Motion Control of Hydraulic Actuators In the Presence of Discrete Pressure Rail Switching

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Abstract—Conventional off-highway mobile machines are hydraulically actuated and have poor energy efficiencies. The goal to increase system efficiency and to reap the benefits of electrification have led to the creation of a novel Hybrid Hydraulic-Electric Architecture (HHEA) which could significantly increase efficiency, decrease the sizes and cost of electrical components, and maintain control performance. The key concept is to utilize a set of common pressure rails to provide the majority of power via the power dense hydraulics and to modulate that power by small electric components, which are less power dense, for precise control.

The paper presents the motion control strategy for this architecture. The architecture presents a distinct challenge in that as the energy minimization algorithm selects different pressure rails, the system could experience large discrete jumps in pressures that impact control performance, especially when the electric components are torque limited. To meet this challenge, a passivity based back-stepping controller is used as the nominal controller between pressure rail switching events and a separate transition controller is used to deal with the pressure rail switching events. Two transition controllers are proposed, one uses the electric motor torque as the input, and the other also uses valve timing as a second input. For both transition controllers, a least norm control approach is used to steer the system states to the exact desired states at the end of the short transition period (~40 ms). The transition controllers are able to reduce the tracking error and the required electric motor torque, and hence achieve better control performance even with smaller electric components.

I. INTRODUCTION

Conventional heavy-duty off-highway vehicles used in the construction, agricultural and turf industries are hydraulically actuated to take advantage of hydraulics’ unsurpassed power density. However, the efficiencies of these machines are low (~21% from engine to output power [1]) due to the use of throttling for control, components operating at partial displacements where they are less efficient, and the inability to recuperate energy from overrunning loads.

A novel system architecture that combines the power density advantage of hydraulics with the efficiency and controllability advantages of electric actuation has been proposed to increase efficiency [2]. This Hybrid Hydraulic-Electric Architecture (HHEA) (Fig. 1) uses a set of common pressure rails (CPRs) and an electric motor driven pump/motor to control each degree-of-freedom. Here, the flow and hence the actuator speed can be directly controlled via the electric motor drive. At each moment, a pair of pressure rails for the inlet to the pump/motor ($P_A$) and for the return port of the hydraulic actuator ($P_B$) is selected to provide a hydraulic force that is close to the desired actuator force. This way, the electric motor is responsible only to make up for the difference. Consequently, the majority of the power is transmitted hydraulically and the electric motors serve only to modulate that power. Since electric machines tend to have much lower power density and higher cost than hydraulics, the HHEA enables high power machines to reap the benefits of electrification at lower cost and with smaller components. The HHEA is inherently throttle-less and is capable of capturing regenerative energy from the load. Previous studies have shown that the system architecture can potentially reduce energy consumption by over 55-65% compared to the baseline load sensing system for a variety of off-road mobile machines [2].

This paper addresses the motion control aspect of the HHEA. Precise motion control is important for off-road

Fig. 1. Top: The hybrid hydraulic-electric architecture (HHEA) with 3 services and 3 pressure rails at 0 MPa, 17.5 MPa, and 35 MPa. The electric generator/motor at the engine is optional. Bottom: Hydraulic-Electric Control Module (HECM) for a linear actuator.
vehicles whose large part of utility depends on the machines being dexterous and performing tasks exactly as commanded. The HHEA poses a unique challenge to motion control due to the discrete pressure changes when the selected pressure rails for the pump/motor inlet and the actuator return switch from one selection to another in order to minimize system efficiency or to keep the system within the torque capability of electric motor.

The proposed motion control strategy is to utilize two different controllers: 1) for the duration between consecutive pressure rail transitions, and 2) for the short time when the pressure rails undergo discrete switching. A passivity based back-stepping controller is used for a former, while an open-loop least-norm controller is used for the latter. This partition is necessary because the electric motor has limited torque capability so that it will not be able to completely compensate for the discrete pressure changes due to rail switches. The passivity based back-stepping control is similar to that in [3] where a hydraulic-transformer is used for control. Control in the presence of discrete pressure switches is also encountered in the control of the switched mode hydraulic transformer [4]. There, a bumpless transfer with a delayed action was adopted. The solution proposed in the present paper is more comprehensive.

