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PASSIVE CONTROL OF FLUID POWERED HUMAN POWER AMPLIFIERS

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Abstract

A unified control framework is proposed for the control of fluid powered human power amplifiers. Human power amplifiers are mechanical tools that humans operate directly and the human force is amplified hydraulically or pneumatically. The tool and the environment forces would therefore appear to the human as having been scaled down. Ideally, the assistive force is transparent to the human such that the tool should feel haptically like any other passive mechanical tool to the human. To ensure safety and coupling stability, it is helpful that the controlled human power amplifier interacts with the human opeartor and other environments passively. In the proposed unified control framework, the hydraulic or pneumatic actuator is controlled to interact virtually with a fictitious inertia. The control problem then becomes one of coordinating the velocities between virtual inertia and the actuator. Both processes of augmentation with virtual inertia, and coordination can be done passively; the latter via the energy preserving passive decomposition approach into locked and shaped systems. The overall control system can be shown to possess the needed passivity property. The control schemes for both the hydraulic and pneumatic cases have been experimentally implemented.

Keywords:
Passivity, coupling stability, coordination, exoskeleton, virtual system.

INTRODUCTION

Human amplifiers, extenders or exoskeleton are tools that humans manipulate directly, but have the ability to attenuate or amplify the apparent power that the human exerts. By being physically connected to the task, the control is more intuitive for the human operator. In particular, the operator controls and is informed by the machine via physical quantities like power, forces and displacements, as in the use of common mechanical tools such as hammer, scissors etc. Hydraulics and pneumatics actuations have better power densities than electromagnetic actuation. However, we must consider the fluid compressibility and the fact that a hydraulic actuator is mainly a velocity source whereas an electromechanical actuator is typically treated as a torque/force source.

The control objective is to amplify the power that the human exerts on the machine. Since human power amplifiers interact physically with the work environment and the human, the interactive stability with a variety such environments are needed to ensure safety. For this reason, an additional requirement is that the human power amplifer can interact with the human operator at the handle as well as with its physical environment in an energetic passive manner. Since most physical environments and human dynamics are energetically passive or can be shown to behave nearly energetically passive, ensuring that the human power amplifer behave passively can ensure the system can operate stably in a many situations. Even with active environments, infinite net external energy input is needed to make an energetically passive human power amplifier unstable.

The human power amplifier control problem can be posed simplistically as a force tracking problem [1] - i.e. the actuator force is controlled to be a scaled copy of the applied human force. However, this requires the use of velocity positive feedback which has a destabilizing effect in the presence of uncertainty, slow sampling or time delay. Without the velocity positive feedback, force tracking performance degrades especially during free-space operation. Passivity property cannot be easily ensured structurally.

An alternate passive control structure was proposed in [2] for hydraulically actuated human power amplifiers. Instead of using a force tracking paradiagm, the control problem becomes one of coordinating the velocity of the system with that of a fictitious passive mechanical system that is coupled to the human power amplifier via the measured force and the valve command input. In essence, the velocity feedforward term is now generated by the velocity of the fictitious mechanical system. The control law for achieving coordination is accomplished via a passive decomposition [3], [4], [5] into a shape

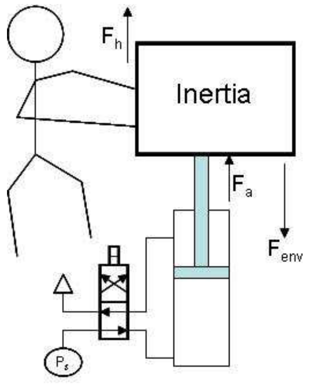

Figure 1. 1 DOF fluid powered human power amplifier setup

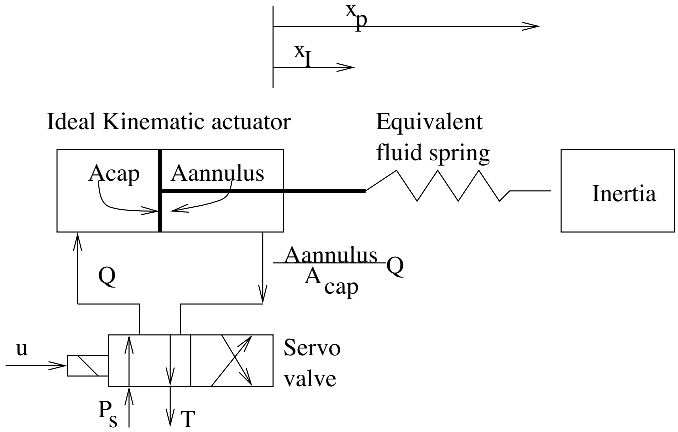

Figure 2. Each hydraulic actuator is modeled as an ideal kinematic actuator and an equivalent fluid spring.

