PASSIVE BILATERAL TELEOPERATION OF A HYDRAULIC ACTUATOR USING AN ELECTROHYDRAULIC PASSIVE VALVE

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Abstract

A passive control scheme for the bilateral teleoperation of a electrohydraulic actuator and a motorized joystick 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 using previously developed active feedback to passify a four way proportional directional control valve, and then by the design of an intrinsically passive teleoperation controller. The coordination error between the joystick and the hydraulic actuator converges to zero for sufficiently low manipulation bandwidth. Experimental results verify the characteristics of the control scheme.

Keywords: hydraulic teleoperation, bilateral teleoperation, passive control, passivity, electrohydraulic actuator

1 Introduction

Many hydraulic systems are required to touch and contact their physical environments in applications such as earth-digging, the transport of materials etc. Of these, many are also controlled by human operators on-site via control levers 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 to the environment is limited. The well known passivity theorem (Vidyasagar, 1993) ensures that a passive system can interact stably with *any* strictly passive system which includes all physical objects and environments that do not contain an energy source. Even human operators have been shown to be indistinguishable from a passive system (Hogan, 1989). Notice that this interaction stability property cannot be guaranteed by merely stable systems, as stable systems can be destabilized when physically coupled with another stable system.

The inherent safety that passive systems afford was exploited in electromechanical machines that interact with humans such as smart exercise machines (Li and Horowitz, 1997), bilateral teleoperated manipulators (Anderson and Spong, 1989; Lee and Li, 2003; Lee and Li, 2002), haptic interface (Hannaford and Ryu, 2002), Cobots (Colgate et al, 1996), coding natural robot autonomous behavior (Li and Horowitz, 1999), and robot contour following with and without force control (Li and Horowitz, 2001; Li and Li, 2000). The first application of the passivity concept to hydraulic systems seems to be (Li, 2000).

Previous attempts to improve operation productivity and safety of hydraulic construction equipment from a controls point of view include human remote control with the use of vision feedback investigated in (Chen et al, 1996); supervisory control approaches for autonomous operation investigated in (Chen et al, 1996; Cannon and Singh, 2000) in which the soil / machine interactions are explicitly modeled and taken into account; and artificial intelligence approach for autonomous operation as investigated in (Bradley and Seward,

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1998). Teleoperation approaches in which direct human control is assumed, were studied both in Europe (Starzewski and Skibiniewski, 1989; Luengo and Barrientos, 1998) and by the University of British Columbia (UBC) group in Canada (Parker et al, 1993; Salcudean et al, 1999; Salcudean et al, 1998; Salcudean et al, 1997; Tafazoli et al, 1996; Tafazoli et al, 1999). The control goal undertaken in the pioneering work on the teleoperation of excavators by Lawrence, Salcudean and co-workers at UBC is to install an ideal transparent mapping between human / machine interaction, and the machine / environment interaction. Their approaches are mainly based on impedance control (especially hybrid impedance control) with the assumption that the dynamics of the hydraulics can be abstracted after the use of simple low level control. Because ideal transparency is required in their problem formulation, some knowledge of the environment impedances has to be assumed in this approach. These impedances cannot be precisely known in reality. Such control schemes, unfortunately, cannot guarantee passivity. In most teleoperation scenarios, it is necessary to develop a controller that apart from other objectives, also guarantees the stability of operator-machine-work environment interaction. Our research approach to guaranteeing this interaction stability with a broad class of environments is by ensuring that the machine is passive. This is in contrast to previous approaches in which passivity is not guaranteed.

In this paper, a control methodology for a bilateral teleoperated hydraulic actuator and a force feedback joystick is proposed. 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. The control scheme 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 (Li and Lee, 2003; Lee and Li, 2002) are similar except that they are concerned with electromechanical, rather than hydraulic machines. The proposed control scheme ensures that the system has the passivity property without abstracting away the valve dynamics. 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 in (Li, 2000) in which passification methods are proposed, via either structural modification or active feedback control, to transform a single-stage four way proportional directional control valve into a passive two port device. Similar approach was later developed for two-stage valves (Krishnaswamy and Li, 2002), and generalized to the passification of other mechatronic devices (Li and Ngwompo, 2002).

In the present paper, we develop a teleoperation control system based on a valve that has been passified using the active feedback passification method in (Li, 2000). The teleoperation controller, which controls both the passified valve and the motorized joystick, is designed to be intrinsically passive with respect to an appropriate supply rate so that the overall system is *always* passive, and which, for sufficiently slow manipulation, achieves asymptotic coordination of the joystick and the hydraulic actuator.

The rest of this paper is organized as follows. The key concepts of passivity are reviewed in section II. The models for the various subsystems are described in section III. Section IV reviews the technique for passifying the four way proportional directional control valve, which is the first step in the control design process. Section V describes the passive teleoperation control law. Implementation results are given in Section VI. Section VII contains concluding remarks.

2 Preliminaries

Consider a dynamical system with input u and output y. A supply rate for a system can be defined to be any function of the input/output pair $s:(u, y) \to s(u, y) \in R$ that is L_{1e} integrable in time. Following (Willems, 1972), a system is said to be dissipative or passive with respect to the supply rate $s(u, y)$ if, for a given initial condition, there exists a constant c so that for all time t and for all input $u(\cdot)$,

$$-\int_0^t s(u(\tau), y(\tau))d\tau \le c^2 \tag{1}$$

For physical systems, useful supply rates are those associated with power input into the system. For most physical systems, $u(t)$ and $y(t)$ are the complementary effort/flow or flow/effort variable pairs. For such sys tems, the supply rate formed by the inner product of the effort (e.g. force, pressure, voltage) and the flow (e.g. velocity, flow, current) variables quantifies the physical power input to the dynamical system.

The inequality 1 then, means that the net energy that can be extracted from a passive system is finite and bounded by the initial energy of the dynamical system, c^2.

For a two-port system that interacts with two environments via the input/output variables (u_1, y_1), and (u_2, y_2) at the two respective ports, a system is said to be passive with a power scaling $\rho > 0$, if

$$\int_0^t [\rho u_1^T(\tau) y_1(\tau) + u_2^T(\tau) y_2(\tau)] d\tau \ge -c^2 \tag{2}$$

Here, the supply rate for the two-port system $s((u_1, y_1), (u_2, y_2))$ consists of the sum of the power input at port 1 scaled by ρ and the power input at port 2. For the teleoperator system, port 1 can be the human interaction port, and port 2 is the work environment port. Then, $\rho > 1$ and $\rho < 1$ correspond to the human power amplification and attenuation factors respectively.

To demonstrate that a system is passive with respect to a supply rate $s(u, y)$, it suffices to define a storage function $W(x)$, which is a positive scalar function of the state x, with the property that:

$$\frac{d}{dt} W(x(t)) \le s(u, y)$$

Integrating with respect to time, gives

$$\int_0^t s(u, y)\, d\tau \geq -W(x(0))$$

For physical systems, the total energies of the systems are good candidates for storage functions.

3 System Modelling and Control Objectives

The hydraulic teleoperation system consists of three physical subsystems, a direct acting four way proportional directional flow control valve (together with a pump that supplies a constant supply pressure), a one degree of freedom double ended hydraulic actuator, and a motorized joystick. The teleoperation controller, usually implemented on a computer, is to be developed later. The configuration of the teleoperated system is shown in Fig. 2. The model of each hardware subsystem is now described.