The rest of the paper is organized as follows. Section II describes the HHEA and our control objectives. The nominal controller is briefly presented in Section III. Two versions of the transition controllers are presented in section IV. Simulation results are presented in section V. Concluding remarks are given in section VI.

II. SYSTEM DESCRIPTION AND CONTROL OBJECTIVES

In the HHEA, there is set of $N$ common pressure rails each at a different nominal pressure $(P_{R1}, \ldots, P_{RN})$. The rails were fed by a common pump/motor and the pressure at each rail is regulated by a hydraulic accumulator. For each degree-of-freedom, there is a hydraulic-electric control module (HECM) that combines hydraulic power from the pressure rails with electric power to actuate the linear degree-of-freedom (See Fig. 1-bottom). Ignoring pressure and inertia dynamics for the moment, if the pressure rails with pressures $P_{Ri}$ and $P_{Rj}$ are selected for the inlet of the pump/motor $P_B$ in the HECM and the return of the hydraulic actuator $P_A$, and the electric motor torque is $T_m$, the resultant actuator force and speed are

$$F_{act} = P_{Ri}A_{cap} - (P_{Rj} + \frac{2\pi}{D} T_m)A_{rod}$$

$$= \left( P_{Ri}A_{cap} - P_{Rj}A_{rod} \right) - \frac{2\pi}{D} T_m$$

$$\dot{x} = -\frac{D}{2\pi A_{rod}} \omega$$

where $D$ is the pump/motor displacement, $A_{cap}$ and $A_{rod}$ are the cap and rod side piston areas, $F_{rail}$ and $F_{ elect}$ are the portions of the actuator force provided by the selected pressure rails and by the electric motor. There are $N^2$ options of $F_{rail}$ based on different selections of pressure rails. Thus, from (1), by choosing $F_{rail}$ close to the desired $F_{act}$, the electric motor torque can be minimized (and the electric motor can be downsized). In general, the choice of $(P_{Ri}, P_{Rj})$ is to minimize the energy loss globally.

In the presence of pressure and inertia dynamics, the system dynamics are:

$$M\ddot{x} = P_{cap}A_{cap} - P_{rod}A_{rod} - F_L$$

$$\dot{P}_{rod} = \frac{\beta}{V_{rod}(x)} (Q + A_{rod}\dot{x})$$

$$Q = \frac{D}{2\pi \omega}$$

$$J\omega = \frac{(P_B - P_{rod})D}{2\pi} + T_m$$

where $F_L(t)$ is the load on the actuator, $P_{cap} = P_A$ and $P_{rod}$ are the actuator chamber pressures, $\beta$ is the bulk modulus of the fluid, $V_{rod}(x)$ is the rod side fluid volume, $J$ is the inertia of the electric motor and the pump/motor, and $P_B$ is the inlet pressure of the pump/motor. $P_A$ (i.e. $P_{cap}$) and $P_B$ are connected to the selected pressure rails so that $P_A, P_B \in \{P_{R1}, \ldots, P_{RN}\}$. Because these selections can change instantaneously, $P_B(t)$ and $P_{cap}(t)$ can undergo discrete jumps. In between rail transitions, $P_B$ and $P_{cap}$ are constant.

The control objective is to utilize the motor torque $T_M$ to track the desired position $x_d(t)$ trajectory specified by the operator. It is assumed that $x_d(t), \dot{x}_d(t), \ddot{x}_d(t)$ and as well as the load on the actuator $F_L(t)$ are known. In addition, the desired rail selections at any time are assumed to have been determined by another controller concerned with maximizing energy saving. However, the controller is expected to have some flexibility in terms of the exact time when the switching occurs.

