Pressure Feedback Control Of Electro-Hydraulic Actuators Using Fixed Displacement Hydraulic Machines

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Abstract.

This paper presents the development of a control strategy for Electro-Hydraulic Actuator (EHA) systems based on fixed displacement hydraulic machines. Although some previous studies on EHAs show benefits in terms of energy efficiency and regeneration capability compared to conventional hydraulic systems, few studies focus on the dynamic performance of EHAs. Unlike traditional valve-controlled systems, an EHA based on a fixed displacement pump typically relies on variation of the electric motor shaft speed to control the actuator velocity. This study shows that pressure feedback can be used to control the speed of the electric machine, thus achieving the required actuator machine dynamic response. A simulation model is developed by including the EM, the hydraulic system, and the actuation system, which can give a realistic prediction of the EHA performance for a typical duty cycle. An open-circuit EHA previously proposed by the authors is used to validate the control strategy, and the simulation results show superior dynamic performance, exhibiting a reduction in response time and oscillatory behavior, compared to a feedforward EHA controller. Following the method presented in this study, it is possible to design a controller with a pressure feedback strategy for EHA systems with different size and actuation requirements.

Keywords. EHA, Dynamic performance, Pressure feedback control.

1. INTRODUCTION

The electrification trend affecting the off-highway vehicle market has increased the interest in high-efficiency electro-hydraulic actuator (EHA), which can address the limitation of efficiency of traditional valve-controlled actuator systems. Particularly attractive are solutions that combine a variable speed electric machine with a fixed displacement hydraulic machine due to considerations of cost and efficiency [1, 2]. Compared to the traditional valve-controlled system, an EHA based on a fixed displacement pump typically controls the actuation velocity by varying the pump flow through the electric machine (EM) speed. Therefore, it is arguable that such an EHA can adopt the same control as a conventional hydraulic system to achieve the same dynamic performance, such as limiting the oscillations of the actuator.

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With the purpose of reducing the actuator oscillations, pressure feedback control is proved to be effective for traditional valve-controlled systems and for displacement-controlled systems. The pressure feedback is implemented with high-pass filter to eliminate the faster dynamic from the pulsations, while using the information from the velocity response in the control loop. This method has been proved effective to improve dynamic performance of hydraulic system, such as adaptive damping. This is documented in several studies: for example, [3, 4] describe the use of the pressure feedback strategy in a pump-controlled actuator to achieve active damping and to eliminate unstable switching between pumping and motoring that can occur in a closed-circuit system. [5] applies an active oscillation damping strategy in a displacement-controlled actuator to increase the productivity of the machine and comfort of the operator. Ding et al. [6] discuss the case of vibration reduction of an excavator with an independent metering system. Pressure feedback can also be used for the so-called ride control, to reduce cabin vibrations based on the commanded motion of the loader function [7]. Pedersen and Andersen well explained the principle of pressure feedback in hydraulic control systems through detailed mathematical analysis, and provided guidelines to tune the control parameters using the root-locus method, and on the selection between high and low pass filter [8].

Nevertheless, all the mentioned examples are based on valve control, where the idea is to control the flow rate by adjusting the opening of the valves. In this study, a novel method is studied by pump control, i.e., controlling the pump speed to adjust flow rate with pressure feedback strategy. In fact, instantaneous pump displacement control was achieved using a servo valve in [3], while in the other cases, the control action was performed directly by a metering control element. Therefore, for an EHA system, the main question arises on the opportunity to utilize a pressure feedback strategy to increase the damping of the system. The dynamics of an electric machine (EM) is very different from that of hydraulic valves. Taking the permanent magnet synchronous machine (PMSM) as an instance, in speed control mode, the torque command is converted to speed via a PI control structure. The motor speed response depends on the control parameters, as well as the machine parameters such as inertia and torque limit [9]. The EM usually cannot respond as fast as hydraulic control valves unless the maximum torque available is very high with respect to the system torque requirements. This means an EM rotor and shaft inertia much higher than a valve spool, which limits the acceleration.

This study aims to address the challenges mentioned above and provides the development of a pressure feedback controller for the EHA system. The pressure feedback signal from the actuator is processed via a high-pass filter with a feedback gain, and then summed with the speed command to the EM before its torque-speed PI controller. A simulation model validated with experimental results that integrate the EM control, hydraulic system, and the mechanism of the actuator is used to validate the pressure feedback controller effectiveness. On the basis of the simulation results, the damping vibration is reduced with the closed-loop pressure feedback control compared to a feedforward-controlled EHA.

The structure of the paper is as follows: in Section 2 the reference system will be introduced, focusing on the control structure; Section 3 gives the method to formulate the controller; finally, the simulation model and results will be shown in Section 4 before concluding the work in Section 5.