and a locked system, from which a plethora of control laws can be designed to stabilize the shape system. A significant advantage of the control is that the passivity property is enforced by the control structure itself, and is therfore more robust. In this paper, we show that this control structure can be generalized for pneumatic systems as well.

PROBLEM FORMULATION

For simplicity, we consider only one degree of freedom motion. Extension to fully coupled multi-DOFs dynamics can be done but will detract from the key concept. For each degree of freedom, the generalized inertia $M_p > 0$ is acted on by the generalized human input F_{human} and environment force F_{env}, as well as the hydraulic / pneumatic actuator force F_a:

$$M_p \ddot{x}_p = F_{human} + F_{env} + F_a \tag{1}$$

We assume that the displacement of the human amplifier x_p is measured; and both the applied actuator force F_a and the applied human force F_{human} are measured by force sensors. The environment force F_{env} which includes interaction force with all environment other than the human operator as well as friction, is not measured. The actuator force is generated by a hydraulic/pneumatic actuator (or motor), controlled by a four-way proportional valve with a constant pressure supply. The valve bandwidth is sufficiently high so that the valve command corresponds statically to spool displacement. The models for the actuator force F_a will be considered in the next section.

It is desired that the closed loop system would enable the human operator and the work environment to interact via a rigid mechanical tool except that the human would feel as if he/she is $\rho + 1$ times stronger. Therefore, the target dynamics would be

$$M_L \ddot{x}_p = (\rho + 1) F_{human} + F_{env} \tag{2}$$

where M_L is the inertia of the rigid mechanical tool, which in the proposed control structure will be slightly different from M_p in (1).

The human power amplifier interacts with both the human operator and the work environment via the effort and flow variables of $(F_{human}, \dot{x}_p)$ and $(F_{env}, \dot{x}_p)$. To ensure safety, it is also desired that this interaction satisfies an energetic passivity with power scaling condition. Let the supply rate be the sum of the mechanical power input from the environment and the scaled mechanical power input by the human:

$$P_{total}(F_{human}, F_{env}, \dot{x}_p) := \{(1+\rho) F_{human} + F_{env}\} \dot{x}_p \tag{3}$$

The desired passivity condition is that there exists $c^2 \geq 0$ such that for any human and environment input F_{human} and F_{env} and for any $t \geq 0$,

$$\int_0^t P_{total}((F_{human}, \dot{x}_p), (F_{env}, \dot{x}_p)) d\tau \geq -c^2 \tag{4}$$

This condition implies that the net amount of scaled energy that can be extracted from the system by the work environment and the human are limited by c^2 and $c^2/(1+\rho)$ respectively.

MODELING OF ACTUATOR FORCE

We now model a fluid power actuator with capside and rodside areas of A_1, and A_2, area ratio of $r := A_1/A_2$. The actuator is controlled by a matched, symmetric and critically lapped four-way valve with fast spool dynamics so that the command input corresponds to the valve area opening. We consider the hydraulic and the pneumatic cases separately.

Hydraulic case

A useful way to account for the fluid compressibility in a hydraulic actuator is to model the actuator as consisting of an ideal kinematic actuator (with displacement x_I) interacting with the system inertia (with displacement x_p) via an equivalent spring with a compression (Figure 2):

$$\Delta = x_I - x_p \tag{5}$$

The position dependent spring force $F_a(x_I, \Delta)$ encompasses the compressibility of the fluid in the actuator and the fluid line, as well as other mechanical compressibility. Inclusion of the compressibility effect is essential for modeling the force exerted by the hydraulic actuator.