III. NOMINAL CONTROL: PASSIVITY BASED BACK-STEPPING

In between rail transitions, a non-linear feedback control is used for trajectory tracking. Since a hydraulic actuator has been shown to be a passive two port system [5], a passivity based back-stepping control design that makes use of the natural energetic interaction is used. An energy based Lyapunov function is used to establish stability and convergence. By relying on the inherent physical structure, stability does not require the control to cancel specific terms, and there are fewer controller parameters to be tuned. Similar to the situation in [6], the controller proposed in [5] where flow $Q$ in (5) is the input has to be extended to use electric motor torque $T_M$ in (6) as the input.

Referring to (3)-(6), control design proceeds from assuming successively $P_{rod}, Q$, and finally $T_M$ as the inputs. At each step, the Lyapunov function for proving stability is successively extended to include additional states by adding to it the energy associated with that state. The procedure is summarized below.

Let $e = x - x_d$ be the tracking error and assuming $\dot{x}$ is the control input, the control law

$$r := \dot{x} - \lambda_p e; \quad \lambda_p > 0$$

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leads to the stable dynamics $\dot{e} = -\lambda_p e$. Next assume that $P_{rod}$ is the control input. The extra state to be included is: $e_v := \dot{x} - v$. The control law and Lyapunov function and its derivative are:

$$P_{rod,d} = \frac{1}{A_{rod}} \left[ P_{cap} A_{cap} - F_L - M \dot{r} + K_e e_v + K_p e \right]$$  \hspace{1cm} (8)

$$V_2 := \frac{1}{2} M e_v^2 + \frac{1}{2} K_p e^2$$  \hspace{1cm} (9)

$$\dot{V}_2 := -K_e e_v^2 - \lambda_p K_p e^2 + \tilde{P} A_{rod}$$  \hspace{1cm} (10)

where $\tilde{P} := P_{rod} - P_{rod,d}$ is the pressure error. Thus, if $\tilde{P} \to 0$, then $e \to 0$, and $e_v \to 0$.

Next we assume that the output flow of the pump/motor $Q$ is the control input. The control law and the Lyapunov function are:

$$Q_d = A_{rod} \dot{r} + \frac{V_{rod}(x)}{\beta} \tilde{P} - \lambda_3 \tilde{P}$$  \hspace{1cm} (11)

$$\dot{V}_3 := V_2 + V_{rod}(x) W_{V}(\tilde{P}, P_{rod,d})$$  \hspace{1cm} (12)

where $W_{V}(\tilde{P}, P_{rod,d})$ is the energy density associated with compressing the fluid from $P_{rod,d}$ to $P_{rod}$ [7]. It can be shown [7] that when $\lambda_3$ is sufficiently large, $V_3$ satisfies:

$$\dot{V}_3 \leq -[e \quad e_v \quad \tilde{P}] M \begin{bmatrix} e \\ e_v \\ \tilde{P} \end{bmatrix} + \Phi_v(\tilde{P}) \tilde{Q}$$  \hspace{1cm} (13)

where $M$ is positive definite, $\tilde{Q} = Q - Q_d$ and $\Phi_v(\tilde{P}) := \tilde{P} + W_{V}(\tilde{P}, P_{rod,d})$ is the hydraulic effort (conjugate to hydraulic flow $Q$). Note that the desired $Q_d$ can be referred to the desired pump/motor speed $\omega_d$ via

$$Q_d = \frac{D}{2\pi \omega_d}$$

Finally, with the actual control input $T_m$ as the control input, the control law and Lyapunov function are:

$$T_m = \frac{D}{2\pi} (P_{rod} - P_B - \tilde{P}) - \lambda_4 \tilde{\omega}$$  \hspace{1cm} (14)

$$V_4 := V_3 + \frac{1}{2} J \tilde{\omega}^2$$  \hspace{1cm} (15)

$$\dot{V}_4 \leq -[e \quad e_v \quad \tilde{P}] M \begin{bmatrix} e \\ e_v \\ \tilde{P} \end{bmatrix} - \lambda_4 \tilde{\omega}^2$$  \hspace{1cm} (16)

where $\tilde{\omega} = \omega - \omega_d$. This shows that $e, e_v, \tilde{P}, \tilde{\omega}$ all converge to 0 exponentially.