2. REFERENCE SYSTEM

2.1. Hydraulic layout

The architecture of the studied system along with the reference machine is shown in Figure 1. It is considered as a throttle-less circuit with a hydraulic energy efficiency that can reach 84.7\%, as well documented in previous work [12]. A two quadrant hydraulic pump (HP) driven by a variable electrical machine (EM) controls the speed of the cylinder based on an open circuit configuration. The directional valve (4/3 DV) is only used to switch among the working modes to allow for a four-quadrant operation of the system. A BPV valve is used to allow for slow actuation velocity whenever the speed demand is lower than the minimum recommended pump speed operating region, and for fast actuation velocities in overrunning loads. Although the circuit studied in this paper includes a BPV that can be used for active damping, this might not be the case in other circuits that can achieve low speed modes without the need of a throttling element, for this reason the authors conceived the idea examining the potential of using an electrical motor to increase the damping of the system. The detailed working principle, control logic [10,11] and the performance of this EHA layout [11] are explained in previous studies by the authors.

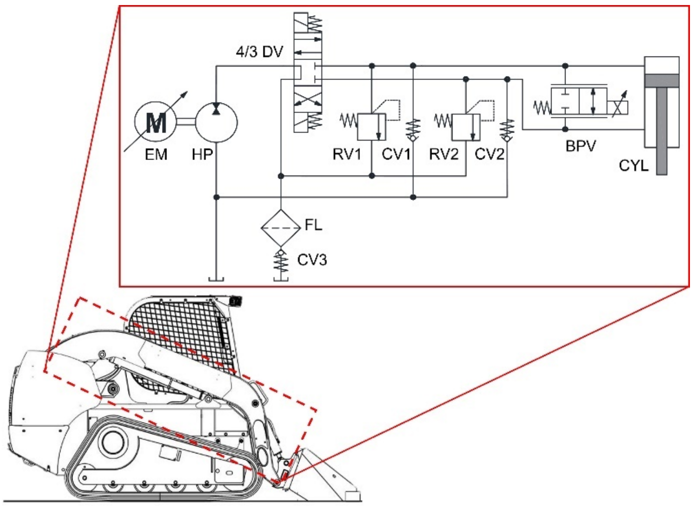

Figure 1. Hydraulic circuit of the reference machine

2.2. Control structure

Figure 2 represents the control flow of the studied EHA system. The operator sets the desired actuator speed by a joystick command. This speed command is properly converted to an equivalent EM shaft speed command. The details of this process are given in [10]. The speed command is an input to the EM inverter (power electronics), which is converted to a torque command with a PI controller that uses speed feedback. With the torque command, the EM and the HP can reach the desired speed and supply flow to the hydraulic system [13].

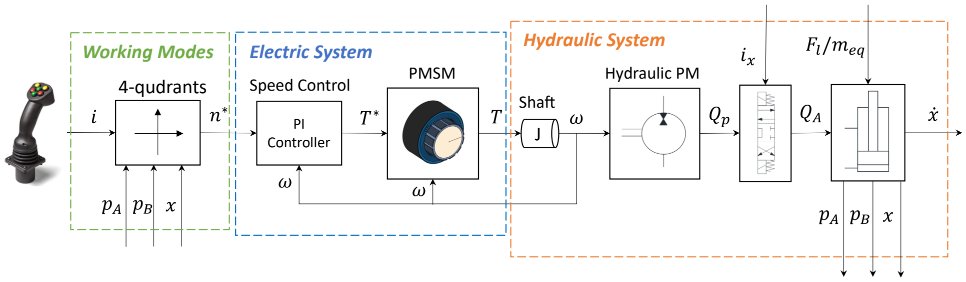

Figure 2. Hydraulic circuit of the reference machine

The working modes block includes the control logic for 4-quadrant operation, such as commanding speed to the electric machine accordingly (e.g., reverse speed for energy regeneration", and use the bypass valve for extreme low-speed operation. For brevity, more details can be found in other work of the authors [10-13].

This control logic requires the use of position and pressure sensors to identify the working modes of the EHA. The pressure sensors also allow one to implement the pressure feedback controller.

3. Formulation of the controller

3.1. *System linearization*

The linear model can be viewed as two separate parts: First, the boom actuation that includes the hydraulic system and the boom mechanism; Second, the electrical machine control.

3.1.1. Hydraulic boom actuation

For control design and analysis, a linearized model of the system is developed, starting from the boom mechanism highlighted in Figure 1. For the development of the linear model, the boom mechanism is simplified to a mass-spring-damper system. Figure 3 shows the scheme of the linear model that includes a hydraulic cylinder and the boom mechanism.