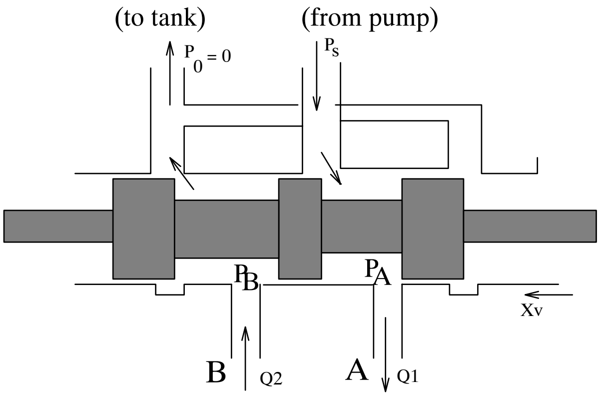


Fig. 1: *Four way, three land proportional directional control valve*

3.1 Single Stage Control Valve Model

Consider a symmetric, matched, critically centred four way proportional directional control valve which is shown in Fig. 1. Assume that the valve is supplied by a constant pressure source, and is connected to a flow-conserving device, such as a double-ended actuator or a motor so that the output and return flows are the same. Let x_v be the spool displacement, and $P_L = P_A$ ñ P_B is the load pressure, i.e. the differential pressure between the two flow ports connected to the valve. Then the output flow Q_L is given by (Merritt, 1967):

$$Q_L(x_v, P_L) := \begin{cases} \dfrac{C_d}{\sqrt{\rho_h}} w x_v \sqrt{P_s - \dfrac{x_v}{|x_v|} P_L}, & \operatorname{sgn}(x_v) P_L < P_s \\ -\dfrac{C_d}{\sqrt{\rho_h}} w x_v \sqrt{-P_s + \dfrac{x_v}{|x_v|} P_L}, & \operatorname{sgn}(x_v) P_L \geq P_s \end{cases} \tag{3}$$

where $C_d > 0$ is the orifice discharge 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 Eq. 3 is commonly used. Following (Li, 2000), Eq. 3 can be rewritten as:

$$Q_L(x_v, P_L) = K_q x_v - K_t(x_v, P_L) P_L \tag{4}$$

where

$$K_q = C_d w \sqrt{\frac{P_s}{\rho_h}} \tag{5}$$

$$K_t(x_v, P_L) = \int_0^1 \frac{\partial Q_L}{\partial P_L}(x_v, P_L \cdot l)\, dl \tag{6}$$

are the no-load flow gain, and the mean slope of the P_L - Q_L curve between the zero load pressure point and the operating load pressure point respectively. In Eq. 6, l is a dummy integration variable. Notice that $K_t(x_v, P_L)$ is non-negative for all x_v and P_L. Note that from Eq. 4, that $K_t(x_v, P_L)P_L$ is the difference between the actual load flow $Q_L(x_v, P_L)$ and that predicted by the no-load flow gain. Eq. 4 means that the valve can be interpreted to comprise an ideal modulated flow source $K_q x_v$, and a shunt (bleed-off) conductance $K_t(x_v, P_L)$. For further details, please see (Li, 2000).

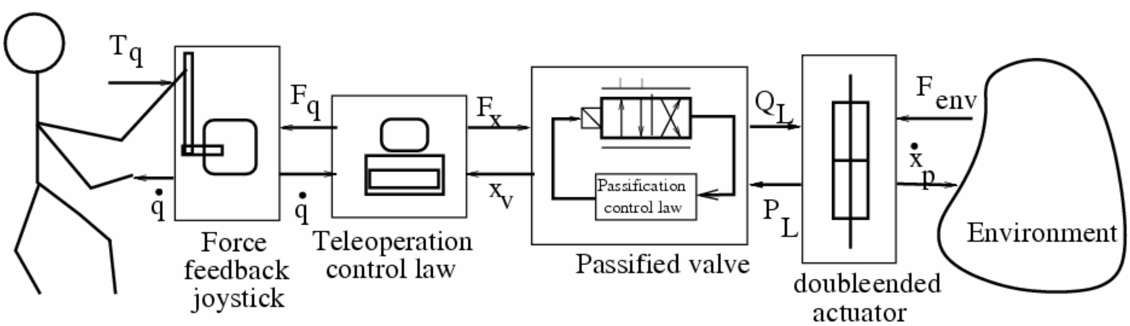


Fig. 2: *The teleoperated system consists of 1) a motorized force feedback joystick, 2) a teleoperation controller, 3) a passified valve with an embedded power supply, 4) a hydraulic actuator.*

For the purpose of the present paper, we can identify from Eq. 3, a load adjusted flow gain $\bar{K}_{q}(\text{sgn}(x_v), P_L)$, namely,

$$\bar{K}_{q}(\text{sgn}(x_v), P_L) = \begin{cases} \dfrac{C_d}{\sqrt{\rho_h}} w \sqrt{P_s - \dfrac{x_v}{|x_v|} P_L}, & \text{sgn}(x_v) P_L < P_s \\ -\dfrac{C_d}{\sqrt{\rho_h}} w \sqrt{-P_s + \dfrac{x_v}{|x_v|} P_L}, & \text{sgn}(x_v) P_L \geq P_s \end{cases} \quad (7)$$

such that

$$Q_L = \bar{K}_q(\text{sgn}(x_v), P_L) x_v \quad (8)$$

Notice that $\bar{K}_q(\text{sgn}(x_v), P_L)$ depends only on $\text{sgn}(x_v)$ but not on x_v itself; and as $\text{sgn}(x_v)P_L$ increases from $0 \rightarrow P_s$, $\bar{K}_q(\text{sgn}(x_v), P_L)$ decreases from $\bar{K}_q > 0$ to 0. Therefore, whenever $|P_L| < P_s$, it is possible to choose an appropriate desired spool position x_v^d to achieve any desired flow rate Q_L^d (less than the saturated flow limit):

$$x_v^d = \frac{Q_L^d}{\bar{K}_q(\text{sgn}(Q_L^d), P_L)} \quad (9)$$

where $\text{sgn}(x_v^d)$ is chosen to be the sign of Q_L^d.

Let the dynamics of the spool be given by:

$$\varepsilon \ddot{x}_v = u \quad (10)$$

where ε is the mass 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 the centring spring forces, as well as the steady state spring and transient damping forces manifested by the steady state and transient flow forces. One way to consider Eq. 10 is that it represents the resulting dynamics when these spool forces have been cancelled out. However, as will be apparent, since the passification control law to be developed relies on terms that are of the same forms as these spring and damping forces, these forces can in fact be considered part of the passification control law, and thus do not need to be cancelled out deliberately. In actual implementation, these forces were not cancelled.

As shown in (Li, 2000), the proportional directional control valve is not passive as a one port device with respect to the hydraulic power $s(P_L, Q_L) := -P_L Q_L$ as the supply rate for any constant $x_v \neq 0$. Intuitively, this is because the valve will be delivering hydraulic energy whenever the load pressure $\text{sgn}(x_v)P_L$ is smaller than the hydraulic supply pressure P_s. It will be shown in section IV, that via control, the valve can be passified to be a passive two port device.

3.2 Hydraulic Actuator Model

The hydraulic actuator is assumed to be a double ended cylinder with annulus area A_p. Further, assuming incompressible fluid flow within the actuator and no leakage between actuator chambers, the actuator kinematic and force balance equations are given by:

$$A_p \dot{x}_p = Q_L; \quad P_L = \frac{1}{A_p} F_{env} \quad (11)$$

where x_p is the actuator displacement, F_{env} is the environment force acting on the piston, Q_L is the flow through the actuator and P_L is the load pressure which is the differential pressure between the actuator chambers. If Q_L is provided by the valve as in Eq. 8, then:

$$\dot{x}_p = \frac{Q_L}{A_p} = \frac{\bar{K}_q(\text{sgn}(x_v), P_L)}{A_p} x_v \quad (12)$$

By taking the product of the load pressure P_L and the load flow rate Q_L given in Eq. 11, it can be seen that the actuator is passive with respect to the supply rate:

$$s_{piston}((F_{env}, \dot{x}_p), (Q_L, P_L)) := -F_{env}\dot{x}_p + P_L Q_L \quad (13)$$

since $s_{piston}((F_{env}, \dot{x}_p), (Q_L, P_L)) \equiv 0$.