**IV. TRANSITION CONTROL: LEAST NORM CONTROL**

The nominal controller guarantees good tracking performance under nominal conditions. However, when the pressure rails switch, pressure error can instantaneously become very large. As seen from (14), it demands a significant torque input that far exceeds the capability of the electric motor/drive. This is especially significant when both pressure rails are switched at the same time. Recall that the premise of the system architecture is that it requires only small electric machines. One can saturate the torque at its maximum level and hope for the best. This unfortunately leads to quite significant increase in tracking error and it takes a long time for the controller to recover (see Fig. 2 for an example).

In this section, we design a controller during a short transition period (about 40ms) during which the effect of the pressure rail changes on the motion tracking is minimized while keeping the control input to a realizable value. After the transition period, control is handed back to the nominal controller. In this particular approach, the transition control aims to reach a state with zero error at the end of the transition.

**A. Least Norm Control**

The rod side chamber volume $V_{rod}(x)$ is assumed to remain constant during the very short transition period. The state dynamics in (3)-(6) thus become a linear time-invariant system:

$$\dot{X} = AX + BP \begin{bmatrix} F_L(t) \\ P_{cap}(t) \\ P_B(t) \end{bmatrix} + BU T_M(t)$$  \hspace{1cm} (17)

where

$$A = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & -\frac{1}{2\pi} & 0 \\ 0 & 0 & 0 & -\frac{D}{2\pi J} \end{bmatrix}$$

$$B_P = \begin{bmatrix} -1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}$$

Here, we assume that the time courses of $P_{cap}(t) = P_A(t)$ and $P_B(t)$ are known. In particular $P_B(t)$ switches instantaneous to the new selected rail pressure at the transition time, whereas $P_{cap}(t)$ will switch after a slight delay (to be determined).

Given the initial state $X(t_0)$ at the beginning of the transition period, and the known time courses of $P_B(t)$ and $P_{cap}(t)$, our goal is to use motor torque $T_M(t)$ to control the state to the desired values $X_f$ at the end of the transition period $t_f$. A computationally efficient way for computing the control is via least norm control - i.e. the control that satisfies the requirement and minimizes the $L_2$ norm of $T_M(t)$.

1 For continuous actuator load and if both the prior and next rail selections are within the electric motor torque capability, the electric motor should have sufficient capability to cope with the rare instances when only one pressure rail switches so that a separate transition control may not be necessary.
The solution to the linear system (17) is:

\[ X(t_f) = \Phi(t_f, t_0)X(t_0) + \int_{t_0}^{t_f} \Phi(t_f, \tau)B_P \begin{bmatrix} F_L(\tau) \\ P_{cap}(\tau) \\ P_B(\tau) \end{bmatrix} d\tau \]

\[ + \int_{t_0}^{t_f} \Phi(t_f, \tau)B_L T_M(\tau) d\tau \]

(18)

where \( \Phi(t_2, t_1) \) is the transition matrix of \( A, L_r : \mathbb{R}^{[t_0, t_f]} \rightarrow \mathbb{R}^1 \) denotes the reachability map. Defining

\[ \Delta x := X_f - \Phi(t_f, t_0)X(t_0) - \int_{t_0}^{t_f} \Phi(t_f, t_\tau)B_P \begin{bmatrix} F_L(\tau) \\ P_{cap}(\tau) \\ P_B(\tau) \end{bmatrix} d\tau \]

the least norm control problem is to find \( T_M \):

\[ \min_{T_M(\cdot)} \int_{t_0}^{t_f} T_M^2(\tau) d\tau \quad \text{s.t.} \quad \Delta x = L_r[T_M(\cdot)] \]