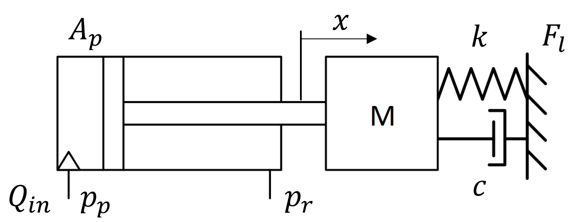

Figure 3. Linearized model of the boom cylinder mechanism

Based on the linear model, applying Newton's second law, the equation of motion of the cylinder is described as in equation (1), derived from the force balance of the hydraulic cylinder. The spring stiffness can be assumed as zero (i.e., $k \approx 0$), and the equivalent mass is calculated numerically from the kinematic model.

$$\ddot{x} = \frac{1}{M_{eq}}\left(A_p p_p - A_r p_r - c\dot{x} - kx - F_L\right) \tag{1}$$

The pressure build-up inside the cylinder chambers is given in equation (2) and equation (3), for piston side and rod side, respectively. Internal cylinder leakage is neglected for this analysis (i.e., $K_l = 0$).

$$\dot{p}_p = \frac{K_{oil}}{V_p}\left[Q_p - A_p\dot{x} - K_L\left(p_p - p_r\right)\right] \tag{2}$$

$$\dot{p}_r = \frac{K_{oil}}{V_r}\left[-Q_r + A_r\dot{x} + K_L\left(p_p - p_r\right)\right] \tag{3}$$

The hydraulic capacitance C_h can be defined as in equation (4)

$$C_h = \frac{V}{K} \tag{4}$$

The hydraulic capacitance of the piston and rod chamber side is defined in equation (5) and (6), respectively.

$$C_{h,p} = \frac{V_p}{K_{oil}} = \frac{V_{line,p} + A_p \cdot x}{K_{oil}} \tag{5}$$

$$C_{h,r} = \frac{V_r}{K_{oil}} = \frac{V_{line,r} + A_r \cdot (l - x)}{K_{oil}} \tag{6}$$

For a concise expression, the pressure differential across the cylinder is defined in equation (7), so that the force can be obtained as equation (8).

$$\Delta p^* = p_p - \frac{A_r}{A_p} \cdot p_r = p_p - \frac{1}{\lambda} \cdot p_r \tag{7}$$

$$F_{cyl} = A_p \Delta p^* = A_p p_p - A_r p_r \tag{8}$$

As a result, the mechanical system can be simplified to two equations in terms of force balance and pressure build-up. First, by combining equation (1) and (8) to form (9). Second, by substituting equation (4), (7) into equation (2) and (3) to form (10).

$$\ddot{x} = \frac{1}{M_{eq}}\left(A_p \Delta p^* - c\dot{x} - F_L\right) \tag{9}$$

$$\Delta\dot{p}^* = \frac{Q_{in}}{C_{h,eq}} - \frac{A_p\dot{x}}{C_{h,eq}} \tag{10}$$

Based on these two equations, a state-space representation of the linearized mechanism can be represented as in equation (11) and (12).

$$\begin{bmatrix} \Delta\dot{p}^* \\ \ddot{x} \end{bmatrix} = \begin{bmatrix} 0 & -\frac{A_p}{C_{h,eq}} \\ \frac{A_p}{M_{eq}} & -\frac{c}{M_{eq}} \end{bmatrix} \begin{bmatrix} \Delta p^* \\ \dot{x} \end{bmatrix} + \begin{bmatrix} \frac{1}{C_{h,eq}} & 0 \\ 0 & -\frac{1}{m_{eq}} \end{bmatrix} \begin{bmatrix} Q_{in} \\ F_l \end{bmatrix} \tag{11}$$

$$\begin{bmatrix} F_l \\ \dot{x} \end{bmatrix} = \begin{bmatrix} A_p & 0 \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \Delta p^* \\ \dot{x} \end{bmatrix} \tag{12}$$

The equivalent hydraulic capacitance $C_{h,eq}$ is defined as equation (13).

$$C_{h,eq} = \left(\frac{1}{C_{h,p}} + \frac{1}{\lambda^2 C_{h,r}} \right)^{-1} \tag{13}$$

Based on the equations above, the actuation velocity $\dot{x}$ can be represented as an s-function shown in equation (14), with the inlet flow rate Q_{in} and the load force F_l as the input.

$$\dot{x}(s) = \frac{1}{\left(s^2 + \frac{c}{M_{eq}} s + \frac{A_p^2}{M_{eq} C_{h,eq}} \right)} \left(\frac{A_p}{M_{eq} C_{h,eq}} Q_{in} - \frac{s}{M_{eq}} F_l \right) \tag{14}$$

The actuation velocity follows a second-order response, and the natural frequency and damping ratio can be obtained as in equation (15) and (16).

$$\omega_n = \sqrt{\frac{A_p^2}{M_{eq} C_{h,eq}}} \tag{15}$$

$$\zeta = \frac{c}{2} \sqrt{\frac{C_{h,eq}}{A^2 M_{eq}}} \tag{16}$$

In this hydraulic model, the losses of the pump (e.g., leakage) and throttles losses are also included. However, this information is more about the efficiency performance of the system, while the dynamic behavior is mainly determined by the mechanism with low natural frequency.