3.3 Motorized Joystick Model

The motorized joystick dynamics are given by:

$$M_q \ddot{q} = F_q + T_q \quad (14)$$

where $M_q > 0$ is the joystick inertia, F_q is the control (motor) force, and T_q is the force supplied by the operator. Using the kinetic energy

$$W_{joystick}(q, \dot{q}) = \frac{1}{2} M_q \dot{q}^2$$

as the candidate storage function, and by differentiating it, it is easy to show that the motorized joystick is passive with respect to the supply rate,

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

which is the total power input.

3.4 Control Objectives

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

1. the joystick position and the hydraulic actuator piston position are coordinated. Thus, we want the coordination error $E(t)$ to converge to 0:

$$E(t) := \alpha q(t) - x_p(t) \rightarrow 0 \quad (15)$$

2. the closed loop control system shown in Fig. 1 is passive with respect to the supply rate:

$$s_{total}((T_q, \dot{q}), (F_{env}, \dot{x}_p)) := \rho T_q \dot{q} - F_{env} \dot{x}_p \quad (16)$$

where $T_q \dot{q}$ is the power input by the human operator, $-F_{env}\dot{x}_p$ is the power input by the work environment, and $\rho > 1$ is the power scaling factor.

4 Passification of the Control Valve

As mentioned in Section I, the first step in the control design is to passify the valve. In (Li, 2000), two methodologies are proposed to render 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. One approach is based on structural modification, and the other is based on active feedback control. The following theorem from (Li, 2000) pertains to the latter which is the approach utilized in the present paper. Please refer to (Li 2000) for further interpretations and details.

Theorem 1 (Li, 2000) Let F_x be the auxiliary, exogenous command input to the valve, and let

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

Consider the control law for the spool given by

$$\begin{aligned} u &= -B\dot{x}_v - AP_L - \gamma B x_v \\ &+ \frac{\varepsilon}{B}\left\{\left[\hat{\dot{F}}_x(t) - A\hat{\dot{P}}_L(t)\right] - g_2(t)\,\mathrm{sgn}(z(t))\right\} \\ &+ F_x \end{aligned} \tag{17}$$

where $B > 0$ is a damping coefficient, $A > 0$ is the pressure feedback gain, $\gamma > 0$ is a positive spring rate, and symbol ^ denotes the best estimate of the argument $(\cdot)$

If $g_2(t)$ in Eq.17 is defined so that

$$\begin{aligned} & g_2(t) > \mathrm{sgn}(z(t)) \\ & \left\{\left[\hat{\dot{F}}_x(t) - A\hat{\dot{P}}_L(t)\right] - (\dot{F}_x(t) - A\dot{P}_L(t))\right\} \end{aligned} \tag{18}$$

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

$$s((P_L, F_x), x_v) := \frac{K_q}{A} F_x x_v + [-P_L Q_L] \tag{19}$$

which consists of a ìcommand powerî term and a hydraulic power output term.

Proof: We define the following storage function,

$$W = \frac{K_q}{A}\left[\frac{1}{2}Bx_v^2 + \frac{\varepsilon}{2\gamma}z^2\right]$$

which consists of ìenergiesî associated with the states x_v and z. Differentiating this storage function, we get:

$$\begin{aligned} \dot{W} &\le \frac{K_q}{A}F_x x_v - P_L(K_q x_v) - \frac{K_q B}{A\gamma}z^2 \\ &\le \frac{K_q}{A}F_x x_v - P_L Q_L \end{aligned} \tag{20}$$

where the first inequality is due to the definition of $g_2(t)$, and the second inequality is derived using the fact that,

$$-\frac{K_q B}{A\gamma}z^2 - K_t(x_v, P_L)P_L^2 \le 0$$

as $\frac{K_q B}{A\gamma} > 0$ and so is $K_t(x_v,P_L)$. The passivity property of the valve is obtained by integrating Eq. 20.

Remark 0 *B, A* and γ can be arbitrarily defined as long as they are positive. However, they affect the apparent power scaling of the passified valve between the command port and they hydraulic port (given by K_q / A) as well as the valve bandwidth (see Remark 4).

Remark 1 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. $P_L Q_L$ is the hydraulic power output, and $F_x x_v$ can be considered a fictitious command power. In this case, the 2-port device has associated with it a power scaling factor of K_q / A.

Remark 2 Notice that the spool control force u in Eq.17 contains a spring term $-\gamma B x_v$, and a damping term $-B\dot{x}_v$. Moreover, recall that flow rate is monotonic with respect to spool opening, and steady flow forces are proportional to the flow rate squared, and inversely proportional to the orifice area; transient flow forces are proportional to the rate of change of flow rate. Thus the steady flow force can be modeled as a spring force and the transient flow force (approximately) as a damping force acting on the spool (Strictly speaking, modelling the transient flow force as a damping force is an approximation due to its dependence on the rate of change of pressure. This dependence is however traditionally considered insignificant to the valve dynamics (see Merritt, 1967 p.104 for a discussion)). Hence, the spring forces due to the centering spring and steady flow force can be (approximately) subsumed in the spring term $-\gamma B x_v$. Similarly, the damping forces due to viscous damping and transient flow forces can be subsumed in the damping term $-B\dot{x}_v$ in Eq. 17.

Remark 3 The passification control requires estimates of the time derivatives of the load pressure P_L and of the auxiliary valve command F_x. These can be computed from the pressure chamber dynamics and from the teleoperation control (to be designed). In implementation, these estimates are obtained by direct backward numerical differentiation. The error in this estimate is taken care of, while preserving passivity, by the robustness term ($g_2(t)$ sgn($z(t)$)) in Eq. 17, but at the expense of added dissipation.

Remark 4 When the estimate of the term $\dot{F}_x(t) - A\dot{P}_L(t)$ is accurate, the valve behaviour is given by the transfer function:

$$\frac{x_v(s)}{F_x(s) - AP_L(s)} = \frac{s + B/\varepsilon}{B\left[s(s + B/\varepsilon) + \gamma B/\varepsilon\right]} \tag{21}$$

where s is the Laplace variable. Thus, for low frequency operation, the passified valve dynamics are given by:

$$F_x - AP_L = K_{sp}x_v \tag{22}$$

where $K_{sp} := \gamma B$ is the equivalent spring rate. If the dynamics of the passified valve are designed to be critically damped, the repeated pole will be given by ñ$B/(2\varepsilon)$, so that the bandwidth within which the low frequency approximation is appropriate will be $B/(2\varepsilon)$. The subsequent teleoperation controller design will be based on the low frequency approximation of the passive vale dynamics as given by Eq. 22. Thus, for good performance, the bandwidth of operation should be less than $B\,/\,(2\varepsilon)$ rad/s.

From Eq. 20, we see that the dissipation term is ñ$K_q B/(A\gamma)\cdot z^2$. This is an artifact of the specific passification method in Theorem 1. For steady state operation, $\gamma x_v \approx z$, so this dissipation term is given by $-\frac{K_q}{A} K_{sp} x_v^2$.

Thus, the equivalent spring rate K_{sp} contributes to the dissipation in the passified valve. As will be shown later, this is reflected in the haptics property of the teleoperator.

Remark 5 Although the passification control law Eq. 17 - Eq. 19 is developed based on the assumption that the valve is connected to a flow conserving device, a similar passification control law can be similarly obtained when the valve¥s outlet and return flows are in a constant and known ratio (e.g. when connecting to a single ended cylinder). This can be done by first rederiving the equivalent flow / pressure relation Eq. 3 for the matched symmetric 4-way valve for this case, and by defining the load pressure P_L to be the flow ratio weighed pressure difference between the two ports.