Since the system (17) can be shown to be completely controllable from \( T_M \), the reachability map \( L_r \) is surjective and the least norm control is given by [8]:

\[ T_M(\cdot) = L^*_r(L_rL^*_r)^{-1} \Delta x \]

(20)

where \( L^*_r \) is the adjoint of \( L_r \). Explicitly, it is written as:

\[ T_M(\tau) = B_L^T \Phi^T(t_f, \tau)G^{-1} \Delta x \]

(21)

\[ G = \int_{t_0}^{t_f} \Phi(t_f, t_\tau)B_L B_L^T \Phi^T(t_f, \tau) d\tau \]

(22)

where \( G \) is the reachability Gramian.

**B. Scaling for different transitions**

The least norm control solution in (19) and (21) can be computed efficiently for different rail switch conditions. In (21), the transition matrix \( \Phi(t_f, \tau) \) and the Gramian \( G \) can be precomputed. Furthermore, if \( P_B(t) \) is a step change and \( P_{cap}(t) \) is a delayed step change with a known delay, \( \Delta x \) in (19) can be expressed as a linear function of \( X(t_0), X_f, \) the new value of \( P_B \), and the old and new values of \( P_{cap} \). Therefore, the least-norm control at any time \( T_M(\tau) \) can be computed by multiplying a precomputed kernel with a scaling matrix.

Although the rod side chamber volume \( V_{rod}(x) \) can be assumed to remain constant during the transition time, it differs for different transitions. Hence the precomputed kernel needs to be parameterized by the fluid volume.

**C. Transition control with motor torque and valve-timing**

The transition control strategy in section IV-A assumes instantaneous changes from the pressure of one pressure rail to another, and that the electric motor torque \( T_M \) is the only control input. In reality, during rail switching, the switching valve connected to the rail prior to the switch has to be closed and the valve connected to the rail to be connected has to be opened. These valve transitions take time during which the relevant chamber pressure transitions from one pressure to another. This reality suggests that valve timing, especially on the capside of the actuator, can be a second control input during transition.

To utilize this secondary control input, we partition the transition control into two sub-problems: 1) the electric motor torque \( T_M \) is used to control the pump/motor speed \( \omega \) and the rod side chamber pressure \( P_{rod} \), and 2) the capside valve timing is used to more precisely control the actuator position and velocity \( x \) and \( \dot{x} \).

1) **Rod-side pressure control:** Assuming the piston position and velocity are indeed precisely controlled, the rod-side dynamics are:

\[ \begin{bmatrix} \dot{P} \\ \dot{\omega} \end{bmatrix} = \begin{bmatrix} \frac{\beta D}{2\pi V_{rod}} & 0 \\ \frac{\beta A_{rod} \omega}{2\pi J} P_B(t) \end{bmatrix} \begin{bmatrix} P \\ \omega \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{1}{J} \end{bmatrix} T_M(t) \]

(23)

Similar to the least norm control for the full system in section IV-A, \( T_M(\cdot) \) is computed as the least norm control that steers the initial states from \( (P(t_0), \omega(t_0)) \) to the desired values at \( t = t_f \) with the least \( L_2 \) effort.

2) **Cap-side control:** With consideration of the valve dynamics during switching, the actuator and cap side chamber pressure dynamics are given by:

\[ M \ddot{x} = P_{cap}(t)A_{cap} - P_{rod}(t)A_{rod} - F_L(t) \]

(24)

\[ \dot{P}_{cap} = \frac{\beta}{V_{cap}(x)} (Q_{valve} - A_{cap} \dot{x}) \]

(25)

\[ Q_{valve} = k_v x_{v1}(t) \text{sign}(P_{rod}^r - P_{cap}) \sqrt{|P_{rod}^r - P_{cap}|} + k_v x_{v2}(t) \text{sign}(P_{rod}^r - P_{cap}) \sqrt{|P_{rod}^r - P_{cap}|} \]

(26)

Here \( V_{cap}(x) := V_{cap} - A_{cap} x \) is the capside fluid volume, \( P_{rod}^r \) and \( P_{cap}^r \) are the pressures of the previous and next selected rail pressures, \( x_{v1} \) and \( x_{v2} \) are the valve openings for the connections to these rails, and \( k_v \) is the valve constant for both valves.