3.1.2. Electrical machine control

Regarding the EM control, the PI controller with the speed feedback shown in Figure 2 can be further detailed in Figure 4. To achieve a fast response of the EHU, whenever the user sends a command to the EM, the maximum torque is commanded to accelerate the shaft to the desired speed, before it goes back down to the steady state torque required by the pump. This maximum torque is limited based on the EM size, and is used to protect the EM from overheating. Thus, a saturation block is added to the PI output to protect the EM from overheating. The anti-windup strategy can be added to this control flow for better performance.

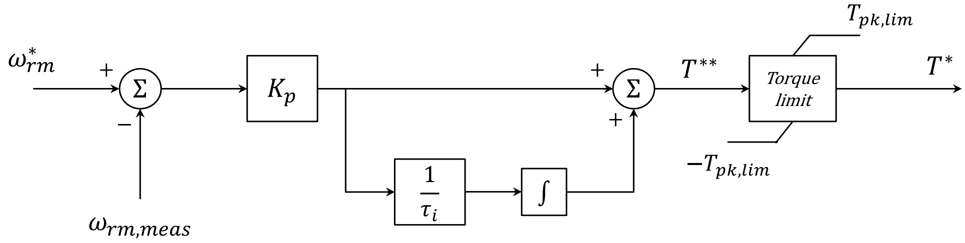

Figure 4. PI controller of the electric machine

Given a mechanical load on the EM, the dynamics of the EM mechanical parts is governed by equation (17). T_l computes as in equation (18). Considering the mechanical losses based on an efficiency map and assuming the inlet pressure to be equal to the tank pressure, we have $\Delta p^* \approx \Delta p$. The coefficient B is usually neglected and set at zero in this study. It may be possible to measure this value in the future study.

$$\dot{\omega} = \frac{1}{J}(T_e - B\omega - T_l) \tag{17}$$

$$T_l = \frac{\Delta p^* V_d}{2\pi} \tag{18}$$

The flow supplied by the EHU is represented in equation (19).

$$Q_p = n \cdot V_d \cdot \eta_{vol} \tag{19}$$

Combined with the linearized model given in equation (11) and (12), the linearized EHA system can be described as in equation (20) and (21), where the inputs are the torque T and the load force F_l.

$$\begin{bmatrix} \ddot{x} \\ \Delta\dot{p}^* \\ \dot{\omega} \end{bmatrix} = \begin{bmatrix} -\frac{c}{m_{eq}} & \frac{A_A}{m_{eq}} & 0 \\ -\frac{A_A}{C_{eq}} & 0 & C_0/C_{eq} \\ 0 & -C_0/J & -B/J \end{bmatrix} \begin{bmatrix} \dot{x} \\ \Delta p^* \\ \omega \end{bmatrix} + \begin{bmatrix} 0 & -\frac{1}{m_{eq}} \\ 0 & 0 \\ 1/J & 0 \end{bmatrix} \tag{20}$$

$$\begin{bmatrix} \dot{x} \\ F_l \\ \omega \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & A_p & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} \dot{x} \\ \Delta p^* \\ \omega \end{bmatrix} \tag{21}$$

Based on the linearized EHA model, the poles of the system can be calculated for stability analysis. Figure 5 graphically represents the system poles. The shaft inertia was estimated using the available CAD designs of the HP and the EM. The real pole represents the impact of the EHU speed response to a torque command. Two complex conjugate poles have a small damping ratio, corresponding to the behavior of the boom mechanism. The real pole is closer to the imaginary axis, thus dominating the response time. When increasing the inertia of the EHU shaft, the response time can increase. The PI controller described in Figure 4 addresses the real pole and maintains the fast response. Therefore, to reduce vibration, the feedback controller should focus on the complex conjugate poles.

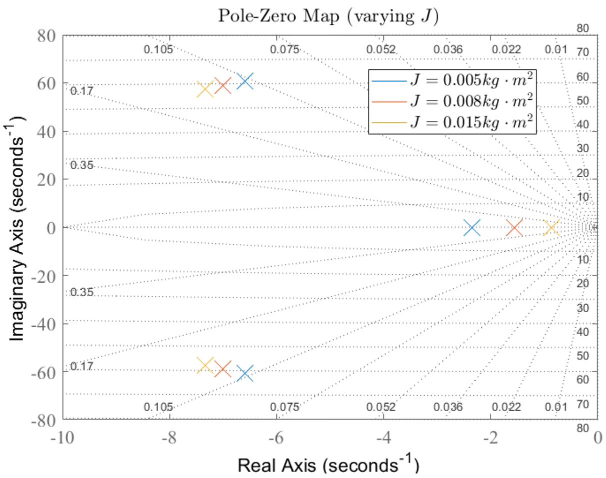

Figure 5. Pole position of the linearized EHA model with different rotational inertia

3.2. Pressure feedback

Figure 6 shows the controller with the pressure feedback controlling the EM speed. The pressure feedback controller includes a high-pass filter and a gain K_f which can be represented as in equation (22). The oscillation with the frequency lower than $1/\tau$ can be roughly excluded, thus the controller is effective for a certain bandwidth. The feedback control block between the working modes and electric system works as a summation point when the pressure feedback is activated. It also includes basic logic control so that the operator can disable the feedback function or adjust the control parameters, which is not the focus of the current study. Therefore, in the following sections, this block simply represents a summation point.

