2 *Under the valve passification control* (9)-(10) *and the teleoperation controller* (19)-(22), *the complete teleoperated hydraulic actuator system is* passive *with respect to the supply rate given by* (14). *Furthermore, if*

a. the passified valve dynamics are adequately approximated by the low frequency approximant (13),

b. for all $t \geq 0$, the load pressure $|P_L(t)|$ is strictly less than P_s so that there exists $a > 0$, such that $\bar{K}_t(t) \geq a > 0$,

c. the initial state of the fictitious flywheel $\dot{f}(0)$ in (19) *is initialized according to* (23),

then,

1. *the fictitous flywheel speed $|\dot{f}(t \geq 0)|$ will always be greater than the threshold f_0;*
2. *the coordination error $E(t) = \alpha\dot{q} - \dot{x}_p$ converges exponentially to 0;*
3. *as $t \to \infty$, the haptics property of the joystick will be given by* (24).

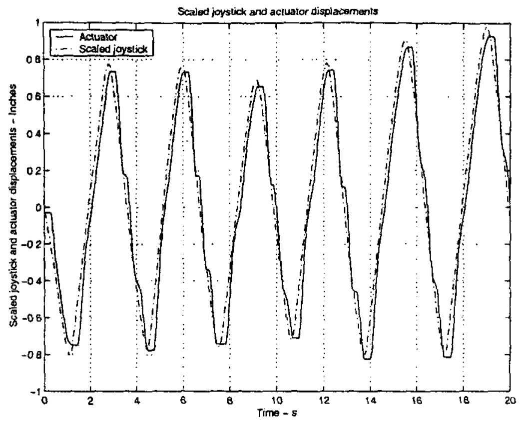

Figure 2: Displacements of scaled joystick and actuator during joystick manipulation

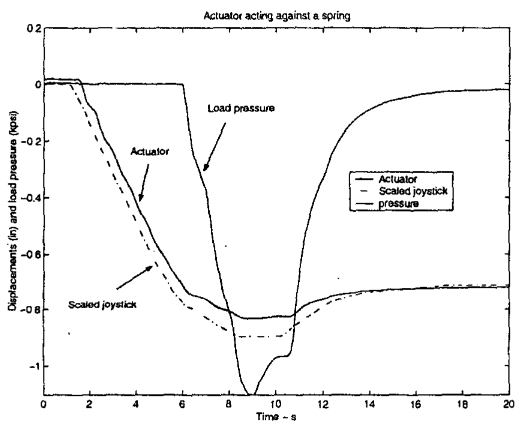

Figure 3: Displacements of scaled joystick and actuator when actuator is interaction with a spring.

5 Experimental results

The proposed teleoperation control scheme was implemented experimentally using a Parker-Hannafin D1FS direct acting proportional control valve. The joystick is actuated by a MicroMo DC motor. Figure 2 shows the response of the joystick and the actuator when only the joystick is manipulated ($F_{env} = 0$). The parameters used are $B/\epsilon = 30rad/s$, $\alpha = 3$, $\rho = 40000$. The maximum coordination error is 0.05in. Notice the relatively sluggish response of the actuator when the joystick is slow. This is probably due to significant deadband in the valve with overlapped. The response of the system in the presence of environment force is shown in Fig. 3. In this scenario, $\alpha = 0.3$ and $\rho = 2000$ to emphasize the joystick motion. The user first manipulates the actuator towards a leaf spring and then releases it. Throughout the manipulation, the actuator and the joystick are coordinated ($\pm 0.06in$ error). Notice especially the spring back effect when the user releases the joystick.

6 Concluding Remarks

A passive bilateral teleoperation control scheme is proposed for a single degree of freedom electrohydraulic actuator and a motorized joystick. The design method uses an active feedback passified valve, and uses the low frequency approximation to the passified valve dynamics. To further improve performance, high frequency passified valve dynamics as well as the the presence of deadband should be taken into account.

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