Various open loop and closed loop methods can be applied to control the actuator position \( x(t) \) using valve openings \( x_{v1}(t) \) and \( x_{v2}(t) \) as control input. In general, we seek \( x_{v1}(t) \) and \( x_{v2}(t) \) to be monotonic. In the results below, an open loop approach is used in which the valve opening profiles are predetermined. A closed loop approach, e.g. passivity based back-stepping, can also be used.

**V. RESULTS AND DISCUSSION**

The control strategy is illustrated for a desired trajectory and a rail transition in which \( P_A(t) \) switches from \( 30 \text{ MPa} \) to \( 40 \text{ MPa} \), and \( P_B(t) \) switches from \( 0 \text{ MPa} \) to \( 20 \text{ MPa} \) (see insets of Fig. 2).

Figure 2 shows the tracking result with the back-stepping control with and without the electric motor torque being saturated at \( \pm 350 \text{ N-m} \). Note that the back-stepping controller is able to track the desired trajectory except at the instances of rail transition. Without saturation, the tracking error is of the order of \( 0.2 \text{ mm} \) but at the expense of very high torque (\( 4000 \text{ N-m} \)). When the motor torque is limited, 7mm peak error occurs and the tracking is not recovered for 100 ms.

To demonstrate the transition control, we focus on the sample transition in Fig. 2. Transition control using least
Fig. 2. Trajectory tracking performance and torque input using the nominal back-stepping controller only using a desired trajectory and a sample pressure switch: with and without torque saturation at $\pm 350\text{Nm}$.

Fig. 3. Performance for different transition times - least norm control using only motor torque as input. Time delay of the $P_A$ switching is 8 ms.

Fig. 4. Torque inputs for different transition times - least norm control using only motor torque as input. Time delay of the $P_A$ switching is 8 ms.

Fig. 5. Torque input for various cap-side delays - least norm control using only motor torque as input.

The transition control using both motor torque and cap-side valve timing are investigated next. Figures 6-7 show the performance of this approach and the valve opening profiles and the resultant cap-side pressures. The transition time is kept at 40ms, which is a good trade-off between performance and torque input. Notice that the tracking error has been significantly reduced to less than 0.3 mm compared to the 1-1.5mm in Fig. 3. Despite the improvement, Fig. 8 shows that motor torque requirement is slightly less than the transition control that uses motor torque as the only control input.

Figure 9 compares the tracking error using the back-stepping controller with motor torque saturated at $\pm 350\text{Nm}$, and the combination of back-stepping and the least norm transition controller using only motor torque as control input. The transition controller is able to reduce the peak errors and ensure that error is zero at the end of the transition and then it hands over the control to the back-stepping controller. The transition controller also makes sure that the transition time is short and as it can be seen the transition time is 40 ms with the combination of the two controllers while it is 100ms with just the back-stepping controller.

VI. Conclusion

In the proposed hybrid hydraulic-electric architecture for off-road mobile machines, switching of the common pressure rails are fundamental to realizing the premise of transmitting the majority of the power via hydraulics while small electric
machines are used to modulate that power. This paper has presented an approach of dealing with the adverse effect on control performance when pressure rails are switched. While a small amount of tracking error is tolerated during transition, a linear least norm transition controller ensures that the tracking error returns to zero at the end of the transition when the nominal back-stepping controller takes over again. A hardware in the loop (HIL) setup is being constructed to validate the proposed approach experimentally.

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