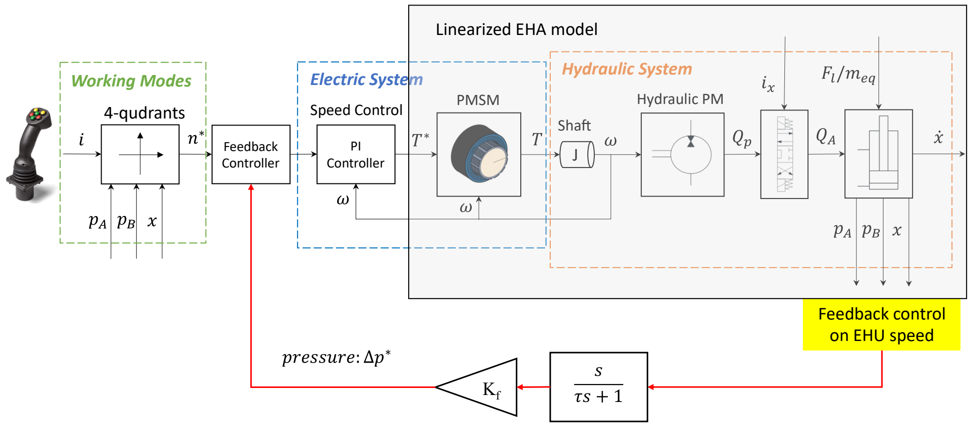

Figure 6. Controller with pressure feedback of the EHA system

$$H(s) = K_f \frac{s}{\tau s+1} \tag{22}$$

Focusing on the damping vibration, the complex conjugate poles are from the state-space representation given in equation (11). By setting the EM speed ω as input (i.e., replacing the flow input Q_{in}), the state-space representation can be rewritten as in equation (23). By comparing (11) and (23), it can be found the updated model is about the definition of the input.

$$\begin{bmatrix} \Delta\dot{p}^* \\ \ddot{x} \end{bmatrix} = \begin{bmatrix} 0 & -\frac{A_p}{C_{h,eq}} \\ \frac{A_p}{m_{eq}} & -\frac{c}{M_{eq}} \end{bmatrix} \begin{bmatrix} \Delta p^* \\ \dot{x} \end{bmatrix} + \begin{bmatrix} \frac{V_d}{2\pi C_{h,eq}} & 0 \\ 0 & -\frac{1}{M_{eq}} \end{bmatrix} \begin{bmatrix} \omega \\ F_l \end{bmatrix} \tag{23}$$

The plant described in (23) denotes as $G(s)$ With the pressure feedback controller, $H(s)$ given in (22), the transfer function can be stated in equation (24).

$$\frac{y}{u} = \frac{G(s)}{1+G(s)H(s)} \tag{24}$$

Setting the time constant within a reasonable range (e.g., $t \in (\frac{1.8}{\omega_n}, \frac{2.6}{\omega_n})$) according to [8], a root-locus plot can be obtained by varying the value of the feedback gain K_f. Figure 7 shows the root-locus plot.

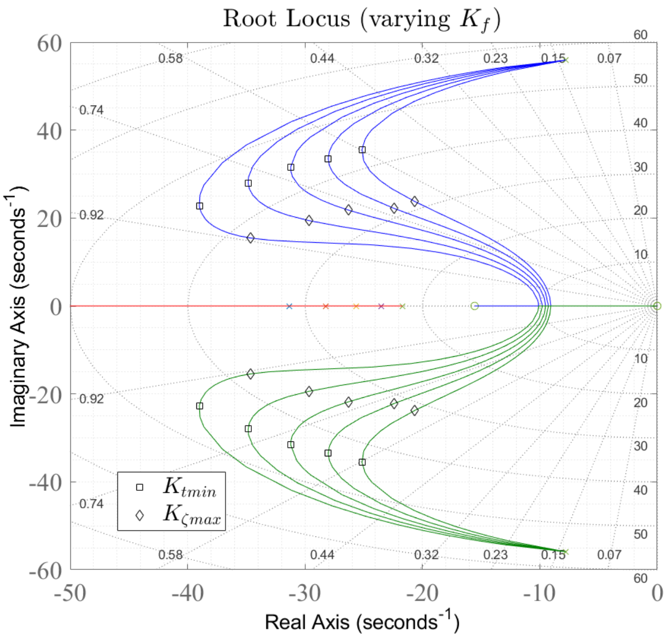

Figure 7. Root locus plot of the EHA system with pressure feedback controller

In the figure, the value between K_{tmin} (i.e., minimum response time) and $K_{\zeta max}$ (i.e., maximum damping ratio with relative high frequency) is suggested for K_f. However, because the dynamic behavior is estimated at the beginning, when tuning K_f a greater range deserves attempt. This root locus plot includes the motor model, but the PI-control is simplified as shown in equation (23). The input of the model, flow rate Q is translated to motor speed ω in a directly way. The derivation assumes: the speed response of electric machine is much faster than that of boom mechanism. In this study, the assumption is effective and the system performance is stable.