Let V_1 and V_2 be the chamber volumes that include dead volumes,

$$V_1 = V_{10} + A_1 x_I = A_1(L_{10} + x_I)$$
$$V_2 = V_{20} - A_2 x_I = A_2(L_{20} - x_I)$$

Using basic bulk modulus equation, we have:

$$F_a(x_I, \Delta) = \beta\left(\frac{A_1^2}{V_1} + \frac{A_2^2}{V_2}\right)\Delta \tag{6}$$
$$= \beta\left(\frac{A_1}{L_{10} + x_I} + \frac{A_2}{L_{20} - x_I}\right)\Delta \tag{7}$$
$$= K(x_I)\Delta \tag{8}$$

where $K(x_I)$ is the position dependent linear spring stiffness.

The ideal actuator speed $\dot{x}_I$ would be the acuator speed when compressibility is neglected. For a four-way proportional control valve, it is related to the command u as follows:

$$\dot{x}_I = \gamma_q(\text{sgn}(u), F_a)u \tag{9}$$

$$\gamma_q(\text{sgn}(u), F_a) = \begin{cases} \frac{w}{A_1}\left(\frac{r^3}{r^3+1}\right)^{\frac{1}{2}}\sqrt{P_s - F_a/A_1}, & u \geq 0 \\ \frac{w}{A_1}\left(\frac{r^2}{r^3+1}\right)^{\frac{1}{2}}\sqrt{P_s + rF_a/A_1}, & u < 0 \end{cases} \tag{10}$$

where $\gamma_q(\text{sgn}(u), F_a)$ is the loaded velocity gain, P_s is the supply pressure, and w is the area gradient.

Pneumatic case

In the pneumatic case, the definition and dynamics of x_I and of the equivalent spring are not as obvious. Suppose that the air mass and volume (including dead volume) in each actuator chamber are m_1, V_1 and m_2, V_2. Since

$$V_1 = V_{10} + A_1 x_p = A_1(L_{10} + x_p)$$
$$V_2 = V_{20} - A_2 x_p = A_2(L_{20} - x_p)$$

The force exerted by the actuator can be rewritten as,

$$F_a = P_1 A_1 - P_2 A_2 - P_{atm}(A_1 - A_2)$$
$$= RT\left[\frac{m_1 A_1}{V_1} - \frac{m_2 A_2}{V_2}\right] - P_{atm}(A_1 - A_2)$$
$$= RT\left[\frac{m_1}{L_{10} + x_p} - \frac{m_2}{L_{20} - x_p}\right] - P_{atm}(A_1 - A_2) \tag{11}$$

The last term models the effect of the atmospheric pressure acting on piston rod area. Define x_I' to be the actuator position when $F_a = -P_{atm}(A_1 - A_2)$. Hence,

$$x_I' = \frac{m_1}{m_1 + m_2}L_o - L_{10} \tag{12}$$
$$\Delta = x_I - x_p \tag{13}$$

where $L_o = L_{10} + L_{20}$. Let $\mathbf{m} = (m_1, m_2)$. The force exerted by the actuator can then be written as

$$F_a = \bar{K}(\mathbf{m}, x_p)(x_I' - x_p) - P_{atm}(A_1 - A_2) \tag{14}$$
$$\bar{K}(\mathbf{m}, x_p) := \frac{RT(m_1 + m_2)}{(L_{10} + x_p)(L_{20} - x_p)} \tag{15}$$

We now define

$$x_I := x_I'(\mathbf{m}) - \frac{P_{atm}(A_1 - A_2)}{\bar{K}(\mathbf{m}, x_p)} \tag{16}$$
$$\Delta := x_I - x_p \tag{17}$$

We have

$$F_a(\mathbf{m}, x_p) = \bar{K}(\mathbf{m}, x_p)\Delta \tag{18}$$

This defines a nonlinear and air mass dependent spring element.

For moderate actuator force, $x_I' - x_p$ is small compared to $L_{10} + x_I'$ and $L_{20} - x_I'$. Then

$$K(\mathbf{m}) := \bar{K}(\mathbf{m}, x_I') \approx \bar{K}(\mathbf{m}, x_p) \tag{19}$$

Furthermore,if $L_{10} \approx L_{20}$, the error in the approximation is only second order in the compression. Eq.(19) is advantageous because, in the isothermal situation, x_p enters only through Δ but not the stiffness term, and x_I is a function of $\mathbf{m}$ only but not of x_p, similar to the hydraulics case. This assumption is made hence forth so that:

$$F_a(\mathbf{m}, \Delta) := K(\mathbf{m})\Delta \tag{20}$$
$$\Delta := x_I(\mathbf{m}) - x_p \tag{21}$$

which is a linear spring with stiffness being dependent on the amount of gas in the actuator chambers.