Notice, that the motorized joystick, hydraulic actuator, and the passive valves are all passive two-port devices. The following lemma shows that interconnection of passive two-port subsystems with compatible supply rates result in passive systems.

Lemma 1 Consider the two two-port systems A and B with respective port variables (u^1_A, y^1_A), (u^2_A, y^2_A), and (u^1_B, y^1_B), (u^2_B, y^2_B) (see Fig. 3). Suppose that system A is passive with respect to the supply rate

$$S_A\left(\left(u_A^1, y_A^1\right), \left(u_A^2, y_A^2\right)\right) = u_A^1 \cdot y_A^1 + y_1 u_A^2 \cdot y_A^2$$

and system B is passive with respect to the supply rate

$$S_B\left(\left(u_B^1, y_B^1\right), \left(u_B^2, y_B^2\right)\right) = -u_B^1 \cdot y_B^1 + y_2 u_B^2 \cdot y_B^2$$

where $\gamma_1 > 0$, $\gamma_2 > 0$. Then, the interconnection given by $u^2_A := y^1_B$, $u^1_B := y^2_A$ 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 $\int_0^T S_{AB}(\cdot\,,\cdot), (\cdot\,,\cdot) d\tau$

As an example, this lemma shows that the interconnection of the passified valve (using Theorem 1) and the double ended hydraulic actuator in Fig. 1 is passive with respect to the supply rate:

$$S_{\text{valve/piston}}\left(\left(F_{\text{env}}, \dot{x}_p\right), \left(F_X, x_V\right)\right) = -F_{\text{env}}\dot{x}_p + \frac{K_q}{A} F_X x_V$$

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 $(\dot{q}, F_q)$, (x_v, F_x).

Then the interconnection of the hydraulic actuator Eq. 11, the passive valve, the controller, and the motorized joystick is passive with respect to the supply rate:

$$S_{\text{total}}\left(\left(F_{\text{env}}, \dot{x}_p\right), \left(T_q, \dot{q}\right)\right) := -F_{\text{env}}\dot{x}_p + \rho T_q \dot{q} \quad (23)$$

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

$$S_{\text{controller}}\left(\left(x_V, F_X\right), \left(\dot{q}, F_q\right)\right) = -\frac{K_q}{A} F_X x_V - \rho F_q \dot{q} \quad (24)$$

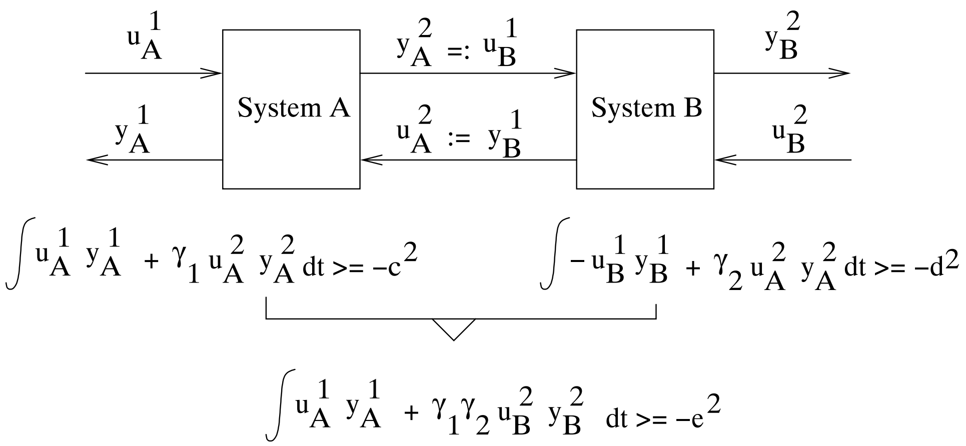


Fig. 3: *Interconnection of two passive two-port systems with compatible supply rates is a passive two-port system*

5 Passive Teleoperator Controller

To preserve passivity, the controller will be designed so that it is passive with respect to the supply rate in Eq. 24 as specified by Corollary 1. In addition, the motions of the joystick and of the hydraulic actuator must also be coordinated so that the coordination error $E := \alpha q - x_p$, where α is the kinematic scaling factor, converges to zero.

The control law will be designed to have good coordination performance for relatively slow manipulations, specifically when the passified valve dynamics can be well approximated by the static relationship Eq. 22:

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

where $K_{sp}x_v = F_x - AP_L$, is determined by the control used in the passification algorithm in Theorem 1. According to Remark 4, when the passive valve dynamics are designed to be critically damped, Eq. 22 will be valid if the frequency of operation is lower than $B/2\varepsilon$. 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.* Similarly, in the event of flow saturation, we expect the coordination performance to degrade, however the passivity property of the teleoperation system will still be retained.

5.1 Ideal Control

If the static valve dynamics is valid, we can manipulate F_x to control x_v via Eq. 22, which in turn controls $\dot{x}_p$ via Eq. 12, in order to make $x_p(t) \to \alpha q(t)$. One such control is:

$$F_x = AP_L + \frac{K_{sp}A_p}{K_q(t)}[\lambda(t)E(t) + \alpha\dot{q}(t)] \tag{25}$$

where $\lambda(t) > 0$ will be determined later. Because $Q_L = A_p\dot{x}_p = \bar{K}_q(t)x_v$ (Eq. 12), this control law will generate $\dot{x}_p = \lambda(t)E + \alpha\dot{q}$, so that

$$\dot{E} = \alpha\dot{q} - \dot{x}_p = -\lambda(t)E \tag{26}$$

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 Eq. 24 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

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

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

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

$$F_q = \frac{K_q}{\rho A}\left[-\frac{K_{sp}A_p}{\bar{K}_q(t)}\alpha x_v - \frac{\alpha A_p}{\bar{K}_q(t)}AP_L\right] \tag{27}$$

The control law to be designed below will generate the joystick control F_q that converges to Eq. 27 when $E \to 0$.

5.2 Dynamic Passive Control

A dynamic control law is now proposed that guarantees that the desired passivity property in Corollary 1 is satisfied. In addition, the control law should generate the valve control Eq. 25 at nearly all times, and the joystick control Eq. 27 after $E \to 0$. The controller contains the dynamics of a fictitious flywheel, with inertia M_f and speed $\dot{f}$ (implemented in software), which is used to store energy temporarily. The control outputs and the flywheel dynamic update law are given by:

$$\begin{pmatrix} \frac{A_p}{K_q}F_q \\ F_x \\ M_f\ddot{f} \end{pmatrix} = \Omega_2(t)\begin{pmatrix} E \\ \dot{q} \\ x_v \\ \dot{f} \end{pmatrix} \tag{28}$$

where

$$\Omega_2(t) = \begin{pmatrix} -\gamma\alpha & 0 & -\frac{\alpha K_{sp}A_p}{\bar{K}_q(t)} & -\frac{\alpha A_p AP_L}{\bar{v}\bar{K}_q(t)} \\ \frac{\lambda(t)K_{sp}A_p}{\bar{K}_q(t)} & \frac{\alpha K_{sp}A_p}{\bar{K}_q(t)} & 0 & \frac{AP_L}{\bar{v}} \\ 0 & \frac{\alpha A_p AP_L}{\bar{v}\bar{K}_q(t)} & -\frac{AP_L}{\bar{v}} & 0 \end{pmatrix} \tag{29}$$

$$\bar{v} = \begin{cases} \dot{f} & \text{if } |\dot{f}| > f_0 \\ \text{sgn}(\dot{f})f_0 & \text{otherwise} \end{cases} \tag{30}$$

$$\lambda(t) = \frac{\gamma\bar{K}_q^2(t)}{K_{sp}A_p^2} \geq 0 \tag{31}$$

and $\gamma > 0$ is a gain. Notice that $\lambda(t)$ will be strictly positive if $|P_L| < P_s$. Moreover, F_x will be exactly Eq. 25 when $\bar{v} = \dot{f}$, and F_q will be exactly Eq. 27 when $E(t) = \dot{E}(t) = 0$ and $\bar{v} = \dot{f}$.