4. SIMULATION

4.1. Simulation model

An integrated EHA model that includes all the electrical-hydraulic-mechanical (EHM) domains in a single simulation platform is developed in Amesim environment. The coupling between the EHM parts of the model along with the control logic allows to simulate the system dynamics in a realistic way.The data for the valve dynamics were selected based on the valve data sheet, while the EM maximum torque is based on a prototype of the unit built at the authors lab at Purdue University. Figure 8 shows the scheme of the integrated EHA model. The details to develop such a lumped parameter model are explained in previous work by the authors [11], where the model is validated in terms of functionality and energy performance.

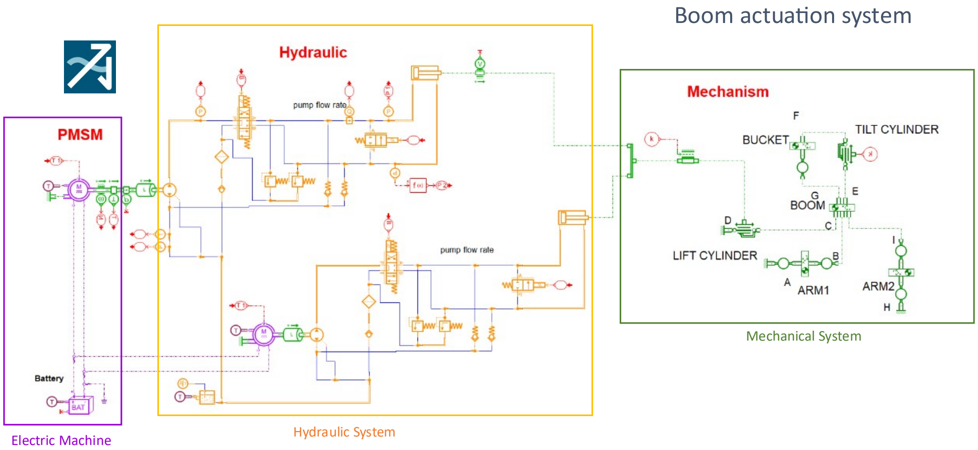

Figure 8. Integrated simulation model of the proposed EHA

The main parameters of the simulation model are listed in Table 1 for the EHU, while Table 2 gives the control parameters. The simulation will vary the feedback gain K_f to determine the optimal controller setup.

Parameters of components		Value [unit]
Hydraulic pump	Nominal displacement V_d	10.69 [cc/rev]
	Maximum speed n_{max}	4300 [rpm]

	Minimum speed n_{min}	500 [rpm]
	Maximum pressure p_{max}	200 [bar]
Electric machine	Nominal speed n_{nom}	4300 [rpm]
	Nominal torque T_{nom}	20.87 [Nm]
	Nominal current I_{nom}	24.22 [A]
	Number of poles P	10 [-]
	Shaft inertial J	0.008 $[kgm^2]$
	Rated voltage U_n	244 $[V_{rmx}]$
	Rated current I_n	7.32 $[A_{rmx}]$

Table 1 Parameters of the hydraulic pump and electric machine.

Parameters	**Value [unit]**
Natural frequency ω_n	56.45 [rad/s]
Feedback high-pass time constant τ	0.055 [1/s]
EM proportional gain K_p	10 [rpm]
EM integral gain K_i	2000 [-]
EM anti-windup gain K_{aw}	1000 [-]

Table 2 Control parameters in the simulation model.

4.2. Linear model parameters

The parameters used in the linear mechanism model are tuned based on the actual system size described in [11]. The optimal parameters are listed in Table 3.

Parameters	**Value [unit]**
Equivalent mass M_{eq}	4500 [kg]
Damping constant c	70000 [N/(m/s)]
Load force F_l	16000 [N]

Length of cylinder stroke l	0.8865 [m]
Cylinder piston side area A_p	3.837e-3 $[m^2]$
Cylinder rod side area A_r	2.828e-3 $[m^2]$
Oil bulk modulus K	1.4815e3 [MPa]

Table 3 Parameters used in the linearized model.

4.3. Results

First, the model is used to test the system response based on a feed-forward controller. Then, the proposed pressure feedback controller is compared with the open-loop control strategy.