Mass flow rate through the valve to either chamber can be obtained by modeling it as flow through converging-diverging nozzle. Although highly nonlinear with respect to upstream and downstream pressures, they are linear with respect to valve opening and hence command u. Hence, $\dot{x}_I$ and $\dot{\mathbf{m}}$ are of the form:

$$\dot{x}_I = \gamma_I(\text{sgn}(u), \mathbf{m}, F_a)u \tag{22}$$
$$\dot{\mathbf{m}} = \gamma_q(\text{sgn}(u), \mathbf{m}, F_a)u \tag{23}$$

Passivity property of hydraulic / pneumatic actuator

Consider the storage function for the hydraulic actuator,

$$W_{hyd} = \int_0^{\Delta} F_a(x_I, \sigma)d\sigma = \frac{1}{2}K(x_I)\Delta^2 \tag{24}$$

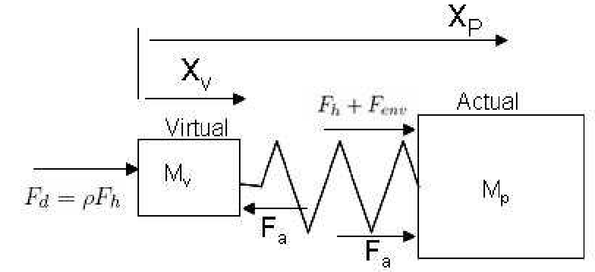

Figure 3. Virtual coordination control strategy

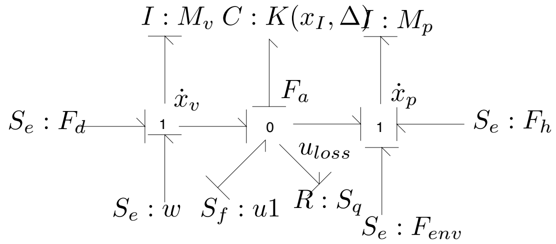

Figure 4. Bond graph of hydraulic human power amplifier with a virtual mechanical system. u_{loss} and w are dissipative term and extra control signal that are not considered in this paper.

Then,

$$\dot{W}_{hyd} = F_a(x_I, \Delta)(\dot{x}_I - \dot{x}_p) + \frac{1}{2}\frac{\partial K}{\partial x_I}\Delta^2 \dot{x}_I \quad (25)$$

$$= F_a(x_I, \Delta)\left\{-\dot{x}_p + \left[1 + \frac{\Delta}{2K(x_I, \Delta)}\frac{\partial K}{\partial x_I}\right]\dot{x}_I\right\} \quad (26)$$

$$= F_a(x_I, \Delta)\left\{-\dot{x}_p + \gamma_1(x_I, F_a, sgn(u))u\right\} \quad (27)$$

where

$$\gamma_1(x_I, F_a) = \left[1 + \frac{\Delta}{2K(x_I, \Delta)}\frac{\partial K}{\partial x_I}\right]\gamma_q(\text{sgn}(u), F_a)$$

Similar result is obtained for the pneumatic actuator with the storage function given by:

$$W_{pneu} = \int_0^\Delta F_a(\mathbf{m}, \sigma)d\sigma = \frac{1}{2}K(\mathbf{m})\Delta^2 \quad (28)$$

Then,

$$\dot{W}_{pneu} = F_a(\mathbf{m}, \Delta)(\dot{x}_I - \dot{x}_p) + \frac{1}{2}\frac{\partial K}{\partial \mathbf{m}}\Delta^2 \dot{\mathbf{m}} \quad (29)$$

$$= F_a(\mathbf{m}, \Delta)(-\dot{x}_p + \left[\frac{\partial x_I}{\partial \mathbf{m}} + \frac{\Delta}{2K(\mathbf{m}, \Delta)}\frac{\partial K}{\partial \mathbf{m}}\right]\dot{\mathbf{m}}) \quad (30)$$

$$= F_a(\mathbf{m}, \Delta)(-\dot{x}_p + \gamma_1(\mathbf{m}, F_a, \text{sgn}(u))u) \quad (31)$$

where

$$\gamma_1(\mathbf{m}, F_a, \text{sgn}(u)) = \left[\frac{\partial x_I}{\partial \mathbf{m}} + \frac{\Delta}{2K(\mathbf{m}, \Delta)}\frac{\partial K}{\partial \mathbf{m}}\right]\gamma_q(\text{sgn}(u), \mathbf{m}, F_a).$$