Theorem 2 The controller given in Eq. 28-Eq. 31 is a passive two-port system with respect to the supply rate Eq. 24:

$$s_{controller}((x_v, F_x), (\dot{q}, F_q)) = -\frac{K_q}{A}F_x x_v - \rho F_q\dot{q} \tag{32}$$

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

$$s_{total}((F_{env}, \dot{x}_p), (T_q, \dot{q})) := -F_{env}\dot{x}_p - \rho T_q\dot{q} \tag{33}$$

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

Proof: Consider the storage function

$$W = \frac{1}{2}\gamma E^2 + \frac{1}{2}M_f\dot{f}^2 \tag{34}$$

Then, if we define

$\psi^{\mathrm{T}}(t) = \begin{bmatrix} E & \dot{q} & x_{\mathrm{V}} & \dot{f} \end{bmatrix}$, we have

$$\dot{W} + \frac{A_{\mathrm{P}}}{K_{\mathrm{q}}} F_{\mathrm{q}} \dot{q} + F_{\mathrm{X}} x_{\mathrm{V}} = \psi^{\mathrm{T}}(t) \Omega(t) \psi(t) = 0$$

with the matrix

$$\Omega(t) = \begin{pmatrix} 0 & \gamma\alpha & -\dfrac{\gamma \bar{K}_{\mathrm{q}}}{A_{\mathrm{P}}} & 0 \\ -\gamma\alpha & 0 & -\dfrac{\alpha K_{\mathrm{sp}} A_{\mathrm{P}}}{\bar{K}_{\mathrm{q}}(t)} & -\dfrac{\alpha A_{\mathrm{P}} A P_{\mathrm{L}}}{\bar{v} \bar{K}_{\mathrm{q}}(t)} \\ \dfrac{K_{\mathrm{sp}} A_{\mathrm{P}}}{\bar{K}_{\mathrm{q}}(t)} & \dfrac{\alpha K_{\mathrm{sp}} A_{\mathrm{P}}}{\bar{K}_{\mathrm{q}}(t)} & 0 & \dfrac{A P_{\mathrm{L}}}{\bar{v}} \\ 0 & \dfrac{\alpha A_{\mathrm{P}} A P_{\mathrm{L}}}{\bar{v} \bar{K}_{\mathrm{q}}(t)} & -\dfrac{A P_{\mathrm{L}}}{\bar{v}} & 0 \end{pmatrix} \tag{35}$$

Notice, that the last 3 rows of $\Omega(t)$ are $\Omega_2(t)$ in Eq. 28, and that $\gamma \dot{E} = \Omega_{1*}(t) \Psi(t)$, where $\Omega_{1*}(t)$ denotes the first row of $\Omega(t)$. With the choice of $\lambda(t)$ in Eq. 31, $\Omega(t)$ will be skew symmetric. Hence,

$$W(t) - W(0) = \int_0^t s_{\text{controller}}\left((x_{\mathrm{V}}, F_{\mathrm{X}}), (\dot{q}, F_{\mathrm{q}}) \right) d\tau$$

$$\Rightarrow \int_0^t s_{\text{controller}}\left((x_{\mathrm{V}}, F_{\mathrm{X}}), (\dot{q}, F_{\mathrm{q}}) \right) d\tau \ge -W(0)$$

5.3 Initialization of Flywheel Speed

If the flywheel speed can be guaranteed to be always larger than the design threshold, $\left|\dot{f}(t)\right| \ge f_0, \quad \forall t$, then the F_{x} entry in the controller in Eq. 28 is the same as Eq. 25, which in turn ensures that the coordination error dynamics are given by Eq. 26 which are convergent. For this to be the case, we must be able to initialize $\dot{f}(0)$ such that $\left|\dot{f}(t)\right| \ge f_0, \quad \forall t$. Suppose that at some time t, $\dot{f}(t) > f_0$, by considering the dynamics of the fictitious flywheel as given by the last row of Eq. 28, and utilizing Eq. 30,

$$\begin{aligned} \frac{d}{dt}\left[\frac{1}{2} M_{\mathrm{f}} \dot{f}^2 \right] &= \dot{f} \left(\frac{\alpha A_{\mathrm{P}}}{\bar{v} \bar{K}_{\mathrm{q}}(t)} A P_{\mathrm{L}} \dot{q} - \frac{A P_{\mathrm{L}}}{\bar{v}} x_{\mathrm{V}} \right) \\ &= \frac{A P_{\mathrm{L}}}{\bar{K}_{\mathrm{q}}(t)} (\alpha A_{\mathrm{P}} \dot{q} - \bar{K}_{\mathrm{q}}(t) x_{\mathrm{V}}) = \frac{A P_{\mathrm{L}}}{\bar{K}_{\mathrm{q}}(t)} \dot{E} \end{aligned} \tag{36}$$

The last equality is because $\bar{K}_{\mathrm{q}}(t) x_{\mathrm{V}} = A_{\mathrm{P}} \dot{x}_{\mathrm{P}} = A_{\mathrm{P}} (\alpha \dot{q} - \dot{E})$.

When $\dot{f}(t) > f_0$, the control law Eq. 28 ensures that the coordination error dynamics $\dot{E} = -\lambda(t) E$ where $\lambda(t) \ge 0$. Moreover, $|P_{\mathrm{L}}(t)|$ is strictly smaller than the supply pressure P_{s}, there exists $a > 0$, such that $\bar{K}_{\mathrm{q}}(t) \ge a > 0$. These suggest that the initial flywheel speed $\dot{f}(0)$ should satisfy,

$$\frac{1}{2} M_{\mathrm{f}} \left(\dot{f}^2(0) - f_0^2 \right) > \frac{A P_{\mathrm{L}}}{a} |E(0)| \tag{37}$$

Theorem 3 If the flywheel speed in the controller Eq. 28 is initialized according to Eq. 37, then, the energy in the flywheel will not be lower than f_0. Hence, $\bar{v}(t) = \dot{f}(t)$ in Eq. 30 at all times.

Proof: Suppose that Eq. 37 is true. We shall prove by contradiction. Assume that $t_1 \ge 0$ is the first instance at which $\left|\dot{f}(t)\right| = f_0$. Therefore, on the interval $t \in [0, t_1)$, $\left|\dot{f}(t)\right| > f_0$. Thus, Eq. 36 and Eq. 26 apply for $t \in [0, t_1)$. Equation 36 implies that

$$\dot{f}^2(t) > f_0^2 + \frac{2 A P_{\mathrm{L}}}{a M_{\mathrm{f}}} (E(t) - E(0)) \tag{38}$$

and Eq. 26 implies that $|E(t)$ ñ $E(0)| \le |E(0)|$. Therefore, given Eq. 37, on $t \in [0, t_1]$, $\dot{f}^2(t) > \varepsilon > 0$. Hence, $\dot{f}^2(t_1) \ge \varepsilon > 0$, which is a contradiction.

We summarize the properties of the passive hydraulic teleoperation controller.