Figure 9 shows the simulated step response of the actuator velocity and the corresponding displacement with feed-forward control relevant to Figure 2. The response has three phases as highlighted in the plot. In phase 1, an undershoot occurs due to the transition of the on/off valve. A 3mm undershoot of actuator displacement can be noticed. Nevertheless, for a machine with hydraulic cylinders which have a 900 mm length of stroke, it is negligible. This problem is addressed in [14] by using a pressure control, but it is not in the scopes of this study. The ramp in phase 2 represents the response of the EM, in terms of speed. Phase 3 is the focus of this study, which shows the damping oscillation of the actuator.

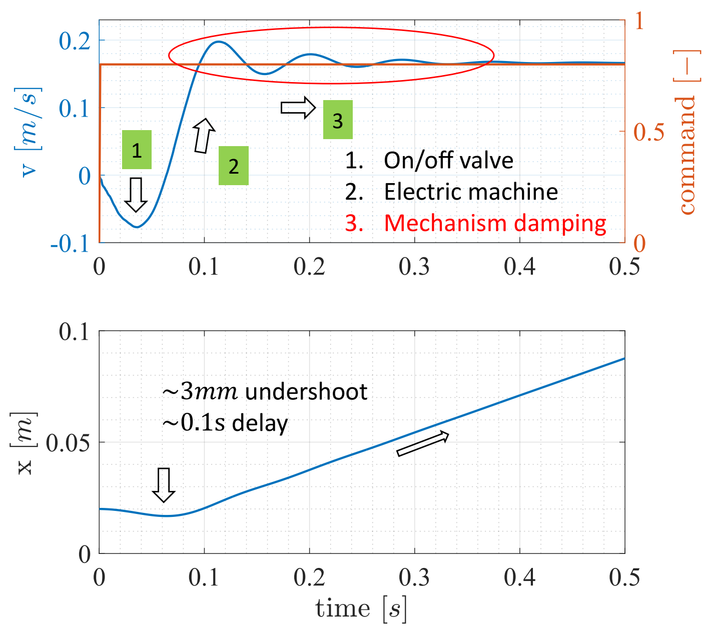

Figure 9. Step response of the EHA with feedforward control

Figure 10 shows a comparison of the velocity response of the hydraulic cylinder with a pressure feedback control (using different gains K_f), and without feedback. The red curve is the baseline response without feedback. The black line represents the optimal result, which effectively reduces the oscillation of the system. The blue and green dotted lines give results with different feedback gain values.

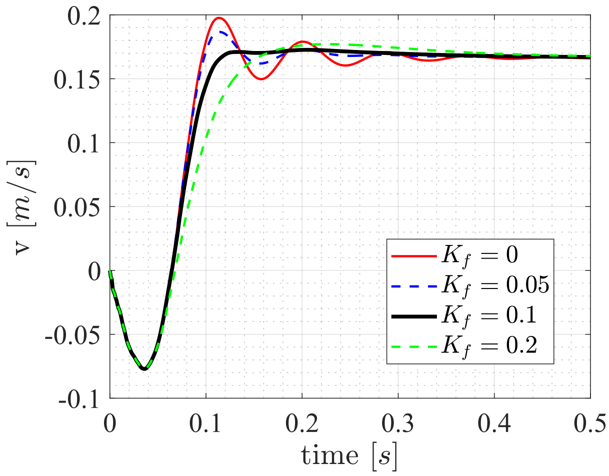

Figure 10. Step response of the EHA with pressure feedback

Focusing on the prime mover, Figure 11 shows a comparison of the torque and speed response with or without feedback control. In the feed-forward case (baseline), given a speed command, the maximum torque command is given to the EM to achieve the fastest response. However, this results in a low damping ratio in the actuator. With pressure feedback, the torque command of the EM is adjusted, leading to the change in the acceleration slope. The EM speed response in the rising time region also changes and can be seen in Figure 11 between 0.05 seconds and 0.14 seconds. This change in EM speed increases the damping ratio of the actuator and results in better dynamic performance of the actuator as shown in Figure 10.

The torque response appears not natural compared to original open-loop controller. This is due to the need of slowing down the motor speed for better dynamic performance with feedback controller. As a result, the torque does not need to keep the maximum in the motor acceleration phase. This is a novel method studied based on simulation model, so such torque performance still results in smooth speed control. For future work, experimental validation can be significant to investigate more about torque control and speed response.