For either the hydraulic / pneumatic case, the compressibility is shown to be a passive 2-port capacitive field (like a spring), with a mechanical port via $(F_a, \dot{x}_p)$ that interacts with system inertia M_p in Eq.(1) and the valve command port via $(F_a, -\gamma_1 u)$.

VIRTUAL COORDINATION CONTROL FORMULATION

With the actuator compressibility shown to be a spring like object with one end interacting with inertia of the machine, the approach is to so control the valve so that the other end of the spring is interacting with a small virtual mass which is acted upon by the desired force $F_d = \rho F_h$ (Fig. 3). If the virtual mass (M_v) and the system inertia M_p are exactly coordinated, M_p and M_v become a single rigid inertia which is acted upon by the desired actuator force F_d and the human and environment forces F_h and F_{env}. We illustrate this process for the hydraulics case only. The pneumatic case can be obtained by substituting x_I by $\mathbf{m}$ for the most parts.

Consider the dynamics of a virtual mass and its coupling to the fluid power actuator given by:

$$\begin{aligned} M_v\ddot{x}_v &= F_d - F_a(x_I, \Delta) \\ u &= \frac{1}{\gamma_1}x_v + \frac{1}{\gamma_q}u_1 \end{aligned} \quad (32)$$

where $F_d(t)$ is the desired actuator force typically given by $F_d(t) := \rho F_h(t)$. Notice that the coupling between the virtual inertia and the human power amplifier is similar to an integral controller ($u_1 = 0$). u_1 is the control term that will cause coordination $\dot{x}_v \to \dot{x}_p$.

The coupled system is given by:

$$\begin{aligned} M_p\ddot{x}_p &= F_{human} + F_{env} + F_a(x_I, \Delta) \\ M_v\ddot{x}_v &= F_d - F_a(x_I, \Delta) \\ \dot{\Delta} &= \dot{x}_I - \dot{x}_p \\ \dot{x}_I &= \frac{\gamma_q}{\gamma_1}\dot{x}_v + u_1 \end{aligned} \quad (33)$$

which has a bond graph representation shown in Fig. 4.

Using the storage function,

$$W_{total} = W_{hyd} + \frac{1}{2}M_p\dot{x}_p^2 + \frac{1}{2}M_v\dot{x}_v^2 \quad (34)$$

where W_{hyd} is the storage function for the capacitive field in (24). From this, it can be shown that the coupled system (33) satisfies the passivity property:

$$\begin{aligned} &\int_0^t [F_d\dot{x}_v + (F_h + F_{env})\dot{x}_p]d\tau \\ &+ \int_0^t [\frac{\gamma_1}{\gamma_q}F_a u_1 - F_d v_E]d\tau \geq -c_0^2 \end{aligned} \quad (35)$$

for some c_0 and for all $t \geq 0$. Thus, if the control terms u_1 are designed such that for some c_1, and for all $t \geq 0$,

$$\int_0^t [\frac{\gamma_1}{\gamma_q}F_a u_1 - F_d(t)v_E]d\tau \leq c_1^2 \quad (36)$$

where $v_E = \dot{x}_v - \dot{x}_p$, then, for $F_d(t) = \rho F_h(t)$, the human power amplifier under control will satisfy the desired passivity property in (4).