Theorem 4 Under the valve passification control Eq. 17 and the teleoperation controller Eq. 28 - Eq. 31, the complete teleoperated hydraulic actuator system consisting of the motorized joystick Eq. 14, the proportional directional control valve Eq. 3 and the hydraulic actuator Eq. 11 is passive with respect to the supply rate given by Eq. 23,

$$s_{\text{total}}((F_{\text{env}}, \dot{x}_{\mathrm{p}}), (T_{\mathrm{q}}, \dot{q})) := -F_{\text{env}} \dot{x}_{\mathrm{p}} + \rho T_{\mathrm{q}} \dot{q} \tag{39}$$

i.e., given initial conditions, there exists $c \in R$, so that for any environment and human input $F_{\text{env}}(\cdot)$ and $T_{\mathrm{q}}(\cdot)$, and for all times $t \ge 0$,

$$\int_0^t [-F_{\text{env}}(\tau) \dot{x}_{\mathrm{p}}(\tau) + \rho T_{\mathrm{q}}(\tau) \dot{q}(\tau)] d\tau \ge -c^2 \tag{40}$$

Furthermore, if

a) the passified valve dynamics are adequately approximated by the low frequency approximant Eq. 22,
b) for all $t \ge 0$, the load pressure $|P_{\mathrm{L}}(t)|$ is strictly less than P_{s} so that there exists $a > 0$, such that $\bar{K}_t(t) \ge a > 0$,
c) the initial state of the fictitious flywheel $\dot{f}(0)$ in Eq. 28 is initialized according to Eq. 37, then,

1. the fictitous flywheel speed $\left|\dot{f}(t \ge 0)\right|$ will always be greater than the threshold f_0;
2. the coordination error $E(t) = \alpha q(t)$ ñ $x_{\mathrm{p}}(t)$ converges exponentially to 0;
3. *as* $t \to \infty$, the haptics property of the joystick is given by Eq. 41:

$$M_{\mathrm{q}} \ddot{q} = -\left(\frac{\alpha K_{\mathrm{q}} K_{\mathrm{sp}} A_{\mathrm{p}}}{A \rho \bar{K}_{\mathrm{q}}(t)} \right) \dot{q} - \left(\frac{\alpha K_{\mathrm{q}}}{\rho \bar{K}_{\mathrm{q}}(t)} \right) F_{\text{env}} + T_{\mathrm{q}} \tag{41}$$

Proof: The passivity property of the teleoperation system is proved in theorem 2. The existence of a positive lower bound for the flywheel speed has been proved in theorem 3.

To prove that the coordination error converges exponentially, differentiate the Lyapunov function $\frac{1}{2} E^2$, followed by substituting Eq. 12 for $\dot{x}_p$, followed by substituting Eq. 22 for x_{v} and finally substituting the valve control input Eq. 28 for F_{x}. This results in the following equation:

$$\frac{d}{dt}\frac{1}{2}E^2 = -\lambda(t)E^2 \qquad (42)$$

Since $\lambda(t)$ has been chosen such that $\lambda(t) \geq \underline{\lambda} > 0$, the exponential convergence property of the coordination error is obtained.

The haptics property of the joystick can be proved by substituting the joystick control Eq. 25 for F_q in the joystick dynamics Eq. 14 and using the fact that $E(t) \rightarrow 0$ as $t \rightarrow \infty$.

Remark: It should be pointed out that the coordination control law Eq. 27 and the passive implementation Eq. 28 - Eq. 31 do not require a force sensor to measure the human force at the force feedback joystick.

5.4 Haptics Property and Design Tradeoff

Consider the haptics property Eq. 41, which describes what the human operator feels when operating the joystick.

Eq. 41 can be interpreted to be a scaled version of 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 (Li and Lee, 2003), and (Lee and Li, 2002) that the teleoperator should appear to be a rigid mechanical tool to the human operator.

For a lossless two port system with power scaling ρ and kinematic scaling α, the expected force scaling factor is α/ρ. The actual force scaling, however, is $\frac{\alpha K_q}{\rho \bar{K}_q(t)} \geq \alpha/\rho$ for $\mathrm{sgn}(x_v)P_L > 0$.

Similarly, the coefficient $\frac{\alpha K_q K_{sp} A_p}{A\rho \bar{K}_q(t)}$ also increases as $\mathrm{sgn}(x_v)P_L$ increases. 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 (because of the shunt conductance $K_t(x_v,P_L)$ in Eq. 4). Because of the imposed intrinsic passivity, these inefficiencies are experienced by the human operator.

The passified valve's equivalent spring rate K_{sp} in Eq. 22 contributes to the damping and sluggishness of the joystick. Thus, as suggested in Remark 4, K_{sp} corresponds to the power loss in the passified valve. Ideally, it should be small to decrease the joystick damping so as to increase the sensitivity of the human operator in sensing the work environment. Unfortunately, this would compromise the bandwidth in which the low frequency approximation is valid. To see this, consider the passified spool dynamics in Eq. 21. Suppose that they are designed to be critically damped (with $\gamma = B/(4\varepsilon)$), then Eq. 21 becomes:

$$\frac{x_v(s)}{F_x(s) - AP_L(s)} = \frac{s + B/\varepsilon}{B(s + B/(2\varepsilon))^2} \qquad (43)$$

Hence, the bandwidth of the valve is given by:

$$B_W := B/(2\varepsilon) \qquad (44)$$

so that $K_{sp} = B_w^2\varepsilon$, where ε is the mass of spool in Eq. 10. Therefore, for the proposed control method, there is a tradeoff between bandwidth B_w and haptics property which is limited by the spool mass ε (smaller the better).

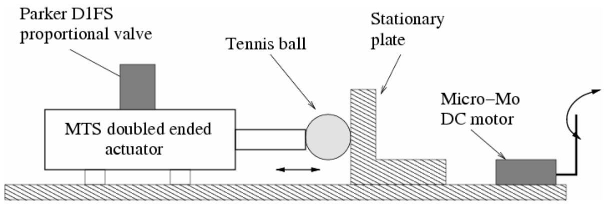


Fig. 4: *Single DOF hydraulic teleoperation setup. The double ended actuator controlled by a proportional valve is interacting with a tennis ball. The joystick is motorized by a small D.C. motor. The schematic of the setup is shown in Fig. 1*

6 Experimental Results

The proposed teleoperation control scheme was implemented experimentally using the setup in Fig. 4. It consists of a Parker-Hannifin D1FS direct acting proportional control valve and a MTS double ended actuator (with an annulus area of approximately $6.5\cdot 10^{-4}$m^2). The proportional control valve provides the spool displacement signal. The valve outlets are instrumented with pressure sensors. The double ended hydraulic actuator is instrumented with an LVDT (Linear Variable Displacement Transducer). A pressure compensated hydraulic power supply with a maximum pressure of 10.3 MPa, and a maximum flow of 37.3 LPM is used in the experiment. The joystick is actuated by a MicroMo DC motor (Max torque 0.53 Nm with a 25:1 gear head) which is instrumented with an encoder. However, since the applied human force is not needed for the control law, a force sensor is not necessary at the motorized joystick.

In the first set of experiments, the hydraulic actuator is unconstrained (i.e. $F_{env} = 0$). Figure 5 shows the response of the joystick and the actuator when only the joystick is manipulated by the human operator. The parameters used are $B/\varepsilon = 30$ rad/s, $\alpha = 3$, $\rho = 40000$. The choice of ρ determines the sensitivity of the system (the smaller the more sensitive). The choice depends on the intended application and the hardware with regard to the force level that the hydraulic machine is expected to encounter, and the allowable force that the force feedback joystick can provide as dictated by the hardware and safety. In our experiments, the maximum coordination error is 1.2 mm. Notice that the maximum error occurs when the joystick is nearly stationary. This is because of the significant deadband that exists in the proportional valve (see plot of spool displacement in Fig. 5).