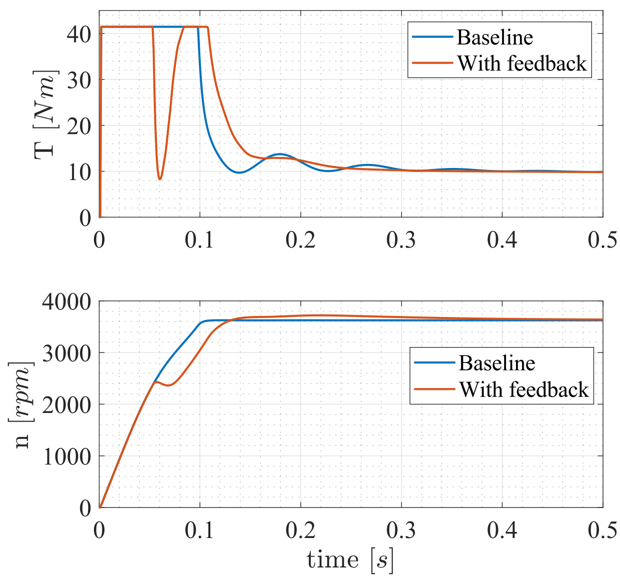

Figure 11. Step response of the EHA with pressure feedback

5. CONCLUSION

This paper investigates the implementation of a pressure feedback strategy that acts on the prime mover of an EHA, aiming to reduce the damping oscillation and thus improve the dynamic performance. The paper derives the controller on the basis of a system linearization. After that, a lumped-parameter simulation model is developed to validate the controller design. Based on the simulation results, the damping oscillation of the actuation velocity response is effectively reduced with pressure feedback. The response time to steady state is much less (from 0.4s to 0.1s), and the overshoot due to the oscillation of about 15\% is eliminated via the pressure feedback controller. Experimental validation of the study may be a focus of future work. Since this architecture is implemented with a bypass valve, it would be used to compare the effectiveness of both methods.

6. NOMENCLATURE

$\ddot{x}$	Acceleration
Δp	Pressure differential
η_{vol}	Volumetric efficiency
λ	Area ratio
ω	Angular velocity
ω_n	Natural frequency
ζ	Damping ratio

A	Area
B	Viscous coefficient of the machine bearings
C_h	Hydraulic capacitance
c	Viscous friction
F	Force
I	Current
J	EM and pump inertia
K	Oil bulk modulus
k	Spring stiffness
K_f	Feedback gain
l	Length of cylinder
M_{eq}	Equivalent mass
n	Motor speed
p	Pressure
Q	Flow rate
T	Torque
U	Voltage
V	Volume
V_d	Pump displacement
x	Cylinder position

7. ACKNOWLEDGMENT

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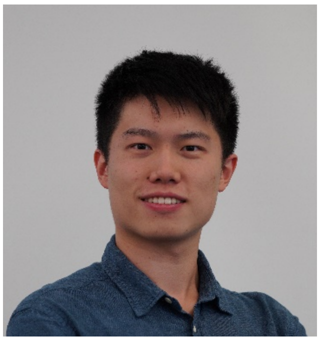
Biographies



Dr. Shaoyang Qu is currently a electrified systems engineer in Bosch Rexroth US. His work mainly focuses on the R&D work on electrified solutions of the state-of-the-art hydraulic systems and the development of next generation electrification platforms. He is also responsible for the customer commissioning, system solutions, and engineering support of eLION products in North America. He got the Ph.D. degree in the School of Mechanical Engineering at Purdue University, where he worked in Maha Fluid Power Research Center on electrification of mobile hydraulics since 2018. Before that, he attended Tsinghua University in Beijing, China, where he received his B.E. in Mechanical Engineering and B.S. in Business administration in 2018.

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Dr. Andrea Vacca is the director of the Maha Fluid Power Research Center, the largest academic research center dedicated to fluid power research in the United States. His research focuses on several aspects of hydraulic control technology including new concepts to perform hydraulic actuations, new designs and modeling of positive displacement machines, electrification of fluid power systems, modeling of the properties of hydraulic fluids, reduction of noise emissions from hydraulic components.

Dr. Vacca holds about 20 patents/patent applications and he is author of about 200 technical papers, and of the textbook “Hydraulic Fluid Power”. He is the Editor in Chief of the International Journal of Fluid Power, one of the directors of the Global Fluid Power Society (GFPS) and a former chair of the Society of Automotive Engineers (SAE) - Fluid Power Division and of the American Society of Mechanical Engineers (ASME) - Fluid Power System and Technology Division. In 2019, Andrea Vacca was awarded the 2019 Joseph Bramah Medal from the Institution of Mechanical Engineers for his contributions to global fluid power research, particularly related to gear pumps.

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Dr. Enrique Busquets is currently responsible for the service and aftermarket business in North America. In addition, his current role in engineering has been the engineering director with regional product and business development responsibility on electronics, software, telematics, and electrification at Bosch Rexroth North America since April 2021. Additionally, Enrique has been responsible for the testing and validation infrastructure in North America since 2018 and innovations since 2016.

Previously, Enrique Busquets was the engineering manager responsible for systems and software development at Bosch Rexroth North America from 2018 to 2021. The technology focus was on mobile applications with electronified hydraulics and electrified systems.

Enrique holds a bachelor's degree from the University of Texas at El Paso and a master's, and doctorate degree in mechanical engineering from Purdue University with emphasis on hydraulics and electronic controls.

In addition to his professional activities, Dr. Enrique Busquets is the Bosch Rexroth industry sponsor and representative at the Maha research center and the National Fluid Power association.