Coordinate transformation into locked and shape systems

To see the effect of the control u_1 on the dynamically coupled system and the coordination error, consider the coordinate transformation:

$$\begin{pmatrix} v_L \\ v_E \end{pmatrix} = \begin{pmatrix} M_p/M_L & M_v/M_L \\ 1 & -1 \end{pmatrix} \begin{pmatrix} \dot{x}_p \\ \dot{x}_v \end{pmatrix} \quad (37)$$

$$M_L := M_v + M_p \quad (38)$$

$$M_E := \left(\frac{1}{M_p} + \frac{1}{M_v} \right)^{-1} = \frac{M_p M_v}{M_p + M_v} \quad (39)$$

(v_L, v_E) are referred to as the locked and shape system velocities, since v_E measures the difference between the speeds of the virtual and actuator manipulator, and $v_L = \dot{x}_v = \dot{x}_p$ when $v_E = 0$. Hence, v_L is the speed of the manipulator and the virtual manipulator when they are locked in place. M_L and M_E are respectively the locked system inertia, and the shape system inertia.

Eqs. (37)-(39) is an instance of decomposing the velocity space of the coupled system (i.e. $\Re^2 = \{(\dot{x}_p, \dot{x}_v)\}$) into a component given by the coordination error $v_E = \dot{x}_p - \dot{x}_v$ and its orthogonal complement v_L with respect to $\mathrm{diag}(M_p, M_v)$ being the metric. This decomposition approach can be generalized to nonlinear multi-DOF mechanical systems as well [3], [4], [5].

The dynamics in the transformed coordinates are given by:

$$M_L \dot{v}_L = F_h + F_{env} + F_d \quad (40)$$

$$M_E \dot{v}_E = \underbrace{\frac{M_v}{M_L}(F_h + F_{env}) - \frac{M_p}{M_L} F_d(t)}_{F_E} + F_s(\Delta) \quad (41)$$

$$\dot{\Delta} = -v_E + u_2 \quad (42)$$

We can associate storage functions for the locked and shape systems respectively as,

$$W_L = \frac{1}{2} M_L v_L^2; \quad W_S = \frac{1}{2} M_E v_E^2 + \int_0^{\Delta} F_s(\delta) d\delta \quad (43)$$

Notice that the coordinate transformation Eqs.(37)-(39) preserves kinetic energies and storage functions in that the sum of the kinetic energies and storage functions, respectively, of the locked and shape systems are exactly those of the coupled system [6]

$$\kappa := \frac{1}{2} M_p \dot{x}_p^2 + \frac{1}{2} M_v \dot{x}_v^2 = \frac{1}{2} M_L v_L^2 + \frac{1}{2} M_E v_E^2 \quad (44)$$

$$W_c := W_L + W_S \quad (45)$$

This means that by ensuring that the shape and locked systems are passive, the coupled system in Eq.(33) will also be passive.

Shape system control

The shape system control objective is to make $v_E \to 0$ in:

$$M_E \dot{v}_E = F_E(t) + F_a(x_I, \Delta) \quad (46)$$

$$\dot{\Delta} = \dot{x}_I - \dot{x}_p \quad (47)$$

$$\dot{x}_I = \frac{\gamma_q}{\gamma_1} \dot{x}_v + u_1 \quad (48)$$

A variety of control algorithms can be designed for u_1 to ensure that $v_E \to 0$ exponentially. A backstepping control law is used to obtain the experimental results in this paper.

Locked system control

If $F_d := \rho F_h$, the locked system dynamics becomes

$$M_L \dot{v}_L = \underbrace{(\rho + 1) F_h + F_{env}}_{F_{total}} \quad (49)$$

which is the desired dynamics of a passive rigid mechanical tool.

When $v_E \to 0$, we also have the RHS in Eq.(41) being zero on average:

$$0 = F_E + F_s(\Delta)$$

$$0 = \frac{M_v}{M_L} F_{total} + F_a(x_I, \Delta) - F_d(t)$$

Since F_{total} is the total forcing term for the locked system (49) scaled by M_v/M_L (which is small if M_v is small), thus unless the locked system undergoes large acceleration, F_{total} is typically small. Therefore, we have the original force control objective being approximately satisfied:

$$F_a(x_I, \Delta) \approx F_d(t)$$

This is despite the specified objective (because of the dynamics of the added M_v inertia) being a coordination control problem.

EXPERIMENTAL RESULTS

Hydraulic case

The hydraulic human power amplifier has two DOF. The pitch motion is actuated by a linear actuator, and the reach motion actuated by a hydraulic motor and a rack-and-pinion drive. The reach direction has a range of 0.4m, and a range from 40^o to 90^o in the pitch direction. Load cells are used to measure force signals from the human and the actuator, along both the directions of motion. Experiments are performed at a supply pressure of $6894.75 N/m^2$. Moog series 7 servo valves are used to control both the actuator and motor.