In the second set of experiments, the hydraulic actuator pushes a tennis ball against a steel block. In this case, we set $\rho = 900$, $\alpha = 0.15$ to emphasize the force reflection property of the control scheme. In this scenario, the human operator first manipulates the actuator towards the tennis ball, and contact is made at about 2 sec. The operator then repeatedly pushes on the ball and releases the joystick. Observe, from Fig. 6-7, that throughout the experiment, the coordination error is also within 1.1 mm. Note that as the contact force (F_{env}) increases, the spool displacement is decreased due to the pressure feedback in the valve passification control. Notice particularly that after the human has released the joystick, as the tennis bounces back and pushes the actuator backward, the joystick also springs back.

Estimate of $\frac{\varepsilon}{B} \cdot \frac{d}{dt}(F_x - AP_L)$ is used in the passification control Eq. 17 of the valve. A bound on the estimation error is used in the robustness term. Figure 6-7 show the signal

F_x - AP_L and the error in predicting $\frac{d}{dt}(F_x - AP_L)$.

Since ε/B = 0.049s, the robustness term in Eq. 17 is only 5-8 % of the signal F_x - AP_L.

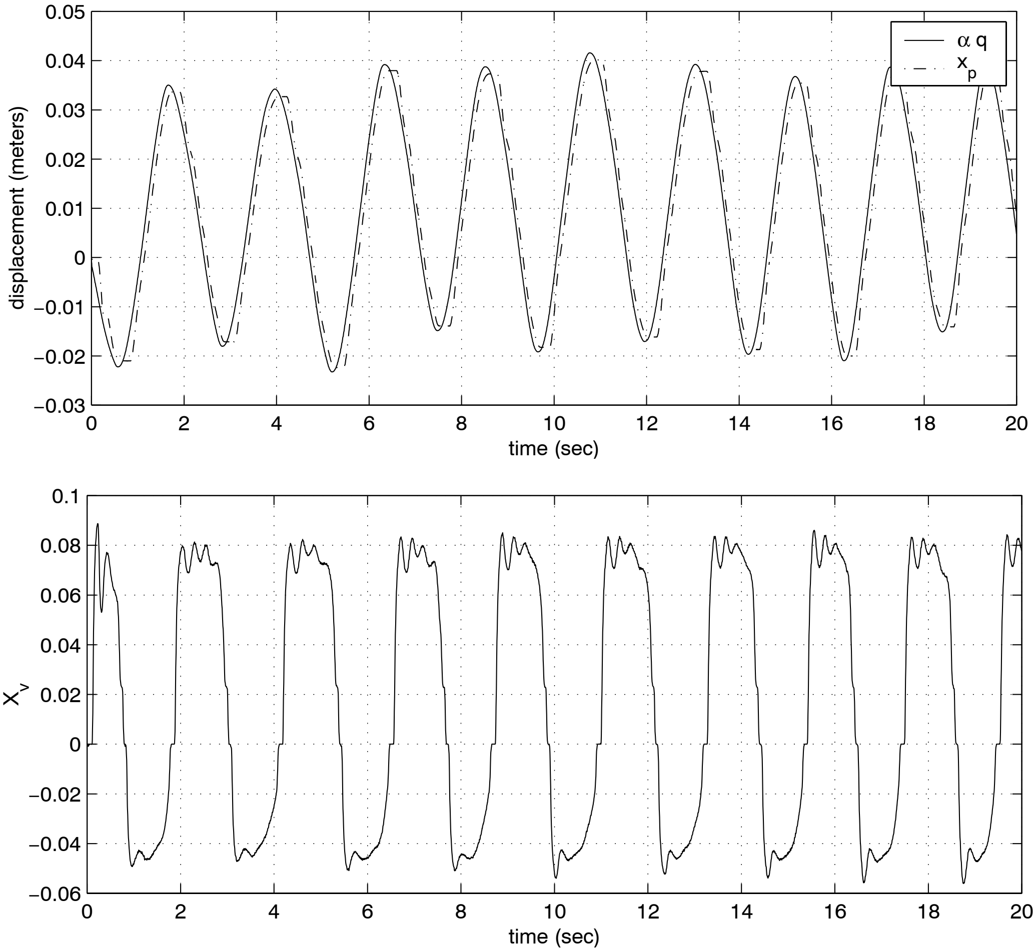
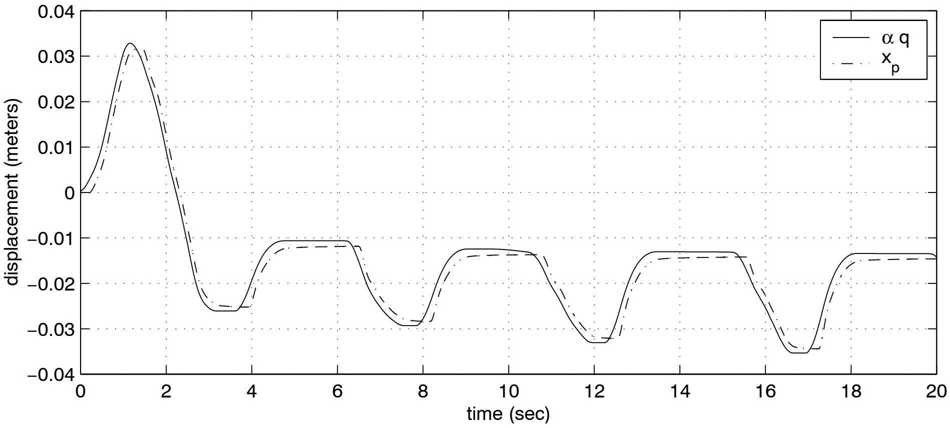
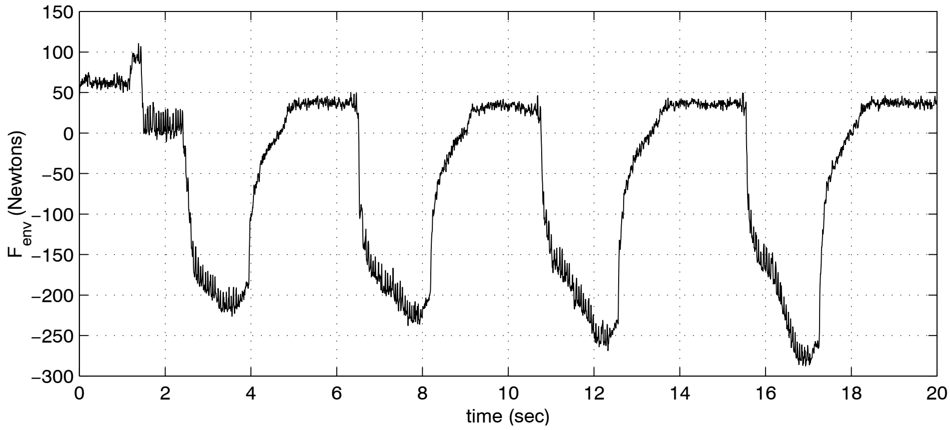


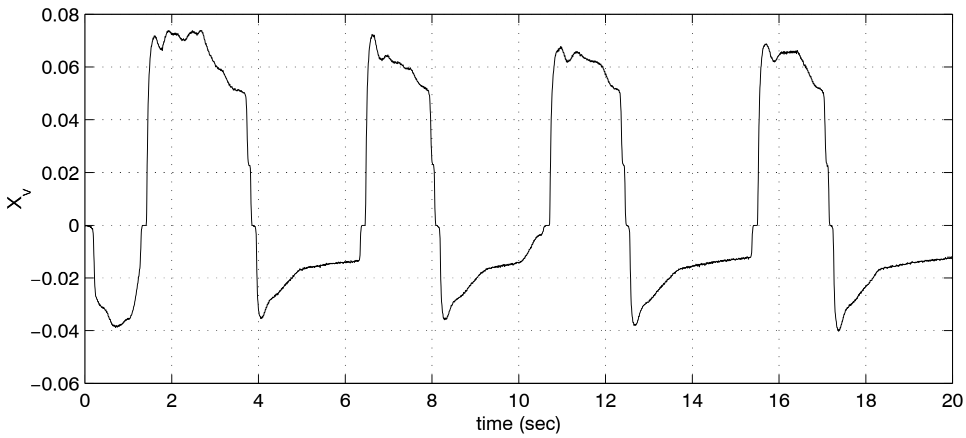
Fig. 5: *Unconstrained joystick manipulation experiment. Top: displacements of scaled joystick and actuator displacements; Bottom: spool displacement normalized to maximum spool LVDT range (unitless)*



a) Displacements of hydraulic actuator (dotted) and α-scaled joystick displacement (solid)

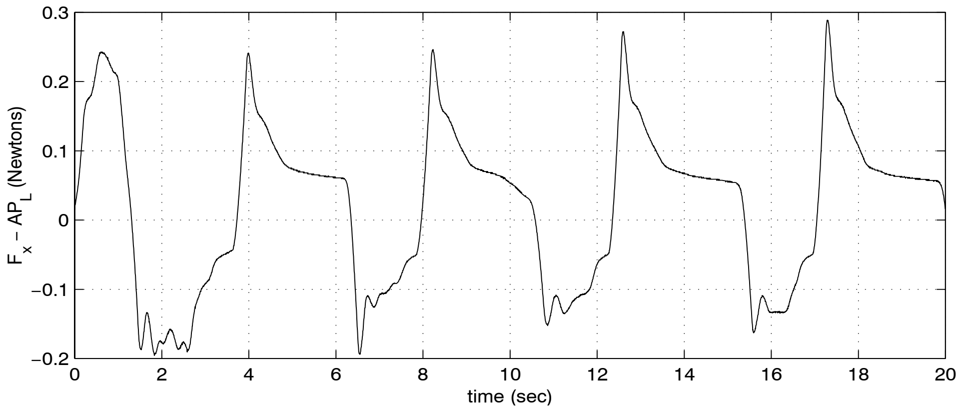


b) Environment force ($-F_{env}$) acting on the hydraulic actuator (estimated from pressure sensors)

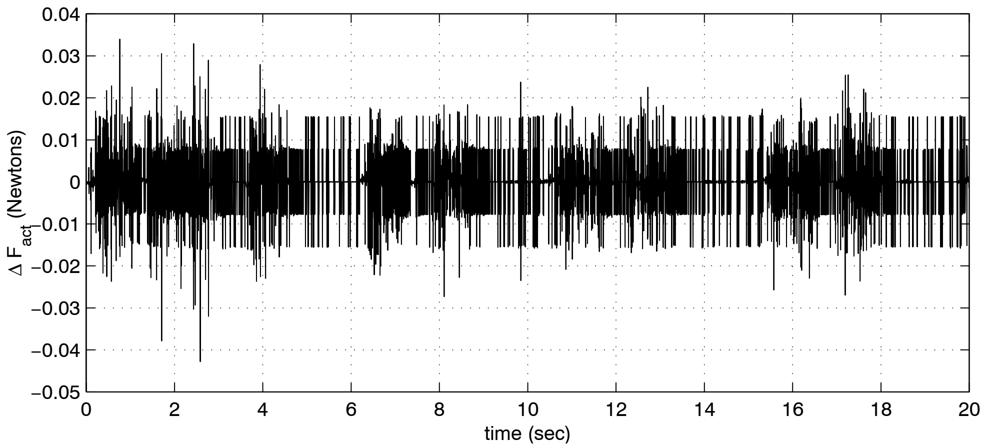


c) Spool displacement x_v normalized to maximum spool LVDT range (unitless).

Fig. 6: *Experimental results when actuator interacts with a tennis ball*



a) Total valve command: F_x - AP_L



b) Robust passivity term is given by the error in estimating $\frac{\varepsilon}{B}\frac{d}{dt}[F_x - AP_L]$

Fig. 7: *Control signals when the actuator is interacting with a tennis ball*

7 Conclusions

A passive bilateral teleoperation control scheme has been proposed for an electrohydraulic actuator and a motorized joystick. The developed passive control ensures that the teleoperated machine behaves like a passive mechanical tool. Under sufficiently low manipulation bandwidth, coordination of the master joystick and the slave actuator is achieved. The efficacy of the control scheme has been experimentally validated. Compared to other control schemes such as impedance control, by ensuring passivity, the coupling stability of the hydraulic teleoperator with a broad class of passive objects can be guaranteed. Moreover, the control scheme does not require knowledge of the impedances of the environment with which it is in contact.

Two areas of performance improvements are desirable. Firstly, the coordination performance can be improved if the deadband of the valve can be taken into account, without violating passivity. Secondly, the current control law presents a tradeoff between bandwidth (and hence coordination performance) and haptics property (in terms of large damping effect). This conflict is an artifact of the combination of valve passification and control methodologies. If alternate passification strategies (Li and Ngwompo, 2002) are adopted or a control scheme that takes into account the valve dynamics is developed, this conflict may be overcome.

In this paper, the effect of fluid compressibility is neglected. In the presence of fluid compressibility, the outlet and return flows of the valve will not be exactly the same even when a double ended actuator is used, so that the concept of a single load flow Q_L is not strictly valid. However, to the extent that the single load flow assumption is valid, the passivity property of the system will still be valid if the pressure feedbacks are obtained close to the valve ports. On the other hand, coordination performance will be worse when compressibility effects are significant (high pressure, large fluid capacitance). In our more recent and current research, valve passification as well as passive teleoperation algorithms have been developed that enable us to allow for different valve outlet and return flows, valve dynamics and fluid compressibility. However, by demanding better coordination control, these control laws are more complex, and all require the use of a force sensor at the joystick to measure the applied human force. The relatively simple control proposed in the present paper on the other hand, does not require the use of force sensor on the joystick.

Nomenclature

A_p	Actuator cross-sectional area
A	Area for pressure feedback
c	A scalar
B	Damping coefficient
B_w	Passified valve bandwidth
C_d	Discharge coefficient
E	Coordination error
F_x, F_{act}	Spool stroking forces
F_q	Control input force
F_f	Fictitious flywheel actuation force
F_{env}	Environment force
$\dot{f}$	Fictitious flywheel speed
f_0	Fictitious flywheel speed threshold
M_f	Fictitious flywheel inertia
P_s	Supply pressure
P_L	Difference in pressure between the two actuator chambers
q	Joystick position
$Q_L(x_v, P_L)$, Q_L	Flow rate out of / into the valve
Q_L^d	Desired flow rate
K_q	Flow rate gain
$\overline{K}_q(x_v, P_L)$	Nonlinear load adjusted flow gain
K_{sp}	Equivalent spring stiffness of passified valve
$K_t(x_v, P_L)$	Nonlinear equivalent shunt conductance
M	Joystick inertia
$s(\cdot,\cdot)$	Various supply rates
T_q	Human input force
u	Spool control force
w	Valve area gradient
W	Various storage functions
x_v	Spool displacement
x_v^d	Desired spool position
x_p	Actuator piston position
α	Kinematic scaling
ε	Spool inertia
γ	Positive scalar
ρ	Power scaling
ρ_h	Hydraulic fluid density
$\lambda(t)$	Coordination error convergence rate

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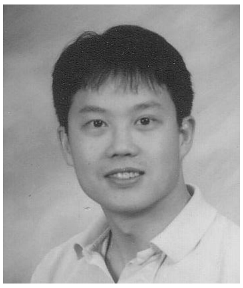

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