With a torque amplification factor of 3, force tracking results are illustrated: when the system is suddenly loaded with a 9.7kg block (Fig. 5) in free motion, and when it is pushed against a rigid obstacle Fig. 6. In all cases, force tracking results are quite reasonable.

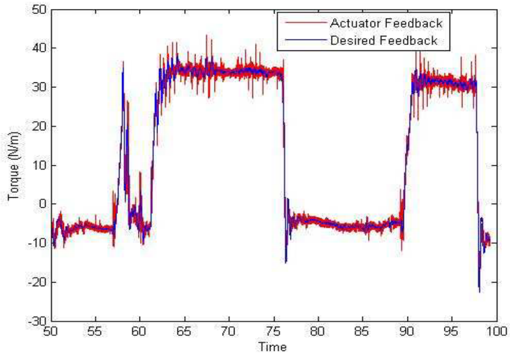

Figure 5. Hydraulic force tracking in free motion with load added and removed.

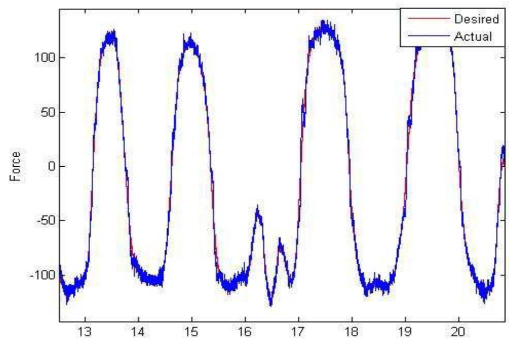

Figure 7. Pneumatic force tracking in Free motion.

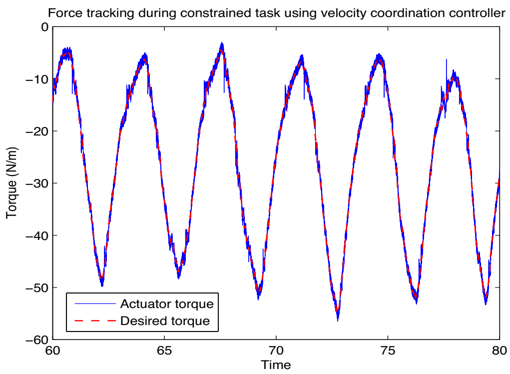

Figure 6. Hydraulic force tracking during constrained task.

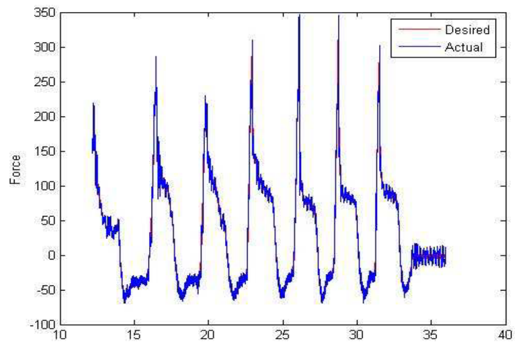

Figure 8. Pneumatic force tracking during repeated hard contacts.

Pneumatic case

The pneumatic human power amplifier is a one degree of freedom setup consisting of a pneumatic actuator supporting a 4kg load. The human interacts with the setup through a load cell. The force exerted by the actuator is determined by measuring the pressure in the chambers on either side of the piston. The actuator has in-built position feedback sensor which is used to determine the piston position. All the experiments are done at a supply pressure of 5.7 bar (gauge). A Festo 5/3 proportional servo valves is used for controlling the air flow rate.

The amplification gain for pneumatic actuator is set at 5. Fig. 7 shows the force tracking result when moving a load, Fig. 8 shows the force tracking results when the actuator hits an obstacle. The system is not destabilized by sudden contact and is quite safe to use.

CONCLUSIONS

A unified control framework hydraulic and pneumatic human power amplifier is proposed. The control law converts a force control objective into a coordination control paradigm. The key aspect lies in modeling the compressibility effect and imposing a control structure that respects the power continuity. Experimental results demonstrate the usefulness of the framework.

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